CHAPTER 12

REFRIGERATION LOAD

THE segments of total refrigeration load are (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 the refrigeration of specific foods can be found in Chapters 14 and 16 through 28.

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, W
A = outside area of section, m^2
Δt = difference between outside air temperature and air temperature of the refrigerated space, °C

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, $W/(m^2 \cdot K)$
x = wall thickness, m
k = thermal conductivity of wall material, $W(m \cdot K)$
h_i = inside surface conductance, $W/(m^2 \cdot K)$
h_o = outside surface conductance, $W/(m^2 \cdot K)$

A value of 1.6 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 .

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 decrease with age due to factors discussed in Chapter 23 of the 2001 *ASHRAE Handbook—Fundamentals*). Chapter 25 of the 2001 *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 2001 *ASHRAE Handbook—Fundamentals* describes how to calculate heat gain through walls and roofs with thermal bridges.The metal surfaces of prefabricated or insulated panels have a negligible effect on thermal performance and should not be considered in calculating the U-factor.

Table 1 Thermal Conductivity of Cold Storage Insulation

Insulation	Thermal Conductivity k, W/(m·K)
Polyurethane board (R-11 expanded)	0.023 to 0.026
Polyisocyanurate, cellular (R-141b expanded)	0.027
Polystyrene, extruded (R-142b)	0.035
Polystyrene, expanded (R-142b)	0.037
Corkboard[b]	0.043
Foam glass[c]	0.044

[a] Values are for a mean temperature of 24°C and insulation is aged 180 days.
[b] Seldom used insulation. Data is only for reference.
[c] Virtually no effects due to aging.

Table 2 Minimum Insulation Thickness

Storage Temperature	Expanded Polyisocyanurate Thickness	
	Northern U.S.	Southern U.S.
°C	mm	mm
10 to 16	50	50
4 to 10	50	50
−4 to 4	50	75
−9 to −4	75	75
−18 to −9	75	100
−26 to −18	100	100
−40 to −26	125	125

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 °C	South Wall °C	West Wall °C	Flat Roof °C
Dark-colored surfaces				
Slate roofing	5	3	5	11
Tar roofing				
Black paint				
Medium-colored surfaces				
Unpainted wood	4	3	4	9
Brick				
Red tile				
Dark cement				
Red, gray, or green paint				
Light-colored surfaces				
White stone	3	2	3	5
Light colored cement				
White paint				

Note: Add °C 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 the heat load. The 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 2001 *ASHRAE Handbook—Fundamentals* may be used to calculate this load if moisture permeable materials are used.

Chapter 27 of the 2001 *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 2001 *ASHRAE Handbook—Fundamentals*. Chapter 29 of the 2001 *ASHRAE Handbook—Fundamentals* discusses load calculation procedures in greater detail.

PRODUCT LOAD

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

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

$$Q_1 = mc_1\,(t_1 - t_2) \tag{4}$$

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

$$Q_2 = mc_1\,(t_1 - t_f) \tag{5}$$

3. Heat removed to freeze the product:

$$Q_3 = mh_{if} \tag{6}$$

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

$$Q_4 = mc_2\,(t_f - t_3) \tag{7}$$

where

Q_1, Q_2, Q_3, Q_4 = heat removed, kJ
m = mass of product, kg
c_1 = specific heat of product above freezing, kJ/(kg·K)
t_1 = initial temperature of product above freezing, °C
t_2 = lower temperature of product above freezing, °C
t_f = freezing temperature of product, °C
h_{if} = latent heat of fusion of product, kJ/kg
c_2 = specific heat of product below freezing, kJ/(kg·K)
t_3 = final temperature of product below freezing, °C

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

$$q = \frac{Q_2 + Q_3 + Q_4}{3600\,n} \tag{8}$$

where

q = average cooling load, kW
n = allotted time, h

Equation (8) only applies to uniform entry of the product into storage. The refrigeration load created by nonuniform loading of a warm product may be much greater over a short period. See Chapter 14 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 8. 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 334 kJ/kg. Most food products freeze in the range of −3 to −0.5 °C. When the exact freezing temperature is not known, assume that it is −2°C.

