Credits

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Current seafood process changes relating to controlled freezing, transport, and storage are improving quality. Freezing and storage help to even out seasonal catch rates, and provide a more uniform supply of high quality raw material to consumers and to secondary processors. Some processors are recognizing the need to exercise greater control over their products to be shipped to international customers. Many look to high quality cold storage as a means of stabilizing temperatures and accumulating complete lots, which might then be shipped directly to buyers. The advantage is minimal repacking, transfer, and temperature fluctuation. Small coastal businesses, too, look for opportunities for added value processing and for maximum value retention. Both require quality low temperature storage.

Such opportunities point to a need for cold storage facilities that are carefully planned, constructed, and operated and that support maximum seafood quality at acceptable cost. Beyond this manual, information on cold storage planning and design is available through a variety of sources. Books include those by Dellino (1997), Hallowell (1980), and Stoecker (1998). Companies such as Krack Corp. (1992) and St. Onge, Ruff and Associates (2005) supply good design and planning information. The Food and Agriculture Organization (FAO) of the United Nations has reproduced an excellent series of reports by Graham (1977, 1984), Löndahl (1981), and Johnston et al. (1994); many of these and others are available on the FAO Web site, www.fao.org/documents/default.asp. Following a 2003 conference in Anchorage an online proceedings was produced, covering topics on the theme “A public seafood processing and cold storage facility: is it right for your community?” (see www.uaf.edu/MAP/workshops/cold-storage/index.html). The “Industrial Refrigeration Workshop” presented semi-annually in recent years by Kansas State University Continuing Education program has assembled industry experts to address key design issues for large ammonia refrigeration systems, including cold stores. And the best sources to address planning and design questions are those with the vast experience of good industry contractors, consultants, and manufacturers.

Appendix A of this manual lists a number of U.S. companies producing or supplying small cold storage facilities. In many cases the product literature supplied by these companies is a valuable source of design and planning information. Appendix B references a cold storage decision model that uses cost figures from this manual (Planning Seafood Cold Storage) and product flow-through based on historic landings in Oregon (Burden et al. 2003). It is available on the Web site of the Coastal Oregon Marine Experiment Station: marineresearch.oregonstate.edu/main_page.htm.

This manual is written for people in the planning stages of small cold storage facilities—seafood processors, port managers, and city planners, among many others. As the title implies, it is to support planning and not design. A project moving beyond the planning stage requires the tools, experience, and skills of professional engineers.

The optimum size of a cold storage facility in terms of building and operating costs, was specified by Löndahl (1981) to be in excess of 500,000 cubic feet. This implies capacities exceeding 7 or 8 million pounds. Most projects contemplated for coastal communities are smaller than that. This manual targets one-million-pound facilities and smaller, as described in several scenarios in Chapter 3. A larger facility (4 million pounds) is also included in Chapter 3, with the clarification that for projects at the upper end of this manual’s range, confidence in planning cost estimates decreases.

Note that profitability of independent facilities varies also with the percentage of room that is
filled, and with the number of “turns” or frequency of loading/unloading. A facility with infrequent product movement may require 60% occupancy to break even (D. Thomas 2003). These trade-offs can be weighed within the decision model of Burden et al. (2003).

Estimates of cost and refrigeration capacity vary not only with size and construction options, but with holding temperatures as well. In this manual we consider a range from –10°F to –40°F, with the understanding that most quality seafoods should store at temperatures no warmer than –20°F. Confidence in estimates falls as storage limits approach –40°F, a temperature often considered important for the processing of specialty, high-valued products of salmon and albacore. And although most discussion focuses on facilities to support frozen storage temperatures, some of this information might be adapted for chilled (unfrozen) storage as well.
Suppose you’ve decided to build a cold storage facility adjacent to your plant.

- What is the nature of the building site? What will it take to prepare it for construction? What will the foundation involve? How far is the site from freezers, and how much temperature increase can you tolerate between freezer and cold store? Will trucks, ships, or barges have easy access for rapid discharge of goods?

- How big will the storage be? How much product must be stored at any one time? Will a temperature-controlled loading and staging dock be required? How accessible do the products need to be, so how many aisles or racks must be included? How much will it cost to build?

- What kind of electrical (or fuel) power will be required and is currently available? Will forklift trucks be required and how will they be powered? What are the anticipated costs of power and other operating expenses?

- What products will be stored and how will they be packaged? How long will they be stored and at what temperature must each be held in the room? How many temperature levels must be provided, and so how many different rooms and controls? How much temperature fluctuation can various products tolerate, and so how uniform must the room temperature be held?

- Who will install and maintain the refrigeration? What are the pros and cons of various refrigerants and systems? What kinds of heat loads must you anticipate and report to the refrigeration contractor? Will product loaded into the room have a higher temperature than that maintained in the room? How often will doors be opened or lift trucks and workers be present?

Contemplating these and many other questions is an important first step in planning a cold storage facility. This report addresses some of the points to help processors and planners formulate ideas, plans, and questions as they deal with contractors and suppliers. The ultimate goal is to encourage well-designed cold storage facilities that will support local processing of high quality and affordable seafoods.
Chapter 1: Buildings

The initial step in a cold store plan is to consider the structure—its size, capacity, and construction, and its ability to keep out the heat.

**GENERAL DESCRIPTION**

All buildings discussed in this manual are single-story structures, following current practice. Walls, ceiling, and in some cases floors, typically consist of insulated panels similar to those in Figure 1. Outside surfaces can be galvanized steel, or otherwise coated steel or aluminum. Selection depends on needs for toughness, brightness of color, ease of cleaning, or perhaps some requirement to satisfy a local sanitation ordinance. All must enable a sufficiently tight connection to prevent moisture infiltration. Prices vary.

Panels are typically assembled on site, with the nature of structural support defining two general size categories. Smaller-sized rooms are formed by panels that fasten together to support themselves, in many cases with the aid of a latching system built into the edges of the panels. For the system shown in Figure 2, the assembler uses a large hexagonal set-screw wrench to tighten a cam-operated lock that holds panels together, generally against a gasket sealing the joint. In this way, walls, ceiling, and floor can be bound into a finished, pre-designed box such as the small walk-in model shown in Figure 3. Designers have used various schemes to transmit the holding force throughout the panel to prevent the latch from pulling out. In some cases, the latch mechanism is attached to a heavy wood or high-density foam frame. At least one manufacturer has tied latch mechanisms together with metal straps imbedded in the insulation. The quality of panels, based on uniform shape, tight connection, surface coatings, latch mechanisms, and homogeneous insulation, are

**Figure 1. One manufacturer's insulated panel design**

(courtesy Bally Refrigerated Boxes)

1. Outside metal skin. Galvanized steel, patterned aluminum, white or sand tan polyester over galvanized steel, Galvalume or stainless steel.

2. Bally wash primer for optimum foam adhesion.

3. Urethane insulation, foamed-in-place (poured, not frothed). R-value: 34 for 4" panel; 42 for 5" panel.

4. Tongues and grooves on panel edges are accurately molded urethane.

5. Patented cam-action Speed-Lok joining mechanism.

6. Heavy-gauge steel straps connect locking arms with locking pins on opposite edges of each panel.

7. Interior metal skin. Galvanized steel, patterned aluminum, white or sand tan polyester over galvanized steel, Galvalume or stainless steel.

8. Interior metal floor panel skin. Heavy-gauge galvanized steel, optional smooth aluminum or stainless steel.

9. Exterior metal floor panel skin. Usually supplied in same finish as vertical panels. Edges capped with matching metal when stainless steel, white or sand tan polyester over galvanized steel are specified for verticals.
Figure 2. A cam lock mechanism for joining panels
(courtesy Hussman Corp.)

Figure 3. A typical modular walk-in
(courtesy Ram Freezers and Coolers Mfg. Inc.)
best judged by inspection and by interviewing past customers.

The use of cam-locked panels gives a certain flexibility in re-sizing or re-shaping rooms. However, as size increases, a need emerges for a structure that provides support beyond that of the panels themselves. This is of particular importance for outside buildings subjected to high wind and snow loads. One designer suggests that this category arises as ceiling spans exceed 12 or 15 feet. Some of these buildings have framing internal to the insulated envelope, as Figure 4 shows. Others have insulation supported inside a structure frame and external walls (Figure 5).

There are available many fine companies whose major expertise and service is the design and construction of cold storage facilities. Following the planning phase of a project, this is the recommended route when entering the design and construction phases. For smaller installations such as walk-in freezers and coolers, most companies supplying components—panels, doors, monitoring instruments—can also give design recommendations based on experience with their product. Appendix A has an incomplete list of businesses that manufacture or supply components for small cold storage facilities.

