produced by wind and movement of construction equipment. Most frequently the steel and concrete will not be placed in a symmetrical pattern and frequently large impact loads will occur. Because of the many varied conditions, it is frequently impossible to determine with any great precision the loads which the form must carry. The designer must therefore make safe assumptions by which the forms can be designed such that failure will not result.

Lateral pressures in forms for walls and columns are influenced by a number of factors: weight of concrete, height of placing, vibration, temperature, size and shape of form, amount and distribution of reinforcing steel, and several other variables. Formulas have been suggested for computing safe lateral pressures to be used in form design. However, because of limited test data, they are not generally accepted by all engineers.

Form Liners Absorbent form liners are occasionally applied to the surface of forms to extract the water from the surface of the concrete, eliminate air and water voids, and produce a concrete of uniform appearance with surfaces which are superior in durability and resistance to abrasion.

The vacuum process, whereby water is absorbed from the concrete through a special form liner made of two layers of screen or wire mesh covered by a layer of cloth, has a similar effect and if properly used reduces the water content of the concrete to a depth of several inches.

Neoprene and other types of rubber have been successfully used as liners in precasting work in which a number of units are made from one form. Rubber is particularly suited for patterned work.

Plastic form liners make it easy to obtain a textured surface or a glossy smooth surface. Generally speaking, plastic liners are easily cleaned and if not too thin are suitable for a number of reuses.

Removal of Forms The time that forms should remain in place depends on the character of the members and weather conditions. The strength of concrete must be ascertained before removing the forms. Unless special precautions are taken, concrete should not be placed below 40°F. Fresh concrete should never be subjected to temperatures below freezing. As an approximate guide for the minimum time for form removal, the following rules, which assume moist curing at not less than 70°F for the first 24 h, may be observed.

WALLS IN MASS WORK. In summer, 1 day; in cold weather, 3 days.

THIN WALLS. In summer, 1 day; in cold weather, 5 days.

COLUMNS. In summer, 1 day; in cold weather, 4 days, provided girders are shored to prevent appreciable weight reaching the columns.

SLABS UP TO 7-FT SPAN. In summer, 4 days; in cold weather, 2 weeks.

BEAMS AND GIRDER SIDES. In summer, 1 day; in cold weather, 5 days.

BEAMS AND GIRDERS AND LONG-SPAN SLABS. In summer, 7 days; in cold weather, 2 weeks.

CONDUITS. 2 or 3 days, provided there is not a heavy fill upon them.

ARCHES. If a small size, 1 week; large arches with a heavy dead load, 3 weeks.

Forms for prestressed members may be removed when sufficient prestressing has been applied to enable them to carry their dead loads and the expected construction loads.

EVALUATION OF EXISTING CONCRETE STRUCTURES

This section is intended to give some guidance to the persons responsible for inspections of structures, whether these are done routinely, or before changing a loading or use, or during remodeling, or when something suspicious is found. Serious problems clearly need to be evaluated by a structural engineer experienced in such evaluations, testing, and renovation; such problems should not be evaluated by one who is primarily a structural designer.

The following are some signs of distress. Any cracks wider than about 0.02 in (0.5 mm) are potentially serious, particularly if there are more than a few. If the cracks are inclined like the shear crack shown in Fig. 12.3.1, they are an additional warning sign and should be investigated promptly. At interior supports and other locations where the beams are continuous with the columns, these cracks may also start at the top surface. Any cracking parallel to the member axis is potentially serious.

Rust stains and streaks from cracks indicate corrosion problems. Corrosion, especially from salt, disrupts the concrete surrounding the steel long before the bar areas are significantly reduced, because rust occupies much more volume than the steel it replaces. This rusting causes internal cracking, which can often be detected by tapping on the concrete surface with a hammer (1-lb size), which produces a distinctive ''hollow'' sound. ''Stalactites'' growing from the bottoms of members may be either salt or lime, but both indicate water-penetration problems and the need for waterproofing work.

Concrete exposed to freezing and thawing or to attack by some chemicals may crumble and disintegrate, leading to loss of section area and protective cover on the reinforcement. Large deflections and/or slopes in members may be indicative of impending distress or disaster, but sometimes members were not built very straight, and in such cases the warning sign is actually a *change* in the conditions.

Any reinforced concrete building designed before about 1956 has a potential weakness in the shear strength of the beams and girders because of a deficiency in the codes of that era, and major changes in loading and seemingly minor signs of distress should be investigated carefully. This problem may also exist in highway bridges designed before 1974. Prestressed members should not have this problem.

Repairs are made by injecting epoxy into cracks, by surface patching, by chipping away significant volumes of concrete and replacing it, and by other means. The repair materials range from normal concretes to highly modified concrete and polymer materials.

12.4 AIR CONDITIONING, HEATING, AND VENTILATING

By Norman Goldberg

REFERENCES: ASHRAE Handbooks ''Fundamentals,'' ''Systems and Equipment,'' and ''Applications.'' Stamper and Koral, ''Handbook of Air Conditioning, Heating and Ventilating,'' Industrial Press. Carrier, ''Handbook of Air Conditioning Design,'' McGraw-Hill.

Air conditioning is the process of treating air to meet the requirements of a conditioned space by controlling its temperature, humidity, cleanliness, and distribution. This section presents standards, basic data, and physical laws for use in the design of air conditioning and related heating and ventilating systems.

COMFORT INDEXES

The human body generates heat and disipates that heat to the surrounding air by sensible flow and the evaporationg of moisture.

Effective Temperature Effective temperature (ET) combines the effect of ambient temperature and humidity into a single index.

ASHRAE Comfort Chart The American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE) comfort envelope shown in Fig. 12.4.1 is based on clothing and activity. Comfort varies with skin temperature and skin wettedness.

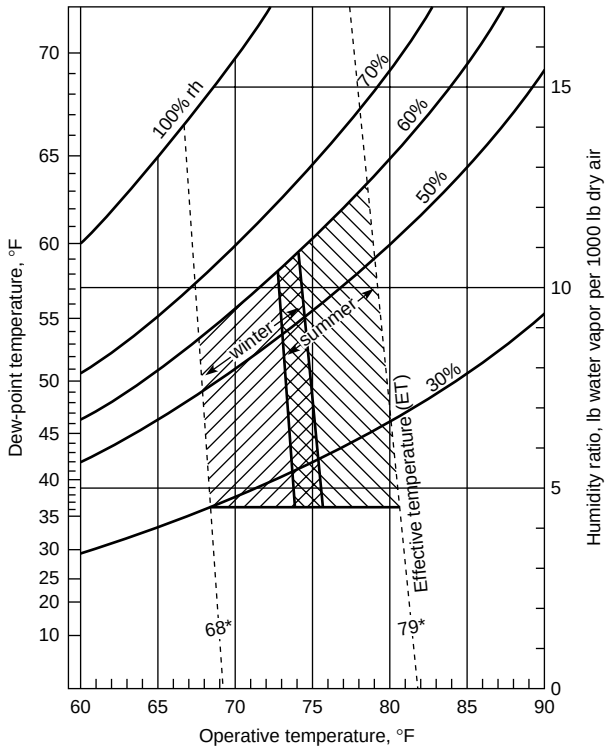


Fig. 12.4.1 Standard effective temperatures and ASHRAE comfort zones. *(ASHRAE ''Handbook of Fundamentals,'' 1993.)*

Temperature-Humidity Index

The term temperature-humidity index (THI) is used to describe the combined effects of temperature and humidity on comfort experienced by people. Since individual reactions can vary considerably from person to person, this quantity should be considered as a guide rather than as an absolute. However, relatively few people will feel discomfort when the THI is 70 or below. By the time it reaches 75, about half the people will be uncomfortable. An index of 80 in a work area may cause a decrease in workers' efficiency. To compute THI, any one of the following formulas may be used:

$$\begin{aligned} \text{THI} &= 0.4(t_d + t_w) + 15 \\ &= 0.55t_d + 0.2t_{dp} + 17.5 \\ &= t_d - (0.55 - 0.55\ \text{RH})(t_d - 58) \end{aligned} \qquad (12.4.1)$$

where t_d = dry-bulb temperature, °F; t_w = wet-bulb temperature, °F; t_{dp} = dew-point temperature, °F; and RH = relative humidity, expressed as a decimal. To convert to degrees Celsius, $t_c = \frac{5}{9}(t_f - 32)$.

Wind-Chill Index

The wind-chill index attempts to describe how much heat the body will lose under certain conditions of wind and temperature. It is determined empirically by an equation which is used to describe the rate of heat loss from a litre cylinder of water at 33°C (91.4°F) as a function of ambient temperature and wind velocity. The formulas for the wind-chill index (WCI) are:

$$\text{WCI} = (10.45 - V + 10\sqrt{V})(33 - t_a) \qquad \text{kcal/m}^2\text{/h} \qquad (12.4.2)$$

where V = metres per second, t_a = °C.

$$\text{WCI} = (10.45 - 0.447V + 6.6854\sqrt{V})(91.4 - t_a) \qquad (12.4.3)$$

where V = miles per hour, t_a = °F.

Instead of using the WCI to express the severity of a cold environment, meteorologists use an index derived from the WCI called the *equivalent wind chill temperature*. This is the ambient temperature that would produce, in a calm wind (defined for this application as 4 mi/h), the same WCI as the actual combination of air temperature and wind velocity. Equivalent wind chill temperature $t_{eq,wc}$ can be calculated by:

$$t_{eq,wc} = -0.0818\ (\text{WCI}) + 91.4 \qquad (12.4.4)$$

where $t_{eq,wc}$ is expressed as a temperature, °F. (See Table 12.4.1.)

INDOOR DESIGN CONDITIONS

Indoor design conditions are determined by the application, usually either the comfort of the people occupying the space or the control of conditions within that space to facilitate a process.

Comfort Conditions Typical indoor design conditions for summer and winter comfort are shown in Table 12.4.2.

Conditions for Process Design These vary widely, depending on the process. See the manufacturer's requirements and/or the ASHRAE Handbook ''Applications.''

Table 12.4.1 Equivalent Wind Chill Temperature of Cold Environments*

Wind Speed, mi/h	Actual Thermometer Reading, °F											
	50	40	30	20	10	0	−10	−20	−30	−40	−50	−60
	Equivalent Chill Temperature, °F											
0	50	40	30	20	10	0	−10	−20	−30	−40	−50	−60
5	48	37	27	16	6	−5	−15	−26	−36	−47	−57	−68
10	40	28	16	3	−9	−21	−34	−46	−58	−71	−83	−95
15	36	22	9	−5	−18	−32	−45	−59	−72	−86	−99	−113
20	32	18	4	−11	−25	−39	−53	−68	−82	−96	−110	−125
25	30	15	0	−15	−30	−44	−59	−74	−89	−104	−119	−134
30	28	13	−3	−18	−33	−48	−64	−79	−94	−110	−125	−140
35	27	11	−4	−20	−36	−51	−67	−83	−98	−114	−129	−145
40†	26	10	−6	−22	−38	−53	−69	−85	−101	−117	−133	−148
Little danger: In less than 5 h, with dry skin. Maximum danger from false sense of security. (WCI less than 1400)					**Increasing danger:** Danger of freezing exposed flesh within one minute. (WCI between 1400 and 2000)			**Great danger:** Flesh may freeze within 30 seconds. (WCI greater than 2000)				

* Cooling power of environment expressed as an equivalent temperature under calm conditions.
† Winds greater than 43 mi/h have little added chilling effect.
SOURCE: U.S. Army Research Institute of Environmental Medicine.

Table 12.4.2 Recommended Inside Design Conditions

Type of application	Summer			Winter				
				With humidification			Without humidification	
	Dry bulb, °F (°C)	Rel. hum., %	Temp. swing, °F (°C)*	Dry bulb, °F (°C)	Rel. hum., %	Temp. swing, °F (°C)‡	Dry bulb, °F	Temp. swing, °F (°C)‡
General comfort (apartment, house, hotel, office, hospital, school, etc.)	74–76 (23–24)	50–45	2 (1.1)	68–72 (20–22)	35–30	3 to 4 (1.6–2.2)	70–74	4 (2.2)
Retail shops (short-term occupancy) (bank, barber or beauty shop, department store, supermarket, etc.)	74–76 (23–24)	50–45	2 (1.1)	68–72 (20–22)	35–30†	3 to 4 (1.6–2.2)	70–74	4 (2.2)
Low-sensible-heat-factor applications (high latent load) (auditorium, church, bar, restaurant, kitchen, etc.)	74–76 (23–24)	55–50	2 (1.1)	68–72 (20–22)	40–35	2 to 3 (1.1–1.7)	70–74	4 (2.2)
Factor comfort (assembly areas, machining rooms, etc.)	76–78 (24–26)	55–45	2 (1.1)	68–70 (18–21)	35–30	4 to 6 (2.2–3.3)	65–70	6 (3.3)

* Temperature swing is above the thermostat setting at peak summer load conditions.
† Winter humidification in retail clothing stores is recommended to maintain the quality texture of goods.
‡ Temperature swing is below the thermostat setting at peak winter load conditions (no lights, people, or sun).
°C = ⅝(°F − 32)
SOURCE: Adapted from Carrier Corp. ''System Design Manual,'' with permission.

OUTDOOR DESIGN CONDITIONS

Outdoor design conditions are based primarily on the geographical location but they are affected to some extent by the criticality of the application and by the economics of maintaining indoor conditions at specified design levels during unusual outdoor weather conditions.

Outdoor design conditions for comfort applications are usually not the most severe conditions that have been experienced in a locality, and recommended levels may be exceeded for an average of 5 percent of the hours during the cooling season.

Frequently used winter outside dry-bulb design temperature are shown in Fig. 12.4.2. Summer outdoor design dry-bulb, wet-bulb, and dew-point temperatures for comfort applications are shown in Figs. 12.4.3 to 12.4.5, respectively.

Design of critical air-conditioning systems and selection of atmospheric evaporative cooling equipment are normally based on more severe outdoor conditions than for comfort applications. Figure 12.4.6 shows wet-bulb temperatures in the United States that are exceeded 1 percent of the time during the cooling season.

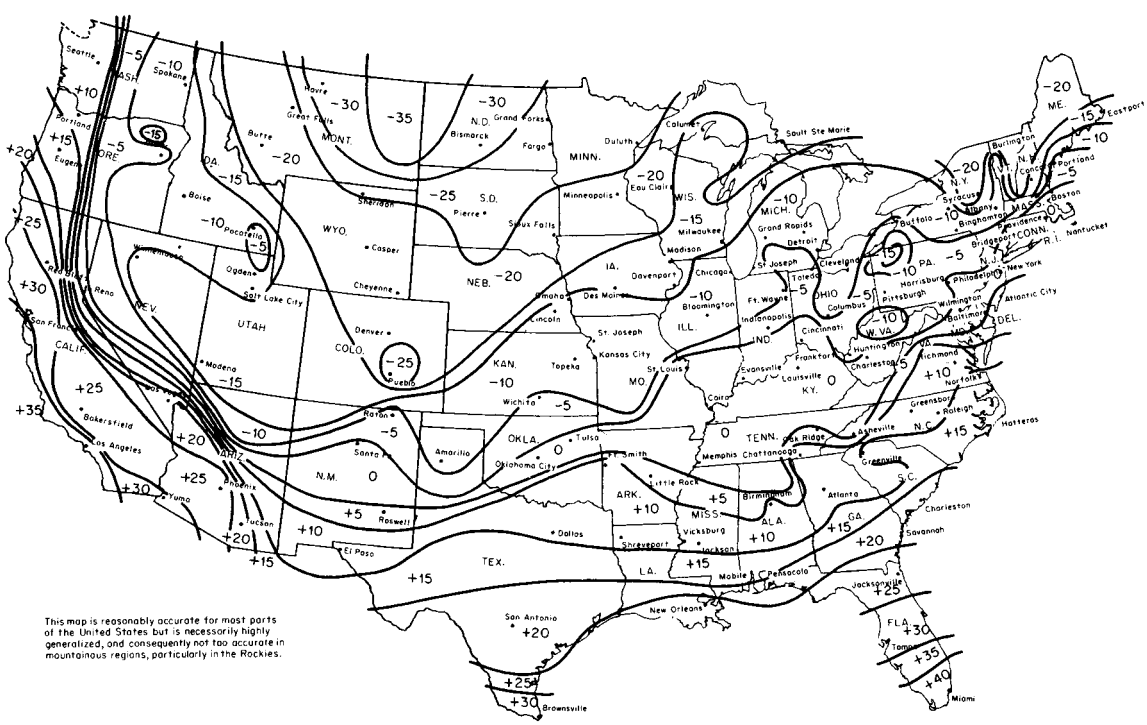


Fig. 12.4.2 Isotherms of winter outdoor temperatures, °F. *[E. Stamper and L. Koral (eds.), ''Handbook of Air Conditioning, Heating and Ventilating,'' 3d ed., 1979, Industrial Press.]*

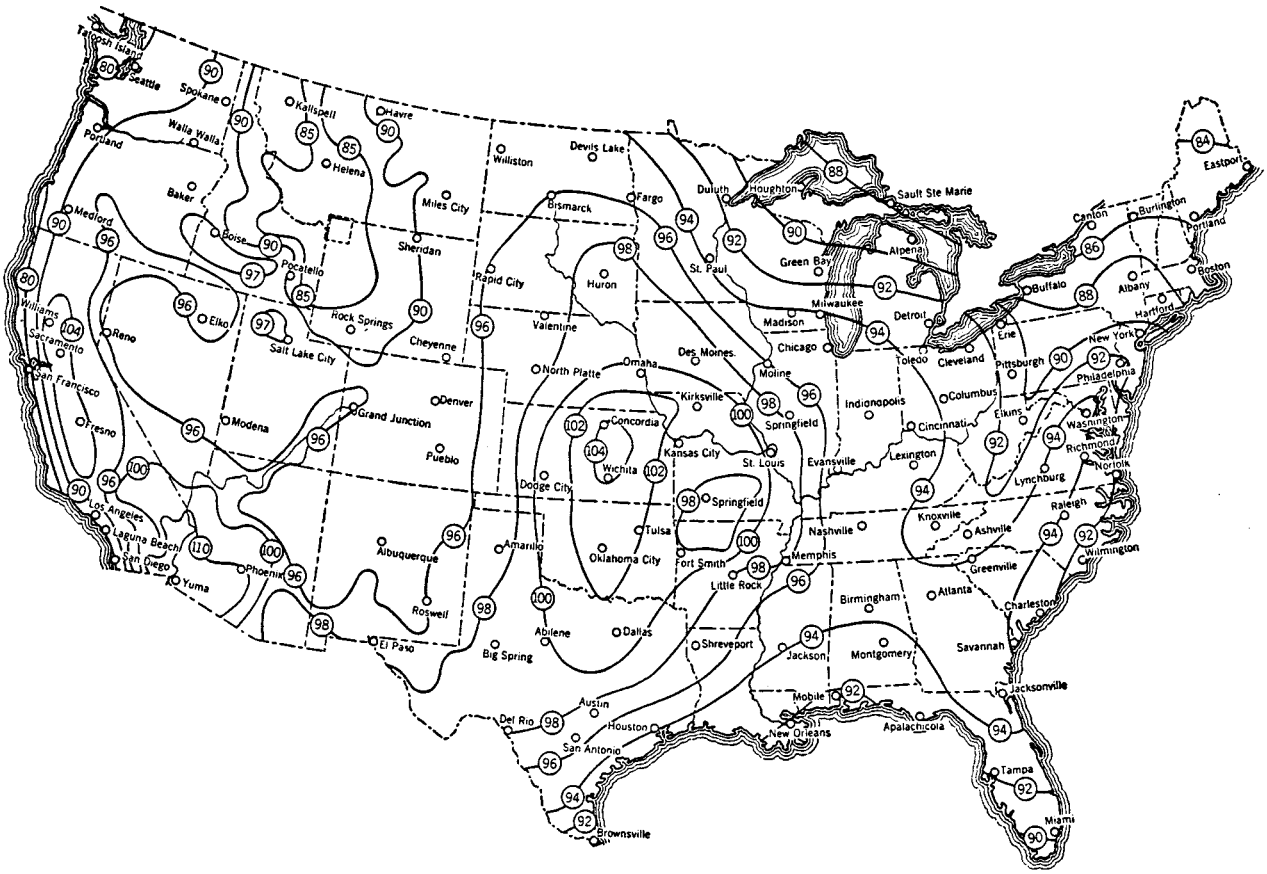


Fig. 12.4.3 Summer dry-bulb temperature data. *(Marley Co.)*

Ventilation and Infiltration

Ventilation Ventilation rates must be adequate to control levels of indoor contaminants and provide makeup air for exhaust systems. It must be recognized that the introduction of outdoor air into a conditioned space represents a significant load on the system since it affects both temperature and moisture levels.

Mechanical ventilation is controllable and is the desirable way to introduce outdoor air into a space. Infiltration is uncontrolled airflow through cracks and other openings and, where feasible, it should be limited. ASHRAE Standard 62-1989, ''Ventilation for Indoor Air Quality,'' presents recommendations for outdoor air rates to maintain acceptable indoor air quality for a wide variety of applications. Requirements of local building codes may differ and may require more, or accept less, ventilation air. An extract from the ASHRAE Standard 62-1989, ''Table of Air Requirements,'' is shown in Table 12.4.3.

Infiltration Infiltration during the summer is due primarily to wind velocity creating a positive pressure on the windward side of a building. During the winter, when the density of indoor and outdoor air differ significantly, stack effect adds substantially to the wind effect. Completely offsetting infiltration by the introduction of sufficient outdoor air through the air-conditioning system is costly. Consequently, outdoor air infiltration should be curtailed by the use of vestibules and, if feasible, revolving doors at frequently used entrances, and tightly fitting windows and/or storm sash throughout. See Tables 12.4.4 and 12.4.5.

Flow of Water Vapor through Building Construction Components Water vapor flow through building construction adds to the latent load, and although it is usually a minor load factor in comfort applications, it can affect the integrity of insulation and other building components.

Vapor transmission is determined by the differences in vapor pressure and the permeability of the construction, but can be reduced by the use of vapor barriers on the high-vapor-pressure side of the construction. In applications where interior design dew points are either high or low, the vapor transmission should be calculated.

The permeance of some materials that will affect the transmission of water vapor is shown in Table 12.4.6.

Infiltration of Moisture

Infiltration of air, and particularly moisture, into a conditioned space is frequently a source of sizable heat gain or loss.

The moisture entering a building as water vapor may be expressed as

$$W_t = W_{trans} + W_{inf} + W_{vent} \tag{12.4.5}$$

where W_t = total weight of vapor, grains (grams). Assuming that W_{trans} is negligible as a load factor,

$$W_1 = W_{inf} + W_{vent} \tag{12.4.6}$$

W_{inf} = air-infiltrated vapor = $W(M_o - M_i)$grains(grams)

$$= W(M_o - M_i)\,\Delta p \text{ grains (grams)} \tag{12.4.7}$$

W_{vent} = ventilation air vapor

$$= W(M_o - M_i) \text{ grains (grams)} \tag{12.4.8}$$

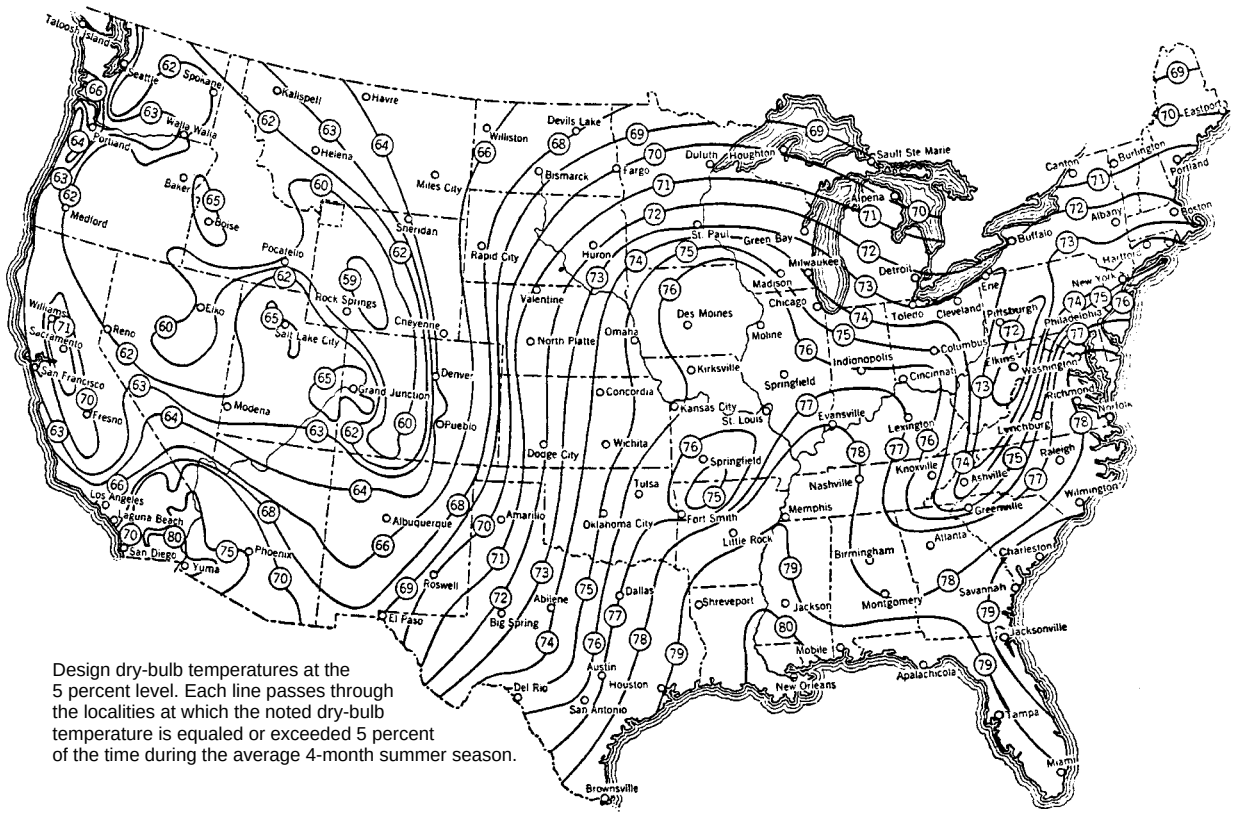


Fig. 12.4.4 Summer wet-bulb temperature data. *(Marley Co.)*

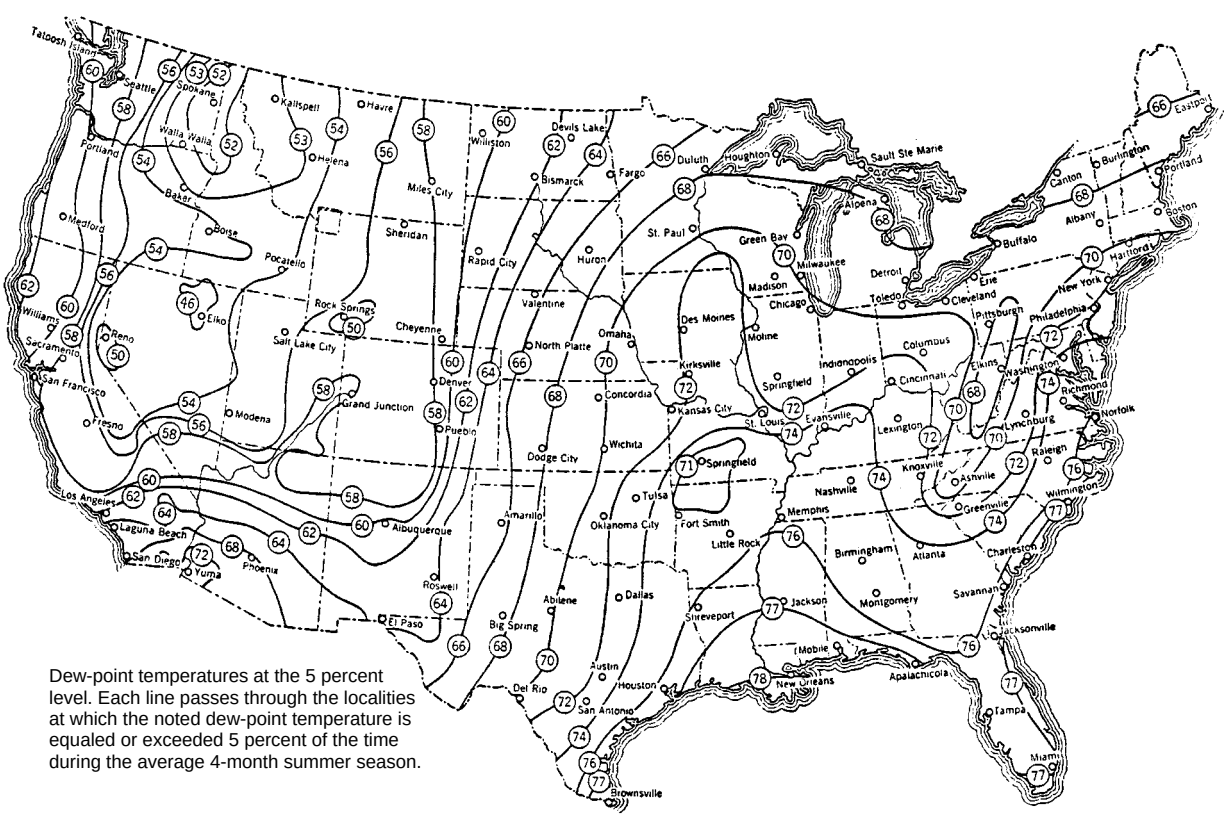
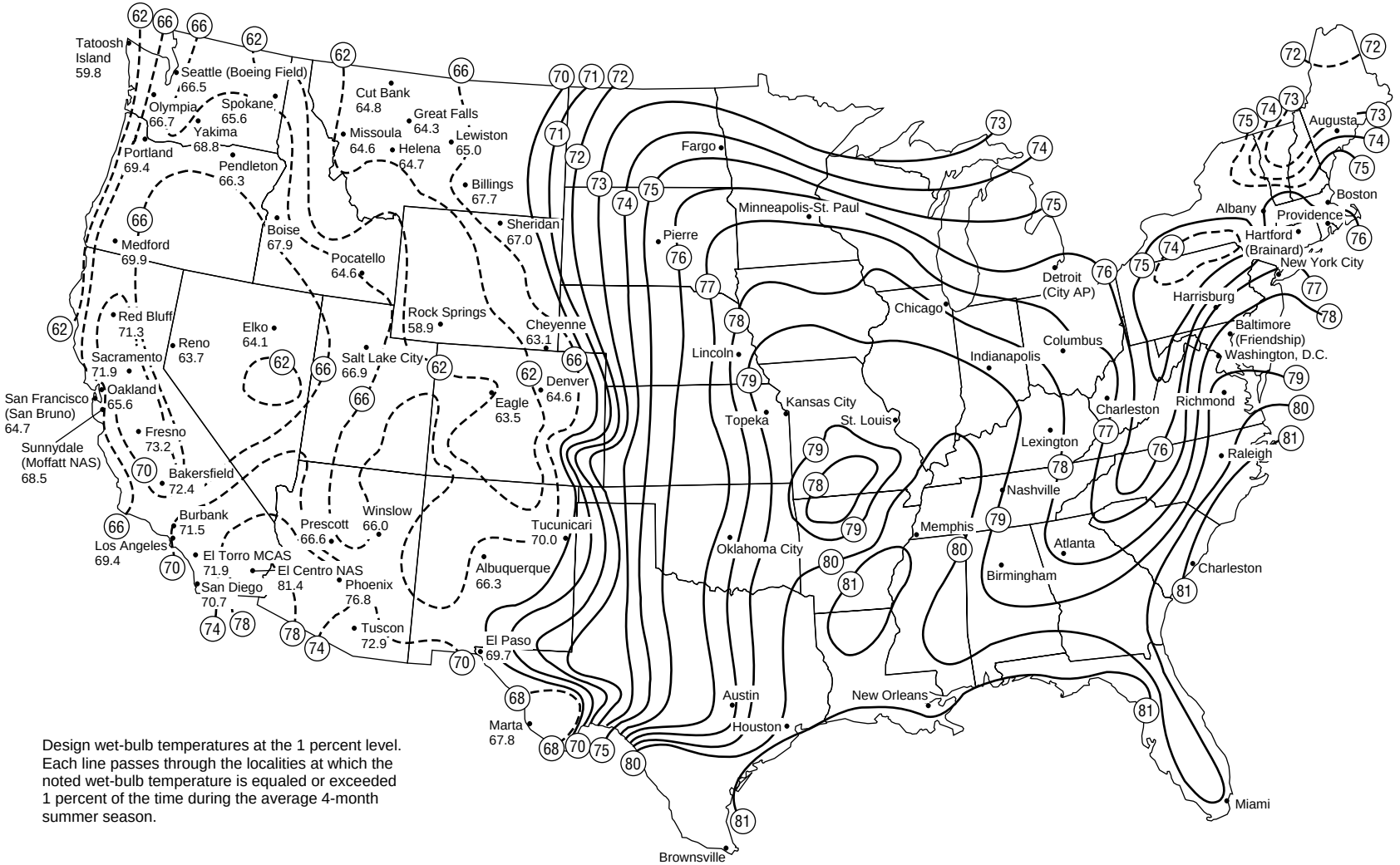


Fig. 12.4.5 Summer dew-point temperature data. *(Marley Co.)*



Design wet-bulb temperatures at the 1 percent level. Each line passes through the localities at which the noted wet-bulb temperature is equaled or exceeded 1 percent of the time during the average 4-month summer season.

Fig. 12.4.6 High wet-bulb temperature data. *(Fluor Products Co.)*

Table 12.4.3 Outdoor Air Requirements for Ventilation*

Application	Estimated maximum† occupancy, persons/1000 ft² or 100 m²	Outdoor air requirements				Comments
		cfm/person	L/s·person	cfm/ft²	L/s·m²	
	Commercial facilities					
Hotels, motels, resorts, dormitories						
Bedrooms				30	15	Per room.
Living rooms				30	15	Per room.
Baths				35	18	Per room. Installed capacity for intermittent use.
Lobbies	30	15	8			
Conference rooms	50	20	10			
Assembly rooms	120	15	8			
Dormitory sleeping areas	20	15	8			See also food and beverage services, merchandising, barber and beauty shops, garages.
Gambling casinos	120	30	15			Supplementary smoke-removal equipment may be required.
Offices						
Office space	7	20	10			Some office equipment may require local exhaust.
Reception areas	60	15	8			
Telecommunication centers and data entry areas	60	20	10			
Conference rooms	50	20	10			Supplementary smoke-removal equipment may be required.
Public spaces						
Corridors and utilites				0.05	0.25	
Public restrooms, cfm/wc or cfm/urinal		50	25			Normally supplied by transfer air. Local mechanical exhaust with no recirculation recommended.
Locker and dressing rooms				0.5	2.5	
Smoking lounge	70	60	30			
Food and beverage service						
Dining rooms	70	20	10			Supplementary smoke-removal equipment may be required.
Cafeteria, fast food	100	20	10			
Bars, cocktail lounges	100	30	15			
Kitchens (cooking)	20	15	8			Exhaust min. 1.5 cfm/ft²
Retail stores, sales floors, and show room floors						
Basement and street	30			0.30	1.50	
Upper floors	20			0.20	1.00	
Storage rooms	15			0.15	0.75	
Dressing rooms				0.20	1.00	
Malls and arcades	20			0.20	1.00	
Shipping and receiving	10			0.15	0.75	
Warehouses	5			0.05	0.25	
Smoking lounge	70	60	30			Normally supplied by transfer air, local mechanical exhaust; exhaust with no recirculation recommended.
Sports and Amusement						
Spectator areas	150	15	8			When internal combustion engines are operated for maintenance of playing surfaces, increased ventilation rates may be required.
Game rooms	70	25	13			
Ice arenas (playing areas)				0.50	2.50	
Swimming pools (pool and deck area)				0.50	2.50	Higher values may be required for humidity control.
Playing floors (gymnasium)	30	20	10			
Ballrooms and discos	100	25	13			
Bowling alleys (seating areas)	70	25	13			
Theaters						Specifal ventilation will be needed to eliminate special stage effects (e.g., dry ice vapors, mists, etc.)
Ticket booths	60	20	10			
Lobbies	150	20	10			
Auditorium	150	15	8			
Stages, studios	70	15	8			
Miscellaneous						
Photo studios	10	15	8			
Darkrooms	10			0.50	2.50	
Pharmacy	20	15	8			
Bank vaults	5	15	8			
Duplicating, printing				0.50	2.50	Installed equipment must incorporate positive exhaust and control (as required) of undesirable contaminants (toxic or otherwise).

Table 12.4.3 Outdoor Air Requirements for Ventilation* (*Continued*)

Application	Estimated maximum† occupancy, persons/1000 ft² or 100 m²	Outdoor air requirements				Comments
		cfm/person	L/s·person	cfm/ft²	L/s·m²	
Institutional Facilities						
Education						
Classroom	50	15	8			
Laboratories	30	20	10			Specifal contaminant control systems may be required for processes or functions including laboratory animal occupancy.
Training shop	30	20	10			
Music rooms	50	15	8			
Libraries	20	15	8			
Locker rooms				0.50	2.50	
Corridors				0.10	0.50	
Auditoriums	150	15	8			
Smoking lounges	70	60	30			Normally supplied by transfer air. Local mechanical exhaust with no recirculation recommended.
Hospitals, nursing and convalescent homes						
Patient rooms	10	25	13			Special requirements or codes and pressure relationships may determine minimum ventilation rates and filter efficiency. Procedures generating contaminants may require higher rates.
Medical procedure	20	15	8			
Operating rooms	20	30	15			
Recovery and ICU	20	15	8			
Autopsy rooms				0.50	2.50	Air shall not be recirculated into other spaces.
Physical Therapy	20	15	8			
Correctional facilities						
Cells	20	20	10			
Dining halls	100	15	8			
Guard stations	40	15	8			

* This table prescribes supply rates of acceptable outdoor air required for acceptable indoor air quality. These values have been chosen to control CO_2 and other contaminants with an adequate margin of safety and to account for health variations among people, varied activity levels, and a moderate amount of smoking. Rationale for CO_2 control is presented in Appendix D of Standard 62-1989.

† Net occupable space.

SOURCE: Abstracted from ASHRAE Standard 62-1989, with permission.

Table 12.4.4 Infiltration through Doors—Winter*

15 mi/h (24 km/h) wind velocity†; doors on one or adjacent windward sides‡

	ft³/min per ft² area (m³/min per m²)§									
			Average use							
	Infrequent use		1 and 2 story buildings		Tall buildings, ft (m)					
	ft³/min per ft²	m³/min per m²	ft³/min per ft²	m³/min per m²	50	(15.25 m)	100	(30.5 m)	200	(61 m)
Revolving door	1.6	0.48	10.5	3.15	12.6	3.78	14.2	4.26	17.3	5.19
Glass door [3⁄16 in (5 mm) crack]	9.0	2.70	30.0	9.00	36.0	10.80	40.5	12.15	49.5	14.85
Wood door 3 × 7 ft (0.9 × 2 m)	2.0	0.60	13.0	3.90	15.5	4.65	17.5	5.25	21.5	6.45
Garage and shipping-room door	4.0	1.2	9.0	2.70						

SOURCE: Carrier Corporation ''System Design Manual,'' Part 1, Load Estimating, 1970.

* All values are based on the wind blowing directly at the window or door. When the prevailing wind direction is oblique to the window or doors, multiply the values by 0.60 and use the total window and door areas on the windward side(s).

† Based on a wind velocity of 15 mi/h (24 km.p.a). For design wind velocities different from the base, multiply the table values by the ratio of velocities.

‡ Stack effect in tall buildings may also cause infiltration on the leeward side. To evaluate this, determine the equivalent velocity V_i and subtract the design velocity V. The equivalent velocity is

$$V_i = \sqrt{V^2 - 1.75a} \text{ (upper section)}$$
$$= \sqrt{V^2 - 1.75b} \text{ (lower section)}$$

where a and b are the distances above and below the midheight of the building, respectively, in feet.

Multiple the table values by the ratio $(V_i - V)/15$ for one-half of the windows and doors on the leeward side of the building. (Use values under one- and two-story building for doors on leeward side of tall buildings.)

§ Doors on opposite sides increase values 25%.

Table 12.4.5 Infiltration Through Windows and Doors—Crack Method—Summer–Winter*
Double-hung windows, unlocked on windward side.

	cfm per linear foot of crack											
	Wind velocity, mi/h											
	5		10		15		20		25		30	
Type of double-hung window	No W strip	W strip	No W strip	W strip	No W strip	W strip	No W strip	W strip	No W strip	W strip	No W strip	W strip
Wood sash												
Average Window	0.12	0.07	0.35	0.22	0.65	0.40	0.98	0.60	1.33	0.82	1.73	1.05
Poorly Fitted Window	0.45	0.10	1.15	0.32	1.85	0.57	2.60	0.85	3.30	1.18	4.20	1.53
Poorly Fitted—with Storm Sash	0.23	0.05	0.57	0.16	0.93	0.29	1.30	0.43	1.60	0.59	2.10	0.76
Metal Sash	0.33	0.10	0.78	0.32	1.23	0.53	1.73	0.77	2.3	1.00	2.8	1.27

SOURCE: Carrier ''Handbook of Air Conditioning Design,'' McGraw-Hill, 1965.
* Infiltration caused by stack effect must be calculated separately during the winter.
† No allowance has been made for usage.

Table 12.4.6 Water Vapor Transmission Through Various Materials

Description of material or construction	Permeance, Btu/(h · 100 ft²) (gr/lb diff) latent heat
Packaging materials	
Cellophane, moisture-proof	0.01–0.25
Glassine (1 ply waxed or 3 ply plain)	0.0015–0.006
Kraft paper soaked with paraffin wax, 4.5 lb per 100 ft²	1.4–3.1
Pliofilm	0.01–0.025
Paint films	
2 coats aluminum paint, estimated	0.05–0.2
2 coats asphalt paint, estimated	0.05–0.1
2 coats lead and oil paint, estimated	0.1–0.6
2 coats water emulsion, estimated	5.0–8.0
Papers	
Duplex or asphalt laminae (untreated)	
30-30-30, 3.1 lb per 100 ft²	0.15–0.27
30-60-30, 4.2 lb per 100 ft²	0.051–0.091
Kraft paper	
1 sheet	8.1
2 sheets	5.1
Aluminum foil on one side of sheet	0.016
Aluminum foil on both sides of sheet	0.012
Sheathing paper	
Asphalt impregnated and coated, 7 lb per 100 ft²	0.02–0.10
Slaters felt, 6 lb per 100 ft², 50% saturated with tar	1.4
Roofing Felt, saturated and coated with asphalt	
25 lb/ft²	0.015
50 lb/ft²	0.011
Tin sheet with 4 holes 1/16-in diameter	0.17
Crack 12 in long by 1/32 in wide (approximate from above)	5.2

Painted surfaces: Two coats of a good vapor seal paint on a smooth surface give a fair vapor barrier. More surface treatment is required on a rough surface than on a smooth surface. Data indicates that either asphalt or aluminum paint is good for vapor seals.

