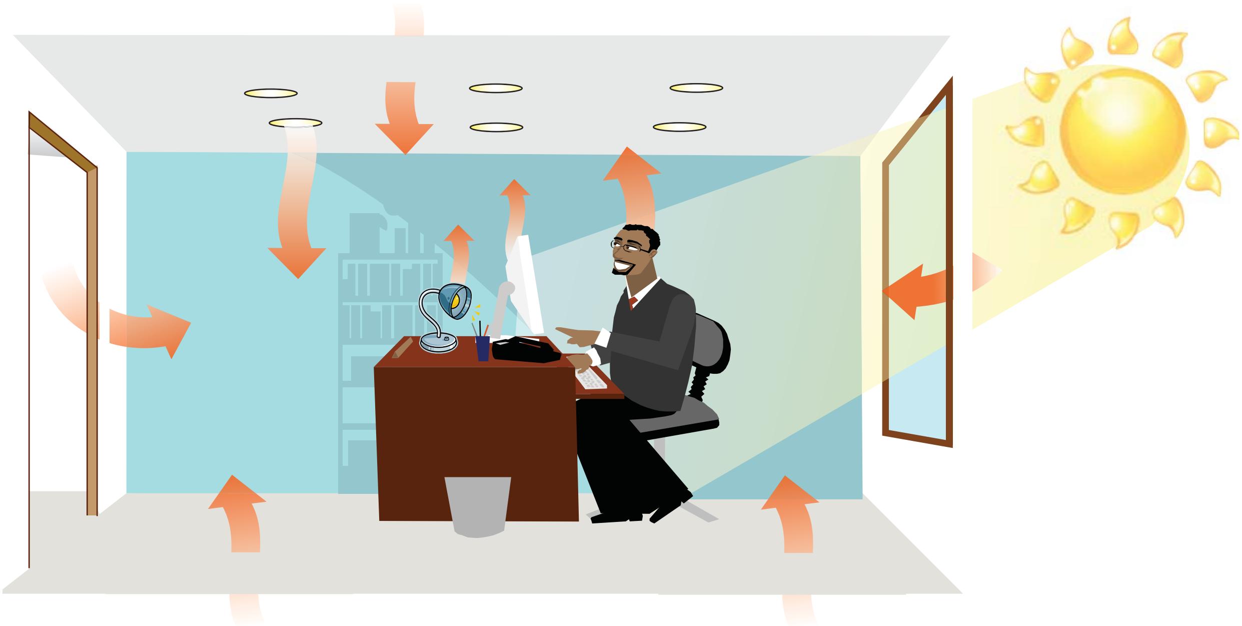
# Air Conditioning Clinic

## Cooling and Heating Load Estimation

One of the Fundamental Series



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# Cooling and Heating Load Estimation

## One of the Fundamental Series

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# Preface

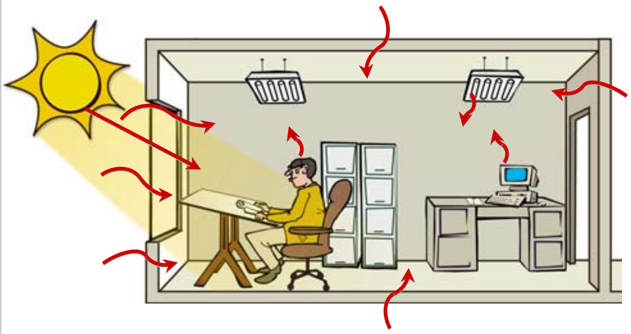


Figure 1

The Trane Company believes that it is incumbent on manufacturers to serve the industry by regularly disseminating information gathered through laboratory research, testing programs, and field experience.

The Trane Air Conditioning Clinic series is one means of knowledge sharing. It is intended to acquaint a nontechnical audience with various fundamental aspects of heating, ventilating, and air conditioning. We have taken special care to make the clinic as uncommercial and straightforward as possible. Illustrations of Trane products only appear in cases where they help convey the message contained in the accompanying text.

This particular clinic introduces the reader to **cooling and heating load estimation**. It is intended to introduce the concepts of estimating building cooling and heating loads and is limited to introducing the components that make up the load on a building, the variables that affect each of these components, and simple methods used to estimate these load components. It is not intended to teach all the details or latest computerized techniques of how to calculate these loads.

If you are interested in learning more about the specific techniques used for cooling and heating load estimating, this booklet includes several references in the back.

# Contents

period one

# Human Comfort

notes

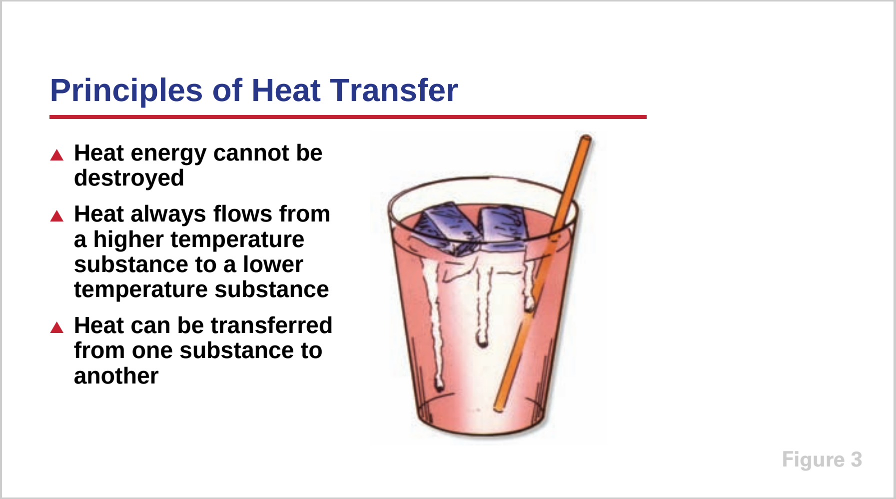
## Cooling and Heating Load Estimation

*period one*

Human Comfort

Figure 2

Heating and air conditioning systems use the principles of heat transfer to maintain comfortable indoor conditions for people.



## Principles of Heat Transfer

- **Heat energy cannot be destroyed**
- **Heat always flows from a higher temperature substance to a lower temperature substance**
- **Heat can be transferred from one substance to another**

Figure 3

The three basic principles of heat transfer discussed in this clinic are:

1) Heat energy cannot be destroyed; it can only be transferred to another substance.

To produce cooling, heat must be removed from a substance by transferring the heat to another substance. This is commonly referred to as the principle of "conservation of energy." Ice cubes are typically placed in a beverage to cool it before being served. As heat is transferred from the beverage to the ice, the temperature of the beverage is lowered. The heat removed from the beverage is not destroyed, but instead is absorbed by the ice, melting the ice from a solid to a liquid.

notes

2) Heat energy naturally flows from a higher-temperature substance to a lower-temperature substance, in other words, from hot to cold.

Heat cannot naturally flow from a cold substance to a hot substance. Consider the example of the beverage and the ice cubes. Because the temperature of the beverage is higher than the temperature of the ice cubes, heat will always flow from the beverage to the ice cubes.

3) Heat energy is transferred from one substance to another by one of three basic processes: conduction, convection, or radiation.

Methods of Heat Transfer

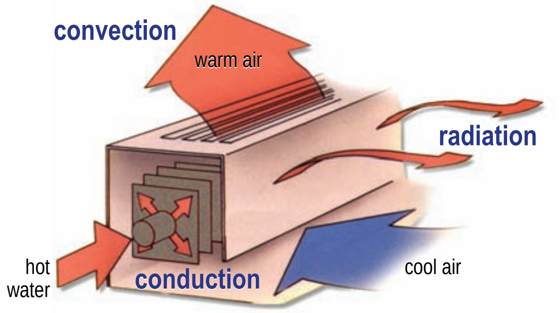


Figure 4

The device shown is a baseboard convector that is commonly used for heating a space. It can be used to demonstrate all three processes of transferring heat.

Hot water flows through a tube inside the convector, warming the inside surface of the tube. Heat is transferred, by conduction, through the tube wall to the slightly cooler fins that are attached to the outside surface of the tube. **Conduction** is the process of transferring heat through a solid.

The heat is then transferred to the cool air that comes into contact with the fins. As the air is warmed and becomes less dense, it rises, carrying the heat away from the fins and out of the convector. This air movement is known as a convection current. **Convection** is the process of transferring heat as the result of the movement of a fluid. Convection often occurs as the result of the natural movement of air caused by temperature (density) differences.

Additionally, heat is radiated from the warm cabinet of the convector and warms cooler objects within the space. **Radiation** is the process of transferring heat by means of electromagnetic waves, emitted due to the temperature difference between two objects. An interesting thing about radiated heat is that it does not heat the air between the source and the object it contacts; it only heats the object itself.

### Measuring Heat Quantity

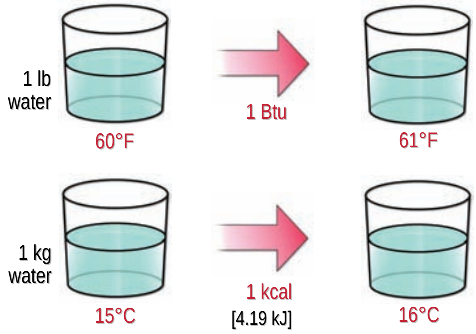


Figure 5

In the I-P system of units, the unit for measuring the quantity of heat is the **British Thermal Unit** (**Btu**). The Btu is defined as the quantity of heat energy required to raise the temperature of 1 lb of water 1°F.

Similarly, in the Système International (SI) system, heat quantity can be expressed using the unit kiloJoule (kJ). A kcal is defined as the amount of heat energy required to raise the temperature of 1 kg of water 1°C. One kcal is equal to 4.19 kJ.

In heating and cooling applications, however, emphasis is placed on the rate of heat transfer, that is, the quantity of heat that flows from one substance to another within a given period of time. This rate of heat flow is commonly expressed in terms of Btu/hr—the quantity of heat, in Btu, that flows from one substance to another during a period of 1 hour.

Similarly, in the SI system of units, the rate of heat flow is expressed in terms of kilowatts (kW). One kW is equivalent to 1 kJ/sec. One kilowatt describes the quantity of heat, in kJ, that flows from one substance to another during a period of 1 second. Finally, the rate of heat flow may often be expressed in terms of watts (W). One kW is equivalent to 1000 W.

notes

## Sensible versus Latent Heat

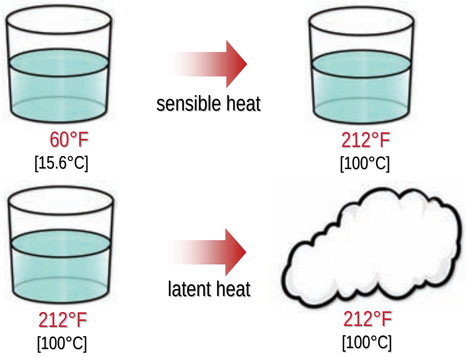


Figure 6

The process of comfort heating and air conditioning is simply a transfer of energy from one substance to another. This energy can be classified as either sensible or latent heat energy.

**Sensible heat** is heat energy that, when added to or removed from a substance, results in a measurable change in dry-bulb temperature.

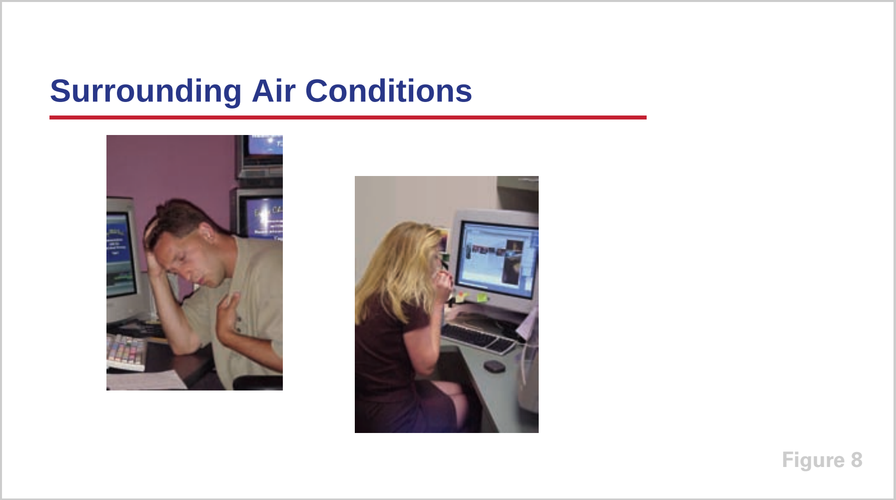
Changes in the **latent heat** content of a substance are associated with the addition or removal of moisture. Latent heat can also be defined as the "hidden" heat energy that is absorbed or released when the phase of a substance is changed. For example, when water is converted to steam, or when steam is converted to water.

## Heat Generated by People

Figure 7

The necessity for comfort air conditioning stems from the fact that the metabolism of the human body normally generates more heat than it needs. This heat is transferred by convection and radiation to the environment surrounding the body. The average adult, seated and working, generates excess heat at the rate of approximately 450 Btu/hr [132 W]. About 60% of this heat is transferred to the surrounding environment by convection and radiation, and 40% is released by perspiration and respiration.

As the level of physical activity increases, the body generates more heat in proportion to the energy expended. When engaged in heavy labor, as in a factory for example, the body generates 1,450 Btu/hr [425 W]. At this level of activity, the proportions reverse and about 40% of this heat is transferred by convection and radiation and 60% is released by perspiration and respiration.



Surrounding Air Conditions

Figure 8

In order for the body to feel comfortable, the surrounding environment must be of suitable temperature and humidity to transfer this excess heat. If the temperature of the air surrounding the body is too high, the body feels uncomfortably warm. The body responds by increasing the rate of perspiration in order to increase the heat loss through evaporation of body moisture. Additionally, if the surrounding air is too humid, the air is nearly saturated and it is more difficult to evaporate body moisture.

If the temperature of the air surrounding the body is too low, however, the body loses more heat than it can produce. The body responds by constricting the blood vessels of the skin to reduce heat loss.

## Factors Affecting Human Comfort

- Dry-bulb temperature
- Humidity
- Air movement
- Fresh air
- Clean air
- Noise level
- Adequate lighting
- Proper furniture and work surfaces

Figure 9

The term “comfort” is often used to define a broader set of conditions than just temperature and humidity. Air movement, adequate fresh air, cleanliness of the air, noise levels in the space, adequate lighting, and proper furniture and work surfaces, are just a few of the other variables that contribute to making a space comfortable for its occupants. This clinic, however, will focus only on the aspects of thermal comfort.

Thermal comfort depends on creating an environment of dry-bulb temperature, humidity, and air motion, that is appropriate for the activity level of the people in the space. This environment allows the body’s rate of heat generation to balance with the body’s rate of heat loss.

notes

## Indoor Design Conditions

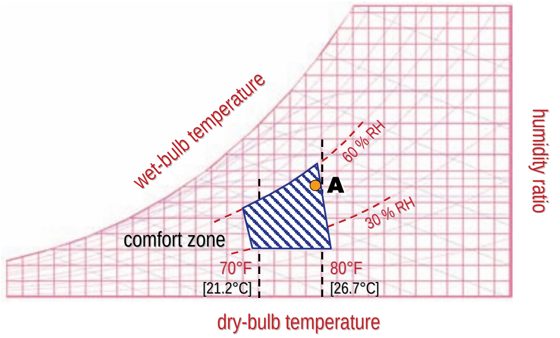


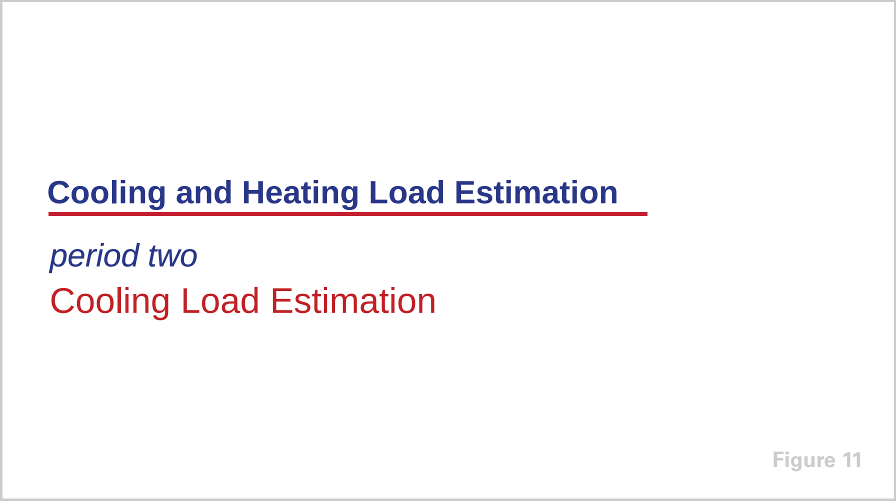
Figure 10

Research studies have been conducted to show that, with a specific amount of air movement, thermal comfort can be produced with certain combinations of dry-bulb temperature and relative humidity. When plotted on a psychrometric chart, these combinations form a range of conditions for delivering acceptable thermal comfort to 80% of the people in a space. This "comfort zone" and the associated assumptions are defined by ASHRAE Standard 55, *Thermal Environmental Conditions for Human Occupancy*.

Determining the desired condition of the space is the first step in estimating the cooling and heating loads for the space. In this clinic, we will choose 78ºF [25.6ºC] dry-bulb temperature and 50% relative humidity **(A)** as the desired indoor condition during the cooling season.

## period two

# Cooling Load Estimation



Cooling and Heating Load Estimation

*period two*

Cooling Load Estimation

Figure 11

The selection of heating, ventilating, and air conditioning (HVAC) system components and equipment should always be based on an accurate determination of the building heating and cooling loads.

During this period we will estimate the cooling loads for a single space in a single-story office building. In Period Four we will estimate the heating loads for this same space. As stated in the preface, this clinic is intended to introduce the concepts of estimating building cooling and heating loads and is not intended to cover all of the details.

The Cooling Load Temperature Difference/Solar Cooling Load/ Cooling Load Factor (CLTD/SCL/CLF) load estimation method, used throughout Period Two, is a simplified hand calculation procedure developed long ago by ASHRAE. Because of its simplicity, it is the most common method used for basic instruction on estimating cooling loads.

period two

# Cooling Load Estimation

notes

**Cooling Load Components**

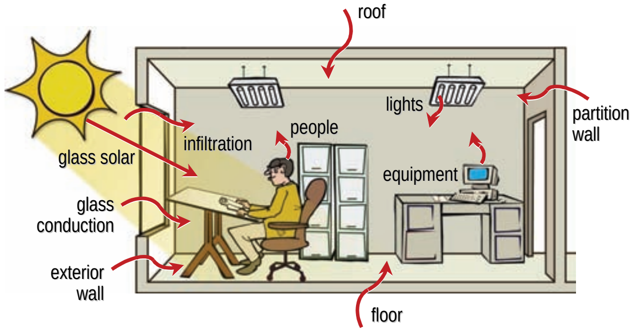


Figure 12

The space cooling load is the rate at which heat must be removed from a space in order to maintain the desired conditions in the space, generally a dry-bulb temperature and relative humidity. The cooling load for a space can be made up of many components, including:

- Conduction heat gain from outdoors through the roof, exterior walls, skylights, and windows. (This includes the effects of the sun shining on these exterior surfaces.)
- Solar radiation heat gain through skylights and windows.
- Conduction heat gain from adjoining spaces through the ceiling, interior partition walls, and floor.
- Internal heat gains due to people, lights, appliances, and equipment in the space.
- Heat gain due to hot, humid air infiltrating into the space from outdoors through doors, windows, and small cracks in the building envelope.

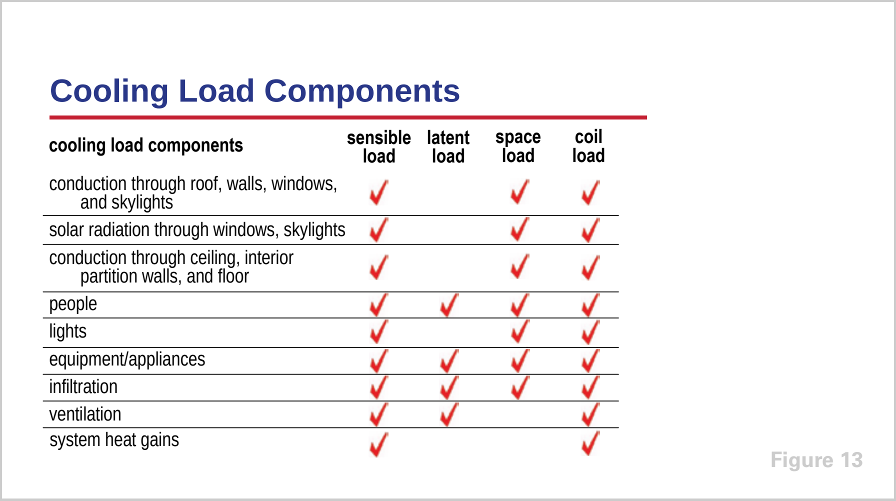
In addition, the cooling coil in the building HVAC system has to handle other components of the total building cooling load, including:

- Heat gain due to outdoor air deliberately brought into the building for ventilation purposes.
- Heat generated by the fans in the system and possibly other heat gains in the system.

Throughout this period, we will assume that the space has no plenum (the space between the ceiling and roof). Therefore, all of the heat gain due to the roof and lighting affects the space directly.

period two

# Cooling Load Estimation



## Cooling Load Components

| cooling load components | sensible load | latent load | space load | coil load |
|---|---|---|---|---|
| conduction through roof, walls, windows, and skylights | ✓ | | ✓ | ✓ |
| solar radiation through windows, skylights | ✓ | | ✓ | ✓ |
| conduction through ceiling, interior partition walls, and floor | ✓ | | ✓ | ✓ |
| people | ✓ | ✓ | ✓ | ✓ |
| lights | ✓ | | ✓ | ✓ |
| equipment/appliances | ✓ | ✓ | ✓ | ✓ |
| infiltration | ✓ | ✓ | ✓ | ✓ |
| ventilation | ✓ | ✓ | | ✓ |
| system heat gains | ✓ | | | ✓ |

Figure 13

These load components contribute sensible and/or latent heat to the space. Conduction through the roof, exterior walls, windows, skylights, ceiling, interior walls, and floor, as well as the solar radiation through the windows and skylights, all contribute only sensible heat to the space.

The people inside the space contribute both sensible and latent heat. Lighting contributes only sensible heat to the space, while equipment in the space may contribute only sensible heat (as is the case for a computer) or both sensible and latent heat (as is the case for a coffee maker). Infiltration generally contributes both sensible and latent heat to the space.

The cooling coil has to handle the additional components of ventilation and system heat gains. Ventilation contributes both sensible and latent heat to the coil load. Other heat gains that occur in the HVAC system (from the fan, for example) generally contribute only sensible heat.

**Time of Peak Cooling Load**

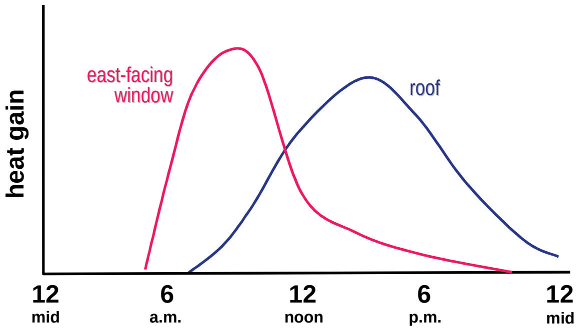


Figure 14

One of the more difficult aspects of estimating the maximum cooling load for a space is determining the time at which this maximum load will occur. This is because the individual components that make up the space cooling load often peak at different times of the day, or even different months of the year.

For example, the heat gain through the roof will be highest in the late afternoon, when it is warm outside and the sun has been shining on it all day. Conversely, the heat gain due to the sun shining through an east-facing window will be highest in the early morning when the sun is rising in the east and shining directly into the window.

Determining the time that the maximum total space cooling load occurs will be discussed later in this clinic.

**Example Office Space (Room 101)**

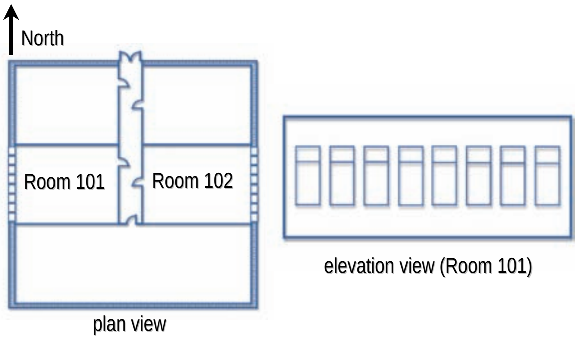


Figure 15

Room 101 is the space that we will use as an example throughout this clinic. The windows face west and the solar heat gain through these windows will

notes

peak in the late afternoon when the sun is setting and shining directly into the windows. Because of this, we will assume that the maximum cooling load for our example space occurs at 4 p.m.

For this example, the following criteria will be used as a basis for estimating the space cooling and heating loads.

