CHAPTER 13

REFRIGERATION LOAD

TOTAL refrigeration load includes (1) transmission load, which is heat transferred into the refrigerated space through its surface; (2) product load, which is heat removed from and produced by products brought into and kept in the refrigerated space; (3) internal load, which is heat produced by internal sources (e.g., lights, electric motors, and people working in the space); (4) infiltration air load, which is heat gain associated with air entering the refrigerated space; and (5) equipment-related load.

The first four segments of load constitute the net heat load for which a refrigeration system is to be provided; the fifth segment consists of all heat gains created by the refrigerating equipment. Thus, net heat load plus equipment heat load is the total refrigeration load for which a compressor must be selected.

This chapter contains load calculating procedures and data for the first four segments and load determination recommendations for the fifth segment. Information needed for refrigeration of specific foods can be found in Chapters 15 and 17 to 29.

TRANSMISSION LOAD

Sensible heat gain through walls, floor, and ceiling is calculated at steady state as

$$q = UA\,\Delta t \tag{1}$$

where

q = heat gain, Btu/h
A = outside area of section, ft^2
Δt = difference between outside air temperature and air temperature of the refrigerated space, °F

The overall coefficient of heat transfer U of the wall, floor, or ceiling can be calculated by the following equation:

$$U = \frac{1}{1/h_i + x/k + 1/h_o} \tag{2}$$

where

U = overall heat transfer coefficient, Btu/h·ft^2·°F
x = wall thickness, in.
k = thermal conductivity of wall material, Btu·in/h·ft^2·°F
h_i = inside surface conductance, Btu/h·ft^2·°F
h_o = outside surface conductance, Btu/h·ft^2·°F

A value of 1.6 Btu/h·ft^2·°F for h_i and h_o is frequently used for still air. If the outer surface is exposed to 15 mph wind, h_o is increased to 6 Btu/h·ft^2·°F.

With thick walls and low conductivity, the resistance x/k makes U so small that $1/h_i$ and $1/h_o$ have little effect and can be omitted from the calculation. Walls are usually made of more than one material; therefore, the value x/k represents the composite resistance of the materials. The U-factor for a wall with flat parallel surfaces of materials 1, 2, and 3 is given by the following equation:

$$U = \frac{1}{x_1/k_1 + x_2/k_2 + x_3/k_3} \tag{3}$$

The thermal conductivity of several cold storage insulations are listed in Table 1. These values increase with age due to factors discussed in Chapter 23 of the 2005 *ASHRAE Handbook—Fundamentals*. Chapter 25 of the 2005 *ASHRAE Handbook—Fundamentals* includes more complete tables listing the thermal properties of various building and insulation materials.

Table 2 lists minimum insulation thicknesses of expanded polyisocyanurate board recommended by the refrigeration industry. These thicknesses may need to be increased to offset heat gain caused by building components such as wood and metal studs, webs in concrete masonry, and metal ties that bridge across the insulation and reduce the thermal resistance of the wall or roof. Chapter 25 of the 2005 *ASHRAE Handbook—Fundamentals* describes how to calculate heat gain through walls and roofs with thermal bridges. Metal surfaces of prefabricated or insulated panels have a negligible effect on thermal performance and need not be considered in calculating the U-factor.

In most cases, the temperature difference Δt can be adjusted to compensate for solar effect on heat load. Values in Table 3 apply over a 24 h period and are added to the ambient temperature when calculating wall heat gain.

Table 1 Thermal Conductivity of Cold Storage Insulation

Insulation	Thermal Conductivity[a] k, Btu·in/h·ft^2·°F
Polyurethane board (R-11 expanded)	0.16 to 0.18
Polyisocyanurate, cellular (R-141b expanded)	0.19
Polystyrene, extruded (R-142b)	0.24
Polystyrene, expanded (R-142b)	0.26
Corkboard[b]	0.30
Foam glass[c]	0.31

[a]Values are for a mean temperature of 75°F, and insulation is aged 180 days.
[b]Seldom-used insulation. Data are only for reference.
[c]Virtually no effects from aging.

Table 2 Minimum Insulation Thickness

Storage Temperature, °F	Expanded Polyisocyanurate Thickness	
	Northern U.S., in.	Southern U.S., in.
50 to 60	2	2
40 to 50	2	2
25 to 40	2	3
15 to 25	3	3
0 to 15	3	4
–15 to 0	4	4
–40 to –15	5	5

The preparation of this chapter is assigned to TC 10.8, Refrigeration Load Calculations.

Table 3 Allowance for Sun Effect

Typical Surface Types	East Wall, °F	South Wall, °F	West Wall, °F	Flat Roof, °F
Dark-colored surfaces				
Slate roofing	8	5	8	20
Tar roofing				
Black paint				
Medium-colored surfaces				
Unpainted wood	6	4	6	15
Brick				
Red tile				
Dark cement				
Red, gray, or green paint				
Light-colored surfaces				
White stone	4	2	4	9
Light colored cement				
White paint				

Note: Add to the normal temperature difference for heat leakage calculations to compensate for sun effect. Do not use for air-conditioning design.

In most cases, the temperature difference Δt can be adjusted to compensate for solar effect on heat load. Values given in Table 3 apply over a 24 h period and are added to the ambient temperature when calculating wall heat gain.

Latent heat gain due to moisture transmission through walls, floors, and ceilings of modern refrigerated facilities is negligible. Data in Chapter 25 of the 2005 *ASHRAE Handbook—Fundamentals* may be used to calculate this load if moisture permeable materials are used.

Chapter 28 of the 2005 *ASHRAE Handbook—Fundamentals* gives outdoor design temperatures for major cities; values for 0.4% should be used.