Aluminum foil on paper: This material should also be applied over a smooth surface and joints lapped and sealed with asphalt. The vapor barrier should always be placed on the side of the wall having the higher vapor pressure if condemnation of moisture in wall is possible.

Application: The heat gain due to water vapor transmission through walls may be neglected for the normal air-conditioning or refrigeration job. This latent gain should be considered for air-conditioning jobs where there is a great vapor pressure difference between the room and the outside, particularly when the dew point inside must be low. Note that moisture gain due to infiltration usually is of much greater magnitude than moisture transmission through building structures.

Conversion factors: To convert above table values to:

grain/(hr) (sq ft) (inch mercury vapor pressure difference), multiply by 9.8. grain/(hr) (sq ft) (pounds per sq inch vapor pressure difference), multiply by 20.0.

To convert Btu latent heat to grains, multiply by 7000/1060 = 6.6.

SOURCE: Carrier ''Air Conditioning Design Manual,'' 1965.

Table 12.4.7 Overall Heat-Transfer Coefficient *U*
Outside air 15 mi/h wind, inside still air.

Example	Construction	Btu/(h·ft²·°F)	W/(m²·K)
Frame walls	Wood siding, insulation board, air space, gypsum board	0.19	1.079
Frame partition	Gypsum board, air space, gypsum board	0.34	1.931
Frame construction ceilings and floors	Linoleum or tile, felt, plywood, wood subfloor, air space, metal lath, plaster	0.23	1.306
Pitched roofs	Asphalt shingles, building paper, wood sheathing, air space, gypsum, lath, plaster	0.28	1.590
Masonry wall	Face brick 4 in, common brick 4 in	0.48	2.725
	Face brick 4 in, common brick 4 in, air space, gypsum lath, plaster	0.29	1.647
Masonry partition	Cement block (cinder aggregate), plaster on both sides	0.31	1.760
Flat masonry roof	Built-up roofing, roof insulation 1 in, concrete slab 4 in, air space, metal lath, plaster	0.18	1.022

SOURCE: ASHRAE ''Handbook of Fundamentals,'' 1981.

Table 12.4.8 Coefficient Heat Transmission *U* for Windows and Skylights
Air-to-air heat transfer, Btu/(h·ft²·°F) and W/(m²·K); outside air 0°F, 15 mi/h (24 km/h) wind, no solar radiation; inside still air.

	Vertical glass sheets				Horizontal glass sheets			
	Outdoor exposure		Indoor exposure		Outdoor exposure		Indoor exposure	
	Btu/(h·ft²·°F)	W/(m²·K)	Btu/(h·ft²·°F)	W/(m²·K)	Btu/(h·ft²·°F)	W/(m²·K)	Btu/(h·ft²·°F)	W/(m²·K)
Common window glass, single sheet	1.13	6.416	0.75	4.259	1.22	6.927	0.96	5.450
Common window glass, two sheets, 1-in air space (2.54 cm)	0.53	3.009	0.45	2.555	0.63	3.577	0.56	3.180

SOURCE: ASHRAE ''Handbook of Fundamentals,'' 1981.

Table 12.4.9 Coefficient of Heat Transmission *U* for Wood Doors

	Single		With glass storm door	
Construction	Btu/(h·ft²·°F)	(m²·K)	Btu/(h·ft²·°F)	W/(m²·K)
1 in-thick solid door ($2\frac{5}{32}$ in) (2 cm)	0.64	3.634	0.37	2.101
2 in-thick solid door ($1\frac{5}{8}$ in) (4 cm)	0.43	2.442	0.28	1.590
Door containing wood or glass panels	0.85	4.826	0.39	2.214

SOURCE: ASHRAE ''Handbook of Fundamentals,'' 1981.

Table 12.4.10 Surface Conductances and Resistances for Air

Position of surface	Direction of heat flow	Nonreflective $\varepsilon = 0.90$				Surface emissivity, reflective $\varepsilon = 0.20$				Reflective $\varepsilon = 0.05$			
		C	*C′*	*R*	*R′*	*C*	*C′*	*R*	*R′*	*C*	*C′*	*R*	*R′*
Still air:													
Horizontal	Upward	1.63	9.26	0.61	0.108	0.91	5.17	1.10	0.193	0.76	4.32	1.32	0.232
Vertical	Horizontal	1.46	8.29	0.68	0.121	0.74	4.20	1.35	0.238	0.59	3.35	1.70	0.299
Horizontal	Downward	1.08	6.76	0.92	0.148	0.37	2.10	2.70	0.476	0.22	1.25	4.55	0.800
Moving air (any position):													
15 mi/s (24 km/h) wind (for winter)	Any	6.00	34.07	0.17	0.029								
7½ mi/h (12 km/h) wind (for summer)	Any	4.00	22.71	0.25	0.044								

SOURCE: Adapted from ASHRAE ''Handbook of Fundamentals,'' 1993.
NOTES: *C* = conductance, Btu/(h·ft²·°F temp. diff.)
C′ = conductance W/(m²·K).
R = resistance = 1/*C*.
R′ = resistance = 1/*C′*.

Table 12.4.11 Determination of *U* value Resulting from Addition of Insulation to Any Given Building Section

Given building section property*, †				Added R‡, § (R')													
				$R = 4$	$R' = 0.70$	$R = 6$	$R' = 1.05$	$R = 8$	$R' = 1.41$	$R = 12$	$R' = 2.13$	$R = 16$	$R' = 2.86$	$R = 20$	$R' = 3.57$	$R = 24$	$R' = 4.17$
U	U'	R	R'	U	U'	U	U'	U	U'	U	U'	U	U'	U	U'	U	U'
1.00	5.68	1.00	0.18	0.20	1.13	0.14	0.79	0.11	0.62	0.08	0.45	0.06	0.34	0.05	0.28	0.04	0.23
0.90	5.11	1.11	0.20	0.20	1.13	0.14	0.79	0.11	0.62	0.08	0.45	0.06	0.34	0.05	0.28	0.04	0.23
0.80	4.54	1.25	0.22	0.19	1.08	0.14	0.79	0.11	0.62	0.08	0.45	0.06	0.34	0.05	0.28	0.04	0.23
0.70	3.97	1.43	0.25	0.19	1.08	0.13	0.74	0.11	0.62	0.07	0.40	0.06	0.34	0.05	0.28	0.04	0.23
0.60	3.41	1.67	0.29	0.19	1.08	0.13	0.74	0.10	0.57	0.07	0.40	0.06	0.34	0.05	0.28	0.04	0.23
0.50	2.84	2.00	0.35	0.18	1.02	0.13	0.74	0.10	0.57	0.07	0.40	0.06	0.34	0.05	0.28	0.04	0.23
0.40	2.27	2.50	0.44	0.16	0.91	0.12	0.68	0.10	0.57	0.07	0.40	0.05	0.28	0.05	0.28	0.04	0.23
0.30	1.70	3.33	0.59	0.14	0.79	0.11	0.62	0.09	0.51	0.07	0.40	0.05	0.28	0.04	0.23	0.04	0.23
0.20	1.14	5.00	0.88	0.11	0.62	0.09	0.51	0.08	0.45	0.06	0.34	0.05	0.28	0.04	0.23	0.03	0.17
0.10	0.57	10.00	1.75	0.06	0.34	0.06	0.34	0.06	0.34	0.05	0.28	0.04	0.23	0.04	0.23	0.03	0.17
0.08	0.45	12.50	2.22	0.06	0.34	0.05	0.28	0.05	0.28	0.04	0.23	0.04	0.23	0.03	0.17	0.03	0.17

SOURCE: Abstracted from ASHRAE ''Handbook of Fundamentals,'' 1981, with permission.
NOTE: U and R expressed in U.S. customary units. U' and R' in SI units.
* For U and R values not shown in table, interpolate as necessary.
† Enter column 1 with U or R of the design building section.
‡ Under appropriate column heading for Added R, find U value of resulting design section.
§ If the insulation occupies a previously considered air space, an adjustment must be made in the given building section R value.

where Δp = vapor-pressure difference through flow path, in Hg (N/m^2); W = weight of air, lb (kg); M_o = moisture content of outside air, g/lb (g/kg); M_i = moisture content of inside air, g/lb (g/kg).

For a given locality, the moisture content of the outdoor air may be determined from Fig. 12.4.5 (summer dew-point temperature data) and the appropriate psychrometric chart.

Conductance Coefficients Tables 12.4.7 to 12.4.9 list conductance heat-transfer coefficients U for various building components. Table 12.4.10 shows conductance heat transfer coefficients for reflective and nonreflective surfaces. Table 12.4.11 facilitates the computation of the enhancement of the U value by the addition of insulation. Table 12.4.12 presents an example of calculation of U for an assembly of various building components.

AIR CONDITIONING

Cooling Load

Cooling load Q_1, the total simultaneous cooling load, expressed in Btu/hour or (watts):

$$Q_t = Q_{ext} + Q_{int} + Q_{outside\ air} \tag{12.4.9}$$

$$Q_{ext} = \text{external heat gains} = Q_{transmission} + Q\text{-}_{solar} \tag{12.4.10}$$

For walls and roofs

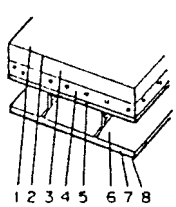
$$Q_{tr-sol} = AU\,(\text{Sa}\,\Delta t) \qquad \text{Btu/h (watts)} \tag{12.4.11}$$

Table 12.4.12 Example of Calculation of Coefficient of Transmission *U* for Composite Building Components

U and R expressed in U.S. customary units, U' and R' in SI units

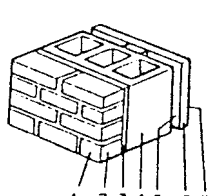
Example 1. Coefficients of transmission U of flat masonry roofs with built-up roofing

Construction (heat flow up)	Resistance R	Resistance R'
1. Outside surface (15 mi/h wind 24 km/h)	0.17	0.03
2. Built-up roofing ⅜ in (9 mm)	0.33	0.06
3. Roof insulation (none)		
4. Concrete slab (lt. wt. agg.) (2 in) (50 mm)	2.22	0.39
5. Corrugated metal	0	0
6. Air space	0.85	0.15
7. Metal lath and ¾ in (19 mm) plaster (lt. wt. agg.)	0.47	0.08
8. Inside surface (still air)	0.61	0.11
Total resistance	4.65	0.82
$U = 1/R = 1/4.65 =$	0.22	
$U' = 1/R' = 1/0.82 =$		1.22



Example 2. Coefficients of transmission U of masonry walls

Construction (beat flow up)	Resistance R	Resistance R'
1. Outside surface (15 mi/h wind) (24 km/h)	0.17	0.03
2. Face brick (4 in) (100 mm)	0.44	0.08
3. Cement mortar (½ in) (12.5 mm)	0.10	0.02
4. Concrete block (cinder agg.) (8 in) (200 mm)	1.72	0.30
5. Air space (reflective)	2.80	0.49
6. Gypsum wallboard, foil back (½ in) (12.5 mm)	0.45	0.08
7. Inside surface (still air)	0.68	0.12
Total resistance	6.36	1.12
$U = 1/R = 1/6.36 =$	0.36	
$U' = 1/R' = 1/1.12 =$		0.89



SOURCE: Abstracted from ASHRAE ''Handbook of Fundamentals,'' 1981, with permission.

Table 12.4.13 Total Equivalent Temperature Differentials for Calculating Heat Gain through Sunlit Walls

North latitude wall facing	Sun time: A.M. 8 D	8 L	10 D	10 L	12 D	12 L	2 D	2 L	P.M. 4 D	4 L	6 D	6 L	8 D	8 L	10 D	10 L	12 D	12 L	South latitude wall facing
	Exterior color of wall: D = dark, L = light																		
	Group A																		
NE	27	16	31	18	26	17	24	17	24	18	23	17	20	15	17	13	15	11	SE
E	32	18	41	24	37	22	29	20	28	20	26	19	23	16	20	14	18	13	E
SE	25	15	36	21	38	23	33	21	28	20	26	18	22	16	19	14	18	12	NE
S	14	9	20	13	28	18	33	22	31	21	25	18	20	15	17	13	15	11	N
SW	17	11	20	13	24	16	34	22	42	27	41	26	28	19	20	14	18	12	NW
W	17	11	20	13	24	16	30	20	42	27	48	30	33	22	22	15	19	13	W
NW	14	9	17	11	21	14	23	17	31	21	38	25	28	19	18	13	16	11	SW
N	14	9	15	10	18	12	20	15	21	16	21	16	18	14	14	11	12	9	S
	Group B																		
NE	12	7	27	14	31	17	30	19	31	21	30	22	27	20	21	17	16	13	SE
E	14	8	34	18	45	24	43	25	39	25	35	24	30	22	23	18	17	14	E
SE	9	5	25	13	39	21	44	26	41	26	37	25	31	23	24	18	17	14	NE
S	4	3	7	4	18	11	32	19	41	26	39	27	33	24	25	19	18	15	N
SW	5	3	7	4	11	7	23	15	41	26	54	34	51	33	38	25	26	19	NW
W	6	4	7	4	11	4	18	12	35	23	55	34	59	37	43	28	30	20	W
NW	5	3	6	4	11	7	17	12	26	18	41	27	47	31	36	24	25	18	SW
N	6	4	9	5	12	8	18	12	22	17	25	20	27	21	22	17	16	14	S
	Group C																		
NE	9	6	19	10	26	15	28	17	29	18	29	20	28	20	24	19	20	16	SE
E	10	7	22	12	36	19	40	23	39	23	36	24	33	23	28	20	22	17	E
SE	8	6	16	9	29	16	38	21	39	24	37	24	34	23	28	21	23	17	NE
S	7	5	7	4	12	7	22	14	32	20	36	24	34	24	29	21	23	17	N
SW	9	6	8	5	10	6	16	10	28	18	42	26	48	30	42	28	35	22	NW
W	10	7	9	5	10	6	14	9	24	16	40	25	52	32	47	30	37	24	W
NW	8	6	8	5	9	6	13	9	19	14	30	20	40	27	38	26	30	21	SW
N	7	5	8	5	10	7	14	9	18	13	22	16	25	19	23	18	19	16	S
	Group D																		
NE	8	5	19	10	28	15	29	17	30	19	30	21	28	21	24	19	19	16	SE
E	9	6	23	12	38	20	42	24	40	24	37	24	33	23	27	20	21	17	E
SE	7	5	16	9	30	16	40	22	41	25	38	25	34	24	28	21	22	17	NE
S	5	4	6	4	12	7	23	14	34	21	38	25	35	24	29	21	23	17	N
SW	8	5	7	4	9	6	16	10	30	19	44	28	51	32	43	28	33	22	NW
W	8	6	7	5	9	6	14	9	25	16	42	27	55	34	49	31	37	25	W
NW	7	5	7	4	9	6	13	9	20	14	31	21	42	28	40	27	31	21	SW
N	6	4	8	5	10	6	14	10	19	14	23	17	25	19	24	19	19	16	S
	Group E																		
NE	10	6	23	12	30	16	30	18	30	20	30	21	28	21	23	18	18	14	SE
E	11	6	28	15	42	22	43	24	39	24	36	24	32	23	25	19	19	15	E
SE	8	5	20	11	35	19	42	24	41	25	38	25	33	23	26	20	20	16	NE
S	4	3	6	4	15	9	28	17	38	24	39	26	34	24	27	20	20	16	N
SW	6	4	7	4	10	6	19	12	35	22	49	31	52	33	41	27	30	21	NW
W	7	5	7	4	10	6	16	11	30	20	48	31	57	36	47	30	34	23	W
NW	6	4	6	4	10	6	15	10	23	16	36	24	45	30	38	26	28	20	SW
N	6	4	8	5	11	7	16	11	21	15	24	18	26	20	23	18	18	15	S
	Group F																		
NE	9	7	14	9	21	12	25	15	27	17	29	19	28	20	26	19	23	17	SE
E	10	8	17	10	28	15	35	19	37	22	37	23	35	23	31	22	26	19	E
SE	10	7	13	8	22	12	31	17	36	21	37	23	35	23	32	22	27	19	NE
S	9	7	7	5	10	6	17	10	26	16	32	20	33	22	31	22	27	19	N
SW	12	9	10	6	9	6	13	8	22	14	33	21	42	27	42	27	37	25	NW
W	14	9	11	7	10	6	12	8	19	12	31	20	43	27	46	29	41	27	W
NW	12	8	9	6	9	6	11	8	16	11	24	16	33	22	36	24	33	23	SW
N	8	7	8	6	9	6	12	8	15	11	19	14	22	17	23	18	21	17	S

SOURCE: Adapted from Carrier ''System Design Manual,'' 1970.
NOTE: To convert temperature differentials in °F to °C, multiply by 5/9.

where Sa Δt = sol-air equivalent temperature differential, °F (°C). Tables 12.4.13 and 12.4.14.

Heat Transmission

Heat gain through exterior walls and roofs is highly variable and is caused by a combination of solar heat and the temperature difference between indoor and outdoor air. Total heat flow is affected by the type and weight of the construction, exposure, sun time, latitude, and design conditions. The equivalent temperature difference from Tables 12.4.13 and 12.4.14 combined with the appropriate heat-transfer coefficients U for the construction, Tables 12.4.15 and 12.4.16, permits estimating the total heat flow through walls and roofs.

Heat flow through interior construction is essentially constant and may be determined by using the actual temperatures on either side of the construction and the appropriate U values.

Solar Heat Gain

Heat gain through glass is a combination of the heat gains by conduction and solar radiation. Both factors are highly variable and one or both are affected by the type of glass, shading, day of the year, time of day, location and orientation angle to the sun, and outside and inside temperatures.

For glass:

$$Q_{glass} = Q_u + Q_{solar} \quad (12.4.12)$$

$$= A_1 U(t_o - t_i) + A_2(\text{MSHGF})(\text{CLF})(\text{SC}) \quad (12.4.13)$$

where A_1 = area of total glass, ft² (m²); A_2 = area of sunlit glass, ft² (m²); U = heat-transfer coefficient (Table 12.4.16); t_0 = outside design temperature, °F (°C) (Fig. 12.4.4); t_i = inside design temperature, °F (°C) (Table 12.4.2); MSHGF = maximum solar heat gain factor (Table 12.4.17); CLF = cooling load factor (Tables 12.4.18, 12.4.19, and 12.4.20); SC = shading coefficient (Tables 12.4.21, 12.4.22, 12.4.23, and 12.4.24).

Note that the data in Table 12.4.17 is for vertical and horizontal glass; the data in Tables 12.4.18 to 12.4.24 addresses the effect of heat storage within the space.

Internal Heat Gains

Internal heat gains are primarily from lighting, equipment, and occupants. They are independent of outdoor temperature and solar effect and include both sensible and latent heat.

$$Q_{int} = \text{internal heat gains} = Q_{people} + Q_{lighting} + Q_{equipment} \quad (12.4.14)$$

In some applications such as general office space, internal heat gains are relatively constant; in others where equipment cycles and/or occupancy is not constant, these loads will vary widely.

Heat gains from people vary with the activity and the proportion of men and women. The proportion of sensible latent heat varies with the temperature of the space. Table 12.4.25 lists adjusted heat gains from people for different activities and typical applications.

Lighting heat gains are based on the total electrical wattage of the operating lights, including ballasts, converted to BTU/hr. Note that under some circumstances the entire electrical input to the lights may not immediately become part of the cooling load. This depends on the type and arrangement of the lights, type of air supply and return, space furnishings, the thermal characteristics of the space, and length of time the lights will be on. Unless all of these can be forseen with a high degree of certainty, and lighting represents a significant portion of the cooling load, most designers will use the full electrical input of the operating lights as the lighting load.

Representative heat gains at various lighting levels for the most common lighting fixtures are shown in Fig. 12.4.7. Curves A, B, and C are drawn for fixtures with newer electronic ballasts and high-efficiency lamps. For a given footcandle level of illumination, they use about 40

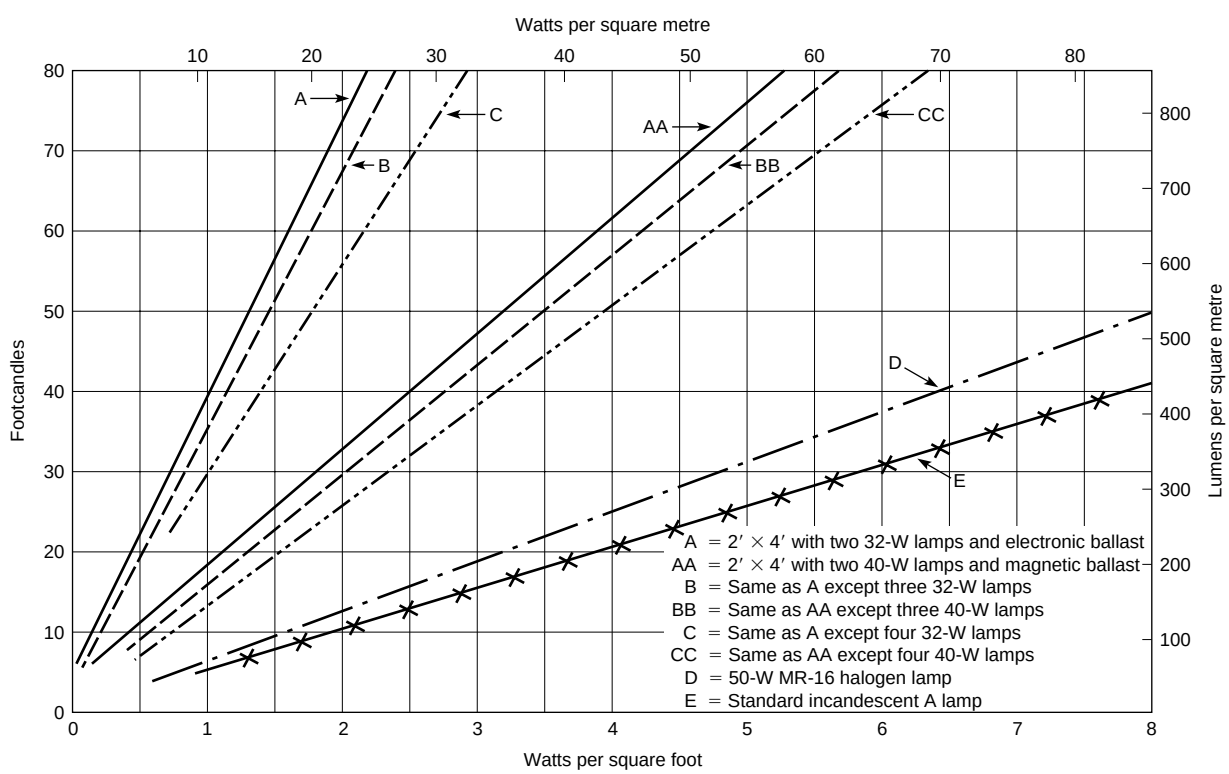


Fig. 12.4.7 Heat gain from typical lighting fixtures. *(Courtesy: S. Ricketts.)*

Table 12.4.14 Total Equivalent Temperature Differentials for Calculating Heat Gain through Flat Roofs

Description of roof construction*, †	Wt., lb/ft²	kg/m²	*U* value Btu/(h·ft²·°F)	*U* value W/(m²·K)
2 in (50 mm) insulation + steel siding	7.8	38.06	0.125	0.710
2 in (50 mm) insulation + 1 in (25 mm) wood‡	8.5	41.48	0.122	0.693
2 in (50 mm) insulation + 2.5 in (62.5 mm) wood‡	13.1	63.93	0.117	0.664
2 in (50 mm) insulation + 4 in (100 mm) wood‡	17.8	88.86	0.113	0.642
2 in (50 mm) insulation + 2 in (50 mm) h.w. concrete	28.8	140.54	0.122	0.693
4 in (100 mm) l.w. concrete	17.8	86.86	0.213	1.209
2 in (50 mm) insulation + 4 in (100 mm) h.w. concrete	52.1	254.25	0.120	0.681
2 in (50 mm) insulation + 6 in (150 mm) h.w. concrete	75.4	367.95	0.117	0.664

* Includes outside surface resistance, ½ in (12.5 mm) membrane and ⅜ in (9 mm) felt on the top and inside surface resistance on the bottom.
† Dark roof (D) = 0.30; light roof (L) = 0.15.
‡ Nominal thickness of wood.

Explanation: Total heat transmission from solar radiation and temperature difference between outdoor and room air, Btu/(h)(ft²)(W/m²) of roof area = equivalent temperature differential from above table × heat-transmission coefficient for summer, Btu/(h·ft²·°F) [W/(m²·K)].

Application. These values may be used for all normal air conditioning estimated; usually without correction (except as noted below in latitude 0 to 50° north or south when the load is calculated for the hottest weather.

Corrections. The values in the table were calculated for an inside temperature of 75°F (24°C) and an outdoor maximum temperature of 95°F (35°C) with an outdoor daily range of 21°F (12°C). The table remains approximately correct for other outdoor maximum (93 to 102°F) (34 to 39°C) and other outdoor daily ranges (16 to 34°F) (9 to 19°C) provided the outdoor daily average temperature remains approximately 85°F (30°C). If the room air temperature is different from 75°F (24°C) and/or the outdoor daily average temperature is different from 85°F (30°C), the following rules can be applied:

1. For room air temperature less than 75°F (24°C) add the difference between 75°F (24°C) and room air temperature; if greater than 75°F (24°C), subtract the difference.
2. For outdoor daily average temperature less than 85°F (30°C) subtract the difference between 85°F (30°C) and the daily average temperature, if greater than 85°F (30°C), add the difference.

Attics or other spaces between the roof and ceiling. If the ceiling is insulated and a fan is used for positive ventilation in the space between the ceiling and roof, the total temperature differential for calculating the room load may be decreased by 25 percent. If the attic space contains a return duct or other air plenum, care should be taken in determining the portion of the heat gain that reaches the ceiling.

Light color. Credit should not be taken for light-colored roofs except where the permanence of light color is established by experience as in rural areas or where there is little smoke.

For solar transmission in other months. The table values of temperature differentials that were calculated for July 21 will be approximately correct for a roof in the following months.

North latitude		South latitude	
Latitude, deg	Months	Latitude, deg	Months
0	All months	0	All months
10	All months	10	All months
20	All months except Nov., Dec., Jan.	20	All months except May, June, July
30	Mar., Apr., May, June, July, Aug., Sept.	30	Sept., Oct., Nov., Dec., Jan., Feb., March
40	April, May, June, July, Aug.	40	Oct., Nov., Dec., Jan., Feb.
50	May, June, July	50	Nov., Dec., Jan.

percent of the energy of earlier fixtures with magnetic ballasts and standard 40-W lamps; see curves AA, BB, and CC. Curves D and E show heat gains from recessed incandescent fixtures.

Heat gain from equipment located within the conditioned space is often a significant portion of the air-conditioning load. Lists of equipment that will be installed, manufacturers' heat release (or wattage) data, and anticipated use and load factors should be obtained from the user early in the design process. Tables 12.4.26, 12.4.27, and 12.4.28 show anticipated gains from a variety of office equipment, restaurant appliances and electric motors. For restaurant cooking equipment, a properly designed, mechanically exhausted hood will reduce the total load of the hooded equipment by about 50 percent.

Miscellaneous Heat Gains Additional heat gains that add to the system load are the heat added at the supply and return fans and heat transferred from unconditioned areas to the distribution ductwork. An additional load on a chilled-water system is the heat equivalent of the chilled-water pump motor work.

Fan heat is the heat added by the fan motor, and manifests itself as the total power input to the fan motor when the motor is located within the airstream or conditioned space, or the net output of the fan motor when it is located out of the airstream or conditioned space. Since air can be considered to be a perfect gas, all the fan heat is added at the fan and the conditioned air temperature will increase as it flows through the fan. If the motor is in the airstream, its inefficiency will add to fan heat.

The contribution of fan heat to temperature rise Δt (°F) is computed as follows:

$$\Delta t = Q_{motor}/1.08 \quad \text{ft}^3/\text{min} \tag{12.4.15}$$

where Q_{motor} is obtained from data in Table 12.4.28.

Sun time																			
A.M.						P.M.													
8		10		12		2		4		6		8		10		12			
D	L	D	L	D	L	D	L	D	L	D	L	D	L	D	L	D	L	λ	δ
Light construction roofs exposed to sun																			
24	8	61	29	88	46	96	53	81	46	48	30	10	8	2	2	−3	−3	0.99	1
8	0	41	18	72	36	90	48	88	49	65	38	30	19	9	7	1	0	0.93	2
1	−2	19	6	43	20	65	33	76	41	72	40	53	31	33	20	18	11	0.68	4
Medium construction roofs exposed to sun																			
6	1	13	4	28	12	45	22	58	30	63	34	56	31	43	25	32	18	0.48	5
2	−2	23	9	49	23	70	36	79	43	71	40	49	29	28	17	15	9	0.73	3
1	−3	28	11	59	28	82	43	88	48	74	42	44	27	19	12	6	4	0.87	3
Heavy construction roofs exposed to sun																			
7	2	15	6	30	13	46	23	58	30	61	33	54	30	41	23	31	17	0.45	5
15	7	17	7	25	11	36	17	46	23	51	27	50	27	43	24	36	20	0.30	6

Table 12.4.15 Descriptions of Wall Construction

		Weight		*U* value	
Group	Components	lb/ft²	kg/m²	Btu/(h·ft²·°F)	W/(m²·K)
A	1 in (2.5 cm) stucco + 4 in (10 cm) l.w. concrete block + air space	28.6	139.63	0.267	1.516
	1 in (2.5 cm) stucco + air space + 2 in (5 cm) insulation	16.3	79.58	0.106	0.601
B	1 in (2.5 cm) stucco + 4 in (10 cm) common brick	55.9	272.90	0.393	2.232
	1 in (2.5 cm) stucco + 4 in (10 cm) l.w. concrete	62.5	305.13	0.481	2.731
C	4 in (10 cm) face brick + 4 in (10 cm) l.w. concrete block + 1 in (2.5 cm) insulation	62.5	305.13	0.158	0.897
	1 in (2.5 cm) stucco + 4 in (10 cm) h.w. concrete + 2 in (5 cm) insulation	62.9	307.08	0.114	0.647
D	1 in (2.5 cm) stucco + 8 in (20 cm) l.w. concrete block + 1 in (2.5 cm) insulation	41.4	202.11	0.141	0.801
	1 in (2.5 cm) stucco + 2 in (5 cm) insulation + 4 in (10 cm) h.w. concrete block	36.6	178.68	0.111	0.630
E	4 in (10 cm) face brick + 3 in (10 cm) l.w. concrete block	62.2	303.66	0.333	1.891
	1 in (2.5 cm) stucco + 8 in (20 cm) h.w. concrete block	56.6	276.32	0.349	1.982
F	4 in (10 cm) face brick + 4 in (10 cm) common brick	89.5	436.94	0.360	2.044
	4 in (10 cm) face brick + 2 in (5 cm) insulation + 4 in (10 cm) l.w. concrete block	62.5	305.13	0.103	0.585

Table 12.4.16 Coefficients of Transmission *U* of Windows, Skylights, and Light Transmitting Partitions
These values are for heat transfer from air to air, Btu/(h · ft² · °F).

Part A—Vertical panels (exterior windows, sliding patio doors, and partitions) —flat glass, glass block, and plastic sheet

Description	Exterior[a] Winter	Exterior[a] Summer	Interior
Flat glass[b]			
Single glass	1.10	1.04	0.73
Insulating glass—double[c]			
0.1875-in airspace[d]	0.62	0.65	0.51
0.25-in airspace[d]	0.58	0.61	0.49
0.5-in airspace[e]	0.49	0.56	0.46
0.5-in airspace, low-emittance coating[f]			
$e = 0.20$	0.32	0.38	0.32
$e = 0.40$	0.38	0.45	0.38
$e = 0.60$	0.43	0.51	0.42
Insulating glass—triple[c]			
0.25-in airspaces[d]	0.39	0.44	0.38
0.5-in airspaces[g]	0.31	0.39	0.30
Storm windows			
1-in to 4-in airspace[d]	0.50	0.50	0.44
Plastic sheet			
Single glazed			
0.125-in thick	1.06	0.98	—
0.25-in thick	0.96	0.89	—
0.5-in thick	0.81	0.76	—
Insulating unit—double[c]			
0.25-in airspace[d]	0.55	0.56	—
0.5-in airspace[c]	0.43	0.45	—
Glass block[h]			
6 × 6 × 4 in thick	0.60	0.57	0.46
8 × 8 × 4 in thick	0.56	0.54	0.44
—with cavity divider	0.48	0.46	0.38
12 × 12 × 4 in thick	0.52	0.50	0.41
—with cavity divider	0.44	0.42	0.36
12 × 12 × 2 in thick	0.60	0.57	0.46

Part B—Horizontal panels (skylights)—flat glass, glass block, and plastic domes

Description	Exterior[a] Winter[i]	Exterior[a] Summer[j]	Interior[l]
Flat glass[e]			
Single glass	1.23	0.83	0.96
Insulating glass—doubles[c]			
0.1875-in airspace[d]	0.70	0.57	0.62
0.25-in airspace[d]	0.65	0.54	0.59
0.5-in airspace[c]	0.59	0.49	0.56
0.5-in airspace, low-emittance coating[f]			
$e = 0.20$	0.48	0.36	0.39
$e = 0.40$	0.52	0.42	0.45
$e = 0.60$	0.56	0.46	0.50
Glass block[h]			
11 × 11 × 3 in thick with cavity divider	0.53	0.35	0.44
12 × 12 × 4 in thick with cavity divider	0.51	0.34	0.42
Plastic domes[k]			
Single-walled	1.15	0.80	—
Double-walled	0.70	0.46	—

Part C—adjustment factors for various window and sliding patio door types (multiply *U* values in parts A and B by these factors)

Description	Single glass	Double or triple glass	Storm windows
Windows			
All Glass[l]	1.00	1.00	1.00
Wood sash—80% glass	0.90	0.95	0.90
Wood sash—60% glass	0.80	0.85	0.80
Metal sash—80% glass	1.00	1.20[m]	1.20[m]
Sliding patio doors			
Wood frame	0.95	1.00	—
Metal frame	1.00	1.10[m]	—

SOURCE: Abstracted from ASHRAE ''Handbook of Fundamentals,'' 1977, with permission.
[a] See Part C for adjustment for various window and sliding patio door types.
[b] Emittance of uncooled glass surface = 0.84.
[c] Double and triple refer to the number of lights of glass.
[d] 0.125-in glass.
[e] 0.25-in glass.
[f] Coating on either glass surface facing airspace; all other glass surfaces uncoated.
[g] Window design: 0.25-in glass–0.125-in glass–0.25-in glass.
[h] Dimensions are nominal.
[i] For heat flow up.
[j] For heat flow down.
[k] Based on area of opening, not total surface area.
[l] Refers to windows with negligible opaque area.
[m] Values will be less than these when metal sash and frame incorporate thermal breaks. In some thermal break designs, *U* values will be equal to or less than those for the glass. Window manufacturers should be consulted for specific data.

Table 12.4.17 Maximum Solar Heat Gain Factor (MSHGF), Btu/h·ft², for Sunlit Glass

0°N Lat

	N	NNE/ NNW	NE/ NW	ENE/ WNW	E/ W	ESE/ WSW	SE/ SW	SSE/ SSW	S	Hor.
Jan.	34	34	88	177	234	254	235	182	118	296
Feb.	36	39	132	205	245	247	210	141	67	306
Mar.	38	87	170	223	242	223	170	87	38	303
Apr.	71	134	193	224	221	184	118	38	37	284
May	113	164	203	218	201	154	80	37	37	265
June	129	173	206	212	191	140	66	37	37	255
July	115	164	201	213	195	149	77	38	38	260
Aug.	75	134	187	216	212	175	112	39	38	276
Sep.	40	84	163	213	231	213	163	84	40	293
Oct.	37	40	129	199	236	238	202	135	66	299
Nov.	35	35	88	175	230	250	230	179	117	293
Dec.	34	34	71	164	226	253	240	196	138	288

8°N Lat

	N	NNE/ NNW	NE/ NW	ENE/ WNW	E/ W	ESE/ WSW	SE/ SW	SSE/ SSW	S	Hor.
Jan.	32	32	71	163	224	250	242	203	162	275
Feb.	34	34	114	193	239	248	219	165	110	294
Mar.	37	67	156	215	241	230	184	110	55	300
Apr.	44	117	184	221	225	195	134	53	39	289
May	74	146	198	220	209	167	97	39	38	277
June	90	155	200	217	200	141	82	39	39	269
July	77	145	195	215	204	162	93	40	39	272
Aug.	47	117	179	214	216	186	128	51	41	282
Sep.	38	66	149	205	230	219	176	107	56	290
Oct.	35	35	112	187	231	239	211	160	108	288
Nov.	33	33	71	161	220	245	233	200	160	273
Dec.	31	31	55	149	215	246	247	215	179	265

16°N Lat

	N	NNE/ NNW	NE/ NW	ENE/ WNW	E/ W	ESE/ WSW	SE/ SW	SSE/ SSW	S	Hor.
Jan.	30	30	55	147	210	244	251	223	199	248
Feb.	33	33	96	180	231	247	233	188	154	275
Mar.	35	53	140	205	239	235	197	138	93	291
Apr.	39	99	172	215	227	204	150	77	45	289
May	52	132	189	218	215	179	115	45	41	282
June	66	142	194	217	207	167	99	41	41	277
July	55	132	187	214	210	174	111	44	42	277
Aug.	41	100	168	209	219	196	143	74	46	282
Sep.	36	50	134	196	227	224	191	134	93	282
Oct.	33	33	95	174	223	237	225	183	150	270
Nov.	30	30	55	145	206	241	247	220	196	246
Dec.	29	29	41	132	198	241	254	233	212	234

24°N Lat

	N	NNE/ NNW	NE/ NW	ENE/ WNW	E/ W	ESE/ WSW	SE/ SW	SSE/ SSW	S	Hor.
Jan.	27	27	41	128	190	240	253	241	227	214
Feb.	30	30	80	165	220	244	243	213	192	249
Mar.	34	45	124	195	234	237	214	168	137	275
Apr.	37	88	159	209	228	212	169	107	75	283
May	43	117	178	214	218	190	132	67	46	282
June	55	127	184	214	212	179	117	55	43	279
July	45	116	176	210	213	185	129	65	46	278
Aug.	38	87	156	203	220	204	162	103	72	277
Sep.	35	42	119	185	222	225	206	163	134	266
Oct.	31	31	79	159	211	237	235	207	187	244
Nov.	27	27	42	126	187	236	249	237	224	213
Dec.	26	26	29	112	180	234	247	247	237	199

32°N Lat

	N	NNE/ NNW	NE/ NW	ENE/ WNW	E/ W	ESE/ WSW	SE/ SW	SSE/ SSW	S	Hor.
Jan.	24	24	29	105	175	229	249	250	246	176
Feb.	27	27	65	149	205	242	248	232	221	217
Mar.	32	37	107	183	227	237	227	195	176	252
Apr.	36	80	146	200	227	219	187	141	115	271
May	38	111	170	208	220	199	155	99	74	277
June	44	122	176	208	214	189	139	83	60	276
July	40	111	167	204	215	194	150	96	72	273
Aug.	37	79	141	195	219	210	181	136	111	265
Sep.	33	35	103	173	215	227	218	189	171	244
Oct.	28	28	63	143	195	234	239	225	215	213
Nov.	24	24	29	103	173	225	245	246	243	175
Dec.	22	22	22	84	162	218	246	252	252	158

40°N Lat

	N (Shade)	NNE/ NNW	NE/ NW	ENE/ WNW	E/ W	ESE/ WSW	SE/ SW	SSE/ SSW	S	Hor.
Jan.	20	20	20	74	154	205	241	252	254	133
Feb.	24	24	50	129	186	234	246	244	241	180
Mar.	29	29	93	169	218	238	236	216	206	223
Apr.	34	71	140	190	224	223	203	170	154	252
May	37	102	165	202	220	208	175	133	113	265
June	48	113	172	205	216	199	161	116	95	267
July	38	102	163	198	216	203	170	129	109	262
Aug.	35	71	135	185	216	214	196	165	149	247
Sep.	30	30	87	160	203	227	226	209	200	215
Oct.	25	25	49	123	180	225	238	236	234	177
Nov.	20	20	20	71	151	201	237	248	250	132
Dec.	18	18	18	60	135	188	232	249	253	113

48°N Lat

	N (Shade)	NNE/ NNW	NE/ NW	ENE/ WNW	E/ W	ESE/ WSW	SE/ SW	SSE/ SSW	S	Hor.
Jan.	15	15	15	53	118	175	216	239	245	85
Feb.	20	20	36	103	168	216	242	249	250	138
Mar.	26	26	80	154	204	234	239	232	228	188
Apr.	31	61	132	180	219	225	215	194	186	226
May	35	97	158	200	218	214	192	163	150	247
June	46	110	165	204	215	206	180	148	134	252
July	37	96	156	196	214	209	187	158	146	244
Aug.	33	61	128	174	211	216	208	188	180	223
Sep.	27	27	72	144	191	223	228	223	230	182
Oct.	21	21	35	96	161	207	213	241	242	136
Nov.	15	15	15	52	115	172	212	234	240	85
Dec.	13	13	13	36	91	156	195	225	233	65

56°N Lat

	N (Shade)	NNE/ NNW	NE/ NW	ENE/ WNW	E/ W	ESE/ WSW	SE/ SW	SSE/ SSW	S	Hor.
Jan.	10	10	10	21	74	126	169	194	205	40
Feb.	16	16	21	71	139	184	223	239	244	91
Mar.	22	22	65	136	185	224	238	241	241	149
Apr.	28	58	123	173	211	223	223	213	210	195
May	36	99	149	195	215	218	206	187	181	222
June	53	111	160	199	213	213	196	174	168	231
July	37	98	147	192	211	214	201	183	177	221
Aug.	30	56	119	165	203	216	215	206	203	193
Sep.	23	23	58	126	171	211	227	230	231	144
Oct.	16	16	20	68	132	176	213	229	234	91
Nov.	10	10	10	21	72	122	165	190	200	40
Dec.	7	7	7	7	47	92	135	159	171	23

60°N Lat

	N (Shade)	NNE/ NNW	NE/ NW	ENE/ WNW	E/ W	ESE/ WSW	SE/ SW	SSE/ SSW	S	Hor.
Jan.	7	7	7	7	46	88	130	152	164	21
Feb.	13	13	13	58	118	168	204	225	231	68
Mar.	20	20	56	125	173	215	234	241	242	128
Apr.	27	59	118	168	206	222	225	220	218	178
May	43	98	149	192	212	220	211	198	194	208
June	58	110	162	197	213	215	202	186	181	217
July	44	97	147	189	208	215	206	193	190	207
Aug.	28	57	114	161	199	214	217	213	211	176
Sep.	21	21	50	145	160	202	222	229	231	123
Oct.	14	14	14	56	111	159	193	215	221	67
Nov.	7	7	7	7	45	86	127	148	160	22
Dec.	4	4	4	4	16	51	76	100	107	9

64°N Lat

	N (Shade)	NNE/ NNW	NE/ NW	ENE/ WNW	E/ W	ESE/ WSW	SE/ SW	SSE/ SSW	S	Hor.
Jan.	3	3	3	3	15	45	67	89	96	8
Feb.	11	11	11	43	89	144	177	202	210	45
Mar.	18	18	47	113	159	203	226	236	239	105
Apr.	25	59	113	163	201	219	225	225	224	160
May	48	97	150	189	211	220	215	207	204	192
June	62	114	162	193	213	216	208	196	193	203
July	49	96	148	186	207	215	211	202	200	192
Aug.	27	58	109	157	193	211	217	217	217	159
Sep.	19	19	43	103	148	189	213	224	227	101
Oct.	11	11	11	40	83	135	167	191	199	46
Nov.	4	4	4	4	15	44	66	87	93	8
Dec.	0	0	0	0	1	5	11	14	15	1

SOURCE: Abstracted from ASHRAE ''Handbook of Fundamentals,'' 1989, with permission.