- Open-plan office space located in a single-story office building in St. Louis, Missouri.
- Floor area = 45 ft × 60 ft [13.7 m × 18.3 m].
- Floor-to-ceiling height = 12 ft [3.7 m] (no plenum between the space and roof).
- Desired indoor conditions = 78°F [25.6°C] dry-bulb temperature, 50% relative humidity during cooling season; 72°F [22.2°C] dry-bulb temperature during heating season.
- West-facing wall, 12 ft high × 45 ft long [3.7 m × 13.7 m], constructed of 8 in. [203.2 mm] lightweight concrete block with aluminum siding on the outside, 3.5 in. [88.9 mm] of insulation, and ½ in. [12.7 mm] gypsum board on the inside.
- Eight clear, double-pane (¼ in. [6.4 mm]) windows mounted in aluminum frames. Each window is 4 ft wide × 5 ft high [1.2 m × 1.5 m].
- Flat, 45 ft × 60 ft [13.7 m × 18.3 m] roof constructed of 4 in. [100 mm] concrete with 3.5 in. [90 mm] insulation and steel decking.
- Space is occupied from 8:00 a.m. until 5:00 p.m. by 18 people doing moderately active work.
- Fluorescent lighting in space = 2 W/ft$^2$ [21.5 W/m$^2$].
- Computers and office equipment in space = 0.5 W/ft$^2$ [5.4 W/m$^2$], plus one coffee maker.

In order to simplify this example, we will assume that, with the exception of the west-facing exterior wall, room 101 is surrounded by spaces that are air conditioned to the same temperature as this space.

## Outdoor Design Conditions

| | 0.4% | | 1% | | 2% | |
|---|---|---|---|---|---|---|
| | DB | WB | DB | WB | DB | WB |
| **St. Louis, Missouri** | 95°F<br>[35°C] | 76°F<br>[25°C] | 93°F<br>[34°C] | 75°F<br>[24°C] | 90°F<br>[32°C] | 74°F<br>[23°C] |

Figure 16

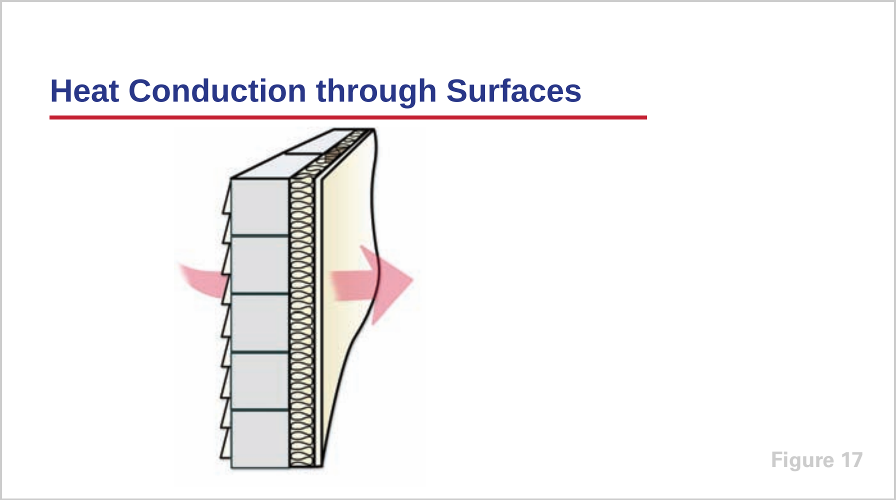
### Outdoor Design Conditions

In Period One, we discussed the indoor conditions required for thermal comfort. The next step toward estimating the cooling load of a space is to determine the highest, frequently-occurring outdoor air temperature. In the summer, for example, when the temperature outside is high, heat transfers from outdoors to indoors, thus contributing to the heat gain of the space.

Obviously, HVAC systems would be greatly oversized if cooling load calculations were based on the most extreme outdoor temperature ever recorded for the location. Instead, outdoor design temperatures are based on their frequency of occurrence. Design outdoor conditions for many locations can be found in the *ASHRAE Handbook—Fundamentals*.

Figure 16, the cooling outdoor design conditions for St. Louis, Missouri, includes three columns of dry-bulb temperatures and corresponding wet-bulb temperatures. The first column heading, 0.4%, means that the dry-bulb temperature in St. Louis exceeds 95°F [35°C] for only 0.4% of all of the hours in an average year (or 35 hours). Also, 76°F [25°C] is the wet-bulb temperature that occurs most frequently when the dry-bulb temperature is 95°F [35°C]. The second column heading, 1%, means that the temperature exceeds 93°F [34°C] for only 1% of all of the hours in an average year (or 87.6 hours). When the dry-bulb temperature is 93°F [34°C], the wet-bulb temperature that occurs most frequently is 75°F [24°C]. For our example, we will use the more severe 95°F [35°C] dry bulb and 76°F [25°C] wet bulb for the outdoor design conditions.

The tables published by ASHRAE include more weather data that can be useful for sizing certain HVAC system components, but that discussion is outside the scope of this clinic.



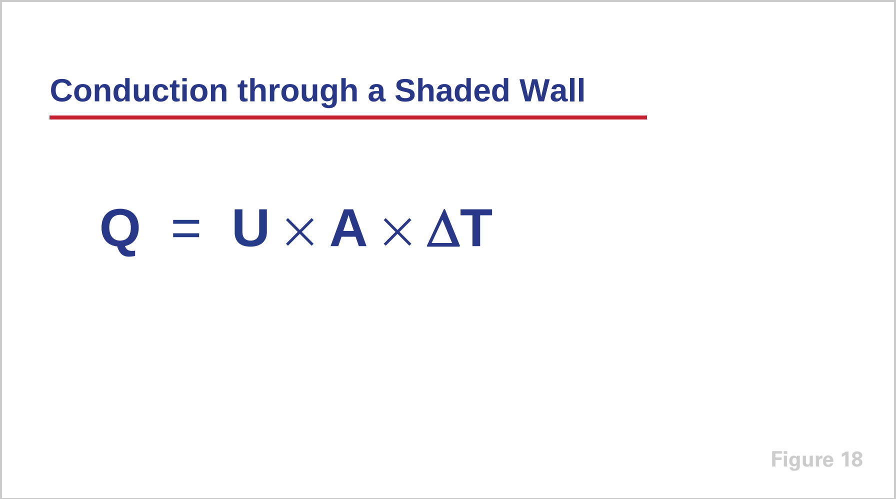
## Heat Conduction through Surfaces

Figure 17

**Conduction through Surfaces**

Conduction is the process of transferring heat through a solid, such as a wall, roof, floor, ceiling, window, or skylight. Heat naturally flows by conduction from a higher temperature to a lower temperature. Generally, when estimating the maximum cooling load for a space, the temperature of the air outdoors is higher than the temperature of the air indoors.

We will focus on the most common conduction heat gains to a space: through the roof, external walls, and windows.



## Conduction through a Shaded Wall

$$Q = U \times A \times \Delta T$$

Figure 18

Although often not applicable, a simplifying assumption when estimating the conduction heat gain through an exterior surface is to assume that the surface is completely shaded at all times. With this assumption, the amount of heat transferred through the surface is a direct result of the temperature difference between the space and outdoors. This assumption, however, does not include the additional heat transfer that occurs because of the sun shining on the surface. This will be discussed next.

The amount of heat transferred through a *shaded* exterior surface depends on the area of the surface, the overall heat transfer coefficient of the surface, and the dry-bulb temperature difference from one side of the surface to the other. The equation used to predict the heat gain by conduction is:

$$Q = U \times A \times \Delta T$$

where,

$Q$ = heat gain by conduction, Btu/hr [W]

$U$ = overall heat-transfer coefficient of the surface, Btu/hr • ft$^2$ • °F [W/m$^2$ • °K]

$A$ = area of the surface, ft$^2$ [m$^2$]

$\Delta T$ = dry-bulb temperature difference across the surface, °F [°C]

In the case of a shaded exterior surface, this temperature difference is the design outdoor dry-bulb temperature ($T_o$) minus the desired indoor dry-bulb temperature ($T_i$).

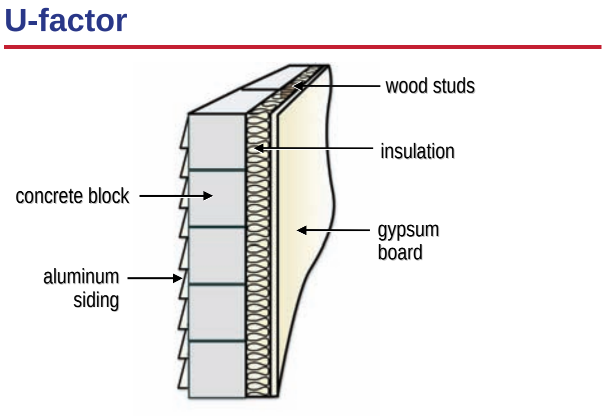


Figure 19

The overall heat transfer coefficient is also called the **U-factor**. The U-factor describes the rate at which heat will be transferred through the structure.

Walls and roofs are typically made up of layers of several materials. The U-factor for a specific wall or roof is calculated by summing the thermal resistances (R-values) of each of these layers and then taking the inverse. The *ASHRAE Handbook—Fundamentals* tabulates the thermal resistance of many common materials used in constructing walls, roofs, ceilings, and floors.

The wall in our example space is comprised of:

- aluminum siding (R = 0.61 $ft^2 \bullet hr \bullet °F/Btu$ [0.11 $\bullet m^2 \bullet °K/W$])
- 8 in. [200 mm] lightweight concrete block (R = 2.0 [0.35])
- 3.5 in. [90 mm] of fiberglass insulation (R = 13.0 [2.29])
- ½ in. [12.7 mm] gypsum board (R = 0.45 [0.08])

Additionally, there is a film of air on the outside surface of the wall (R = 0.25 [0.044], assuming air moving at 7.5 mph [12 km/hr] during the summer) and another film of air on the inside surface of the wall (R = 0.68 [0.12], assuming still air).

## U-factor for Example Wall

**thermal resistance (R)**

| | | |
|---|---|---|
| $R_{outdoor\text{-}air\ film}$ | 0.25 | [0.04] |
| $R_{siding}$ | 0.61 | [0.11] |
| $R_{concrete\ block}$ | 2.00 | [0.35] |
| $R_{insulation}$ | 13.00 | [2.29] |
| $R_{gypsum\ board}$ | 0.45 | [0.08] |
| $R_{indoor\text{-}air\ film}$ | 0.68 | [0.12] |
| $R_{total}$ | 16.99 | [2.99] |

$$U = \frac{1}{R_{total}}$$

$$U = 0.06\ Btu/hr \cdot ft^2 \cdot °F$$

$$[\ U = 0.33\ W/m^2 \cdot °K\ ]$$

Figure 20

The U-factor of this wall is calculated by adding the thermal resistances of each of these layers and then taking the inverse.

$$U = \frac{1}{R_{\substack{outdoor\\air\ film}} + R_{siding} + R_{\substack{concrete\\block}} + R_{insulation} + R_{\substack{gypsum\\board}} + R_{\substack{indoor\\air\ film}}}$$

$$U = \frac{1}{0.25 + 0.61 + 2.0 + 13 + 0.45 + 0.68} = \frac{1}{16.99} = 0.06\ Btu/hr \bullet ft^2 \bullet °F$$

$$\left[U = \frac{1}{0.04 + 0.11 + 0.35 + 2.29 + 0.08 + 0.12} = \frac{1}{2.99} = 0.33\ W/m^2 \bullet °K\right]$$

notes

The U-factor of the roof in our example is calculated in a similar manner.

$$U = \frac{1}{R_{\substack{\text{outdoor}\\\text{air film}}} + R_{\substack{\text{built up}\\\text{roofing}}} + R_{\text{insulation}} + R_{\substack{\text{lightweight}\\\text{concrete}}} + R_{\substack{\text{metal}\\\text{decking}}} + R_{\substack{\text{indoor}\\\text{air film}}}}$$

$$U = \frac{1}{0.25 + 0.33 + 13 + 3.12 + 0 + 0.92} = \frac{1}{17.62} = 0.057 \text{ Btu/hr} \bullet \text{ft}^2 \bullet {}^\circ\text{F}$$

$$\left[U = \frac{1}{0.04 + 0.06 + 2.29 + 0.54 + 0 + 0.16} = \frac{1}{3.09} = 0.323 \text{ W/m}^2 \bullet {}^\circ\text{K}\right]$$

## Conduction through a Shaded Wall

$$Q_{\text{wall}} = 0.06 \times 380 \times (95 - 78) = 388 \text{ Btu/hr}$$

$$[\, Q_{\text{wall}} = 0.33 \times 36.3 \times (35 - 25.6) = 113 \text{ W} \,]$$

Figure 21

If the west-facing wall of our example space was completely shaded at all times, the conduction heat gain due to the wall would be 388 Btu/hr [133 W].

Conduction heat gain through the west-facing wall (assume shaded at all times):

- U-factor = 0.06 Btu/hr • $\text{ft}^2$ • °F [0.33 W/$\text{m}^2$ • °K]
- Total area of wall + windows = 12 ft × 45 ft = 540 $\text{ft}^2$ [3.7 m × 13.7 m = 50.7 $\text{m}^2$]
- Area of windows = 8 windows × (4 ft × 5 ft) = 160 $\text{ft}^2$
  [8 × (1.2 m × 1.5 m) = 14.4 $\text{m}^2$]
- Net area of wall = 540 – 160 = 380 $\text{ft}^2$ [50.7 – 14.4 = 36.3 $\text{m}^2$]
- $\Delta T$ = outdoor temperature (95°F [35°C]) – indoor temperature (78°F [25.6°C])

$$Q = U \times A \times \Delta T$$

$$Q = 0.06 \times 380 \times (95 - 78) = 388 \text{ Btu/hr}$$

$$[Q = 0.33 \times 36.3 \times (35 - 25.6) = 113 \text{ W}]$$

notes

**Sunlit Surfaces**

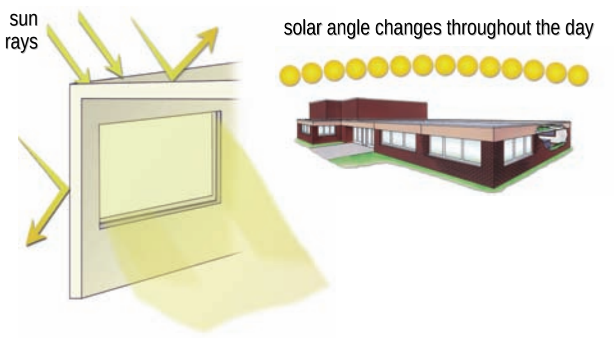


Figure 22

Most exterior surfaces of a building, however, are exposed to direct sunlight during some portion of the day. Solar heat energy is generated by the sun and radiated to earth. Radiant heat is similar to light, in that it travels in a straight line and can be reflected from a bright surface. Both light and radiant heat can pass through a transparent surface (such as glass), yet neither can pass directly through an opaque or non-transparent surface (such as a brick wall). When the sun's rays strike an opaque surface, however, a certain amount of radiant heat energy is transferred to that surface, resulting in an increase in the surface temperature. The amount of heat transferred depends primarily on the color and smoothness of the surface, and the angle at which the sun's rays strike the surface.

When the sun's rays strike the surface at a 90° angle, the maximum amount of radiant heat energy is transferred to that surface. When the same rays strike that same surface at a lesser angle, less radiant heat energy is transferred to the surface. The angle at which the sun's rays strike a surface depends upon the latitude, the time of day, and the month of the year. Due to the rotation of the earth throughout the day, and the earth orbiting the sun throughout the year, the angle at which the sun's rays strike a surface of a building is constantly changing. This varies the intensity of the solar radiation on an exterior surface of a building, resulting in a varying amount of solar heat transferred to the surface throughout the day and throughout the year.

As mentioned previously, the assumption that the surface is completely shaded does not account for the additional heat gain that occurs when the sun shines on a surface. Solar heat, therefore, must be considered, as it constitutes an important part of the total cooling load of most buildings.

notes

## Time Lag

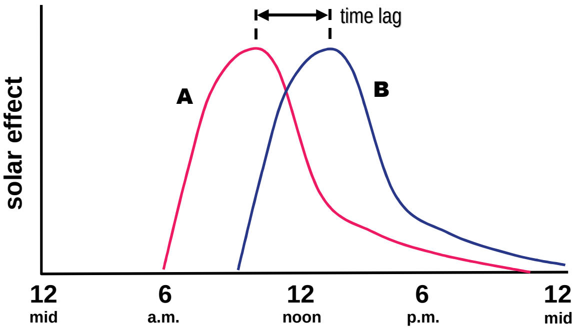


Figure 23

The walls and roof that make up a building's envelope have the capacity to store heat energy. This property delays the heat transfer from outdoors to the space. The time required for heat to be transferred through a structure into the space is called the **time lag**.

For example, the heat that is transferred through a sunlit wall into a space is the result of sunlight that fell on the outer surface of the wall earlier in the day. Curve **A** shows the magnitude of the solar effect on the exterior wall. Curve **B** shows the resulting heat that is transferred through the wall into the space. This delay in the heat gain to the space is the time lag. The magnitude of this time lag depends on the materials used to construct the particular wall or roof, and on their capacity to store heat.

period two

# Cooling Load Estimation

notes

## Conduction through Sunlit Surfaces

$$Q = U \times A \times CLTD$$

Figure 24

A factor called the **cooling load temperature difference (CLTD)** is used to account for the added heat transfer due to the sun shining on exterior walls, roofs, and windows, and the capacity of the wall and roof to store heat. The CLTD is substituted for ΔT in the equation to estimate heat transfer by conduction.

$Q = U \times A \times CLTD$

## CLTD Factors for West-Facing Wall

| | hour | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | **1** | **2** | **3** | **4** | **5** | **6** | **7** | **8** | **9** | **10** | **11** | **12** | **13** | **14** | **15** | **16** | **17** | **18** | **19** | **20** | **21** | **22** | **23** | **24** |
| CLTD (°F) | 35 | 30 | 25 | 21 | 17 | 14 | 11 | 8 | 7 | 6 | 6 | 7 | 8 | 10 | 12 | 16 | 22 | 30 | 37 | 44 | 48 | 48 | 45 | 41 |
| CLTD (°C) | 19 | 17 | 14 | 12 | 9 | 8 | 6 | 4 | 4 | 3 | 3 | 4 | 4 | 6 | 7 | 9 | 12 | 17 | 21 | 24 | 27 | 27 | 25 | 23 |

Figure 25

This particular table includes CLTD factors for a west-facing wall similar to the type used in our example building. It should be noted that the data in this table are based on the following assumptions:

- 78°F [25.6°C] indoor air
- 95°F [35°C] maximum outdoor air
- Average outdoor daily temperature range of 21°F [11.7°C]

- $21^{st}$ day of July
- 40° north latitude
- Dark-colored surface

Tables for various wall and roof types, as well as correction factors for applications that differ from these assumptions, can be found in the *1997 ASHRAE Handbook—Fundamentals* and ASHRAE's *Cooling and Heating Load Calculation Principles* manual.

The wall in our example is classified as Wall Type 9. At 4 p.m. (Hour 17 in this table), the CLTD for a west-facing wall of this type is 22°F [12°C]. This means that, even though the actual dry-bulb temperature difference is only 17°F (95°F – 78°F) [9.4°C (35°C – 25.6°C)], the sun shining on the outer surface of this wall increases the "effective temperature difference" to 22°F [12°C].

Notice that the CLTD increases later in the day, and then begins to decrease in the evening as the stored heat is finally transferred from the wall into the space.

**Table 1. CLTDs for Sunlit Walls (40° North Latitude, July 21), °F**

| | Wall Type 9 | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Hour | | | | | | | | | | | | | | | | | | | | | | | |
| | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 |
| N | 17 | 15 | 13 | 11 | 9 | 7 | 5 | 4 | 4 | 4 | 5 | 7 | 8 | 10 | 12 | 15 | 17 | 19 | 21 | 22 | 23 | 23 | 22 | 20 |
| NE | 18 | 15 | 13 | 11 | 9 | 7 | 5 | 5 | 6 | 10 | 16 | 20 | 23 | 25 | 26 | 27 | 27 | 28 | 28 | 27 | 26 | 25 | 23 | 20 |
| E | 20 | 17 | 14 | 12 | 10 | 8 | 6 | 5 | 7 | 12 | 19 | 26 | 32 | 36 | 37 | 37 | 37 | 36 | 34 | 33 | 31 | 29 | 26 | 23 |
| SE | 20 | 17 | 15 | 12 | 10 | 8 | 6 | 5 | 6 | 9 | 13 | 19 | 25 | 31 | 34 | 36 | 37 | 36 | 35 | 34 | 32 | 29 | 26 | 23 |
| S | 21 | 18 | 15 | 12 | 10 | 8 | 6 | 5 | 4 | 3 | 4 | 6 | 10 | 14 | 20 | 25 | 29 | 33 | 34 | 34 | 32 | 30 | 27 | 24 |
| SW | 31 | 26 | 22 | 18 | 15 | 12 | 9 | 7 | 6 | 5 | 5 | 6 | 8 | 10 | 14 | 19 | 26 | 33 | 39 | 43 | 45 | 44 | 40 | 36 |
| W | 35 | 30 | 25 | 21 | 17 | 14 | 11 | 8 | 7 | 6 | 6 | 7 | 8 | 10 | 12 | 16 | 22 | 30 | 37 | 44 | 48 | 48 | 45 | 41 |
| NW | 29 | 25 | 21 | 17 | 14 | 11 | 9 | 7 | 5 | 5 | 5 | 6 | 7 | 9 | 11 | 14 | 18 | 22 | 28 | 34 | 37 | 38 | 36 | 33 |

**Table 2. CLTDs for Sunlit Walls (40° North Latitude, July 21), °C**

| | Wall Type 9 | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Hour | | | | | | | | | | | | | | | | | | | | | | | |
| | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 |
| N | 9 | 8 | 7 | 6 | 5 | 4 | 3 | 2 | 2 | 2 | 3 | 4 | 4 | 6 | 7 | 8 | 9 | 11 | 12 | 12 | 13 | 13 | 12 | 11 |
| NE | 10 | 8 | 7 | 6 | 5 | 4 | 3 | 3 | 3 | 6 | 9 | 11 | 13 | 14 | 14 | 15 | 15 | 16 | 16 | 15 | 14 | 14 | 13 | 11 |
| E | 11 | 9 | 8 | 7 | 6 | 4 | 3 | 3 | 4 | 7 | 11 | 14 | 18 | 20 | 21 | 21 | 21 | 20 | 19 | 18 | 17 | 16 | 14 | 13 |
| SE | 11 | 9 | 8 | 7 | 6 | 4 | 3 | 3 | 3 | 5 | 7 | 11 | 14 | 17 | 19 | 20 | 21 | 20 | 19 | 19 | 18 | 16 | 14 | 13 |
| S | 12 | 10 | 8 | 7 | 6 | 4 | 3 | 3 | 2 | 2 | 2 | 3 | 6 | 8 | 11 | 14 | 16 | 18 | 19 | 19 | 18 | 17 | 15 | 13 |
| SW | 17 | 14 | 12 | 10 | 8 | 7 | 5 | 4 | 3 | 3 | 3 | 3 | 4 | 6 | 8 | 11 | 14 | 18 | 22 | 24 | 25 | 24 | 22 | 20 |
| W | 19 | 17 | 14 | 12 | 9 | 8 | 6 | 4 | 4 | 3 | 3 | 4 | 4 | 6 | 7 | 9 | 12 | 17 | 21 | 24 | 27 | 27 | 25 | 23 |
| NW | 16 | 14 | 12 | 9 | 8 | 6 | 5 | 4 | 3 | 3 | 3 | 3 | 4 | 5 | 6 | 8 | 10 | 12 | 16 | 19 | 21 | 21 | 20 | 18 |

Source: *1997 ASHRAE Handbook—Fundamentals*, Chapter 28, Table 32

## Conduction through Sunlit Surfaces

$Q_{wall} = 0.06 \times 380 \times 22 = 502$ Btu/hr

$Q_{roof} = 0.057 \times 2{,}700 \times 80 = 12{,}312$ Btu/hr

[ $Q_{wall} = 0.33 \times 36.3 \times 12 = 144$ W ]

[ $Q_{roof} = 0.323 \times 250.7 \times 44 = 3{,}563$ W ]

Figure 26

Using CLTD instead of $\Delta T$, we will determine the heat gain, by conduction, through the west-facing wall and the roof. The U-factors are the same as those calculated on Figure 20.

Conduction heat gain through the west-facing sunlit wall:

- U-factor = 0.06 Btu/hr • $ft^2$ • °F [0.33 W/$m^2$ • °K]
- Net area of wall = 380 $ft^2$ [36.3 $m^2$]
- $CLTD_{hour=17}$ = 22°F [12°C]

$Q = 0.06 \times 380 \times 22 = 502$ Btu/hr

[$Q = 0.33 \times 36.3 \times 12 = 144$ W]

Table 3 [Table 4] includes CLTD factors for several types of roofs. The roof in our example building is classified as Roof Type 2. The data in Table 3 [Table 4] are based on assumptions similar to the CLTD table for walls. At Hour 17, the CLTD for a flat roof of this type is 80°F [44°C].