**SIZING AND LAYOUT**

The focus of this manual is on single-story structures. Some cold storage units of the past have been built in the shape of a multistory cube. This keeps the surface-to-volume ratio low and so minimizes heat gain from the outside. Such a design would also maximize the amount that could be stored on a given piece of land. Two disadvantages of the multistory design were cost and a more difficult loading and unloading access (ASHRAE 1982). Given the current high quality insulations available, attempts to minimize surface area are likely not worthwhile from a heat leakage point of view. And modern use of racking and storage/retrieval systems enables single-story building heights approaching 100 feet (Stoecker 1998).

Figure 5 suggests another option in the early decision process—the loading dock. This is standard on large facilities. Although it occupies space and adds cost, a loading dock has several advantages. It serves as a staging area for sorting or storing product coming in or out. Its low temperature minimizes heat and moist air entering the cold room; separate refrigeration coils in the staging area can in fact dehumidify the air, further reducing the problem of moisture entering the cold room. In some cases, designers will add heating elements in the floor to dry it and prevent formation of ice.

Several factors influence the size of the cold-storage structure. Assume first that adequate land and financing are available. The key factors affecting building size are then the mass of product and
Building design for storage facilities must consider the time of storage (Cole 2004). And these would depend in turn on landing and freezing rates of seafood, contract storage for others, rate of turnover, access to product, and need for aisle space. Dividing walls will also affect the available building footprint for the storage facility layout. Installing multiple rooms held at different temperatures will require such divisions. So too may requirements to separate seafood products from other foods such as fruit, vegetables, or meat. This might be a state requirement, as in Alaska (E.F. Thomas 2003) or the requirement of a given customer.

The volume occupied by a product depends on packaging and product form. Graham (1984) gives “stowage rates” (product densities) for a long list of packaged and unpackaged seafoods. Table 1 provides a few examples of how this value varies. Note that the density of fish muscle might be on the order of 64 pounds per cubic foot.

The building will not be filled with product—aisle space must be available for lift trucks or carts. The more product diversity and turnover, the more accessible the stored product must be. This means a higher percentage of space is required for aisles. Cole (2003) reports that use of 45-50% of the floor space for product storage is about the best you can do. Another way to anticipate floor space uses a term called “space ratio.” This is the ratio of the cold room floor space to the total number of pallets that could be stored. Designers of large rooms anticipate a range of 10-20 ft² per pallet position, with 10 considered a starting point for planning purposes (Piho 2000).

Rack framework is often required, particularly if pallets are to be stacked more than two high (e.g., Figure 6). Spacing at the outer walls results from curbs (e.g., Figure 4), generally recommended to prevent wall damage from lift trucks and pallets. Good cold air circulation between pallets and the outer wall also requires a space of 6 to 12 inches (Young 1990). Graham (1977) recommended 4 inches of clearance between product and floor.
and 8 inches of clearance at the walls—all of these clearances designed to minimize local heat gain. One serious result of such gain is sublimation—that is, loss of surface ice, also called freezer burn—and quality loss. See Chapters 5 and 6 for more on this topic. Provision of adequate ceiling space must also accommodate structure, air handling or coil systems, lights, and in many cases sprinkler systems.

Graham’s 1984 report presents a number of cold storage layouts for different mixes of the products listed in Table 1. Two of his examples (Figure 7) represent smaller capacity rooms (220,000 pounds) and demonstrate several points. First, pallets (represented by the small rectangles) are placed against a curb away from the wall, to allow cold air circulation between the product and wall. Second, aisles are most efficient if laid out in a straight line. Widths must be sufficient to accommodate the turning radius of whatever carts or forklift trucks are to be used. Of the two examples shown in Figure 8, the “counter balance truck” tends to have wider (25-40%) turning radius requirements than the “reach truck” (Löndahl 1981). One designer suggested that the 6½ foot aisle of Figure 7a is workable only with special lift truck designs; 12 feet is a preferable width for the counterbalanced forklift (Figure 8) that would be more common in small cold stores (Cooksey 2003, Showell 1997).

A third point demonstrated in Figure 7 is that loading patterns depend entirely on the products and their requirements for access. Figure 7a shows access to each pallet. Total store density (or stowage rate) when full would be 5.3 pounds per cubic foot. The packing in the store of Figure 7b shows less access but a higher total density of 7 pounds per cubic foot. Table 2 shows density and other dimensions of interest, taken from the examples presented in Graham’s report. Maximum stowage rates in working cold stores are obviously much lower than the product densities of Table 1, and generally higher in larger stores.

### INSULATION

It is the insulation in walls, ceiling, and floor that minimizes the influx of heat. The type and thickness will influence both the cost and effectiveness of the cold store.

Two types of insulation are commonly available in panels for frozen storage: polyurethane foam and expanded polystyrene (EPS). A few panel manufacturers also list fiberglass, although this material is likely specified only for coolers with temperatures above freezing (Hallowell 1980). Fiberglass, without an effective and long-lasting moisture barrier, would tend to take up water, sag, and lose its insulating quality. Regardless of the insulating value that any material has when new, all will deteriorate, at varying rates, if not sealed to keep out water vapor. ASHRAE (The American Society of Heating, Refrigerating, and Air Conditioning Engineers) gives particular emphasis to this point: “The success or failure of an insulation envelope is due directly and entirely to the vapor barrier systems to prevent water vapor transmission into and through the insulation” (ASHRAE 1982). See the next section for more on this topic.

Polyurethane, sometimes referred to as “urethane foam” is a rigid, usually buff-colored foam that forms when a couple of chemical components are mixed together. Polyisocyanurate is closely related to the chemistry of polyurethane, but has a better resistance to combustion (Russell 1997). Both have very similar insulating and density characteristics. One of the components contributing to urethane formation is a “blowing agent,” which

<table>
<thead>
<tr>
<th>Product</th>
<th>Density (lb/ft³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>IQF fillets in polystyrene trays with stretch wrap, in cartons</td>
<td>8-11</td>
</tr>
<tr>
<td>IQF fillets in bulk catering packs, in cartons</td>
<td>15-21</td>
</tr>
<tr>
<td>Fish portions, in cartons</td>
<td>18-27</td>
</tr>
<tr>
<td>Fish portions in sauce, in cartons</td>
<td>25-27</td>
</tr>
<tr>
<td>Fillet blocks</td>
<td>37-52</td>
</tr>
<tr>
<td>Whole, gutted cod in large blocks</td>
<td>40-55</td>
</tr>
<tr>
<td>Whole, gutted cod, stowed as single fish</td>
<td>25-30</td>
</tr>
<tr>
<td>Whole, gutted halibut, in wooden boxes</td>
<td>30-35</td>
</tr>
<tr>
<td>Whole salmon, stowed loose in wooden boxes</td>
<td>33-35</td>
</tr>
<tr>
<td>Shelled shrimp in blocks</td>
<td>45-55</td>
</tr>
<tr>
<td>Breaded shrimp in consumer packs, master carton</td>
<td>25-30</td>
</tr>
</tbody>
</table>

*IQF = individually quick frozen
Source: Graham 1984
Figure 6. Static racks
(from Toole 1990)

Figure 7. Two layouts for 220,000 lb capacity
(from Graham 1984)

Figure 7a.
Access to each pallet.
Pallets stacked two high; 15½’ ceiling.

Figure 7b.
Access to five different products. Pallets stacked two high; 15½’ ceiling.
then becomes the gas occupying the cells that make up the foam. In the past, this was typically R-11 or R-12, both CFC refrigerants that are no longer legal because of their harmful environmental effects. The alternatives used these days include some of the newer “safe” refrigerants; all appear to slightly lessen the insulating value of the foam (Stoecker 1998). Whatever is used, there will be some falloff in value anyway, as the original gases are slowly replaced with air.

Although polyurethane foam is sometimes sprayed onto surfaces with a gun (as in a fish hold or existing structure), panels are more often “foamed in place” or “poured in place.” With the right technique, a predetermined amount of chemicals poured into the bottom of a vertical mold will react and expand (about 30-fold) into a uniform-density foam. The foam fills the void and adheres to the panel skin, which is typically made of coated sheet metal. Common panel thicknesses are between 2 and 8 inches, depending on the manufacturer. Urethane used for wall panels is frequently advertised as being “closed cell,” implying that water vapor is effectively (although not completely) sealed out. The sheet metal skin acts as an additional moisture barrier.