Example 1. 100 kg of lean beef is to be cooled from 18 to 4°C, then frozen and cooled to −18°C. The moisture content is 69.5%, so the latent heat is estimated as 233 kJ/kg. Estimate the cooling load.

Solution:

Specific heat of beef before freezing is listed in Table 3, Chapter 8 as 3.23 kJ/(kg·K); after freezing, 1.68 kJ/(kg·K).

To cool from 18 to 4°C in a chilled room:

$$100 \times 3.23\,(18 - 4) = 4520 \text{ kJ}$$

To cool from 4°C to freezing point in freezer:

$$100 \times 3.23[4 - (-2)] = 1940 \text{ kJ}$$

To freeze: $100 \times 233 = 23\,300$ kJ

To cool from freezing to storage temperature:

$$100 \times 1.68\,[(-2) - (-18)] = 2690 \text{ kJ}$$

Total: $4520 + 1940 + 23\,300 + 2690 = 32\,450$ kJ

(Example 3 in Chapter 8 shows an alternative calculation method.)

Fresh fruits and vegetables respire and release heat during storage. This heat produced by respiration varies with the product and its temperature; the colder the product, the less heat of respiration. Table 9 in Chapter 8 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 (4) through (8) 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 4.

Table 4 Heat Gain from Typical Electric Motors

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

(Columns A, B, C: Location of Motor and Driven Equipment with Respect to Conditioned Space or Airstream)

Fork Lifts. Fork lifts in some facilities can be a large and variable contributor to the load. While many fork lifts 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 fork lift while it is elevating or lowering forks is different than when it is moving.

Processing Equipment. Grinding, mixing, or even cooking equipment may be in the refrigerated areas of food processing plants. Other 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, and this load varies depending on such factors 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

$$q_p = 272 - 6t \quad (9)$$

where t is the temperature of the refrigerated space in °C. Table 5 shows the average load from people in a refrigerated space as calculated from Equation (9).

When people first enter a storage they bring in additional surface heat. As a result, when many people enter and leave every few minutes the load is greater than that listed in Table 5 and must be adjusted. A conservative adjustment would be to multiply the values in calculated in Equation (9) by 1.25.

Table 5 Heat Equivalent of Occupancy

Refrigerated Space Temperature, °C	Heat Equivalent/Person, W
10	210
5	240
0	270
–5	300
–10	330
–15	360
–20	390

Note: Heat equivalent may be estimated by $q_p = 272 - 6t$(°C)

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.

Infiltration by Air Exchange

Infiltration most commonly occurs because of air density differences between rooms (Figures 1 and 2). 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:

$$q_t = qD_tD_f(1 - E) \quad (10)$$

where

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

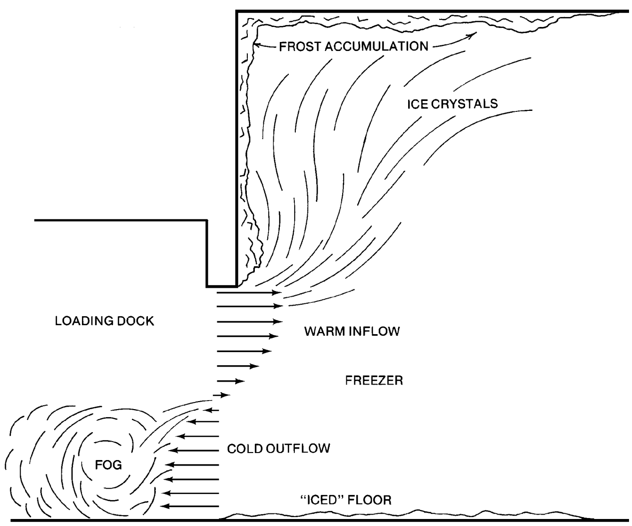

Fig. 1 Flowing Cold and Warm Air Masses that Occur for Typical Open Freezer Doors