Table 12.4.18 Cooling Load Factors (CLF) for Glass without Interior Shading, in North Latitude Spaces Having Carpeted Floors

Dir.	Room mass	Solar time																							
		0100	0200	0300	0400	0500	0600	0700	0800	0900	1000	1100	1200	1300	1400	1500	1600	1700	1800	1900	2000	2100	2200	2300	2400
	L	0.00	0.00	0.00	0.00	0.01	0.64	0.73	0.74	0.81	0.88	0.95	0.98	0.98	0.94	0.88	0.79	0.79	0.55	0.31	0.12	0.04	0.02	0.01	0.00
N	M	0.03	0.02	0.02	0.02	0.02	0.64	0.69	0.69	0.77	0.84	0.91	0.94	0.95	0.91	0.86	0.79	0.79	0.56	0.32	0.16	0.10	0.07	0.05	0.04
	H	0.10	0.09	0.08	0.07	0.07	0.62	0.64	0.64	0.71	0.77	0.83	0.87	0.88	0.85	0.81	0.75	0.76	0.55	0.34	0.22	0.17	0.15	0.13	0.11
	L	0.00	0.00	0.00	0.00	0.01	0.51	0.83	0.88	0.72	0.47	0.33	0.27	0.24	0.23	0.20	0.18	0.14	0.09	0.03	0.01	0.00	0.00	0.00	0.00
NE	M	0.01	0.01	0.00	0.00	0.01	0.50	0.78	0.82	0.67	0.44	0.32	0.28	0.26	0.24	0.22	0.19	0.15	0.11	0.05	0.03	0.02	0.02	0.01	0.01
	H	0.03	0.03	0.03	0.02	0.03	0.47	0.71	0.72	0.59	0.40	0.30	0.27	0.26	0.25	0.23	0.20	0.17	0.13	0.08	0.06	0.05	0.05	0.04	0.04
	L	0.00	0.00	0.00	0.00	0.00	0.42	0.76	0.91	0.90	0.75	0.51	0.30	0.22	0.18	0.16	0.13	0.11	0.07	0.02	0.01	0.00	0.00	0.00	0.00
E	M	0.01	0.01	0.00	0.00	0.01	0.41	0.72	0.86	0.84	0.71	0.48	0.30	0.24	0.21	0.18	0.16	0.13	0.09	0.04	0.03	0.02	0.01	0.01	0.01
	H	0.03	0.03	0.03	0.02	0.02	0.39	0.66	0.76	0.74	0.63	0.43	0.29	0.24	0.22	0.20	0.18	0.15	0.12	0.08	0.06	0.05	0.05	0.04	0.04
	L	0.00	0.00	0.00	0.00	0.00	0.27	0.58	0.81	0.93	0.93	0.81	0.59	0.37	0.27	0.21	0.18	0.14	0.09	0.03	0.01	0.00	0.00	0.00	0.00
SE	M	0.01	0.01	0.01	0.00	0.01	0.26	0.55	0.77	0.88	0.87	0.76	0.56	0.37	0.29	0.24	0.20	0.16	0.11	0.05	0.04	0.03	0.02	0.02	0.01
	H	0.04	0.04	0.03	0.03	0.03	0.26	0.51	0.69	0.78	0.78	0.68	0.51	0.35	0.29	0.25	0.22	0.19	0.15	0.09	0.08	0.07	0.06	0.05	0.05
	L	0.00	0.00	0.00	0.00	0.00	0.07	0.15	0.23	0.39	0.62	0.82	0.94	0.93	0.80	0.59	0.38	0.26	0.16	0.06	0.02	0.01	0.00	0.00	0.00
S	M	0.01	0.01	0.01	0.01	0.01	0.07	0.14	0.22	0.38	0.59	0.78	0.88	0.88	0.76	0.57	0.38	0.28	0.18	0.09	0.06	0.04	0.03	0.02	0.02
	H	0.05	0.05	0.04	0.04	0.03	0.09	0.15	0.21	0.35	0.54	0.70	0.79	0.79	0.69	0.52	0.37	0.29	0.21	0.13	0.10	0.09	0.08	0.07	0.06
	L	0.00	0.00	0.00	0.00	0.00	0.04	0.09	0.13	0.16	0.19	0.23	0.39	0.62	0.82	0.94	0.94	0.81	0.54	0.19	0.07	0.03	0.01	0.00	0.00
SW	M	0.02	0.02	0.01	0.01	0.01	0.05	0.09	0.13	0.16	0.19	0.22	0.38	0.60	0.78	0.89	0.89	0.77	0.52	0.20	0.10	0.07	0.05	0.04	0.03
	H	0.07	0.06	0.05	0.05	0.04	0.07	0.11	0.14	0.16	0.18	0.21	0.35	0.55	0.71	0.80	0.79	0.69	0.48	0.20	0.14	0.11	0.10	0.08	0.07
	L	0.00	0.00	0.00	0.00	0.00	0.03	0.07	0.10	0.13	0.15	0.16	0.18	0.31	0.55	0.78	0.92	0.93	0.73	0.25	0.10	0.04	0.01	0.01	0.00
W	M	0.02	0.02	0.01	0.01	0.01	0.04	0.07	0.10	0.13	0.14	0.16	0.17	0.30	0.53	0.74	0.87	0.88	0.69	0.24	0.12	0.07	0.05	0.04	0.03
	H	0.06	0.06	0.05	0.04	0.04	0.06	0.09	0.11	0.13	0.15	0.16	0.17	0.28	0.49	0.67	0.78	0.79	0.62	0.23	0.14	0.11	0.09	0.08	0.07
	L	0.00	0.00	0.00	0.00	0.00	0.04	0.09	0.14	0.17	0.20	0.22	0.23	0.24	0.31	0.53	0.78	0.92	0.81	0.28	0.10	0.04	0.02	0.01	0.00
NW	M	0.02	0.02	0.01	0.01	0.01	0.05	0.10	0.13	0.17	0.19	0.21	0.22	0.23	0.30	0.52	0.75	0.88	0.77	0.26	0.12	0.07	0.05	0.04	0.03
	H	0.06	0.05	0.05	0.04	0.04	0.07	0.11	0.14	0.17	0.19	0.20	0.21	0.22	0.28	0.48	0.68	0.79	0.69	0.23	0.14	0.10	0.09	0.08	0.07
	L	0.00	0.00	0.00	0.00	0.00	0.08	0.25	0.45	0.64	0.80	0.91	0.97	0.97	0.91	0.80	0.64	0.44	0.23	0.08	0.03	0.01	0.00	0.00	0.00
Hor.	M	0.02	0.02	0.01	0.01	0.01	0.08	0.24	0.43	0.60	0.75	0.86	0.92	0.92	0.87	0.77	0.63	0.45	0.26	0.12	0.07	0.05	0.04	0.03	0.02
	H	0.07	0.06	0.05	0.05	0.04	0.11	0.25	0.41	0.56	0.68	0.77	0.83	0.83	0.80	0.71	0.59	0.44	0.28	0.17	0.13	0.11	0.10	0.09	0.08

SOURCE: Abstracted from ASHRAE ''Handbook of Fundamentals,'' 1989, with permission.
Values for nominal 15 ft × 15 ft × 10 ft high space, with ceiling, and 50% or less glass in exposed surface at listed orientation.
L = lightweight construction.
M = mediumweight construction.
H = heavyweight construction.

Table 12.4.19 Cooling Load Factors (CLF) for Glass without Interior Shading, in North Latitude Spaces Having Uncarpeted Floors

Dir.	Room mass	Solar time																							
		0100	0200	0300	0400	0500	0600	0700	0800	0900	1000	1100	1200	1300	1400	1500	1600	1700	1800	1900	2000	2100	2200	2300	2400
	L	0.00	0.00	0.00	0.00	0.01	0.64	0.73	0.74	0.81	0.88	0.95	0.98	0.98	0.94	0.88	0.79	0.79	0.55	0.31	0.12	0.04	0.02	0.01	0.00
N	M	0.12	0.09	0.07	0.06	0.05	0.33	0.45	0.53	0.61	0.69	0.76	0.82	0.85	0.86	0.85	0.81	0.80	0.70	0.60	0.43	0.32	0.24	0.19	0.15
	H	0.24	0.21	0.19	0.18	0.16	0.43	0.48	0.51	0.56	0.61	0.66	0.71	0.73	0.74	0.73	0.71	0.71	0.62	0.52	0.42	0.36	0.32	0.29	0.26
	L	0.00	0.00	0.00	0.00	0.01	0.51	0.83	0.88	0.72	0.47	0.33	0.27	0.24	0.23	0.20	0.18	0.14	0.09	0.03	0.01	0.00	0.00	0.00	0.00
NE	M	0.03	0.02	0.02	0.02	0.02	0.24	0.45	0.57	0.58	0.49	0.41	0.36	0.32	0.29	0.27	0.24	0.21	0.17	0.13	0.10	0.07	0.06	0.05	0.04
	H	0.08	0.07	0.07	0.06	0.06	0.27	0.43	0.49	0.45	0.37	0.32	0.29	0.28	0.27	0.26	0.24	0.22	0.19	0.16	0.14	0.12	0.11	0.10	0.09
	L	0.00	0.00	0.00	0.00	0.00	0.42	0.76	0.91	0.90	0.75	0.51	0.30	0.22	0.18	0.16	0.13	0.11	0.07	0.02	0.01	0.00	0.00	0.00	0.00
E	M	0.03	0.02	0.02	0.02	0.01	0.20	0.41	0.57	0.65	0.64	0.55	0.44	0.36	0.31	0.26	0.23	0.19	0.16	0.12	0.09	0.07	0.06	0.04	0.04
	H	0.08	0.08	0.07	0.06	0.06	0.24	0.40	0.50	0.53	0.50	0.41	0.33	0.30	0.28	0.26	0.24	0.22	0.19	0.16	0.14	0.13	0.11	0.10	0.09
	L	0.00	0.00	0.00	0.00	0.00	0.27	0.58	0.81	0.93	0.93	0.81	0.59	0.37	0.27	0.21	0.18	0.14	0.09	0.03	0.01	0.00	0.00	0.00	0.00
SE	M	0.04	0.03	0.02	0.02	0.02	0.13	0.31	0.48	0.62	0.69	0.69	0.61	0.50	0.41	0.35	0.30	0.25	0.20	0.15	0.12	0.09	0.07	0.06	0.05
	H	0.10	0.09	0.08	0.08	0.07	0.18	0.32	0.45	0.53	0.56	0.54	0.47	0.39	0.35	0.32	0.29	0.26	0.23	0.19	0.17	0.15	0.14	0.12	0.11
	L	0.00	0.00	0.00	0.00	0.00	0.07	0.15	0.23	0.39	0.62	0.82	0.94	0.93	0.80	0.59	0.38	0.26	0.16	0.06	0.02	0.01	0.00	0.00	0.00
S	M	0.05	0.04	0.04	0.03	0.02	0.05	0.09	0.14	0.24	0.38	0.53	0.65	0.72	0.71	0.63	0.52	0.42	0.33	0.24	0.18	0.14	0.11	0.09	0.07
	H	0.13	0.12	0.10	0.09	0.09	0.11	0.14	0.17	0.25	0.36	0.47	0.55	0.58	0.56	0.49	0.41	0.36	0.30	0.25	0.21	0.19	0.17	0.16	0.14
	L	0.00	0.00	0.00	0.00	0.00	0.04	0.09	0.13	0.16	0.19	0.23	0.39	0.62	0.82	0.94	0.94	0.81	0.54	0.19	0.07	0.03	0.01	0.00	0.00
SW	M	0.08	0.07	0.05	0.04	0.03	0.05	0.07	0.09	0.12	0.15	0.17	0.26	0.40	0.54	0.66	0.73	0.72	0.61	0.43	0.31	0.23	0.17	0.13	0.10
	H	0.15	0.14	0.12	0.11	0.10	0.11	0.12	0.14	0.15	0.17	0.18	0.26	0.37	0.48	0.56	0.59	0.57	0.47	0.33	0.27	0.23	0.21	0.19	0.17
	L	0.00	0.00	0.00	0.00	0.00	0.03	0.07	0.10	0.13	0.15	0.16	0.18	0.31	0.55	0.78	0.92	0.93	0.73	0.25	0.10	0.04	0.01	0.01	0.00
W	M	0.08	0.07	0.05	0.04	0.04	0.04	0.06	0.08	0.10	0.12	0.13	0.15	0.21	0.35	0.50	0.63	0.71	0.67	0.46	0.33	0.24	0.18	0.14	0.11
	H	0.14	0.13	0.12	0.11	0.10	0.10	0.11	0.12	0.13	0.14	0.15	0.16	0.21	0.33	0.45	0.54	0.58	0.52	0.33	0.26	0.22	0.19	0.18	0.16
	L	0.00	0.00	0.00	0.00	0.00	0.04	0.09	0.14	0.17	0.20	0.22	0.23	0.24	0.31	0.53	0.78	0.92	0.81	0.28	0.10	0.04	0.02	0.01	0.00
NW	M	0.08	0.06	0.05	0.04	0.03	0.05	0.07	0.10	0.13	0.15	0.17	0.19	0.20	0.24	0.36	0.51	0.64	0.66	0.46	0.32	0.23	0.17	0.13	0.10
	H	0.13	0.12	0.11	0.10	0.09	0.10	0.12	0.13	0.15	0.16	0.17	0.18	0.19	0.23	0.33	0.46	0.55	0.53	0.33	0.25	0.21	0.18	0.16	0.15
	L	0.00	0.00	0.00	0.00	0.00	0.08	0.25	0.45	0.64	0.80	0.91	0.97	0.97	0.91	0.80	0.64	0.44	0.23	0.08	0.03	0.01	0.00	0.00	0.00
Hor.	M	0.07	0.06	0.05	0.04	0.03	0.06	0.14	0.26	0.40	0.53	0.64	0.73	0.78	0.80	0.77	0.70	0.59	0.45	0.33	0.24	0.19	0.14	0.11	0.09
	H	0.16	0.15	0.13	0.12	0.11	0.13	0.20	0.29	0.39	0.48	0.56	0.61	0.65	0.65	0.63	0.57	0.49	0.40	0.32	0.28	0.25	0.22	0.20	0.18

SOURCE: Abstracted from ASHRAE ''Handbook of Fundamentals,'' 1989, with permission.
Values for nominal 15 ft × 15 ft × 10 ft high space, with ceiling, and 50% or less glass in exposed surface at listed orientation.
L = lightweight construction.
M = mediumweight construction.
H = heavyweight construction.

Table 12.4.20 Cooling Load Factors (CLF) for Glass with Interior Shading, North Latitudes (All Room Constructions)

Fenestration facing	Solar time																							
	0100	0200	0300	0400	0500	0600	0700	0800	0900	1000	1100	1200	1300	1400	1500	1600	1700	1800	1900	2000	2100	2200	2300	2400
N	0.08	0.07	0.06	0.06	0.07	0.73	0.66	0.65	0.73	0.80	0.86	0.89	0.89	0.86	0.82	0.75	0.78	0.91	0.24	0.18	0.15	0.13	0.11	0.10
NNE	0.03	0.03	0.02	0.02	0.03	0.64	0.77	0.62	0.42	0.37	0.37	0.37	0.36	0.35	0.32	0.28	0.23	0.17	0.08	0.07	0.06	0.05	0.04	0.04
NE	0.03	0.02	0.02	0.02	0.02	0.56	0.76	0.74	0.58	0.37	0.29	0.27	0.26	0.24	0.22	0.20	0.16	0.12	0.06	0.05	0.04	0.04	0.03	0.03
ENE	0.03	0.02	0.02	0.02	0.02	0.52	0.76	0.80	0.71	0.52	0.31	0.26	0.24	0.22	0.20	0.18	0.15	0.11	0.06	0.05	0.04	0.04	0.03	0.03
E	0.03	0.02	0.02	0.02	0.02	0.47	0.72	0.80	0.76	0.62	0.41	0.27	0.24	0.22	0.20	0.17	0.14	0.11	0.06	0.05	0.05	0.04	0.03	0.03
ESE	0.03	0.03	0.02	0.02	0.02	0.41	0.67	0.79	0.80	0.72	0.54	0.34	0.27	0.24	0.21	0.19	0.15	0.12	0.07	0.06	0.05	0.04	0.04	0.03
SE	0.03	0.03	0.02	0.02	0.02	0.30	0.57	0.74	0.81	0.79	0.68	0.49	0.33	0.28	0.25	0.22	0.18	0.13	0.08	0.07	0.06	0.05	0.04	0.04
SSE	0.04	0.03	0.03	0.03	0.02	0.12	0.31	0.54	0.72	0.81	0.81	0.71	0.54	0.38	0.32	0.27	0.22	0.16	0.09	0.08	0.07	0.06	0.05	0.04
S	0.04	0.04	0.03	0.03	0.03	0.09	0.16	0.23	0.38	0.58	0.75	0.83	0.80	0.68	0.50	0.35	0.27	0.19	0.11	0.09	0.08	0.07	0.06	0.05
SSW	0.05	0.04	0.04	0.03	0.03	0.09	0.14	0.18	0.22	0.27	0.43	0.63	0.78	0.84	0.80	0.66	0.46	0.25	0.13	0.11	0.09	0.08	0.07	0.06
SW	0.05	0.05	0.04	0.04	0.03	0.07	0.11	0.14	0.16	0.19	0.22	0.38	0.59	0.75	0.83	0.81	0.69	0.45	0.16	0.12	0.10	0.09	0.07	0.06
WSW	0.05	0.05	0.04	0.04	0.03	0.07	0.10	0.12	0.14	0.16	0.17	0.23	0.44	0.64	0.78	0.84	0.78	0.55	0.16	0.12	0.10	0.09	0.07	0.06
W	0.05	0.05	0.04	0.04	0.03	0.06	0.09	0.11	0.13	0.15	0.16	0.17	0.31	0.53	0.72	0.82	0.81	0.61	0.16	0.12	0.10	0.08	0.07	0.06
WNW	0.05	0.05	0.04	0.03	0.03	0.07	0.10	0.12	0.14	0.16	0.17	0.18	0.22	0.43	0.65	0.80	0.84	0.66	0.16	0.12	0.10	0.08	0.07	0.06
NW	0.05	0.04	0.04	0.03	0.03	0.07	0.11	0.14	0.17	0.19	0.20	0.21	0.22	0.30	0.52	0.73	0.82	0.69	0.16	0.12	0.10	0.08	0.07	0.06
NNW	0.05	0.05	0.04	0.03	0.03	0.11	0.17	0.22	0.26	0.30	0.32	0.33	0.34	0.34	0.39	0.61	0.82	0.76	0.17	0.12	0.10	0.08	0.07	0.06
Hor.	0.06	0.05	0.04	0.04	0.03	0.12	0.27	0.44	0.59	0.72	0.81	0.85	0.85	0.81	0.71	0.58	0.42	0.25	0.14	0.12	0.10	0.08	0.07	0.06

SOURCE: Abstracted from ASHRAE ''Handbook of Fundamentals,'' 1989, with permission.

Table 12.4.21 Shading Coefficients for Single Glass and Insulating Glass*

Single glass				
			Shading coefficient	
Type of glass	Nominal thickness,† in (mm)	Solar trans.†	$b_0 = 4.0$	$b_0 = 3.0$
Regular sheet	3/32, 1/8 (2, 3)	0.87	1.00	1.00
Regular plate/float	1/4 (6)	0.80	0.95	0.97
	3/8 (9)	0.75	0.91	0.93
	1/2 (12)	0.71	0.88	0.91
Gray sheet	1/8 (3)	0.59	0.78	0.80
	3/16 (5)	0.74	0.90	0.92
	7/32 (6)	0.45	0.66	0.70
	7/32 (6)	0.71	0.88	0.90
	1/4	0.67	0.86	0.88
Heat-absorbing plate/float§	3/16 (5)	0.52	0.72	0.75
	1/4 (6)	0.47	0.70	0.74
	3/8 (9)	0.33	0.56	0.61
	1/2 (12)	0.24	0.50	0.57

Insulating glass*					
		Solar trans.†		Shading coefficient	
Type of glass	Nominal thickness, ‡ in (mm)	Outer pane	Inner pane	$b_0 = 4.0$	$b_0 = 3.0$
Regular sheet out, regular sheet in	3/32, 1/8 (2, 3)	0.87	0.87	0.90	0.90
Regular plate/float out, regular plate/float in	1/4 (6)	0.80	0.80	0.83	0.83
Heat-absorbing plate/float out, regular plate/float in	1/4 (6)	0.46	0.80	0.56	0.58

SOURCE: Abstracted from ASHRAE ''Handbook of Fundamentals,'' 1993, with permission.
* Refers to factory-fabricated units with 3/16 (5), 1/4 (6), or 1/2 (12) in (mm) airspace or to prime windows plus storm windows.
† Refer to manufacturer's literature for values.
‡ Thickness of each pane of glass, not thickness of assembled unit.
§ Refers to gray-, bronze-, and green-tinted heat-absorbing plate/float glass.

Table 12.4.22 Shading Coefficients for Single Glass with Indoor Shading by Venetian Blinds and Roller Shades

			Type of shading				
			Venetian blinds		Roller shade		
					Opaque		Translucent
Type of glass	Nominal thickness,* in (mm)	Solar trans.†	Medium	Light	Dark	White	Light
Regular sheet Regular plate/float Regular pattern Heat-absorbing pattern Gray sheet	3/32–1/4 (2–6) 1/4–1/2 (6–12) 1/8–9/32 (3–7) 1/8 (3) 3/16, 7/32 (5, 6)	0.87–0.80 0.80–0.71 0.87–0.79 0.74,0.71	0.64	0.55	0.59	0.25	0.39
Heat-absorbing plate/float‡ Heat-absorbing pattern Gray sheet	3/16, 1/4 (5, 6) 3/16, 1/4 (5, 6) 1/8, 7/32 (3, 5)	0.46 0.59,0.45	0.57	0.53	0.45	0.30	0.36
Heat-absorbing plate/float or pattern		0.44–0.30	0.54	0.52	0.40	0.28	0.32
Heat-absorbing plate/float‡	3/8 (10)	0.34					
Heat-absorbing plate or pattern		0.29–0.15 0.24	0.42	0.40	0.36	0.28	0.31
Reflective coated glass							
SC§ = 0.30			0.25	0.23			
0.40			0.33	0.29			
0.50			0.42	0.38			
0.60			0.50	0.44			

SOURCE: Abstracted from ASHRAE ''Handbook of Fundamentals,'' 1993, with permission.
* Refer to manufacturer's literature for values.
† For vertical blinds with opaque white and beige louvers in the tightly closed position. SC is 0.25 and 0.29 when used with glass of 0.71 to 0.80 transmittance.
‡ Refers to gray-, bronze-, and green-tinted heat-absorbing plate/float glass.
§ Shading coefficient for glass with no shading device.

Table 12.4.23 Shading Coefficients for Insulating Glass* with Indoor Shading by Venetian Blinds and Roller Shades

Type of glass	Nominal thickness, each light in (mm)	Solar trans†		Type of shading				
				Venetial blinds‡		Roller shade		
						Opaque		Translucent
		Outer pane	Inner pane	Medium	Light	Dark	White	Light
Regular sheet out, regular sheet in	3/32, 1/8 (2, 3)	0.87	0.87	0.57	0.51	0.60	0.25	0.37
Regular plate/float out,	1/4 (6)	0.80						
Regular plate/float in			0.80					
Heat absorbing plate/float§ out, regular plate/float in	1/4 (6)	0.46	0.80	0.39	0.36	0.40	0.22	0.30
Reflective coated glass								
SC¶ = 0.20				0.19	0.18			
0.30				0.27	0.26			
0.40				0.34	0.33			

SOURCE: Abstracted from ASHRAE ''Handbook of Fundamentals,'' 1993, with permission.
* Refers to factors-fabricated units with 3/16 (2), 1/4 (3), or 1/2 (12), in (mm) airspace, or to prime windows plus storm windows.
† Refer to manufacturer's literature for exact values.
‡ For vertical blinds with opaque white or beige louvers, tightly closed, SC is approximately the same as for opaque white roller shades.
§ Refers to bronze- or green-tinted heat-absorbing plate/flat glass.
¶ Shading coefficient for glass with no shading device.

Table 12.4.24 Shading Coefficients for Double Glazing with Between-Glass Shading

Type of glass	Nominal thickness, each pane	Solar trans.*		Description of airspace	Type of shading		
					Venetian blinds		Louvered sun screen
		Outer pane	Inner pane		Light	Medium	
Regular sheet out, regular sheet in	3/32, 1/8 (2, 3)	0.87	0.87	Shade in contact with glass or shade separated from glass by air space	0.33	0.36	0.43
Regular plate out, regular plate in	1/4 (6)	0.80	0.80	Shade in contact with glass void-filled with plastic			0.49
Heat-absorbing, plate/float† out,	1/4 (6)	0.46	0.80	Shade in contact with glass or shade separated from glass by air space	0.28	0.30	0.37
Regular plate in				Shade in contact with glass voids filled with plastic			0.41

SOURCE: Abstracted from ASHRAE ''Handbook of Fundamentals,'' 1993, with permission.
* Refer to manufacturer's literature for exact values.
† Refers to gray-, bronze-, and green-tinted heat-absorbing plate/float glass.

Table 12.4.25 Rates of Heat Gain from Occupants of Conditioned Spaces*†

Degree of activity	Typical application	Total heat adult male, Btu/h	Total heat adjusted,‡ Btu/h	Sensible heat, Btu/h	Latent heat, Btu/h
Seated at theater	Theater—Matinee	390	330	225	105
Seated at theater	Theater—Evening	390	350	245	105
Seated, very light work	Offices, hotels, apartments	450	400	245	155
Moderately active office work	Offices, hotels, apartments	475	450	250	200
Standing, light work; walking	Department store, retail store	550	450	250	200
Walking; standing	Drug store, bank	550	500	250	250
Sedentary work	Restaurant§	490	550	275	275
Light bench work	Factory	800	750	275	475
Moderate dancing	Dance hall	900	850	305	545
Walking 3 mi/h; light machine work	Factory	1,000	1,000	375	625
Bowling¶	Bowling alley	1,500	1,450	580	870
Heavy work	Factory	1,500	1,450	580	870
Heavy machine work; lifting	Factory	1,600	1,600	635	965
Athletics	Gymnasium	2,000	1,800	710	1,090

SOURCE: Abstracted from ASHRAE ''Handbook of Fundamentals,'' 1989, with permission.
* Tabulated values are based on 75°F room dry-bulb temperature. For 80°F room dry-bulb, the total heat remains the same, but the sensible heat values should be decreased by approximately 20%, and the latent heat values increased accordingly.
† All values are rounded to nearest 5 Btu/h.
‡ *Adjusted heat gain* is based on normal percentage of men, women, and children for the application listed, with the postulate that the gain from an adult female is 85% of that for an adult male, and that the gain from a child is 75% of that for an adult male.
§ Adjusted total heat gain for *Sedentary work, Restaurant*, includes 60 Btu/h for food per individual (30 Btu/h sensible and 30 Btu/h latent).
¶ For *Bowling*, figure one person per alley actually bowling, and all others as sitting (400 Btu/h) or standing and walking slowly (550 Btu/h).

Table 12.4.26 Rate of Heat Gain from Selected Office Equipment

Appliance	Size	Maximum input rating, Btu/h	Standby input rating, Btu/h	Recommended rate of heat gain, Btu/h
Cheek processing workstation	12 pockets	16,400	8,410	8,410
Computer devices				
Card puncher		2,730 to 6,140	2,200 to 4,800	2,200 to 4,800
Card reader		7,510	5,200	5,200
Communication/transmission		6,140 to 15,700	5,600 to 9,600	5,600 to 9,600
Disk drives/mass storage		3,410 to 34,100	3,412 to 22,420	3,412 to 22,420
Magnetic ink reader		3,280 to 16,000	2,600 to 14,400	2,600 to 14,400
Microcomputer	16 to 640 Kbyte*	340 to 2,050	300 to 1,800	300 to 1,800
Minicomputer		7,500 to 15,000	7,500 to 15,000	7,500 to 15,000
Optical reader		10,240 to 20,470	8,000 to 17,000	8,000 to 17,000
Plotters		256	128	214
Printers				
Letter quality	30 to 45 char/min	1,200	600	1,000
Line, high speed	5,000 or more lines/min	4,300 to 18,100	2,160 to 9,040	2,500 to 13,000
Line, low speed	300 to 600 lines/min	1,540	770	1,280
Tape drives		4,090 to 22,200	3,500 to 15,000	3,500 to 15,000
Terminal		310 to 680	270 to 600	270 to 600
Copiers/Duplicators				
Blue print		3,930 to 42,700	1,710 to 17,100	3,930 to 42,700
Copiers (large)	30 to 67* copies/min	5,800 to 22,500	3,070	5,800 to 22,500
Copiers (small)	6 to 30* copies/min	1,570 to 5,800	1,020 to 3,070	1,570 to 5,800
Feeder		100	—	100
Microfilm printer		1,540	—	1,540
Sorter/collator		200 to 2,050	—	200 to 2,050
Electronic equipment				
Cassette recorders/players		200	—	200
Receiver/tuner		340	—	340
Signal analyzer		90 to 2,220	—	90 to 2,220
Mail processing				
Folding machine		430	—	270
Inserting machine	3,600 to 6,800 pieces/h	2,050 to 11,300	—	1,330 to 7,340
Labeling machine	1,500 to 30,000 pieces/h	2,050 to 22,500	—	1,330 to 14,700
Postage meter		780	—	510
Word processors/typewriters				
Letter-quality printer	30 to 45 char/min	1,200	600	1,000
Phototypesetter		5,890	—	5,180
Typewriter		270	—	230
Word processor		340 to 2,050	—	300 to 1,800
Vending machines				
Cigarette		250	51 to 85	250
Cold food/beverage		3,920 to 6,550	—	1,960 to 3,280
Hot beverage		5,890	—	2,940
Snack		820 to 940	—	820 to 940
Miscellaneous				
Bar-code printer		1,500	—	1,260
Cash register		200	—	160
Coffee maker	10 cups	5,120	—	3,580 sensible 1,540 latent
Microfiche reader		290	—	290
Microfilm reader		1,770	—	1,770
Microfilm reader/printer		3,920	—	3,920
Microwave oven	1 ft^3	2,050	—	1,360
Paper shredder		850 to 10,240	—	680 to 8,250
Water cooler	32 qt/h	2,390	—	5,970

SOURCE: Abstracted from ASHRAE ''Handbook of Fundamentals,'' 1993, with permission.
* Input is not proportional to capacity.

Heat transfer to the distribution duct depends on the duct area exposed, the amount of insulation on the duct, and the temperature differential between the air in the duct and the surroundings.

Outside Air

Outside air loads depend on the amount of outside air intake and its condition. Table 12.4.3 shows recommended outdoor air ventilation rates for different occupancies. Tables 12.4.4 and 12.4.5 can be used to determine the quantity of outside air that will infiltrate because of wind effects. In all cases, sufficient outside air should be provided to balance mechanical exhaust systems.

$$\begin{aligned} Q_{outside\ air} &= q_{sensible} + q_{latent} \quad \text{Btu/h (watts)} \\ &= AU(t_o - t_i)\ \text{ft}^3/\text{min} \\ &\quad + 0.68\ (M_o - M_i)\ \text{ft}^3/\text{min} \quad \text{Btu/h} \end{aligned} \tag{12.4.16a}$$

where M_o = outside moisture content at design wet bulb, °F, gr/lb; and M_i = inside moisture content at design relative humidity, gr/lb. A comparable equation in SI units is

$$\begin{aligned} Q_{outside\ air} &= 1.2(t'_o - t'_i)\ \text{m}^3/s \\ &\quad + 42.6\ (M'_o - M'_i)\ \text{m}^3/\text{s} \quad \text{kW} \end{aligned} \tag{12.4.16b}$$

Table 12.4.27 Heat Gain from Restaurant Appliances
Not hooded,* gas burning and steam heated.

Appliance	Type of control	Miscellaneous data	Manufacturer's max. rating, Btu/h	Maintaining rate, Btu/h	Recommended heat gain from average use		
					Sensible heat, Btu/h	Latent heat, Btu/h	Total heat, Btu/h
Gas burning							
Coffee brewer ½ gal Warmer, ½ gal	Man. Man.	Combination brewer and warmer	3,400 500	 500	1,350 400	350 100	1,700 500
Coffee brewer unit with tank		4 brewers and 4½-gal tank			7,200	1,800	9,000
Coffee urn, 3 gal	Auto.	Black finish	3,200	3,900	2,900	2,900	5,800
Coffee urn, 3 gal	Auto.	Nickel plated		3,400	2,500	2,500	5,000
Coffee urn, 5 gal	Auto.	Nickel plated		4,700	3,900	3,900	7,800
Food warmer, values per sq. ft top surface	Man.	Water bath type	2,000	900	850	450	1,300
Fry kettle, 15 lb fat	Auto.	Frying area 10 × 10	14,250	3,000	4,200	2,800	7,000
Fry kettle, 28 lb fat	Auto.	Frying area 11 × 16	24,000	4,500	7,200	4,800	12,000
Grill, Broil-O-Grill Top burner Bottom burner	Man.	Insulated 22,000 Btu/h 15,500 Btu/h	37,000		14,400	3,600	18,000
Stoves, short order—closed top; values per sq. ft top surface	Man.	Ring-type burners 12,000 to 22,000 Btu each	14,000		4,200	4,200	8,400
Stoves, short order—closed top; values per sq. ft top surface	Man.	Ring-type burners 10,000 to 12,000 Btu each	11,000		3,300	3,300	6,600
Toaster, continuous	Auto.	2 slices wide, 360 slices/h	12,000	10,000	7,700	3,300	11,000
Steam heated							
Coffee urn, 3 gal 3 gal 5 gal	Auto. Auto. Auto.	Black finish Nickel plated Nickel plated			2,900 2,400 3,400	1,900 1,600 2,300	4,800 4,000 5,700
Coffee urn, 3 gal 3 gal 5 gal	Man. Man. Man.	Black finish Nickel plated Nickel plated			3,100 2,600 3,700	3,100 2,600 3,700	6,200 5,200 7,400
Food warmer, per sq. ft top surface	Auto.				400	500	900
Food warmer, per sq. ft top surface	Man.				450	1,150	1,500
Coffee brewer, ½ gal Warmer, ½ gal	Man. Man.		2,240 306	306 306	900 230	220 90	1,120 320
4 coffee brewing units with 4½-gal tank	Auto.	Water heater, 2,000 W Brewers, 2,960 W	16,900		4,800	1,200	6,000
Coffee urn, 3 gal 3 gal 5 gal	Man. Auto. Auto.	Black finish Nickel plated Nickel plated	11,900 15,300 17,000	3,000 2,600 3,600	2,600 2,200 3,400	1,700 1,500 2,300	4,300 3,700 5,700
Doughnut machine	Auto.	Exhaust system to outdoors, ½-hp motor	16,000		5,000		5,000
Egg boiler	Man.	Med. ht., 550 W Low ht., 275 W	3,740		1,200	800	2,000
Food warmer with plate warmer, per sq. ft top surface	Auto.	Insulated, separate heating unit for each pot, plate warmer in base	1,350	500	350	350	700
Food warmer without plate warmer, per sq. ft top surface	Auto.	Insulated, separate heating unit for each pot	1,020	400	200	350	550
Fry kettle, 11½ lb fat	Auto.		8,840	1,100	1,600	2,400	4,000
Fry kettle, 25 lb fat	Auto.	Frying area 12 × 14 in	23,800	2,000	3,800	5,700	9,500
Griddle, frying	Auto.	Frying top 18 × 14 in	8,000	2,800	3,100	1,700	4,800
Grille, meat	Auto.	Cooking area 10 × 12 in	10,200	1,900	3,900	2,100	6,000
Grille, sandwich	Auto.	Grill area 12 × 12 in	5,600	1,900	2,700	700	3,400
Roll warmer	Auto.	One drawer	1,500	400	1,100	100	1,200
Toaster, continuous	Auto.	2 slices wide, 360 slices/h	7,500	5,000	5,100	1,300	6,400
Toaster, continuous	Auto.	4 slices wide, 720 slices/h	10,200	6,000	6,100	2,600	8,700
Toaster, pop-up	Auto.	2 slices	4,150	1,000	2,450	450	2,900
Waffle iron	Auto.	One waffle 7-in diam.	2,480	600	1,100	750	1,850
Waffle iron for ice cream sandwich	Auto.	12 cakes, each 2½ × 3¾ in	7,500	1,500	3,100	2,100	5,200

SOURCE: Carrier manual.
* If properly designed positive exhaust hood is used, multiple recommended value by 0.50.

Table 12.4.28 Heat Gain from Typical Electric Motors

Motor nameplate or rated horsepower	Motor type	Nominal r/min	Full load motor efficiency, %	Location of motor and driven equipment with respect to conditioned space or airstream: A — Motor in, driven equipment in, Btu/h	B — Motor out, driven equipment in, Btu/h	C — Motor in, driven equipment out, Btu/h
0.05	Shaded pole	1,500	35	360	130	240
0.08	Shaded pole	1,500	35	580	200	380
0.125	Shaded pole	1,500	35	900	320	590
0.16	Shaded pole	1,500	35	1,160	400	760
0.25	Split phase	1,750	54	1,180	640	540
0.33	Split phase	1,750	56	1,500	840	660
0.50	Split phase	1,750	60	2,120	1,270	850
0.75	3-phase	1,750	72	2,650	1,900	740
1	3-phase	1,750	75	3,390	2,550	850
1.5	3-phase	1,750	77	4,960	3,820	1,140
2	3-phase	1,750	79	6,440	5,090	1,350
3	3-phase	1,750	81	9,430	7,640	1,790
5	3-phase	1,750	82	15,500	12,700	2,790
7.5	3-phase	1,750	84	22,700	19,100	3,640
10	3-phase	1,750	85	29,900	24,500	4,490
15	3-phase	1,750	86	44,400	38,200	6,210
20	3-phase	1,750	87	58,500	50,900	7,610
25	3-phase	1,750	88	72,300	63,600	8,680
30	3-phase	1,750	89	85,700	76,300	9,440
40	3-phase	1,750	89	114,000	102,000	12,600
50	3-phase	1,750	89	143,000	127,000	15,700
60	3-phase	1,750	89	172,000	153,000	18,900
75	3-phase	1,750	90	212,000	191,000	21,200
100	3-phase	1,750	90	283,000	255,000	28,300
125	3-phase	1,750	90	353,000	318,000	35,300
150	3-phase	1,750	91	420,000	382,000	37,800
200	3-phase	1,750	91	569,000	509,000	50,300
250	3-phase	1,750	91	699,000	636,000	62,900

SOURCE: Adapted from ASHRAE ''Handbook of Fundamentals,'' 1993, with permission.

where M'_o = outside moisture content at design wet bulb, °C, g/kg; and M'_i = inside moisture content at design relative humidity, g/kg. t'_o, t'_i are expressed in °C.

Moisture Load

Moisture as a cooling load is a latent heat gain expressed in BTU per hour and is computed with the following equations:

$$Q_t = Q_{inf} + Q_{vent} \quad (12.4.17)$$

$$Q_{inf} = 0.68 \times \text{ft}^3/\text{min} \times (M_o - M_i) \quad \text{Btu/h} \quad (12.4.18a)$$

$$Q_{vent} = 0.68 \times \text{ft}^3/\text{min} \times (M_o - M_i) \quad \text{Btu/h} \quad (12.4.19a)$$

where 0.68 = 60/13.5 × 1,076/7,000; 60 = min/h; 13.5 = specific volume of moist air at 70°F dB and 50% RH; 1,076 = average heat removal required to condense 1 lb of water vapor from the room air; and 7,000 = grains per pound.

Comparable equations in SI units are:

$$Q_{inf} = 42.6\ \text{m}^3/\text{s}\ (M'_o - M'_i) \quad \text{kW} \quad (12.4.18b)$$

$$Q_{vent} = 42.6\ \text{m}^3/\text{s}\ (M'_o - M'_i) \quad \text{kW} \quad (12.4.19b)$$

M'_o = moisture content of outside air, g/kg; and M'_i = moisture content of inside air, g/kg.

HEATING

Heating Load

$$Q_t = Q_{tr} + Q_{inf} + Q_{vent} \quad (12.4.20a)$$

where Q_1 = total heating load, Btu/h; Q_{tr} = transmission load = $AU(t_i - t_o)$, Btu/h; Q_{inf} = infiltration load = 1.08 (ft³/min)$(t_i - t_o)$, Btu/h; Q_{vent} = ventilation load = 1.08 (ft³/min)$(t_i - t_o)$, Btu/h; A = area through which heat flow occurs, ft²; U = overall heat-transfer coefficient = $1/R_t$ Btu/(h)(ft²)(°F); t_o = outside-air design dry-bulb temperature, °F (Fig. 12.4.1); t_i = inside design dry-bulb temperature, °F (Table 12.4.1); R_t = thermal resistance, °F/(Btu·h·ft²) = $R_1 + R_2 + R_3 + \cdots + R_n$; and R_1, R_2, . . . = resistances to heat flow to the individual components of a composite construction. See Fig. 12.4.2 and Table 12.4.2.