Varying the chemical components will vary the density. This in turn affects the heat-insulating properties and the compressive strength of the foam. For foam panels to be used as insulation, densities are typically around 2 pounds per cubic foot. Although polyurethane will not readily burn with a match, uncoated or unlined foam can burn violently in small enclosures (such as fish holds). Chemical additives are generally used to limit flammability and thus to meet local codes.

The second common insulation used in modular panels is expanded polystyrene (EPS), sometimes referred to as “styrene” in product literature. It is the light-weight, white foam similar to that used for picnic coolers and coffee cups. This low-density plastic material forms in a mold when high pressure–high temperature steam is applied to small polystyrene beads. Thus the blowing agent in this insulation fills the void cavities with water vapor, which is soon replaced by air. Manufacturers and the literature report that EPS has insulating properties that are fairly constant over time.

When the polystyrene material is extruded instead of molded, the result is a denser, closed-cell foam. Thus it has a better compressive strength and is much more resistant to moisture penetration. Sometimes referred to as “XPS,” it has the trade name Styrofoam when marketed by the Dow Chemical Co.

EPS is usually less dense than urethane—typically 1 pound per cubic foot, versus 2 for urethane—but can be made with densities of 1.5 pounds per cubic foot. This would have almost double the compressive strength of the lower
 density materials. Dense EPS or XPS can both be used for floors. Table 3 summarizes the important properties of these insulations. Because the insulating properties of EPS are not as good as urethane, a greater wall thickness is required to achieve the same insulating value. Greater thickness does not necessarily imply greater cost; check with vendors. EPS adheres well (or can be laminated) to metal, wood, or drywall, thus giving panels reasonable strength in construction.

### About Vapor Barriers

Regardless of the quality or thickness of insulation, there will be some temperature gradient, and therefore some small trickle of heat flowing through the wall. Figure 9 gives a typical example.

Air outside the wall holds some amount of moisture or water vapor, as measured by relative humidity (RH)—80% in this example. That is, the air at this temperature holds 80% of the vapor it could hold. (At 100% humidity, it would start to rain.) At higher temperatures, the air could hold more vapor; at lower temperatures, less. Following the temperature gradient from the outer wall surface into its center, we find temperatures decreasing. If the outside air also has been able to diffuse into the wall (with its water vapor content unchanged), the relative humidity edges up, reaching 100%, also called the dew point temperature—at about 64°F in this example. At that point, the vapor begins to condense into water. The insulating value of the wall in this water-saturated area takes a plunge. Farther into the wall, the temperature reaches the freezing point of water. Ice forms, further degrading the insulating properties. Expansion of an internal ice block forming over a long period of time will destroy the wall.

![Figure 9. The case for vapor barriers](image)

---

### Table 2. Specifications for cold store layouts

<table>
<thead>
<tr>
<th>Nominal capacity (lb)</th>
<th>Total floor area curb to curb (ft²)</th>
<th>Total room volume (ft³)</th>
<th>Approx. inside total dimension L x W x H (ft)</th>
<th>Number of pallets high</th>
<th>Approximate stowage rate (lb/ft³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>220,000</td>
<td>2,700</td>
<td>43,700</td>
<td>69 × 40½ × 15½</td>
<td>2</td>
<td>5.4</td>
</tr>
<tr>
<td>220,000</td>
<td>2,060</td>
<td>33,480</td>
<td>69 × 31 × 15½</td>
<td>2</td>
<td>7.1</td>
</tr>
<tr>
<td>1,100,000</td>
<td>11,300</td>
<td>179,070</td>
<td>171½ × 67 × 15½</td>
<td>2</td>
<td>6.4</td>
</tr>
<tr>
<td>1,100,000</td>
<td>7,290</td>
<td>115,740</td>
<td>89 × 81½ × 15½</td>
<td>2</td>
<td>10.1</td>
</tr>
<tr>
<td>1,100,000</td>
<td>6,920</td>
<td>179,770</td>
<td>148½ × 46½ × 25</td>
<td>4</td>
<td>6.6</td>
</tr>
<tr>
<td>1,100,000</td>
<td>5,790</td>
<td>150,080</td>
<td>78 × 74½ × 25</td>
<td>4</td>
<td>7.6</td>
</tr>
<tr>
<td>1,100,000</td>
<td>4,490</td>
<td>108,740</td>
<td>77½ × 59½ × 23½</td>
<td>4</td>
<td>10.9</td>
</tr>
<tr>
<td>1,100,000</td>
<td>5,520</td>
<td>177,050</td>
<td>119½ × 47½ × 31</td>
<td>5</td>
<td>6.6</td>
</tr>
<tr>
<td>1,100,000</td>
<td>4,680</td>
<td>150,200</td>
<td>76 × 63 × 31</td>
<td>5</td>
<td>7.9</td>
</tr>
<tr>
<td>1,100,000</td>
<td>3,740</td>
<td>112,040</td>
<td>64½ × 59½ × 29</td>
<td>5</td>
<td>10.9</td>
</tr>
<tr>
<td>4,400,000</td>
<td>10,730</td>
<td>319,280</td>
<td>116 × 94 × 29</td>
<td>5</td>
<td>14.2</td>
</tr>
</tbody>
</table>

Source: Graham 1984
This process keeps going because the “vapor pressure”—that is, the local partial pressure of water vapor—outside the wall is higher than the vapor pressure at lower temperatures. This difference in partial pressures continues to drive the vapor into the wall where it first condenses, then freezes. The solution? A vapor barrier on the warm side. Stoecker (1998) reports that a 12 mil minimum thickness of polyethylene is a common recommendation of designers. Most panels come with a metal covering that serves as the vapor barrier. The next issue is the joints, which need special materials and sealing procedures. An experienced designer should be engaged to properly address this very critical concern.

About R-Value

The property describing the ease with which heat conducts through a material is its thermal conductivity, \( k \). In the IP (inch-pound) system of units, \( k \) is the rate of heat flow (Btu per hr) that will pass through a unit cross-sectional area (1 square foot) of the insulation that is one unit (1 inch) thick, when a temperature difference of one unit (1 °F) exists between its back and front surfaces.

A good insulating material has a low thermal conductivity value in the neighborhood of \( k = 0.2 \). This is in contrast to dry sawdust (\( k = 0.4 \)), plywood (\( k = 0.8 \)), cement (\( k = 5.0 \)), and steel (\( k = 314 \)).

The \( R \)-value, then, is the reciprocal of \( k \) multiplied by the material thickness. All of the \( R \)-values reported in the United States result from \( k \) described in English (IP) units, and from thickness expressed in inches. While \( k \) describes how well heat will flow through a material, \( R \) describes how good an insulator it is. For a typical insulation layer 1 inch thick, the \( R \)-value is

\[
R = \frac{1}{k} (L) = \frac{1}{0.2} (1) = 5
\]

A 4-inch-thick piece of that insulation would have an \( R \)-value of \( 5 \times 4 = 20 \).

### Table 3. Insulation properties

<table>
<thead>
<tr>
<th></th>
<th>Polyurethane</th>
<th>XPS(^d)</th>
<th>EPS (^c)</th>
<th>EPS (^e)</th>
<th>Fiberglass</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal conductivity ( k )(^b)</td>
<td>0.14(^b) (0.16(^b))</td>
<td>0.19</td>
<td>0.24(^c)</td>
<td>0.22(^c)</td>
<td>0.25(^d)</td>
</tr>
<tr>
<td>R-value for 1” thickness</td>
<td>7.1 (6.5(^c))</td>
<td>5.4(^c)</td>
<td>4.17</td>
<td>4.54</td>
<td>4.0</td>
</tr>
<tr>
<td>Compressive strength (psi, at 10% deformation)</td>
<td>14.5(^f)</td>
<td>18-40</td>
<td>11.6(^g)</td>
<td>20(^g)</td>
<td>–</td>
</tr>
</tbody>
</table>

\(^a\) \( k \) expressed as (Btu-in)/(hr-ft\(^2\)-°F).
\(^b\) For 25°F aged polyurethane faced with aluminum foil; source: ASHRAE 1985.
\(^c\) Measured at 25°F; source: ASHRAE 1985.
\(^d\) Normally manufactured with binder to make semi-rigid batt; source: ASHRAE 1985.
\(^f\) Source: Russell 1997.
\(^g\) Source: Zeroloc.

\(^\) At 75°F.