Conduction heat gain through the roof:

- U-factor = 0.057 Btu/hr • $ft^2$ • °F [0.323 W/$m^2$ • °K]
- Area of roof = 45 ft × 60 ft = 2,700 $ft^2$ [13.7 m × 18.3 m = 250.7 $m^2$]
- $CLTD_{hour=17}$ = 80°F [44°C]

$Q = 0.057 \times 2{,}700 \times 80 = 12{,}312$ Btu/hr

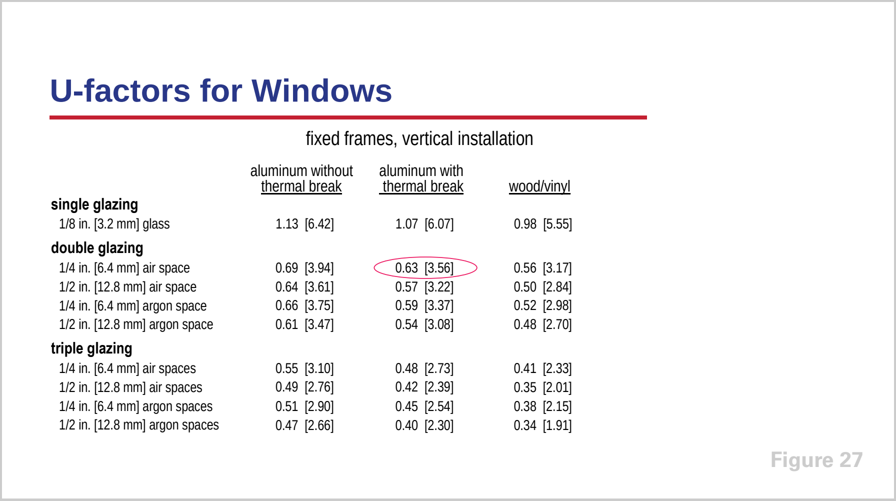
[$Q = 0.323 \times 250.7 \times 44 = 3{,}563$ W]

**Table 3. CLTDs for Flat Roofs (40° North Latitude, July 21), °F**

| | Hour | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Roof Type | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 |
| 1 | 0 | -2 | -4 | -5 | -6 | -6 | 0 | 13 | 29 | 45 | 60 | 73 | 83 | 88 | 88 | 83 | 73 | 60 | 43 | 26 | 15 | 9 | 5 | 2 |
| 2 | 2 | 0 | -2 | -4 | -5 | -6 | -4 | 4 | 17 | 32 | 48 | 62 | 74 | 82 | 86 | 85 | 80 | 70 | 56 | 39 | 25 | 15 | 9 | 5 |
| 3 | 12 | 8 | 5 | 2 | 0 | -2 | 0 | 5 | 13 | 24 | 35 | 47 | 57 | 66 | 72 | 74 | 73 | 67 | 59 | 48 | 38 | 30 | 23 | 17 |
| 4 | 17 | 11 | 7 | 3 | 1 | -1 | -3 | -3 | 0 | 7 | 17 | 29 | 42 | 54 | 65 | 73 | 77 | 78 | 74 | 67 | 56 | 45 | 34 | 24 |
| 5 | 21 | 16 | 12 | 8 | 5 | 3 | 1 | 2 | 6 | 12 | 21 | 31 | 41 | 51 | 60 | 66 | 69 | 69 | 65 | 59 | 51 | 42 | 34 | 27 |

**Table 4. CLTDs for Flat Roofs (40° North Latitude, July 21), °C**

| | Hour | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Roof Type | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 |
| 1 | 0 | -1 | -2 | -3 | -3 | -3 | 0 | 7 | 16 | 25 | 33 | 41 | 46 | 49 | 49 | 46 | 41 | 33 | 24 | 14 | 8 | 5 | 3 | 1 |
| 2 | 1 | 0 | -1 | -2 | -3 | -3 | -2 | 2 | 9 | 18 | 27 | 34 | 41 | 46 | 48 | 47 | 44 | 39 | 31 | 22 | 14 | 8 | 5 | 3 |
| 3 | 7 | 4 | 3 | 1 | 0 | -1 | 0 | 3 | 7 | 13 | 19 | 26 | 32 | 37 | 40 | 41 | 41 | 37 | 33 | 27 | 21 | 17 | 13 | 9 |
| 4 | 9 | 6 | 4 | 2 | 1 | -1 | -2 | -2 | 0 | 4 | 9 | 16 | 23 | 30 | 36 | 41 | 43 | 43 | 41 | 37 | 31 | 25 | 19 | 13 |
| 5 | 12 | 9 | 7 | 4 | 3 | 2 | 1 | 1 | 3 | 7 | 12 | 17 | 23 | 28 | 33 | 37 | 38 | 38 | 36 | 33 | 28 | 23 | 19 | 15 |



### U-factors for Windows

fixed frames, vertical installation

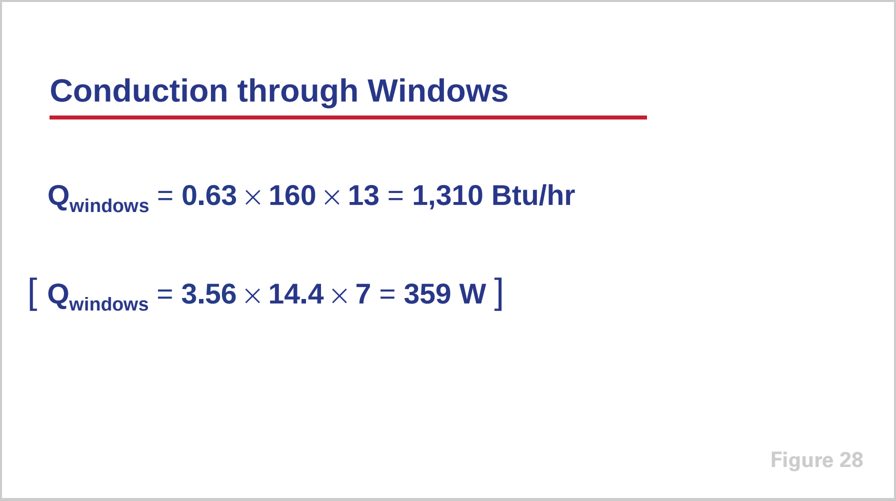
| | aluminum without thermal break | aluminum with thermal break | wood/vinyl |
|---|---|---|---|
| **single glazing** | | | |
| 1/8 in. [3.2 mm] glass | 1.13 [6.42] | 1.07 [6.07] | 0.98 [5.55] |
| **double glazing** | | | |
| 1/4 in. [6.4 mm] air space | 0.69 [3.94] | 0.63 [3.56] | 0.56 [3.17] |
| 1/2 in. [12.8 mm] air space | 0.64 [3.61] | 0.57 [3.22] | 0.50 [2.84] |
| 1/4 in. [6.4 mm] argon space | 0.66 [3.75] | 0.59 [3.37] | 0.52 [2.98] |
| 1/2 in. [12.8 mm] argon space | 0.61 [3.47] | 0.54 [3.08] | 0.48 [2.70] |
| **triple glazing** | | | |
| 1/4 in. [6.4 mm] air spaces | 0.55 [3.10] | 0.48 [2.73] | 0.41 [2.33] |
| 1/2 in. [12.8 mm] air spaces | 0.49 [2.76] | 0.42 [2.39] | 0.35 [2.01] |
| 1/4 in. [6.4 mm] argon spaces | 0.51 [2.90] | 0.45 [2.54] | 0.38 [2.15] |
| 1/2 in. [12.8 mm] argon spaces | 0.47 [2.66] | 0.40 [2.30] | 0.34 [1.91] |

Figure 27

Estimating the heat gain by conduction through a window is very similar to walls and roofs.

The data on this slide is an excerpt from the *1997 ASHRAE Handbook—Fundamentals* and includes U-factors for common window assemblies.

The windows in our example are double-pane windows with a ¼-inch [6.4 mm] air space between the panes. Assuming that the windows are fixed (not operable), with aluminum frames and a thermal break, the U-factor is 0.63 Btu/hr • ft$^2$ • °F [3.56 W/m$^2$ • °K].



### Conduction through Windows

$$Q_{\text{windows}} = 0.63 \times 160 \times 13 = 1{,}310 \text{ Btu/hr}$$

$$\left[\ Q_{\text{windows}} = 3.56 \times 14.4 \times 7 = 359 \text{ W}\ \right]$$

Figure 28

Using this U-factor, we will determine the heat gain, by conduction, through the eight west-facing windows. Table 5 [Table 6] is an excerpt from the *1997 ASHRAE Handbook—Fundamentals* and includes CLTD factors for glass. It is also based on similar assumptions as the CLTD tables for walls and roofs. At Hour 17, the CLTD for a glass window is 13°F [7°C].

notes

Conduction heat gain through the west-facing windows:

- U-factor = 0.63 Btu/hr • ft$^2$ • °F [3.56 W/m$^2$ • °K]
- Total area of glass = 8 windows × (4 ft × 5 ft) = 160 ft$^2$
  [8 × (1.2 m × 1.5 m) = 14.4 m$^2$]
- CLTD$_{hour=17}$ = 13°F [7°C]

$Q = 0.63 \times 160 \times 13 = 1{,}310$ Btu/hr

[$Q = 3.56 \times 14.4 \times 7 = 359$ W]

**Table 5. CLTDs for Glass, °F**

| Hour | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 |
| 1 | 0 | -1 | -2 | -2 | -2 | -2 | 0 | 2 | 4 | 7 | 9 | 12 | 13 | 14 | 14 | 13 | 12 | 10 | 8 | 6 | 4 | 3 | 2 |

**Table 6. CLTDs for Glass, °C**

| Hour | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 |
| 1 | 0 | -1 | -1 | -1 | -1 | -1 | 0 | 1 | 2 | 4 | 5 | 7 | 7 | 8 | 8 | 7 | 7 | 6 | 4 | 3 | 2 | 2 | 1 |

Source: *1997 ASHRAE Handbook—Fundamentals*, Chapter 28, Table 34

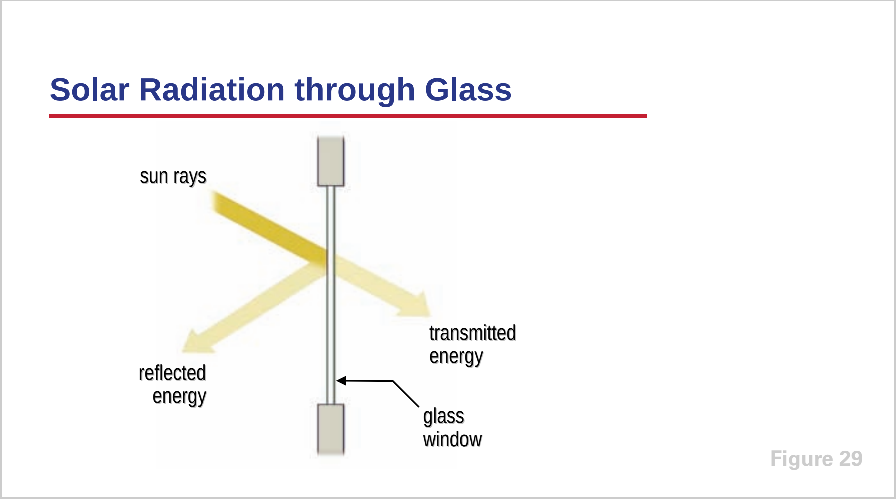
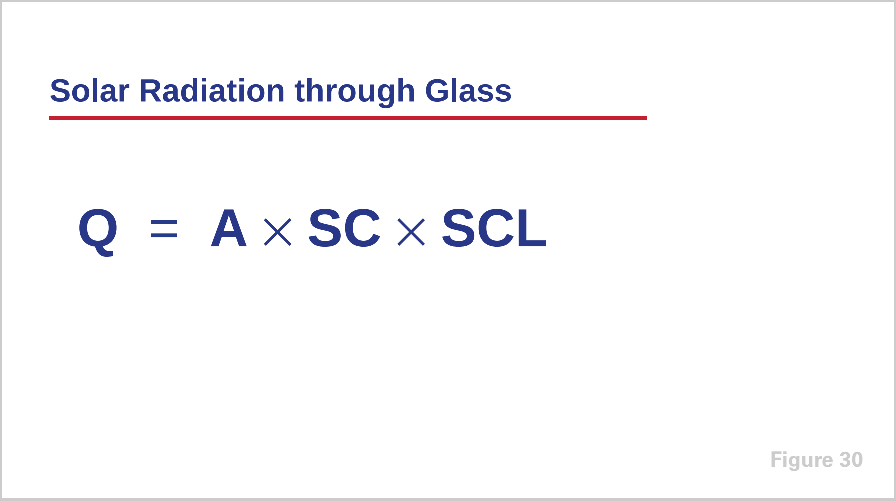

Solar Radiation through Glass

Figure 29

### Solar Radiation through Glass

Previously, we estimated the heat transferred through glass windows by the process of conduction. A large part of the solar heat energy that shines on a window or skylight is radiated through the glass and transmitted directly into the space. The amount of solar heat radiated through the glass depends primarily on the reflective characteristics of the glass and the angle at which the sun's rays strike the surface of the glass.

While glass windows of double- or triple-pane construction do an excellent job of reducing heat transfer by conduction, they do not appreciably reduce the amount of solar radiation directly into a space. To limit the amount of solar radiation entering the space, heat-absorbing glass, reflective glass, or internal or external shading devices can be used.



**Solar Radiation through Glass**

$$Q = A \times SC \times SCL$$

Figure 30

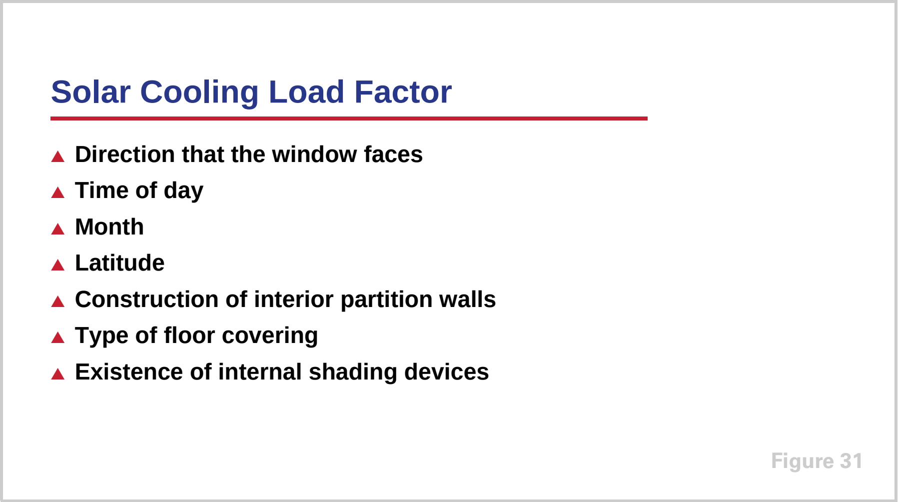
Although the *1997 ASHRAE Handbook—Fundamentals* contains new, more advanced methods of estimating solar heat gain through glass, they are beyond the scope of this clinic.

The equation used to predict the solar heat gain through glass is:

$$Q = A \times SC \times SCL$$

where,

- Q = heat gain by solar radiation through glass, Btu/hr [W]
- A = total surface area of the glass, $ft^2$ [$m^2$]
- SC = shading coefficient of the window, dimensionless
- SCL = solar cooling load factor, Btu/hr • $ft^2$ [$W/m^2$]



## Solar Cooling Load Factor

- Direction that the window faces
- Time of day
- Month
- Latitude
- Construction of interior partition walls
- Type of floor covering
- Existence of internal shading devices

Figure 31

The **solar cooling load (SCL) factor** is used to estimate the rate at which solar heat energy radiates directly into the space, heats up the surfaces and furnishings, and is later released to the space as a sensible heat gain. Similar to CLTD, the SCL factor is used to account for the capacity of the space to absorb and store heat.

The value of SCL is based on several variables, including the direction that the window is facing, time of day, month, and latitude. These four variables define the angle at which the sun's rays strike the surface of the window. The next two variables, the construction of the interior partition walls and the type of floor covering, help define the capacity of the space to store heat. This affects the time lag between the time that the solar radiation warms up the space furnishings and the time that the heat is released into the space. The last variable, whether or not internal shading devices are installed, affects the amount of solar heat energy passing through the glass.

The *1997 ASHRAE Handbook—Fundamentals* contains tables of SCL values for common space types, based on combinations of these variables. Table 7 [Table 8] is an excerpt from the handbook and includes SCL factors for a space type similar to the one in our example. The space in our example is classified as Space Type A. The data in this table is based on the 21st day of July and 40° north latitude. At Hour 17, the SCL for the west-facing windows in our example space is 192 Btu/hr • $\text{ft}^2$ [605 $\text{W/m}^2$].

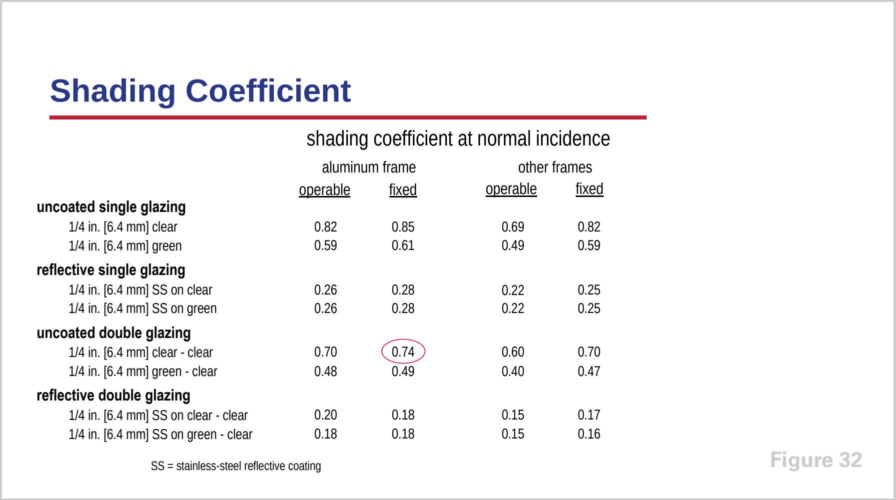
**Table 7. SCL for Sunlit Glass (40° North Latitude, July 21), Btu/hr • ft$^2$**

| | Space Type A | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Hour | | | | | | | | | | | | | | | | | | | | | | | |
| | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 |
| N | 0 | 0 | 0 | 0 | 1 | 25 | 27 | 28 | 32 | 35 | 38 | 40 | 40 | 39 | 36 | 31 | 31 | 36 | 12 | 6 | 3 | 1 | 1 | 0 |
| NE | 0 | 0 | 0 | 0 | 2 | 85 | 129 | 134 | 112 | 75 | 55 | 48 | 44 | 40 | 37 | 32 | 26 | 18 | 7 | 3 | 2 | 1 | 0 | 0 |
| E | 0 | 0 | 0 | 0 | 2 | 93 | 157 | 185 | 183 | 154 | 106 | 67 | 53 | 45 | 39 | 33 | 26 | 18 | 7 | 3 | 2 | 1 | 0 | 0 |
| SE | 0 | 0 | 0 | 0 | 1 | 47 | 95 | 131 | 150 | 150 | 131 | 97 | 63 | 49 | 41 | 34 | 27 | 18 | 7 | 3 | 2 | 1 | 0 | 0 |
| S | 0 | 0 | 0 | 0 | 0 | 9 | 17 | 25 | 41 | 64 | 85 | 97 | 96 | 84 | 63 | 42 | 31 | 20 | 8 | 4 | 2 | 1 | 0 | 0 |
| SW | 0 | 0 | 0 | 0 | 0 | 9 | 17 | 24 | 30 | 35 | 39 | 64 | 101 | 133 | 151 | 152 | 133 | 93 | 35 | 17 | 8 | 4 | 2 | 1 |
| W | 1 | 0 | 0 | 0 | 0 | 9 | 17 | 24 | 30 | 35 | 38 | 40 | 65 | 114 | 158 | 187 | 192 | 156 | 57 | 27 | 13 | 6 | 3 | 2 |
| NW | 1 | 0 | 0 | 0 | 0 | 9 | 17 | 24 | 30 | 35 | 38 | 40 | 40 | 50 | 84 | 121 | 143 | 130 | 46 | 22 | 11 | 5 | 3 | 1 |
| HOR | 0 | 0 | 0 | 0 | 0 | 24 | 69 | 120 | 169 | 211 | 241 | 257 | 259 | 245 | 217 | 176 | 125 | 70 | 29 | 14 | 7 | 3 | 2 | 1 |

**Table 8. SCL for Sunlit Glass (40° North Latitude, July 21), W/m$^2$**

| | Space Type A | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Hour | | | | | | | | | | | | | | | | | | | | | | | |
| | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 | 13 | 14 | 15 | 16 | 17 | 18 | 19 | 20 | 21 | 22 | 23 | 24 |
| N | 0 | 0 | 0 | 0 | 3 | 79 | 85 | 88 | 101 | 110 | 120 | 126 | 126 | 123 | 113 | 98 | 98 | 113 | 38 | 19 | 9 | 3 | 3 | 0 |
| NE | 0 | 0 | 0 | 0 | 6 | 268 | 406 | 422 | 353 | 236 | 173 | 151 | 139 | 126 | 117 | 101 | 82 | 57 | 22 | 9 | 6 | 3 | 0 | 0 |
| E | 0 | 0 | 0 | 0 | 6 | 293 | 495 | 583 | 576 | 485 | 334 | 211 | 167 | 142 | 123 | 104 | 82 | 57 | 22 | 9 | 6 | 3 | 0 | 0 |
| SE | 0 | 0 | 0 | 0 | 3 | 148 | 299 | 413 | 473 | 473 | 413 | 306 | 198 | 154 | 129 | 107 | 85 | 57 | 22 | 9 | 6 | 3 | 0 | 0 |
| S | 0 | 0 | 0 | 0 | 0 | 28 | 54 | 79 | 129 | 202 | 268 | 306 | 302 | 265 | 198 | 132 | 98 | 63 | 25 | 13 | 6 | 3 | 0 | 0 |
| SW | 0 | 0 | 0 | 0 | 0 | 28 | 54 | 76 | 95 | 110 | 123 | 202 | 318 | 419 | 476 | 479 | 419 | 293 | 110 | 54 | 25 | 13 | 6 | 3 |
| W | 3 | 0 | 0 | 0 | 0 | 28 | 54 | 76 | 95 | 110 | 120 | 126 | 205 | 359 | 498 | 589 | 605 | 491 | 180 | 85 | 41 | 19 | 9 | 6 |
| NW | 3 | 0 | 0 | 0 | 0 | 28 | 54 | 76 | 95 | 110 | 120 | 126 | 126 | 158 | 265 | 381 | 450 | 410 | 145 | 69 | 35 | 16 | 9 | 3 |
| HOR | 0 | 0 | 0 | 0 | 0 | 76 | 217 | 378 | 532 | 665 | 759 | 810 | 816 | 772 | 684 | 554 | 394 | 221 | 91 | 44 | 22 | 9 | 6 | 3 |

Source: *1997 ASHRAE Handbook—Fundamentals*, Chapter 28, Table 36



## Shading Coefficient

shading coefficient at normal incidence

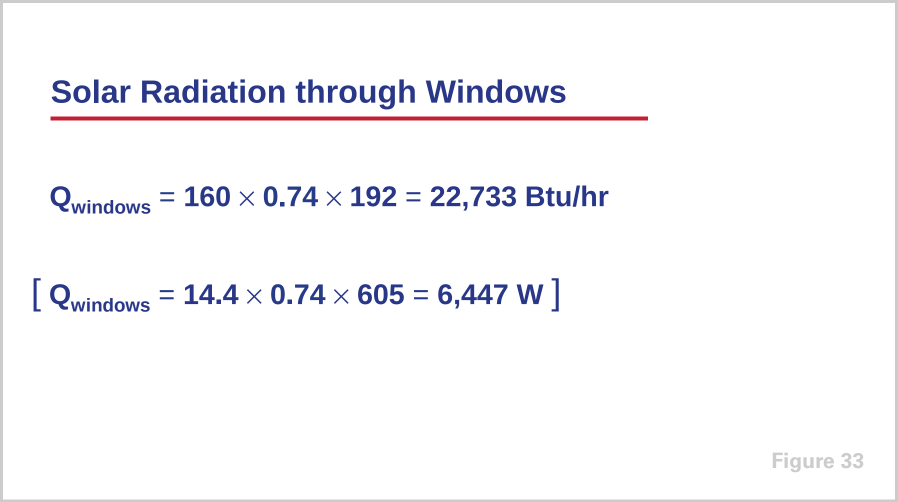
| | aluminum frame operable | aluminum frame fixed | other frames operable | other frames fixed |
|---|---|---|---|---|
| **uncoated single glazing** | | | | |
| 1/4 in. [6.4 mm] clear | 0.82 | 0.85 | 0.69 | 0.82 |
| 1/4 in. [6.4 mm] green | 0.59 | 0.61 | 0.49 | 0.59 |
| **reflective single glazing** | | | | |
| 1/4 in. [6.4 mm] SS on clear | 0.26 | 0.28 | 0.22 | 0.25 |
| 1/4 in. [6.4 mm] SS on green | 0.26 | 0.28 | 0.22 | 0.25 |
| **uncoated double glazing** | | | | |
| 1/4 in. [6.4 mm] clear - clear | 0.70 | 0.74 | 0.60 | 0.70 |
| 1/4 in. [6.4 mm] green - clear | 0.48 | 0.49 | 0.40 | 0.47 |
| **reflective double glazing** | | | | |
| 1/4 in. [6.4 mm] SS on clear - clear | 0.20 | 0.18 | 0.15 | 0.17 |
| 1/4 in. [6.4 mm] SS on green - clear | 0.18 | 0.18 | 0.15 | 0.16 |

SS = stainless-steel reflective coating

Figure 32

The **shading coefficient (SC)** is an expression used to define how much of the radiant solar energy, that strikes the outer surface of the window, is actually transmitted through the window and into the space. The shading coefficient for a particular window is determined by comparing its reflective properties to a standard reference window. The table on this slide includes shading coefficients for common window systems. When the value for the shading coefficient decreases, more of the sun’s rays are reflected by the outer surface of the glass.

The windows in our example space are constructed of two panes of ¼-inch [6.4 mm] clear glass with an air space between the panes. The glass is mounted in an aluminum frame and the windows are fixed (not operable). The SC for this type of window is 0.74.