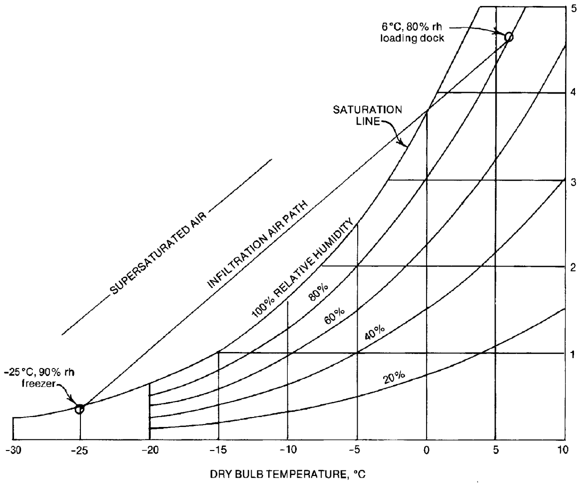

Fig. 2 Psychrometric Depiction of Air Exchange for Typical Freezer Doorway

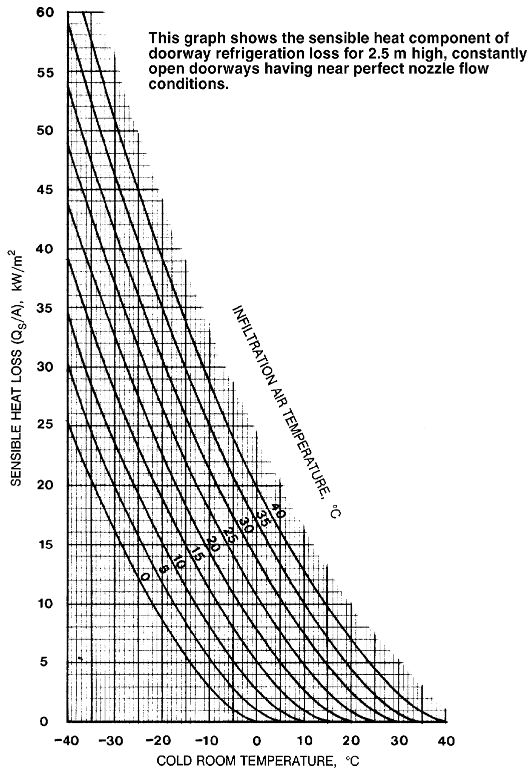

Fig. 3 Sensible Heat Gain by Air Exchange for Continuously Open Door with Fully Established Flow

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

$$q = 0.221A\,(h_i - h_r)\,\rho_r\,(1 - \rho_i/\rho_r)^{0.5}\,(gH)^{0.5}F_m \tag{11}$$

where

q = sensible and latent refrigeration load, kW
A = doorway area, m²
h_i = enthalpy of infiltration air, kJ/kg
h_r = enthalpy of refrigerated air, kJ/kg
ρ_i = density of infiltration air, kg/m³
ρ_r = density of refrigerated air, kg/m³
g = gravitational constant = 9.81 m/s²

Table 6 Sensible Heat Ratio *Rs* for Infiltration from Outdoors to Refrigerated Spaces

Outdoor Cond. DB °C	WB °C	rh, %	Cold Space at 90% rh Dry-Bulb Temperature, °C −30	−25	−20	−15	−10	−5	0	5	10	15
30	19.7	30	0.76	0.75	0.74	0.73	0.72	0.72	0.73	0.77	0.87	—
	21.8	40	0.71	0.69	0.68	0.66	0.65	0.63	0.63	0.64	0.68	0.83
	23.9	50	0.66	0.64	0.62	0.60	0.59	0.57	0.56	0.55	0.56	0.62
	25.8	60	0.62	0.60	0.58	0.56	0.54	0.52	0.50	0.48	0.48	0.49
35	19.0	20	0.80	0.79	0.78	0.77	0.77	0.77	0.79	0.84	0.96	—
	21.6	30	0.72	0.71	0.69	0.68	0.67	0.66	0.67	0.68	0.72	0.86
	24.0	40	0.66	0.64	0.63	0.61	0.59	0.58	0.57	0.57	0.58	0.63
	26.3	50	0.61	0.59	0.57	0.55	0.53	0.52	0.50	0.49	0.48	0.50
	28.3	60	0.56	0.54	0.53	0.51	0.49	0.47	0.45	0.43	0.42	0.41
40	20.7	20	0.76	0.75	0.74	0.73	0.72	0.72	0.73	0.75	0.82	0.98
	23.6	30	0.68	0.66	0.65	0.63	0.62	0.61	0.60	0.61	0.62	0.68
	26.2	40	0.61	0.59	0.58	0.56	0.54	0.53	0.52	0.51	0.50	0.52
	28.6	50	0.55	0.54	0.52	0.50	0.48	0.47	0.45	0.43	0.42	0.42