Additional information on thermal insulation may be found in Chapters 23 and 24 of the 2005 *ASHRAE Handbook—Fundamentals.* Chapter 30 of the 2005 *ASHRAE Handbook—Fundamentals* discusses load calculation procedures in greater detail.

Heat Gain from Cooler Floors

Heat gain through cooler concrete slab floors is predicted using procedures developed by Chuangchid and Krarti (2000), who developed a simplified correlation of the total slab heat gain for coolers based on analytical results reported earlier by Chuangchid and Krarti (1999). Parameters in the solution include slab size and thermal resistance, insulation thermal resistance, soil thermal conductivity, water table depth and temperature, and indoor and outdoor air temperatures. The design procedure accommodates four slab insulation configurations: no insulation, uniform horizontal insulation, partial-horizontal-perimeter insulation, and partial-vertical-perimeter insulation. The slab size characteristic is expressed as the ratio of slab area A to exposed slab perimeter P. The result is an estimate of the annual mean and amplitude cooler floor heat gain that, when combined, gives the instantaneous floor heat at a specific time of year. The time-variation of the ground-coupled heat gain $q(t)$ for cooler slabs is

$$q(t) = q_m - q_a \cos[\omega(t - \phi)] \qquad (4)$$

where

- q_m = annual mean slab floor heat gain, Btu/h
- q_a = amplitude of annual variation slab floor heat gain, Btu/h
- ϕ = phase lag between cooler floor heat gain and outdoor air temperature variation, days
- t = time, days
- ω = constant for annual angular frequency, 0.0172 radians/day

These quantities are functions of input parameters such as building dimensions, soil properties, and insulation thermal resistance. Note that the time of origin (t = 0) in Equation (4) corresponds to the time when the soil surface temperature is minimum (typically January 15 in most U.S. locations). Details of the calculation method are given by Chuangchid and Krarti (2000).

Table 4 Example Input Data Required to Estimate Cooler Floor Heat Gain

Required Information	Example Values
Soil thermal conductivity k_s	10.47 Btu·in/h·ft^2·°F
Soil thermal diffusivity α_s	7.66 × 10^{-6} ft^2/s
Total floor area A	32 ft × 50 ft = 1600 ft^2
Exposed perimeter P	164 ft
Slab thickness y	4 in.
Slab thermal resistance R_f	3.33 ft^2·h·°F/Btu
Insulation thermal resistance R_i (partial uniform, partial, vertical)	20 ft^2·h·°F/Btu
Partial insulation length c	3 ft
Cooler inside temperature T_r	35°F
Annual mean T_m	40°F
Annual amplitude T_a	60°F
Water table depth b	Ignored (if depth is greater than 7 ft, water table effect can be ignored)

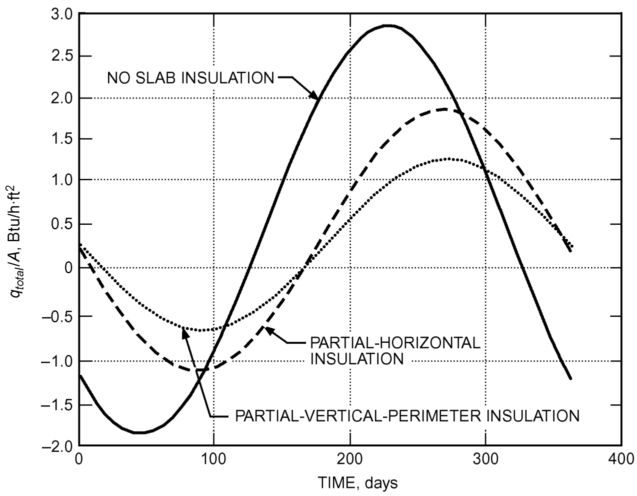

Fig. 1 Variation of Cooler Floor Heat Gain over One Year for Conditions in Table 4

For the conditions in Table 4, results for cooler slab floor heat gain are computed based on the method. As shown in Figure 1, heat gain through the cooler floor significantly varies and may even be negative at certain times of the year. The influence of partial-horizontal and partial-vertical-perimeter insulation on cooler floor heat gain for Table 4 conditions can also be seen in Figure 1.

Figure 2 shows the results for maximum heat transfer per unit area q_{max}/A by calculation where typical cold and warm climate temperatures are applied. These outdoor temperatures are given in Table 5. q_{max}/A is the sum of the mean and amplitude of heat gain per unit area and is, therefore, the maximum heat transfer rate that occurs at some time during the year. The influence of floor slab width and length is captured by the A/P parameter. Widths of both 10 and 30 m were used to generate Figure 2. The fact that the q_{max}/A curves in Figure 2 coalesce for different slab areas while still having the same A/P ratio and the same indoor, annual mean outdoor, and annual amplitude outdoor temperatures suggests that slab area does not affect the cooler floor slab heat gain rate per unit area with floor areas greater than about 100 m^2.