XPS = extruded polystyrene; EPS = expanded polystyrene.
Buildings

is mounted on the ceiling of a 24-foot regulation tunnel ventilated with moving air. After a gas burner ignites the ceiling at one end, data on the spread of the flame along the length of the tunnel are used to calculate an FSI relative to two other materials. One is a nonflammable cement board, FSI = 0; the other is a select grade red oak, FSI = 100. The amount of smoke developed during burning, and fuel used in the gas burners, are also recorded with this test.

Specifications

Thermal conductivity values vary slightly with density and temperature. For urethane, the insulating properties will slightly deteriorate with age, as air slowly diffuses into the cells and replaces the gas originally used as a blowing agent (ASHRAE 1985). And values of all materials differ somewhat according to the source of the information. For example, various sources give $k$ values for urethane (2 pounds per cubic foot) over a range of 0.11 to 0.17. Table 3 gives some specifications for three insulating materials. Note that the total $R$-value for a 4-inch wall of urethane, according to this table, is $4 \times 7.1 = 28.6$. To achieve the same $R$-value with low-density EPS, a wall thickness of $28.6/4.17 = 7$ inches is needed. Note also that if a manufacturer reports an $R$-value of 34 for the urethane insulation in their 4-inch panel, it means that the manufacturer is assuming the thermal conductivity ($k$) to be about 0.12 (instead of 0.14 used in Table 3). Such a low value is generally expected for new insulation.

Foundations

The foundation for the cold store must meet several criteria. It must support the weight of the room or building; it must be insulated; it must provide good underground drainage, ventilation, and in many cases, heat; and it must have a load-bearing surface strong enough for the product and for the loading equipment being used.

For all but the smallest walk-ins, consult a reputable design engineer to lay out the required foundation for the room or building. The foundation can be a conventional network of concrete footings or a “floating” reinforced concrete slab. In some coastal sites, pilings may be required. The permafrost of Western Alaska may require a frame resting on pilings, or maybe a series of wooden cribs that rest on the surface (Henzler 1990). In any case, the foundation must support the weight of the building and its contents with enough safety margin to withstand local earthquake hazards.

Particularly under rooms used for frozen storage, insulation, moisture barriers, drainage, and often heat are all important. Without total drainage, water will collect under the floor and turn to ice in locations where the temperature falls below 32°F. As ice begins to accumulate, it can heave and crack the floor, potentially distorting the building, as shown in Figure 10. This problem is similar to that resulting from lack of a proper moisture barrier in walls and ceiling.

The solution to this is proper insulation in the floor (4 inches is typical), a good vapor barrier, ground water drainage, and very often some source of heat. In some areas, ambient air remains above freezing temperatures, so the “heat” could be supplied by warm air flowing under the floor. Figure 11 shows a foundation that lets air circulate under the insulated floor and ensures good drainage. A “gravity vent” means that the pipes lay at an angle (50:1 slope was recommended by one designer); the heavier cold air flows by gravity out one side, warm air flows in the other. The ventilation pipes are typically 4-6 inches in diameter, perforated to accept drainage water, and laid out on 4-6 foot centers. This system has a few drawbacks, as pointed out by one contractor. Settlement, weeds, brush,
animals, and even future construction can all plug these vent holes (St. Onge, Ruff and Associates 1995).

In Figure 11 the insulated enclosure rests on top of the concrete slab. One alternative is the pit type construction of Figure 12, which places the cold store floor at the same level as that of the floor outside. Note that in both cases, a “thermal break” minimizes the conduction heat leakage coming from the warm floor outside the room. The thermal conductivity of concrete is more than 25 times that of insulating materials.

In most coastal locations, the cold, damp atmosphere and high groundwater tables will require active heating of the foundation. The option most commonly used for very large rooms (greater than 16,000 square feet or so) is to cast a grid of plastic tubing into the sub-slab and circulate a warm glycol solution (Russell 1997). A logical source of heat is unused heat from the refrigeration system—possibly the cylinder head coolers in ammonia compressors, a heat exchanger removing discharge gas superheat, or the condenser. Smaller rooms might use electric resistance heater elements embedded under the insulated floor. Another option for smaller rooms might be to use vent pipes as described earlier in Figures 11 and 12. Fans would circulate air heated by the refrigeration condenser. In all cases, it is important to have thermocouples in the foundation. These monitor heater performance and ensure that soil temperature remains above freezing. And in most if not all cases, energy conservation experts would encourage designers to consider thicker floor insulation to minimize both the energy used to heat the floor, and the refrigeration power required to then remove it.

For very small walk-ins, a foundation heater is not usually required. But most vendors would recommend that an air space be provided between the floor and foundation, using wooden spacers. Note that with raised flooring, interior or exterior ramps are needed to enable use of carts or pallet jacks (Figure 13). Loads become hard to move and can easily tip if a wheel goes off the edge.

All insulations used in the floor—particularly urethane and EPS—are rigid enough to handle

---

**Figure 11. Gravity vent systems or forced draft vent system**

(from Russell 1997)
the total distributed loads imposed by products, lift trucks, and workers. Such stationary load-bearing capacities of 600 pounds per square foot are common. Rolling loads, i.e., under a wheel or roller, are far higher and therefore require wear surfaces over the insulation. If products are carried by hand, as in small walk-ins, a wear surface of 16 or 14 gage galvanized or stainless steel is adequate. On floors supporting hand-lift trucks, ¼ inch steel or aluminum diamond tread, or ¾ inch plywood, is often employed. Higher loads and the use of mechanical forklift trucks generally call for a floor wear surface consisting of 4 inches of reinforced concrete as shown in Figures 11 and 12. A 1½ inch coated plywood floor resting on 2-by-6s, spaced on 12 inch centers, has also been employed for this situation (Young 1990).
ACCESSORIES

Cold storage rooms, from the smallest walk-ins to the largest warehouses, consist of a foundation, an enclosure, and refrigeration machinery. There are many other pieces that make up the cold store system. We’ll call these “accessories.”

Doors

Doorways must have sufficient width and height to allow easy passage of the required forklift trucks or handcarts hauling the product. Open doors, however, create the problems depicted in Figure 14, a result of the density difference between the inside and outside air. The result is in-flowing warm and moist air, significant unwanted heat loads, temperature fluctuations, ice on the floors, frost blocking evaporator fins, and other ugly possibilities.

There are some measures that will minimize this. Formulas used by designers show that the rate of heat entering through an open door is quite sensitive to the height of the doorway—it is proportional to \((\text{height})^{1.5}\). Consequently, you should plan a doorway that is no higher than necessary—a decision that is linked to the cargo carriers to be used. Fast-acting doors that slide vertically or horizontally minimize opening time and, therefore, heat load. Vertically hung plastic strips can reduce infiltration heat through an open doorway by up to 90%, and “air curtains” that result from jets of air shooting down or across the open doorway can be as much as 80% effective (Stoecker 1998).

Most small “walk-ins” have hinged doors that often come as part of a prefabricated panel (e.g., Figure 3). These require heating elements within
the frame to prevent ice formation (Figure 15), although this is an undesirable energy cost. ASHRAE (1994) recommends that these hinged doors (presumably without additional curtains) should not be used for rooms colder than 0°F.

In all cases, doorways need to have a pattern of protective posts (Figure 16); without them, door damage from moving equipment is almost assured.

**Pressure Relief Ports**

As temperature rapidly drops in a cold room or freezer, so will the internal pressure. Such problems are more acute in small rooms than in larger warehouses. A small pressure drop can damage the room by buckling panels and opening seams. Consider that a pressure difference of just 0.1 psi in a small 14 × 20 foot walk-in exerts a ceiling load of over 2 tons. To prevent such damage, designers use formulas and experience to specify an appropriate number and size of pressure relief ports. Figure 17 shows one example for very small rooms; heater wires on all versions prevent blockage due to the formation of ice.

**Shelves and Racks**

Every planner will face a different set of needs for shelving or racking. Decisions depend on the layout of the room, the product mix and weight, how accessible different products have to be, the strength and size-uniformity of cartons and pallets, and the materials handling equipment to be used. Options include single-deep (e.g., Figure 6) and double-deep racks, with design and access depending upon the type of forklift truck (e.g., Figure 8) to be used (Cole 2004). A more detailed description of these and other racking options can be found in Showell (1997) and by contacting suppliers and contracting engineers. Many vendors of walk-ins

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**Figure 14.** Flowing cold and warm air masses that occur when typical freezer doors are open

(from ASHRAE 1985)
will also supply shelving; racks are always regarded as a separate expense.