H = doorway height, m
F_m = density factor

$$F_m = \left(\frac{2}{1 + (\rho_r/\rho_i)^{1/3}}\right)^{1.5} \tag{12}$$

(Chapter 6 of the 2001 *ASHRAE Handbook—Fundamentals* and the ASHRAE Psychrometrics Chart list air enthalpy and density values.)

Equation (13), when used with Figure 3, is a simplification of Equation (11):

$$q = 0.577WH^{1.5}\left(\frac{Q_s}{A}\right)\left(\frac{1}{R_s}\right) \tag{13}$$

where

q = sensible and latent refrigeration load, kW
Q_s/A = sensible heat load of infiltration air per square metre of doorway opening as read from Figure 3, kW/m²
W = doorway width, m
R_s = sensible heat ratio of the infiltration air heat gain, from Tables 6 or 7 (or from a psychrometric chart)

The values of R_s in Tables 6 and 7 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 together with loss of accuracy due to simplification results in a maximum error for Equation (13) of approximately 4%.

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

$$D_t = \frac{(P\theta_p + 60\theta_o)}{3600\ \theta_d} \tag{14}$$

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

Table 7 Sensible Heat Ratio *Rs* for Infiltration from Warmer to Colder Refrigerated Spaces

Warm Space Temp. °C	Warm Space rh, %	Cold Space at 90% rh Dry-Bulb Temperature, °C –40	–35	–30	–25	–20	–15	–10	–5	0
20	100	0.62	0.60	0.57	0.55	0.53	0.50	0.44	0.44	0.41
	80	0.67	0.65	0.63	0.61	0.58	0.56	0.53	0.51	0.48
	60	0.73	0.71	0.69	0.68	0.65	0.63	0.61	0.60	0.59
	40	0.80	0.79	0.78	0.76	0.75	0.73	0.73	0.73	0.76
	20	0.89	0.88	0.88	0.87	0.87	0.87	0.80	0.93	—
15	100	0.67	0.65	0.63	0.60	0.58	0.55	0.51	0.48	0.45
	80	0.72	0.70	0.68	0.66	0.63	0.61	0.58	0.55	0.53
	60	0.77	0.76	0.74	0.72	0.70	0.68	0.66	0.65	0.66
	40	0.84	0.83	0.81	0.80	0.79	0.78	0.78	0.79	0.87
	20	0.91	0.91	0.90	0.90	0.90	0.91	0.93	—	—
10	100	0.72	0.70	0.68	0.65	0.62	0.59	0.56	0.52	0.48
	80	0.76	0.75	0.73	0.70	0.68	0.65	0.63	0.60	0.59
	60	0.81	0.80	0.78	0.76	0.75	0.73	0.71	0.71	0.77
	40	0.87	0.86	0.85	0.84	0.83	0.82	0.83	0.88	—
	20	0.93	0.93	0.92	0.92	0.92	0.94	0.99	—	—
5	100	0.77	0.75	0.73	0.70	0.67	0.64	0.60	0.55	0.49
	80	0.81	0.79	0.77	0.75	0.72	0.70	0.67	0.65	0.68
	60	0.85	0.83	0.82	0.80	0.79	0.77	0.77	0.79	—
	40	0.89	0.88	0.88	0.87	0.86	0.86	0.89	—	—
	20	0.95	0.94	0.94	0.94	0.95	0.98	—	—	—
0	100	0.81	0.79	0.77	0.74	0.71	0.67	0.63	0.56	—
	80	0.84	0.83	0.81	0.79	0.76	0.74	0.71	0.71	—
	60	0.88	0.86	0.85	0.84	0.82	0.81	0.83	0.98	—
	40	0.92	0.91	0.90	0.89	0.89	0.91	0.98	—	—
		0.96	0.96	0.96	0.96	0.98	—	—	—	—
-5	100	0.85	0.83	0.81	0.79	0.75	0.71	0.65	—	—
	80	0.88	0.86	0.85	0.83	0.81	0.78	0.78	—	—
	60	0.91	0.90	0.88	0.87	0.87	0.87	0.98	—	—
	40	0.94	0.93	0.93	0.92	0.93	0.98	—	—	—
	20	0.97	0.97	0.97	0.98	—	—	—	—	—
-10	100	0.88	0.87	0.85	0.82	0.79	0.74	—	—	—
	80	0.91	0.89	0.88	0.86	0.85	0.84	—	—	—
	60	0.93	0.92	0.91	0.91	0.91	0.97	—	—	—
	40	0.95	0.95	0.95	0.95	0.98	—	—	—	—
	20	0.98	0.98	0.99	—	—	—	—	—	—
-15	100	0.91	0.90	0.88	0.85	0.81	—	—	—	—
	80	0.93	0.92	0.91	0.89	0.89	—	—	—	—
	60	0.95	0.94	0.94	0.94	0.98	—	—	—	—
	40	0.97	0.97	0.97	0.99	—	—	—	—	—
	20	0.99	0.99	1.00	—	—	—	—	—	—