Table 5 Typical Annual and Annual Amplitude Outdoor Temperatures for Warm and Cold Climates

Temperature	Typical Cold Climate, °F	Typical Warm Climate, °F
Annual mean T_m	40	70
Annual amplitude T_a	60	30

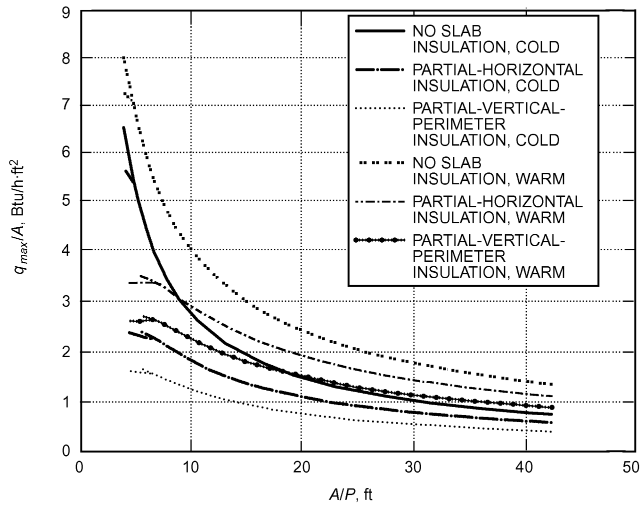

Fig. 2 Variation of q_{max}/A with A/P Using Conditions from Tables 4 and 5

PRODUCT LOAD

The primary refrigeration loads from products brought into and kept in the refrigerated space are (1) heat that must be removed to bring products to storage temperature and (2) heat generated by products (mainly fruits and vegetables) in storage. The quantity of heat to be removed can be calculated as follows:

1. Heat removed to cool from initial temperature to some lower temperature above freezing:

$$Q_1 = mc_1(t_1 - t_2) \tag{5}$$

2. Heat removed to cool from initial temperature to freezing point of product:

$$Q_2 = mc_1(t_1 - t_f) \tag{6}$$

3. Heat removed to freeze product:

$$Q_3 = mh_{if} \tag{7}$$

4. Heat removed to cool from freezing point to final temperature below freezing point:

$$Q_4 = mc_2(t_f - t_3) \tag{8}$$

where

Q_1, Q_2, Q_3, Q_4 = heat removed, Btu
m = weight of product, lb
c_1 = specific heat of product above freezing, Btu/lb·°F
t_1 = initial temperature of product above freezing, °F
t_2 = lower temperature of product above freezing, °F
t_f = freezing temperature of product, °F
h_{if} = latent heat of fusion of product, Btu/lb
c_2 = specific heat of product below freezing, Btu/lb·°F
t_3 = final temperature of product below freezing, °F

The refrigeration capacity required for products brought into storage is determined from the time allotted for heat removal and assumes that product is properly exposed to remove the heat in that time. The calculation is

$$q = \frac{Q_2 + Q_3 + Q_4}{n} \tag{9}$$

where

q = average cooling load, Btu/h
n = allotted time, h

Equation (9) only applies to uniform entry of product into storage. The refrigeration load created by nonuniform loading of a warm product may be much greater over a short period. See Chapter 15 for information on calculating the cooling load of warm product.

Specific heats above and below freezing for many products are given in Table 3 of Chapter 9. A product's latent heat of fusion may be estimated by multiplying the water content of the product (expressed as a decimal) by the latent heat of fusion of water, which is 144 Btu/lb. Most foods freeze between 26 and 31°F. When the exact freezing temperature is not known, assume that it is 28°F.

Example 1. 200 lb of lean beef is to be cooled from 65 to 40°F, then frozen and cooled to 0°F. The moisture content is 69.5%, so the latent heat is estimated as 144 × 0.695 = 100 Btu/lb. Estimate the cooling load.

Solution:

Specific heat of beef before freezing is listed in Table 3, Chapter 9, as 0.84 Btu/lb·°F; after freezing, 0.51 Btu/lb·°F.

To cool from 65 to 40°F in a chilled room:

$$200 \times 0.84(65 - 40) = 4200 \text{ Btu}$$

To cool from 40°F to freezing point in freezer:

$$200 \times 0.84(40 - 28) = 2016 \text{ Btu}$$

To freeze: $200 \times 100 = 20{,}000$ Btu

To cool from freezing to storage temperature:

$$200 \times 0.51(28 - 0) = 2856 \text{ Btu}$$

Total: $4200 + 2016 + 20{,}000 + 2856 = 29{,}072$ Btu

(Example 3 in Chapter 9 shows an alternative calculation method.)

Fresh fruits and vegetables respire and release heat during storage. This respiration heat varies by product and its temperature; the colder the product, the less heat of respiration. Table 9 in Chapter 9 gives heat of respiration rates for various products.

Calculations in Example 1 do not cover heat gained from product containers brought into the refrigerated space. When pallets, boxes, or other packing materials are a significant portion of the total mass introduced, this heat load should be calculated.

Equations (5) to (9) are used to calculate the total heat gain. Any moisture removed appears as latent heat gain. The amount of moisture involved is usually provided by the end-user as a percentage of product mass; so, with such information, the latent heat component of the total heat gain may be determined. Subtracting the latent heat component from the total heat gain determines the sensible heat component.

INTERNAL LOAD

Electrical Equipment. All electrical energy dissipated in the refrigerated space (from lights, motors, heaters, and other equipment) must be included in the internal heat load. Heat equivalents of electric motors are listed in Table 6.

Forklifts. Forklifts in some facilities can be a large and variable contributor to the load. Although many forklifts may be in a space at one time, they do not all operate at the same energy level. For example, the energy used by a forklift while it is elevating or lowering forks is different than when it is moving.