**Lighting**

Manufacturers of small walk-ins have various guidelines on how much lighting is appropriate. For example, one recommends a 100 watt light for each 100 square feet of floor space. Designers of larger buildings have used lower power intensities, in the range of 50-80 watts per 100 square feet (Kornelias 1990, Piho 2000, Kelleher et al. 2001). All of these figures assume fluorescent lamps, not incandescent. In any case, it may be the actual required lighting (rather than power) that dictates the lamps used. For example the Alaska Department of Environmental Conservation (2002) requires that storage areas have lighting to provide “at least 20 foot-candles of light, evenly distributed.” And in all cases, some kind of shielding will be required over the light fixture to protect food packages from broken glass.

Heightened concern for energy conservation has led to a number of lighting innovations, often now standard on many projects. Motion sensors can activate bi-level lighting, dimming after some short period of inactivity. The resulting power reduction of 50% is typical (Wilcox 1999). And such
alternative fixtures as metal halide and high pressure sodium, can deliver the required illumination using lower rates of electrical energy.

**Fire Protection**

Particularly with the urethane and EPS foams used as insulation, most rooms with a floor area larger than 400 square feet will require some kind of fire protection. More than just insulation can burn—Stoecker (1998) points out the significant flammability of other cold room materials such as wooden pallets, cardboard cartons, and plastic wrap. Fire protection options include an approved thermal barrier on the surface of the walls, a ceiling-mounted sprinkler system, or both. If the room is located within another building, these measures will likely be required both inside and outside the cold room.

Although there appear to be some general guidelines on when and where sprinkler systems are required, and how a configuration might be influenced by exit locations and material selection, nothing is universal. At least two entities must be consulted in the planning stage of the project. One is the state office responsible for building codes, or in some cases the state fire marshall. They in turn will likely refer to some set of industry or international codes, as defined for example by the Uniform Building Code, the National Fire Protection Association (NFPA), or some combination of these and others. Most are periodically updated; to add to the confusion, many states will modify various parts to address local concerns. Regardless of requirements given by the state office, one must also check in with a second entity, your insurance carrier, who may require the more stringent measures.

A Sitka, Alaska, cold storage unit built in the early 1990s had sufficient thermal barriers on the walls so that no sprinkler system was required (Williams 1991). The usual practice, however, is to install dry sprinklers with melting triggers, mounted on the ceiling. Some use a dual temperature sensor, which means they won’t go off if hit (i.e., water is pumped in only if the valve opens and the sensor records a high temperature). At some maximum ceiling height (maybe 30 or 40 feet), regulations may require a second row of sprinklers to be built into the racks to ensure spray coverage at lower levels. This is good to avoid, because they are susceptible to damage; use of new high-volume heads, called “early suppression fast response” (ESFR) sprinklers can mitigate such a requirement in high-ceiling rooms (Vallort 2003).

**Miscellaneous**

Many additional accessories must be considered. Some may improve convenience, safety, and product quality. Some are necessary; some are required.
• Materials handling equipment. This has been discussed earlier. Except for the smallest walk-in stores, these represent a significant expense to purchase, operate, and maintain. Some forklift trucks are powered by compressed natural gas or liquefied petroleum gas. Many are powered by DC batteries that require charging stations in the plant. Even when operated using off-peak electrical rates, Cole (2004) estimates that such stations can represent 15-20% of the operation's total electrical power cost.

• Room temperature monitors and recorders. An accurate measure and record of temperatures may be required by a customer, the local state agency (e.g., Alaska DEC 2002), or the U.S. Food and Drug Administration (Peterson 1992).

• Cleanable surfaces on floors and walls, corner fillets on floors, possible placement of floor drains.

• Alarms that will indicate high room temperature (or any number of other refrigeration settings).

• Special floor surfacing or matting to prevent slipping.

• Heavy “kick plates” on doors in small walk-ins, including various bumper guards or curbing to minimize door and wall damage.

• Inside safety push handles or automatic openers for doors.

• Foot-operated treadles (pedals) on small walk-ins for hands-free door opening.

• Various roof kits and door awnings or drip shields (rain hoods) for outdoor installation of small walk-ins.

PERMITS AND CODES

Any building project will require permits from a number of agencies; the scope and magnitude will depend on location and other factors. The experience of one Oregon contractor found a range of permitting costs, from a few hundred dollars to 3% of the total (Ristau 2002). This section notes a few of the agencies that might be involved, using the State of Alaska as an example. Check your local requirements.

A building permit will be required that may also cover the fire protection system; a separate permit from the local or state fire marshall may be required. Regulations governing the design and construction of seafood processing facilities will have to be followed. In Alaska, this would be the state Department of Environmental Conservation (Alaska DEC 2002). Although not specific in detail, these regulations generally address the sanitation and food safety issues noted in the previous section (floor surfaces, temperature monitoring, light guards, etc.). One organization that does specify details in numbers and design specifications is the National Sanitation Foundation (now NSF International, www.nsf.org), a private nonprofit organization. Its tests and product certification creates standards used by industry and agencies. Their Standard No. 7 is entitled “Commercial Refrigerators and Freezers” (NSF International 2001) and details many of the sanitation and safety issues covered by DEC. NSF approval is in most cases sufficient to gain approval by the Alaska DEC (Soares 1991).

One who stores seafood products is defined in Alaska as a “processor” (E.F. Thomas 2003) and in Alaska will likely need to file an “Intent to Operate” form, the results of which support application to several state agencies. Permits from the state DEC and the U.S. Environmental Protection Agency (EPA) would likely be necessary if there will be any effluent from seafood processing. The ultimate result of these applications is a Seafood Processors Permit, renewed annually.

As described in a following section, refrigeration for most of the small facilities addressed in this manual uses HFC (hydrofluorocarbon) refrigerants. For larger facilities ammonia is a more effective choice, and this may call for other permits and programs. If the plant were to contain ammonia in excess of some limit (currently 10,000 pounds), additional permits are required. OSHA (Occupational Safety and Health Agency) will require an approved program (PSM, Process Safety Management) to ensure the safety of workers in the plant. EPA will require an approved program (RMP, Risk Management Program) that will ensure the safety of the public against catastrophic releases of ammonia and other chemicals. Compliance is both valuable and expensive.
Chapter 2: Refrigeration

The wide-ranging topic of refrigeration is changing as we grow more concerned with costs and efficiencies, energy utilization, and environmental issues. The intent of this manual is to aid in planning and decision-making. While most refrigeration issues are best addressed by industry engineers, designers, and contractors, reflection on a few topics may help the buyer or planner resolve questions as the design and selection proceeds.

HEAT LOADS
Designers size the refrigeration system to remove heat from many sources, typically under heavy-load (if not worst-case) conditions. Graham (1977) calculated the daily average heat load on a 35,000 cubic foot room having 2,150 square feet of floor space. Its capacity would be on the order of a half-million pounds. He estimated an average total heat load of 100,000 Btu per hour that would be distributed as shown in Table 4. Consider each of these:

Heat-flow through boundaries depends primarily on the overall temperature difference, and on insulation thickness and quality. This well-insulated example room was maintained at –22°F with an assumed outside temperature of 95°F.

The optimum insulation thickness reflects a trade-off among costs covering insulation, refrigeration machinery, and refrigeration energy. Designers can evaluate those trade-offs for each situation. Energy efficiency experts lean toward thicker insulation as electrical costs continue to climb. Graham (1984) also pointed out a minimum insulation thickness, below which outside surface temperature could fall as low as the dew point, causing condensation on the outer walls. This thickness, however, would almost always be less than the economic optimum.

Table 5 gives one designer’s insulation recommendations for a range of operating conditions. Air infiltration through open or poorly fitting doors adds a significant heat load. Although open doors counted for just 19% of the heat gain in this example building, ASHRAE (1994) reports that infiltration air in a distribution-type warehouse can account for more than half of the refrigeration load. And open doors have been shown to account for 70% of the total heat load in some older, multi-story buildings (Fleming 1976). The usual result of such high infiltration loads: the room cannot maintain its uniform design temperature, and this will affect food quality. In addition, warm air brings moisture that frosts up refrigeration coils, lowering efficiency and requiring more frequent defrosting (Figure 14).

Plastic strip curtains, air curtains, fast-acting doors, and automatic closers are important factors to reduce this load. Heat and moist air will also flow in when a door does not close tightly, due to carelessness or to ice formation. Empirical equations reported by ASHRAE (1994) show such a load to be significant.

Lights contribute the heat-equivalent of the power supplied—1,000 watts (3,412 Btu per hour) in the Table 4 example.