## Solar Radiation through Windows

$$Q_{windows} = 160 \times 0.74 \times 192 = 22{,}733 \text{ Btu/hr}$$

$$[\ Q_{windows} = 14.4 \times 0.74 \times 605 = 6{,}447 \text{ W}\ ]$$

Figure 33

Now, determine the heat gain by solar radiation through the windows on the west-facing wall of our example space:

Solar radiation heat gain through the windows on the west-facing wall:

- Total area of glass = 8 windows × (4 ft × 5 ft) = 160 $\text{ft}^2$
  [8 × (1.2 m × 1.5 m) = 14.4 $\text{m}^2$]
- SC = 0.74
- $SCL_{hour=17}$ = 192 Btu/hr • $\text{ft}^2$ [605 W/$\text{m}^2$]

$$Q = 160 \times 0.74 \times 192 = 22{,}733 \text{ Btu/hr}$$

$$[Q = 14.4 \times 0.74 \times 605 = 6{,}447 \text{ W}]$$

## Shading Devices

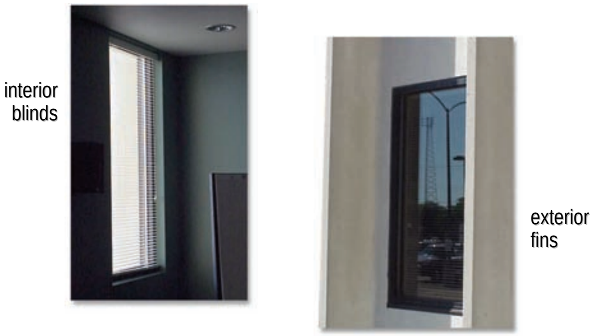


Figure 34

Installing internal shading devices, such as venetian blinds, curtains, or drapes, can reduce the amount of solar heat energy passing through a window. The effectiveness of these shading devices depends on their ability to reflect the incoming solar radiation back through the window, before it is converted into heat inside the space. Light-colored blinds or drapes lined with light-colored materials, therefore, are more effective than dark-colored shading devices. The type of internal shading device used affects the shading coefficient of the window-and-shading-device combination.

External shading devices, such as overhangs, vertical fins, or awnings, can also reduce the amount of solar heat energy passing through a window. They can be used to reduce the area of the glass surface that is actually impacted by the sun's rays.

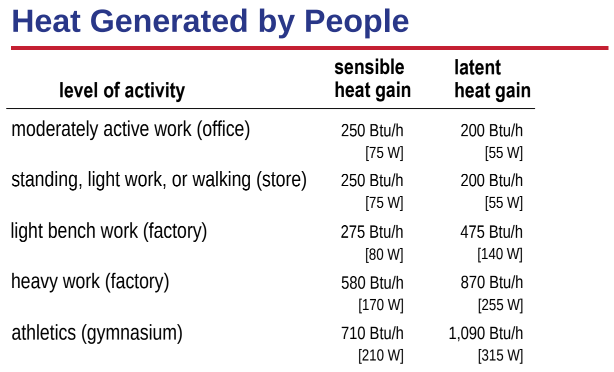
Internal Heat Gains

Figure 35

### Internal Heat Gains

The next component of the space cooling load is the heat that originates within the space. Typical sources of internal heat gain are people, lights, cooking processes, and other heat-generating equipment, such as motors, appliances, and office equipment.

While all of these sources contribute sensible heat to the space, people, cooking processes, and some appliances (such as a coffee maker) also contribute latent heat to the space.



Heat Generated by People

| level of activity | sensible heat gain | latent heat gain |
|---|---|---|
| moderately active work (office) | 250 Btu/h<br>[75 W] | 200 Btu/h<br>[55 W] |
| standing, light work, or walking (store) | 250 Btu/h<br>[75 W] | 200 Btu/h<br>[55 W] |
| light bench work (factory) | 275 Btu/h<br>[80 W] | 475 Btu/h<br>[140 W] |
| heavy work (factory) | 580 Btu/h<br>[170 W] | 870 Btu/h<br>[255 W] |
| athletics (gymnasium) | 710 Btu/h<br>[210 W] | 1,090 Btu/h<br>[315 W] |

Figure 36

As mentioned in Period One, people generate more heat than is needed to maintain body temperature. This surplus heat is dissipated to the surrounding air in the form of sensible and latent heat. The amount of heat released by the body varies with age, physical size, gender, type of clothing, and level of physical activity. This table is an excerpt from the *1997 ASHRAE Handbook—Fundamentals*. It includes typical sensible and latent heat gains per person, based on the level of physical activity. The heat gains are adjusted to account for the normal percentages of men, women, and children in each type of space.

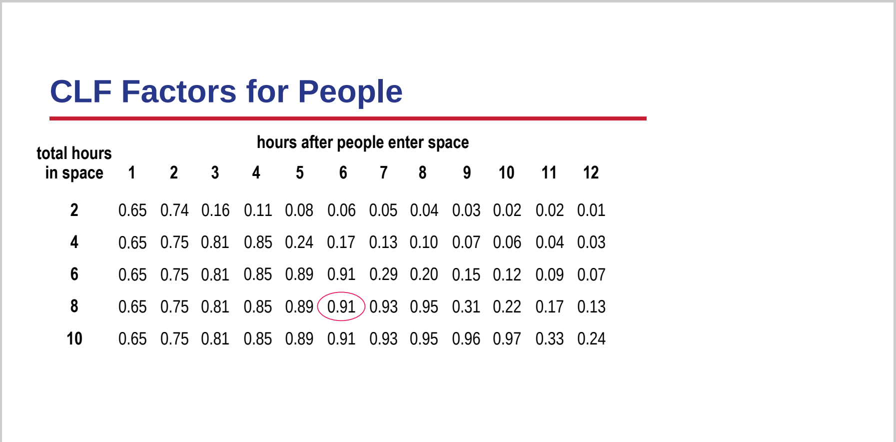
The equations used to predict the sensible and latent heat gains from people in the space are:

$Q_S$ = number of people × sensible heat gain/person × CLF

$Q_L$ = number of people × latent heat gain/person

where,

- $Q_S$ = sensible heat gain from people, Btu/hr [W]
- $Q_L$ = latent heat gain from people, Btu/hr [W]
- CLF = cooling load factor, dimensionless



## CLF Factors for People

| total hours in space | hours after people enter space<br>1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 2 | 0.65 | 0.74 | 0.16 | 0.11 | 0.08 | 0.06 | 0.05 | 0.04 | 0.03 | 0.02 | 0.02 | 0.01 |
| 4 | 0.65 | 0.75 | 0.81 | 0.85 | 0.24 | 0.17 | 0.13 | 0.10 | 0.07 | 0.06 | 0.04 | 0.03 |
| 6 | 0.65 | 0.75 | 0.81 | 0.85 | 0.89 | 0.91 | 0.29 | 0.20 | 0.15 | 0.12 | 0.09 | 0.07 |
| 8 | 0.65 | 0.75 | 0.81 | 0.85 | 0.89 | 0.91 | 0.93 | 0.95 | 0.31 | 0.22 | 0.17 | 0.13 |
| 10 | 0.65 | 0.75 | 0.81 | 0.85 | 0.89 | 0.91 | 0.93 | 0.95 | 0.96 | 0.97 | 0.33 | 0.24 |

Figure 37

Similar to the use of the CLTD for conduction heat gain and SCL for solar heat gain, the **cooling load factor (CLF)** is used to account for the capacity of the space to absorb and store heat. Some of the sensible heat generated by people is absorbed and stored by the walls, floor, ceiling, and furnishings of the space, and released at a later time. Similar to heat transfer by conduction through an external wall, the space can therefore experience a time lag between the time that the sensible heat is originally generated and the time that it actually contributes to the space cooling load. For heat gain from people, the value of CLF depends on 1) the construction of the interior partition walls in the space, 2) the type of floor covering, 3) the total number of hours that the space is occupied, and 4) the number of hours since the people entered the space.

Figure 37, CLF Factors for People, is an excerpt from the *1997 ASHRAE Handbook—Fundamentals*. It shows that one hour after people enter the space, 35% (1– 0.65) of the sensible heat gain from the people is absorbed by the surfaces and furnishings in the space, and 65% is the actual cooling load in the space. Following the table to the right, however, you see that, as the people are in the space for a longer period of time, the surfaces and furnishings of the space can no longer absorb as much heat, and they release the heat that was absorbed earlier in the day. For example, if the people enter the space at 8 a.m. and remain for a total of 8 hours, at 2 p.m. (6 hours after entering) 91% of the sensible heat gain from the people is seen as a cooling load in the space. Only 9% is absorbed by the surfaces and furnishings of the space.

If the space is not maintained at a constant temperature during the 24-hour period, however, the CLF is assumed to equal 1.0. Most air-conditioning systems designed for non-residential buildings either shut the system off at night or raise the temperature set point to reduce energy use. Thus, it is uncommon to use a CLF other than 1.0 for the cooling load due to people.

**Heat Gain from People**

$Q_{\text{sensible}} = 18 \times 250 \times 1.0 = 4{,}500\ \text{Btu/hr}$

$Q_{\text{latent}} = 18 \times 200 = 3{,}600\ \text{Btu/hr}$

$\left[\ Q_{\text{sensible}} = 18 \times 75 \times 1.0 = 1{,}350\ \text{W}\ \right]$

$\left[\ Q_{\text{latent}} = 18 \times 55 = 990\ \text{W}\ \right]$

Figure 38

Determine the internal heat gain from people in our example space. Based on the table in Figure 36, people participating in moderately active office work generate 250 Btu/hr [75 W] sensible heat and 200 Btu/hr [55 W] latent heat.

Internal heat gain from people:

- Number of people = 18
- Sensible heat gain/person = 250 Btu/hr [75 W]
- Latent heat gain/person = 200 Btu/hr [55 W]
- CLF = 1.0 (because the space temperature set point is increased at night)

$Q_S$ = 18 people × 250 Btu/hr per person × 1.0 = 4,500 Btu/hr

[$Q_S$ = 18 people × 75 W per person × 1.0 = 1,350 W]

$Q_l$ = 18 people × 200 Btu/hr per person = 3,600 Btu/hr

[$Q_l$ = 18 people × 55 W per person = 990 W]

## Heat Gain from Lighting

$$Q = \text{watts} \times 3.41 \times \text{ballast factor} \times \text{CLF}$$

$$[\,Q = \text{watts} \times \text{ballast factor} \times \text{CLF}\,]$$

Figure 39

Heat generated by lights in the space is a significant contribution to the cooling load. For example, a 120-watt light fixture generates 410 Btu/hr [120 W] of heat—approximately the same amount of heat gain generated by an average office worker.

Additionally, when estimating the heat gain from fluorescent lights, approximately 20% is added to the lighting heat gain to account for the additional heat generated by the ballast.

The equation used to estimate the heat gain from lighting is:

$$Q = \text{watts} \times 3.41 \times \text{ballast factor} \times \text{CLF}$$

$$[Q = \text{watts} \times \text{ballast factor} \times \text{CLF}]$$

where,

- Q = sensible heat gain from lighting, Btu/hr [W]
- Watts = total energy input to lights, W
- 3.41 = conversion factor from W to Btu/hr (when using I-P units)
- Ballast factor = 1.2 for fluorescent lights, 1.0 for incandescent lights
- CLF = cooling load factor, dimensionless

Similar to the sensible heat gain from people, a cooling load factor (CLF) can be used to account for the capacity of the space to absorb and store the heat generated by the lights. If the lights are left on 24 hours a day, or if the air-conditioning system is shut off or set back at night, the CLF is assumed to be equal to 1.0.

## Heat Gain from Lighting

$$Q_{\text{lights}} = 5{,}400 \times 3.41 \times 1.2 \times 1.0 = 22{,}097 \text{ Btu/hr}$$

$$\left[\, Q_{\text{lights}} = 5{,}400 \times 1.2 \times 1.0 = 6{,}480 \text{ W} \,\right]$$

Figure 40

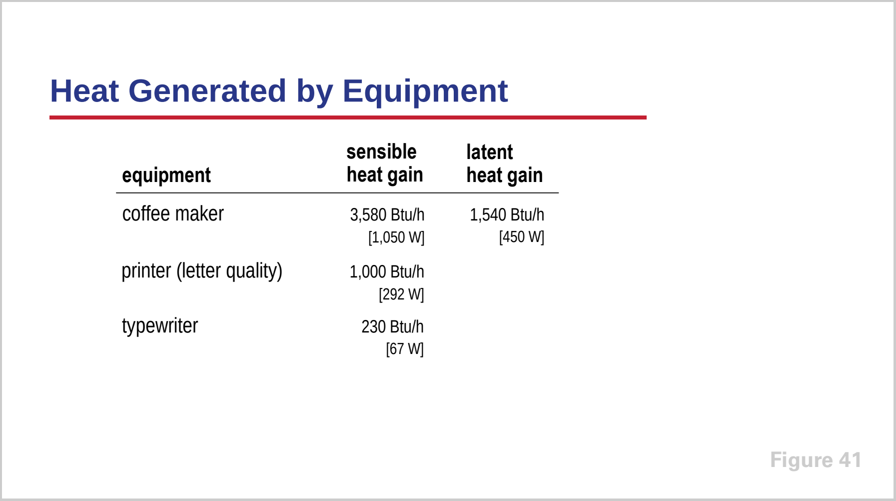
Next, determine the internal heat gain from lighting in our example space.

Internal heat gain from lighting:

- Amount of lighting in space = 2 W/ft$^2$ [21.5 W/m$^2$]
- Floor area = 45 ft × 60 ft = 2,700 ft$^2$ [13.7 m × 18.3 m = 250.7 m$^2$]
- Total lighting energy = 2 W/ft$^2$ × 2,700 ft$^2$ = 5,400 W
  [21.5 W/m$^2$ × 250.7 m$^2$ = 5,400 W]
- Ballast factor = 1.2 (fluorescent lights)
- CLF = 1.0 (because the space temperature set point is increased at night)

$$Q = 5{,}400 \times 3.41 \times 1.2 \times 1.0 = 22{,}097 \text{ Btu/hr}$$

$$[Q = 5{,}400 \times 1.2 \times 1.0 = 6{,}480 \text{ W}]$$



## Heat Generated by Equipment

| equipment | sensible heat gain | latent heat gain |
|---|---|---|
| coffee maker | 3,580 Btu/h<br>[1,050 W] | 1,540 Btu/h<br>[450 W] |
| printer (letter quality) | 1,000 Btu/h<br>[292 W] | |
| typewriter | 230 Btu/h<br>[67 W] | |

Figure 41

There are many types of appliances and equipment in restaurants, schools, office buildings, hospitals, and other types of buildings. This equipment may generate a significant amount of heat and should be accounted for when estimating the space cooling load.

The data on this slide is an excerpt from the *1997 ASHRAE Handbook—Fundamentals*. The handbook contains tables of sensible and latent heat gains from various types of office and restaurant equipment, although data for the actual piece of equipment is preferred, if available.

Using this table, we estimate that the coffee maker contributes 3,580 Btu/hr [1,050 W] of sensible heat and 1,540 Btu/hr [450 W] of latent heat to our example space. Additionally, we are told that there are 0.5 W/ft$^2$ [5.4 W/m$^2$] of computers and other office equipment in the space (floor area = 2,700 ft$^2$ [250.7 m$^2$]).

Therefore, the internal heat gain from computers and office equipment is:

$$\text{Sensible heat gain} = 0.5\ \text{W/ft}^2 \times 2{,}700\ \text{ft}^2 \times 3.41\ \text{Btu/hr/W} = 4{,}604\ \text{Btu/hr}$$

$$[5.4\ \text{W/m}^2 \times 250.7\ \text{m}^2 = 1{,}354\ \text{W}]$$

Similar to the sensible heat gain from people and lighting, tables of cooling load factors (CLF) can be used to refine this estimate. If the equipment is left on 24 hours a day, or if the air-conditioning system is shut off or set back at night, the CLF is assumed to be equal to 1.0. In our example, the CLF is 1.0 because the space temperature set point is increased at night.

notes

## Infiltration

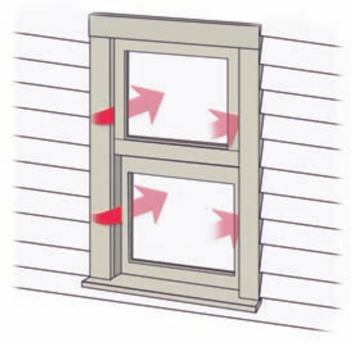


Figure 42

### Infiltration

In a typical building, air leaks into or out of a space through doors, windows, and small cracks in the building envelope. Air leaking **into** a space is called **infiltration**. During the cooling season, when air leaks into a conditioned space from outdoors, it can contribute to both the sensible and latent heat gain in the space because the outdoor air is typically warmer and more humid than the indoor air.

## Methods of Estimating Infiltration

- Air change method
- Crack method
- Effective leakage-area method

Figure 43

Before estimating the heat gain from infiltration, we must first estimate the amount of air that is leaking into the space. There are three methods commonly used to estimate infiltration airflow.

The **air change method** is the easiest, but may be the least accurate of these methods. It involves estimating the number of air changes per hour that can be expected in spaces of a certain construction quality. Using this method, the quantity of infiltration air is estimated using the equation:

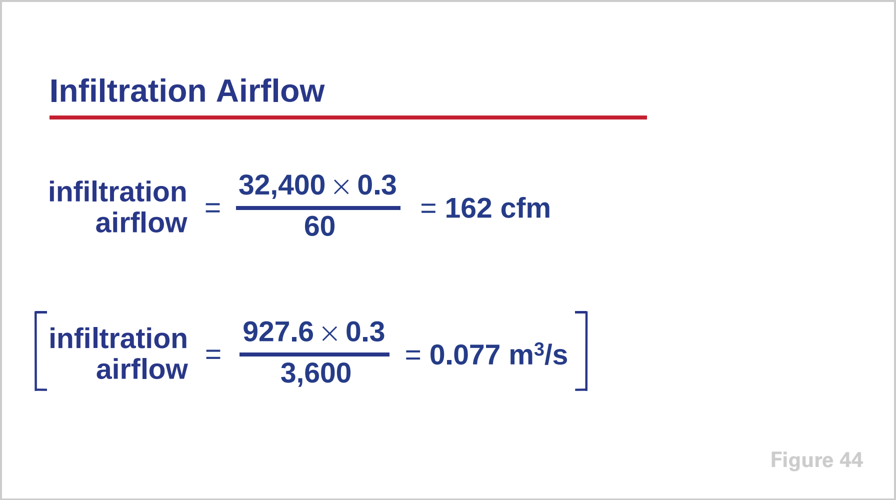
infiltration airflow = (volume of space × air change rate) ÷ 60

[infiltration airflow = (volume of space × air change rate) ÷ 3,600]

where,

- Infiltration airflow = quantity of air infiltrating into the space, cfm [$m^3/s$]
- Volume of space = length × width × height of space, $ft^3$ [$m^3$]
- Air change rate = air changes per hour
- 60 = conversion from hours to minutes
- 3,600 = conversion from hours to seconds

The **crack method** is a little more complex and is based upon the average quantity of air known to enter through cracks around windows and doors when the wind velocity is constant. The **effective leakage-area method** takes wind speed, shielding, and “stack effect” into account, and requires a very detailed calculation.



## Infiltration Airflow

$$\text{infiltration airflow} = \frac{32{,}400 \times 0.3}{60} = 162 \text{ cfm}$$

$$\left[\text{infiltration airflow} = \frac{927.6 \times 0.3}{3{,}600} = 0.077 \text{ m}^3\text{/s}\right]$$

Figure 44

The table below includes estimates for infiltration using the air change method. Assuming that the space in our example is of average construction and kept at a positive pressure relative to the outdoors, we estimate 0.3 air changes/hr of infiltration.

Volume of space = 45 ft × 60 ft × 12 ft = 32,400 $\text{ft}^3$
[13.7 m × 18.3 m × 3.7 m = 927.6 $\text{m}^3$]

$$\text{infiltration airflow} = \frac{32{,}400 \times 0.3}{60} = 162 \text{ cfm}$$

$$\left[\text{infiltration airflow} = \frac{927.6 \times 0.3}{3{,}600} = 0.077 \text{ m}^3\text{/s}\right]$$

**Table 9. Estimates of Infiltration Airflow, Air Changes Per Hour**

| | |
|---|---|
| Neutral pressure, poor construction | 1.0 |
| Neutral pressure, average construction | 0.6 |
| Neutral pressure, tight construction | 0.3 |
| Pressurized, poor construction | 0.5 |
| Pressurized, average construction | [✓ 0.3] |
| Pressurized, tight construction | 0.0 |

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## Heat Gain from Infiltration

$$Q_{sensible} = 1.085 \times \text{airflow} \times \Delta T$$

$$Q_{latent} = 0.7 \times \text{airflow} \times \Delta W$$

$$[\ Q_{sensible} = 1{,}210 \times \text{airflow} \times \Delta T\ ]$$

$$[\ Q_{latent} = 3{,}010 \times \text{airflow} \times \Delta W\ ]$$

Figure 45

The equation used to estimate the sensible heat gain from infiltration is:

$$Q_S = 1.085 \times \text{airflow} \times \Delta T$$

$$[Q_S = 1{,}210 \times \text{airflow} \times \Delta T]$$

where,

- $Q_S$ = sensible heat gain from infiltration, Btu/hr [W]
- 1.085 [1,210] = product of density and specific heat, Btu • min/hr • $ft^3$ • °F [$J/m^3$ • °K]
- Airflow = quantity of air infiltrating the space, cfm [$m^3/s$]
- ΔT = design outdoor dry-bulb temperature minus the desired indoor dry-bulb temperature, °F [°C]

The equation used to estimate the latent heat gain from infiltration is:

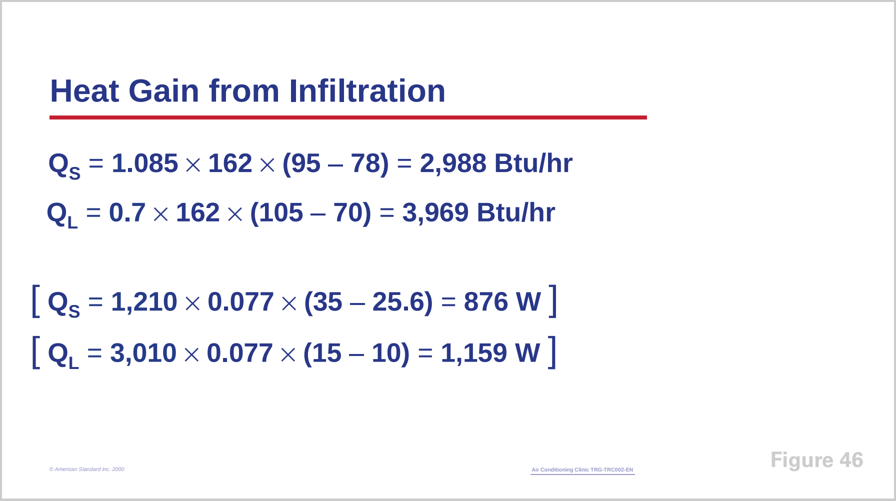
$$Q_L = 0.7 \times \text{airflow} \times \Delta W$$

$$[Q_L = 3{,}010 \times \text{airflow} \times \Delta W]$$

where,

- $Q_L$ = latent heat gain from infiltration, Btu/hr [W]
- 0.7 [3,010] = latent heat factor, Btu • min • lb/hr • $ft^3$ • gr [J • $kg/m^3$ • g]
- Airflow = quantity of air infiltrating the space, cfm [$m^3/s$]
- ΔW = design outdoor humidity ratio minus the desired indoor humidity ratio, grains of water/lb of dry air [grams of water/kg of dry air]

The psychrometric chart can be used to determine the humidity ratio for both outdoor and indoor conditions.



**Heat Gain from Infiltration**

$Q_S = 1.085 \times 162 \times (95 - 78) = 2{,}988$ Btu/hr

$Q_L = 0.7 \times 162 \times (105 - 70) = 3{,}969$ Btu/hr

[ $Q_S = 1{,}210 \times 0.077 \times (35 - 25.6) = 876$ W ]

[ $Q_L = 3{,}010 \times 0.077 \times (15 - 10) = 1{,}159$ W ]

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Figure 46

Heat gain from infiltration:

- Infiltration airflow = 162 cfm [0.077 $m^3/s$]
- Outdoor conditions: 95°F [35°C] dry bulb and 76°F [25°C] wet bulb results in $W_o$ = 105 grains of water/lb dry air [15 grams of water/kg dry air]
- Indoor conditions: 78°F [25.6°C] dry bulb and 50% relative humidity results in $W_i$ = 70 grains of water/lb dry air [10 grams of water/kg dry air]

$Q_S = 1.085 \times 162 \times (95 - 78) = 2{,}988$ Btu/hr

[$Q_S = 1{,}210 \times 0.077 \times (35 - 25.6) = 876$ W]

$Q_L = 0.7 \times 162 \times (105 - 70) = 3{,}969$ Btu/hr

[$Q_L = 3{,}010 \times 0.077 \times (15 - 10) = 1{,}159$ W]

Realize that 1.085 and 0.7 [1,210 and 3,010] are not constants, but are derived from properties of air at “standard” conditions (69°F [21°C] dry air at sea level). Air at other conditions and elevations will cause these factors to change.