Processing Equipment. Grinding, mixing, or cooking equipment may be in refrigerated areas of food processing plants. Other

Table 6 Heat Gain from Typical Electric Motors

				Location of Motor and Driven Equipment with Respect to Conditioned Space or Airstream		
				A	B	C
Motor Name-plate or Rated Horse-power	Motor Type	Nominal rpm	Full Load Motor Effi-ciency, %	Motor in, Driven Equip-ment in, Btu/h	Motor out, Driven Equip-ment in, Btu/h	Motor in, Driven Equip-ment out, Btu/h
0.05	Shaded pole	1500	35	360	130	240
0.08				580	200	380
0.125				900	320	590
0.16				1160	400	760
0.25	Split phase	1750	54	1180	640	540
0.33			56	1500	840	660
0.50			60	2120	1270	850
0.75	3-Phase	1750	72	2650	1900	740
1			75	3390	2550	850
1.5			77	4960	3820	1140
2			79	6440	5090	1350
3			81	9430	7640	1790
5			82	15,500	12,700	2790
7.5			84	22,700	19,100	3640
10			85	29,900	24,500	4490
15			86	44,400	38,200	6210
20			87	58,500	50,900	7610
25			88	72,300	63,600	8680
30			89	85,700	76,300	9440
40				114,000	102,000	12,600
50				143,000	127,000	15,700
60				172,000	153,000	18,900
75			90	212,000	191,000	21,200
100				283,000	255,000	28,300
125				353,000	318,000	35,300
150			91	420,000	382,000	37,800
200				569,000	509,000	50,300
250				699,000	636,000	62,900

heat sources include equipment for packaging, glue melting, or shrink wrapping. Another possible load is the makeup air for equipment that exhausts air from a refrigerated space.

People. People add to the heat load, in amounts depending on factors such as room temperature, type of work being done, type of clothing worn, and size of the person. Heat load from a person q_p may be estimated as

$$q_p = 1295 - 11.5t \tag{10}$$

where t is the temperature of the refrigerated space in °F. Table 7 shows the average load from people in a refrigerated space as calculated from Equation (10).

When people first enter a storage they bring in additional surface heat. Thus, when many people enter and leave every few minutes, the load is greater than that listed in Table 7 and must be adjusted. A conservative adjustment is to multiply the values calculated in Equation (10) by 1.25.

Latent Load. The latent heat component of the internal load is usually very small compared to the total refrigeration load and is customarily regarded as all sensible heat in the total load summary. However, the latent heat component should be calculated where water is involved in processing or cleaning.

INFILTRATION AIR LOAD

Heat gain from infiltration air and associated equipment loads can amount to more than half the total refrigeration load of distribution warehouses and similar applications.

Table 7 Heat Equivalent of Occupancy

Refrigerated Space Temperature, °F	Heat Equivalent/Person, Btu/h
50	720
40	840
30	950
20	1050
10	1200
0	1300
–10	1400

Note: Heat equivalent may be estimated by Equation (10).

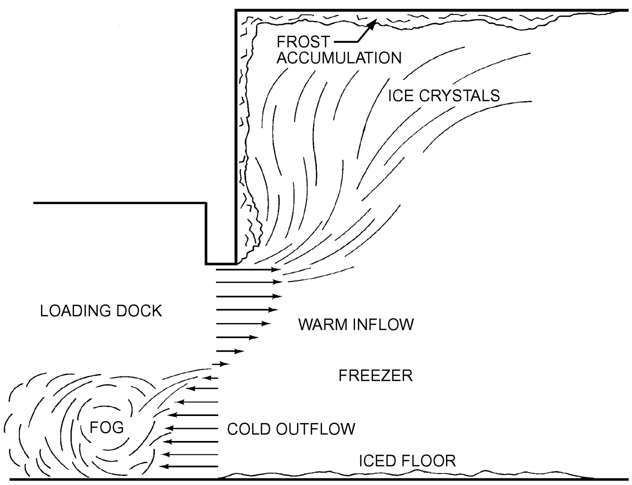

Fig. 3 Flowing Cold and Warm Air Masses for Typical Open Freezer Doors

Infiltration by Air Exchange

Infiltration most commonly occurs because of air density differences between rooms (Figures 3 and 4). For a typical case where the air mass flowing in equals the air mass flowing out minus any condensed moisture, the room must be sealed except at the opening in question. If the cold room is not sealed, air may flow directly through the door (discussed in the following section).

Heat gain through doorways from air exchange is as follows:

$$q_t = qD_tD_f(1 - E) \tag{11}$$

where

q_t = average heat gain for the 24 h or other period, Btu/h
q = sensible and latent refrigeration load for fully established flow, Btu/h
D_t = doorway open-time factor
D_f = doorway flow factor
E = effectiveness of doorway protective device

Gosney and Olama (1975) developed the following air exchange equation for fully established flow:

$$q = 795.6A(h_i - h_r)\rho_r(1 - \rho_i/\rho_r)^{0.5}(gH)^{0.5}F_m \tag{12}$$

where

q = sensible and latent refrigeration load, Btu/h
A = doorway area, ft^2
h_i = enthalpy of infiltration air, Btu/lb
h_r = enthalpy of refrigerated air, Btu/lb
ρ_i = density of infiltration air, lb/ft^3
ρ_r = density of refrigerated air, lb/ft^3
g = gravitational constant = 32.174 ft/s^2
H = doorway height, ft
F_m = density factor

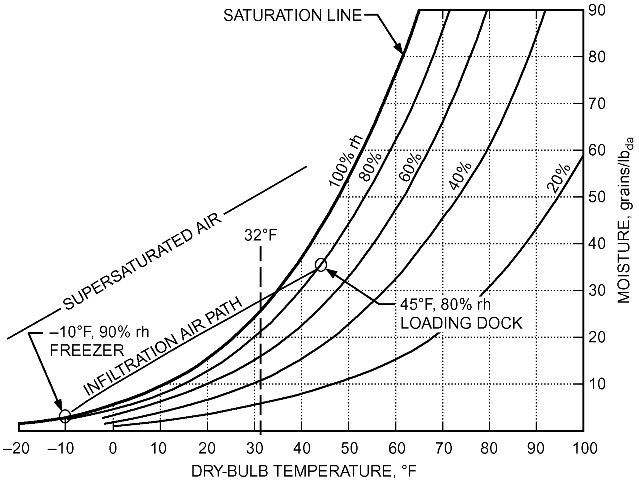

Fig. 4 Psychrometric Depiction of Air Exchange for Typical Freezer Doorway

Table 8 Sensible Heat Ratio R_s for Infiltration from Outdoors to Refrigerated Spaces