A comparable equation in SI units is

$$Q_1 = Q_{tr} + Q_{inf} + Q_{vent} \quad (12.4.20b)$$

where Q_t = total heating load, kW; $Q_{tr} = AU(t_i - t_o)$, kW; Q_{inf} = 1.2 (m³/s)$(t_i - t_o)$, kW; Q_{vent} = 1.2 (m³/s)$(t_i - t_o)$, kW; A = area through which heat flow occurs, m²; U = overall heat-transfer coefficient = $1/R_t$, W/(m²·K); t_o = outside-air design dry-bulb temperature, °C; t_i = inside design dry-bulb temperature, °C; m³/s = cubic meters per second, air; and R_t = thermal resistance, K/(W·m²).

PSYCHROMETRICS

The psychrometric chart can be used to simplify the analysis of air-conditioning processes. The processes can be shown simply and visualized clearly, and the chart lends itself to quick graphical computations. In addition, the state of any air–water vapor mixture can be fixed on the psychrometric chart with reasonable accuracy by means of any two psychrometric characteristics. Usually dry-bulb temperature and wet-bulb temperature are used because they are the simplest to measure.

Figures 12.4.8 and 12.4.9 are psychrometric charts for normal temperatures in inch-pound and SI units, respectively.

Psychrometric charts at different temperatures and barometric pressures are useful for problems falling outside the ranges shown in Figs. 12.4.8 and 12.4.9. A collection of different psychrometric charts, in both USCS and SI units, for low, normal, and high temperatures, at sea

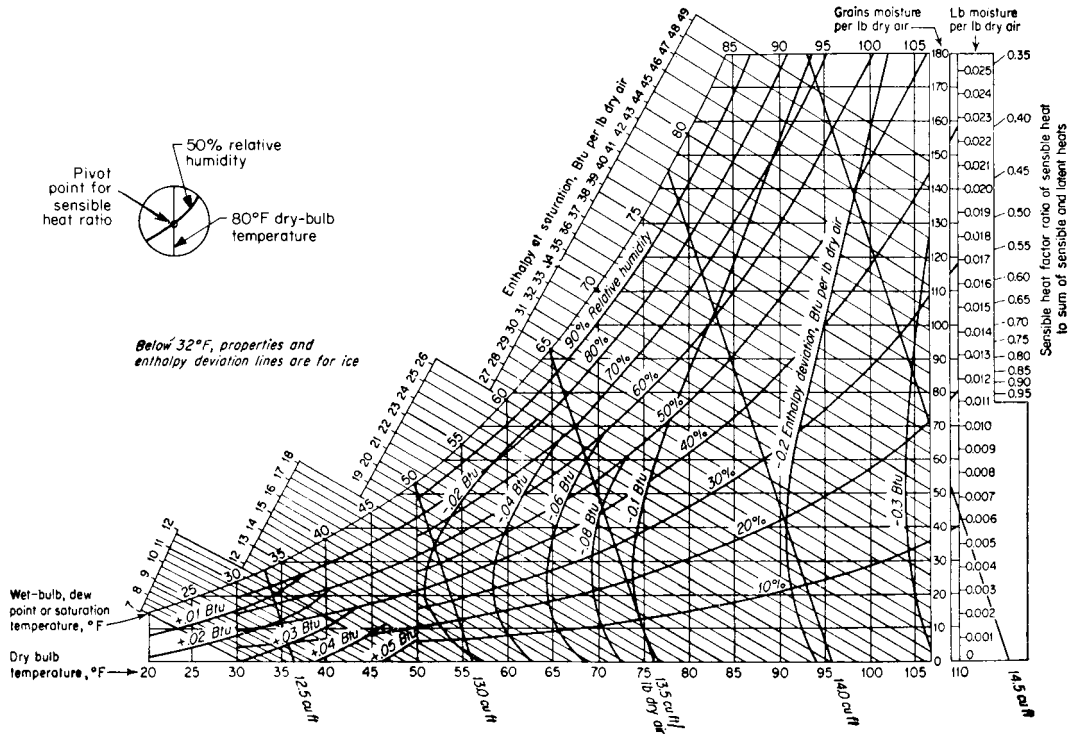


Fig. 12.4.8 Psychrometric chart for normal temperatures in USCS units. *(Carrier Corp.)*

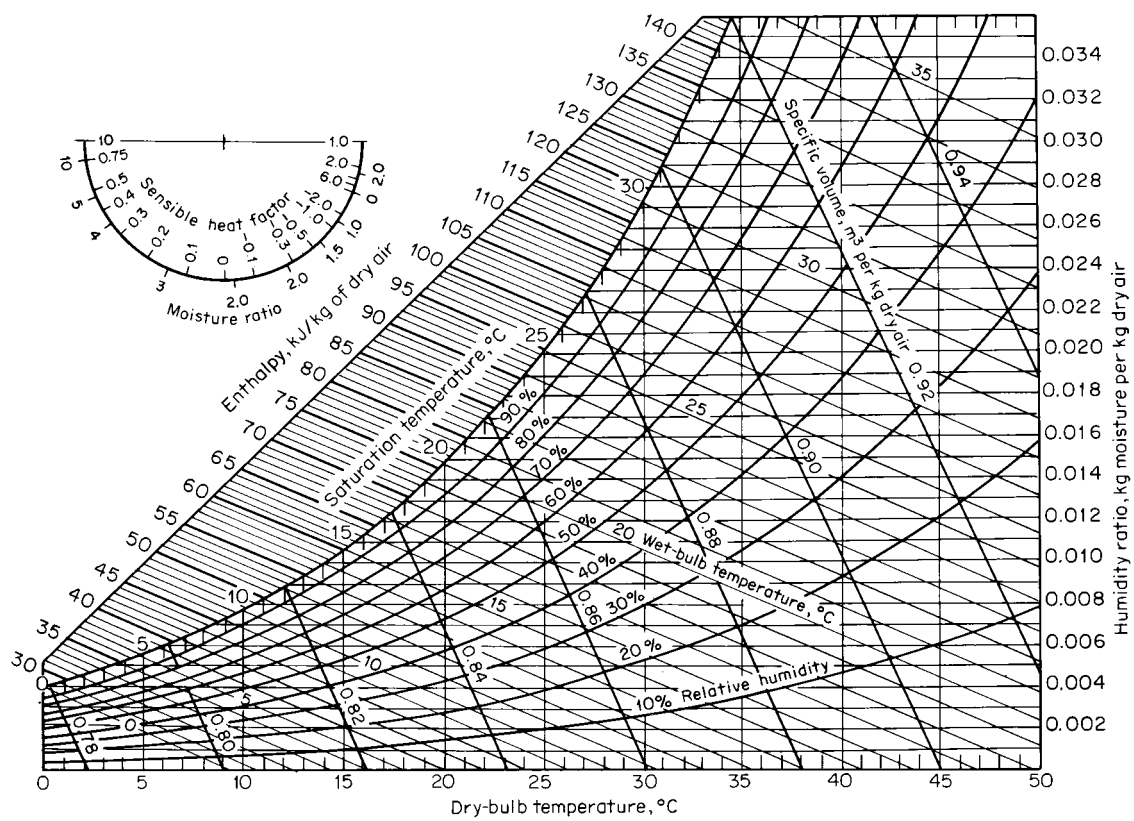


Fig. 12.4.9 Psychrometric chart for normal temperatures in SI units. *(Adapted from material published by Business News Publishing Co., 1975.)*

level and at four elevations above sea level, are available from ASHRAE, the Carrier Corp. of Syracuse, NY, and from other equipment manufacturers.

Figure 12.4.10 shows a psychrometric chart simplified in skeleton form.

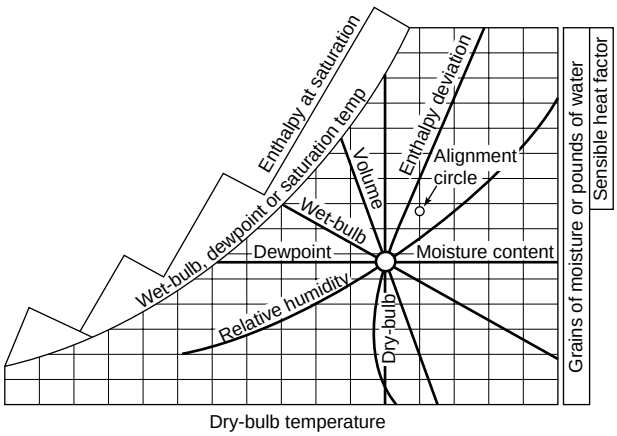


Fig. 12.4.10 Skeleton psychrometric chart.

Terminology

Dry-bulb temperature: The temperature of air as registered by an ordinary thermometer.

Wet-bulb temperature: The temperature registered by a thermometer whose bulb is covered by a wetted wick and exposed to a current of rapidly moving air.

Dew-point temperature: The temperature at which condensation of moisture begins when the air is cooled.

Relative humidity: Ratio of the actual water vapor pressure of the air to the saturated water vapor pressure of the air at the same temperature.

Specific humidity or moisture content: The weight of water vapor in grains (or pounds) of moisture per pound of dry air.

Enthalpy: A thermal property indicating the quantity of heat in the air above an arbitrary datum, in Btu per pound of dry air. The datum for dry air is 0°F and, for the moisture content, 32°F water.

Enthalpy deviation: Enthalpy should be corrected by the enthalpy deviation due to the air not being in the saturated state. On normal air conditioning estimates it is omitted.

Specific volume: The cubic feet of the mixture per pound of dry air.

Sensible heat factor: The ratio of sensible to total heat.

Alignment circle: Used in conjunction with the sensible heat factor to plot the various air-conditioning process lines.

Pounds of dry air: The basis for all psychrometric calculations. Remains constant during all psychrometric processes.

The dry-bulb, wet-bulb, and dew-point temperatures and the relative humidity are so related that, if two properties are known, all other properties shown may then be determined. When **air is saturated,** dry-bulb, wet-bulb, and dew-point temperatures are all equal.

Room sensible heat factor is the ratio of room sensible heat to the total of room sensible and room latent heat.

A line drawn between the room design condition at the slope of the sensible heat factor is the **condition line** and it will intersect the saturation line at the apparatus dew point. See Fig. 12.4.11. In order to maintain design conditions, supply air must be provided to the space at a temperature and moisture level that, when plotted, must fall on the condition line. If air is provided at conditions of apparatus dew point, the minimum quantity of conditioned air would be required to obtain design conditions.

Supply Air Temperature The room cooling load is the sum of external and internal sensible and latent heat gains plus the difference in enthalpy between outside and room air for that portion of outside air which does not contact the cooling-coil surfaces. The percentage of air that passes through a cooling coil untreated is the numerical value of the *coil bypass factor;* e.g., a bypass factor of 20 percent represents a cooling-coil saturation efficiency of 80 percent.

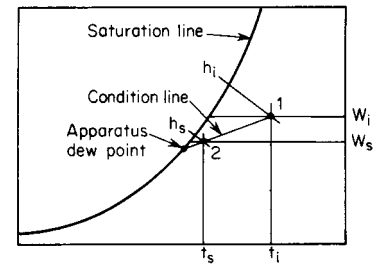


Fig. 12.4.11 Apparatus dew point and condition line.

The ratio of room sensible heat gains to total room sensible and latent heat gains is the **room sensible heat ratio** (RSHR).

$$\text{RSHR} = \frac{Q_{rs}}{Q_{rs} + Q_{rl}} \tag{12.4.21}$$

It represents the ratio of sensible cooling capacity to the total cooling capacity required of the supply air to satisfy room conditions. It is used to plot the slope of the *room-condition line* on a psychometric chart for the determination of the *apparatus dew point* (ADP). See Fig. 12.4.11.

The actual supply air temperature and off coil wet-bulb temperature will depend on the bypass characteristic of the selected cooling coil (Fig. 12.4.11).

Supply Air Rate The rate of supply air required is expressed by

$$Q_{sa} = Q_{rs}/1.08(t_r - t_s) \qquad \text{ft}^3/\text{min} \tag{12.4.22a}$$

In SI units:

$$Q'_{sa} = Q'_{rs}/1.2(t_r - t_s) \qquad \text{m}^3/\text{s} \tag{12.4.22b}$$

where Q_{sa} = supply air, ft³/min (m³/s); Q'_{rs} = room sensible heat, kW; t_r = room design temperature, °F (K); t_s = supply air temperature, °F (K).

The bypass factor is used to express the effectiveness of a cooling coil. It is based on the artificial premise that a portion of the air that enters a cooling coil is cooled completely to saturation, that is, to the apparatus dew point, and the remaining air is not cooled at all. The actual condition of the air that leaves the coil is then represented to be a mixture of the fully conditioned and completely unconditioned air. The ratio of unconditioned air to total air is the **bypass factor**.

The bypass factor is a function of the type and amount of coil surface and the time available for contact as the air passes over the coil. Table 12.4.29 gives approximate bypass factors for various finned coil surfaces and air velocities. A larger bypass factor requires the circulation of more conditioned air; where outside air is introduced, the bypassed air increases the room air load.

Table 12.4.29 Typical Bypass Factors for Unsprayed Coils

Depth of coils (rows)	5-in spacing	
	8 fins/in	14 fins/in
	Face velocity, ft/min	
	300–700	300–700
2	0.42–0.55	0.22–0.38
3	0.27–0.40	0.10–0.23
4	0.19–0.30	0.05–0.14
5	0.12–0.25	0.02–0.09
6	0.08–0.18	0.01–0.06
8	0.03–0.08	

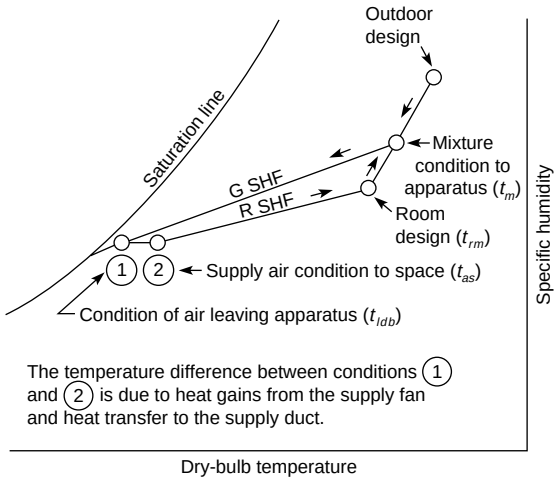


Fig. 12.4.12 Typical cooling process.

Typical Air-Conditioning Cooling Process Figure 12.4.12 depicts a typical cooling process adjusted for fan heating and duct losses.

DUCT DESIGN

Air Velocity Supply and return air ducts and apparatus are sized on the basis of air quantity and are kept within the limitations of allowable friction losses, velocity, and noise. Conveying air at low velocity will be quieter and require less pressure, less fan power and less carefully constructed ductwork than conveying air at medium or high velocity. Lower velocities are preferred where space is available.

Pressure Loss in Duct Systems Pressure losses in duct systems are due to friction of the air in contact with the sides of the duct and dynamic losses caused by changes of duct shape or direction and by obstructions to flow.

$$\text{Friction } H_f = f\frac{L}{D}\left(\frac{V}{4{,}005}\right)^2 \qquad (12.4.23a)$$

where H_f = head loss due to friction, in H_2O: L = length of duct, ft; D = diameter of duct, ft; V = velocity of air, ft/min; and f = nondimensional friction coefficient. A comparable equation in SI units is

$$H_f = f\frac{L}{d}\frac{(V)^2}{2}\rho \qquad (12.4.23b)$$

where H_f = head loss due to friction, N/m^2; L = length of duct, m; D = diameter of duct, m; V = velocity of air, m/s; f = nondimensional friction coefficient; and ρ = density of air, kg/m^3.

$$\text{Dynamic losses } H_v = \text{C}\left(\frac{V}{4{,}005}\right)^2 \qquad (12.4.24a)$$

where H_v = velocity-head loss, in H_2O; C = experimentally determined constant; $V = Q/A$ = air velocity, ft/min; Q = airflow rate, ft^3/min; and A = cross-sectional area of duct, ft^2. A comparable equation in SI units is

$$H'_v = C'\frac{(V')^2}{2} \qquad (12.4.24b)$$

where H'_v = velocity-head loss, N/m^2; V' = air velocity, m/s; and = density of air, kg/m^3. When the equation for H'_v is written in the form above, $C' = C$.

Duct Design Methods Two methods are frequently used: **equal friction** and **static regain**.

Equal Friction The equal friction method is applicable primarily to systems using low or moderate velocities, where the velocity head is not an important factor. A friction drop per 100 ft of length is chosen, and the duct mains and branches are all sized on the basis of this friction drop. This will invariably result in higher velocities in the mains, where they can be tolerated, and low velocities in the branches, where they are desirable. Limitations are established for friction loss per 100 feet and for velocities in the mains and branches.

Static Regain The static regain method is used for both conventional and high velocity systems. It is especially applicable in the latter, where the velocity head may be appreciable. In the static regain method, the static pressure required to give proper airflow through the system outlets is determined, and this pressure is maintained by reducing the velocity at each branch or takeoff, so that the recovery in pressure due to reduction of velocity balances the friction loss in the preceding section of duct. This is possible because of the convertibility of static and velocity pressures. For practical applications it is usually assumed that 50 percent of the velocity pressure available will be converted to static pressure.

$$H_R = 0.5\left(\frac{V_1}{4{,}005}\right)^2 - \left(\frac{V_2}{4{,}005}\right)^2 \qquad (12.4.25a)$$

where H_R = head recovered, in H_2O; V_1 = system inlet velocity, ft/min; and V_2 = system outlet velocity, ft/min. A comparable equation in SI units is

$$H'_R = \frac{0.5(V'_1)^2}{2} - \frac{(V'_2)^2}{2} \qquad (12.4.25b)$$

where $H_{R'}$ = head recovered, N/m^2; V'_1 = system inlet velocity, m/s; and V'_2 = system outlet velocity, m/s.

Figure 12.4.13 may be used to determine directly the friction loss per

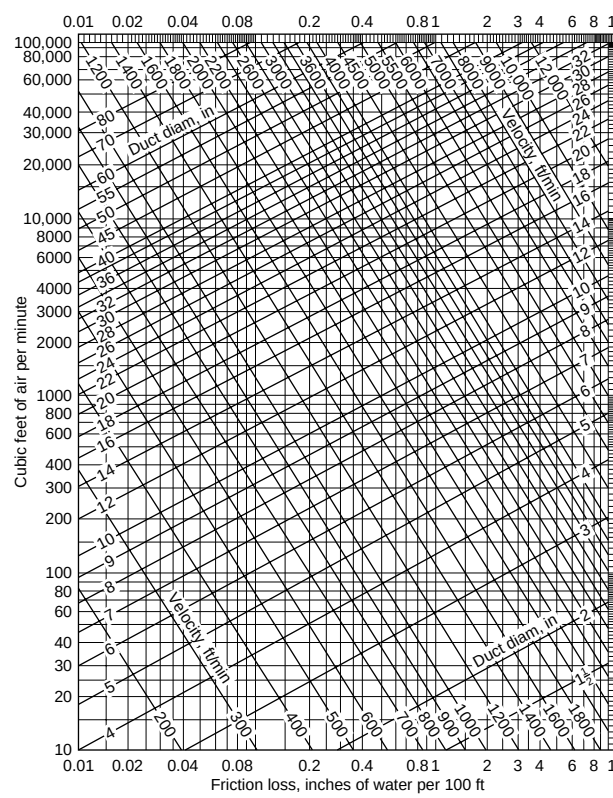


Fig. 12.4.13 Friction loss for usual conditions in round ducts. (This chart applies to smooth, round, galvanized steel ducts, and is based on air at 70°F and 29.96 in Hg absolute pressure. For air of different density, the friction may be assumed to vary directly with the density.)

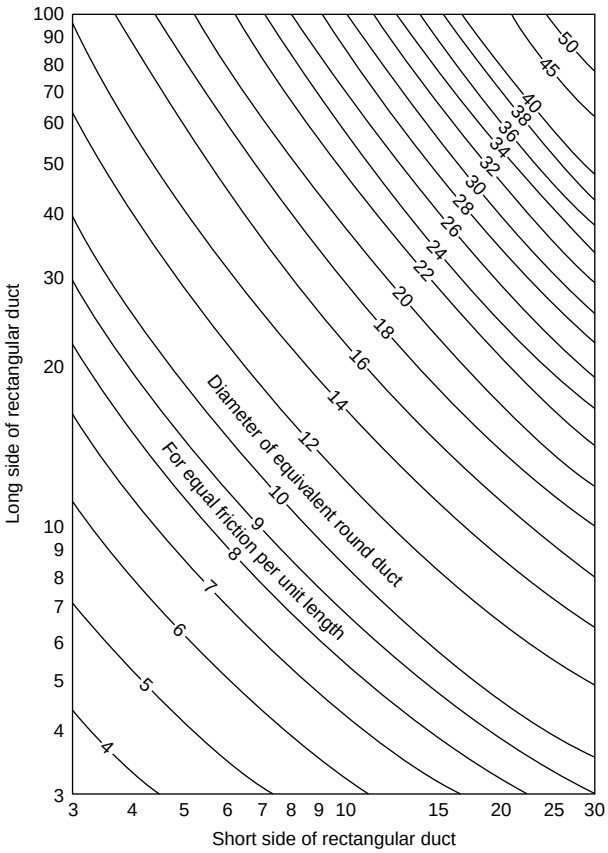


Fig. 12.4.14 Equivalent diameters of round ducts. Use with rectangular ducts, with capacity constant. *(Buffalo Forge.)*

100 ft for round ducts. Figure 12.4.14 provides a conversion from round to rectangular ductwork with the same air capacity. Table 12.4.30 presents a range of friction loss per 100 ft and velocity limits for low- and high-velocity air systems. Typical velocities through some air system components are shown in Table 12.4.31.

Table 12.4.30 Duct System Pressure Losses and Design Velocities

	Pressure loss, in H_2O, gage, per 100 ft	Maximum velocity, ft/min
Low-velocity systems		
Noise critical	0.05–0.07	1,400
Usual design	0.08–0.1	1,800
Straight runs without takeoffs	0.08–0.12	2,400
High-velocity systems		
Usual design	0.4–0.6	2,600
Straight runs without takeoffs	0.6	3,400

Air Distribution

Air outlets are designed to control air motion, noise level, and temperature gradients caused by the introduction of air into a conditioned space and the removal of air from that space.

Supply Outlets Supply outlets (and diffusers) are usually located at near ceiling height with grilles or registers high on the sidewall. Supply outlets may be located below windows or in the floor as long as they satisfy the criteria for the space and are unobstructed. Factors which usually affect the selection of supply outlets are noise, location of outlet, temperature of supply air, and area of diffusion. Manufacturers' performance data for supply outlets should be used as the basis for selection.

Table 12.4.31 Typical Velocities through Air System Components

	Residences		Nonresidential buildings	
Designation	ft/min	m/min	ft/min	m/min
Outdoor-air intakes*	300	80	500	150
Filters*	250	75	300	90
Heating coils*	450	140	500	150
Cooling coils*	450	140	500	150
Air washers	500	150	500	150

SOURCE: Adapted from ASHRAE ''Handbook of Fundamentals,'' 1993, with permission.

* These velocities are for total face area, not the net free area; other velocities in the table are for net free area.

Return Outlets Selection of return outlets (and registers or grilles) is usually governed by face velocity to limit discomfort to occupants from drafts and/or noise. See Table 12.4.32.

Table 12.4.32 Recommended Return Face Velocities

	Velocity over gross area	
Intake location	ft/min	m/min
Above occupied zone	800 up	245 up
Within occupied zone, not near seats	600–800	185–245
Within occupied zone, near seats	400–600	120–185
Door or wall louvers	200–300	60–90
Undercutting of doors (through undercut area)	200–300	60–90

SOURCE: Adapted from ASHRAE, ''Handbook of Fundamentals,'' 1993, with permission.

FANS

Supply and **return fans** must convey the conditioned air through the system efficiently and with a minimum of disturbing noise and vibrations.

Fans produce pressure by increasing airflow velocity. Velocity change is the result of tangential and radial velocity components in the case of centrifugal fans, and of axial and tangential velocity components in the case of axial-flow fans.

Centrifugal fans are most frequently used, but **axial-flow fans** have been applied successfully. Figures 12.4.15 to 12.4.17 show typical performance curves for forward-curved and backward-curved centrifugal fans

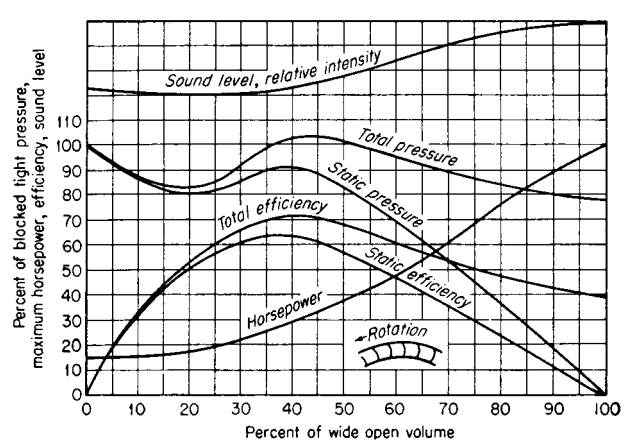


Fig. 12.4.15 Percentage performance curves for a forward-curved blade centrifugal fan.

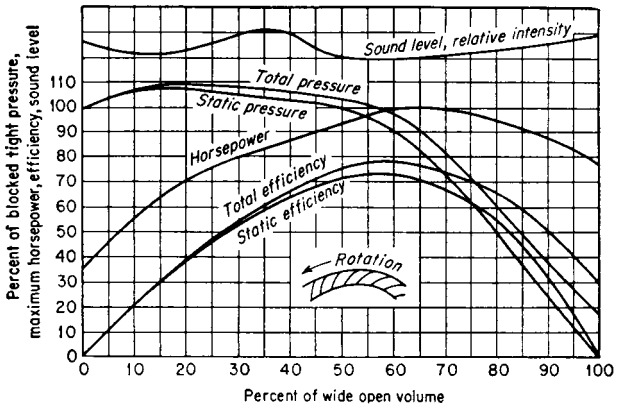


Fig. 12.4.16 Percentage performance curves for a backward-curved blade centrifugal fan.

and axial-flow fans. Table 12.4.33 summarizes the relative characteristics of centrifugal fans. Figure 12.4.18 shows the zone of optimum performance for a backward-curved-blade centrifugal fan.

FAN LAWS

Fan laws in Table 12.4.34 relate the performance variables for any dynamically similar series of fans. (See also Sec. 14.5.) They can be

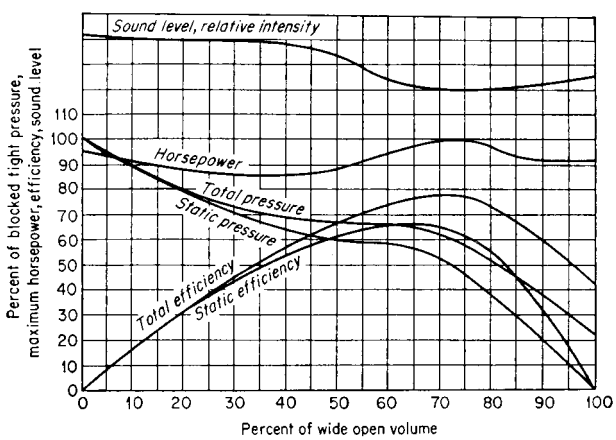


Fig. 12.4.17 Percentage performance curves for an axial fan.

used to predict the performance of any fan in a series when test data are available for a given fan in the series. Fan laws may also be used with a particular fan to determine the effect of speed change. Unless otherwise identified, fan performance data are based on dry air at standard conditions: 14.696 lb/in^2 abs and 70°F (0.075 lb/ft^3). In applications where the fan may be required to handle air or gas at some other density, because of temperature or altitude, fan performance will be affected. Table 12.4.35 shows correction factors to air volume for altitude and temperature.

Table 12.4.33 Relative Characteristics of Centrifugal Fans

Characteristic	Backward curved	Forward curved
First cost	High	Low
Efficiency	High	Low
Stability of operation	Good	Poor
Space required	Medium	Small
Tip speed	High	Low

Testing and Balancing Each fan and air duct system component should be tested, adjusted, and balanced to assure that it conforms to the design requirements.

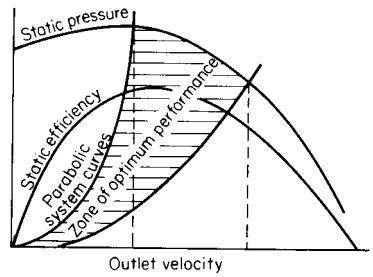


Fig. 12.4.18 Zone of optimum performance for backward-curved blade centrifugal fans.

System and Duct Noise The major sources of noise from air-conditioning systems are fans, diffusers, grilles, ducts, fittings, and vibration. **Sound control** for components consists of selecting devices that meet the design goal under all operating conditions and installing them properly so that no additional sound is generated. The sound power

Table 12.4.34 Fan Laws*†
For all fan laws: $N_{t1} = N_{t2}$ and (Pt. of Rtg.)$_1$ = (Pt. of Rtg.)$_2$

No.	Dependent variables	Independent variables		
1a	$Q_1 = Q_2$	$\times \left(\frac{D_1}{D_2}\right)^3$	$\times \frac{N_1}{N_2}$	$\times 1$
1b	Press.$_1$ = Press.$_2$‡	$\times \left(\frac{D_1}{D_2}\right)^2$	$\times \left(\frac{N_1}{N_2}\right)^2$	$\times \frac{\rho_1}{\rho_2}$
1c	$H_1 = H_2$	$\times \left(\frac{D_1}{D_2}\right)^3$	$\times \left(\frac{N_1}{N_2}\right)^3$	$\times \frac{\rho_1}{\rho_2}$
2a	$Q_1 = Q_2$	$\times \left(\frac{D_1}{D_2}\right)^2$	$\times \left(\frac{\text{Press.}_1}{\text{Press.}_2}\right)^{1/2}$	$\times \left(\frac{\rho_2}{\rho_1}\right)^{1/2}$
2b	$N_1 = N_2$	$\times \left(\frac{D_2}{D_1}\right)$	$\times \left(\frac{\text{Press.}_1}{\text{Press.}_2}\right)^{1/2}$	$\times \left(\frac{\rho_2}{\rho_1}\right)^{1/2}$
2c	$H_1 = H_2$	$\times \left(\frac{D_1}{D_2}\right)^2$	$\times \left(\frac{\text{Press.}_1}{\text{Press.}_2}\right)^{3/2}$	$\times \left(\frac{\rho_2}{\rho_1}\right)^{1/2}$
3a	$N_1 = N_2$	$\times \left(\frac{D_2}{D_1}\right)^3$	$\times \frac{Q_1}{Q_2}$	$\times 1$
3b	Press.$_1$ = Press.$_2$	$\times \left(\frac{D_2}{D_1}\right)^4$	$\times \left(\frac{Q_1}{Q_2}\right)^2$	$\times \frac{\rho_1}{\rho_2}$
3c	$H_1 = H_2$	$\times \left(\frac{D_2}{D_1}\right)^4$	$\times \left(\frac{Q_1}{Q_2}\right)^3$	$\times \frac{\rho_1}{\rho_2}$

NOTE: D = fan size, N = revolutions per minute, ρ = gas density, Q = volume flow rate, P = pressure, and H = horsepower.
* The subscript 1 denotes that the variable is for the fan under consideration.
† The subscript 2 denotes that the variable is for the tested fan.
‡ P_{total} or P_{static}.

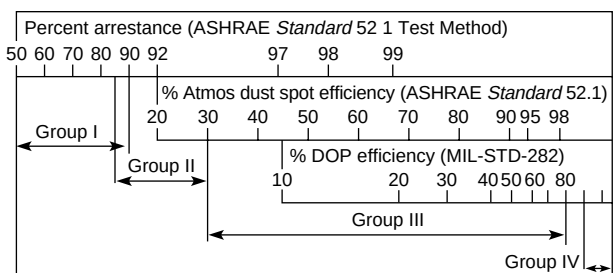
Table 12.4.35 Air Volume Correction Factors for Altitude and Temperature

Altitude, ft (m) above sea level	0	1,000 (305)	2,000 (610)	3,000 (915)	4,000 (1,220)	5,000 (1,525)	6,000 (1,880)	7,000 (2,135)	8,000 (2,440)
Barometric pressure, in Hg (kN/m^2)	29.92 (101.3)	28.86 (97.7)	27.82 (94.2)	26.81 (90.8)	25.84 (87.5)	24.89 (84.3)	23.98 (81.2)	23.09 (78.2)	22.2 (75.2)
Air temp, °F (°C)	Correction factors								
70 (21.1)	1.040	1.003	0.967	0.932	0.898	0.865	0.833	0.803	0.772
100 (38)	0.984	0.948	0.915	0.882	0.850	0.818	0.788	0.759	0.731
150 (66)	0.904	0.872	0.840	0.801	0.781	0.752	0.724	0.698	0.672
200 (93)	0.835	0.805	0.777	0.749	0.722	0.694	0.668	0.645	0.620
250 (121)	0.777	0.749	0.722	0.696	0.671	0.647	0.622	0.599	0.577
300 (149)	0.725	0.699	0.674	0.649	0.628	0.603	0.580	0.560	0.538
350 (177)	0.680	0.656	0.632	0.609	0.588	0.566	0.545	0.525	0.505
400 (204)	0.641	0.618	0.596	0.574	0.553	0.533	0.512	0.495	0.476
450 (232)	0.605	0.583	0.564	0.543	0.523	0.503	0.485	0.467	0.450
500 (260)	0.574	0.553	0.534	0.515	0.496	0.477	0.460	0.443	0.426
550 (288)	0.546	0.526	0.508	0.490	0.472	0.454	0.438	0.421	0.406
660 (316)	0.520	0.501	0.484	0.466	0.449	0.433	0.416	0.401	0.387
650 (343)	0.496	0.478	0.462	0.444	0.428	0.413	0.397	0.383	0.368
700 (371)	0.475	0.458	0.442	0.426	0.411	0.395	0.381	0.367	0.354

SOURCE: ''Bulletin 3576-B, Correction Factors for Temperature and Altitude,'' Buffalo Forge Co., Buffalo, NY.

NOTE: Equivalent = $\frac{\text{ft}^3 \text{ min at actual conditions}}{\text{correction factor}}$.

output of a fan is determined by the type of fan, airflow, and pressure. Sound control in the duct system requires proper duct layout, sizing, and provision for installing duct noise attenuators, if required. The noise generated in a system increases with both duct velocity and system pressure. The ASHRAE Handbook ''Applications'' and other related sources present methods for sound control in air-conditioning systems.



Group I
Panel-type filters of spun glass, open cell foams, expanded metal and screens, synthetics, textile denier woven and nonwoven, or animal hair.

Group II
Pleated panel-type filters of fine denier nonwoven synthetic and synthetic-natural fiber blends, or all natural fiber.

Group III
Extended surface supported and nonsupported filters of fine glass fibers, fine electret synthetic fibers, or wet-laid paper of cellulose-glass or all-glass fibers.

Group IV
Extended-area pleated HEPA-type filters of wet-laid ultra-fine glass fiber paper. Biological grade air filters are generally 95% DOP efficiency; HEPA filters are 99.97 and 99.99%; and ULPA filters are 99.999%.

Notes:
1. Group numbers have no significance other than their use in this figure.
2. Correlation between the test methods shown are approximations for general guidance only.

Fig. 12.4.19 Comparative performance of viscous impingement and dry media filters. *(ASHRAE ''Handbook of Fundamentals,'' 1993.)*

FILTRATION

Filters are provided in air-conditioning systems to assure that the conditioned air will satisfy space cleanliness criteria and to protect air treatment equipment from fouling due to dirt accumulation. Figure 12.4.19 describes types of dry media filters and shows their range of performance. In addition, it shows an approximate comparison of the filtration efficiencies obtained by the use of three different testing methods.

The filters designated as Group I are usually used in least critical applications and as a prefilter to extend the operating life of more costly, higher-efficiency filters. Filters designated as Groups II and III are used in good-quality comfort and process air-conditioning applications. The very high efficiency HEPA and ULPA filters are costly and result in high pressure losses. They are applied in clean rooms and for nuclear and toxic particulate filtration.

HEAT REJECTION APPARATUS

All the heat removed from a conditioned space and the heat equivalent of the work performed by the refrigeration equipment must be removed through the cooling system and dissipated via the refrigeration condenser. Table 12.4.36 lists approximate refrigeration equipment energy consumption values.

In most systems the heat is rejected to the atmosphere, but in some cases, water-cooled refrigeration condensers are used to transfer the rejected heat to surface water or to well water. Some examples of the most common systems used to transfer rejected heat to the atmosphere are shown in Fig. 12.4.20. The **air-cooled condenser and evaporative condenser** are arranged with the refrigeration condenser in the outdoor airstream. Refrigerant is piped directly with no intervening heat transfer. The **air-cooled condenser** is the simplest arrangement, but the condenser must operate at temperatures sufficiently high so that condensing takes place well above the outdoor ambient dry-bulb temperature. The **evaporative condenser** adds a spray system to permit condensing above the outdoor ambient wet-bulb temperature.

Closed-Circuit Coolers For larger systems and where extensive

piping of refrigerant is either not desired or otherwise precluded, a water-cooled condenser is utilized. The rejected heat is transferred to water piped to an outdoor radiator—a closed-circuit dry cooler or evaporatively cooled device. The **closed-circuit dry cooler,** or **radiator,** is a simple indirect cooling arrangement, but can operate at temperatures higher than the air-cooled condenser because of the additional heat-transfer process involved.

Table 12.4.36 Refrigeration Energy Consumption per 12,000 Btu/h Cooling Capacity

Electric consumption, kW/ton cooling capacity		
	Type of heat rejection, kW/ton	
Item	Mechanical-draft cooling tower	Air-cooled condensers
Room-air conditioners		1.2–2.45*
Refrigeration compressors (motor-driven):		
Reciprocating (20 to 100 tons)	1.03–0.86	1.27–1.09
Centrifugal-hermetic	0.94–0.74	
Centrifugal-external drive (350 to 1,250 tons)	1.02–0.81	
Auxiliary equipment:		
Pump, cooling water	0.15	
Blower for air conditioner	0.10	0.10
Blower for rejection air	0.15	0.15

Steam-consumption rate lb/h/ton (kg/h/ton) cooling capacity		
	Steam rate	
Item	lb/h/ton	kg/h/ton
Centrifugal compressors (steam-driven):		
Condensing (27 in vacuum)	14.5	6.6
Noncondensing (14.7 psig)	30–35	13.6–15.9
Absorption chillers (100 to 1,200 ton) (low-pressure steam)	18–18.5	8.2–8.4
Absorption chillers (high-pressure steam)	11–11.5	5–5.2

Gas-consumption rate, Btu/ton (J/ton) cooling capacity		
	Gate rate	
Item	Btu/ton	kJ/ton
Refrigeration compressors (natural-gas-engine-driven)	8,500	8,970

* Depending on the EER (energy efficiency ratio) rating for the different manufacturers.

The dry cooler water circuit operating with antifreeze permits year-round operation.

The **closed-circuit evaporative cooler** has an indirect condenser water circuit, but uses a spray system and evaporative cooling to reduce the head pressure. In both the dry cooler and closed-circuit cooler there is no contact of either condenser water or antifreeze with the outdoor atmosphere; often this proves highly desirable.

Cooling Towers In large systems and systems where maximum overall efficiency is important, a portion of the condenser water is evaporated into the atmosphere to cool the remaining condenser water. The water lost by evaporation must be made up, usually from a domestic water source. Figure 12.4.20 shows two cooling tower configurations. The **counterflow** arrangement requires a spray system and is usually taller than the crossflow type. **Crossflow** towers have simpler gravity distribution systems.

All cooling towers and all evaporative spray systems that operate during subfreezing weather must be protected against freezing. A number of methods are available to add heat to the piping circuits and the basins or sumps.

Condenser Water System For electrically driven refrigeration compressors a temperature differential of 10°F (5.5°C) may be assumed, and for steam-driven equipment a temperature differential of 20°F (11°C) is usual. In the latter case the refrigeration and steam condensers are piping in series with a temperature rise of approximately 10°F (5.5°C) each.

$$\text{Condenser water, gal/min} = \frac{\text{Total heat rejected, Btu/h}}{500 \times \text{temperature differential, °F}} \quad (12.4.26a)$$

In SI units,

$$\text{Condenser flow, m}^3\text{/s} = \frac{\text{Total heat rejected, kW}}{4190 \times \text{temperature differential, °C}} \quad (12.4.26b)$$

Atmospheric Evaporative Cooling Equipment The lowest temperature to which water can be cooled in atmospheric cooling equipment is the wet-bulb temperature of the ambient air.

Water-Cooling Effectiveness in Percent

$$E = \frac{(\text{hot-water temp.} - \text{cold-water temp.}) \times 100}{\text{hot-water temp.} - \text{wet-bulb temp. entering}} \quad (12.4.27)$$

The cold-water temperature must be chosen to place the requirement within the effectiveness range of the equipment used.

Makeup Water for Atmospheric Cooling Equipment Makeup water is introduced to replace losses due to evaporation, drift, and blowdown. If all water were cooled by evaporation, the loss by evaporation for the usual 10°F (5.5°C) cooling range would be:

$$\text{Evaporation, \%} = \frac{Q \times 100}{8.3 \times \text{gal/min} \times \text{hfg}} \quad (12.4.28a)$$

where Q = total heat rejected. Btu/h; gal/min = total condenser water circulated; and hfg = evaporation heat of water, Btu/lb, at ambient design temperature. In SI units:

$$\text{Evaporation, \%} = \frac{Q' \times 100}{1{,}000 \times \text{m}^3\text{/s} \times \text{hfg}} \quad (12.4.28b)$$

where Q' = total heat rejected, kJ/min; m^3/s = total condenser water circulated; and hfg = evaporation heat of water, kJ/kg, at ambient design temperature.

Drift losses depend on the tower design, but generally, from the cooling tower, they are limited to 0.2 percent of the circulated rate.

The makeup water replacing losses due to evaporation, drift, and blowdown introduces dissolved solids into the system.

To prevent excessive concentration, a portion of the circulating water is wasted. The quantity of blowdown depends on the original quantity of dissolved solids in the makeup water and the permissible concentration.