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 (14), 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 the cold outflow is not impeded by obstructions (such as stacked pallets within or adjacent to the flow path either within 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 16 K temperature difference when traffic flow equals one entry and exit per minute through fast-operating doors. Tests by Downing and Meffert (1993) at temperature differences of 7 K and 10 K found a flow factor of 1.1. Based on these results, the recommended flow factor for cyclically operated doors with temperature differentials less than 11°C is 1.1, and the recommended flow factor for higher differentials is 0.8.

The effectiveness *E* of open-doorway protective devices is 0.95 or higher for newly installed strip doors, fast fold doors, and other nontight-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 doors 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 (10).

Infiltration by Direct Flow Through Doorways

A negative pressure created elsewhere in the building because of mechanical air exhaust without mechanical air replenishment is a common cause of heat gain from infiltration of warm air. In refrigerated spaces equipped 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 the wind and the heat gain may be very large. Equation (15) for heat gain from infiltration by direct inflow provides the basis for either correcting the negative pressure or adding to refrigeration capacity:

$$q_t = VA(h_i - h_r)\,\rho_r D_t \tag{15}$$

where

q_t = average refrigeration load, kW
V = average air velocity, m/s
A = opening area, m^2
h_i = enthalpy of infiltration air, kJ/kg
h_r = enthalpy of refrigerated air, kJ/kg
ρ_r = density of refrigerated air, kg/m^3
D_t = decimal portion of time doorway is open

The area *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 0.03 m^2 to over 0.1 m^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 velocity *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 0.3 to 1.5 m/s.

The effectiveness of nontight-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:

$$\text{Sensible heat:}\quad q_s = q_t R_s \tag{16}$$

$$\text{Latent heat:}\quad q_l = q_t(1 - R_s) \tag{17}$$

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

EQUIPMENT RELATED LOAD

Heat gain associated with the operation of the refrigeration equipment 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. The fan motor is mounted in the airstream on many cooling units with propeller fans because the cold air extends the power range of the motor. For example, a standard motor in a –23°C freezer operates satisfactorily at a 25% overload to the rated (nameplate) power. The heat gain from the fan motors should be based on the actual run time. Generally, fans on cooling units are operated continuously except during the defrost period. But, increasingly, fans are being 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, the heat gain from a cooling unit with electric defrost is greater than the same unit with hot gas defrost, and heat gain from a unit with water defrost is even less. The moisture that evaporates into the space during the defrost cycle must also be added to the refrigeration load.