Outdoors		Cold Space at 90% rh Dry-Bulb Temperature, °F									
Temp. °F	rh, %	−30	−20	−10	0	10	20	30	40	50	60
100	50	0.59	0.57	0.55	0.53	0.51	0.49	0.48	0.46	0.45	0.45
	40	0.65	0.63	0.61	0.59	0.57	0.56	0.54	0.53	0.54	0.57
	30	0.71	0.69	0.68	0.66	0.65	0.64	0.63	0.64	0.66	0.76
	20	0.79	0.77	0.76	0.75	0.74	0.74	0.75	0.78	0.87	—
95	60	0.58	0.56	0.54	0.52	0.49	0.47	0.45	0.43	0.42	0.41
	50	0.62	0.60	0.58	0.56	0.54	0.52	0.51	0.49	0.48	0.50
	40	0.67	0.66	0.64	0.62	0.60	0.59	0.57	0.57	0.58	0.64
	30	0.73	0.72	0.70	0.69	0.67	0.66	0.66	0.68	0.72	0.89
90	60	0.61	0.59	0.57	0.55	0.52	0.50	0.48	0.46	0.45	0.45
	50	0.65	0.63	0.61	0.59	0.57	0.55	0.54	0.52	0.52	0.56
	40	0.70	0.68	0.67	0.65	0.63	0.61	0.61	0.61	0.63	0.74
	30	0.76	0.74	0.73	0.71	0.70	0.69	0.70	0.72	0.80	—

$$F_m = \left[\frac{2}{1 + (\rho_r/\rho_i)^{1/3}}\right]^{1.5} \tag{13}$$

(Chapter 6 of the 2005 *ASHRAE Handbook—Fundamentals* and the appropriate ASHRAE psychrometric chart list air enthalpy and density values.)

Equation (14), when used with Figure 5, is a simplification of Equation (12):

$$q = 3790 WH^{1.5}\left(\frac{Q_s}{A}\right)\left(\frac{1}{R_s}\right) \tag{14}$$

where

- q = sensible and latent refrigeration load, Btu/h
- Q_s/A = sensible heat load of infiltration air per square foot of doorway opening as read from Figure 5, ton/ft^2
- W = doorway width, ft
- R_s = sensible heat ratio of the infiltration air heat gain, from Table 7 or 8 (or from a psychrometric chart)

The values of R_s in Tables 8 and 9 are based on 90% rh in the cold room. A small error occurs where these values are used for 80 or 100% rh. This error, with loss of accuracy due to simplification, results in a maximum error for Equation (14) of approximately 4%.

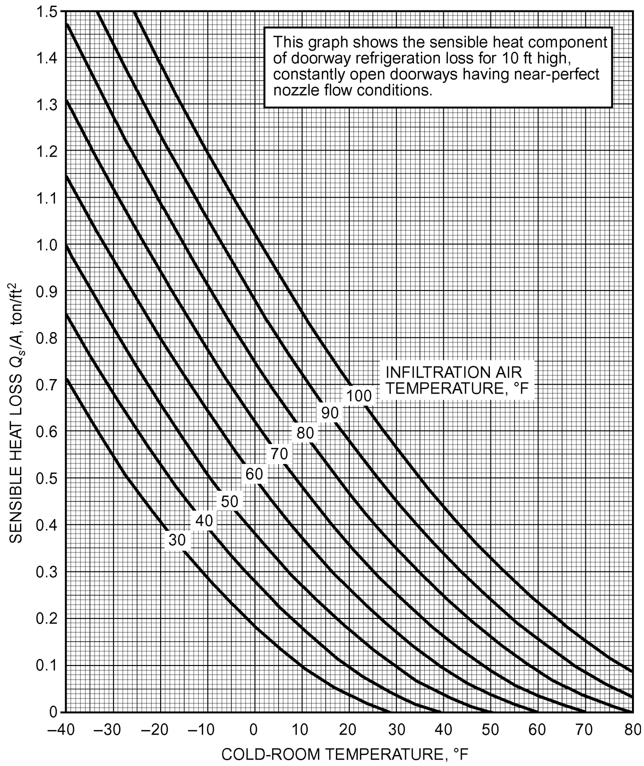

Fig. 5 Sensible Heat Gain by Air Exchange for Continuously Open Door with Fully Established Flow

For cyclical, irregular, and constant door usage, alone or in combination, the doorway open-time factor can be calculated as

$$D_t = \frac{(P\theta_p + 60\theta_o)}{3600\theta_d} \tag{15}$$

where

- D_t = decimal portion of time doorway is open
- P = number of doorway passages
- θ_p = door open-close time, seconds per passage
- θ_o = time door simply stands open, min
- θ_d = daily (or other) time period, h

The typical time θ_p for conventional pull-cord-operated doors ranges from 15 to 25 s per passage. The time for high-speed doors ranges from 5 to 10 s, although it can be as low as 3 s. The time for θ_o and θ_d should be provided by the user. Hendrix et al. (1989) found that steady-state flow becomes established 3 s after the cold-room door is opened. This fact may be used as a basis to reduce θ_p in Equation (15), particularly for high-speed doors, which may significantly reduce infiltration.

The doorway flow factor D_f is the ratio of actual air exchange to fully established flow. Fully established flow occurs only in the unusual case of an unused doorway standing open to a large room or to the outdoors, and where cold outflow is not impeded by obstructions (e.g., stacked pallets in or adjacent to the flow path in or outside the cold room). Under these conditions, D_f is 1.0.