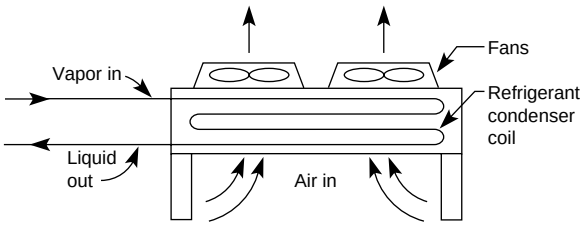
For most open systems, chemical water treatment is used to inhibit corrosion and minimize blowdown.

Water Distribution, Chilled Water Systems' Temperature Differential Design temperature differentials for chilled-water systems are determined by the need to supply chilled water at an average temperature sufficiently low to provide adequate cooling and dehumidification and the desire to minimize the costs of piping, pumping energy, and refrigeration. These differentials normally range from 10°F (5.6°C) for average systems to as much as 15°F (8.3°C) or more for very large systems with widely separated loads.

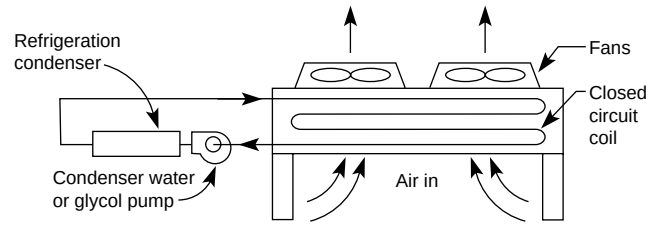
$$\text{Volume flow rate, gal/min} = \frac{(\text{total load} + \text{piping heat gains} + \text{pump heat}), \text{Btu/h}}{500 \times \text{temperature differential, °F}} \quad (12.4.29a)$$

A comparable equation in SI units is

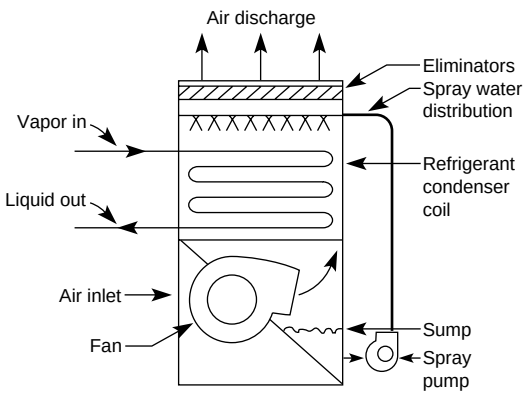
$$\text{Volume flow rate, m}^3\text{/s} = \frac{(\text{total load} + \text{piping heat gains} + \text{pump heat}), \text{kW}}{4{,}190 \times \text{temperature differential, °C}} \quad (12.4.29b)$$



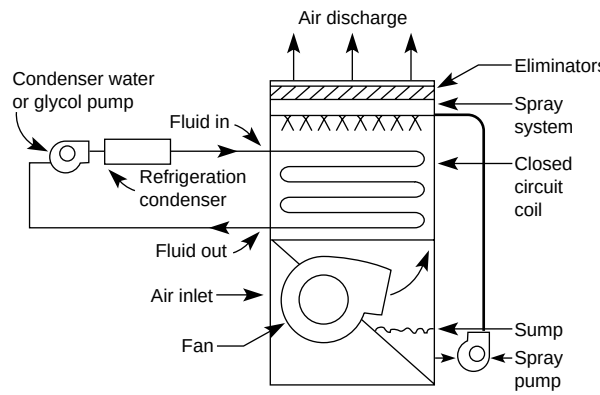
Air-Cooled Condenser



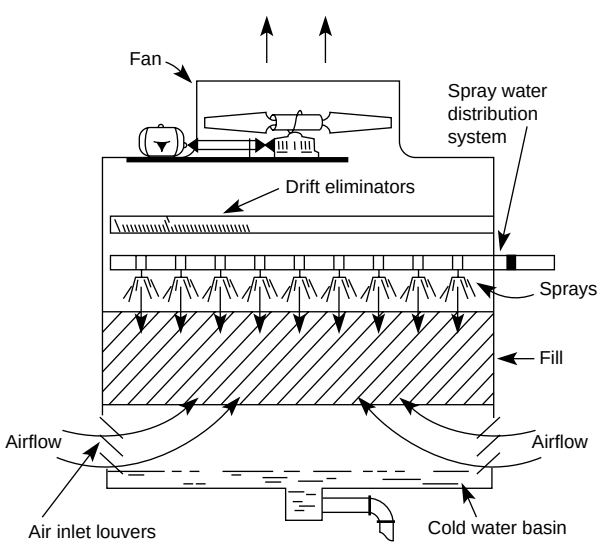
Closed Circuit Dry Cooler



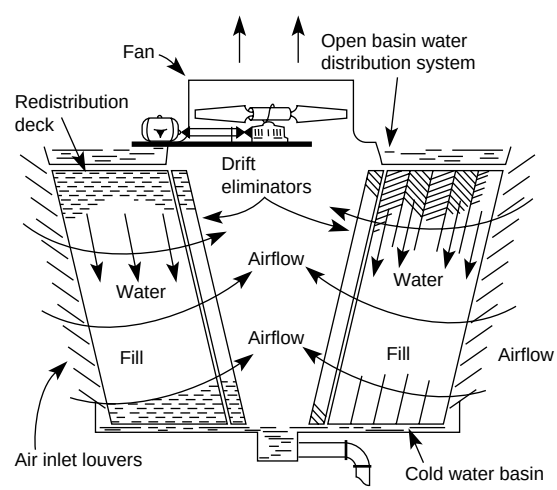
Evaporative Condenser



Closed Circuit Evaporative Cooler



Counterflow Cooling Tower



Crossflow Cooling Tower

Fig. 12.4.20 Heat rejection equipment. *(Courtesy of E. Revzina.)*

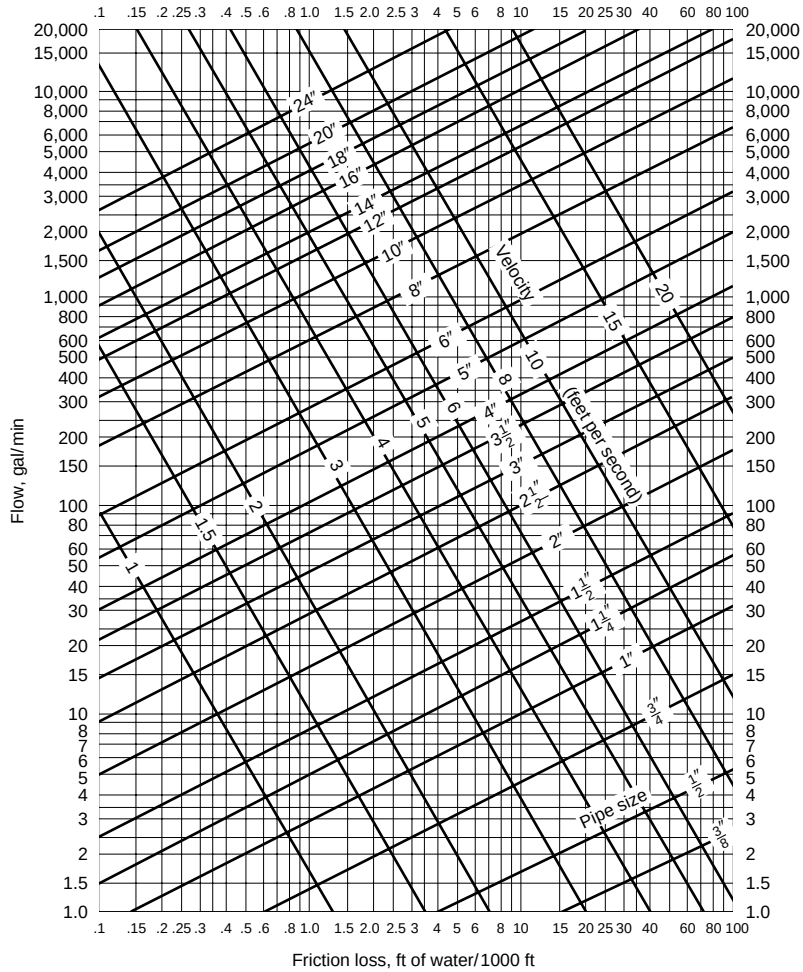


Fig. 12.4.21 Friction loss for open piping systems using schedule 40 pipe.

Table 12.4.37 Low-Pressure Steam System Pipe Capacities (lb/h), with Condensate Flowing with the Steam Flow

Nom. pipe size, in	Pressure drop per 100 ft											
	1/16 psi (1 oz)		1/8 psi (2 oz)		1/4 psi (4 oz)		1/2 psi (8 oz)		3/4 psi (12 oz)		1 psi	
	Saturated pressure (psig)											
	3.5	12	3.5	12	3.5	12	3.5	12	3.5	12	3.5	12
¾	9	11	14	16	20	24	29	35	36	43	42	50
1	17	21	26	31	37	46	54	66	68	82	81	95
1¼	36	45	53	66	78	96	111	138	140	170	162	200
1½	56	70	84	100	120	147	174	210	218	260	246	304
2	108	134	162	194	234	285	336	410	420	510	480	590
2½	174	215	258	310	378	460	540	660	680	820	780	950
3	318	380	465	550	660	810	960	1,160	1,190	1,430	1,380	1,670
3½	462	550	670	800	990	1,218	1,410	1,700	1,740	2,100	2,000	2,420
4	726	800	950	1,160	1,410	1,690	1,980	2,400	2,450	3,000	2,880	3,460
5	1,700	1,430	1,680	2,100	2,440	3,000	3,570	4,250	4,380	5,250	5,100	6,100
6	1,920	2,300	2,820	3,350	3,960	4,850	5,700	7,000	7,200	8,600	8,400	10,000
8	3,900	4,800	5,570	7,000	8,100	10,000	11,400	14,300	14,500	17,700	16,500	20,500
10	7,200	8,800	10,200	12,600	15,000	18,200	21,000	26,000	26,200	32,000	30,000	37,000
12	11,400	13,700	16,500	19,500	23,400	28,400	33,000	40,000	41,000	49,500	48,000	57,500

NOTE: The weight flow rates at 3.5 psig can be used to cover saturated pressures from 1–6 psig and the weight flow rates at 12 psig can be used to cover saturated pressures from 8–16 psig with an error not exceeding 8%.

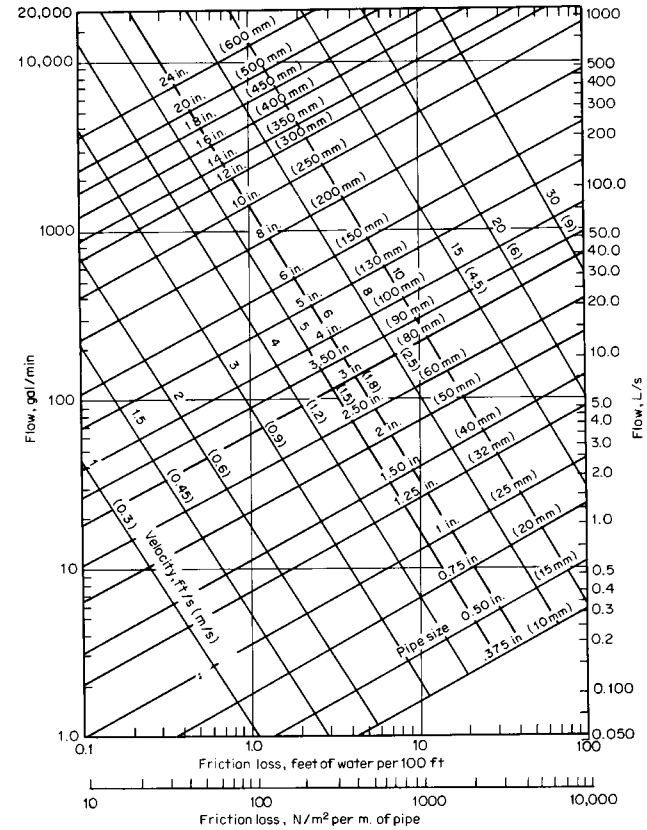


Fig. 12.4.22 Friction loss for closed piping systems using schedule 40 pipe.

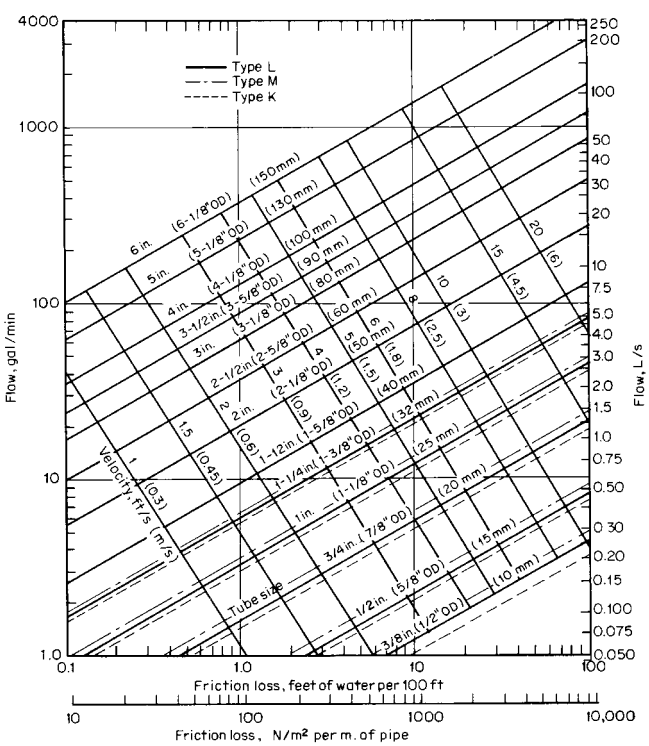


Fig. 12.4.23 Friction loss for closed and open piping systems using copper tubing.

Table 12.4.38 Return Main and Riser Capacities for Low-Pressure Steam Systems

Pipe size, in	Pressure drop per 100 ft														
	1/32 psi (1/2 oz)			1/24 psi (2/3 oz)			1/16 psi (1 oz)			1/8 psi (2 oz)			1/4 psi (4 oz)		
	Wet*	Dry	Vac	Wet*	Dry	Vac	Wet*	Dry	Vac	Wet*	Dry	Vac	Wet*	Dry	Vac
Return mains															
3/4						42			100			142			200
1	125	62		145	71	143	175	80	175	250	103	249	350	115	350
1 1/4	213	130		248	149	244	300	168	300	425	217	426	600	241	600
1 1/2	338	206		393	236	388	475	265	475	675	340	674	950	378	950
2	700	470		810	535	815	1,000	575	1,000	1,400	740	1,420	2,000	825	2,000
2 1/2	1,180	760		1,580	868	1,360	1,680	950	1,680	2,350	1,230	2,380	3,350	1,360	3,350
3	1,880	1,460		2,130	1,560	2,180	2,680	1,750	2,680	3,750	2,250	3,800	5,350	2,500	5,350
3 1/2	2,750	1,970		3,300	2,200	3,250	4,000	2,500	4,000	5,500	3,230	5,680	8,000	3,580	8,000
4	3,880	2,930		4,580	3,350	4,500	5,500	3,750	5,500	7,750	4,830	7,810	11,000	5,380	11,000
5						7,880			9,680			13,700			19,400
6						12,600			15,500			22,000			31,000
Return risers															
3/4		48			48	143		48	175		48	249		48	350
1		113			113	244		113	300		113	426		113	600
1 1/4		248			248	388		248	475		248	674		248	950
1 1/2		375			375	815		375	1,000		375	1,420		375	2,000
2		750			750	1,360		750	1,680		750	2,380		750	3,350
2 1/2						2,180			2,680			3,800			5,350
3						3,250			4,000			5,680			8,000
3 1/2						4,480			5,500			7,810			11,000
4						7,880			9,680			13,700			19,400
5						12,600			15,500			22,000			31,000

* Vacuum values may be used for wet return risers and mains.

Water Distribution, Chilled-Water Systems Temperature Differential

Application	Temp rise °C	Temp rise °F
Close-coupled system on one floor	5–8	3–4.5
Two- or three-story building	8–11	4.5–6
Multistory building	12–20	6.5–11

Water Piping Design

There is a **friction loss** in any pipe through which water is flowing. This loss depends on the following factors:

1. Water velocity
2. Pipe diameter
3. Interior surface roughness
4. Pipe length

System pressure has no effect on the head loss of the equipment in the system. However, higher than normal system pressures may dictate the use of heavier pipe, fittings, and valves along with specially designed equipment.

To properly design a water piping system, the engineer must evaluate not only the pipe friction loss but the loss through valves, fittings, and other equipment. In addition to these friction losses, the use of diversity in reducing the water quantity and pipe size should be considered in designing the water piping system.

Most air conditioning applications use either steel pipe or copper tubing in the piping system. For friction loss values in steel pipe or copper tubing, refer to Figs. 12.4.21 to 12.4.23.

Pipe size determination, especially for very long systems and/or unusual construction or operating cost considerations, is based on an economic analysis, but smaller systems with less critical economic factors are often based on average conditions. In a typical system pipe sizes are selected for a pressure loss of from 3 to 6 ft per 100 ft of pipe.

Pipe Length The total friction loss in a water piping system is the sum of losses in:

1. The total straight lengths of pipe
2. The equivalent lengths of pipe due to fittings, valves, and other elements in the piping system.

Figure 12.4.24 shows the resistance of some valves and fittings in terms of equivalent length of straight pipe.

Steam System Piping

Steam systems which operate below 15 psig are classified as low-pressure systems. Tables 12.4.37 and 38 may be used to determine the size of piping for low-pressure steam and condensate systems pitched in the direction of flow and for risers in which steam and condensate flow in opposite directions.

Table 12.4.39 lists total pressure drop limitations for long-length piping systems. In such systems, pipe sizes may have to be increased to keep total pressure loss in supply and return piping within the system limits. For systems in which pressure may vary, i.e., when steam is supplied by a boiler, the distribution pressure must be based on the minimum pressure that will be available.

Table 12.4.39 Total Pressure Drop for Two-Pipe Low-Pressure Steam Piping Systems

Initial steam pressure, psig	Total pressure drop in supply piping, psi	Total pressure drop in return piping, psi
2	1/2	1/2
5	1 1/4	1 1/4
10	2 1/2	2 1/2
15	3 3/4	3 3/4
20	5	5

Tables 12.4.40 and 41 are used to size piping for use in medium- and high-pressure systems, respectively, and assume piping is installed with proper pitch and adequate provisions to remove condensate.

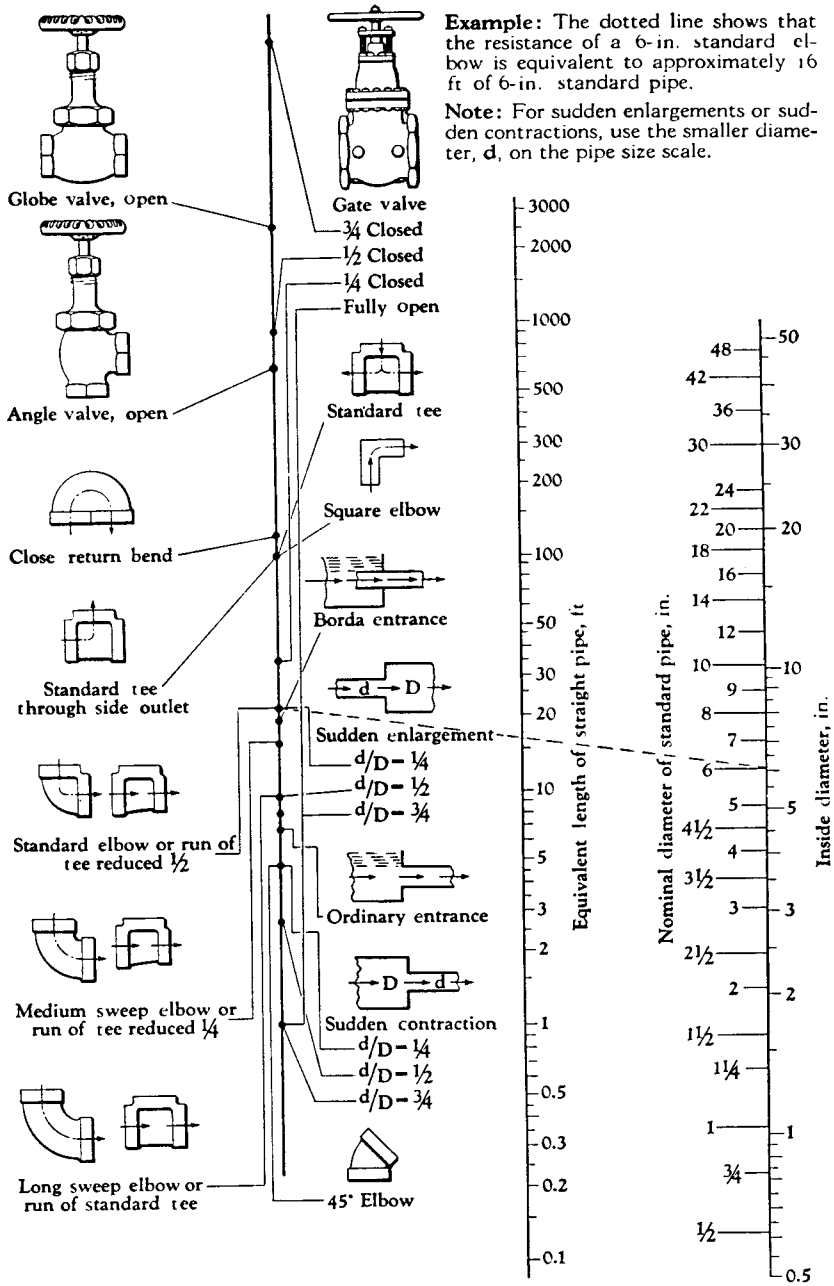


Fig. 12.4.24 Resistance of valves and fittings in terms of equivalent length of straight pipe. *(Crane Co.)*

Table 12.4.40 Medium-Pressure Steam System (30 psig) Pipe Capacities (lb/h)

Pipe size, in	Pressure drop per 100 ft: ⅛ psi (2 oz)	¼ psi (4 oz)	½ psi (8 oz)	¾ psi (12 oz)	1 psi (16 oz)
	Supply mains and risers		25–35 psig—max. error 8%		
¾	15	22	31	38	45
1	31	46	63	77	89
1¼	69	100	141	172	199
1½	107	154	219	267	309
2	217	313	444	543	627
2½	358	516	730	924	1,033
3	651	940	1,330	1,628	1,880
3½	979	1,414	2,000	2,447	2,825
4	1,386	2,000	2,830	3,464	4,000
5	2,560	3,642	5,225	6,402	7,390
6	4,210	6,030	8,590	10,240	12,140
8	8,750	12,640	17,860	21,865	25,250
10	16,250	23,450	33,200	40,625	46,900
12	25,640	36,930	52,320	64,050	74,000
	Return mains and risers		0–4 psig—max. return pressure		
¾	115	170	245	308	365
1	230	340	490	615	730
1¼	485	710	1,025	1,285	1,530
1½	790	1,155	1,670	2,100	2,500
2	1,575	2,355	3,400	4,300	5,050
2½	2,650	3,900	5,600	7,100	8,400
3	4,850	7,100	10,250	12,850	15,300
3½	7,200	10,550	15,250	19,150	22,750
4	10,200	15,000	21,600	27,000	32,250
5	19,000	27,750	40,250	55,500	60,000
4	31,000	45,500	65,500	83,000	98,000

Table 12.4.41 High-Pressure Steam System (150 psig) Pipe Capacities (lb/h)

Pipe size, in	Pressure drop per 100 ft: ⅛ psi (2 oz)	¼ psi (4 oz)	½ psi (8 oz)	¾ psi (12 oz)	1 psi (16 oz)	2 psi (32 oz)	5 psi
	Supply mains and risers			130–180 psig—max, error 8%			
¾	29	41	58	82	116	184	300
1	58	82	117	165	233	369	550
1¼	130	185	262	370	523	827	1,230
1½	203	287	407	575	813	1,230	1,730
2	412	583	825	1,167	1,650	2,000	3,410
2½	683	959	1,359	1,920	2,430	3,300	5,200
3	1,237	1,750	2,476	3,500	4,210	6,000	9,400
3½	1,855	2,626	3,715	5,250	6,020	8,500	13,100
4	2,625	3,718	5,260	7,430	8,400	12,300	19,200
5	4,858	6,875	9,725	13,750	15,000	21,200	33,100
6	7,960	11,275	15,950	22,550	25,200	36,500	56,500
8	16,590	23,475	33,200	46,950	50,000	70,200	120,000
10	30,820	43,430	61,700	77,250	90,000	130,000	210,000
12	48,600	68,750	97,250	123,000	155,000	200,000	320,000
	Return mains and risers			1–20 psig—max, return pressure			
¾	156	232	360	465	560	890	
1	313	462	690	910	1,120	1,780	
1¼	650	960	1,500	1,950	2,330	3,700	
1½	1,070	1,580	2,460	3,160	3,800	6,100	
2	2,160	3,300	4,950	6,400	7,700	12,300	
2½	3,600	5,350	8,200	10,700	12,800	20,400	
3	6,500	9,600	15,000	19,500	23,300	37,200	
3½	9,600	14,400	22,300	28,700	34,500	55,000	
4	13,700	20,500	31,600	40,500	49,200	78,500	
5	25,600	38,100	58,500	76,000	91,500	146,000	
6	42,000	62,500	96,000	125,000	150,000	238,000	

12.5 ILLUMINATION

by Abraham Abramowitz

REFERENCES: Amick, ''Fluorescent Lighting Manual,'' McGraw-Hill. ''IES Lighting Handbook'' (1981). Design publications of General Electric Co., North American Philips Co. (successor to Westinghouse Electric Co. Lamp Division), and GTE-Sylvania.

BASIC UNITS

Candela, cd (formerly candle) is the unit of luminous intensity of a light source. One candela is defined as the luminous intensity in a given direction, of a source that emits monochromatic radiation of frequency 540×10^{12} Hz (approximately 555 nm) and of which the radiant intensity in that direction $\frac{1}{683}$ W per steradian (W/sr).

Lumen, lm, is the unit of luminous flux ϕ. It is equal to the flux on a unit surface all points of which are one unit distant from a uniform point source of one candela. Such a point source emits 4π lumens. (For an additional definition of lumen, see the following material on vision.)

Illuminance E is the density of luminous flux on a surface. If the foot is taken as the unit of length and the flux is uniformly distributed over the surface, the density in **lumens per square foot** is called **footcandles, fc;** in SI units **lumens per square metre, lux (lx),** is used. (One footcandle equals 10.76 lux.) In order to make the units comparable, dekalux (10 lux) is frequently used.

The term *illumination* is frequently used for the word *illuminance.* Modern practice reserves *illumination* for the process of lighting and *illuminance* for the result.

Luminance is the luminance intensity of any surface in a given direction per unit of projected area of the surface as viewed from that direction. The unit of luminance is candela/in^2; in SI units cd/m^2 is used. (1 cd/in^2 = 1,550 cd/m^2.) In general, a luminous surface will have a different luminance when viewed from different angles. An important exception is a **perfectly diffuse reflecting (lambertian) surface** which has a constant luminance regardless of the viewing angle. If such a surface has a luminance of 1 cd/in^2, it emits 452 lm/ft^2. **Footlamberts,** fL, in lumens per square foot, is the unit of luminance applied to this case. While this conversion applies only to the perfectly diffuse case, it is frequently used in all cases. Thus, a perfectly diffuse surface with a luminance of 1 cd/in^2 is said to have a luminance of 452 fL. In practice the average lumens emitted per square foot of surface is taken to be the footlamberts. This conversion practice is deprecated.

Subjective brightness is the subjective attribute of any light sensation giving rise to the whole scale of qualities of becoming bright, light, brilliant, dim, or dark. Unfortunately, the term ''brightness'' often is used when referring to luminance.

NOTE. The above definitions are adapted from the ''IES Lighting Handbook.''

Absorption, reflection, and transmission are the general processes by which incident light flux interacts with a medium. **Absorption** is the process whereby incident flux is dissipated. **Reflection** is the process by which the incident flux leaves a surface or medium from the incident side.

NOTE. Reflection may occur as from a mirror (specular reflection), it may be reflected at angles different from that of the incident flux to incident plane (diffuse reflection), or it may be a combination of the two types of reflection.

Transmission is the process by which incident flux leaves a surface or medium on a side other than the incident side. If the light ray is reduced only in intensity, the transmission is called *regular.* If the ray emerges in all directions, transmission is called *diffuse.* Both modes may exist in combination.

The incident flux ϕ_i equals the flux absorbed ϕ_a, reflected ϕ_r, and transmitted ϕ_t. That is,

$$\phi_i = \phi_a + \phi_r + \phi_t$$

Dividing this equation by ϕ_i, we obtain

$$1 = \phi_a/\phi_i + \phi_r/\phi_i + \phi_t/\phi_i$$

or

$$1 = \alpha + \rho + \tau$$

α is the *absorptance*, ρ is the *reflectance*, and τ is the *transmittance.* In each case, the incident flux may be restricted to a single wavelength, a particular direction, and a given solid angle. These must be specified.

The **wavelength** of electromagnetic radiation is measured in metres. For the frequencies involved in illumination, the wavelength is given in nanometres, nm, equal to 10^{-9} m, and micrometres, μm, equal to 10^{-6} m.

VISION

Most engineering designs, (bridges, structures, roads etc.) are based on strength and are not concerned with the way the human organism reacts. The **response of the eye** is central to illuminating engineering. The **lens** of the **eye** focuses an image on the **retina.** Here a photochemical process takes place which sends nerve impulses to the brain via the optic nerve. The amount of light entering the eye is controlled by the **pupil.** The normal eye automatically accommodates itself to focus on an object, while the pupil adjusts itself to allow for a high or low level of object luminance. The sensors in the eye are known as **rods** and **cones.** The cones are clustered in a small central part of the retina called the **fovea.** They transmit a sharp image to the brain and give color response. Outside the fovea the rods predominate. They give neither a sharp image nor a color response. When the luminance of the visual field is 0.01 fL or lower, as at night, seeing is due to the rods only and is called **scotopic vision.** At higher levels, with the cones primarily involved, seeing is called **photopic vision.** There is an intermediate region called **mesopic vision.**

The response of the eye to colors of different wavelengths is given in Fig. 12.5.1. Note the shift in maximum response at lower luminance levels called the ''Purkinje shift.'' Note that these curves are relative ones, and that the two peaks do not correspond to the same levels of illumination. The **luminous efficacy** (lumen output per radiated watt) is 683 lm/W at the wavelength of maximum photopic response 555 nm. For white light, radiation which has the characteristic of an equal energy spectrum with all the energy in the visual region, it is approximately 220 lm/W.

Spectral Lumen If the response curve of the eye for photopic vision, versus λ in nanometers, is expressed as $k(\lambda)$, and the spectral power function of the source in watts per nanometer is taken to be $Q_e(\lambda)$, then the luminous flux is given by the equation

$$\phi_{\text{lumens}} = 683 \int_{380}^{780} k(\lambda)\, Q_e(\lambda)\, d(\lambda) \qquad (12.5.1)$$

LIGHT METERS

Early light meters compared the luminance of a diffuse highly reflecting surface with that obtained from a calibrated standard. The most common light meter in use today is similar to a photographic exposure meter. A photovoltaic cell is directly connected to a sensitive microammeter calibrated in footcandles (or dekalux). The best meters (called color-corrected) have a response similar to that of the eye in photopic vision. Special shapes are used on the cover to avoid total reflection of

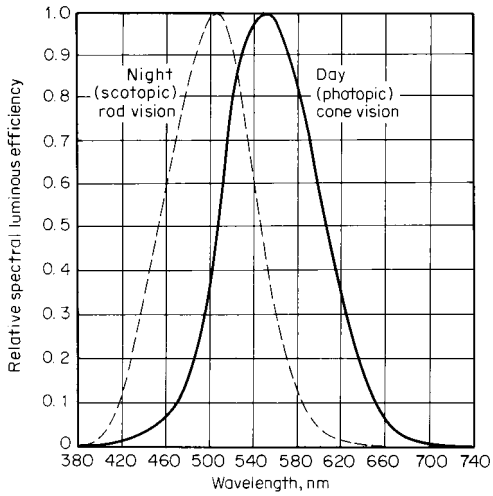


Fig. 12.5.1 Relative spectral luminous efficiency curves for photopic and scotopic vision, showing the Purkinje shift on the wavelength of maximum efficiency. Note the wavelength of the visual region of the electromagnetic spectrum. *(IESNA Lighting Handbook, 5th ed. This material has been modified from its original version and is not reflective of its original form as recognized by the IESNA.*)

light from the glass surface of the cell. Such meters are said to be cosine law corrected. The microammeter is frequently replaced with an electronic amplifier using an analog or digital readout.

LIGHT SOURCES

The original and still major source of light is the sun. Next came fire, derived from candles, oil, and gas lamps. With the discovery of electricity came arc lamps, gas-discharge lamps, and hot-filament lamps. ''Flame'' or hot sources give a continuous spectrum. Gas-discharge devices such as neon lamps and mercury-arc lamps give discrete, or line, spectra. The lines may be modified in various ways: by pressure broadening, use of phosphor coatings (to convert ultraviolet radiation into visible light), and using a mixture of gases. The continuous spectra of phosphors have colors which depend upon the mixture used. Light-emitting diodes, LED, consisting of a layer of two different semiconductors, are in use for display purposes.

Color Temperature and Luminance

In general, three quantities are required to specify the color of a light and its luminous level. However, an approximate designation is used by specifying the temperature of a hot (black-body) emitter whose color almost matches that of the light. The **color temperature** of daylight is about 6000 K and that of tungsten lamps about 2300 to 3300 K.

Table 12.5.1 Approximate Luminances of Various Light Sources (IES)

Light source	Approximate average luminance, cd/in²	Approximate average luminance, kcd/m²
Clear sky	5.16	8
Candle flame (sperm)	6.45	10
60-W inside frosted bulb	77.4	120
60-W ''white bulb''	19.35	30
Fluorescent lamp, cool white, T-12 bulb, medium loading	5.3	8.2
High-intensity mercury-arc type H33, 2.5 atm	968	1,500
Clear glass neon tube 15 mm, 60 mA	1.03	1.6

Different light sources have markedly different luminances as shown in Table 12.5.1. ''Large'' sources have low luminances, while ''small'' sources have high luminances.

Lamps

Electric lamps are the principal source of artificial light in common use. They convert electrical energy into light or radiant energy.

An **incandescent-filament lamp** contains a filament which is heated by the current passing through it. The filament is enclosed in a glass bulb which has a base suitable to connect the lamp to an electrical socket. To prevent oxidation of the filament at elevated temperature, the bulb is evacuated of air or filled with an inert gas. The bulb also serves to control the light from the incandescent filament, which is essentially a point source. High luminance of the source is typically reduced by acid etching to frost the inside surface of the bulb. Silica coating will also provide additional diffusion and can alter the color of the light emitted. Portions of the bulb's interior can be covered with reflecting material to give a predetermined direction to the emitted light. Chemical tinting of clear glass bulbs provides a variety of colors. Whenever the color that is normally produced by an incandescent filament is changed, the filtering process removes from the radiated light the energy of all wavelengths except those necessary to produce the desired color. This subtractive method of color alteration is less efficacious than the generation of light of varying colors by gaseous discharge.

Sizes and **shapes** of lamp bulbs are designated by a letter code followed by a numeral; the letter indicates the shape (Fig. 12.5.2), and the number indicates the diameter of the bulb in eighths of an inch. Thus a T-12 lamp has a tubular shape and is 1⅛ or 1½ in in diameter.



Fig. 12.5.2 Typical filament lamp shapes: S, straight; F, flame; G, globe; A, general service; T, tubular; PS, pear shape; PAR, parabolic; R, reflector.

Incandescent lamps are available with several **types of bases** (Fig. 12.5.3). Most general-service lamps have medium screw bases; larger or smaller screw bases are used depending on lamp wattage. Bipost and prefocus bases accurately position the filament, as in optical projection systems. Bipost lamps also serve where ruggedness and greater heat dissipation are required.

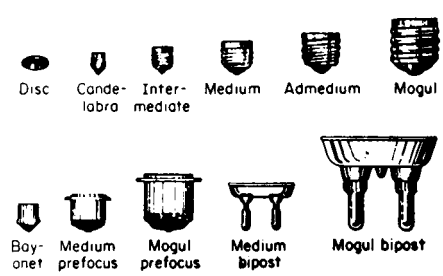


Fig. 12.5.3 Typical incandescent lamp bases.

Incandescent-lamp filaments are generally constructed of tungsten. Tungsten has a high melting point and a low vapor pressure, which permits high operating temperatures without evaporation: the higher the operating temperature, the higher the efficacy (lumens per watt) and the shorter the life. Filament evaporation throughout the life of the lamp causes blackening of the bulb and thinning of the filament with consequent lower light output. Argon-nitrogen gas filling reduces the rate of evaporation. Figure 12.5.4 shows steps in lamp manufacture.

Tungsten filaments are also placed in compact quartz tubes filled with a halogen atmosphere where the tungsten halide lighting source

continuously returns evaporated tungsten particles to the filament. The inside walls do not blacken, and light output remains fairly constant throughout the life of the lamp.

For exposed outdoor lamp signs, **gas-filled incandescent lamps** are often used, especially when high-speed animation is to be depicted by

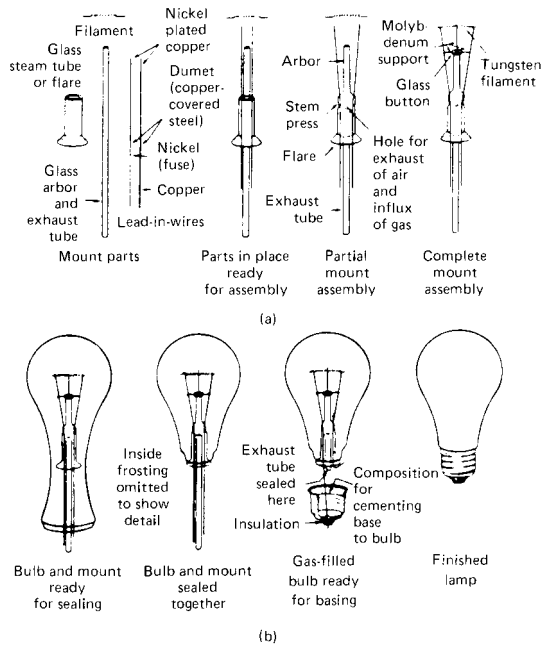


Fig. 12.5.4 Steps in the manufacture of a typical filament incandescent lamp. (*a*) Assembly of inner structure; (*b*) final assembly. *(IESNA. This material has been modified from its original version and is not reflective of its original form as recognized by the IESNA.)*

the on-off sequencing of the lamps. The lamp envelope is filled with a gas which enhances the speed with which the filament is heated and cooled. The very fast quenching of the filament without the usual decay trail of light permits a sharply defined illusion of motion.

A **fluorescent lamp** consists of a glass tube coated on the inside with phosphor powders which fluoresce when excited by ultraviolet light; filament electrodes are mounted in end seals connected to base pins (Fig. 12.5.5). The tube is filled with rare gases (such as argon, neon, and

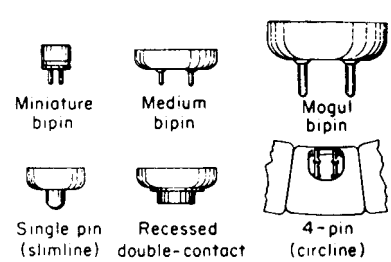


Fig. 12.5.5 Fluorescent lamp bases.

krypton) and a drop of mercury (Fig. 12.5.6), and operates at a relatively low pressure. In operation, electrons are emitted from the hot electrodes. These electrons are accelerated by the voltage across the tube until they collide with mercury atoms, causing them to be ionized and excited. When the mercury atom returns to its normal state, mercury spectral lines in both the visible and the ultraviolet region are generated. The low pressure enhances ultraviolet radiation. The ultraviolet radiation excites the phosphor coating to luminance. The resulting light output is not only much higher than that obtained from the mercury lines alone but also results in a continuous spectrum with colors dependent upon the phosphors used.

As with all gas-discharge devices, these lamps have negative voltampere characteristics. Unless the voltage difference between the applied voltage and the lamp operating voltage is absorbed in some way, damaging currents will result. A reactor is used in series with the lamp. It may be capacitive or inductive (many turns of wire on an iron core). The supply voltage should be at least twice the lamp operating voltage. Where this is not the case, the supply voltage (up to 277 V is used) is stepped up by an autotransformer. The necessary reactance is frequently part of the transformer leakage inductance.

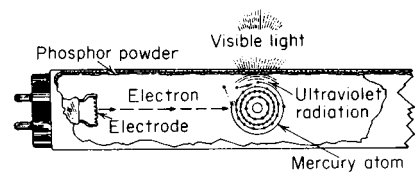


Fig. 12.5.6 Fluorescent lamp operation. *(Westinghouse.)*

For starting purposes, voltages higher than twice lamp operating voltages may be used. For minimum line current a power-factor-correcting capacitor is used and assembled with the autotransformer. A capacitor is put across the lamp to reduce radio-frequency interference with nearby radio receivers. All these elements are placed inside a case filled with a potting compound. The assembly is called a **ballast.** The object of the compound is to reduce noise from the core lamination vibrations and to improve heat dissipation. Built-in thermal protectors, which deenergize the ballast when dangerous temperatures are reached, are now required in the United States. Ballast manufacturers rate their units by noise levels. Some ballasts use solid-state, high-frequency generators.

To start a fluorescent lamp, electron emission from the electrodes must be induced. Two methods are generally employed: (1) the filament electrodes are heated by passing current through them; (2) a high voltage, sufficient to start an electric discharge in the lamp, is applied across it. Once a discharge starts, mercury-ion bombardment keeps the filaments at a hot electron-emitting temperature. Lamps are designed for either type of operation. The first group is further divided into preheat lamps and rapid-start lamps. Some lamps can be used for both types of circuits. The lamp current is carried primarily by electrons emitted from the filaments. The end or weakening of electron emission is an important cause of the end of lamp life.

Preheat circuits contain starters (Fig. 12.5.7) which are switches, closed when power is first applied, permitting current to flow and preheat the electrodes. After a predetermined period of time, the starter switch opens, throwing a potential across the lamp which starts the discharge. Lamps used for preheat circuits have bipin bases (Fig. 12.5.5).

Instant-start circuits have ballasts which apply sufficient voltage across the lamp to induce current flow without preheating the electrodes. **Slimline lamps** are the principal instant-start type. They have single-pin bases because no preheating is required, are available in sizes up to 8 ft in length, and can have varying lumen outputs dependent upon the ballast current rating and wattage. Because of their high starting voltage, they generally employ spring-loaded push-pull lampholders which disconnect the ballast circuit unless the lamps are properly seated in position.