- Density = 0.075 $lb/ft^3$ [1.2 $kg/m^3$]
- Specific heat = 0.24 Btu/lb • °F [1,004 J/kg • °K]
- Latent heat of water vapor = 1,076 Btu/lb [2,503 kJ/kg]

$0.075 \times 0.24 \times 60$ min/hr $= 1.085$ [$1.2 \times 1{,}004 = 1{,}210$]

$$\frac{0.075 \times 1{,}076 \times 60 \text{ min/hr}}{7000 \text{ grains/lb}} = 0.7 \left[\frac{1.2 \times 2{,}503 \times 1{,}000 \text{ J/kJ}}{1{,}000 \text{ g/kg}} = 3{,}010\right]$$

## Summary of Space Cooling Loads

| space load components | sensible load<br>Btu/hr [W] | latent load<br>Btu/hr [W] |
|---|---|---|
| conduction through roof | 12,312 [3,563] | |
| conduction through exterior wall | 502 [144] | |
| conduction through windows | 1,310 [359] | |
| solar radiation through windows | 22,733 [6,447] | |
| people | 4,500 [1,350] | 3,600 [990] |
| lights | 22,097 [6,480] | |
| equipment | 8,184 [2,404] | 1,540 [450] |
| infiltration | 2,988 [876] | 3,969 [1,159] |
| **total space cooling load** | **74,626 [21,623]** | **9,109 [2,599]** |

Figure 47

This completes the estimation of the components of the cooling load for the space. The total space cooling load will be used during a simplified psychrometric analysis in Period Three to determine the quantity and temperature of air required to condition this space.

As mentioned earlier, the space used in this example has no plenum, so all of the heat gain from the roof and lights affects the space directly. Applications in which this assumption does not apply will be discussed briefly in Period Five.

In addition to these space cooling loads, there are other loads that affect the cooling coil in the building HVAC system. These include the load of the outdoor air, deliberately brought into the building for ventilation purposes, and heat generated by the fans in the system. These loads are added to the space load to determine the total cooling load for the building. Estimating these additional components is necessary to properly size the cooling coil for the system.

notes

## Ventilation

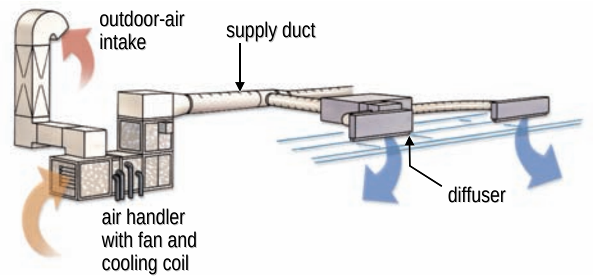
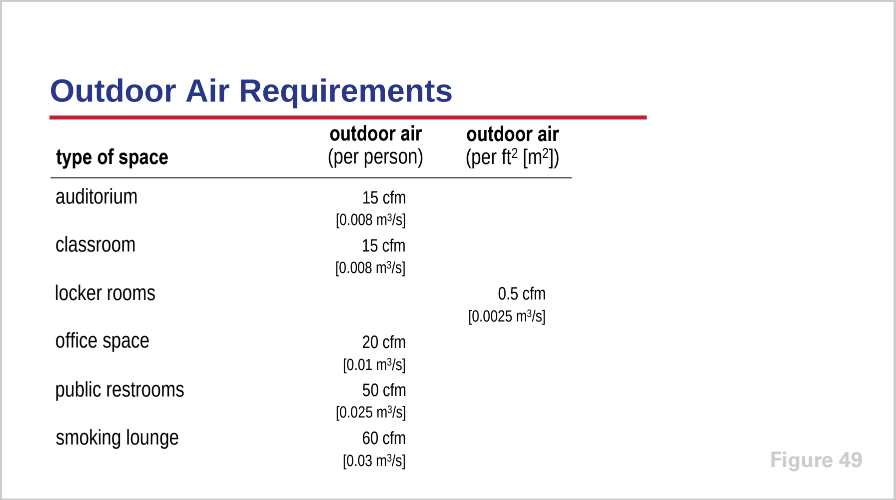


Figure 48

### Ventilation

Outdoor air is often used to dilute or remove contaminants from the indoor air. The intentional introduction of outdoor air into a space, through the use of the building's HVAC system, is called **ventilation**. This outdoor air must often be cooled and dehumidified before it can be delivered to the space, creating an additional load on the air-conditioning equipment.

You should never depend on infiltration to satisfy the ventilation requirement of a space. On days when the outdoor air is not moving (due to wind), the amount of infiltration can drop to zero. Instead, it is common to introduce outdoor air through the HVAC system, not only to meet the ventilation needs, but also to maintain a positive pressure (relative to the outdoors) within the building. This positive pressure reduces, or may even eliminate, the infiltration of unconditioned air from outdoors. To pressurize the building, the amount of outdoor air brought in for ventilation must be greater than the amount of air exhausted through central and local exhaust fans.



## Outdoor Air Requirements

| **type of space** | **outdoor air** (per person) | **outdoor air** (per ft$^2$ [m$^2$]) |
|---|---|---|
| auditorium | 15 cfm<br>[0.008 m$^3$/s] | |
| classroom | 15 cfm<br>[0.008 m$^3$/s] | |
| locker rooms | | 0.5 cfm<br>[0.0025 m$^3$/s] |
| office space | 20 cfm<br>[0.01 m$^3$/s] | |
| public restrooms | 50 cfm<br>[0.025 m$^3$/s] | |
| smoking lounge | 60 cfm<br>[0.03 m$^3$/s] | |

Figure 49

The amount of outdoor air required for a space is often prescribed by local building codes or industry standards. One such standard, ASHRAE Standard 62, *Ventilation for Acceptable Air Quality*, prescribes the quantity of outdoor air required per person (or per unit area) to provide adequate ventilation for various types of spaces. The data in Figure 49 are an excerpt from this standard.

In our example, calculating the required quantity of outdoor air involves multiplying the number of people in the space by the 20 cfm [0.01 m$^3$/s] of outdoor air required per person in an office space.

ventilation airflow = 18 people × 20 cfm/person = 360 cfm

[ventilation airflow = 18 people × 0.01 m$^3$/s/person = 0.18 m$^3$/s]

### Cooling Load Due to Ventilation

$Q_S = 1.085 \times 360 \times (95 - 78) = 6{,}640$ **Btu/hr**

$Q_L = 0.7 \times 360 \times (105 - 70) = 8{,}820$ **Btu/hr**

[ $Q_S = 1{,}210 \times 0.18 \times (35 - 25.6) = 2{,}047$ **W** ]

[ $Q_L = 3{,}010 \times 0.18 \times (15 - 10) = 2{,}709$ **W** ]

Figure 50

The sensible and latent loads from ventilation are calculated using the same equations as for infiltration:

$Q_S = 1.085 \times \text{airflow} \times \Delta T$

$[Q_S = 1{,}210 \times \text{airflow} \times \Delta T]$

$Q_L = 0.7 \times \text{airflow} \times \Delta W$

$[Q_L = 3{,}010 \times \text{airflow} \times \Delta W]$

Cooling load due to the conditioning of ventilation air:

- Ventilation airflow = 360 cfm [0.18 $m^3/s$]
- Outdoor conditions: $T_o$ = 95°F [35°C], $W_o$ = 105 grains of water/lb dry air [15 grams of water/kg dry air]
- Indoor conditions: $T_i$ = 78°F [25.6°C], $W_i$ = 70 grains of water/lb dry air [10 grams of water/kg dry air]

$Q_S = 1.085 \times 360 \times (95 - 78) = 6{,}640$ Btu/hr

$[Q_S = 1{,}210 \times 0.18 \times (35 - 25.6) = 2047 \text{ W}]$

$Q_L = 0.7 \times 360 \times (105 - 70) = 8{,}820$ Btu/hr

$[Q_L = 3{,}010 \times 0.18 \times (15 - 10) = 2{,}709 \text{ W}]$

## System Heat Gains

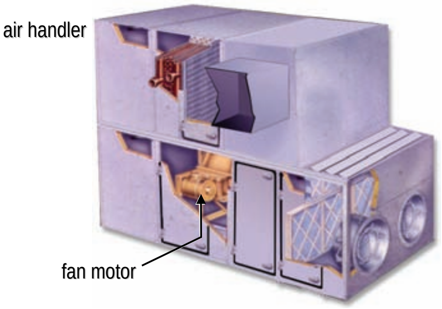


Figure 51

### System Heat Gains

There may be others sources of heat gain within the HVAC system. One example is the heat generated by fans. When the supply fan, driven by an electric motor, is located in the conditioned airstream, it adds heat to the air. Heat gain from a fan is associated with three energy conversion losses.

Fan motor heat is due to the energy lost in the conversion of electrical energy (energy input to the motor) to mechanical energy (rotation of the motor shaft). It is dissipated as heat from the motor and is represented by the inefficiency of the motor.

fan motor heat gain = power input to motor × (1 – motor efficiency)

If the fan motor is also located within the conditioned airstream, such as inside the cabinet of an air handler (as shown in Figure 51), it is considered an instantaneous heat gain to the airstream. If it is located outside the conditioned airstream, it is considered a heat gain to the space where it is located.

Fan-blade heat gain is due to the energy lost in the conversion of mechanical energy to kinetic energy (moving of the air). It is dissipated as heat from the fan blades, it is considered an instantaneous heat gain to the airstream, and it is represented by the inefficiency of the fan.

fan blade heat gain = power input to fan × (1 – fan efficiency)

Finally, the remaining (useful) energy input to the fan, the energy used to pressurize the supply duct system, is eventually converted to heat as the air travels through the ductwork. For simplicity, most designers assume that this heat gain occurs at a single point in the system, typically at the location of the fan.

duct friction heat gain = power input to fan × fan efficiency

### Components of Fan Heat

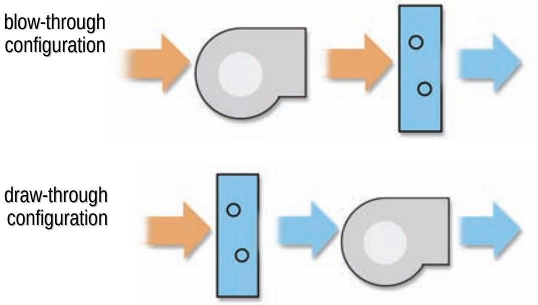


Figure 52

It is important to know where the fan heat gain occurs with respect to the cooling coil. If the fan is located upstream and blows air through the cooling coil, the fan heat causes an increase in the temperature of the air entering the coil.

If, however, the fan is located downstream and draws air through the cooling coil, the fan heat causes an increase in the temperature of the air supplied to the space.

### Heat Gain in Ductwork

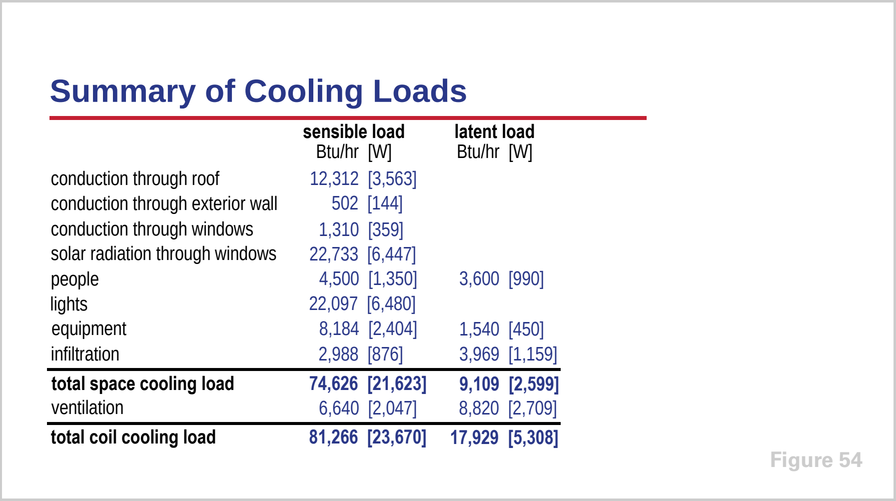
Figure 53

Another source of heat gain in the system may be heat that is transferred to the conditioned air through the walls of the supply and return ductwork. For example, if the supply ductwork is routed through an unconditioned space, such as a ceiling plenum or an attic, heat can be transferred from the air surrounding the duct to the supply air.

Supply ductwork is generally insulated to prevent this heat gain and the associated increase in temperature of the supply air. An increased supply air temperature requires a greater amount of supply air to maintain the desired space conditions, resulting in more fan energy use. Insulation also reduces the risk of condensation on the cool, outer surfaces of the duct.

Return ductwork, on the other hand, is generally not insulated unless it passes through a very warm space. Any heat picked up by the return air is generally heat that would have eventually entered the space as a cooling load. Therefore, the cooling load caused by this heat gain to the return air is not wasted.

For the example used in this clinic, we will assume that fan heat gains and other system heat gains are negligible.



## Summary of Cooling Loads

| | sensible load | | latent load | |
|---|---|---|---|---|
| | Btu/hr | [W] | Btu/hr | [W] |
| conduction through roof | 12,312 | [3,563] | | |
| conduction through exterior wall | 502 | [144] | | |
| conduction through windows | 1,310 | [359] | | |
| solar radiation through windows | 22,733 | [6,447] | | |
| people | 4,500 | [1,350] | 3,600 | [990] |
| lights | 22,097 | [6,480] | | |
| equipment | 8,184 | [2,404] | 1,540 | [450] |
| infiltration | 2,988 | [876] | 3,969 | [1,159] |
| **total space cooling load** | **74,626** | **[21,623]** | **9,109** | **[2,599]** |
| ventilation | 6,640 | [2,047] | 8,820 | [2,709] |
| **total coil cooling load** | **81,266** | **[23,670]** | **17,929** | **[5,308]** |

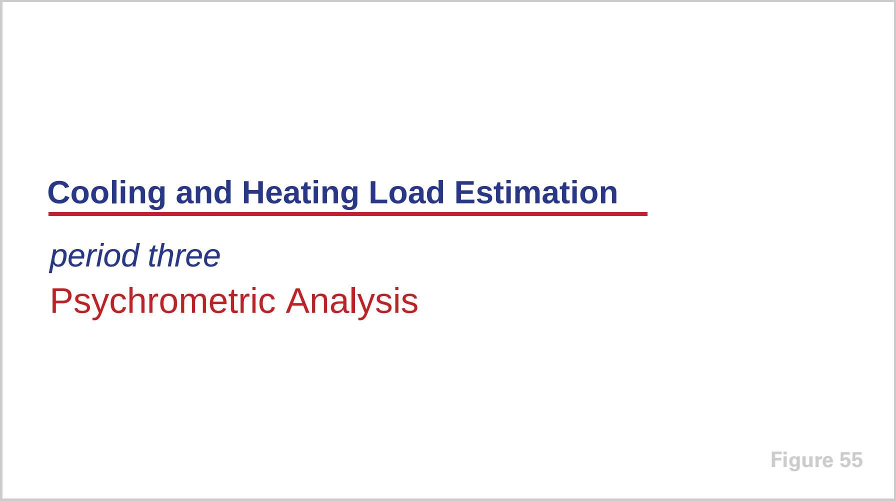
Figure 54

In summary, the total cooling load for our example space is made up of the following components:

- Conduction heat gain from outdoors through the roof and west-facing exterior wall and windows
- Solar radiation heat gain through the west-facing windows
- Internal heat gains from people, lights, office equipment, and a coffee maker in the space
- Heat gain due to hot, humid air infiltrating into the space from outdoors

In addition, the cooling coil in the building HVAC system has to cool the outdoor air that is deliberately brought into the building for ventilation purposes.

We will use these results in the next period to conduct a simplified psychrometric analysis of our example space.



## Cooling and Heating Load Estimation

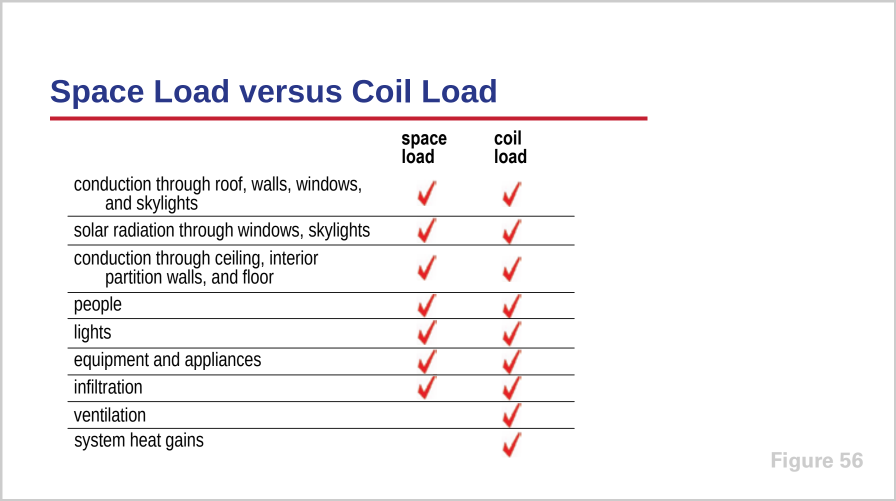
*period three*

Psychrometric Analysis

Figure 55

In this period we will use the results from our example cooling load calculations to 1) determine the sensible heat ratio for the space, 2) perform a simplified psychrometric analysis to calculate the supply airflow and temperature required to properly condition the space, and 3) calculate the required capacity of the cooling coil.

Then we will analyze a system that serves multiple spaces to determine the design airflow for the supply fan and the total building cooling load.



## Space Load versus Coil Load

| | space load | coil load |
|---|---|---|
| conduction through roof, walls, windows, and skylights | ✓ | ✓ |
| solar radiation through windows, skylights | ✓ | ✓ |
| conduction through ceiling, interior partition walls, and floor | ✓ | ✓ |
| people | ✓ | ✓ |
| lights | ✓ | ✓ |
| equipment and appliances | ✓ | ✓ |
| infiltration | ✓ | ✓ |
| ventilation | | ✓ |
| system heat gains | | ✓ |

Figure 56

### Single-Space Analysis

The first step in our psychrometric analysis is to determine which components of the cooling load are space loads and which only affect the coil load. This is important because, although all heat gains that occur inside the building contribute to the total load on the cooling coil, only those heat gains that occur within the space need to be offset by the cool air supplied to the space. Notice that all space loads are also coil loads, but all coil loads are not necessarily also space loads.

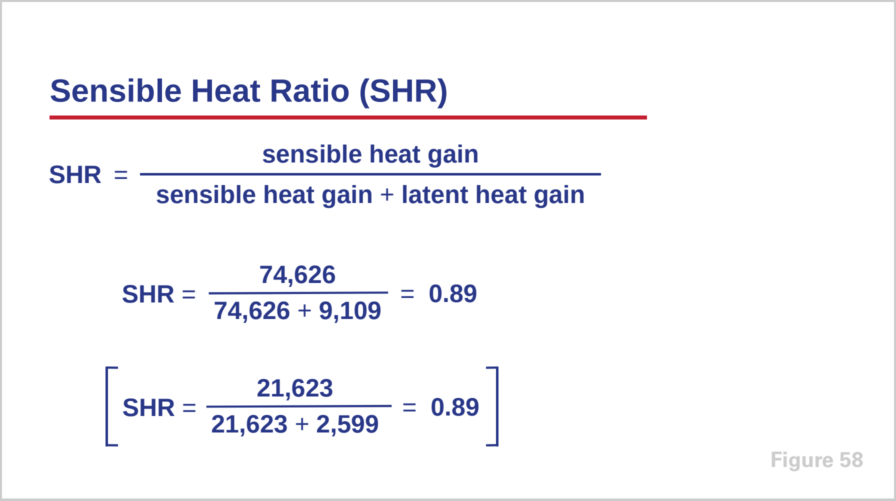
For example, in most buildings, ventilation air is conditioned prior to being delivered to the space. Therefore, the ventilation load adds to the total cooling coil load, but does not add to the space cooling load. Additionally, heat gains that occur within the HVAC system, such as fan heat and duct heat gain, are considered coil loads, but not space loads.

## Space Sensible and Latent Loads

| **space load components** | **sensible load** Btu/hr [W] | **latent load** Btu/hr [W] |
|---|---|---|
| conduction through roof | 12,312 [3,563] | |
| conduction through exterior wall | 502 [144] | |
| conduction through windows | 1,310 [359] | |
| solar radiation through windows | 22,733 [6,447] | |
| people | 4,500 [1,350] | 3,600 [990] |
| lights | 22,097 [6,480] | |
| equipment | 8,184 [2,404] | 1,540 [450] |
| infiltration | 2,988 [876] | 3,969 [1,159] |
| **total space cooling load** | **74,626 [21,623]** | **9,109 [2,599]** |

Figure 57

This table lists only the space cooling loads calculated for the example used in Period Two. The total sensible cooling load for this space is 74,626 Btu/hr [21,623 W] and the total latent cooling load for this space is 9,109 Btu/hr [2,599 W].



## Sensible Heat Ratio (SHR)

$$\text{SHR} = \frac{\text{sensible heat gain}}{\text{sensible heat gain} + \text{latent heat gain}}$$

$$\text{SHR} = \frac{74{,}626}{74{,}626 + 9{,}109} = 0.89$$

$$\left[\text{SHR} = \frac{21{,}623}{21{,}623 + 2{,}599} = 0.89\right]$$

Figure 58

The proportions of sensible and latent heat must be known in order to determine the proper condition of the air being supplied to cool the space. This **sensible heat ratio (SHR)** is the ratio of sensible heat gain to total (sensible plus latent) heat gain, and is defined as follows:

$$\text{SHR} = \frac{\text{sensible heat gain}}{\text{sensible heat gain} + \text{latent heat gain}}$$

$$\text{SHR} = \frac{74{,}626\ \text{Btu/hr}}{74{,}626\ \text{Btu/hr} + 9{,}109\ \text{Btu/hr}}\ \left[\frac{21{,}623\ \text{W}}{21{,}623\ \text{W} + 2{,}599\ \text{W}}\right] = 0.89$$

The SHR for our example space is 0.89. That is, 89% of the cooling load for this space is sensible and 11% is latent.

## Single-Space Analysis

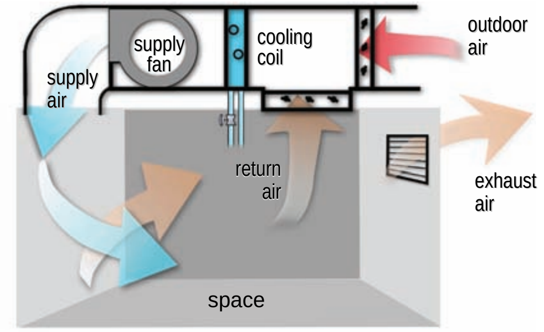
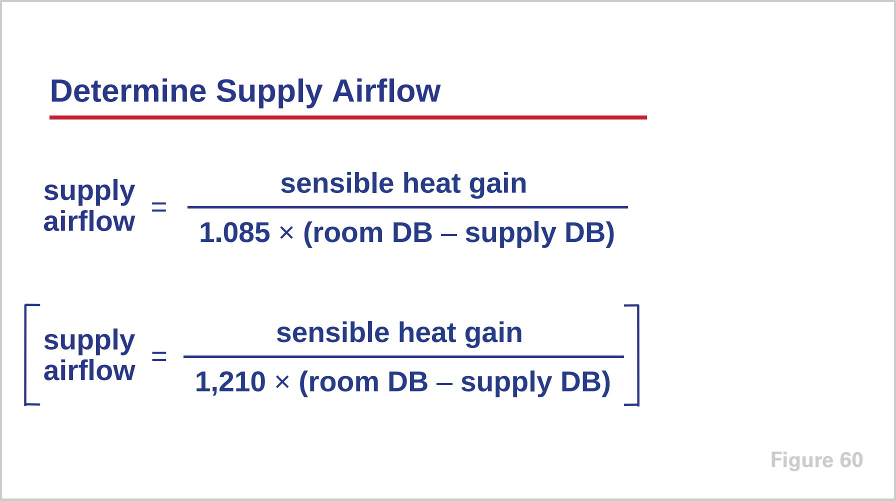


Figure 59

After the SHR has been determined for the space, a simple psychrometric analysis can be performed to determine the quantity of air that must be supplied to condition this space, and the proper temperature of that air. The next few illustrations will use some of the basic concepts presented in the *Psychrometry* Air Conditioning Clinic.

This analysis assumes that the example space is served by its own dedicated air-conditioning system, consisting of a cooling coil and supply fan.

notes



## Determine Supply Airflow

$$\text{supply airflow} = \frac{\text{sensible heat gain}}{1.085 \times (\text{room DB} - \text{supply DB})}$$

$$\left[\text{supply airflow} = \frac{\text{sensible heat gain}}{1{,}210 \times (\text{room DB} - \text{supply DB})}\right]$$

Figure 60

The quantity of air required to offset the space sensible heat gain is determined using the following formula:

$$\text{supply airflow} = \frac{\text{sensible heat gain}}{1.085 \times (\text{room DB} - \text{supply DB})}$$

$$\left[\text{supply airflow} = \frac{\text{sensible heat gain}}{1{,}210 \times (\text{room DB} - \text{supply DB})}\right]$$

where,

- Sensible heat gain = sensible heat gain in the space, Btu/hr [W]
- 1.085 [1,210] = product of density and specific heat, Btu • min/hr • $\text{ft}^3$ • °F [J/$\text{m}^3$ • °K]
- Supply airflow = quantity of air supplied to the space, cfm [$\text{m}^3$/s]
- Room DB = desired space dry-bulb temperature, °F [°C]
- Supply DB = supply air dry-bulb temperature, °F [°C]

Remember that 1.085 [1,210] is not a constant—it is derived from the density and specific heat of the air at actual conditions.