Hendrix et al. (1989) found that a flow factor D_f of 0.8 is conservative for a 28°F temperature difference when traffic flow equals one entry and exit per minute through fast-operating doors. Downing and Meffert (1993) found a flow factor of 1.1 at temperature differences of 12 and 18°F. Based on these results, the recommended flow factor for cyclically operated doors with temperature differentials less than 20°F is 1.1, and the recommended flow factor for higher differentials is 0.8.

Table 9 Sensible Heat Ratio R_s for Infiltration from Warmer to Colder Refrigerated Spaces

Warm Space		Cold Space at 90% rh Dry-Bulb Temperature, °F									
Temp. °F	rh, %	–40	–30	–20	–10	0	10	20	30	40	50
70	100	0.60	0.58	0.56	0.53	0.50	0.47	0.44	0.41	0.37	0.34
	80	0.66	0.64	0.61	0.59	0.56	0.53	0.50	0.48	0.46	0.44
	60	0.72	0.70	0.68	0.66	0.63	0.61	0.59	0.58	0.59	0.64
	40	0.79	0.78	0.76	0.75	0.73	0.72	0.71	0.73	0.80	—
60	100	0.66	0.64	0.62	0.59	0.56	0.52	0.49	0.45	0.41	0.35
	80	0.71	0.69	0.67	0.64	0.62	0.59	0.56	0.53	0.52	0.53
	60	0.77	0.75	0.73	0.71	0.69	0.67	0.65	0.65	0.70	—
	40	0.83	0.82	0.81	0.79	0.78	0.77	0.78	0.83	—	—
50	100	0.72	0.70	0.67	0.64	0.61	0.57	0.53	0.49	0.43	—
	80	0.76	0.74	0.72	0.70	0.67	0.64	0.61	0.59	0.62	—
	60	0.81	0.80	0.78	0.76	0.74	0.72	0.71	0.75	—	—
	40	0.87	0.86	0.84	0.83	0.82	0.82	0.85	—	—	—
40	100	0.77	0.75	0.72	0.69	0.66	0.62	0.57	0.51	—	—
	80	0.81	0.79	0.77	0.74	0.72	0.69	0.66	0.67	—	—
	60	0.85	0.84	0.82	0.80	0.78	0.77	0.79	0.99	—	—
	40	0.90	0.89	0.88	0.87	0.86	0.88	0.97	—	—	—
30	100	0.82	0.80	0.77	0.74	0.70	0.66	0.59	—	—	—
	80	0.85	0.83	0.81	0.79	0.76	0.73	0.73	—	—	—
	60	0.88	0.87	0.86	0.84	0.83	0.83	0.94	—	—	—
	40	0.92	0.91	0.90	0.90	0.91	0.96	—	—	—	—
20	100	0.86	0.84	0.82	0.79	0.75	0.69	—	—	—	—
	80	0.89	0.87	0.85	0.83	0.81	0.80	—	—	—	—
	60	0.91	0.90	0.89	0.88	0.88	0.95	—	—	—	—
	40	0.94	0.94	0.93	0.94	0.97	—	—	—	—	—
10	100	0.90	0.88	0.86	0.83	0.78	—	—	—	—	—
	80	0.92	0.90	0.89	0.87	0.86	—	—	—	—	—
	60	0.94	0.93	0.92	0.92	0.96	—	—	—	—	—
	40	0.96	0.96	0.96	0.98	—	—	—	—	—	—
0	100	0.92	0.91	0.89	0.85	—	—	—	—	—	—
	80	0.94	0.93	0.92	0.91	—	—	—	—	—	—
	60	0.96	0.95	0.95	0.97	—	—	—	—	—	—
	40	0.97	0.97	0.98	—	—	—	—	—	—	—

The effectiveness E of open-doorway protective devices is 0.95 or higher for newly installed strip, fast-fold, and other non-tight-closing doors. However, depending on the traffic level and door maintenance, E may quickly drop to 0.8 on freezer doorways and to about 0.85 for other doorways. Airlock vestibules with strip or push-through doors have an effectiveness ranging between 0.95 and 0.85 for freezers and between 0.95 and 0.90 for other doorways. The effectiveness of air curtains ranges from very poor to more than 0.7. For a wide-open door with no devices, $E = 0$ in Equation (11).

Infiltration by Direct Flow Through Doorways

Negative pressure elsewhere in the building, created by mechanical air exhaust without mechanical air replenishment, is a common cause of heat gain from infiltration of warm air. In refrigerated spaces with constantly or frequently open doorways or other through-the-room passageways, this air flows directly through the doorway. The effect is identical to that of open doorways exposed to wind, and heat gain may be very large. Equation (16) for heat gain from infiltration by direct inflow provides the basis for either correcting negative pressure or adding to refrigeration capacity:

$$q_t = 60VA(h_i - h_r)\rho_r D_t \tag{16}$$

where

q_t = average refrigeration load, Btu/h
V = average air velocity, ft/min
A = opening area, ft^2
h_i = enthalpy of infiltration air, Btu/lb
h_r = enthalpy of refrigerated air, Btu/lb
ρ_r = density of refrigerated air, lb/ft^3
D_t = decimal portion of time doorway is open

A is the smaller of the inflow and outflow openings. If the smaller area has leaks around truck loading doors in well-maintained loading docks, the leakage area can vary from 0.3 ft^2 to over 1.0 ft^2 per door. For loading docks with high merchandise movement, the facility manager should estimate the time these doors are fully or partially open.