Instant-start lamps are sometimes available with bipin bases similar to those used in preheat lamps. In these instances the lead wires from the pins are connected together inside the lamp. These lamps, marked ''instant start,'' are not interchangeable with rapid-start equipment. Cold-cathode lamps employ the instant-start principle, have cylindrical iron electrodes, and tend to be less efficacious at shorter lengths because of

high wattage losses at the electrodes. They are limited to low current densities because the electrodes operate at temperatures below that necessary for thermionic emission. Cold-cathode lamps, whose operation is not affected by dimming or flashing, have long life and are generally used for custom-built shapes and patterns that require bending, such as for electric signs.

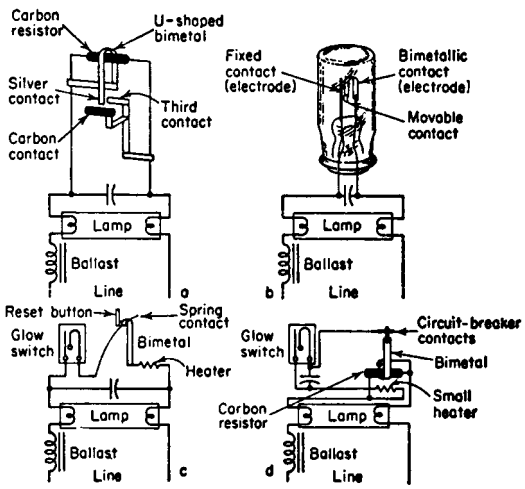


Fig. 12.5.7 Starter switches for preheat cathode circuits. *(IESNA)*

Rapid-start circuit ballasts have separate windings for the electrodes which are immediately and continuously heated when the circuit is energized. This rapid heating causes sufficient ionization in the lamp for a discharge to start from the voltage of the main ballast windings. Two-lamp rapid-start ballasts are of the series sequence type, in which the lamps start in sequence and, when fully lighted, operate in series.

A new type **screw-in fluorescent lamp** with built-in ballast can be used in a standard medium-screw socket. These lamps consume less power than incandescent lamps for the same luminance; accordingly, albeit their first cost is significantly higher, they are expected to prove to be more economical by virtue of their reduced power consumption and much longer life. The verdict of the consuming public is yet to come as significant numbers of them make their way into household and commercial applications.

Fluorescent lamps used in low-ambient-temperature applications, as in outdoor signs, are of the **high-output (HO)** type, and require special high-output ballasts to permit the lamps to maintain their luminance at lower operating temperatures.

Typical fluorescent lamp circuits are shown in Fig. 12.5.8.

High-intensity-discharge lamps consist of tubes in which electric arcs in a variety of materials are produced. Outer glass jackets provide thermal insulation in order to maintain the arc tube temperature. The temperature and amount of material is controlled so that the discharge operates in a vapor pressure of several atmospheres. This results in enhancing the radiation in the visible region.

Mercury-vapor lamps consist of mercury-argon-filled quartz tubes surrounded by a nitrogen-filled glass jacket. Clear lamps radiate the visible mercury lines (bluish green). Ultraviolet radiation is absorbed to some extent by the outer jackets. The color of the light and the lumen output is improved by coating the inside of the outer jackets with a phosphor. When excited by the ultraviolet radiation of the arc, the phosphors add light in the red part of the spectrum to the output. The resulting lamps are called white, color improved, or deluxe white. The lamps start by a discharge in argon between an electrode and a starting electrode (see Fig. 12.5.9). As the mercury vaporizes, the pressure builds up and the discharge transfers to a mercury discharge. This takes several minutes. After shutdown, the lamps cannot be restarted until the inner tube pressure drops so that an argon discharge can start.

Metal halide (multivapor) lamps use small quantities of sodium, thallium, scandium, dysprosium, and indium iodides in addition to the usual mercury-argon mix. Color is improved and output substantially increased over high-intensity-discharge lamps using mercury alone. While the construction is similar to mercury lamps, a bimetal switch is built into the lamp to short out the starting resistor after the lamps start. A vacuum jacket is used around the quartz discharge tube (see Fig. 12.5.9).

High-pressure sodium-vapor lamps use metallic sodium sealed in translucent aluminum oxide tubes. This material is used to withstand the corrosive effect of hot sodium vapor. For starting purposes a xenon fill gas and a sodium-mercury amalgam is used. Arc temperatures are maintained by an outer vacuum jacket. The lamp is started by generating a high-voltage pulse for about a microsecond (see Fig. 12.5.9).

High-pressure discharge lamps, like fluorescent lamps, require ballasts. These provide the necessary voltage, reactances, and power-factor-correcting capacitors. Typical circuits are shown in Fig. 12.5.8.

Table 12.5.2 Comparable Luminous Efficacies (lumens/watt)* (IES)

Lamp	Lumens/watt
Tungsten incandescent	8–33
High-intensity mercury†	24–63‡
Fluorescent†	19–100‡
Metal halide (multivapor)†	69–125
High-pressure sodium†	73–140

* Constantly being improved.
† Ballast losses not included.
‡ Depends upon lamp size, type, and color.

Comparative lamp efficacies (lumens/watt) are given in Table 12.5.2. Lamp data for commonly used incandescent, fluorescent, and high-intensity-discharge lamps are listed in Tables 12.5.3, 12.5.4, and 12.5.5.

Luminaires

Luminaires are generally categorized as **industrial, commercial,** or **residential.** Use within these categories usually determines the quality and ruggedness of materials of construction. Generally speaking, style, ornament, and in most cases low cost are prime considerations for residential fixtures. Industrial fixtures require low maintenance, low operating cost, efficiency, and durability. Commercial fixtures combine the elements of all of these and place heavy emphasis on visual comfort.

Luminaires are classified by the International Commission on Illumination (ICI) in accordance with the percentages of total luminaire output emitted above and below the horizontal (Fig. 12.5.10). Industrial fixtures usually are direct or semidirect.

Table 12.5.3 Incandescent-Lamp Data

Watts	Bulb size	Initial lumens	Rated life, h
25	A-19	230	2,500
40	A-19	474	1,500
60	A-19	1,060	1,000
75	A-19	1,190	750
100	A-19	1,740	750
150	A-21	2,873	750
200	A-23	4,000	750
300	PS-30	6,130	750
500	PS-35	10,675	1,000
750	PS-52	16,935	1,000
1,000	PS-52	23,510	1,000

For general-service lamps 115-, 120-, and 125-V service, inside frosted.
NOTE: Lamps are constantly being improved. The latest manufacturer's data should be used for accuracy.
SOURCE: ''IESNA Handbook,'' 8th ed., 1993, reprinted with permission. (This material has been modified from its original version and is not reflective of its original form as recognized by the IESNA.)

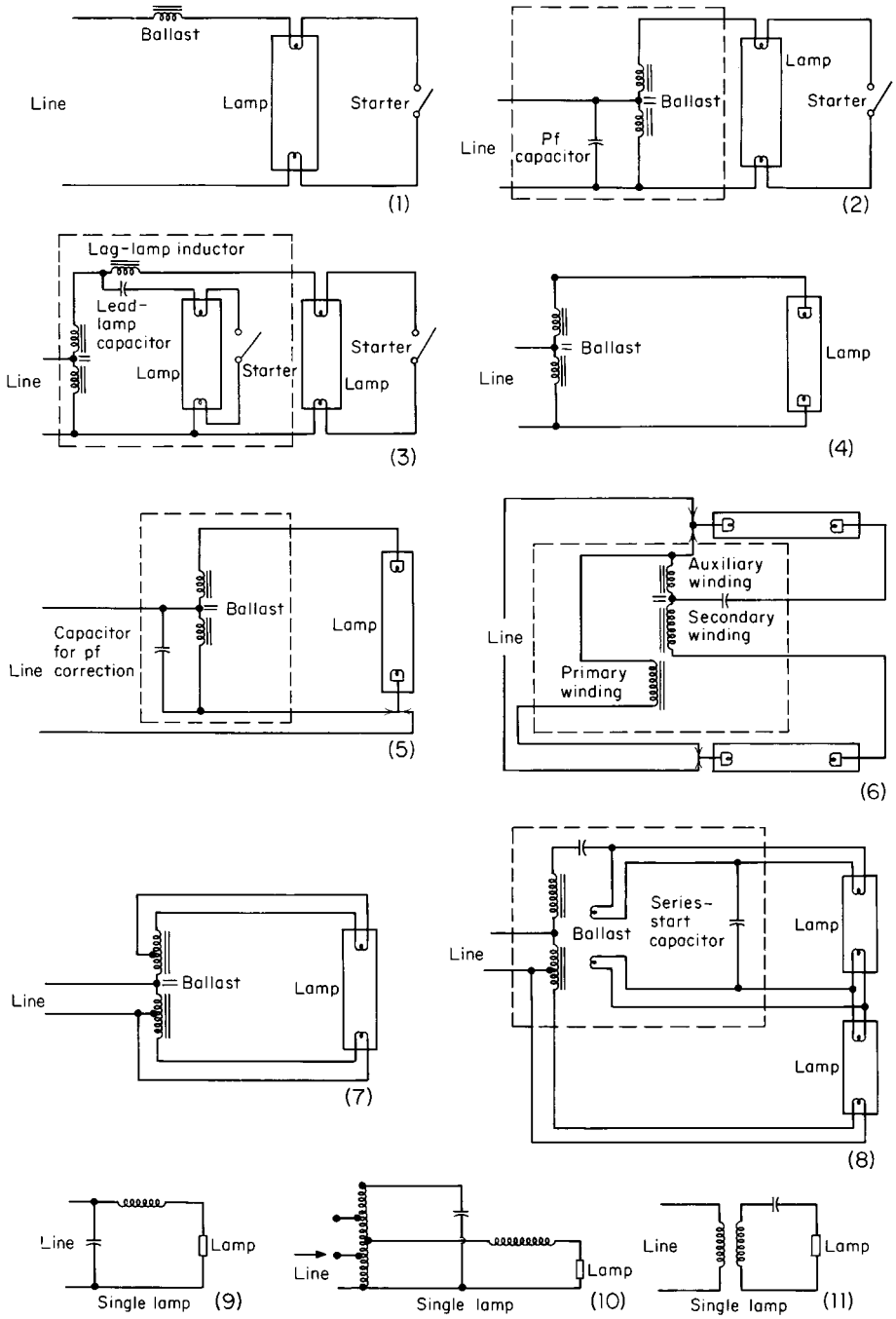


Fig. 12.5.8 Typical circuits for fluorescent and high-intensity discharge lamps. (1) Basic preheat circuit; (2) preheat circuit with autotransformer to step up voltage and capacitor to correct power factor; (3) leadlag preheat; (4) basic instant-start circuit; (5) instant-start circuit showing disconnect lampholder; (6) typical series instant-start circuit; (7) basic rapid-start circuit; (8) two-lamp series lead circuit; (9) mercury reactor ballast circuit; (10) mercury autotransformer circuit; (11) mercury stabilizing ballast circuit. *(GE.)*

Table 12.5.4 Fluorescent Lamp Data*

Lamp designation	Nominal length mm	Nominal length in	Approx lamp Current (ma)	Approx lamp Volts	Approx lamp Watts	Single-lamp circuit watts Ballast	Single-lamp circuit watts Total	Two-lamp circuit watts Ballast	Two-lamp circuit watts Total	Cool white lumens at 100 h	Rated average life, h 3 h burning/start
Preheat starting†											
F15T12	450	18	325	47	15	4.5	19.5	9	39	830	9,000
F20T12	600	24	380	57	20.5	5	25.5	10	51	1,283	9,000
F30T8	900	36	355	99	30.5	10.5	41	17	78	2,330	7,500
F40T12	1,200	48	430	101	40	12	52	16	96	2,150	15,000
F90T12	1,500	60	1,500	65	90	20	110	24	204	6,025	9,000
Rapid start† (lightly loaded lamps)											
F30T12	900	36	430	81	33.5	10.5	44			2,210	18,000
F40T12	1,200	48	430	101	41	13	54	13	95	3,150	21,000
Rapid start† (medium loaded lamps)											
F48T12	1,200	48	800	78	63		85		146	4,300	12,000
F72T12	1,800	72	800	117	87		106		200	6,650	12,000
F96T12	2,400	96	800	153	113		140		252	9,150	12,000
Rapid start† (highly loaded lamps and power grove§)											
F48T12/48PG17	1,200	48	1,500	84	116		146		252	6,900/7,450	9,000
F72T12/72PG17	1,800	72	1,500	125	168		213		326	10,640/11,500	9,000
F96T12/96PG17	2,400	96	1,500	163	215		260		450	15,250/16,000	9,000
Instant start† (slimline)											
F48T12 lead/lag	1,200	48	425	100	39			26	104	3,000	7,500–12,000
F48T12 series	1,200	48	425	100	39			17	95	3,000	7,500–12,000
F72T12 lead/lag	1,800	72	425	149	57			47	161	4,585	7,500–12,000
F72T12 series	1,800	72	425	149	57			25	139	4,585	7,500–12,000
F96T12 lead/lag	2,400	96	425	197	75			40	190	6,300	12,000
F96T12 series	2,400	96	425	197	75			22	172	6,300	12,000
Circline lamps¶											
C8T9	200 OD	8¼ OD	370	61	22.5	7.5	30			1,065	12,000
C12T9	300 OD	12 OD	425	81	33	9	42			1,870	12,000
C16T9	400 OD	16 OD	415	108	41.5	16.5	58			2,580	12,000

SOURCE: Adapted from ''IESNA Handbook,'' 8th ed., 1993, reprinted with permission. (This material has been modified from its original version and is not reflective of its original form as recognized by the IESNA.)

* Lamps are continuously being improved. For design purposes consult the latest manufacturers' data. Data shown is for standard lamps. Energy-saving ballasts and fluorescent lamps are available.

† The first number is the ''nominal'' lamp wattage, while the second number is the tube diameter in eighths of an inch.

§ General Electric Co. trademark.

¶ The first number is the nominal outside diameter of the lamp, while the second number is the tube diameter in eighths of an inch.

Table 12.5.5 High-Intensity-Discharge Lamp Data*

Watt	Nominal lamp Voltage	Nominal lamp Amperes	Approx ballast loss, watts	Approx initial lumens† (100 h)	Life, h
Mercury lamps					
100	130	0.85	10–35	2,500–4,400	24,000+
175	130	1.5	15–35	6,000–8,600	24,000+
250	130	2.1	25–35	8,000–13,000	24,000+
400	135	3.2	20–55	15,000–23,000	24,000+
700	265	2.8	35–65	36,000–43,000	24,000+
1,000	265	4.0	40–90	43,000–63,000	24,000+
Metal-halide lamps					
175	130	1.4	35	12,000–14,000	7,500
400	135	3.2	60	31,000–40,000	15,000–20,000
1,000	250	4.3	50–100	105,000–125,000	10,000–12,000
High-pressure sodium-vapor lamps					
250	100	3.0	55–60	25,000–30,000	24,000
400	100	4.7	65–75	47,500–50,000	24,000

SOURCE: Abstracted from ''IES Lighting Handbook'' and General Electric Co. data.

* Lamps are continuously being improved. For design purposes, consult the latest manufacturers' data.

† Depending upon ballast used, lamps may have outputs which change with burning position.

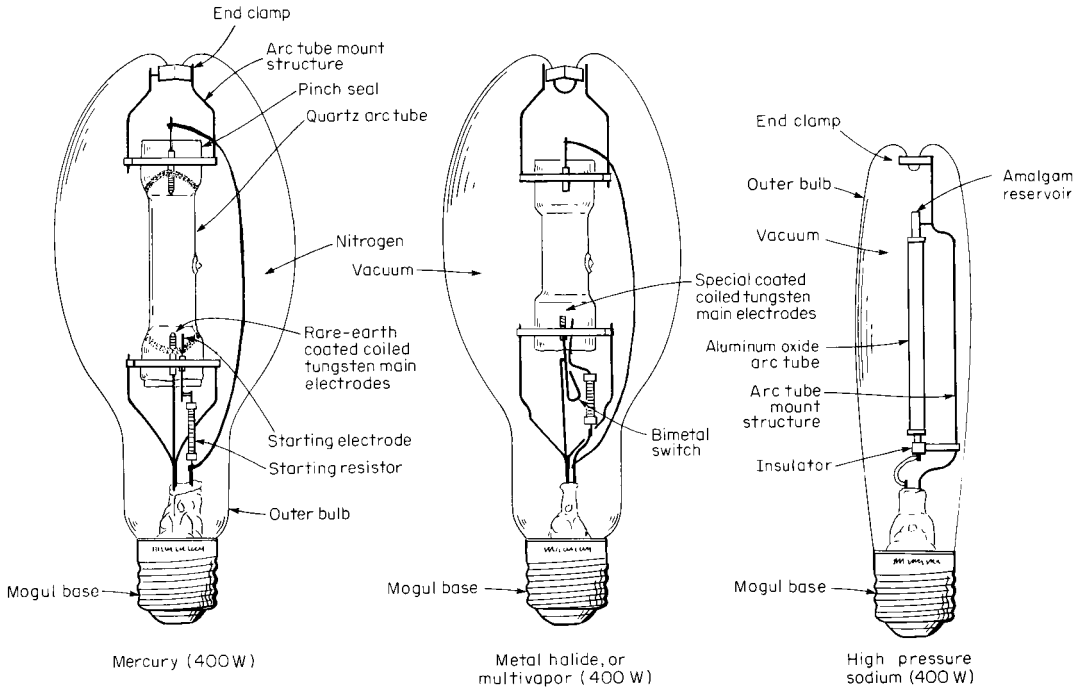


Fig. 12.5.9 High-output lamps. *(GE.)*

Luminaires control the source of light so that it can be better used for a given seeing task. Materials used in luminaires are designed to reflect, refract, diffuse, or obscure light.

Reflectors are commonly made of specular alzak aluminum, glass, baked-enamel steel, and porcelain-enamel steel.

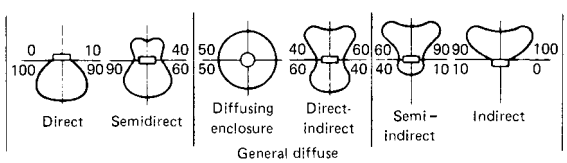


Fig. 12.5.10 Light distribution curves by ICI classifications of luminaires. Upward and downward components are in percentage ranges. *(IES.)*

Lenses with prismatic patterns refract light sources to disperse the rays or to direct them most effectively. Lenses are of glass, acrylic plastic (Plexiglas), or polystyrene. **Glass** is generally superior for incandescent fixtures because of its heat resistance. Fluorescent lamps with **acrylic** lenses have a life at least twice that of **polystyrene** because of their color stability. **Translucent** glass and plastic are used in bottom lenses, louvers, and side panels to diffuse light and to obscure light sources. Baked-enamel steel **louvers,** in egg-crate, concentric ring, or cellular configurations are widely used to shield light sources from normal viewing angles.

Fixture bodies, trims, and lens frames are commonly constructed of steel, electrogalvanized, and/or treated with a rust-inhibiting coating, and painted with several coats of baked enamel. Stamped, spun, cast, and die cast aluminum are also used.

PRESCRIBING ILLUMINATION

The object of a **lighting design** is to provide sufficient illuminance for a given seeing task without introducing discomfort. Sufficient light is not difficult to obtain with modern light sources, but unless properly placed and controlled, uncomfortable, glaring light will result.

A given task has a size, luminance contrast, luminance, and color. The luminance of a perfectly diffusing reflecting surface is given by

$$L = E\rho \tag{12.5.2}$$

where E is illuminance in footcandles, ρ its reflectance (ratio of reflected light flux to incident flux), and luminance L is in footlamberts. The contrast C between two adjacent areas is given by

$$C = (L_b - L_o)/L_b \tag{12.5.3}$$

where L_b is the luminance of the larger or background area and L_o is the object or task luminance for a given illuminance E. Substituting Eq. (12.5.2) into Eq. (12.5.3),

$$C = \frac{E\rho_b - E\rho_o}{E\rho_b} = (\rho_b - \rho_o)/\rho_b \tag{12.5.4}$$

Thus, contrast is basically a function of the task-to-background reflectance. Most surfaces are not perfectly diffuse and require the use of the luminance factor β (ratio of actual luminance for a given viewing angle to the illumination under actual conditions) instead of ρ. The contrast which can just be seen or detected (minimum perceptible contrast) is a function of background luminance for a given task.

Unfortunately most visual tasks have mirrorlike (specular) or semimirrorlike surfaces. This results in reducing the work contrast. Where a highly luminous object such as an incandescent-lamp filament is reflected from a polished surface, ''reflected glare'' results. Reflections of a large luminous area by matte or semimatte material results in ''veiling glare.'' For work on a horizontal surface, the line of sight from the eye for most people is from 0 to 40° from the vertical, with a peak angle of about 25°. If light from the source is directly reflected into the eyes, contrast and resulting visibility are greatly reduced. When a highly luminous source is directly reflected into the eye, it is possible to completely obliterate a task. Veiling reflections are controlled by proper luminaire design and placement of fixtures. Some designs keep flux out of the 0 to 40° zone. As far as possible the work area should have a

matte surface without shining details, and light should come from the side or behind the worker.

In addition to veiling reflectance, there is a reduction in contrast due to light directly entering the eye from the source. This is called the disability glare effect. It produces a light veil over the image of the task on the retina. It is not a serious problem in interior lighting, but it is important in roadway lighting and similar situations.

Visual-Comfort Criteria High luminances directly or reflected in the field of view can cause discomfort without necessarily interfering with seeing even though visual performance may be impaired. This discomfort glare can be caused by direct glare from sources which have too high a luminance, are inadequately shielded, or have too great an area. Lighting systems are rated by a **visual-comfort probability, VCP,** expressed as a percentage of people who, if seated in the most undesirable location, will be expected to find it acceptable. (For a complete description of VCP, see the IESNA Handbook.) If the following conditions are met, direct glare will not be a problem in lighting installations:

1. The VCP is 70 or more.
2. The ratio of maximum-to-average luminaire luminance does not exceed 5 : 1 (preferably 3 : 1) at 45, 55, 65, 75, and 85° from the nadir crosswise and lengthwise.
3. Maximum luminaire luminances crosswise and lengthwise do not exceed the following values:

Angle above nadir, deg	*Maximum luminance*	
	cd/m²	*fL*
45	7,710	2,250
55	5,500	1,605
65	3,860	1,125
75	2,570	750
85	1,695	495

Design of Interior Lighting Systems

Lighting is as much an art as a science. While many studies have been made on what constitutes adequate lighting along with proper quality, the effect to be achieved depends upon the designer. In this section emphasis will be primarily on achieving adequate illumination.

The design approach is to consider the space to be lighted and the task to be performed. An illuminance is then selected. A suitable luminaire is picked, and calculations are made in order to determine the number and layout of the fixtures. The overall quality is then checked. If unsatisfactory, a new layout is made. An economic study is made to check costs. If these are too high, new layouts are studied until all design restraints are met.

Selection of illuminance Levels

From 1958, the Illuminating Engineering Society (IES) published single-value illuminance levels. Their latest publication, the 1993 ''IESNA Lighting Handbook,'' gives a range of values which permits lighting designers to tailor lighting systems to specific needs. This flexibility permits levels to be adjusted for (1) the visual task; (2) the age of the observers; (3) the need for speed and/or accuracy for visual performance; (4) the reflectance of the task. An illumination-level guide for selected tasks is given in Table 12.5.6. The data are based on an assumption of average conditions for people, tasks, and visual performance requirements. For other conditions see the 1993 ''IESNA Lighting Handbook.''

Room, Furniture, and Equipment Finishes

The color and finish of rooms, furniture, and equipment are important in the overall lighting design. Best results are obtained when the lighting designer coordinates his or her work with the architect, interior decorator, or plant designer.

Table 12.5.6 Illuminance Guide for Selected Tasks

	Footcandles (lm/ft)	Lux (lm/m²)
Commercial drafting		
Conventional	150*	1,600*
Libraries		
Reading good print, typed originals	30	320
Reading small print, handwriting, photocopies	75	800
Active stacks (vertical, illumination)	30	320
Offices		
Conference rooms—conferring	30	320
Conference rooms—typical visual tasks	75–100	800–1,080
Corridors, stairs, elevators	20	220
General tasks, varying difficulty	100	1,080
Lobbies, reception areas	30	320
Private	75	880
Rest rooms	30	320
Video display areas	75	800
School		
Classrooms, laboratories	75	800
Shops	100	1,080
Sight-saving rooms, hearing-impaired classes	150	1,600
Store		
Mass merchandizing, high activity	100	1,080
Self-service	200	2,200
Circulation, low activity	30	320
Feature displays, low, medium, high activity	150,* 300,* 500*	1,600,* 3,200*, 5,400*
Industrial		
Garages		
Repair	75	800
Active traffic areas	15	160
Loading platform	20	220
Machine shops and assembly areas		
Rough bench-machine work, simple assembly	50	540
Medium bench-machine work, moderately difficult assembly	100*	1,080*
Difficult machine work, assembly	150*	1,600*
Fine bench-machine work, assembly	300*	3,200*
Receiving and shipping	30	320
Warehouse storage rooms		
Active large items	15	160
Active small items, labels	30	320
Inactive	5	54
Outdoor areas		
Storage yards		
Active	20	220
Inactive	1	11
Parking areas		
Open, high activity	2	22
Open, medium activity	1	11
Covered parking, pedestrian areas	5	54
Covered night entrance	5	54
Covered day entrance	50	54

SOURCE: Adapted from General Electric Co. design data.

* Requires supplementary lighting. Care should be taken that the supplementary lighting does not introduce direct and reflected glare.

A color scheme should be selected to give light reflectance values as follows:

Area or unit	*Percent reflectance*
Ceilings	70–90
Floors	20–40*
Walls, draperies	40–60†
Bench top, desks, machine, and equipment	25–45

* In storage areas, keep reflectance of aisle floors as high as possible in order to reflect light onto the lower shelves. This should also be done where the underside of objects has to be seen.

† These values should be 30 to 40 where video display terminals (VDTs) are used to avoid veiling reflections in the VDT faces.

The color and finish of a space and equipment therein sets the psychological feel of the space. For example, the trend is away from drab finishes on machinery and dark gray filing cabinets. Colors such as yellow, orange, red, and light gray seem to advance toward the eye. They tend to make large spaces feel smaller. Receding colors such as violet, blue, blue green, and dark grays make small spaces feel larger. Some colors are used for safety purposes. Various areas are painted to designate safe or hazardous locations in a fashion similar to piping identification discussed in Sec. 8. These colors have been carefully standardized in ANSI Z53.1-1979.

Designating Color

In order to be able to obtain designed values of a lighting system, it is necessary to be able to specify the exact color wanted. Many methods have been devised for so doing. One method uses carefully controlled sets of colored chips, each one of which has a particular designation. The desired color is matched against these chips. The **Munsell system** uses *scales of hue* (the actual basic color such as red), *value* (a 10-step scale ranging from black through grays to white), and *chroma* (the amount of gray mixed in with the color). This system is used by many manufacturers to designate their colors. The **Ostwald system** describes color in terms of *color content, white content,* and *black content.* The **Inter-Society Color Council–National Bureau of Standards** (ISCC-NBS) **method** designates one-inch square samples with names. For color designation by the **ICI method,** a spectroradiometric curve of the source is determined together with a spectrophotometric curve of the reflecting or transmitting surface. By mathematical manipulation using spectral tristimulus values, chromaticity coordinates are obtained. See the IESNA Handbook for details. Chromaticity coordinates are extensively used for fluorescent lamp colors. These coordinates can be measured directly by photoelectric colorimeters. They are designed with filter photocell responses to be close to each of the ICI tristimulus values. Built-in logic circuitry results in direct reading of the chromaticity. Incandescent and vapor-type lamp colors are specified by color temperature.

LIGHTING DESIGN

Interior lighting is designed by the **lumen** method. This takes into account the interreflections of light inside a room. The average illuminance on the work plane equals the incident luminous flux ϕ divided by the area, or $E = \phi/A$. Lumens reaching the work plane is equal to lamp lumens multiplied by the **coefficient of utilization** CU. This factor is a function of room size, shape, and finish, mounting height of fixture, and type of luminaire used. The lumens ϕ_L initially available from the lamps may be reduced by ambient temperature, lower voltage, and the ballast used. As time goes by the room surfaces and luminaires become dirty, which further reduces the illuminance. In addition, lamp output falls, and some of them burn out. The total effect of all these factors is expressed by the *light-loss factor* LLF. The maintained illuminance E_m is the initial illumination times the LLF, or

$$E_m = (\phi_L \times \text{CU} \times \text{LLF})/A \qquad (12.5.5)$$

The required maintained illuminance is selected from Table 12.5.6 or from the more extensive data in the IESNA Handbook. A fixture and lamp is selected, and Eq. (12.5.5) is solved for the necessary lamp flux ϕ_L. The number of luminaires N is found by dividing the total lamp lumens ϕ_L by the lumens per fixture ϕ_F. A trial layout is then made. A simple layout keeps spacing between units equal to twice the distance between fixtures and wall. Spacing is checked against the maximum allowable luminaire spacing from manufacturers' data to ensure uniform illumination. However, this criterion results in inadequate lighting near the walls. In order to light desks and benches along the walls, a spacing of 2½ ft from the luminaire center to the wall is used. The ends of fluorescent luminaire rows should be 6 to 12 in from the walls with a maximum distance of 2 ft.

Wall and ceiling cavity luminances can be obtained by using luminance coefficients (LC) for the fixtures (see the IESNA Handbook). For interior areas, maximum luminance ratios should be 3 : 1 or 1 : 3 between tasks and immediate surround, and 10 : 1 or 1 : 10 between tasks and remote surfaces. To ensure eye comfort, the visual-comfort probability (VCP) is investigated.

The **coefficient of utilization** is found by using the **zonal-cavity** method. In this method effects of the room proportion, luminaire suspension lengths, and work-plane height on the CU are found by dividing the room into three cavities as shown in Fig. 12.5.11. For each cavity a cavity ratio is calculated:

$$\text{Cavity ratio} = \frac{5h\,(\text{room length} + \text{room width})}{(\text{room length}) \times (\text{room width})} \qquad (12.5.6)$$

where $h = h_{RC}$ for the room cavity ratio RCR; $= h_{CC}$ for the ceiling cavity ratio CCR; and $= h_{FC}$ for the floor cavity ratio FCR.

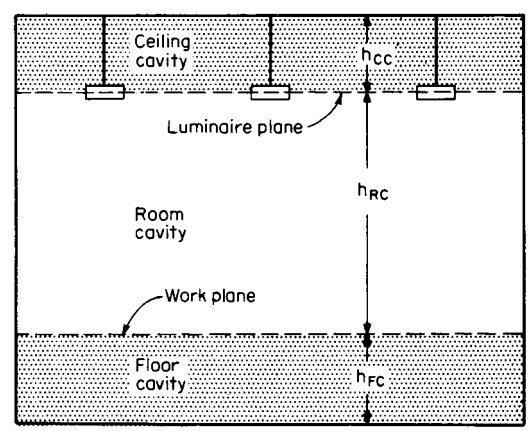


Fig. 12.5.11 The three cavities used in the zonal cavity method.

Table 12.5.7 is used to obtain a single effective ceiling cavity reflectance ρ_{CC} and a single effective floor cavity reflectance ρ_{FC}. For surface-mounted and recessed luminaires, CCR = 0 and the ceiling reflectance is used as ρ_{CC}. Figure 12.5.12 gives CU for selected fixtures. In using Fig. 12.5.12, interpolation may be necessary. Additional fixture data are given in the IES Handbook. Fixture manufacturers furnish such data for their units. Those data should be used for the best accuracy. If the effective floor cavity reflectance ρ_{FC} differs from 20 percent, an adjustment is made by using Table 12.5.8.

For simplicity in calculating the light-loss factor, the effects of ambient temperature, luminaire voltage variation, ballasts, and burnouts will be neglected. Room-surface dirt depreciation factors are shown in Fig. 12.5.13; luminaire dirt depreciation factors are in Fig. 12.5.14. The importance of frequent cleaning is evident. Categories are given for each fixture in Fig. 12.5.12. Lamp lumen depreciation (LLD) depends upon when lamps are replaced before complete burnout. If replacement

Table 12.5.7 Percent Effective Ceiling or Floor Cavity Reflectances for Various Reflectance Combinations (IES)

Cavity ratio	% base* reflectance:	90										80										70										60										50									
	% wall reflectance:	90	80	70	60	50	40	30	20	10	0	90	80	70	60	50	40	30	20	10	0	90	80	70	60	50	40	30	20	10	0	90	80	70	60	50	40	30	20	10	0	90	80	70	60	50	40	30	20	10	0
0.2		89	88	88	87	86	85	85	84	84	82	79	78	78	77	77	76	76	75	74	72	70	69	68	68	67	67	66	66	65	64	60	59	59	59	58	57	56	56	55	53	50	50	49	49	48	48	47	46	46	44
0.4		88	87	86	85	84	83	81	80	79	76	79	77	76	75	74	73	72	71	70	68	69	68	67	66	65	64	63	62	61	58	60	59	59	58	57	55	54	53	52	50	50	49	48	48	47	46	45	45	44	42
0.6		87	86	84	82	80	79	77	76	74	73	78	76	75	73	71	70	68	66	65	63	69	67	65	64	63	61	59	58	57	54	60	58	57	56	55	53	51	51	50	46	50	48	47	46	45	44	43	42	41	38
0.8		87	85	82	80	77	75	73	71	69	67	78	75	73	71	69	67	65	63	61	57	68	66	64	62	60	58	56	55	53	50	59	57	56	55	54	51	48	47	46	43	50	48	47	45	44	42	40	39	38	36
1.0		86	83	80	77	75	72	69	66	64	62	77	74	72	69	67	65	62	60	57	55	68	65	62	60	58	55	53	52	50	47	59	57	55	53	51	48	45	44	43	41	50	48	46	44	43	41	38	37	36	34
1.2		85	82	78	75	72	69	66	63	60	57	76	73	70	67	64	61	58	55	53	51	67	64	61	59	57	54	50	48	46	44	59	56	54	51	49	46	44	42	40	38	50	47	45	43	41	39	36	35	34	29
1.4		85	80	77	73	69	65	62	59	57	52	76	72	68	65	62	59	55	53	50	48	67	63	60	58	55	51	47	45	44	41	59	56	53	49	47	44	41	39	38	36	50	47	45	42	40	38	35	34	32	27
1.6		84	79	75	71	67	63	59	56	53	50	75	71	67	63	60	57	53	50	47	44	67	62	59	56	53	47	45	43	41	38	59	55	52	48	45	42	39	37	35	33	50	47	44	41	39	36	33	32	30	26
1.8		83	78	73	69	64	60	56	53	50	48	75	70	66	62	58	54	50	47	44	41	66	61	58	54	51	46	42	40	38	35	58	55	51	47	44	40	37	35	33	31	50	46	43	40	38	35	31	30	28	25
2.0		83	77	72	67	62	56	53	50	47	43	74	69	64	60	56	52	48	45	41	38	66	60	56	52	49	45	40	38	36	33	58	54	50	46	43	39	35	33	31	29	50	46	43	40	37	34	30	28	26	24
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2.6		81	74	67	62	56	51	46	42	38	35	73	66	60	55	50	45	41	38	34	31	65	59	54	49	45	40	35	33	30	28	58	53	48	43	39	35	31	28	26	24	50	46	41	37	34	30	26	23	21	20
2.8		81	73	66	60	56	49	44	40	36	34	73	65	59	53	48	43	39	36	32	29	65	59	53	48	43	38	33	30	28	26	58	53	47	43	38	34	29	27	24	22	50	46	41	36	33	29	25	22	20	19
3.0		80	72	64	58	52	47	42	38	34	30	72	65	58	52	47	42	37	34	30	27	64	58	52	47	42	37	32	29	27	24	57	52	46	42	37	32	28	25	23	20	50	45	40	36	32	28	24	21	19	17
3.2		79	71	63	56	50	45	40	36	32	28	72	65	57	51	45	40	35	33	28	25	64	58	51	46	40	36	31	28	25	23	57	51	45	41	36	31	27	23	22	18	50	44	39	35	31	27	23	20	18	16
3.4		79	70	62	54	48	43	38	34	30	27	71	64	56	49	44	39	34	32	27	24	64	57	50	45	39	35	29	27	24	22	57	51	45	40	35	30	26	23	20	17	50	44	39	35	30	26	22	19	17	55
3.6		78	69	61	53	47	42	36	32	28	25	71	63	54	48	43	38	32	30	25	23	63	56	49	44	38	33	28	25	22	20	57	50	44	39	34	29	25	22	19	16	50	44	39	34	29	25	21	18	16	14
3.8		78	69	60	51	45	40	35	31	27	23	70	62	53	47	41	36	31	28	24	22	63	56	49	43	37	32	27	24	21	19	57	50	43	38	33	29	24	21	19	15	50	44	38	34	29	25	21	17	15	13
4.0		77	69	58	51	44	39	33	29	25	22	70	61	53	46	40	35	30	26	22	20	63	55	48	42	36	31	26	23	20	17	57	49	42	37	32	28	23	20	18	14	50	44	38	33	28	24	20	17	15	12
4.2		77	62	57	50	43	37	32	28	24	21	69	60	52	45	39	34	29	25	21	18	62	55	47	41	35	30	25	22	19	16	56	49	42	37	32	27	22	19	17	14	50	43	37	32	28	24	20	17	14	12
4.4		76	61	56	49	42	36	31	27	23	20	69	60	51	44	38	33	28	24	20	17	62	54	46	40	34	29	24	21	18	15	56	49	42	36	31	27	22	19	16	13	50	43	37	32	27	23	19	16	13	11
4.6		76	60	55	47	40	35	30	26	22	19	69	59	50	43	37	32	27	23	19	15	62	53	45	39	33	28	24	21	17	14	56	49	41	35	30	26	21	18	16	13	50	43	36	31	26	22	18	15	13	10
4.8		75	59	54	46	39	34	28	25	21	18	68	58	49	42	36	31	26	22	18	14	62	53	45	38	32	27	23	20	16	13	56	48	41	34	29	25	21	18	15	12	50	43	36	31	26	22	18	15	12	09
5.0		75	59	53	45	38	33	28	24	20	16	68	58	48	41	35	30	25	21	18	14	61	52	44	36	31	26	22	19	16	12	56	48	40	34	28	24	20	17	14	11	50	42	35	30	25	21	17	14	12	09
6.0		73	61	49	41	34	29	24	20	16	11	66	55	44	38	31	27	22	19	15	10	60	51	41	35	28	24	19	16	13	09	55	45	37	31	25	21	17	14	11	07	50	42	34	29	23	19	15	13	10	06
7.0		70	58	45	38	30	27	21	18	14	08	64	53	41	35	28	24	19	16	12	07	58	48	38	32	26	22	17	14	11	06	54	43	35	30	24	20	15	12	09	05	49	41	32	27	21	18	14	11	08	05
8.0		68	55	42	35	27	23	18	15	12	06	62	50	38	32	25	21	17	14	11	05	57	46	35	29	23	19	15	13	10	05	53	42	33	28	22	18	14	11	08	04	49	40	30	25	19	16	12	10	07	03
9.0		66	52	38	31	25	21	16	14	11	05	61	49	36	30	23	19	15	13	10	04	56	45	33	27	21	18	14	12	09	04	52	40	31	26	20	16	12	10	07	03	48	39	29	24	18	15	11	09	07	03
10.0		65	51	36	29	22	19	15	11	09	04	59	46	33	27	21	18	14	11	08	03	55	43	31	25	19	16	12	10	08	03	51	39	29	24	18	15	11	09	07	02	47	37	27	22	17	14	10	08	06	02

* Ceiling, floor, or floor of cavity.