Some of the 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 those 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 –1°C. 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 –30°C the theoretical contribution to total refrigeration load due to fan power and coil defrosting alone can exceed, for many cases, 15% of the total load. This percentage assumes proper control of defrosting so that the space is not heated excessively.

SAFETY FACTOR

Generally, the calculated load is increased by a factor of 10% to allow for possible discrepancies between the 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, little data are available to predict heat gain from coil defrosting. For this reason, the experience of existing 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 the cooling coils also affects the performance of the cooling units and, therefore, the 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 the airflow through the cooling coils more readily. Removing this type of frost requires more frequent and longer periods of defrosting of the 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

Summar: 33°C db, 27°C wb
Comments: 0.4% summer and winter conditions
Room dimensions: 40.5 by 68 by 9 m
Floor area: 2743 m^2
Pallet positions: 4800
Turns per yr: 20
Use factor: 90%

Ambient Design Conditions

Design Room Temperature, –23°C

	Sun Effect, K	Surface Temperature, °C
Roof	5.5	39
Floor	0	16
Wall, East	0	33
North	0	33
West[a]	0	–2
South[b]	0	7

[a]Adjacent to refrigerated meat room held at –2°C
[b]Adjacent to refrigerated truck dock held at 7°C

Insulation Thickness

	m	k, W/(m·K)	U, W/(m^2·K)	R, m^2·K/W
Roof	0.152	2.048	3.414	7.441
Floor	0.152	2.711	4.518	5.620
Wall, East	0.102	1.745	4.363	5.822
North	0.102	1.745	4.363	5.822
West	0.102	1.745	4.363	5.822
South	0.102	1.745	4.363	5.822

k = thermal conductivity U = k/m thickness R = thermal resistance

Solution:

Heat Transmission

	Length, m	Width, m	Height, m	Adjacent Temp.	U, W/(m^2·K)	Area A, m^2	ΔT, K	Load kW*
Roof	40.5	67.7	0	38.9	0.1344	2743	62	22.9
Floor	40.5	67.7	0	15.5	0.1779	2743	39	19.0
Walls, East	67.7	0	9.2	33.3	0.1718	618	57	6.01
North	40.5	0	9.2	33.3	0.1718	370	57	3.62
West	67.7	0	9.2	–2.2	0.1718	618	21	2.25
South	40.5	0	9.2	7.2	0.1718	370	31	1.93
							Safety, 20%	3.17
							Total Transmission Load, kW	66.89

*$UA\Delta t$ U = Heat transfer coefficient

Product (see Chapter 8, Thermal Properties of Foods)

Pallets per 24 h: 420
Kilograms per pallet: 1134
Mass flow: 19 845 kg/h
Temperature in: –15°C
Temperature out: –23°C
Specific heat: 1.88 kJ/(kg·K)

Product Load = (19 845)(8)(1.88)/3600 = 82.91 kW

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, m	2.4	2.4	2.4
Door Height, m	3.0	3.0	3.0
Enthalpy, h_i	33.5	101.9	22.1
Enthalpy, h_r	–4.7	–4.7	–4.7
Density, ρ_r	1.41	1.41	1.41
Density, ρ_i	1.25	1.12	1.30
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, kW	16.21	8.97	2.78
Total load, kW	48.67	8.97	2.78
		Infiltration Load, kW	60.45

$$\text{Infiltration Load} = 0.221A\,(h_i - h_r)\rho_r(1 - \rho_i/\rho_r)^{0.5}(g_c H)^{0.5}F_m[D_f D_t(1 - E_f)]$$

Infiltration (continued)

A = doorway area, m^2
h_f = enthalpy °C incoming air through doorway from adjacent area, kJ/kg
h_r = enthalpy °C room air, kJ/kg
ρ_r = density °C colder, room, air, kg/m^3
ρ_i = density °C warmer, incoming, air, kg/m^3
g_c = gravitational constant, 9.81 m/s^2
H = doorway height, m
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: 10.8 W/m^2
Floor area: 2743 m^2