Table 4. Heat loads calculated for a room of 35,000 cubic feet

<table>
<thead>
<tr>
<th>Heat flow through the boundaries</th>
<th>32%</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air infiltration</td>
<td>19%</td>
</tr>
<tr>
<td>Lights</td>
<td>4%</td>
</tr>
<tr>
<td>Personnel</td>
<td>3%</td>
</tr>
<tr>
<td>New product loaded</td>
<td>33%</td>
</tr>
<tr>
<td>Fans</td>
<td>4%</td>
</tr>
<tr>
<td>Defrost</td>
<td>5%</td>
</tr>
</tbody>
</table>

Source: Graham 1977
Planning Seafood Cold Storage

Personnel working in the room generate heat that must be removed by the system. This example assumed two workers generating a heat load at rates commonly listed in handbooks such as ASHRAE (1985). Any additional heat contributed by mechanical loading equipment was not considered in Table 4.

When new product is loaded into the cold store, it will typically have a temperature higher than the intended storage temperature. This may be due to early removal from an adjacent freezer; more commonly, the product simply takes on some heat as it travels to the cold storage facility. This heat must be removed by the cold store’s refrigeration system. An ASHRAE (1982) handbook suggests that designers assume that incoming product will be 10-20°F warmer than the room. Graham used a value of 18°F to arrive at his load calculations of Table 4.

Any increasing temperature of a “frozen” product will actually indicate a partial thawing. This is because the liquid fraction of any food is a solution of salts, enzymes, and other proteins. Therefore, over the commercial range of frozen temperatures, it will never completely solidify. There are several consequences of loading elevated-temperature product into the cold room:

- The refrigeration machinery in the cold store may become overloaded, causing the room to begin to warm. Cold storage refrigeration is sized not to freeze things, but to maintain low temperatures.
- Fluctuating cold store room temperature will cause fluctuating temperature on the surface of products that are already there. This leads to sublimation, or ice transforming directly to water vapor, which causes dried-out surface patches, also called “freezer burn.” Frost deposited on the inside surfaces of packaging is another sign of fluctuating temperature.
- incompletely frozen product will finish freezing in the cold store. Given a sufficiently high entering temperature (e.g., 20-25°F), freezing times can be lengthy and this can seriously affect quality. One observation of a particularly severe example showed freezing time to be on the order of weeks. Part II of this manual covers in detail the effects on seafood quality.

Fans generate the heat equivalent of the horsepower used to drive them; three ¹⁄₃-HP fans were assumed for the room of Table 4. (1 HP is the same as 2,544 Btu per hour.) Recent work has demonstrated the ability of variable frequency drives to slow fans when full-speed is not needed and thus to diminish this component of the refrigeration load (Wilcox 1999, Kolbe et al. 2004). Although not a large part of the load in this example, it can be significant in blast freezers and chilled-storage warehouses.

Defrost heat might be applied for an hour or so each day, contributing typically 5-6% of the total heat load. Electric resistance heaters, hot refrigerant gas, and warm water are all used to melt and remove frost from coils and fins.

The heat load distribution in Table 4 represents just one scenario. If we were to vary temperature of incoming product, or to increase the rate of lift-truck activity, the load and the load distribution could be significantly altered.

---

**Table 5. Likely cold store insulation thickness required (inches)**

<table>
<thead>
<tr>
<th>Material</th>
<th>Assumed thermal conductivity</th>
<th>Ambient temp. (°F)</th>
<th>Storage temp. (°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Polystyrene</td>
<td>$k = 0.242$</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>68</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td></td>
<td>86</td>
<td>7</td>
</tr>
<tr>
<td></td>
<td></td>
<td>104</td>
<td>9</td>
</tr>
<tr>
<td>Polyurethane</td>
<td>$k = 0.16$</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>68</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>86</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td></td>
<td>104</td>
<td>6</td>
</tr>
</tbody>
</table>

*a expressed as Btu per (hour) (sq. ft.) (°F temperature difference per inch)*

Source: Graham 1984
MACHINERY OPTIONS

Designers use a heat-load analysis, as Graham has done in the previous section, to arrive at the total refrigeration requirement. The scope and details of this analysis will depend upon the stage of planning. Some rules-of-thumb might be used for a “seat-of-the-pants” estimate, one having a wide range of uncertainty. As a project approaches a more serious level, a tighter design will be needed.

Once the heat load is determined, designers can then select refrigeration machinery from equipment performance information. Consider a quick calculation from the 100,000 Btu per hour load of Graham’s example. This would actually be the load averaged over a 24-hour day, but it would have to be removed by machinery that operates about 18 hours per day. Thus refrigeration capacity would have to be 133,000 Btu per hour (24/18 \times 100,000), or 147,000 Btu per hour if a 10% safety factor is thrown in (ASHRAE 1985). This is the same as 12.2 refrigeration tons (TR).

Note that the term “refrigeration ton” is just another term for the rate of heat removal. It is equivalent to the amount of heat absorbed by 1 ton of ice melting in 24 hours. One refrigeration ton = 12,000 Btu per hour.

Various rules-of-thumb have been used for such rooms. For “intermediate rooms” up to 5,000 square feet with 15 to 20 foot ceilings, Hallowell (1980) suggested one refrigeration ton per 200 square feet of floor. He was assuming a warmer (–10°F) room temperature; his predicted capacity (10.7 TR) would be lower than what we calculated above. Thus such rules should be considered only as rough starting points. Design procedures for small rooms can be found in a number of handbooks supplied by equipment manufacturers; examples are those by Krack (1992) and Bohn (Undated).

About Refrigeration Compressors

Most of the small systems considered in this manual would employ the use of reciprocating piston compressors. It is only in the large ammonia systems that screw compressors become more efficient and cost effective.

Compressors in the smaller rooms will generally drive “dry expansion” systems in which liquid refrigerant sprays (and “expands”) into the low-pressure evaporator tubes. As the liquid collects heat from the surrounding air, it evaporates, leaving the evaporator as a “dry” gas. The cycle continues as the compressor collects and compresses this gas, which exits at a high pressure and temperature. Its heat of vaporization is then removed in the condenser, and it flows as a high pressure liquid to the evaporator, where the cycle begins again.

Some of the compressors on small systems are “hermetic” or “semi-hermetic,” while others are the open-type. Hermetics have the motor and compressor sealed in a single, welded-shell housing. Semi-hermetics have the motor and compressor in a housing that encloses the shaft that connects them. Semi-hermetics can be disassembled easily for repair; hermetics cannot. In both of these cases, the compressor turns at motor speed, and refrigerant rarely leaks from the compressor because no shaft seals are exposed. Open-type compressors have an exposed shaft with a possibly increased chance of leakage, but they have advantages. Speed can be controlled so the compressor can be slowed down; a slower speed would increase operating life and would decrease compressor capacity to match a changing load. Open compressors tend to be more efficient than the hermetics, because refrigerant gas is not being used to cool the motor. It is also easier to modify and control motors for a graduated (soft) start in those small systems where the available power (primarily high starting current) is limited. Note that compressors are almost always driven by electric motors, although other options are possible. As electricity rates increase, diesel engine drives could be considered as an alternative.

Often the refrigeration load of the cold room is just one of several—others might be due to ice machines, blast or plate freezers, or a chill-room. It is in such cases—when refrigeration systems become large and/or temperatures are very low—that engineers will consider many other options. For example:

- At low refrigerant temperatures (colder than about –20°F), two-stage refrigeration should be considered (Fenton 2000). “Two-stage” can take many forms. One is essentially two cycles in which the one removing heat from the low temperature load (the “low-stage”), will dump it off to a second higher-temperature cycle; the second “high-stage” dumps its heat load to the ambient air outside the plant.
• “Flooded systems” have pumps that send low-pressure liquid refrigerant flowing through the evaporators in the cold room, at 3 or 4 times the rate needed. An exit mixture of liquid and gas then separates downstream.

• And for large systems with varying loads and multiple compressors, the use of screw compressors will become cost effective. These expensive units are very efficient at full load, but not at partial load. So large systems will have a combination of compressors, arranged so that “recips” will handle varying loads while “screws” will run full-on or full-off.

About Refrigerants

Until recent years, the smaller cold store refrigeration systems commonly used either R-502 or R-22. These are among a series of compounds called “halocarbons” which users commonly referred to as “freons.” This term was actually a trade name of the Dupont Chemical Co., one of the big manufacturers. Ammonia (also called R-717) is a naturally occurring substance and has long been used in larger systems.