## Determine Supply Airflow

$$\text{supply airflow} = \frac{74{,}626}{1.085 \times (78 - 55)} = 2{,}990 \text{ cfm}$$

$$\left[\text{supply airflow} = \frac{21{,}623}{1{,}210 \times (25.6 - 12.8)} = 1.40 \text{ m}^3\text{/s}\right]$$

Figure 61

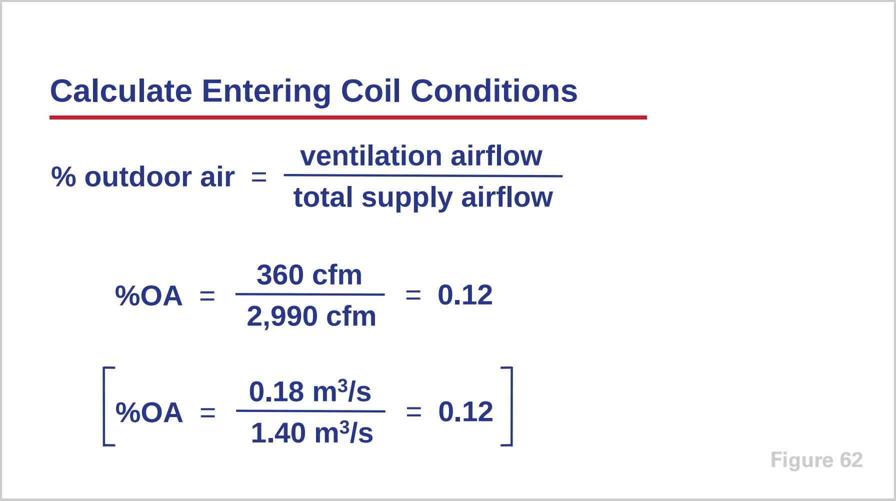
The next step is to either assume the supply air dry-bulb temperature and calculate the supply airflow, or assume the supply airflow and calculate the supply air temperature.

For our example, we will assume that the supply air dry-bulb temperature is 55°F [12.8°C]. Based on this assumption, the quantity of air required to offset the sensible heat gain in this space is 2,990 cfm [1.40 $m^3$/s].

$$\text{supply airflow} = \frac{74{,}626 \text{ Btu/hr}}{1.085 \times (78°\text{F} - 55°\text{F})} = 2{,}990 \text{ cfm}$$

$$\left[\text{supply airflow} = \frac{21{,}623 \text{ W}}{1{,}210 \times (25.6°\text{C} - 12.8°\text{C})} = 1.40 \text{ m}^3\text{/s}\right]$$

The rest of the analysis will fine-tune this assumption.



## Calculate Entering Coil Conditions

$$\%\text{ outdoor air} = \frac{\text{ventilation airflow}}{\text{total supply airflow}}$$

$$\%\text{OA} = \frac{360\text{ cfm}}{2{,}990\text{ cfm}} = 0.12$$

$$\left[\%\text{OA} = \frac{0.18\text{ m}^3/\text{s}}{1.40\text{ m}^3/\text{s}} = 0.12\right]$$

Figure 62

Next, we need to calculate the condition of the air entering the cooling coil. This air is a mixture of return air (RA) from the space and outdoor air (OA). In Period Two, we determined that 360 cfm [0.18 m$^3$/s] of outdoor air is required to properly ventilate this space.

The percentage of the total supply airflow that is made up of outdoor air is determined as follows:

$$\%\text{ outdoor air (OA)} = \frac{\text{ventilation airflow}}{\text{total supply airflow}}$$

$$\%\text{ outdoor air (OA)} = \frac{360\text{ cfm}}{2{,}990\text{ cfm}}\ \left[\frac{0.18\text{ m}^3/\text{s}}{1.40\text{ m}^3/\text{s}}\right] = 0.12$$

This indicates that 12% of the air entering the cooling coil is outdoor air and 88% is air being recirculated from the space.

### Calculate Entering Coil Conditions

95°F × 0.12 = 11.4°F
78°F × 0.88 = 68.6°F
mixture = 80.0°F

[35°C × 0.12 = 4.2°C
25.6°C × 0.88 = 22.5°C
mixture = 26.7°C]

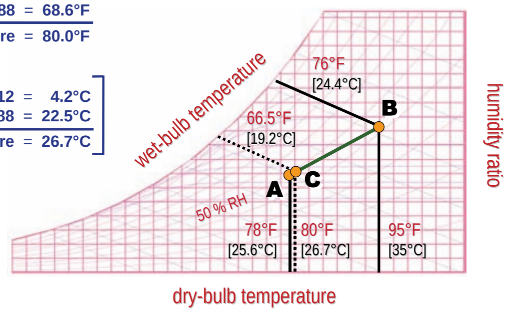


Figure 63

Assuming that the air being recirculated from the space is the same condition as the space, we can determine the condition of the air entering the cooling coil. First, the dry-bulb temperature of this air mixture is determined as follows:

- Outdoor air conditions: 95°F [35°C] dry bulb, 76°F [24.4°C] wet bulb
- Recirculated air conditions: 78°F [25.6°C] dry bulb, 50% relative humidity

(95°F × 0.12) + (78°F × 0.88) = 80°F

[(35°C × 0.12) + (25.6°C × 0.88) = 26.7°C]

Using the psychrometric chart, the condition of this air mixture (**C**) must fall on a line connecting the condition of the recirculated air (**A**) and the condition of the outdoor air (**B**). The wet-bulb temperature that marks the intersection of the connecting line and the 80°F [26.7°C] dry-bulb temperature mark is approximately 66.5°F [19.2°C]. Because the recirculated air constitutes a larger percentage (88%) of the mixture, the mixed-air condition (**C**) is much closer to the recirculated air condition (**A**) than the outdoor design condition (**B**).

## Determine Supply Air Temperature

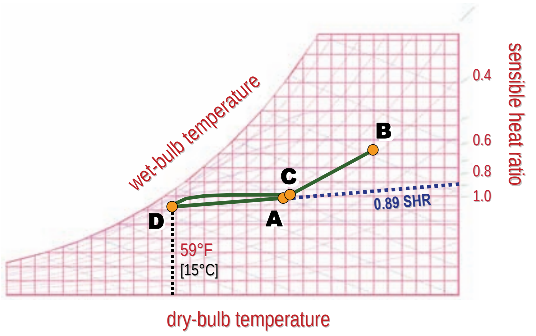
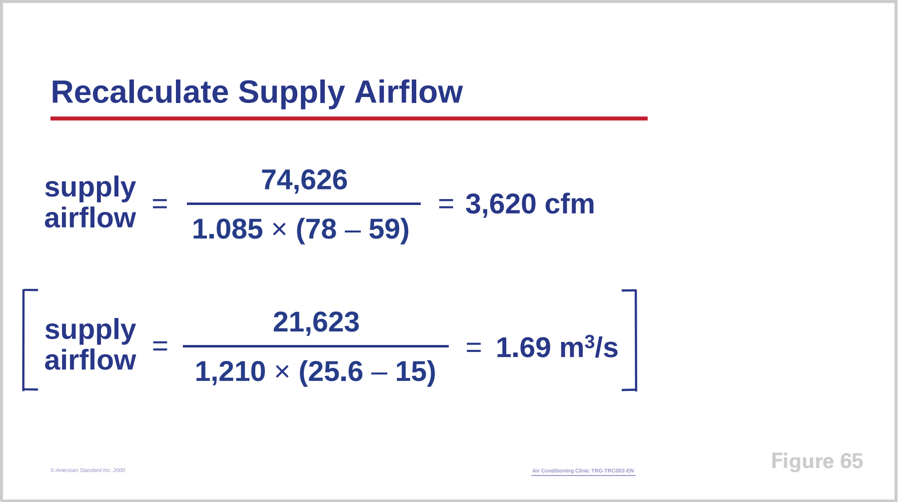


Figure 64

The next step is to determine the supply-air condition (dry-bulb and wet-bulb temperatures) necessary to absorb the sensible and latent heat in the space.

A sensible-heat-ratio line is drawn by connecting the 0.89 value on the SHR scale with the index point. Since the index point is the same as the desired space condition for this example (**A**), this line is extended until it intersects the saturation curve. If the desired space condition was different, a line would be drawn parallel to the 0.89 SHR line through the space condition.

Using the curvature of the nearest two coil curves as a guide, draw a curve from the mixed-air condition (**C**) until it intersects the SHR line. This point of intersection (**D**) represents the supply-air condition that will offset the space sensible and latent heat gains in the correct proportions required to maintain the desired space condition. Here, this supply-air condition is 59°F dry bulb, 57.4°F wet bulb [15°C dry bulb, 14.1°C wet bulb].



**Recalculate Supply Airflow**

$$\text{supply airflow} = \frac{74{,}626}{1.085 \times (78 - 59)} = 3{,}620 \text{ cfm}$$

$$\left[\text{supply airflow} = \frac{21{,}623}{1{,}210 \times (25.6 - 15)} = 1.69 \text{ m}^3\text{/s}\right]$$

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Air Conditioning Clinic TRG-TRC002-EN

Figure 65

Because this resulting supply air temperature is different than the 55ºF [12.8ºC] that we assumed, we need to recalculate the supply airflow.

$$\text{supply airflow} = \frac{74{,}626 \text{ Btu/hr}}{1.085 \times (78°\text{F} - 59°\text{F})} = 3{,}620 \text{ cfm}$$

$$\left[\text{supply airflow} = \frac{21{,}623 \text{ W}}{1{,}210 \times (25.6°\text{C} - 15°\text{C})} = 1.69 \text{ m}^3\text{/s}\right]$$

To complete another iteration of this analysis, we would 1) calculate the new percentage of outdoor air (10%), 2) determine the new entering coil conditions (79.7ºF dry bulb, 66.2ºF wet bulb [26.5ºC dry bulb, 19ºC wet bulb]), and 3) draw a curve from this new mixed-air condition until it intersects the 0.89 SHR line. This second iteration will likely result in the same supply air condition as the first iteration.

This simple psychrometric analysis indicates that our example space requires 3,620 cfm of air at 59°F dry bulb, 57.4°F wet bulb [1.69 $\text{m}^3$/s of air at 15°C dry bulb, 14.1°C wet bulb] to offset the sensible and latent heat gains for the space in the correct proportions.

For more information on the process of analyzing an HVAC system on the psychrometric chart, refer to the *Psychrometry* Air Conditioning Clinic.

notes

## Total Cooling Load on Coil

| | Room 101<br>Btu/hr [W] |
|---|---|
| total space sensible load | 74,626 [21,623] |
| total space latent load | 9,109 [2,599] |
| ventilation | 15,460 [4,756] |
| **total coil cooling load** | **99,195 [28,978]** |

Figure 66

Finally, the cooling coil in the air-conditioning system that serves this space must be capable of handling the sensible and latent loads for the space, plus any additional loads that affect the coil only. In our example, outdoor air for ventilation is the only additional load affecting the cooling coil. Based on the calculations performed in Period Two, the total load on the cooling coil is 99,195 Btu/hr (8.3 refrigeration tons) [28,978 W].

## Multiple-Space Analysis

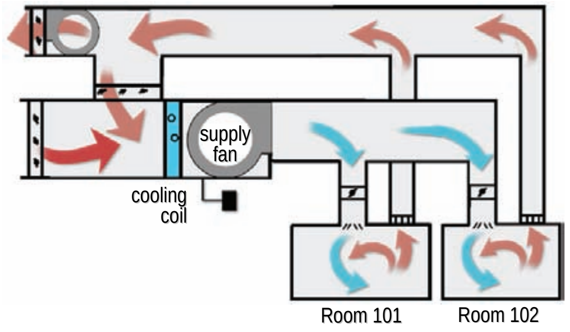


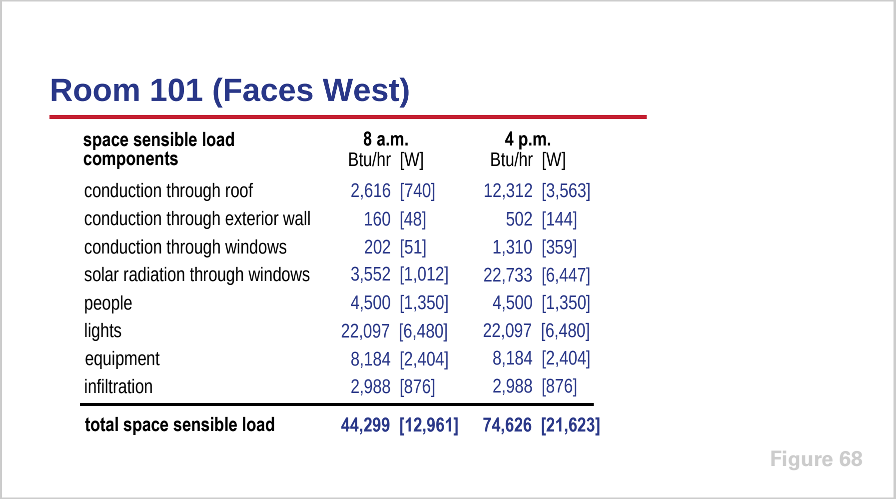
Figure 67

### Multiple-Space Analysis

So far in this clinic, we have calculated the cooling load for a single space and learned how to size the fan and cooling coil for a system serving that space. Referring back to the example office space in Figure 15, now we will consider an example where rooms 101 (west-facing wall) and 102 (east-facing wall) are served by the same air-conditioning system.

As shown in the previous section, if each space is conditioned by a separate system, then the fan and coil would be sized to handle the maximum load for the particular space. If, however, a single system is used to condition several spaces in a building, the method used to size the fan and coil depends on whether the system is a constant-volume (CV) or variable-air-volume (VAV) system. The example system shown here has two spaces. If the supply fan delivers a constant volume of air, the fan must be sized by summing the peak sensible loads for each of the spaces it serves. If, however, it is a VAV system and the fan delivers a varying amount of air to the system, the fan is sized based on the one-time, worst-case airflow requirement of all of the spaces it serves.

The next example will help explain this important difference.



## Room 101 (Faces West)

| **space sensible load components** | **8 a.m.** |  | **4 p.m.** |  |
|---|---|---|---|---|
|  | Btu/hr | [W] | Btu/hr | [W] |
| conduction through roof | 2,616 | [740] | 12,312 | [3,563] |
| conduction through exterior wall | 160 | [48] | 502 | [144] |
| conduction through windows | 202 | [51] | 1,310 | [359] |
| solar radiation through windows | 3,552 | [1,012] | 22,733 | [6,447] |
| people | 4,500 | [1,350] | 4,500 | [1,350] |
| lights | 22,097 | [6,480] | 22,097 | [6,480] |
| equipment | 8,184 | [2,404] | 8,184 | [2,404] |
| infiltration | 2,988 | [876] | 2,988 | [876] |
| **total space sensible load** | **44,299** | **[12,961]** | **74,626** | **[21,623]** |

Figure 68

Room 101 is the same space that we have used in the two previous periods of this clinic. In Period Two, we calculated the components of the space cooling load at 4 p.m. This table shows the components of the space sensible load at both 8 a.m. and 4 p.m. (The cooling loads at 8 a.m. were calculated using the same methods introduced in Period Two.)

Because room 101 has several west-facing windows, the peak (highest) space sensible load occurs in the late afternoon when the sun is shining directly through the windows.

## Room 102 (Faces East)

| **space sensible load components** | **8 a.m.**<br>Btu/hr [W] | **4 p.m.**<br>Btu/hr [W] |
|---|---|---|
| conduction through roof | 2,616 [740] | 12,312 [3,563] |
| conduction through exterior wall | 160 [48] | 844 [252] |
| conduction through windows | 202 [51] | 1,310 [359] |
| solar radiation through windows | 21,667 [6,138] | 3,078 [874] |
| people | 4,500 [1,350] | 4,500 [1,350] |
| lights | 22,097 [6,480] | 22,097 [6,480] |
| equipment | 8,184 [2,404] | 8,184 [2,404] |
| infiltration | 2,988 [876] | 2,988 [876] |
| **total space sensible load** | **62,414 [18,087]** | **55,313 [16,158]** |

Figure 69

Room 102 is exactly the same as room 101, except that the exterior wall faces east instead of west. Because of this difference, these two rooms will not only have different cooling loads, but the peak space cooling load for room 102 will occur at a different time of day than for room 101.

Because room 102 has several east-facing windows, the peak space cooling load occurs in the morning when the rising sun shines directly through the windows.

## “Sum-of-Peaks” versus “Block”

| **space sensible load** | **8 a.m.**<br>Btu/hr [W] | **4 p.m.**<br>Btu/hr [W] |
|---|---|---|
| Room 101 (faces west) | 44,299 [12,961] | **74,626 [21,623]** |
| Room 102 (faces east) | **62,414 [18,087]** | 55,313 [16,158] |

**sum-of-peaks** = **74,626** + **62,414** = **137,040 Btu/hr**
[21,623 + 18,087 = 39,710 W]

**block** = **74,626** + **55,313** = **129,939 Btu/hr**
[21,623 + 16,158 = 37,781 W]

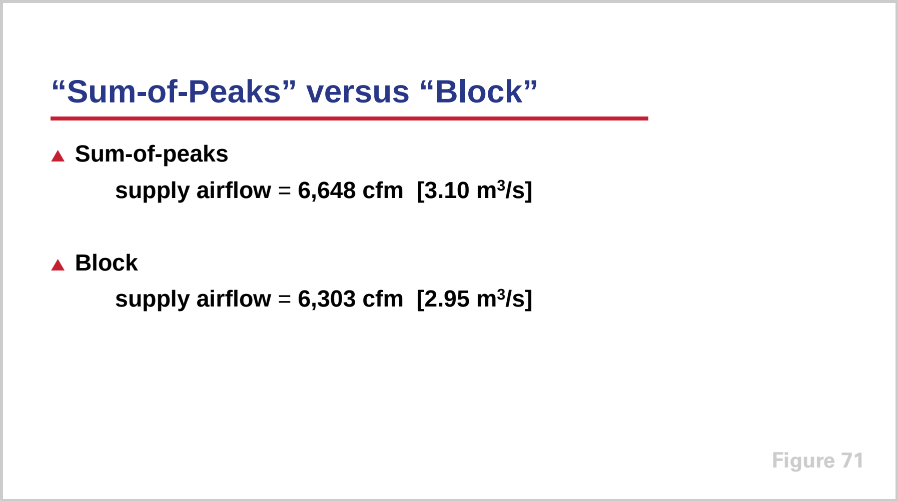
Figure 70

As seen in this example, the peak space loads do not necessarily occur at the same time for all spaces served by the same system. As discussed earlier in this period, the supply airflow for a space is calculated based on the maximum sensible load for that space—74,626 Btu/hr [21,623 W] for room 101 and 62,414 Btu/hr [18,087 W] for room 102. If the supply fan in this example delivers a constant volume of air at all times, it must be sized by summing the peak

notes

sensible loads of both spaces—137,040 Btu/hr [39,710 W]. This is called the **sum-of-peaks** load.

Although rooms 101 and 102 peak at different times of the day, there will be a single instance in time when the *sum* of these two space loads is highest. This is called the **block** load. If these two spaces are served by a single VAV system, in which the supply fan delivers a varying amount of air to the system, the fan only needs to be sized for the time when the sum of the space sensible loads is the highest—129,939 Btu/hr [37,781 W].



## “Sum-of-Peaks” versus “Block”

- **Sum-of-peaks**
  **supply airflow = 6,648 cfm [3.10 m$^3$/s]**
- **Block**
  **supply airflow = 6,303 cfm [2.95 m$^3$/s]**

Figure 71

Assume that the supply air dry bulb is the 59°F [15°C] that was calculated during the psychrometric analysis. The sum-of-peaks and block airflows for sizing the supply fan in these two cases can then be calculated as follows:

$$\text{sum-of-peaks supply airflow} = \frac{137{,}040\ \text{Btu/hr}}{1.085 \times (78°\text{F} - 59°\text{F})} = 6{,}648\ \text{cfm}$$

$$\left[\text{sum-of-peaks supply airflow} = \frac{39{,}710\ \text{W}}{1{,}210 \times (25.6°\text{C} - 15°\text{C})} = 3.10\ \text{m}^3/\text{s}\right]$$

$$\text{block supply airflow} = \frac{129{,}939\ \text{Btu/hr}}{1.085 \times (78°\text{F} - 59°\text{F})} = 6{,}303\ \text{cfm}$$

$$\left[\text{block supply airflow} = \frac{37{,}781\text{W}}{1{,}210 \times (25.6°\text{C} - 15°\text{C})} = 2.95\ \text{m}^3/\text{s}\right]$$

This is the reason that VAV systems can use smaller supply fans than constant-volume systems.

notes

## “Block” Cooling Load

| loads at 4 p.m. | Room 101 Btu/hr [W] | Room 102 Btu/hr [W] |
|---|---|---|
| total space sensible load | 74,626 [21,623] | 55,313 [16,158] |
| total space latent load | 9,109 [2,599] | 9,109 [2,599] |
| ventilation | 15,460 [4,756] | 15,460 [4,756] |
| **total coil cooling load** | **99,195 [28,978]** | **79,882 [23,513]** |

**block cooling load (4 p.m.)** = **99,195** + **79,882** = **179,077 Btu/hr**
[28,978 + 23,513 = 52,491 W]

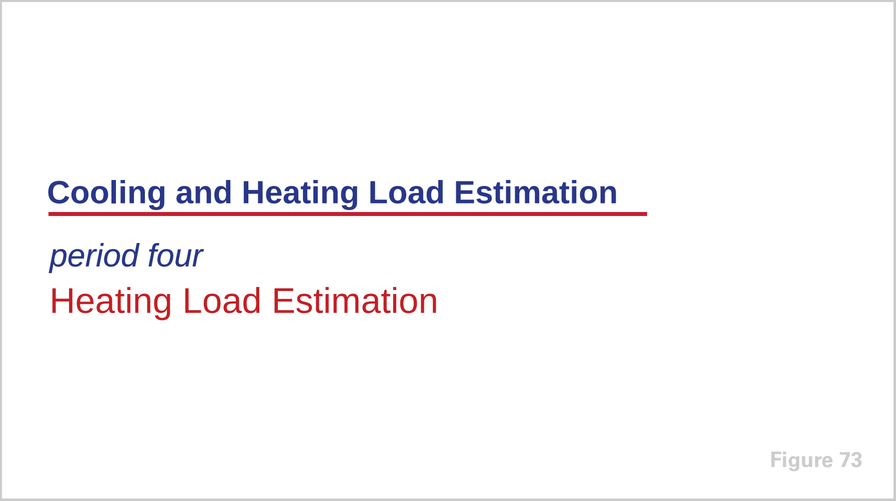
Figure 72

Similar to the single-space example, the cooling coil in the HVAC system serving these two spaces must be capable of handling the space sensible and latent loads, plus any additional loads that only affect the coil. Again, in our example, outdoor air for ventilation is the only additional load affecting the coil.

In general, a cooling coil in a system serving multiple spaces is sized based on the block cooling load. In our example, this block load occurs at 4 p.m., the time when the sum of the space loads for rooms 101 and 102, plus the ventilation load, is the highest. Based on the calculations performed in Period Two and earlier in this period, the block load for sizing the cooling coil in this multiple-space system is 179,077 Btu/hr (14.9 refrigeration tons) [52,491 W].

# period four
# Heating Load Estimation

notes



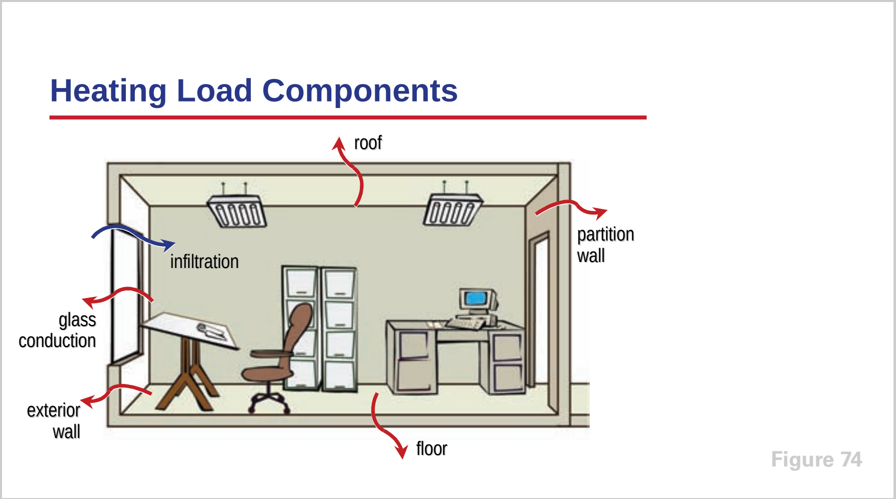
## Cooling and Heating Load Estimation

*period four*

Heating Load Estimation

Figure 73

The space heating load is the rate at which heat must be added to a space in order to maintain the desired conditions in the space, generally a dry-bulb temperature. During this period, we will estimate the heating load for the same office space that was used for the example in Period Two.