To evaluate V, the magnitude of negative pressure or other flow-through force must be known. If differential pressure across a doorway can be determined, velocity can be predicted by converting static head to velocity head. However, attempting to estimate differential pressure is usually not possible; generally, the alternative is to assume a commonly encountered velocity. The typical air velocity through a door is 60 to 300 ft/min.

The effectiveness of non-tight-closing devices on doorways subject to infiltration by direct airflow cannot be readily determined. Depending on the pressure differential, its tendency to vary, and the ratio of inflow area to outflow area, the effectiveness of these devices can be very low.

Sensible and Latent Heat Components

When calculating q_t for infiltration air, the sensible and latent heat components may be obtained by plotting the infiltration air path on the appropriate ASHRAE psychrometric chart, determining the air sensible heat ratio R_s from the chart, and calculating as follows:

$$\text{Sensible heat: } q_s = q_t R_s \tag{17}$$

$$\text{Latent heat: } q_l = q_t(1 - R_s) \tag{18}$$

where $R_s = \Delta h_s/\Delta h_t$.

EQUIPMENT RELATED LOAD

Heat gains associated with refrigeration equipment operation consist essentially of the following:

- Fan motor heat where forced-air circulation is used
- Reheat where humidity control is part of the cooling
- Heat from defrosting where the refrigeration coil operates at a temperature below freezing and must be defrosted periodically, regardless of the room temperature

Fan motor heat must be computed based on the actual electrical energy consumed during operation. Propeller fan motors are mounted in the airstream on many cooling units because the cold air extends the power range of the motor. For example, a standard motor in a –10°F freezer operates satisfactorily at a 25% overload to the rated (nameplate) power. Heat gain from fan motors should be based on the actual run time. Generally, fans on cooling units operate continuously except during the defrost period. However, fans may be cycled on and off to control temperature and save energy.

Cole (1989) characterized and quantified the heat load associated with defrosting using hot gas. Other common defrost methods use electricity or water. Generally, heat gain from a cooling unit with electric defrost is greater than the same unit with hot-gas defrost; heat gain from a unit with water defrost is even less. Moisture that evaporates into the space during defrost must also be added to the refrigeration load.

Some heat from defrosting is added only to the refrigerant, and the rest is added to the space. To accurately select refrigeration equipment, a distinction should be made between equipment heat loads that are in the refrigerated space and those that are introduced directly to the refrigerating fluid.

Equipment heat gain is usually small at space temperatures above approximately 30°F. Where reheat or other artificial loads are not imposed, total equipment heat gain is about 5% or less of the total load. However, equipment heat gain becomes a major portion of the total load at freezer temperatures. For example, at –20°F the theoretical contribution to total refrigeration load from fan power and coil defrosting alone can exceed, for many cases, 15% of the total load (assuming proper control of defrosting so that the space is not heated excessively).

ASHRAE research project RP-1094 (Sherif et al. 2002) found that, where excessive moisture persists during normal freezer operation and/or freezer air becomes supersaturated for an extended period of time, significantly more heat gain from coil defrosting should be included in calculating refrigeration load. Excessive moisture and supersaturated air result if care is not taken to prevent air infiltration into the freezer and/or if a high coil temperature difference (TD) is imposed. Coil TD is the difference between the temperatures of air entering the coil and refrigerant inside the coil. Excessive coil TDs typically exist if entering air temperature rises, refrigerant temperature drops, or both. Coil refrigerant temperatures typically drop as an intuitive response by freezer operators to the rising air temperature inside the freezer. Air temperatures typically rise after excessive coil frost build-up, resulting in coil heat transfer performance degradation. Excessive frost build-up typically occurs if supersaturated air is allowed inside the freezer. Sherif et al. (2002) suggested guidelines to prevent this chain reaction of events by using the saturation curve of the psychrometric chart as a guide to select the proper combination of coil entering air temperature, refrigerant temperature, and sensible heat ratio for a given freezer temperature. The study imposed upper limits on the allowable coil TD for prescribed values of these variables. In cases where excessive moisture and/or supersaturated air are allowed for an extended period of time during the refrigeration portion of the refrigeration/defrost (R/D) cycle, the study recommends that the fan/defrost contribution be raised from 15% to as high as 30%, depending on the degree of supersaturation and how long it prevailed in the freezer.

SAFETY FACTOR

Generally, the calculated load is increased by a factor of 10% to allow for possible discrepancies between design criteria and actual operation. This factor should be selected in consultation with the facility user and should be applied individually to the first four heat load segments.

A separate factor should be added to the coil-defrosting portion of the equipment load for freezer applications that use dry-surface refrigerating coils. However, few data are available to predict heat gain from coil defrosting. For this reason, the experience of similar facilities should be sought to obtain an appropriate defrosting safety factor. Similar facilities should have similar room sensible heat ratios.

The nature of frost accumulation on cooling coils also affects cooling unit performance and, therefore, refrigeration load. A very-low-density frost forms under certain conditions, particularly where the room sensible heat ratio is more than a few points below 1.0. This type of frost is difficult to remove and tends to block airflow through the cooling coils more readily. Removing this frost requires more frequent and longer periods of defrosting of cooling units, which increases the refrigeration load.