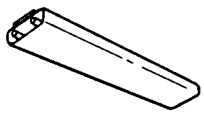
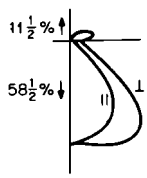
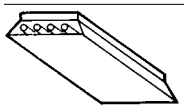
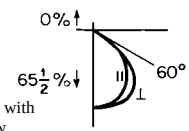
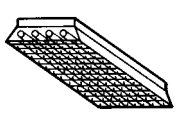
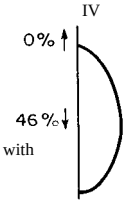
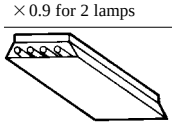
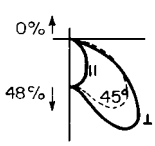
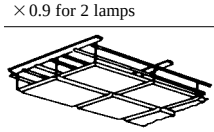
Table 12.5.7 Percent Effective Ceiling or Floor Cavity Reflectance for Various Reflectance Combinations (*Continued*)

Cavity ratio	% base* reflectance:	40										30										20										10										0									
	% wall reflectance:	90	80	70	60	50	40	30	20	10	0	90	80	70	60	50	40	30	20	10	0	90	80	70	60	50	40	30	20	10	0	90	80	70	60	50	40	30	20	10	0	90	80	70	60	50	40	30	20	10	0
0.2		40	40	39	39	39	38	38	37	36	36	31	31	30	30	29	29	29	28	28	27	21	20	20	20	20	20	19	19	19	17	11	11	11	10	10	10	10	09	09	09	02	02	02	01	01	01	01	00	00	0
0.4		41	40	39	39	38	37	36	35	34	34	31	31	30	30	29	28	28	27	26	25	22	21	20	20	20	19	19	18	18	16	12	11	11	11	11	10	10	09	09	08	04	03	03	02	02	02	01	01	00	0
0.6		41	40	39	38	37	36	34	33	32	31	32	31	30	29	28	27	26	26	25	23	23	21	21	20	19	19	18	18	17	15	13	13	12	11	11	10	10	09	08	08	05	05	04	03	03	02	02	01	01	0
0.8		41	40	38	37	36	35	33	32	31	29	32	31	30	29	28	26	25	25	23	22	24	22	21	20	19	19	18	17	16	14	15	14	13	12	11	10	10	09	08	07	07	06	05	04	04	03	02	02	01	0
1.0		42	40	38	37	35	33	32	31	29	27	33	32	30	29	27	25	24	23	22	20	25	23	22	20	19	18	17	16	15	13	16	14	13	12	12	11	10	09	08	07	08	07	06	05	04	03	02	02	01	0
1.2		42	40	38	36	34	32	30	29	27	25	33	32	30	28	27	25	23	22	21	19	25	23	22	20	19	17	17	16	14	12	17	15	14	13	12	11	10	09	07	06	10	08	07	06	05	04	03	02	01	0
1.4		42	39	37	35	33	31	29	27	25	23	34	32	30	28	26	24	22	21	19	18	26	24	22	20	18	17	16	15	13	12	18	16	14	13	12	11	10	09	07	06	11	09	08	07	06	04	03	02	01	0
1.6		42	39	37	35	32	30	27	25	23	22	34	33	29	27	25	23	22	20	18	17	26	24	22	20	18	17	16	15	13	11	19	17	15	14	12	11	09	08	07	06	12	10	09	07	06	05	03	02	01	0
1.8		42	39	36	34	31	29	26	24	22	21	35	33	29	27	25	23	21	19	17	16	27	25	23	20	18	17	15	14	12	10	19	17	15	14	13	11	09	08	06	05	13	11	09	08	07	05	04	03	01	0
2.0		42	39	36	34	31	28	25	23	21	19	35	33	29	26	24	22	20	18	16	14	28	25	23	20	18	16	15	13	11	09	20	18	16	14	13	11	09	98	06	05	14	12	10	09	07	05	04	03	01	0
2.2		42	39	36	33	30	27	24	22	19	18	36	32	29	26	24	22	19	17	15	13	28	25	23	20	18	16	14	12	10	09	21	19	16	14	13	11	09	07	06	05	15	13	11	09	07	06	04	03	02	0
2.4		43	39	35	33	29	27	24	21	18	17	36	32	29	26	24	22	19	16	14	12	29	26	23	20	18	16	14	12	10	08	22	19	17	15	13	11	09	07	06	05	16	13	11	09	08	06	04	03	01	0
2.6		43	39	35	32	29	26	23	20	17	15	36	32	29	25	23	21	18	16	14	12	29	26	23	20	18	16	14	11	09	08	23	20	17	15	13	11	09	07	06	04	17	14	12	10	08	06	05	03	02	0
2.8		43	39	35	32	28	25	22	19	16	14	37	33	29	25	23	21	17	15	13	11	30	27	23	20	18	15	13	11	09	07	23	20	18	16	13	11	09	07	05	03	17	15	13	10	08	07	05	03	02	0
3.0		43	39	35	31	27	24	21	18	16	13	37	33	29	25	22	20	17	15	12	10	30	27	23	20	17	15	13	11	09	07	24	21	18	16	13	11	09	07	05	03	18	16	13	11	09	07	05	03	02	0
3.2		43	39	35	31	27	23	20	17	15	13	37	33	29	25	22	19	16	14	12	10	31	27	23	20	17	15	12	11	09	06	25	21	18	16	13	11	09	07	05	03	19	16	14	11	09	07	05	03	02	0
3.4		43	39	34	30	26	23	20	17	14	12	37	33	29	25	22	19	16	14	11	09	31	27	23	20	17	15	12	10	08	06	26	22	18	16	13	11	09	07	05	03	20	17	14	12	09	07	05	03	02	0
3.6		44	39	34	30	26	22	19	16	14	11	38	33	29	24	21	18	15	13	10	09	32	27	23	20	17	15	12	10	08	05	26	22	19	16	13	11	09	06	04	03	20	17	15	12	10	08	05	04	02	0
3.8		44	38	33	29	25	22	18	16	13	10	38	33	28	24	21	18	15	13	10	08	32	28	23	20	17	15	12	10	07	05	27	23	19	17	14	11	09	06	04	02	21	18	15	12	10	08	05	04	02	0
4.0		44	38	33	29	25	21	18	15	12	10	38	33	28	24	21	18	14	12	09	07	33	28	23	20	17	14	11	09	07	05	27	23	20	17	14	11	09	06	04	02	22	18	15	13	10	08	05	04	02	0
4.2		44	38	33	29	24	21	17	15	12	10	38	33	28	24	20	17	14	12	09	07	33	28	23	20	17	14	11	09	07	04	28	24	20	17	14	11	09	06	04	02	22	19	16	13	10	08	06	04	02	0
4.4		44	38	33	28	24	21	17	14	11	09	39	33	28	24	20	17	14	11	09	06	34	28	24	20	17	14	11	09	07	04	28	24	20	17	14	11	08	06	04	02	23	19	16	13	10	8	06	04	02	0
4.6		44	38	32	28	23	19	16	14	11	08	39	33	28	24	20	17	13	10	08	06	34	29	24	20	17	14	11	09	07	04	29	25	20	17	14	11	08	06	04	02	23	20	17	13	11	08	06	04	02	0
4.8		44	38	32	27	22	19	16	13	10	08	39	33	28	24	20	17	13	10	08	05	35	29	24	20	17	13	10	08	06	04	29	25	20	17	14	11	08	06	04	02	24	20	17	14	11	08	06	04	02	0
5.0		45	38	31	27	22	19	15	13	10	07	39	33	28	24	19	16	13	10	08	05	35	29	24	20	16	13	10	08	06	04	30	25	20	17	14	11	08	06	04	02	25	21	17	14	11	08	06	04	02	0
6.0		44	37	30	25	20	17	13	11	08	05	39	33	27	23	18	15	11	09	06	04	36	30	24	20	16	13	10	08	05	02	31	26	21	17	14	11	08	06	03	01	27	23	18	15	12	09	06	04	02	0
7.0		44	36	29	24	19	16	12	10	07	04	40	33	26	22	17	14	10	08	05	03	36	30	24	20	15	12	09	07	04	02	32	27	21	17	13	11	08	06	03	01	28	24	19	15	12	09	06	04	02	0
8.0		44	35	28	23	18	15	11	09	06	03	40	33	26	21	16	13	09	07	04	02	37	30	23	19	15	12	08	06	03	01	33	27	21	17	13	10	07	05	03	01	30	25	20	15	12	09	06	04	02	0
9.0		44	35	26	21	16	12	10	08	05	02	40	33	25	20	15	12	09	07	04	02	37	29	23	19	14	11	08	06	03	01	34	28	21	17	13	10	07	05	02	01	31	25	20	15	12	09	06	04	02	0
10.0		43	34	25	20	15	12	08	07	05	02	40	32	24	19	14	11	08	06	03	01	37	29	22	18	13	10	07	05	03	01	34	28	21	17	12	10	07	05	02	01	31	25	20	15	12	09	06	04	02	0

* Ceiling, floor, or floor of cavity.

Typical luminaire	Typical distribution and % lamp lumens	Maint. cat.	Max *S/MH* guide§	RCR‡ ↓	ρ_{CC}*→	80			70			50			30			10			0
					ρ_W†→	50	30	10	50	30	10	50	30	10	50	30	10	50	30	10	0
						Coefficients of utilization for 20% effective floor cavity reflectance ($\rho FC = 20$)															
Porcelain-enamel ventilated standard dome with incandescent lamp	0% ↑ 83½% ↓	IV	1.3	0		.99	.99	.99	.97	.97	.97	.93	.93	.93	.89	.89	.89	.85	.85	.85	.83
				1		.87	.84	.81	.85	.82	.79	.82	.79	.77	.79	.76	.74	.76	.74	.72	.71
				2		.76	.70	.65	.74	.69	.65	.71	.67	.63	.69	.65	.62	.66	.63	.60	.59
				3		.66	.59	.54	.65	.59	.53	.62	.57	.53	.60	.56	.52	.58	.54	.51	.49
				4		.58	.51	.45	.57	.50	.45	.55	.49	.44	.53	.48	.44	.51	.47	.43	.41
				5		.52	.44	.39	.51	.44	.38	.49	.43	.38	.47	.42	.37	.46	.41	.37	.35
				6		.46	.39	.33	.46	.38	.33	.44	.38	.33	.43	.37	.33	.41	.36	.32	.31
				7		.42	.34	.29	.41	.34	.29	.40	.33	.29	.39	.33	.29	.38	.32	.28	.27
				8		.38	.31	.26	.37	.31	.26	.36	.30	.26	.35	.30	.25	.34	.29	.25	.24
				9		.35	.28	.23	.34	.28	.23	.33	.27	.23	.32	.27	.23	.32	.26	.23	.21
				10		.32	.25	.21	.32	.25	.21	.31	.25	.21	.30	.24	.21	.29	.24	.20	.19
EAR-38 lamp above 2″ (51 mm) diameter aperture (increase efficiency to 54½% for 3″ (76 mm) diameter aperture	0% ↑ 44½% ↓	IV	0.7	0		.52	.52	.52	.51	.51	.51	.48	.48	.48	.46	.46	.46	.45	.45	.45	.44
				1		.49	.48	.47	.48	.47	.46	.46	.45	.45	.44	.44	.43	.43	.43	.42	.41
				2		.46	.44	.43	.45	.44	.43	.44	.43	.42	.43	.42	.41	.41	.41	.40	.39
				3		.43	.41	.40	.43	.41	.40	.42	.40	.39	.41	.39	.38	.40	.39	.38	.37
				4		.41	.39	.37	.41	.39	.37	.40	.38	.37	.39	.37	.36	.38	.37	.36	.35
				5		.39	.37	.35	.39	.37	.35	.38	.36	.35	.37	.36	.34	.36	.35	.34	.34
				6		.37	.35	.33	.37	.35	.33	.36	.34	.33	.35	.34	.33	.35	.34	.32	.32
				7		.35	.33	.31	.35	.33	.31	.34	.33	.31	.34	.32	.31	.33	.32	.31	.30
				8		.34	.31	.30	.33	.31	.30	.33	.31	.30	.32	.31	.29	.32	.31	.29	.29
				9		.32	.30	.28	.32	.30	.28	.31	.30	.28	.31	.29	.28	.31	.29	.28	.28
				10		.31	.28	.27	.31	.28	.27	.30	.28	.27	.30	.28	.27	.30	.28	.27	.26
Enclosed reflector with an incandescent lamp	0% ↑ 71½% ↓	V	1.4	0		.85	.85	.85	.83	.83	.83	.80	.80	.80	.76	.76	.76	.73	.73	.73	.72
				1		.77	.75	.73	.76	.74	.72	.73	.71	.69	.70	.69	.67	.67	.66	.65	.64
				2		.70	.66	.63	.68	.65	.62	.66	.63	.60	.64	.61	.59	.61	.60	.58	.56
				3		.63	.58	.54	.62	.57	.54	.60	.56	.53	.58	.54	.52	.56	.53	.51	.50
				4		.56	.51	.47	.56	.51	.47	.54	.50	.46	.52	.49	.46	.51	.48	.45	.44
				5		.51	.46	.42	.50	.45	.41	.49	.44	.41	.48	.44	.40	.46	.43	.40	.39
				6		.46	.41	.37	.46	.41	.37	.45	.40	.36	.43	.39	.36	.42	.39	.36	.34
				7		.42	.37	.33	.42	.37	.33	.41	.36	.33	.40	.36	.32	.39	.35	.32	.31
				8		.39	.33	.30	.38	.33	.29	.37	.33	.29	.37	.32	.29	.36	.32	.29	.28
				9		.36	.30	.27	.35	.30	.27	.35	.30	.27	.34	.30	.26	.33	.29	.26	.25
				10		.33	.28	.24	.33	.28	.24	.32	.27	.24	.31	.27	.24	.31	.27	.24	.23
Diffuse aluminum reflector with 35″ CW shielding	17% ↑ 66% ↓ II	II	1.5/1.3	0		.95	.95	.95	.91	.91	.91	.83	.83	.83	.76	.76	.76	.69	.69	.69	.66
				1		.85	.82	.79	.81	.79	.76	.75	.73	.71	.69	.67	.66	.63	.62	.61	.58
				2		.75	.71	.67	.72	.68	.65	.67	.63	.61	.62	.59	.57	.57	.55	.53	.51
				3		.67	.61	.57	.65	.59	.55	.60	.56	.52	.55	.52	.49	.51	.49	.46	.44
				4		.60	.54	.49	.58	.52	.48	.54	.49	.45	.50	.46	.43	.46	.43	.41	.39
				5		.54	.47	.43	.52	.46	.42	.49	.43	.40	.45	.41	.38	.42	.39	.36	.34
				6		.49	.42	.37	.47	.41	.37	.44	.39	.35	.41	.37	.33	.38	.35	.32	.30
				7		.44	.38	.33	.43	.37	.32	.40	.35	.31	.38	.33	.30	.35	.31	.28	.27
				8		.40	.34	.29	.39	.33	.29	.37	.31	.28	.34	.30	.27	.32	.28	.26	.24
				9		.37	.31	.26	.36	.30	.26	.34	.29	.25	.32	.27	.24	.30	.26	.23	.21
				10		.34	.28	.24	.33	.27	.23	.31	.26	.23	.29	.25	.22	.28	.24	.21	.19
Diffuse aluminum reflector with 35″CW × 35″LW shielding	17% ↑ 56½% ↓ II	II	1.5/1.1	0		.91	.91	.91	.86	.86	.86	.77	.77	.77	.68	.68	.68	.61	.61	.61	.57
				1		.80	.77	.75	.76	.74	.71	.69	.67	.65	.62	.60	.59	.55	.54	.53	.50
				2		.71	.67	.63	.68	.64	.60	.61	.58	.55	.55	.53	.51	.50	.48	.46	.43
				3		.63	.58	.53	.60	.55	.51	.55	.51	.47	.50	.46	.44	.45	.42	.40	.38
				4		.57	.51	.46	.54	.49	.44	.49	.45	.41	.45	.41	.38	.41	.38	.35	.33
				5		.51	.45	.40	.49	.43	.39	.45	.40	.36	.41	.37	.34	.37	.34	.31	.29
				6		.46	.40	.35	.44	.38	.34	.41	.36	.32	.37	.33	.30	.34	.30	.28	.26
				7		.42	.36	.31	.40	.35	.30	.37	.32	.29	.34	.30	.27	.31	.28	.25	.23
				8		.38	.32	.28	.37	.31	.27	.34	.29	.26	.31	.27	.24	.29	.25	.23	.21
				9		.35	.29	.25	.34	.28	.25	.31	.27	.23	.29	.25	.22	.27	.23	.21	.19
				10		.33	.27	.23	.31	.26	.22	.29	.24	.21	.27	.23	.20	.25	.21	.19	.17
2 lamp, surface-mounted, bare lamp unit—photometry with 18-in-wide panel above luminaire (lamps on 6-in centers)	9½% ↑ 78% ↓ II	I	1.3	0		1.02	1.02	1.02	.99	.99	.99	.92	.92	.92	.86	.86	.86	.81	.81	.81	.78
				1		.85	.80	.76	.82	.78	.74	.76	.73	.70	.71	.68	.66	.67	.64	.62	.60
				2		.72	.65	.59	.70	.63	.58	.65	.60	.55	.61	.56	.52	.57	.53	.50	.47
				3		.63	.55	.48	.60	.53	.47	.56	.50	.45	.53	.47	.43	.49	.45	.41	.38
				4		.55	.46	.40	.53	.45	.39	.50	.43	.37	.46	.41	.36	.43	.38	.34	.32
				5		.49	.40	.34	.47	.39	.33	.44	.37	.32	.41	.35	.31	.39	.34	.29	.27
				6		.43	.35	.29	.42	.34	.29	.40	.33	.28	.37	.31	.27	.35	.30	.26	.23
				7		.39	.31	.25	.38	.30	.25	.36	.29	.24	.34	.28	.23	.32	.26	.22	.20
				8		.36	.28	.22	.35	.27	.22	.33	.26	.21	.31	.25	.21	.29	.24	.20	.18
				9		.33	.25	.20	.32	.25	.20	.30	.24	.19	.28	.23	.18	.27	.22	.18	.16
				10		.30	.23	.18	.29	.22	.18	.28	.21	.17	.26	.21	.17	.25	.20	.16	.14

Fig. 12.5.12 Coefficients of utilization for typical luminaires. (*Abstracted from IESNA Handbook, 8th ed., 1993. Reprinted with permission. This material has been modified from its original version and is not reflective of its original form as recognized by the IESNA.*)

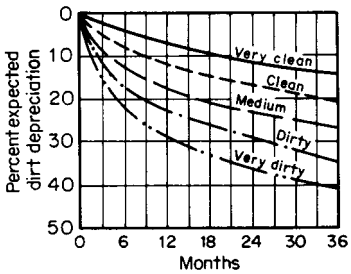
Typical luminaire	Typical distribution and % lamp lumens	Maint. cat.	Max *S/MH* guide§	ρ_{CC}*→	80			70			50			30			10			0
				ρ_W†→	50	30	10	50	30	10	50	30	10	50	30	10	50	30	10	0
				RCR‡ ↓	Coefficients of utilization for 20% effective floor cavity reflectance ($\rho FC = 20$)															
		V	1.5/1.2	0	.81	.81	.81	.78	.78	.78	.72	.72	.72	.66	.66	.66	.61	.61	.61	.59
				1	.71	.68	.66	.68	.66	.63	.63	.61	.59	.58	.57	.56	.54	.53	.52	.50
				2	.63	.58	.55	.60	.56	.53	.56	.53	.50	.52	.50	.47	.48	.46	.45	.43
				3	.56	.50	.46	.54	.49	.45	.50	.46	.43	.47	.43	.41	.43	.41	.39	.37
				4	.50	.44	.40	.48	.43	.39	.45	.40	.37	.42	.38	.35	.39	.36	.34	.32
				5	.45	.39	.34	.43	.38	.34	.40	.36	.32	.38	.34	.31	.35	.32	.30	.28
				6	.40	.34	.30	.39	.34	.30	.37	.32	.28	.34	.30	.27	.32	.29	.26	.25
				7	.37	.31	.27	.35	.30	.26	.33	.29	.25	.31	.27	.24	.30	.26	.23	.22
				8	.33	.28	.24	.32	.27	.23	.30	.26	.23	.29	.25	.22	.27	.24	.21	.20
2 lamp prismatic wraparound—multiply by 0.95 for 4 amps				9	.31	.25	.21	.30	.25	.21	.28	.24	.20	.26	.23	.20	.25	.22	.19	.18
				10	.28	.23	.19	.27	.22	.19	.26	.21	.18	.24	.21	.18	.22	.20	.17	.16
		V	1.4/1.3	0	.71	.71	.71	.70	.70	.70	.66	.66	.66	.64	.64	.64	.61	.61	.61	.60
				1	.64	.62	.60	.63	.61	.60	.60	.59	.58	.58	.57	.56	.56	.55	.54	.53
				2	.57	.54	.51	.56	.53	.51	.54	.52	.50	.52	.50	.48	.51	.49	.47	.46
				3	.51	.47	.44	.50	.46	.43	.49	.45	.43	.47	.44	.42	.46	.43	.41	.40
				4	.46	.41	.38	.45	.41	.37	.44	.40	.37	.42	.39	.36	.41	.38	.36	.35
				5	.41	.36	.33	.40	.36	.32	.39	.35	.32	.38	.35	.32	.37	.34	.31	.30
4 lamp, 2′ (610 mm) wide unit with sharp cutoff (high angle—low luminance) flat prismatic lens—×1.05 for 3 lamps				6	.37	.32	.28	.36	.32	.28	.35	.31	.28	.34	.31	.28	.34	.30	.28	.27
				7	.33	.29	.25	.33	.28	.25	.32	.28	.25	.31	.27	.25	.30	.27	.24	.23
				8	.30	.26	.22	.30	.25	.22	.29	.25	.22	.28	.25	.22	.28	.24	.22	.21
				9	.28	.23	.20	.27	.23	.20	.27	.23	.20	.26	.22	.20	.25	.22	.19	.18
×0.9 for 2 lamps				10	.25	.21	.18	.25	.21	.18	.25	.20	.18	.24	.20	.18	.23	.20	.18	.17
		IV	0.9	0	.55	.55	.55	.54	.54	.54	.51	.51	.51	.49	.49	.49	.47	.47	.47	.46
				1	.49	.48	.46	.48	.47	.46	.46	.45	.44	.45	.44	.43	.43	.42	.42	.41
				2	.44	.42	.40	.43	.41	.39	.42	.40	.38	.40	.39	.37	.39	.38	.37	.36
				3	.40	.37	.34	.39	.36	.34	.38	.36	.33	.37	.35	.33	.36	.34	.32	.32
				4	.36	.33	.30	.36	.33	.30	.35	.32	.30	.34	.31	.29	.33	.31	.29	.28
				5	.33	.30	.27	.33	.29	.27	.32	.29	.27	.31	.28	.26	.30	.28	.26	.25
				6	.30	.27	.24	.30	.27	.24	.29	.26	.24	.29	.26	.24	.28	.25	.24	.23
4 lamp, 2′ (610 mm) wide troffer with 45° white metal louver—×1.05 for 3 lamps				7	.28	.25	.22	.28	.24	.22	.27	.24	.22	.26	.24	.22	.26	.23	.22	.21
				8	.26	.23	.20	.26	.22	.20	.25	.22	.20	.25	.22	.20	.24	.22	.20	.19
				9	.24	.21	.19	.24	.21	.19	.23	.20	.18	.23	.20	.18	.23	.20	.18	.18
×0.9 for 2 lamps				10	.23	.19	.17	.22	.19	.17	.22	.19	.17	.22	.19	.17	.21	.19	.17	.16
		V	N.A.	0	.57	.57	.57	.56	.56	.56	.53	.53	.53	.51	.51	.51	.49	.49	.49	.48
				1	.50	.48	.46	.49	.47	.45	.47	.45	.44	.45	.43	.42	.43	.42	.41	.40
				2	.43	.40	.37	.42	.39	.36	.40	.38	.35	.39	.37	.35	.37	.36	.34	.33
				3	.37	.33	.30	.37	.33	.30	.35	.32	.29	.34	.31	.29	.33	.30	.28	.27
				4	.33	.28	.25	.32	.28	.25	.31	.27	.24	.30	.27	.24	.29	.26	.24	.23
Bilateral batwing distribution—4 lamp, 2′ (610 mm) wide fluorescent unit with flat prismatic lens and overlay—×1.05 for 3 lamps				5	.29	.24	.21	.28	.24	.21	.27	.24	.21	.26	.23	.20	.25	.23	.20	.19
				6	.26	.21	.18	.25	.21	.18	.24	.21	.18	.24	.20	.18	.23	.20	.17	.16
				7	.23	.19	.16	.23	.18	.15	.22	.18	.15	.21	.18	.15	.21	.17	.15	.14
				8	.21	.17	.14	.21	.16	.14	.20	.16	.13	.19	.16	.13	.19	.16	.13	.12
				9	.19	.15	.12	.19	.15	.12	.18	.14	.12	.18	.14	.12	.17	.14	.12	.11
×0.9 for 2 lamps				10	.17	.13	.11	.17	.13	.11	.17	.13	.11	.16	.13	.11	.16	.13	.10	.10
		ρ_{cc} from below ~65%		1			.60	.58	.56	.58	.56	.54								
Diffusing plastic or glass				2			.53	.49	.45	.51	.47	.43								
1. Ceiling efficiency ~60%; diffuser transmittance ≈50%; diffuser				3			.47	.42	.37	.45	.41	.36								
				4			.41	.36	.32	.39	.35	.31								
reflectance ~40%. Cavity with minimum obstructions and painted				5			.37	.31	.27	.35	.30	.26								
				6			.33	.27	.23	.31	.26	.23								
with 80% reflectance paints—use $pc = 70$				7			.29	.24	.20	.28	.23	.20								
				8			.26	.21	.18	.25	.20	.17								
2. For lower reflectance paint or obstructions—use $\rho c = 50$				9			.23	.19	.15	.23	.18	.15								
				10			.21	.17	.13	.21	.16	.13								

* ρ_{cc} = percent effective ceiling cavity reflectance.
† ρ_w = percent wall reflectance.
‡ RCR = room cavity ratio.
§ Maximum *S/MH* guide—ratio of maximum luminaire spacing to mounting or ceiling height above work plane.

Fig. 12.5.12 (*Continued*)

Table 12.5.8 Multiplying Factors for Other than 20 Percent Effective Floor Cavity Reflectance (IES)

% effective ceiling cavity reflectance ρ_{CC}:	80				70				50			30			10		
% wall reflectance ρ_W:	70	50	30	10	70	50	30	10	50	30	10	50	30	10	50	30	10
Room cavity ratio	For 30% effective floor cavity reflectance (20% = 1.00)																
1	1.092	1.082	1.075	1.068	1.077	1.070	1.064	1.059	1.049	1.044	1.040	1.028	1.026	1.023	1.012	1.010	1.008
2	1.079	1.066	1.055	1.047	1.068	1.057	1.048	1.039	1.041	1.033	1.027	1.026	1.021	1.017	1.013	1.010	1.006
3	1.070	1.054	1.042	1.033	1.061	1.048	1.037	1.028	1.034	1.027	1.020	1.024	1.017	1.012	1.014	1.009	1.005
4	1.062	1.045	1.033	1.024	1.055	1.040	1.029	1.021	1.030	1.022	1.015	1.022	1.015	1.010	1.014	1.009	1.004
5	1.056	1.038	1.026	1.018	1.050	1.034	1.024	1.015	1.027	1.018	1.012	1.020	1.013	1.008	1.014	1.009	1.004
6	1.052	1.033	1.021	1.014	1.047	1.030	1.020	1.012	1.024	1.015	1.009	1.019	1.012	1.006	1.014	1.008	1.003
7	1.047	1.029	1.018	1.011	1.043	1.026	1.017	1.009	1.022	1.013	1.007	1.018	1.010	1.005	1.014	1.008	1.003
8	1.044	1.026	1.015	1.009	1.040	1.024	1.015	1.007	1.020	1.012	1.006	1.017	1.009	1.004	1.013	1.007	1.003
9	1.040	1.024	1.014	1.007	1.037	1.022	1.014	1.006	1.019	1.011	1.005	1.016	1.009	1.004	1.013	1.007	1.002
10	1.037	1.022	1.012	1.006	1.034	1.020	1.012	1.005	1.017	1.010	1.004	1.015	1.009	1.003	1.013	1.007	1.002
	For 10% effective floor cavity reflectance (20% = 1.00)																
1	0.923	0.929	0.935	0.940	0.933	0.939	0.943	0.948	0.956	0.960	0.963	0.973	0.976	0.979	0.989	0.991	0.993
2	0.931	0.942	0.950	0.958	0.940	0.949	0.957	0.963	0.962	0.968	0.974	0.976	0.980	0.985	0.988	0.991	0.995
3	0.939	0.951	0.961	0.969	0.945	0.057	0.966	0.973	0.967	0.975	0.981	0.978	0.983	0.988	0.988	0.992	0.996
4	0.944	0.958	0.969	0.978	0.950	0.963	0.973	0.980	0.972	0.980	0.986	0.980	0.986	0.991	0.987	0.992	0.996
5	0.949	0.964	0.976	0.983	0.954	0.968	0.978	0.985	0.975	0.983	0.989	0.981	0.988	0.993	0.987	0.992	0.997
6	0.953	0.969	0.980	0.986	0.958	0.972	0.982	0.989	0.977	0.985	0.992	0.982	0.989	0.995	0.987	0.993	0.997
7	0.957	0.973	0.983	0.991	0.961	0.975	0.985	0.991	0.979	0.987	0.994	0.983	0.990	0.996	0.987	0.993	0.998
8	0.960	0.976	0.986	0.993	0.963	0.977	0.987	0.993	0.981	0.988	0.995	0.984	0.991	0.997	0.987	0.994	0.998
9	0.963	0.978	0.987	0.994	0.965	0.979	0.989	0.994	0.983	0.990	0.996	0.985	0.992	0.998	0.988	0.994	0.999
10	0.965	0.980	0.989	0.995	0.967	0.981	0.990	0.995	0.984	0.991	0.997	0.986	0.993	0.998	0.988	0.994	0.999
	For 0% effective floor cavity reflectance (20% = 1.00)																
1	0.859	0.870	0.879	0.886	0.873	0.884	0.893	0.901	0.916	0.923	0.929	0.948	0.954	0.960	0.979	0.983	0.987
2	0.871	0.887	0.903	0.919	0.886	0.902	0.916	0.928	0.926	0.938	0.949	0.954	0.963	0.971	0.978	0.983	0.991
3	0.882	0.904	0.915	0.942	0.898	0.918	0.934	0.947	0.936	0.950	0.964	0.958	0.969	0.979	0.976	0.984	0.993
4	0.893	0.919	0.941	0.958	0.908	0.930	0.948	0.961	0.945	0.961	0.974	0.961	0.974	0.984	0.975	0.985	0.994
5	0.903	0.931	0.953	0.969	0.914	0.939	0.958	0.970	0.951	0.967	0.980	0.964	0.977	0.988	0.975	0.985	0.995
6	0.911	0.940	0.961	0.976	0.920	0.945	0.965	0.977	0.955	0.972	0.985	0.966	0.979	0.991	0.975	0.986	0.996
7	0.917	0.947	0.967	0.981	0.924	0.950	0.970	0.982	0.959	0.975	0.988	0.968	0.981	0.993	0.975	0.987	0.997
8	0.922	0.953	0.971	0.985	0.929	0.955	0.975	0.986	0.963	0.978	0.991	0.970	0.983	0.995	0.976	0.988	0.998
9	0.928	0.958	0.975	0.988	0.933	0.959	0.980	0.989	0.966	0.980	0.993	0.971	0.985	0.996	0.976	0.988	0.998
10	0.933	0.962	0.979	0.991	0.937	0.963	0.983	0.992	0.969	0.982	0.995	0.973	0.987	0.997	0.977	0.989	0.999



% expected dirt depreciation:	Luminaire distribution type																			
	Direct				Semidirect				Direct-indirect				Semi-indirect				Indirect			
Room cavity ratio	10	20	30	40	10	20	30	40	10	20	30	40	10	20	30	40	10	20	30	40
1	.98	.96	.94	.92	.97	.92	.89	.84	.94	.87	.80	.76	.94	.87	.80	.73	.90	.80	.70	.60
2	.98	.96	.94	.92	.96	.92	.88	.83	.94	.87	.80	.75	.94	.87	.79	.72	.90	.80	.69	.59
3	.98	.95	.93	.90	.96	.91	.87	.82	.94	.86	.79	.74	.94	.86	.78	.71	.90	.79	.68	.58
4	.97	.95	.92	.90	.95	.90	.85	.80	.94	.86	.79	.73	.94	.86	.78	.70	.89	.78	.67	.56
5	.97	.94	.91	.89	.94	.90	.84	.79	.93	.86	.78	.72	.93	.86	.77	.69	.89	.78	.66	.55
6	.97	.94	.91	.88	.94	.89	.83	.78	.93	.85	.78	.71	.93	.85	.76	.68	.89	.77	.66	.54
7	.97	.94	.90	.87	.93	.88	.82	.77	.93	.84	.77	.70	.93	.84	.76	.68	.89	.76	.65	.53
8	.96	.93	.89	.86	.93	.87	.81	.75	.93	.84	.76	.69	.93	.84	.76	.68	.88	.76	.64	.52
9	.96	.92	.88	.85	.93	.87	.80	.74	.93	.84	.76	.68	.93	.84	.75	.67	.88	.75	.63	.51
10	.96	.92	.87	.83	.93	.86	.79	.72	.93	.84	.75	.67	.92	.83	.75	.67	.88	.75	.62	.50

Fig. 12.5.13 Room surface dirt depreciation factors. *(IES)*

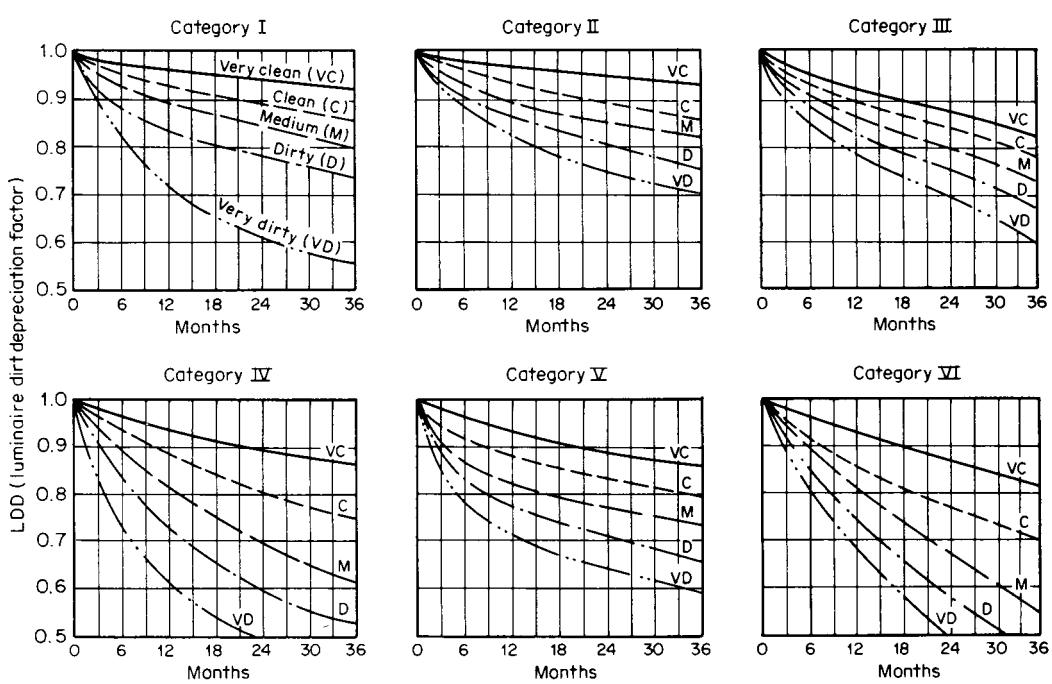


Fig. 12.5.14 Luminaire dirt depreciation (LLD) factors for six luminaire categories (I to VI) and for five degrees of dirtiness. *(IES.)*

GENERAL INFORMATION

Project identification: _____

(Give name of area and/or building and room number)

Average maintained illumination for design: ___ footcandles or ___ lux

Luminaire data:

Manufacturer: _____

Catalog number: _____

Lamp data:

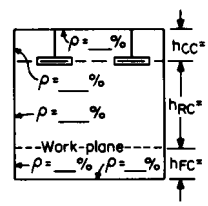
Type and color: _____

Number per luminaire: _____

Total lumens per luminaire: _____

SELECTION OF COEFFICIENT OF UTILIZATION

Step 1: Fill in sketch at right.



Step 2: Determine cavity ratios [Eq. (12.5.6)]

Room cavity ratio RCR = _____

Ceiling cavity ratio CCR: = _____

Floor cavity ratio FCR = _____

room length: _____

room width: _____

Step 3: Obtain effective ceiling cavity reflectance ρ_{CC} from Table 12.5.7 ρ_{CC} = _____

Step 4: Obtain effective floor cavity reflectance ρ_{FC} from Table 12.5.7 ρ_{FC} = _____

Step 5: Obtain coefficient of utilization CU from manufacturer's data (or Fig. 12.5.12 and Table 12.5.8) CU = _____

SELECTION OF LIGHT-LOSS FACTORS

Nonrecoverable		Recoverable	
Luminaire ambient temperature	_____	Room surface dirt depreciation RSDD	_____
Voltage to luminaire	_____	Lamp lumen depreciation LLD	_____
Ballast factor	_____	Lamp burnouts factor LBO	_____
Luminaire surface depreciation	_____	Luminaire dirt depreciation LDD	_____

Total light loss factor, LLF (product of individual factors above) = _____

CALCULATIONS
(Average maintained illumination level)

$$\text{Number of luminaires} = \frac{(\text{footcandles}) \times (\text{area*, ft}^2)}{(\text{lumens per luminaire}) \times (\text{CU}) \times (\text{LLF})}$$

$$= \frac{\qquad\qquad}{\qquad\qquad} =$$

$$\text{Footcandles} = \frac{(\text{number of luminaires}) \times (\text{lumens per luminaire}) \times (\text{CU}) \times (\text{LLF})}{(\text{area*, ft}^2)}$$

$$= \frac{\qquad\qquad}{\qquad\qquad} =$$

Calculated by: _____ Date: _____

*If lux is used, area is in m^2.

Fig. 12.5.15 Average illumination calculation sheet. *[Abstracted from ''IESNA Handbook,'' 8th ed., 1993; reprinted with permission. (This material has been modified from its original version and is not reflective of its original form as recognized by the IESNA.)]*

at 30 percent rated life is used, the LLD for incandescent lamps varies from 78 to 90 percent, with an average about 87 percent. For fluorescent lamps the LLD varies from 67 to 91, with an average about 82 percent. For better values consult the IES Handbook or manufacturers' data.

A design summary sheet is given in Fig. 12.5.15.

While the lumen method is the accepted procedure for calculating interior lighting levels, it is often necessary to have a quick approximation of the quantity of lighting equipment needed to satisfy an illumination-level specification. There are several rules of thumb which serve that purpose.

Spacing Method Table 12.5.9 indicates approximate *base*-maintained footcandle levels according to fixture spacing. For levels other than the base quantity, the level is changed inversely and proportionally to a change in spacing—i.e., doubling the spacing (in one direction) halves the level; halving the spacing doubles the level. Doubling the spacing in both directions reduces the level to one-fourth.

Lumens-per-Square-Foot Method (see footnote for Table 12.5.10);

Less accurate than the previous method, this one permits a fairly reasonable calculation by using the lamp lumens as given in the GE Lamp Catalog and substituting in the formula:

$$\text{Footcandle} = \frac{\text{total lamp lumens per fixture}}{2 \times \text{area per fixture}}$$

Or, for a given footcandle level, transposing the formula to determine area per fixture:

$$\text{Area per fixture} = \frac{\text{total lamp lumens per fixture}}{2 \times \text{footcandles}}$$

The ''2'' in the denominator assumes loss of half the lumens in fixture utilization, lamp depreciation and dirt accumulation. *(Source: General Electric.)*

Watts-per-Square-Foot Method Table 12.5.10 shows another way to arrive at a quick approximation.

Point Method of Design

If uniformity of lighting is to be investigated, or if outdoor lighting is to be designed, the point method is used. Manufacturers furnish candlepower distribution curves for their fixtures. An average curve is given for symmetrical fixtures while curves in various planes are given for asymmetrical ones. The basic equation for calculating the illumination from such curves is

$$E_h = (I_\theta \cos \theta)/D^2 = I_\theta H/D^3 \qquad (12.5.7)$$

Table 12.5.10 Watts-per-Square-Foot Method*

A convenient, popular, quick approximation

Lamp	Per 100 fc
Lucalox	1.6 W/ft²
Multi-vapor	2.5 W/ft²
Fluorescent	3.0 W/ft²
Mercury-vapor	3.5 W/ft²
Incandescent (reflector lamp)	8.5 W/ft²

SOURCE: General Electric Co.

* Both Table 12.5.9 and lumen-per-square-foot method are based upon *large* industrial areas, where room width (W) is six times the fixture mounting height (MH). For medium-sized areas (where W = 3MH), reduce footcandles and increase wattage 15 percent. For small areas (where W = MH), reduce footcandles and increase wattage 50 percent. Lucalox fixtures and incandescent reflector lamps tend to be more efficient and have higher lumen-maintenance characteristics; thus, for these sources increase the footcandle value about 25 percent.

where E_h is the illumination on the horizontal plane, I_θ the candlepower in the given direction, and D the distance of the luminaire to the point P. See Fig. 12.5.16.

$$E_v = (I_\theta \sin \theta)D^2 = I_\theta R/D^3$$

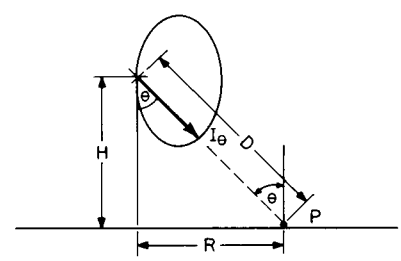


Fig. 12.5.16 Footcandle calculation diagram.

Table 12.5.9 Approximate Footcandle Levels According to Fixture Spacing*

Lighting system		Spacings†				
Lamp	Watts	10 × 10 ft	15 × 15 ft	20 × 20 ft	25 × 25 ft	30 × 30 ft
Lucalox	70	35	15	10	—	—
	100	55	25	15	10	—
	150	95	45	25	15	10
	250	180	80	45	30	20
	400	300	135	75	50	35
	1,000	—	—	210	135	95
Multivapor	175	85	35	20	15	10
	400	200	90	50	35	25
	1,000	—	300	165	105	75
		Continuous rows of 2-lamp fixtures on spacing† of:				
Fluorescent (cool white)		6 ft	8 ft	10 ft	12 ft	15 ft
40W Rapid Start		120	90	70	60	50
75W Slimline		120	90	70	60	50
110W High Output		185	140	110	90	75
215W Power Groove		300	225	180	150	120

SOURCE: General Electric Co.

* See footnote for Table 12.5.10.

† Spacings assumed within maximums established by fixture manufacturer.

For vertical surfaces

$$E_v = (I_\theta \sin \theta)D^2 = I_\theta R/D^3 \quad (12.5.8)$$

Nomograms and graphical solutions are available for Eqs. (12.5.7) and (12.5.8).

THE ECONOMICS OF LIGHTING INSTALLATIONS

The cost of lighting is computed by summing the annual cost of energy; relamping; cost of labor for cleaning, relamping, and servicing; interest; and depreciation.

Different lighting systems can be evaluated by comparing the costs per million lumen hours per luminaire. This is given by Eq. (12.5.9),

$$U = \frac{10}{Q \times D}\left[\frac{(P+h)}{L} + W \times R + \frac{(F+M)}{H}\right] \quad (12.5.9)$$

where U = unit cost of light in dollars per million lumen-hours; Q = mean lamp lumens; D = luminaire dirt depreciation (average between cleanings); P = price of lamp in cents; h = cost (in cents) to replace one lamp; L = average rated lamp life in thousands of hours; W = mean luminaire input watts (lamps plus ballast); R = energy cost in cents per kilowatthour; F = fixed or owning costs in cents per luminaire-year; M = cleaning costs in cents per luminaire-year; and H = annual hours of operation in thousands of hours.

In the above equations the area can be expressed in square metres. These equations are general and basic. With the advent of energy conservation, it has become important to use electric power effectively and efficiently. The IES, in its 1981 Handbook, established a watts per square foot (square metre) for various occupancies, called a *base unit power density* (UPD). These values are used to establish a power limit for a facility. For this purpose approximate values are used for CU, lamp plus ballast, lumens per watt, and LLF. For LLF, 0.7 is used.

Another way to compare installations is to compute the watts per square foot for each proposed installation. This is computed by either method:

$$\text{watts/ft}^2 = \frac{\text{total lamp lumens}}{\text{area, ft}^2} \times \frac{1}{\text{lumens/watt of lamp and ballast}} \quad (12.5.10)$$

$$\text{watts/ft}^2 = \frac{\text{designed illuminance}}{\text{CU} \times \text{LLF}} \times \frac{1}{\text{lumens/watt of lamp and ballast}} \quad (12.5.11)$$

Typical values are shown in Table 12.5.10.