## Heating Load Components

Figure 74

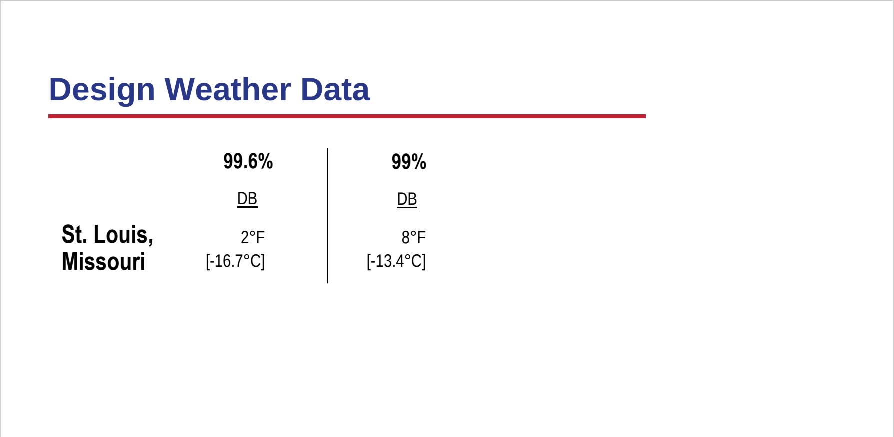
In general, the estimation of heating loads assumes worst-case conditions for the space. The winter design outdoor temperature is used for determining the conduction heat loss through exterior surfaces. No credit is given for heat gain from solar radiation through glass or from the sun's rays warming the outside surfaces of the building. Additionally, no credit is given for internal heat gains due to people, lighting, and equipment in the space.

The heating load for a space can be made up of many components, including:

- Conduction heat loss to the outdoors through the roof, exterior walls, skylights, and windows

- Conduction heat loss to adjoining spaces through the ceiling, interior partition walls, and floor
- Heat loss due to cold air infiltrating into the space from outdoors through doors, windows, and small cracks in the building envelope

In addition, the heating coil in the building HVAC system has to heat up the outdoor air that is deliberately brought into the building for ventilation purposes.



**Design Weather Data**

| | 99.6% | 99% |
|---|---|---|
| | DB | DB |
| St. Louis, Missouri | 2°F [-16.7°C] | 8°F [-13.4°C] |

Figure 75

### Outdoor Design Conditions

Similar to the cooling-design outdoor conditions discussed at the beginning of Period Two, heating-design outdoor conditions for many locations can be found in the *1997 ASHRAE Handbook—Fundamentals*.

For our example building located in St. Louis, Missouri, the heating-design outdoor conditions include two columns of dry-bulb temperatures. The first column heading, 99.6%, means that the dry-bulb temperature in St. Louis exceeds 2°F [-16.7°C] for 99.6% of all the hours in an average year. In other words, the outdoor temperature is colder than 2°F [-16.7°C] for only 0.4% of all hours (or 35 hours) in an average year. The second column heading, 99%, means that the dry-bulb temperature exceeds 8°F [-13.4°C] for 99% of all hours in an average year.

Again, these two columns allow the HVAC system designer to choose how conservative that they wish to be. For our example, we will use the more severe 2°F [-16.7°C] dry-bulb temperature for the outdoor design conditions.

### Conduction Heat Loss

$$Q = U \times A \times \Delta T$$

Figure 76

## Conduction through Surfaces

In a matter similar to Period Two, we will focus on the most common conduction heat losses from a space: through the roof, the exterior walls, and the windows.

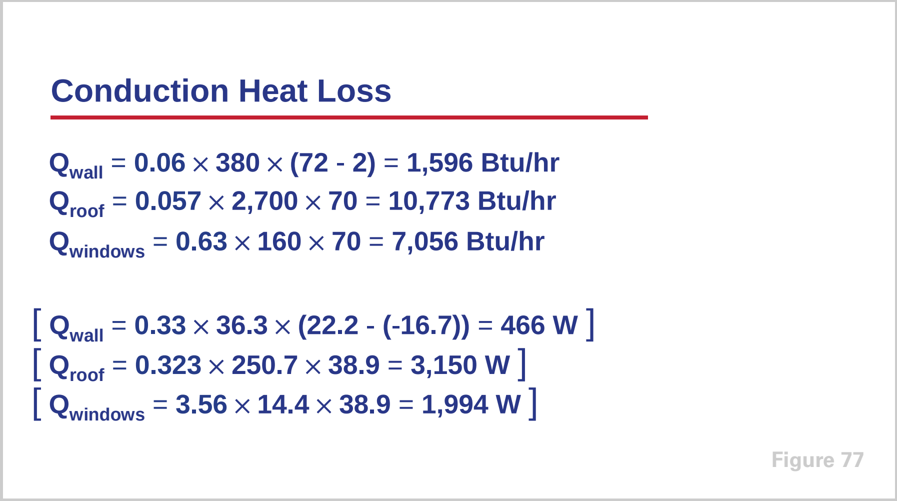
When calculating heating loss by conduction through the roof, the exterior walls, and the windows, no credit is given for the effect of the sun shining on the outside surfaces. With this assumption, the amount of heat transferred through the surface is a direct result of the temperature difference between the outdoor and indoor surfaces ($\Delta T$ is used instead of CLTD).

The amount of heat loss through a roof, an exterior wall, or a window depends on the area of the surface, the overall heat transfer coefficient of the surface, and the dry-bulb temperature difference from one side of the surface to the other. The equation used to predict the heat loss by conduction is:

$$Q = U \times A \times \Delta T$$

where,

- $Q$ = heat loss by conduction, Btu/hr [W]
- $U$ = overall heat-transfer coefficient of the surface, Btu/hr • $ft^2$ • °F [$W/m^2$ • °K]
- $A$ = area of the surface, $ft^2$ [$m^2$]
- $\Delta T$ = desired indoor dry-bulb temperature ($T_i$) minus the design outdoor dry-bulb temperature ($T_o$), °F [°C]



**Conduction Heat Loss**

$Q_{wall} = 0.06 \times 380 \times (72 - 2) = 1{,}596$ Btu/hr
$Q_{roof} = 0.057 \times 2{,}700 \times 70 = 10{,}773$ Btu/hr
$Q_{windows} = 0.63 \times 160 \times 70 = 7{,}056$ Btu/hr

[ $Q_{wall} = 0.33 \times 36.3 \times (22.2 - (-16.7)) = 466$ W ]
[ $Q_{roof} = 0.323 \times 250.7 \times 38.9 = 3{,}150$ W ]
[ $Q_{windows} = 3.56 \times 14.4 \times 38.9 = 1{,}994$ W ]

Figure 77

The U-factors for the roof, wall, and windows will be slightly different during the winter than during the summer. This is due to a change in the outside surface resistance (R = 0.17 versus 0.25 [R = 0.03 versus 0.044]) that is caused by a difference in wind conditions from summer to winter. For simplicity, we will ignore this minor difference and use the U-factors calculated in Period Two for the roof, wall, and windows of our example space. Also, realize that in our example the desired indoor dry-bulb temperature during the heating season is 72°F [22.2°C], different than during the cooling season.

Conduction heat loss through the west-facing wall:

- U-factor for wall = 0.06 Btu/hr • $ft^2$ • °F [0.33 W/$m^2$ • °K]
- Net area of wall = 380 $ft^2$ [36.3 $m^2$]
- $\Delta T$ = indoor temperature (72°F [22.2°C]) – outdoor temperature (2°F [16.7°C])

$Q = 0.06 \times 380 \times (72 - 2) = 1{,}596$ Btu/hr

[$Q = 0.33 \times 36.3 \times (22.2 - (-16.7)) = 466$ W]

Conduction heat loss through the roof:

- U-factor for roof = 0.057 Btu/hr • $ft^2$ • °F [0.323 W/$m^2$ • °K]
- Area of roof = 2,700 $ft^2$ [250.7 $m^2$]
- $\Delta T$ = 70°F [38.9°C]

$Q = 0.057 \times 2{,}700 \times 70 = 10{,}773$ Btu/hr

[$Q = 0.323 \times 250.7 \times 38.9 = 3{,}150$ W]

Conduction heat loss through the west-facing windows:

- U-factor for window = 0.63 Btu/hr • $ft^2$ • °F [3.56 $W/m^2$ • °K]
- Total area of glass = 160 $ft^2$ [14.4 $m^2$]
- $\Delta T$ = 70°F [38.9°C]

Q = 0.63 × 160 × 70 = 7,056 Btu/hr

[Q = 3.56 × 14.4 × 38.9 = 1,994 W]

## Infiltration and Ventilation

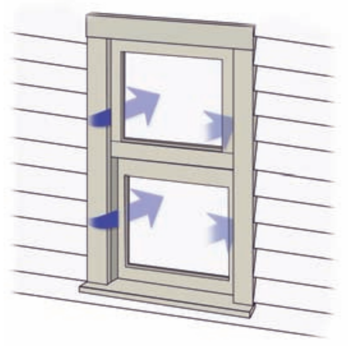


Figure 78

### Infiltration and Ventilation

Similar to the cooling season, during the heating season, air may leak in from outdoors through doors, windows, and small cracks in the building envelope. It contributes to the sensible heat loss of the space because the outdoor air is typically colder than the indoor air.

Additionally, the outdoor air during the heating season is generally drier than the indoor air. If the building requires humidification, infiltration of cold, dry outdoor air adds to the humidification load. This clinic, however, will only focus on the sensible heat loss due to infiltration and its effect on sizing the sensible heating equipment in the HVAC system.

## Infiltration and Ventilation

$$Q_{\text{sensible}} = 1.085 \times \text{airflow} \times \Delta T$$

$$[\; Q_{\text{sensible}} = 1{,}210 \times \text{airflow} \times \Delta T \;]$$

Figure 79

The equation used to calculate the sensible heat loads due to infiltration and ventilation is the same as shown in Period Two:

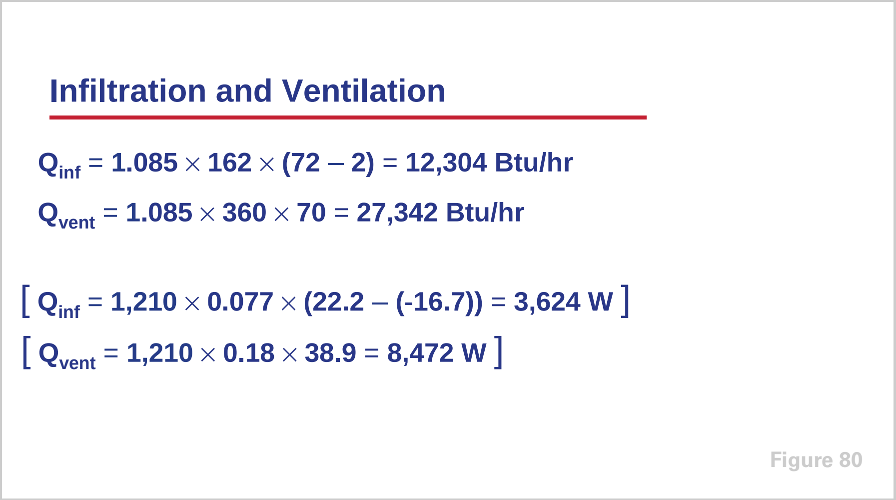
$Q_S = 1.085 \times \text{airflow} \times \Delta T$

$[Q_S = 1{,}210 \times \text{airflow} \times \Delta T]$

where,

- $Q_S$ = sensible heat load due to infiltration or ventilation, Btu/hr [W]
- 1.085 [1,210] = product of density and specific heat, Btu • min/hr • $ft^3$ • °F [$J/m^3$ • °K]
- Airflow = infiltration or ventilation airflow, cfm [$m^3/s$]
- $\Delta T$ = desired indoor dry-bulb temperature minus the design outdoor dry-bulb temperature, °F [°C]

notes



## Infiltration and Ventilation

$Q_{inf} = 1.085 \times 162 \times (72 - 2) = 12{,}304 \text{ Btu/hr}$

$Q_{vent} = 1.085 \times 360 \times 70 = 27{,}342 \text{ Btu/hr}$

$[\ Q_{inf} = 1{,}210 \times 0.077 \times (22.2 - (-16.7)) = 3{,}624 \text{ W}\ ]$

$[\ Q_{vent} = 1{,}210 \times 0.18 \times 38.9 = 8{,}472 \text{ W}\ ]$

Figure 80

Sensible heat loads due to infiltration and the conditioning of ventilation air:

- Infiltration airflow = 162 cfm [0.077 $m^3/s$]
- Ventilation airflow = 360 cfm [0.18 $m^3/s$]
- Outdoor dry-bulb temperature: 2°F [-16.7°C]
- Indoor dry-bulb temperature: 72°F [22.2°C])

Infiltration:

$Q_S = 1.085 \times 162 \times (72 - 2) = 12{,}304$ Btu/hr

[$Q_S = 1{,}210 \times 0.077 \times (22.2 - (-16.7)) = 3{,}624$ W]

Ventilation:

$Q_S = 1.085 \times 360 \times 70 = 27{,}342$ Btu/hr

[$Q_S = 1{,}210 \times 0.18 \times 38.9 = 8{,}472$ W]

## Summary of Heating Loads

| | sensible load Btu/hr | [W] |
|---|---|---|
| conduction through roof | 10,773 | [3,150] |
| conduction through exterior wall | 1,596 | [466] |
| conduction through windows | 7,056 | [1,994] |
| infiltration | 12,304 | [3,624] |
| **total space heating load** | **31,729** | **[9,234]** |
| ventilation | 27,342 | [8,472] |
| **total coil heating load** | **59,071** | **[17,706]** |

Figure 81

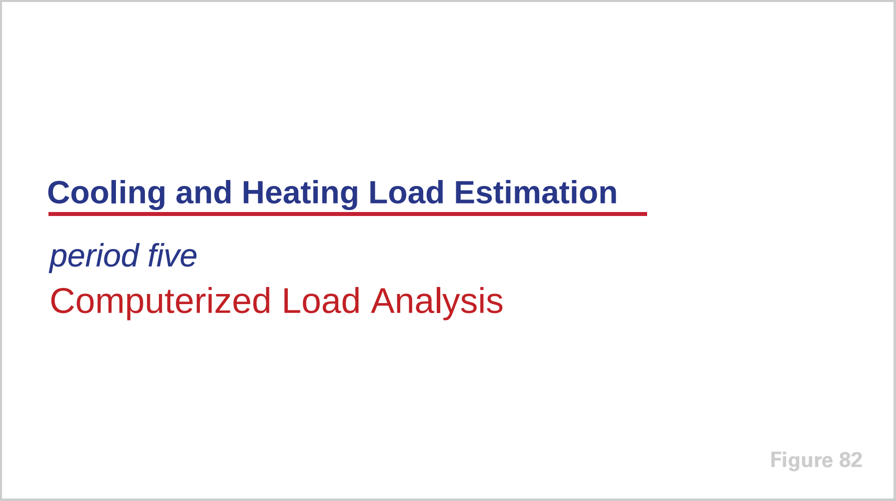
Again, the estimation of heating loads assumes worst-case conditions for the space. No credit is given for solar effects or internal heat gains. The total heating load for our example space is made up of the following components:

- Conduction heat loss through the roof and the west-facing exterior wall and windows
- Cold air infiltrating into the space from outdoors

In addition, the heating coil in the building HVAC system must warm the outdoor air that is deliberately brought into the building for ventilation purposes.

This total heating load, 59,071 Btu/hr [17,706 W], is used to size the heating coils in the HVAC system. For buildings in cold climates, many system designers choose to apply an additional safety factor to the estimated heating load. This is not the case with the cooling load. The reason for this additional concern is that the danger of undersizing the heating system may lead to frozen and/or broken water pipes that can cause extensive damage to the building.

# period five
# Computerized Load Analysis



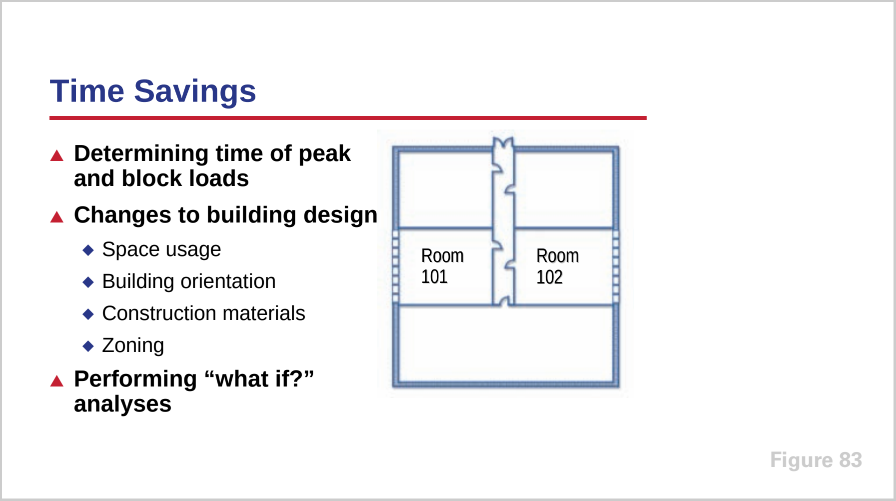
## Cooling and Heating Load Estimation

*period five*

Computerized Load Analysis

Figure 82

There are many commercially-available computer programs that make cooling and heating load calculations faster and easier. This period will briefly introduce some of the benefits of using computer programs to perform a cooling-and-heating-load analysis.



## Time Savings

- **Determining time of peak and block loads**
- **Changes to building design**
  - Space usage
  - Building orientation
  - Construction materials
  - Zoning
- **Performing "what if?" analyses**

Figure 83

### Time Savings

First of all, there are many time-saving benefits of using a computer program to perform load calculations. Hand calculations can be tedious, and numerous iterations or recalculations are much easier when a computer program is used.

As mentioned at the beginning of this clinic, often one of the more difficult aspects of determining the maximum cooling load for a space and system is determining the time at which these maximum loads occur. Predicting the time that the highest space load will occur can be perfected with experience, but calculating the block load for a multiple-space system often requires numerous calculations, for multiple spaces, at multiple points in time.

Additionally, changes that occur to the building design midway through the design process may require the HVAC system designer to recalculate loads. Changes such as a different usage for a space, new construction materials, changing the orientation of the building, or zoning similar spaces together, are common causes for recalculation.

Finally, computer programs make it easier to perform trade-off, or “what if?”, analyses. Examples may include determining the effects of using better windows, installing shading devices, or adding more insulation.

## “Sum-of-Peaks” versus “Block”

| **space sensible load** | **8 a.m.**<br>Btu/hr [W] | **4 p.m.**<br>Btu/hr [W] |
|---|---|---|
| Room 101 (faces west) | 44,299 [12,961] | **74,626 [21,623]** |
| Room 102 (faces east) | **62,414 [18,087]** | 55,313 [16,158] |

**sum-of-peaks** = **74,626** + **62,414** = **137,040 Btu/hr**
[21,623 + 18,087 = 39,710 W]

**block** = **74,626** + **55,313** = **129,939 Btu/hr**
[21,623 + 16,158 = 37,781 W]

Figure 84

As shown during our multiple-space example in Period Three, in order to determine the design airflow for the supply fan, we had to calculate the sensible loads for both spaces at two different times. When done by hand, this required four separate sets of calculations. A computer program would require entering the parameters for these spaces only once. Additionally, it could easily perform these calculations for every hour to find the block load for the system.

Space versus Plenum Loads

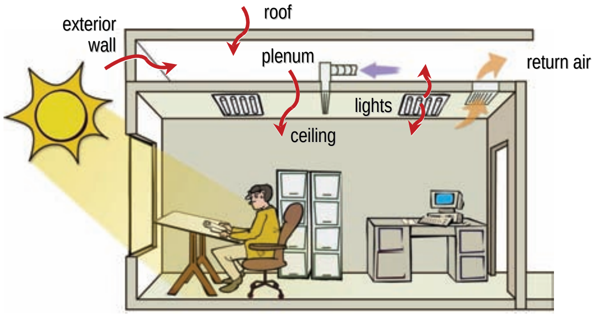


Figure 85

### Space versus Plenum Loads

In the previous examples used throughout this clinic, we have assumed that the spaces do not have a ceiling plenum. The **plenum** is the space between the ceiling and the roof. When a building has a plenum, all heat gain through the roof affects the plenum instead of the occupied space. Additionally, because a portion of the exterior wall is adjacent to the plenum, the heat gain through that part of the wall affects the plenum. Finally, in many types of lighting systems, a portion of the heat generated by the lights is transferred directly into the plenum instead of into the space.

The plenum is not air conditioned, but in this example the return air from the space passes through the plenum on the way back to the air handler. Some of the heat gain to the plenum is absorbed by this return air, increasing the temperature of the air returning to the air handler. The heat gain in the plenum, however, also creates a temperature difference between the plenum and the space, which results in some of this heat being transferred by conduction, through the ceiling and into the space.

# Computerized Load Analysis

notes

### Space versus Plenum Loads

| **sensible load components** | **space**<br>Btu/hr [W] | **plenum**<br>Btu/hr [W] |
|---|---|---|
| conduction through roof | | 8,640 [2,357] |
| conduction through exterior wall | 418 [120] | 84 [24] |
| conduction through windows | 1,310 [359] | |
| solar radiation through windows | 22,733 [6,447] | |
| people | 4,500 [1,350] | |
| lights | 15,468 [4,536] | 6,629 [1,944] |
| equipment | 8,184 [2,404] | |
| infiltration | 2,988 [876] | |
| heat transfer through ceiling | 3,620 [1,020] | -3,620 [-1,020] |
| **total space sensible load** | **59,221 [17,112]** | |

Figure 86

Using the example space from Period Two, the conduction heat gain through the roof, the conduction heat gain through the top section of the exterior wall, and 30% (assumed) of the heat generated by the lights is transferred to the plenum instead of to the space. While some of this heat is absorbed by the return air, 3,620 Btu/hr [1,020 W] is transferred through the ceiling into the space. This heat transfer between the plenum and the space does affect the space sensible load, but it does not affect the coil load. This is because any heat that is not transferred to the space is absorbed by the return air and must be eventually removed by the cooling coil.

The affect of the plenum reduces the total space sensible load in our example from 74,626 Btu/hr [21,623 W] without the plenum to 59,221 Btu/hr [17,112 W] with the plenum. This results in 21% less airflow required to condition this space.

A heat balance can be performed to determine how much heat that originally entered the plenum actually enters the space. This heat balance is an iterative process, because the conduction heat transfer between the plenum and the space depends on the temperature of the air in the plenum and the quantity of return air passing through the plenum. A computer program can perform these calculations much faster than they could be done by hand, due to the number of iterations required to achieve accurate results.

notes

## Ducted Return

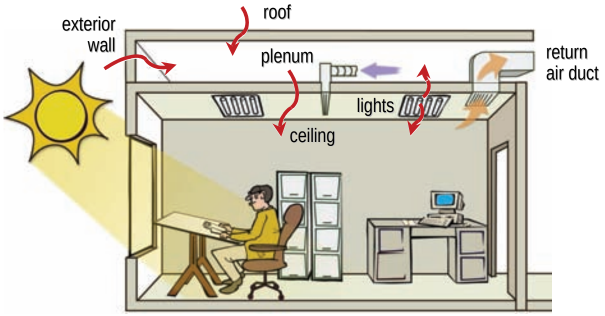


Figure 87

This example space also includes a plenum, but the return air is ducted instead of traveling through the open plenum. In this type of arrangement, a greater percentage of the heat gain to the plenum will be transferred to the space, depending on the location of the return air duct and whether or not it is insulated. If the return air duct is insulated, nearly all of the heat gain to the plenum will be transferred to the space.

## Advanced Calculation Methods

Figure 88

### Advanced Calculation Methods

The CLTD/SCL/CLF method used throughout this clinic is a simplified hand calculation procedure developed by ASHRAE. The tables used in the CLTD/SCL/CLF method were created by using the more advanced transfer-function method (TFM) to model a series of “typical” commercial spaces. Adjustments are required to correct for latitude, month, indoor and outdoor temperatures, and space construction. Because these tables were originally created for a fixed set of applications, ASHRAE recommends that designers use them with caution. In contrast, some of the more advanced computer-based calculation methods are able to more accurately model heat transfer in a wide variety of applications.

ASHRAE has conducted extensive research over the years to improve methods for estimating cooling and heating loads. Even though the CLTD/SCL/CLF method is the most common method used for basic instruction, designers are encouraged to investigate the benefits of more of these advanced techniques.

# period six
# Review

notes

## Cooling and Heating Load Estimation

*period six*

Review

Figure 89

We will now review the main concepts that were covered in this clinic on cooling and heating load estimation.

## Review—Period One

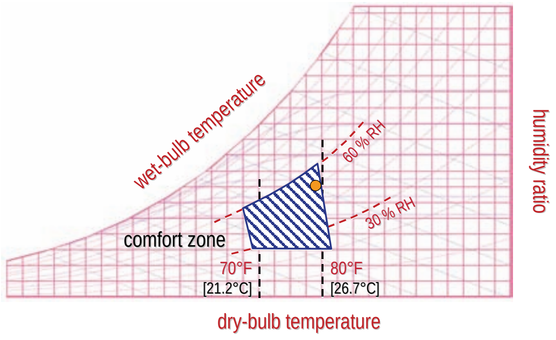


Figure 90

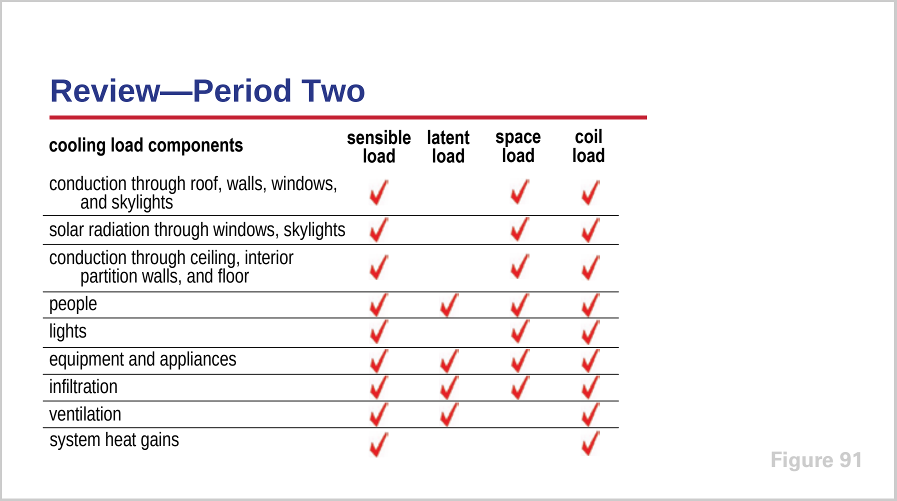
Period One discussed basic principles of heat transfer and human comfort. Although many factors contribute to making a space comfortable, this clinic focused only on the aspect of thermal comfort.