Example 2. Calculate the total refrigeration load for a freezer storage with design criteria as follows:

Design Criteria

Summer: 92°F db, 80°F wb
Comments: 0.4% summer and winter conditions
Room dimensions: 133 by 222 by 30 ft
Floor area: 29,526 ft^2
Pallet positions: 4800
Turns per yr: 20
Use factor: 90%

Ambient Design Conditions

Design Room Temperature, –10°F

		Sun Effect, °F	Surface Temperature, °F
Roof		10	102
Floor		0	60
Wall,	east	0	92
	north	0	92
	west[a]	0	28
	south[b]	0	45

[a]Adjacent to refrigerated meat room held at 28°F
[b]Adjacent to refrigerated truck dock held at 44.5°F

Insulation Thickness

		in.	k, Btu·in/h·ft^2·°F	U, Btu/h·ft^2·°F	R, ft^2·h·°F/Btu
Roof		6	0.142	0.02367	42.25
Floor		6	0.188	0.03133	31.91
Wall,	east	4	0.121	0.03025	33.06
	north	4	0.121	0.03025	33.06
	west	4	0.121	0.03025	33.06
	south	4	0.121	0.03025	33.06

k = thermal conductivity U = k/in. thickness R = thermal resistance

Solution:

Heat Transmission

	Length, ft	Width, ft	Height, ft	Adj. Temp.	U, Btu/h·ft^2·°F	Area A, ft^2	Δt, °F	Load, tons*
Roof	133	222	0	102	0.02367	29,526	112	6.52
Floor	133	222	0	60	0.03133	29,526	70	5.40
Wall,								
east	222	0	30	92	0.03025	6660	102	1.71
north	133	0	30	92	0.03025	3990	102	1.03
west	222	0	30	28	0.03025	6660	38	0.64
south	133	0	30	45	0.03025	3990	55	0.55
							Safety, 20%	3.17
							Total Transmission Load, tons	19.02

*$UA\Delta t$/12,000 U = Heat transfer coefficient

Product (see Chapter 9)
Pallets per 24 h: 420
Pounds per pallet: 2500
Mass flow: 43,750 lb/h
Temperature in: 5°F
Temperature out: –10°F
Specific heat: 0.45 Btu/lb·°F

Product Load = (43,750)(15)(0.45)/12,000 = 24.61 tons

Motors (Other than air unit fan motors): None

Infiltration

Door Openings	**Door Type 1**	**Door Type 2**	**Door Type 3**
From	**Dock**	**Dry**	**Meat**
To	**Freezer**	**Freezer**	**Freezer**
Door width, ft	8	8	8
Door height, ft	10	10	10
Enthalpy h_i	14.4	43.8	9.5
Enthalpy h_r	–2	–2	–2
Density ρ_r	0.0883	0.0883	0.0883
Density ρ_i	0.0782	0.0697	0.081
Doorway flow factor D_f	0.7	0.7	0.7
Doorway time factor D_t	0.14583	0.0219	0.0417
Effectiveness device E_f	0	0	0
Number of doors	3	1	1
Load per door, tons	4.61	2.55	0.79
Total load, tons	13.84	2.55	0.79
		Infiltration Load, tons	17.19

$$\text{Infiltration Load} = \frac{795.6A(h_i - h_r)\rho_r(1 - \rho_i/\rho_r)^{0.5}(gH)^{0.5}F_m[D_f D_t(1 - E_f)]}{12{,}000}$$

A = doorway area, ft^2
h_f = enthalpy incoming air through doorway from adjacent area, Btu/lb
h_r = enthalpy room air, Btu/lb
ρ_r = density room air, lb/ft^3
ρ_i = density incoming air, lb/ft^3
g = gravitational acceleration, 32.2 ft/s^2
H = doorway height, ft
F_m = density factor = $[2/(1 + (\rho_r/\rho_i)^{1/3}]^{1.5}$
D_f = doorway flow factor
D_t = percentage time period doorway is open during 1 h period, average, expressed as a decimal
E_f = effectiveness factor for open-doorway protective device such as air curtain or plastic strip curtain

Lights
Lighting level: 1.0 W/ft^2
Floor area: 29,526 ft^2

Lighting Load = (1.0)(29,526)(3.413)/12,000 = 8.40 tons

People
Number of persons: 3
Room temperature: –10°F

People Load = (3)[1295 – 11.5(–10)]/12,000 = 0.35 tons

Trucks
Number of trucks: 3
Horsepower per truck: 7.5

Truck Load = (3)(7.5)(2545)/12,000 = 4.77 tons

Fans
Number of fans: 15
Horsepower per fan: 1.50 nominal (4960 Btu/h heat gain per Table 6)

Fan Load = (15)(4960)/12,000 = 6.20 tons

Load Summary	Load, tons
Transmission	19.02
Product	24.61
Motor	0.00
Infiltration	17.19
Lighting	8.40
People	0.35
Trucks	4.77
Fan motors	6.20
Subtotal	80.54
Safety 10%	8.05
Total Load, tons	88.59

LOAD DIVERSITY

When computing refrigeration load, the most conservative approach is to calculate each part at its expected **peak value**. The combined result can overstate the actual total load by as much as 20 to 50%.

The reason for such overestimates is that, typically, all the loads do not occur at the same time of day. Furthermore, many of them are not always at their maximum value when they do occur. The consequence is that oversized refrigerating equipment is often installed for a plant. Some of it may not ever run, or if a single piece of equipment runs, it may do so inefficiently.

There are two ways that this mismatch of estimated load to actual load is addressed: (1) a rigorous computational method, or (2) a "rule-of-thumb" adjustment to the final estimate, determined by judgment.

The rigorous approach is to use the **hour-by-hour load calculation** method. In the past, when most calculations were done manually, this procedure was tedious and time consuming and therefore, not often used. The advent of computer software to perform such calculations eases the task. It is necessary, however, to have comprehensive operating data for the facility under consideration so complete and precise information regarding load magnitude and time of occurrence can be input to the computer. Ballard (1992) describes the procedure.

The other method, used by experienced engineers, is to use a **diversity factor**. Based on analysis of the load data and an understanding of how, or more importantly, when or how often each load element will occur, the designer will often apply a factor ranging from 0.7 to 0.85 to the calculated final total load. That result is the load on which the selection of equipment is based.

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