The two halocarbons (R-502 and R-22) in common use for low-temperature refrigeration both contain chlorine in their molecular structure. And both break down in the atmosphere, releasing chlorine in the upper elevations. R-502 is one of a class called chlorofluorocarbons (CFCs) that release the most chlorine. With the recognition that these released chlorine atoms create serious environmental damage, most countries (including the United States in 1995) banned the manufacture of R-502 along with all like refrigerants in the CFC category.

One group of replacements for R-502 will come from a category of compounds called “hydrofluorocarbons” (HFCs) whose molecules contain no atoms of chlorine. They are thus environmentally safe in that regard. Cox (1997), however, argues that they are still a greenhouse gas, contributing both a global warming potential, and breakdown components that cause acid rain. Two HFCs that appear to be common industrial substitutes are R-507 and R-404a; these both have different pressure-temperature characteristics; the latter is actually a blend for which evaporation and condensation do not occur at the same temperature.

The other common halocarbon, R-22, is in yet a third class of compounds, called “hydrochlorofluorocarbons” (HCFCs). They will also break down and release chlorine in the atmosphere, but to a lesser extent. The use of R-22 is being diminished worldwide, with estimates that U.S. production will cease within the next twenty years or so. However, at the present time, R-22 remains a major contender as a low-temperature refrigerant.

With all of the environmental and economic costs of these manufactured synthetic refrigerants, many question the use of them at all. Why not use a “natural” refrigerant, the most common of which is ammonia? It is cheap, has superior levels of energy efficiency, is environmentally harmless, and is easily detected when leaking. The down side is that with high operating pressures, it requires heavier (and more expensive) machinery; steel (vs. copper) piping is required. Operation and servicing require special training. It can be toxic above a certain concentration, explosive above another, and therefore safety measures are important. In summary, ammonia systems are currently cost-effective only above a certain refrigeration capacity. Given progress elsewhere in the world on the use of natural refrigerants (ammonia, carbon dioxide, natural gas, and others), look for future opportunities for ammonia in smaller systems.

About Evaporators/Condensers

An evaporator in the cold room transfers heat from room air to the refrigerant that is vaporizing inside its tubing. The rate of this transfer can be described by the equation:

\[ Q = (U) \times (A) \times (\Delta T) \]

The larger the surface area, A, the more rapid the heat transfer rate, Q. Thus the tubing is typically configured in a bundle with fins added to dramatically expand the surface area. The factor U is a heat transfer coefficient that depends strongly on the velocity of air flow. So the evaporator will have fans both to increase U, and to project the cold air out into the room. The temperature difference between the air and refrigerant is \( \Delta T \).

Maintaining a high humidity in the cold room can reduce drying of exposed product, as Part II of this manual explains. The equation shows that one could increase the room’s relative humidity by oversizing the evaporator (increase A) and operating with a corresponding decrease in \( \Delta T \). The reduction in \( \Delta T \) happens when the refrigerant side is controlled to a higher temperature. The wall tem-
temperature of the fins then goes up, and so less vapor in the air is removed as frost. And by running at higher refrigerant temperature, the refrigeration system efficiency gets better.

The condenser in the refrigeration system removes heat from the hot gas, causing it to return to high-pressure liquid. When cooled by air, these units also consist of banks of finned-tubing having air blown through by fans. So they appear similar to the evaporators described above; the rate of heat transfer can be described by the same equation. Condensers in large systems usually sit on the roof or at some location allowing clear air flow. With fluctuation of refrigeration loads and outside air conditions, capacity can be automatically changed by varying fan speed (influencing $U$ in the above equation). Most current large systems also increase $U$ by spraying water on the finned surfaces. The evaporating water increases the heat transfer rate in these units, which are known as “evaporative condensers.” It is worthwhile having adequately large condensers; one rule of thumb is that a 1% energy savings will result from each 1°F drop in condenser temperature.

Condensers can also be “shell-and-tube” heat exchangers; water flowing through a tube bundle removes heat and condenses refrigerant gas entering the surrounding shell. Sometimes the warm water would be returned to the harbor; more often it is pumped to a rooftop spray tower where it is cooled by air and evaporation.

Small refrigeration systems often have the motor-driven compressor, condenser, pumps or fans, refrigerant reservoir (the “accumulator”), and controls all mounted in a single package. This is called a “condensing unit.” It then becomes a relatively simple installation to wire it into a power source, and run the plumbing to connect it with a remote evaporator.

**CHOOSING A SYSTEM**

Performance data for refrigeration equipment will specify the horsepower needed to drive a compressor. In Graham's example heat load (above), a 40 HP motor would be required to remove the 12.2 TR (Carrier 1984). Note that the motor size will depend a lot on the temperature of refrigerant evaporating on the low side, which in turn depends on the desired room temperature. The low-side “saturated suction temperature” (SST) might be about 10°F colder than the room. As the design room temperature falls, the HP/TR ratio (horsepower per refrigeration ton) goes up—that

<table>
<thead>
<tr>
<th>Floor (ft$^2$)</th>
<th>Volume (ft$^3$)</th>
<th>Design temperature (°F)</th>
<th>Refrigeration capacity (TR)</th>
<th>Refrigeration power (HP)</th>
<th>Floor area per TR</th>
</tr>
</thead>
<tbody>
<tr>
<td>80</td>
<td>640</td>
<td>-10</td>
<td>0.8</td>
<td>2</td>
<td>100</td>
</tr>
<tr>
<td>280</td>
<td>2,250</td>
<td>-10</td>
<td>4.5</td>
<td>15</td>
<td>62</td>
</tr>
<tr>
<td>1,520</td>
<td>15,200</td>
<td>-10</td>
<td>6</td>
<td>18</td>
<td>253</td>
</tr>
<tr>
<td>2068</td>
<td>37,200</td>
<td>-10</td>
<td>10</td>
<td>35</td>
<td>207</td>
</tr>
<tr>
<td>5,328</td>
<td>133,200</td>
<td>-10</td>
<td>12</td>
<td>50</td>
<td>172</td>
</tr>
<tr>
<td>15,000</td>
<td>375,150</td>
<td>-10</td>
<td>45</td>
<td>155</td>
<td>333</td>
</tr>
<tr>
<td></td>
<td></td>
<td>-20</td>
<td>25</td>
<td>215</td>
<td>288</td>
</tr>
</tbody>
</table>

**Table 6. Refrigeration capacity and power recommendations for a range of scenarios. Authors’ approximation from results of Chapter 3 and of Kolbe and Kramer (1993)**
is, more power is required to remove the same heat load. Required motor horsepower also depends (to a lesser extent) on the high side condensing temperature, also called the “saturated discharge temperature” (SDT). In an air-cooled rooftop condensing unit, this might be on the order of 20°F warmer than the outside air temperature on a warm day. As condensing temperature rises, so does the HP/TR ratio.

Table 6 lists a number of refrigeration estimates based on the scenarios both in Chapter 3 and in a previous edition of this manual (Kolbe and Kramer 1993).

For essentially all of the scenarios and examples of Table 6, engineers would specify unitary distributed refrigeration systems (also called “split systems”) using R-22 or one of the HFC refrigerants. In split systems, a number of condensing units might be distributed on the roof of the building. These house the compressor, condenser, and controls. Refrigerant flows to nearby evaporator/blower units located typically just below, in the cold room ceiling. As the size of the system and required refrigeration capacity increase, ammonia equipment becomes more cost effective. It is also more complicated to install and to service. A separate engine room houses the compressors and refrigerant storage. Liquid refrigerant flows some distance to evaporators in the room; condensed gas flows to condensing units, typically mounted on the roof over the engine room. Cole (2004) lists some of the trade-offs of each of these systems. Various designers have used decision breakpoints to begin consideration of ammonia. One is a power requirement of 250 HP; another, room capacity of 2 million pounds. Some consider split systems; some central-engine-room systems. Experience shows a great deal of overlap (St. Onge, Ruff and Associates 1995), and in such a range, details and trade-offs of both alternatives will need to be closely evaluated.
Chapter 3: Six Cold Storage Scenarios

This chapter describes six scenarios providing examples of how size, layout, and costs might vary for different applications. Although hypothetical, they result from interviews with processors, planners, and engineers. Cost and refrigeration estimates are a compilation of results from the commissioned estimates of three engineering contractors having experience in cold storage design.