Thermal comfort depends upon creating an environment of dry-bulb temperature, humidity, and air motion that is appropriate for the activity level of the people in the space. This environment allows the body's rate of heat generation to balance with the body's rate of heat loss. ASHRAE has prescribed a "comfort zone" that can be used as the basis for HVAC system design.

period six

# Review

notes



## Review—Period Two

| cooling load components | sensible load | latent load | space load | coil load |
|---|---|---|---|---|
| conduction through roof, walls, windows, and skylights | ✓ | | ✓ | ✓ |
| solar radiation through windows, skylights | ✓ | | ✓ | ✓ |
| conduction through ceiling, interior partition walls, and floor | ✓ | | ✓ | ✓ |
| people | ✓ | ✓ | ✓ | ✓ |
| lights | ✓ | | ✓ | ✓ |
| equipment and appliances | ✓ | ✓ | ✓ | ✓ |
| infiltration | ✓ | ✓ | ✓ | ✓ |
| ventilation | ✓ | ✓ | | ✓ |
| system heat gains | ✓ | | | ✓ |

Figure 91

Period Two introduced the components of the building cooling load and the concepts used in estimating these loads using a single example space.

The sensible and latent components of the cooling load discussed in this period included:

- Conduction heat gain from outdoors through the roof, exterior walls, and windows
- Solar radiation heat gain through windows
- Conduction heat gain from adjoining spaces
- Internal heat gains from people, lights, appliances, and equipment in the space
- Heat gain from hot, humid air infiltrating into the space from outdoors
- Heat gain from hot, humid outdoor air deliberately brought into the building for ventilation purposes
- Heat generated by the fans in the system

Again, this clinic is not intended to teach all of the details or latest computerized techniques of how to calculate these loads. Instead, it introduces both the concepts of estimating building cooling and heating loads, as well as some simple methods for estimating these load components.

# period six
# Review

## Review—Period Three

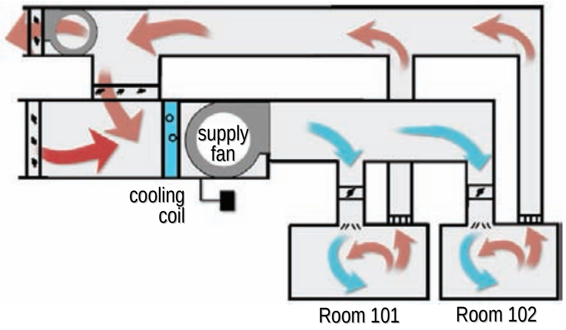


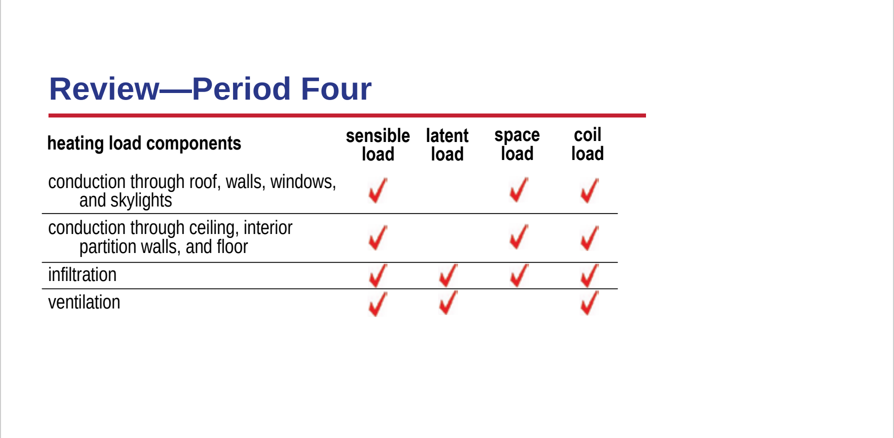
Figure 92

In Period Three, we conducted a simplified psychrometric analysis for this example space. During this analysis we calculated the space sensible heat ratio (SHR), determined the quantity and condition of the air necessary to cool the space, and calculated the required capacity of the cooling coil.

A second space was added to introduce the concept of “sum-of-peaks” versus “block” load. If the supply fan in a multiple-space system delivers a constant volume of air, it must be sized by summing the individual, peak sensible loads of the spaces it serves. This is called the “sum-of-peaks” load. If, however, the supply fan delivers a varying amount of air to the system, it only needs to be sized for the single instance when the sum of all space sensible loads is the highest. This is called the “block” load.

Finally, a cooling coil in a system that serves multiple spaces is generally sized based on the total block cooling load, including the space sensible and latent loads plus any additional loads that only affect the coil. In our example, outdoor air for ventilation was the only additional load affecting the coil.

# period six
# Review



## Review—Period Four

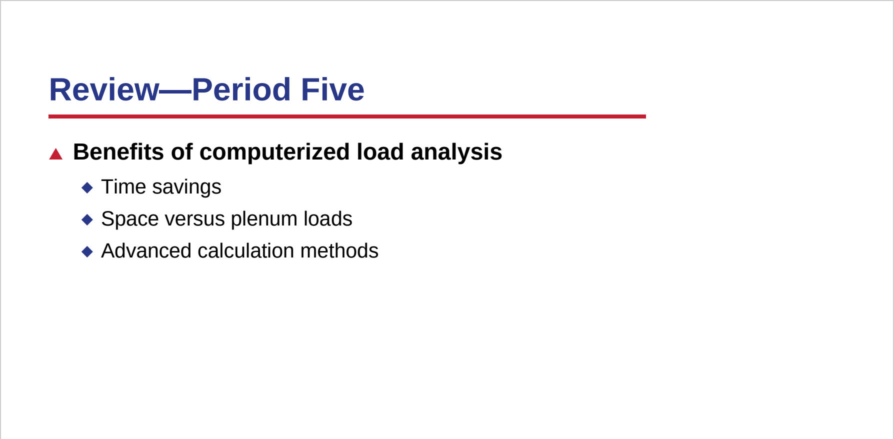
| heading load components | sensible load | latent load | space load | coil load |
|---|---|---|---|---|
| conduction through roof, walls, windows, and skylights | ✓ | | ✓ | ✓ |
| conduction through ceiling, interior partition walls, and floor | ✓ | | ✓ | ✓ |
| infiltration | ✓ | ✓ | ✓ | ✓ |
| ventilation | ✓ | ✓ | | ✓ |

Figure 93

Period Four introduced the components of the building heating load, and the concepts for estimating these loads, using the example space from Period Two.

The components of the heating load discussed in this period included:

- Conduction heat loss to outdoors through the roof, the exterior walls, and the windows
- Heat loss from cold air infiltrating into the space from outdoors
- Heating load due to cold outdoor air deliberately brought into the building for ventilation purposes



## Review—Period Five

- **Benefits of computerized load analysis**
  - Time savings
  - Space versus plenum loads
  - Advanced calculation methods

Figure 94

Many computerized load analysis programs are commercially available to make these calculations quick and simple, eliminating the need for tedious hand calculations and numerous iterations. Period Five briefly introduced some of period six

# Review

notes

the benefits of using a computer program to perform a cooling and heating load analysis.

These programs can save time when determining the time of peak space loads and block loads, recalculating loads when the building design changes, and performing trade-off and “what if?” analyses during building design. They can quickly perform a heat balance between space and plenum loads to more accurately determine the supply airflow required to condition the space. Finally, computers are able to perform some of the more advanced load calculation techniques that can result in improved accuracy.

Figure 95

If you are interested in learning more about the specific techniques used for cooling and heating load estimation, refer to the following references:

- *ASHRAE Handbook – Fundamentals*
- *Cooling and Heating Load Calculation Principles*, ASHRAE
- *Fundamentals of Heating and Cooling Loads*, ASHRAE self-directed learning course
- *Manual N – Commercial Load Calculation*, ACCA

*Psychrometry Air Conditioning Clinic* (Trane Literature order number TRG-TRC001-EN)

For information on Trane’s TRACE™ Load 700 or Load Express™ load-calculation software, visit the Trane C.D.S. Web site at www.tranecds.com.

Visit the ASHRAE Bookstore at www.ashrae.org.

Visit ACCA online at www.acca.org.

For information on additional educational materials available from Trane, contact your local Trane sales office (request a copy of the Educational Material price list—Trane order number EM-ADV1) or visit our online bookstore at www.trane.com/bookstore/.

# Quiz

## Questions for Period 1

**1** Heat always flows from a substance of ________ (higher, lower) temperature to a substance of ________ (higher, lower) temperature.

**2** Which of the three basic processes (conduction, convection, or radiation) is best described as the process of transferring heat due to air moving across the surface of the skin?

## Questions for Period 2

**3** Fill in the following table, indicating whether each component of the space cooling load contributes sensible heat and/or latent heat.

| | Sensible heat gain | Latent heat gain |
|---|---|---|
| conduction through roof | | |
| conduction through exterior walls | | |
| conduction through windows | | |
| solar radiation through windows or skylights | | |
| heat gain from people | | |
| heat gain from lights | | |
| air infiltrating from outdoors through cracks | | |

**4** A wall is made up of the following materials. Given the associated thermal resistance (R-value) for each material, calculate the U-factor for this wall.

| | Thermal resistance hr • ft$^2$ • °F/Btu | [m$^2$ • °K/W] |
|---|---|---|
| outdoor-air film resistance (summer) | 0.25 | [0.044] |
| 4 in. [100 mm] face brick | 0.80 | [0.141] |
| permeable-felt vapor membrane | 0.06 | [0.011] |
| 5.5 in. [140 mm] fiberglass insulation | 21 | [3.67] |
| 0.5 in. [12.7 mm] gypsum board | 0.45 | [0.079] |
| indoor-air film resistance | 0.68 | [0.12] |

# Quiz

**5** Given the following information, calculate the heat gain by conduction through an east-facing wall at Hour 12 during July:

- Wall Type 9
- Area of the wall = 100 $ft^2$ [9.3 $m^2$]
- U-factor of the wall = 0.05 Btu/hr • $ft^2$ • °F [0.284 $W/m^2$ • °K]
- Indoor dry-bulb temperature = 78°F [25.6°C]
- Outdoor dry-bulb temperature = 95°F [35°C]

**6** Given the following information, calculate the heat gain from solar radiation through a south-facing window at Hour 14 in July:

- Space Type A
- Area of the window = 30 $ft^2$ [2.8 $m^2$]
- U-factor of the window = 0.66 Btu/hr • $ft^2$ • °F [3.75 $W/m^2$ • °K]
- Shading coefficient of the window = 0.70
- Indoor dry-bulb temperature = 78°F [25.6°C]
- Outdoor dry-bulb temperature = 95°F [35°C]

**7** Given the following information, calculate both the sensible and latent heat gains due to infiltration of air from outdoors:

- Amount of infiltration = 0.5 air changes/hr
- Volume of space = 3,000 $ft^3$ [85 $m^3$]
- Indoor conditions = 75°F [23.9°C] dry-bulb temperature and 65 grains of water/lb dry air [9.3 grams of water/kg dry air]
- Outdoor conditions = 100°F [37.8°C] dry-bulb temperature and 92 grains of water/lb dry air [13.1 grams of water/kg dry air]

# Quiz

## Questions for Period 3

**8** Given the following components of the cooling load:

| | Sensible heat gain Btu/hr | [W] | Latent heat gain Btu/hr | [W] |
|---|---|---|---|---|
| conduction through roof | 7,000 | [2,050] | | |
| conduction through exterior walls | 500 | [150] | | |
| conduction through windows | 1,400 | [400] | | |
| solar radiation through windows or skylights | 12,200 | [3,600] | | |
| heat gain from people | 4,500 | [1,300] | 3,600 | [1,050] |
| heat gain from lights | 22,100 | [6,500] | | |
| heat gain from office equipment | 9,700 | [2,800] | | |
| air infiltrating through cracks from outdoors | 2,700 | [800] | 2,900 | [850] |
| cooling load due to ventilation brought in by the central HVAC system | 6,000 | [1,750] | 6,500 | [1,900] |
| heat gain from the supply fan | 3,400 | [1,000] | | |

**a** What is the sensible heat ratio for the space?

**b** Assuming the air is supplied to the space at 54°F [12.2°C] dry bulb and the space dry-bulb temperature is 75°F [23.9°C], what quantity of air must be supplied to the space?

**c** What is the total cooling load on the coil?

**9** The following sensible heat gains exist for three spaces being served by the same centralized HVAC system:

| | Total space sensible heat gain Btu/hr [W] | | | |
|---|---|---|---|---|
| | 9 a.m. | | 4 p.m. | |
| Room 201 | 70,000 | [20,500] | 20,000 | [6,000] |
| Room 202 | 10,000 | [3,000] | 85,000 | [25,000] |
| Room 203 | 15,000 | [4,500] | 20,000 | [6,000] |

**a** At what time does the block load occur? (Block load is used to size a variable-air-volume (VAV) supply fan)

**b** What is the block space sensible load?

**c** Suppose that you are sizing a constant-volume supply fan. What is the "sum-of-peaks" space sensible load?

# Quiz

## Questions for Period 4

**10** Which of the following components are ignored when estimating the heating load for a space?

- Exterior walls
- Solar radiation through windows
- Ventilation
- People
- Lights
- Infiltration

**11** Given the following information, calculate the heat loss due to conduction through a west-facing wall at Hour 4 during January:

- Area of the wall = 100 $ft^2$ [9.3 $m^2$]
- U-factor of the wall = 0.05 Btu/hr • $ft^2$ • °F [0.284 W/$m^2$ • °K]
- Indoor dry-bulb temperature = 70°F [21.1°C]
- Outdoor dry-bulb temperature = 0°F [-17.8°C]

## Questions for Period 5

**12** In a space with an open return air plenum, which of the following components would likely add to the heat gain in the plenum instead of, or in addition to, the space?

- Exterior walls
- Ventilation
- People
- Lights
- Roof

# Answers

**1** Higher to lower

**2** Convection

**3**

| | Sensible heat gain | Latent heat gain |
|---|---|---|
| conduction through roof | X | |
| conduction through external walls | X | |
| conduction through windows | X | |
| solar radiation through windows or skylights | X | |
| heat gain due to people | X | X |
| heat gain from lights | X | |
| air infiltrating through cracks from outdoors | X | X |

**4**

| | Thermal resistance hr • ft² • °F/Btu | [m² • °K/W] |
|---|---|---|
| outdoor-air film resistance (summer) | 0.25 | [0.044] |
| 4 in. [100 mm] face brick | 0.80 | [0.141] |
| Permeable-felt vapor membrane | 0.06 | [0.011] |
| 5.5 in. [140 mm] fiberglass insulation | 21 | [3.67] |
| 0.5 in. [12.7 mm] gypsum board | 0.45 | [0.079] |
| indoor-air film resistance | 0.68 | [0.12] |
| total resistance | 23.24 | [4.065] |

$$U = \frac{1}{\text{total resistance}} = \frac{1}{23.24} = 0.043 \text{ Btu/hr} \bullet \text{ft}^2 \bullet {}^\circ\text{F}$$

$$\left[U = \frac{1}{\text{total resistance}} = \frac{1}{4.065} = 0.246 \text{ W/m}^2 \bullet {}^\circ\text{K}\right]$$

# Answers

**5** Q = U × A × CLTD

CLTD (from Table 1 [Table 2] on page 21) = 26°F [14°C]

Q = 0.05 × 100 × 26 = 130 Btu/hr

[Q = 0.284 × 9.3 × 14 = 37 W]

**6** Q = A × SC × SCL

SCL (from Table 7 [Table 8] on page 28) = 84 Btu/hr • $\text{ft}^2$ [265 $\text{W/m}^2$]

Q = 30 × 0.70 × 84 = 1,764 Btu/hr

[Q = 2.8 × 0.70 × 265 = 519 W]

**7**

$$\text{infiltration airflow} = \frac{3{,}000\ \text{ft}^3 \times 0.5\ \text{air changes/hr}}{60\ \text{min/hr}} = 25\ \text{cfm}$$

$$\left[\text{infiltration airflow} = \frac{85\ \text{m}^3 \times 0.5\ \text{air changes/hr}}{3{,}600\ \text{sec/hr}} = 0.012\ \text{m}^3\text{/s}\right]$$

$Q_{\text{sensible}} = 1.085 \times \text{airflow} \times \Delta T = 1.085 \times 25 \times (100 - 75) = 678$ Btu/hr

[$Q_{\text{sensible}} = 1{,}210 \times \text{airflow} \times \Delta T = 1{,}210 \times 0.012 \times (37.8 - 23.9) = 202$ W]

$Q_{\text{latent}} = 0.7 \times \text{airflow} \times \Delta W = 0.7 \times 25 \times (92 - 65) = 473$ Btu/hr

[$Q_{\text{latent}} = 3{,}010 \times \text{airflow} \times \Delta W = 3{,}010 \times 0.012 \times (13.1 - 9.3) = 137$ W]

# Answers

**8 a**

$$\text{SHR} = \frac{\text{sensible heat gain in the space}}{\text{sensible heat gain in the space} + \text{latent heat gain in the space}}$$

where,

space sensible heat gain = 7,000 + 500 + 1,400 + 12,200 + 4,500 + 22,100 + 9,700 + 2,700 = 60,100 Btu/hr [2,050 + 150 + 400 + 3,600 + 1,300 + 6,500 + 2,800 + 800 = 17,600 W]

space latent heat gain = 3,600 + 2,900 = 6,500 Btu/hr [1,050 + 850 = 1,900 W]

$$\text{SHR} = \frac{60{,}100\ \text{Btu/hr}}{60{,}100\ \text{Btu/hr} + 6{,}500\ \text{Btu/hr}}\left[\frac{17{,}600\ \text{W}}{17{,}600\ \text{W} + 1{,}900\ \text{W}}\right] = 0.90$$

**b**

$$\text{supply airflow} = \frac{\begin{matrix}\text{space sensible}\\ \text{heat gain}\end{matrix}}{\begin{matrix}1.085 \times\\ (\text{room DB} - \text{supply DB})\end{matrix}} = \frac{60{,}100}{1.085 \times (75 - 54)} = 2{,}638\ \text{cfm}$$

$$\left[\text{supply airflow} = \frac{\begin{matrix}\text{space sensible}\\ \text{heat gain}\end{matrix}}{\begin{matrix}1{,}210\\ \text{room DB} - \text{supply DB}\end{matrix}} = \frac{17{,}600}{1{,}210 \times (23.9 - 12.2)} = 1.24\ \text{m}^3/\text{s}\right]$$

**c**

Total cooling load on coil = Space sensible load + space latent load + other loads on the system

Total cooling load on coil = 60,100 + 6,500 + 6,000 + 6,500 + 3,400 = 82,500 Btu/hr or 6.9 refrigeration tons

[Total cooling load on coil = 17,600 + 1,900 + 1,750 + 1,900 + 1,000 = 24,150 W or 24.15 kW]

# Answers

**9** **a** Block load occurs at the time when the sum of the space sensible loads, for all spaces being served by the system, is the greatest.

**b** The block load for sizing the VAV supply fan occurs at 4 p.m. and is 125,000 Btu/hr [37,000 W].

| | Total space sensible heat gain Btu/hr [W] | |
|---|---|---|
| | 9 a.m. | 4 p.m. |
| Room 201 | 70,000 [20,500] | 20,000 [6,000] |
| Room 202 | 10,000 [3,000] | 85,000 [25,000] |
| Room 203 | 15,000 [4,500] | 20,000 [6,000] |
| Sum for the hour | 95,000 [28,000] | 125,000 [37,000] |

**c** The sum-of-peaks load for sizing the constant-volume supply fan is 175,000 Btu/hr [51,500 W]

Sum-of-peaks load = 70,000 + 85,000 + 20,000 = 175,000 Btu/hr [20,500 + 25,000 + 6,000 = 51,500 W]

**10** Solar radiation through windows, heat gain from people, and heat gain from lights

**11** $Q = U \times A \times \Delta T$

$Q = 0.05 \times 100 \times (70 - 0) = 350$ Btu/hr

[$Q = 0.284 \times 9.3 \times (21.1 - (-17.8)) = 103$ W]

**12** Part of the heat gain through exterior walls, part of the heat gain from lights, and all of the heat gain through the roof

# Glossary

**ACCA** Air Conditioning Contractors of America

**angle of incidence** The angle at which the sun's rays strike a surface.

**ASHRAE** American Society of Heating, Refrigerating and Air-Conditioning Engineers

**ballast factor** A factor used to account for the additional heat generated by the ballast used with fluorescent lights.

**block load** Calculated by finding the single instance in time when the *sum* of the space loads is the greatest. This method is used for sizing a VAV supply fan which delivers a varying amount of air to the system.

**CLF** Cooling load factor. Similar to CLTD and SCL, this term is used to account for the ability of the space to absorb and store heat.

**CLTD** Cooling load temperature difference. A factor used to determine the conduction heat gain through a sunlit surface of a building. It is used to account for the effects of the temperature difference across the surface and of the solar radiation striking the outside surface.

**CLTD/SCL/CLF method** A simplified hand-calculation procedure, developed by ASHRAE, that uses a set of tabulated data that was generated using the more advanced transfer function method (TFM). Because of its simplicity, it is the most common method used for basic instruction.

**comfort zone** A range of conditions for delivering acceptable thermal comfort to 80% of the people in a space, defined by ASHRAE Standard 55, *Thermal Environmental Conditions for Human Occupancy.*

**conduction** The process of transferring heat through a solid.

**convection** The process of transferring heat via the movement of a fluid, often through the natural movement of air, caused by temperature (density) differences.

**coil curves** A series of curves on a Trane psychrometric chart that represent the changes in dry-bulb and wet-bulb temperatures as air passes through a "typical" cooling coil.

**constant-volume system** A type of air-conditioning system that varies the temperature of a constant volume of air that is supplied to meet the changing load conditions of the space.

**dry-bulb temperature** A measure of the amount of sensible heat in the air.

**heat transfer coefficient** *See* U-factor.

**humidity ratio** Describes the actual weight of the water in an air–water vapor mixture.

**incident angle** *See* angle of incidence.

# Glossary

**infiltration** Air that leaks into a space from the outdoors through small cracks in the building envelope and around doors and windows.

**latent heat** Causes a change in the air's moisture content with no change in dry-bulb temperature.

**outdoor air** Air brought into the building, either by a ventilation system or through openings provided for natural ventilation, from outside the building.

**plenum** The space between the ceiling and roof. When a space includes a plenum, the heat gain through the roof affects the plenum instead of the occupied space.

**psychrometric chart** A tool used to graphically display the properties of air.

**radiation** The process of transferring heat by means of electromagnetic waves, emitted due to the temperature difference between two objects.

**recirculated return air** Air removed from the conditioned space and reused as supply air, usually after passing through an air-cleaning and -conditioning system, for delivery to the conditioned space.

**relative humidity** Comparison of the amount of moisture that a given amount of air *is* holding, to the amount of moisture that the same amount of air *can* hold, at the same dry-bulb temperature.

**return air** Air that is removed from the conditioned space(s) and is either recirculated or exhausted.

**SCL** Solar cooling load factor. A factor used to estimate the rate at which solar heat energy radiates directly into the space, heats up the surfaces and furnishings, and is later released to the space as a sensible heat gain. Similar to CLTD, the SCL is used to account for the ability of the space to absorb and store heat.

**sensible heat** Causes a change in the air's dry-bulb temperature with no change in moisture content.

**sensible heat ratio (SHR)** Ratio of sensible heat gain to total (sensible + latent) heat gain.

**shading coefficient** An expression used to define how much of the radiant solar energy that strikes the surface of the window is actually transmitted through the window and into the space as a heat gain.

**sum-of-peaks load** The sum of the maximum space loads, regardless of when they occur. This method is used for sizing a constant-volume supply fan that must deliver a constant amount of air to the system.

**supply air** Air that is delivered to the conditioned space by mechanical means for ventilation, heating, cooling, humidification, or dehumidification.

# Glossary

**thermal break** A component of a frame, surrounding a door or window, that reduces the conduction heat transfer through the frame.

**time lag** The amount of time required for heat energy to be transmitted through a structure into the space.

**ton of refrigeration** A measure of the rate of heat flow, defined as a transfer of 12,000 Btu/hr.

**transfer function method (TFM)** A computer-based calculation procedure, developed by ASHRAE, that occurs in two steps. First, it calculates the heat gain due to all components, and then it converts this heat gain into the cooling load of the room, accounting for the ability of the space to absorb and store heat.

**U-factor** Describes the rate at which heat will be transferred through the structure.

**variable-air-volume (VAV) system** A type of air-conditioning system that varies the volume of constant-airstream temperature air supplied to meet the changing load conditions of the space.

**ventilation** The intentional introduction of outdoor air into a space through the use of the building's HVAC system.

**wet-bulb temperature** A measure of the dryness of the air by using a thermometer whose bulb is covered by a wet wick.

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