DIMMING SYSTEMS

Dimming systems are used in theaters, auditoriums, ballrooms, etc. Originally, power-consuming rheostats were used. These have been replaced by continuously variable autotransformers, variable reactors, silicon controlled rectifiers (SCR) and triacs. The development of controlled solid-state devices has resulted in small, reliable dimmers which can be readily programmed. Only incandescent and cold-cathode lamps can be dimmed easily. Fluorescent lamps require special ballasts which keep the electrodes hot at all times. Dimmers have been developed for high-intensity discharge (H.I.D.) lamps.

HEAT FROM LIGHTING

Lighting installations are a substantial source of heat, have long been a factor in the design of air-conditioning (cooling) systems, and are increasingly significant in the design of heating systems. The heating effect for 1 W is 3.413 Btu/h. Approximate wattage data for lighting systems at various lighting levels can be calculated by using watts per-square foot calculated from Eq. (12.5.10) or (12.5.11). Heat generated is delivered to surrounding areas in several ways, with energy distribution for fluorescent and incandescent lamps as illustrated in Fig. 12.5.17. With the prevalent high lighting intensities of modern buildings, it is essential to control the heat generated by a lighting system. Substantial portions of the energy which is not radiated into the room may be conducted away from the luminaire by an air stream or by water flowing through a coil attached to the luminaire. In the heating season, this heat

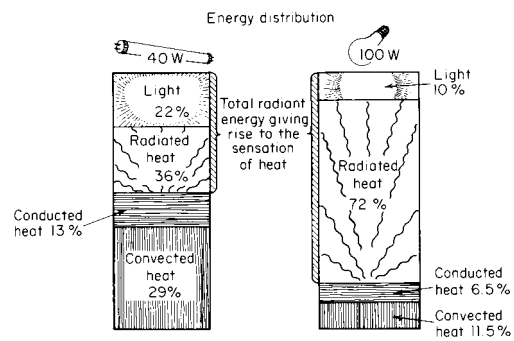


Fig. 12.5.17 Energy distribution of lamps.

energy is delivered to the perimeter of the building for effective space warming. In the cooling season, the heat is rejected to the exterior, thus reducing the load on the cooling system. Air-handling luminaires (Fig. 12.5.18) are receiving wide acceptance.

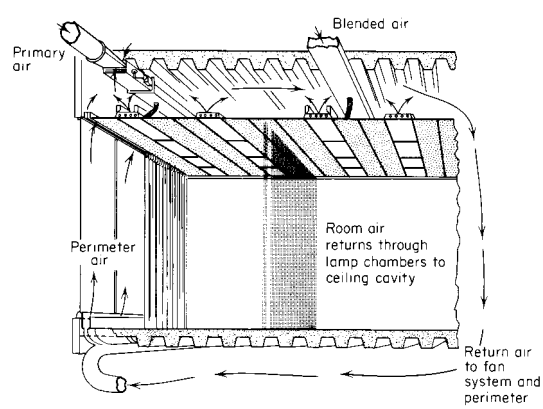


Fig. 12.5.18 Typical air handling system. *(Barber-Colman.)*

12.6 SOUND, NOISE, AND ULTRASONICS

by Benson Carlin and expanded by staff

REFERENCES: ANSI 51.1 Acoustical Terminology. *J Acoust. Soc. Am.* 1929 et seq. Beranek, ''Acoustics,'' McGraw-Hill. Carlin, ''Ultrasonics,'' McGraw-Hill. Morse and Ingard, ''Theoretical Acoustics.'' Harris, ''Handbook of Noise Control,'' McGraw-Hill. Mason, ''Physical Acoustics,'' Academic Press.

DEFINITIONS

Sound is an alteration in pressure, stress, particle displacement, and particle velocity, which is propagated in an elastic material. It is longitudinal in gases but may also be transverse (shear), surface, or other types in elastic media which can support such energy. It may be reflected, diffracted, or refracted at boundaries and under suitable conditions may be changed from one form to another. In longitudinal waves, the molecules move in the direction of wave motion, in the others at right angles to it. Waves may also be plane or circular depending on the source.

In general, the speed of sound V in a medium with mass density ρ is a function of its elastic properties. In solids, $V = \sqrt{E/\rho}$, where E is Young's modulus. In liquids, $V = \sqrt{E'/\rho}$, where E' is the bulk modulus of the liquid (see Table 3.3.2). In gases, the velocity is independent of the pressure, because the elasticity changes to compensate for the density changes; the general equation is $V = \sqrt{kp/\rho}$, where k is the ratio of specific heats and p is the pressure. The velocity in air at 68°F is 1,126 ft/s (33,160 cm/s) and increases by 0.1 percent per °F. In liquids, empirical formulas are easier to use than theoretical ones to predict actual velocities, since velocity varies in a complex way with temperature, pressure, and other factors. With sea water, a standard velocity of 5,100 ft/s (150,000 cm/s) may be used. The velocity of sound in liquids and solids is usually much higher than in gases (see Table 12.6.1 and Kinsler, ''Fundamentals of Acoustics,'' Wiley).

The **frequency** of a sound is the number of periods (cycles) occurring in unit time, customarily expressed as cycles per sec (cps) or ''hertz'' (Hz); kilocycles per sec, kc = 10^3 cps (kHz); megacycles per sec, Mc = 10^6 cps (MHz). Sound frequencies are usually defined as 20 to 20,000 cps **(audible),** higher **(ultrasonic),** and lower **(infrasonic).** Frequencies as high as the thousand-megacycle range (GHz) are now generated (see Table 12.6.2).

The relation between frequency f and wavelength λ is $V = \lambda f$. In air, at 1,126 cps, the wavelength is 1 ft. In nature, the waves may be simple sinusoidal, complex, or explosive (shock) depending on the source. The first is, of course, rare.

Attenuation of sound depends on the media of propagation and the frequency and is caused by absorption, spreading, and scattering. At audible frequencies in air attenuation is small except for the spreading of the energy over wide areas as the sound waves are propagated. By this means the intensity drops according to the inverse square law. However, in other media, the absorption, scattering, or other characteristic may be predominant.

The sound **intensity** is the average rate of sound energy transmitted through a unit area normal to the wave direction at the point considered. This is a definition of power and may be expressed in watts per sq metre. It is usual, however, to express power in **decibels,** dB, which is a term used to give the relative magnitude of two powers by comparing the one under consideration to a standard. The sound-pressure level in decibels, dB, is defined as twenty times the logarithm to the base 10 of the ratio of sound pressure to the reference sound pressure. All values are for air at 20°C and atmospheric pressure. Pressure measurements in air use a pressure reference (rms) of 0.0002 dyne/cm^2; 1 dyne/cm^2 is used underwater.

Intensity references for air are 10^{-16} w/cm^2 [equivalent to a pressure (rms) of 0.0002 dyne/cm^2, and 0.02 erg/cm^{2}s, equivalent to a pressure of 1 dyne/cm^2]. Since the references are equivalent (i.e., the reference pressure corresponds to the reference intensity in this particular case), numerical results are identical for plane waves using either expression $IL = 10 \log (I/I_0)$ or $PL = 20 \log (P_e/P_0)$, where IL and PL are the intensity and pressure levels, I_0 and P_0 are the reference intensity and pressure, P_e is the effective pressure, and I is the intensity in question.

Table 12.6.2 Sound Spectrum

Frequency	Action
20–40 cps	Thunder
128 cps	Average speech (male)
250–2,740 cps	Telephone bandwidth
90–5,000 cps	Radio broadcast
15 cps–15 kc	Limits of average human hearing
10–90 kc	Ultrasonic cleaning
15–50 kc	Ultrasonic depth sounding, sonar
20 kc	Ultrasonic bulgar alarm, control apparatus, door opening
30 kc	Highest frequency obtained by friction
40 kc	Highest frequency of Hartmann generator
48 kc	Bat cries
90 kc	Top limit of tuning fork
100 kc	Highest frequency of Galton whistle
500–15,000 kc	Ultrasonic pulse-echo testing
1,000 kc	Medical therapy
1,500–30,000 kc	Ultrasonic delay lines
15,000 kc	Radar trainer

Table 12.6.1 Velocity of Sound

Material	Sound velocity, ft/s	Density, lb/ft^3	Density × velocity, lb/(ft^2·s)
Aluminum	16,740	168	2.82×10^6
Brass	11,480	530	6.08×10^6
Copper	11,670	555	6.47×10^6
Iron and soft steel	16,410	486	7.98×10^6
Lead	4,026	1125	4.54×10^6
Brick	11,980	125	1.5×10^6
Cork	1,640	15	0.025×10^6
Wood	10,000–15,000	30–50	0.3×10^6–0.75×10^6
Water	4,794	62.4	0.299×10^6
Air, dry, CO_2 free, 32°F	1,088.5	0.0808	88.0
Hydrogen	4,165	0.00560	23.3
Water vapor, 212°F	1,564	0.0372	58.2

NOTE: Approximate values from Smithsonian Tables.

When making measurements with pressure or velocity microphones, it is the pressure level or velocity level which is measured and the relationship between the measurement and the intensity is unknown except in the special cases indicated.

Decibels do not add numerically as linear figures do; i.e., 70 dB + 70 dB = 73 dB since doubling power results in a 3-dB increase in sound pressure. Figure 12.6.1 shows how to add decibels within 14 dB of each other. If the difference is greater between two readings, ignore the weaker one.

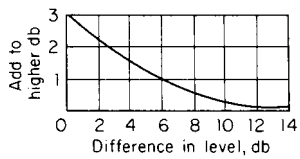


Fig. 12.6.1 Chart for addition of decibels.

Specific Acoustic Impedance The relationship between the pressure and the associated particle velocity at a point in a medium is called the specific acoustic impedance; its unit is the kilogram per meter second or mks rayl. The magnitude ρ_c is called the characteristic impedance of the medium, or the radiation resistance. This applies in the case of plane waves. The ρ_c of material is one of its most useful acoustic characteristics, since by means of it the amount of energy reflected at boundaries may be computed, horns may be analyzed according to the acoustic resistance at throat, and other calculations analogous to those made in electrical design may be carried out.

THE PRODUCTION AND RECEPTION OF SOUNDS

REFERENCES: Rinsler and Frey, ''Fundamentals of Acoustics,'' Wiley. Mason, ''Electromechanical Transducers and Wave Filters,'' Van Nostrand. Olsen, ''Elements of Acoustical Engineering,'' Van Nostrand.

Transducer A device for converting energy from one form to another, e.g., from electrical to acoustic or vice versa, is called a transducer. Among these are loudspeakers, microphones, hydrophones, and piezoelectric and magnetostrictive transducers.

Loudspeakers are usually classified as direct-radiator or horn type. The direct-radiator type consists of a cone, a magnet, a voice coil moving in the magnetic field, a vibrating diaphragm coupled to the cone, and suitable supports. The attachment of a horn improves the impedance match between the speaker and the air since it is essentially an acoustic transformer. The dimensions and flare of the horn contribute to its matching ability.

One of the more common types of horn is exponential, although straight and other types are also possible. In a similar manner, mechanical transformers may be used to concentrate the energy of ultrasonic transducers. In such forms they operate to concentrate rather than to spread the energy. The operation of the speaker may be variously influenced by its enclosure, by the baffle which separates the front from the back radiation, or by its resonances.

Microphones (for gases) and **hydrophones** (for liquids) are transducers for converting mechanical to electrical energy. They may be piezoelectric, electromagnetic, magnetostrictive, or capacitive. The variation in electrical output is proportional to the effect of the acoustic field on the characteristics. Ultrasonic transducers may be any of the above types but are usually crystal (piezoelectric) or magnetostrictive. Among the common piezoelectric materials are quartz, barium titanate, lithium sulfate, ADP (ammonium dihydrogen phosphate), and rochelle salt. In sonar and high-power industrial systems, mosaics of crystals are used; in low-power, high-frequency systems, a single crystal is usual.

Whistles and **sirens** may also be used to produce intense sound fields in gases and liquids. These are devices which produce sound by passing a fluid over an obstacle, thereby creating turbulence in the fluid. When the obstacle is an edge, these are referred to as edge or E tones; when an orifice, as jet tones. Organ pipes, whistles, and nozzles for spraying are devices of this class. Frequencies up to 100,000 cps are possible, although 30,000 cps is the approximate limit at which appreciable power can be generated. Resonators may be placed in the sound field to reinforce it and to stabilize the frequency. These take the form of small pipes tuned to the approximate frequency. Common types of whistles are the Hartmann and Galton (for gases) and the jet edge (for liquids).

Sirens are devices in which a revolving disk with holes in it interrupts a jet from a nearby tube. Compressed air, steam, and water have been used. Frequencies up to 30 kc may be produced at efficiencies of 50 percent approximately; a 1-hp motor produces between 300 and 1,000 W (see also Jones, *J Acoust. Soc. Am.,* 1946).

Transducers are generally driven by electronic generators, motor generators, or air compressors. As receivers, they activate amplifiers or indicating devices.

The Perception of Sound The average young observer perceives sound between 20 and 20,000 cps. High-frequency response deteriorates with advancing age. The ear responds to a wide range of intensities; e.g., between 500 and 5,000 cps, the ratio of tolerated intensities is about 10^{12}. The minimum intensity perceived varies with frequency. Figure 12.6.2 shows the audible frequency and intensity range for a standard listener, where the lowest curve represents the threshold of hearing and the top one the beginning of sensation in the ear. These curves show the pressure levels required for a given tone to sound as loud as the corresponding reference tone of 1,000 cps (see also Fletcher and Munson, *J. Acoust. Soc. Am.,* 1933).

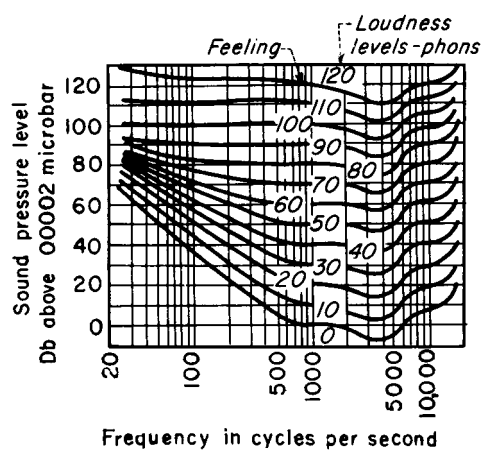


Fig. 12.6.2 Loudness contours.

Loudness is a subjective rather than a purely physical attribute. To provide a qualitative basis, the loudness level in **phons** is defined as the pressure level in decibels of a pure 1,000 cps tone which a typical observer judges to sound as loud as the sound in question. Observers can experimentally judge the loudness of pure or complex tones. However, this does not means that the apparent level is proportional to its level in phons; i.e., a level of 10 phons is not twice as loud as one of 5 phons. An additional expression, **sones,** defined as the loudness of a 1,000 cps tone at 40 dB intensity, is necessary to compare various loudness. The relationship between sones and phons is shown in Fig. 12.6.3.

Quality is a subjective attribute of sound in which equally loud sounds may be distinguished as to kind. Basically, differences in quality arise from differences in the distribution of energy in different parts of the frequency spectrum. In *music* this takes the form of the energy relationship of fundamental and harmonics; in *noise* it is random. These differences affect the sensation of loudness of noise and the psychological annoyance it produces. Shrill, high-pitched, and irregular sounds are usually judged less pleasant than low-pitched and regular sounds.

Among terms used to define quality are **pitch,** determined by fre-

quency (mostly), together with intensity and waveshape (unit is the mel); **timbre,** determined by wave shape (mostly), together with intensity and frequency (see Seashore, ''Psychology of Music,'' Dover).

Masking describes the ability of one sound to make the ear incapable of perceiving a second one. See ANSI 53.20, Psychoacoustical Terminology. It is measured by the shift in the threshold of audibility of the masked sound in decibels. The partial deafening of the ear by the masking effect of noise affords a direct quantitative measure of the interfering effect of the noise. If the masking is measured at several frequencies throughout the audible ranges, the overall pressure level of the sound can be computed. For any given frequency, the masking is expressed in decibels relative to the unmasked threshold for a **critical bandwidth** centered on the frequency of the masked tone. This critical bandwidth is that beyond which an increase in the passband has little effect on the masking of a pure tone at its center frequency. Critical bandwidths vary from 40 cycles at 100 cps to 200 cycles at 6,000 cps.

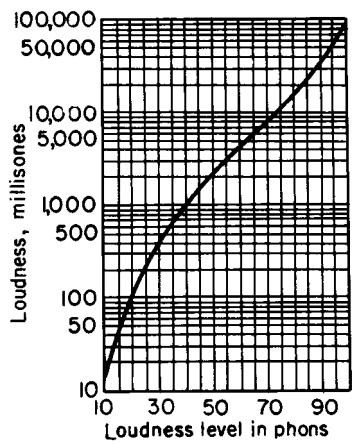


Fig. 12.6.3 Relation between loudness and loudness level.

Noise is an undesired sound. It implies an unwanted disturbance in a useful band which interferes with the useful information. Any elastic structure may produce noise when set into vibration. Generally the motion is an unwanted concomitant of some desired function, e.g., the vibration of a machine tool.

The term has a connotation of unpleasantness in quality or loudness. Typical examples are gear noises, 60-cycle hum, motor traffic, hammer blows, pneumatic-tool operations, and hissing of gases in an orifice. Noise measurements are usually made with a **sound-level meter,** comprising a microphone, attenuator, amplifier, frequency-weighing networks, and an indicator or recorder. See ANSI 51.4, Sound Level Meters. A **sound analyzer** indicates sound pressure as a function of frequency. In some cases electrical filters may be included which permit measurement in certain restricted frequency ranges. By using these instruments the components of the heterogeneous noise may be identified, and this helps to correlate it with its production. Once located, techniques for elimination flow and the results may be used to compute masking. Individual frequencies may be identified by beating against a known source **(beat-frequency oscillator)** until a null is observed.

Contact transducers (vibration pickups) may be used to locate sources of noise, such as in partitions and machine parts. They may be piezoelectric or magnetic and may be used in place of the microphone or the sound-level meter; hydrophones may also be used.

Where the frequency distribution of the noise is significant, an analyzer may be used since the level meter tells nothing about frequency distribution. Analyzers are manufactured in various forms, e.g., the octave-band analyzer, the impact-noise analyzer, and the wave analyzer, each of which uses a different method of finding out which frequency components are present.

Typical sound levels are shown in Table 12.6.3.

Table 12.6.3 Typical Sound Levels

Decibels		
	120	Threshold of feeling
		Thunder, artillery
Deafening	100	Nearby riveter
		Elevated train
	100	Boiler factory
		Loud street noise
Very loud	90	Noisy factory
		Truck unmuffled
	80	Police whistle
		Noisy office
Loud	70	Average street noise
		Average radio
	60	Average factory
		Noisy home
Moderate	50	Average office
		Average conversation
	40	Quiet radio
		Quiet home or private office
Faint	30	Average auditorium
		Quiet conversation
	20	Rustle of leaves
		Whisper
Very faint	10	Soundproof room
		Threshold of audibility
	0	

NOISE CONTROL

REFERENCES: General Radio Co., ''Handbook of Noise Measurement.'' Harris, ''Handbook of Noise Control,'' McGraw-Hill. Bolt, ''Handbook of Acoustic Noise Control.'' Faulkner, ''Handbook of Industrial Noise Control,'' Industrial Press.

Noise control may be carried out at several stages: (1) at the source, by design changes or by quieting procedures, (2) during transmission, by attention to the path by which it is propagated to the listener, and (3) by quieting at the listening position. It may also be controlled architecturally as by the careful placement of necessarily noisy rooms in a building.

The Source By inspection and test procedures, a noise is tracked to its source. In some cases, design procedures which attempt to reduce the vibration or to prevent its radiation may be used. This may require the redesign of elements, such as cams, gears, housings, or provisions for cushioning. Viscous damping materials, e.g., putty or tar, may be applied to the vibratory surfaces in the form of nonhardening plastic mixtures. A machine may be isolated by sections or shock mounts to prevent transmission of vibration from one section to another. Absorbing materials may be placed on walls to absorb sound after it has been radiated.

Transmission Isolation If the vibrations of noisy machinery cannot be suppressed at the source, their transmission to the listener should be impeded. For the higher frequencies constituting noise, the most effective isolation method is the introduction of elastic discontinuities in the structure transmitting the noise (measured by the difference between density-velocity products as given in Table 12.6.1). The discontinuities may be obtained by the use of felt, cork, rubber, or springs in machinery mountings, or by the introduction of alternate lead and cork sheeting at masonry junctions. The isolation treatment should be applied as close to the source as possible in order to eliminate sound radiation from the structures transmitting the vibrations. Where this is not possible, the listening space itself may be isolated. Thus **quiet rooms,** constructed especially for noise measurements, are usually built as separate structures isolated from the main building.

Filtration Some problems of noise transmission through air lend themselves to solution by methods of filtration. Typical examples are the transmission of sound in ventilating ducts and the noise production

at engine exhaust pipes. In each of these cases, the steady flow of gas must not be impeded, but the alternating flow, representing sound transmission, must be effectively suppressed.

For ventilating ducts, an acceptable degree of noise suppression may be obtained by lining the ducts (on at least two nonopposite walls) with an efficient sound absorbent for a distance of 10 to 15 ft from both the inlet and the outlet. Where the length of duct available is insufficient, or where additional noise suppression is required, baffles, covered with absorbing material, may be introduced in the duct. A plenum chamber, used to serve several ducts, should be lined with sound absorbents. If the air velocities are high, it may be necessary to introduce additional baffles at bends in the ducts to avoid noise production through turbulence.

Exhaust mufflers are usually modifications of the elementary low-pass acoustical filter, comprising a through tube to which closed cavities are coupled through small holes at intervals along the tube. Mufflers of this type find application other than in internal combustion engine exhaust systems, and often are termed *passive mufflers.* Typical structures of this type (Fig. 12.6.4) produce little increase in back pressure and considerable attenuation of sound waves having frequencies above a cutoff frequency determined by the size of the holes and cavities. Porous packing, such as steel wool, in the side cavities or studied irregularity in the size and spacing of the cavities will increase the uniformity of noise suppression, whereas increasing the number of side cavities

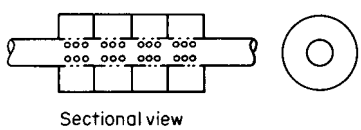


Fig. 12.6.4 Section through a passive exhaust muffler.

and the length of the muffler will increase the amount of suppression. Baffles in the tailpipe or irregular obstructions producing devious flow paths, e.g., the stone-filled pit for stationary engine exhausts, produce muffling action at the expense of appreciable increase in exhaust back pressure.

Shielding of airborne noise must be done by sound-opaque screens large in comparison with the wavelength of the sounds whose transmission they are to impede. This is seldom possible in building interiors except by utilization of building partitions as screens. Sound is transmitted through such partitions principally by minute flexure of the wall as a whole in response to the incident sound pressure on the noisy side, with consequent reradiation on the quiet side. Reduction of sound transmission is obtained by increasing the mass per unit area of the partition, by constructing the partition of material comprising a hollow air-filled or fibrous-filled sandwich with a pattern of holes in one of the surfaces, or by the use of double partitions, vibrationally isolated.

Sound-transmission loss is usually greater for high frequencies than for low and is measured by comparing the average sound level on each side of the partition under standardized conditions. Average values of transmission loss, for frequencies from 125 to 4,000 cps, for typical partitions, are shown in Table 12.6.4. In any specific case, a more exact measure of the effectiveness of an insulating partition can be obtained by direct comparison of the transmission-loss vs. frequency curve for the partition and the intensity vs. frequency curve for the noise. Additional data on transmission loss for a wide variety of building materials and structures are available in the NIST publications TRBM-44; BMS 17 and Supplements 1 and 2.

In general, double partitions (including floated floor constructions) provide greater transmission loss than equally heavy concrete, masonry, or brick walls but, except for special designs, less transmission loss than equally thick masonry walls. Double walls must be constructed carefully to avoid loss of vibration isolation through mechanical bridging between the opposite surfaces. Sound-absorbing fillers (e.g., mineral wool) are usually detrimental to sound insulation if in contact with both interior surfaces, and a single bridging nail may alter significantly the insulating efficiency. For maximum effectiveness, one of the wall surfaces should be hung structurally free at all four edges with the boundary cracks sealed, with felt or asphalt compounds, against sound leakage. Through piping should be made vibrationally discontinuous by introducing canvas or metallic sylphon sections, and clearance holes at the walls should be sealed. Sound leakage through small clearance cracks contributes to the low transmission loss of ordinary doors. Special self-sealing soundproof doors are required to maintain the effectiveness of an efficient sound-insulating partition.

Quieting The sound level established in a room by a noise source is higher than that which the same source would produce in free space on account of successive reflections of sound at the walls. It is the function of **quieting** to avoid such enhancement of noise by providing a high degree of sound absorption at all interior reflecting surfaces exposed to the noise. Commercially available sound-absorbing materials may be cemented to flat surfaces or secured to wood or metal furring strips. They derive their absorbing property either from capillary porosity of the surface or from the dissipative vibration of surface layers. Hanging ''functional absorber'' units comprising vibratile matte surfaces, enclosing a volume of about 1 ft^3, can be used where surface absorbents cannot be installed conveniently. The effectiveness of sound absorbents varies with frequency, usually being greater for high and intermediate than for low frequencies. It may be measured by determining the **absorption coefficient,** defined as the fraction of sound energy diffusely incident on the material that is not reflected, or by determining the **specific acoustic impedance** of the material. The measured absorption coefficient is not a property of the material alone, but depends partly on the size and mounting of the test sample and the size and shape of the test chamber; thus comparison of the coefficients for different materials should be based only on measurements made under identical conditions. Such measurements on a wide variety of materials have been made available by the Acoustical Materials Assoc. (Chicago, Ill.), although it is to be expected that the absorption coefficients effective in various practical applications may differ somewhat from the published values.

Table 12.6.4 Sound-Transmission Loss in Building Partitions

Wall	Thickness, in	Weight lb/ft^2	Transmission loss, dB
Wood	0.2	0.45	18.5
Plate glass	0.25	3.2	27.0
Hollow gypsum tile, unplastered	3	11.1	27.2
Brick wall, unplastered	. . . .	22.0	33
Brick wall, plastered	6	46	43
Brick wall, plastered	10.5	93	49
Double wall; metal lath, ½-in gypsum plaster, on staggered 2 × 4 in wood studs	7.5	19.8	44
Double 3-in hollow gypsum tile, unplastered, 3-in airspace	9	22.0	42.6
1-in Thermax nailed over building paper to 3-in Thermax laid up in mortar, ½-in plaster on both sides	5	15	47
Double 2-in solid-gypsum tile, unplastered, completely isolated structurally by separate foundations, 4-in airspace	8	20.4	59

SOURCE: Based on Sabine, ''Acoustics and Architecture,'' McGraw-Hill.

Table 12.6.5 Sound-Absorption Coefficients

Maker, material thickness	Absorption coef at indicated frequencies						Noise-reduction coef	Weight, lb/ft²	Surface	AMA test No.
	128	256	512	1,024	2,048	4,096				
Armstrong Cork Co.										
Cushiontone A, ¾ in.	0.10	0.28	0.66	0.91	0.82	0.69	0.65	1.05	484 holes per sq ft, 3/16 in diam, 5/8 in deep. Painted by mfr	47–28
Travertone, ¾ in.	0.06	0.23	0.78	0.97	0.84	0.80	0.70	1.20	Fissured, painted	48–58
The Celotex Corp.										
Acousti-Celotex										
Type C-9, ¾ in	0.11	0.23	0.80	0.93	0.58	0.50	0.65	0.96	441 holes per sq ft, 3/16 in diam. Any paint	46–132
Type M-1, 5/8 in	0.07	0.21	0.64	0.86	0.93	0.83	0.65	1.31	676 holes per sq ft, 5/32 in diam. Any paint	46–12
Q-T Ductliner, in	0.21	0.42	0.71	0.86	0.79	0.75	0.70	1.3	Unpainted	A48–10
Johns-Manville Corp.										
Sanacoustic, KK, pad plus metal facing and pad supports 1 9/16 in	0.25	0.58	0.96	0.97	0.85	0.72	0.85	Pad 1.28	4,608 holes per sq ft, 0.068 in diam. Enameled metal pan backed with wool pad	46–88
Fibretone, 13/16 in	0.14	0.37	0.69	0.80	0.76	0.73	0.65	1.17	484 holes per sq ft, 3/16 in diam. Any paint	46–124
Airacoustic, 1 in	0.29	0.31	0.70	0.82	0.79	0.80	0.70	1.50	Unpainted	46–71
National Gypsum Co.										
Acoustifibre, 5/8 in	0.10	0.16	0.62	0.97	0.81	0.73	0.65	0.56	441 holes per sq ft, 3/16 in diam, 5/16 in deep. Painted	46–137
Owens-Corning Fiberglas Corp.										
Fiberglas Acoustical Tile, plain type, ¾ in	0.04	0.20	0.63	0.91	0.82	0.82	0.65	0.69	Painted	A48–99
United States Gypsum Co.,										
Acoustone F, 11/16 in	0.08	0.25	0.76	0.84	0.78	0.73	0.65	1.35	Fissured, painted	46–50
Brick wall, painted	0.012		0.017		0.023		0.02			
Concrete wall or floor	0.01		0.015		0.02		0.02			
Wood floor	0.05		0.03		0.03		0.03			
Cork or rubber tile on concrete			0.03–0.08				0.05			
Glass	0.035		0.027		0.02		0.02			

NOTE: This tabulation is based on *Bull.* XI (1949), Acoustical Materials Assoc. All samples were cemented to plasterboard for test, except that the Sanacoustical unit is attached to wood furring with special clips, and the duct linings are laid on 24 gage sheet iron, nailed to 1 × 3-in wood furring, 24-in O.C.

For ordinary noise quieting, the average of absorption coefficients measured at frequencies of 250, 500, 1,000, and 2,000 cps, called the **noise-reduction coefficient,** may be used. Typical values of this coefficient for representative materials are given in Table 12.6.5. In making quantitative estimates of noise reduction, the **total sound absorption** of the room boundaries may be computed by multiplying the noise-reduction coefficient of each different material present by the total exposed area of that material and summing up the resulting products. The noise reduction is then given by

$$\text{Noise reduction in decibels} = 10 \log \frac{\text{total absorption after treatment}}{\text{total absorption before treatment}}$$

When the frequency spectrum of the offending noise is known, greater precision in calculation of total absorption is obtained by replacing the noise-reduction coefficient by the absorption coefficient measured at the frequency of maximum loudness level from the noise source. Subjective judgments of the loudness reduction obtained by quieting can be estimated by using the noise reduction in decibels in connection with the loudness chart of Fig. 1.2.6.3.

In general, the larger the area of absorbing material introduced and the higher its noise-reduction coefficient, the more effectively the noise is reduced. No amount of quieting treatment can reduce the level of the noise received directly from the source. If full coverage of walls and ceiling is not possible, distribution of the material in several small patches is more effective than the same total area of material concentrated in one location. Similarly, the same area of material is more effective when applied to nonopposite walls and ceiling than when concentrated on either of these areas, and more effective when located near the edges and corners of a given area than when located in the center. Recent advances in noise control have resulted from advances made in signal processing chips and in the significant reduction in the cost of those components. These electronic components are built into electronic equipment, the whole of which constitutes an active noise control system, and which is based on the principle that a sound signal can be canceled by an almost identical sound signal produced 180° out of phase with the first one. This concept of an electronic noise muffler has been successfully adapted to control offensive noise in a variety of industrial applications. Further developments are expected along these lines.

APPLICATIONS

REFERENCES: Carlin, ''Ultrasonics,'' McGraw-Hill. Bergmann, ''Ultrasonics,'' S. Hirzel Verlag. ANSI Z24.18, Ultrasonic Therapeutic Equipment.

Industrial Applications Acoustic waves of high powers used in industrial applications are generally called **sonic** (or **ultrasonic,** when greater than about 20,000 cps). High-intensity sonic waves, in the 10,000 cps to the megacycle range (10^6 cps), are applied to many industrial processes. The effects seem to be a function of cavitation, heating, particle acceleration, short wavelength, and other characteristics of the waves. Application categories are (1) high amplitude and (2) low amplitude.

High-amplitude waves are used in operations such as cleaning, welding, drilling, emulsification, soldering, atomization, chemical and biological applications, medical therapy, and sonar. The energy may be continuous, pulsed, or modulated, in various ways.

Low-amplitude waves are used in operations such as material testing, burglar alarms, delay lines, or medical diagnoses. Any waveshape may

be used; basically, a physical characteristic of the waves, such as velocity, is measured.

Cavitation may be defined as the formation and collapse of gas- or vapor-filled bubbles. Most significant industrial applications, e.g., cleaning, take place during the vaporous phase. The amount and force of cavitation are affected by the character of the liquid and the gas in it. The bubble collapse generates powerful local forces which cause the desired action. Levels of power in the liquid of approximately 3 W/cm^2 are required for intense cavitation and are dependent upon factors such as the liquid, temperature, and external pressure; powers as low as 0.3 W/cm^2 in water will produce a threshold cavitation (see also Briggs et al., *J. Acoust. Soc. Am.*, 1947).

High-Amplitude Applications

(See Fig. 12.6.5.)

Cleaning An ultrasonically agitated bath will erosively clean dirt from immersed articles. A high-power generator, usually electronic, produces sonic energy which is impressed on a transducer to drive the bath. Barium titanate transducers prevail, but magnetostrictive designs may be used. 10,000 to 90,000 cps are commonly used, with the lower frequencies generally more effective. Generator tuning may be manual or by feedback from the transducer controls. Power levels of 5 ± W/cm^2 are commonly used; size (depth) of the tank may affect the output; 50 ± W/gal is a rough empirical relationship; cavitation must exist for effective, speedy cleaning. A proper cleaning solution must be used, i.e., one which supports intense cavitation and also cleans (e.g., water solutions of alkalines or acids, or solvents like hydrofluoroether); temperatures between 120 and 160°F prevail. Among items commonly cleaned are jewelry (lost-wax castings), eyeglass frames, lenses, metal parts, and watches and clocks.

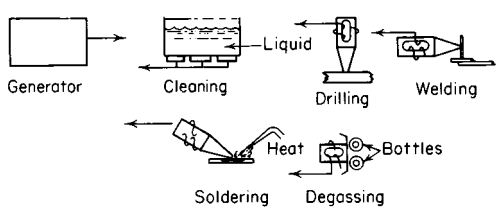


Fig. 12.6.5 High-amplitude systems.

An alternative form of ultrasonic cleaning can be performed by the introduction of an ultrasonic probe in a cleaning bath. The probe can be made to focus a large amount of energy over a small area or volume of the cleaning solution in which the workpiece is submerged. This method fosters removal of tenacious contaminents from normally inaccessible locations.

Foaming of Beverages Air content, which determines the life of a carbonated beverage, is reduced by foaming the bottles or cans before capping. Magnetostrictive transducers at 20,000 cps and 250 W, in contact with the containers, produce enough foam from the CO_2 to expel the air. The ultrasonic power requirement is basically small, but the losses in the coupling dictate the use of large generators. Similar techniques and apparatus may be used for the removal of gases from materials and for chemical effects such as the acceleration of iodine reactions and oxidation.

Soldering Some materials, such as aluminum, oxidize when exposed to air so that soldering is not possible. However, an ultrasonically driven solder bath will cause wetting of the material with solder and tinning of the surface. Magnetostrictive units are indicated because of the temperature requirements, with external heating and high tin-content solders; pots, as well as irons, may be constructed. Applications include aluminum wire, foil capacitors, and the filling of holes in castings (see also Sec. 13).

Welding Similar equipment (between 100 and 5,000 W output) may be used to weld thin metal or thin-to-thick sections. Such units apply the ultrasonics in a shear direction with respect to the parts to be welded. The process depends primarily on the sonic energy, the clamping force, and the amount of external heating. Either spot or lap welds are possible, but in all cases one of the sections must be thin (see also Sec. 13 and ''AWS Welding Handbook'').

Drilling is effected with the same sort of apparatus, but the force is longitudinal rather than shear. An abrasive is flowed over the tool head and is driven by the cavitation against the part to be drilled, causing the material to erode away. Any shape may be obtained in this manner. The head is usually mounted on an apparatus similar to a milling machine. Tolerances depend principally on the physical rigidity of the system and on grit size. Applications include hard materials such as ceramics, jewels, and glasses. The same apparatus has been applied to dental drilling, but the time required and the necessity for use of a slurry have kept it from greater acceptance. However, the method is widely applied to cleaning the surface of the teeth. The technique has been applied to forming lesions in the brain and spinal column of human beings, using a focused beam of sound at 3 ± Mc rather than a velocity transformer.

A **whistle** may be operated in a gas for agglomeration or foam settling and in a liquid for emulsification. Ultrasonic fields introduce additional forces on particles suspended in a gas, causing them to come together. Materials such as smoke, dust, and fog have been experimentally treated in this way. Generally, the whistle drives a resonant cavity, generating about 150 dB intensity (power output of 150 ± W) at 10 to 20 kc. Atomization of a liquid may be effected by introducing the liquid into a strong sonic field, either passing it directly over or through the whistle. The use of waves has been reported for drying solids, such as sugar, for atomically driven sound beacons under the sea, and for emulsification of liquid rubber and other materials.

Sonar Underwater signaling and detection are among the older applications of ultrasonics and comprise the active (pulse-echo, Doppler) and passive (listening) systems. The principles of operation are similar to those of pulse-echo testing in the active case (see Albers, ''Underwater Acoustics Handbook,'' Penn State Univ. Press).

Among the **miscellaneous applications** of high-power ultrasonics is metal treatment, atomization of oil for burners and of cleaning solutions used in the manufacture of delicate electronic devices such as circuit boards, ultrasonic diathermy for bursitis, humidification, and ultrasonic neurosonic surgery.

Testing Materials (See Sec. 5 and McMaster, ''Nondestructive Testing Handbook,'' Ronald.) The most common industrial use of low-power ultrasonic waves is for testing materials. The technique basically depends on the ability of a discontinuity in a material to reflect part of the energy hitting it. Various types of ultrasonic waves such as longitudinal, shear, or surface may be used. Transducers are usually crystal, such as barium titanate or quartz, and measure from ¼ to 1 in. diam (Fig. 12.6.6).

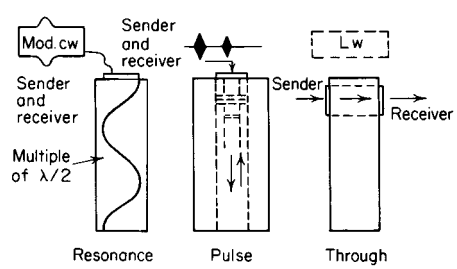


Fig. 12.6.6 Low-amplitude systems.

The basic types of ultrasonic systems are (1) pulse-echo, (2) through-transmission, and (3) resonance. The **pulse-echo system** uses a pulse ranging in length (time) from a fraction of a microsecond to several microseconds and an amplitude from 50 to 250 V radio frequency across the transducer. The pulse travels in the material and is reflected by an interface; the time of travel is measured. The pulse-echo method

has also been applied to medical mapping of the body interior and for tracing brain centerline displacement and heart valve action.

The **through-transmission system** places continuous pulsed or modulated waves on a transducer coupled to one side of a part with pickup on the other side. If a flaw interrupts, the waves do not penetrate the part.

The **resonance system** uses a single transducer and varies the frequency applied to it. Within the applied frequencies is one whose wavelength is related to the thickness of the part in such a way that less power is required of the driving system. This condition is indicated, and since the wavelength within the material is known, the thickness is determined. The most common applications of the resonance system are (1) for thickness measurement when one side only is available, and (2) for finding laminations in thin sections and lack of bond (see also Sec. 5).

The **sonic burglar alarm** depends on the Doppler shift caused in a sonic field by a moving object. The unit operates at 20 kc and will find minimum objects of 0.03 ft^2 in an enclosure of 100 ft^2. Magnetostrictive transducers are used coupled to diaphragms; they are not unlike radio speaker systems in appearance.

Liquid-Level Sensors Ultrasonic devices may be used to measure the level of liquid in a tank either by the pulse-echo technique or by indicating the transducer lead, i.e., a transducer driven by an oscillator where the reaction of the liquid load causes a change in the driving current.

Sonic microscopes are devices in which a beam of sound illuminates a part; the shadows of the field are scanned on a cathode-ray tube (usually constructed of barium titanate) by a flying spot.

SAFETY

Under the Occupational Safety and Health Act (OSHA), definitions have been made which legally define levels either as safe or as hazardous. Moreover, noise is now recognized as a pollutant, both as a nuisance and as the cause of hearing impairment.

General noise levels in the environment have already been defined (Fig. 12.6.2). There is some evidence that noise may cause ailments such as anxiety and heart disorders.

Protection from noise is required when sound levels exceed those in Table 12.6.6 (Table G-16 of the Act), when measured on the A scale at slow response on a standard sound-level meter (except for certain alarms, etc., as provided in the Act). When several successive exposures occur, they are combined (see paragraph 1910.95, reference above).

Conversion from octave-band analysis levels to A-weighted levels may be made from the figure in the Act. This figure has been also adopted by some local or state laws (Fig. 12.6.7).

When the environmental noise is greater than specified in the Act, protection must be provided.

However, when the noise is intermittent, if the peaks occur within 1 s or less, it is considered continuous. When protective equipment is required, it must be provided by a trained person and periodic checks made of the effectiveness.

Table 12.6.6 Permissible Noise Exposures

Duration per day, h	Sound level, dBA slow response
8	90
6	92
4	95
3	97
2	100
1½	102
1	105
½	110
¼ or less	115

In addition to the levels, time of exposure is also involved, as shown in Table 12.6.6.

OSHA also requires, when 90 dBA is exceeded, ''a continuing effective hearing conservation program shall be administered,'' i.e., consisting of periodic hearing checks and noise surveys.

The type of facility required for these tests is spelled out in the reference above.

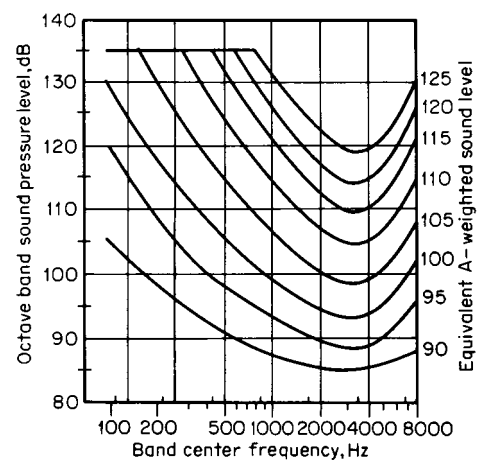


Fig. 12.6.7 Equivalent sound-level contours. Octave-band sound-pressure levels may be converted to the equivalent A-weighted sound level by plotting them on this graph and noting the A-weighted sound level corresponding to the point of highest penetration into the sound-level contours. This equivalent A-weighted sound level, which may differ from the actual A-weighted sound level of the noise, is used to determine exposure limits.