Lighting Load = (10.8)(2743)/1000 = 29.62 kW

People

Number of persons: 3
Room temperature: –23ºC

People Load = (3)[272 – 6(–23)] = 1.23 kW

Trucks

Number of trucks: 3
Kilowatts per truck: 5.6

Truck Load = (3)(5.6) = 16.8 kW

Fans

Number of fans: 15
Kilowatts per fan: 1.10 nominal (1453 W heat gain per Table 4)

Fan Load = (15)(1.453) = 21.8 kW

Load Summary	Load, kW
Transmission	66.89
Product	82.91
Motor	0.00
Infiltration	60.45
Lighting	29.62
People	1.23
Trucks	16.80
Fan motors	21.80
Subtotal	279.70
Safety 10%	27.97
Total Load, kW	307.67

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.

BIBLIOGRAPHY

Ballard, R.N. 1992. Calculating refrigeration loads on an hour-by-hour basis: Part I—Building envelope and Part II—Infiltration and internal heat sources. *ASHRAE Transactions* 98(2).

Cole, R.A. 1984. Infiltration: A load calculation guide. *Proceedings of the International Institute of Ammonia Refrigeration*, 6th annual meeting, San Francisco (February).

Cole, R.A. 1987. Infiltration load calculations for refrigerated warehouses. *Heating/Piping/Air Conditioning* (April).

Cole, R.A. 1989. Refrigeration loads in a freezer due to hot gas defrost, and their associated costs. *ASHRAE Transactions* 95(2):1149-1154.

Dickerson, R.W. 1972. Computing heating and cooling rates of foods. *ASHRAE Symposium Bulletin* No-72-03.

Downing, C.C. and W.A. Meffert. 1993. Effectiveness of cold-storage door infiltration protective devices. *ASHRAE Transactions* 99(2).

Fisher, D.V. 1960. Cooling rates of apples packed in different bushel containers, stacked at different spacings in cold storage. *ASHRAE Transactions* 66.

Gosney, W.B. and H.A.L. Olama. 1975. Heat and enthalpy gains through cold room doorways. *Proceedings of the Institute of Refrigeration, 1975-76* 72:31-41.

Hamilton, J.J., D.C. Pearce, and N.B. Hutcheon. 1959. What frost action did to a cold storage plant. *ASHRAE Journal* 1(4).

Haugh, C.G., W.J. Standelman, and V.E. Sweat. 1972. Prediction of cooling/freezing times for food products. *ASHRAE Symposium Bulletin* NO-72-03.

Hendrix, W.A., D.R. Henderson, and H.Z. Jackson. 1989. Infiltration heat gains through cold storage room doorways. *ASHRAE Transactions* 95(2).

Hovanesian, J.D., H.F. Pfost, and C.W. Hall. 1960. An analysis of the necessity to insulate floors of cold storage rooms at 35°F. *ASHRAE Transactions* 66.

Jones, B.W., B.T. Beck, and J.P. Steele. 1983. Latent loads in low humidity rooms due to moisture. *ASHRAE Transactions* 89(1).

Kayan, C.F. and J.A. McCague. 1959. Transient refrigeration loads as related to energy-flow concepts. *ASHRAE Journal* 1(3).

McQuiston, F.C. and J.D. Spitler. 1994. *Cooling and Heating Load Calculation Manual*, 2nd ed. ASHRAE.

Meyer, C.S. 1964. "Inside-out" design developed for low-temperature buildings. *ASHRAE Journal* 5(4).

Pham, O.T. and D.W. Oliver. 1983. Infiltration of air into cold stores. Meat Industry Research Institute of New Zealand. Presented at IIF-IIR 16th International Congress of Refrigeration, Paris.

Pichel, W. 1966. Soil freezing below refrigerated warehouses. *ASHRAE Journal* 8(10).

Powell, R.M. 1970. Public refrigerated warehouses. *ASHRAE Journal* 12(8).