INTRODUCTION

Six scenarios represent room sizes from the smallest walk-in to a building capable of storing on the order of four million pounds of product. For most of these, estimates include three operating temperatures, –10°F, –20°F, and –40°F. These span the range—from an acceptable temperature for short-term seafood storage, to one suitable for holding very high-quality and high-valued product. The lowest design temperature of –40°F pushes the limit of many designers' experience. Ice cream cold stores might operate at temperatures below –20°F; however, experience with such colder levels is less broad, and contractors report some potential difficulties for –40°F rooms. For example thermal expansion (dimensional changes of materials with temperature), increased brittleness of common steel alloys, and behavior of lights and mechanisms can all pose unique problems. Add an additional level of uncertainty to the numbers associated with this lowest-temperature case.

Each example includes a possible layout that could accommodate the projected use. Just one door is shown in some of the layouts; however, safety codes may require more. We assume use of 4-way pallets having a standard footprint of 40 × 48 inches, as specified by the US Grocers Manufacturing Association. This dimension is also standard in Europe (M. Young 1997).

The spread among estimates is quite broad. One reason is that contractors included different things in their cost breakdown, and we have attempted to sort these out in a reasonable way. Another is that experience and the resulting calculations used to create estimates have focused on large projects. The application here is to small projects, and some rules and computer programs don’t scale down very well. For example such formulas have caused Contractor 3 to overestimate the required refrigeration horsepower, and this has inflated the estimated costs of both machinery and energy. The authors have placed a greater emphasis on the results given by Contractors 1 and 2, when drawing generalities from these tables.

A few more notes on the tables that follow:

1. “Box” includes the purchase cost of the installed building and accessories, including road and rail freight within the Lower 48 states. Also included is an A&E (architecture and engineering) allowance—often 10% is used for larger buildings. Construction of a foundation has been placed separately within the “miscellaneous” category (below).

2. “Refrigeration” includes installed refrigeration machinery and freight, including electrical panels and controls. Contractor 1 assumed halocarbon split systems for the smaller rooms; ammonia (and a central engine room) for the larger. Contractor 2 assumed halocarbon systems for all, and commented that an ammonia option would add significantly to equipment capital cost. Contractor 3 used estimating rules (in cost per square foot) that would apply to larger ammonia systems, which would also include the complexity and expense of a separate engine room.

3. “Miscellaneous” includes
   • Construction of the foundation.
   • Installed sprinkler system.
   • Racking structure, where noted.
The cost of the foundation is the major contributor, and the sprinkler system a minor one. Contractor 1 included the cost of racks; Contractors 2 and 3 did not. One estimate for installed racking systems for each scenario is given below (Cloud 2004). These are “selection racking,” giving high accessibility.

<table>
<thead>
<tr>
<th>Room dimensions (ft)</th>
<th>Racking cost ($)</th>
</tr>
</thead>
<tbody>
<tr>
<td>8 × 10, 8 high</td>
<td>60</td>
</tr>
<tr>
<td>14 × 20, 8 high</td>
<td>300</td>
</tr>
<tr>
<td>38 × 20, 10 high</td>
<td>2,000</td>
</tr>
<tr>
<td>44 × 47, 18 high</td>
<td>9,000</td>
</tr>
<tr>
<td>72 × 74, 25 high</td>
<td>35,000</td>
</tr>
<tr>
<td>122 × 123, 25 high</td>
<td>98,000</td>
</tr>
</tbody>
</table>

4. “Electrical work” denotes the service and materials of an electrical contractor to wire up panels; power and controls for motors driving compressors, pumps, and fans; lighting; door frame heaters; etc.

5. “Total + 10%” gives the total capital cost of units noted above, plus a 10% “contingency” to account for unexpected additions or overruns.

6. “Energy use” assumes that the room operates year-round; machinery runs about 75% of the time. Cost is calculated using an overall industrial rate of $0.07 per kWh. Obviously, total electric power cost will vary proportionally with both electrical rates and operating time. Contractor 1 assumed all sites to be in low-temperature regions of Alaska, resulting in lower values of kWh energy consumed.

7. “Maintenance” implies only that; it does not include any equipment replacement costs.

8. Units used:
   - TR = refrigeration ton. (1 TR = 12,000 Btu per hour).
   - BHP = brake horsepower, or horsepower requirement of the motor driving the compressor.
   - kWh = kilowatt-hour.
A. 8 × 10 WALK-IN

This scenario is adapted from Kolbe and Kramer (1993), using a cost inflation of 25% (BEA 2003). It describes an 8 × 10 walk-in cold room, built to hold about 5,000 pounds of product. The owner would add shelving along each side of the room (Figure 18); a four-foot hinged door is the single access. As presented, it would be assembled within an existing building; a minor additional cost would be required if it were constructed outside, to beef up the roof and provide suitable exterior coatings. These units are typically placed on a concrete slab or leveled gravel foundation, and raised on wood slats to allow a circulation of warm ambient air beneath the floor. Table 7 gives cost results.

Representative details

| Type of Store | Small walk-in; out-of-doors adjoining an existing building. |
| Contents      | 5,000 pounds of packaged bait, whole fish, or blocks. |

<table>
<thead>
<tr>
<th>Estimated Room Size</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Floor Space</td>
<td>80 square feet</td>
</tr>
<tr>
<td>Volume</td>
<td>640 cubic feet</td>
</tr>
</tbody>
</table>

| Layout | 10 × 8 × 8 foot ceiling; single door at one end; product loaded on shelves on each side of the door. |
| Door    | Hinged, 4-foot minimum width |
| Accessories | Door jamb heater, lights, pressure equalizer port, dial thermometer on exterior wall, exterior ramp. |
| Foundation | Existing concrete slab or leveled gravel. |
| Refrigeration | Operating temperature −10°F |
| Capacity | 0.8 TR |
| Compressor Motor | 2 BHP |

Table 7. Contractor estimates for an 8 × 10 walk-in (2003 dollars)

<table>
<thead>
<tr>
<th>Temperature</th>
<th>−10°F</th>
</tr>
</thead>
<tbody>
<tr>
<td>Contractor A</td>
<td></td>
</tr>
<tr>
<td>Contractor B</td>
<td></td>
</tr>
<tr>
<td>Capital expenses</td>
<td></td>
</tr>
<tr>
<td>Box</td>
<td>6,320</td>
</tr>
<tr>
<td>Refrigeration equipment</td>
<td>2,550</td>
</tr>
<tr>
<td>Miscellaneous: freight</td>
<td>620</td>
</tr>
<tr>
<td>Electrical work</td>
<td>1,230</td>
</tr>
<tr>
<td>Total + 10%</td>
<td>11,790</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Annual operating expenses</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Energy cost ($)</td>
<td>2,450</td>
</tr>
<tr>
<td>(kWh)</td>
<td>(35,000)</td>
</tr>
<tr>
<td>Maintenance</td>
<td>300a</td>
</tr>
<tr>
<td>Total</td>
<td>2,750</td>
</tr>
</tbody>
</table>

*aAdded by authors*

Figure 18. Layout for small walk-in
B. 14 × 20 WALK-IN

Frozen storage within a secondary processing operation. The room is designed to stand alone or to locate within an existing building. In either case, a concrete subfloor would have to be either heated or ventilated to prevent freezing and floor-heave. Outdoor construction may require the additional structure of a slanted roof and other roofing or coating materials. The layout pictured in Figure 19 implies owner-supplied shelving or racks along each side of a 4 foot door. Product would likely be hand-stacked from carts or pallet jacks. Table 8 gives estimates.

**Representative Details**

<table>
<thead>
<tr>
<th>Type of Store</th>
<th>Large walk-in; outdoor single story, adjoined to existing building.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Contents</td>
<td>20,000 pounds of processed salmon, halibut, or albacore tuna.</td>
</tr>
<tr>
<td>Estimated room size</td>
<td></td>
</tr>
<tr>
<td>Floor Space</td>
<td>280 square feet.</td>
</tr>
<tr>
<td>Volume</td>
<td>2,250 cubic feet.</td>
</tr>
<tr>
<td>Layout</td>
<td>14 × 20 × 8 foot ceiling; single door at one end; product loaded on shelves on each side of the door.</td>
</tr>
<tr>
<td>Door</td>
<td>Hinged, 4-foot minimum width.</td>
</tr>
<tr>
<td>Accessories</td>
<td>Door frame heater, lights, pressure equalizer port, dial thermometer on exterior wall.</td>
</tr>
<tr>
<td>Foundation</td>
<td>Insulated/ventilated concrete, as necessary.</td>
</tr>
</tbody>
</table>

**Figure 19. Floor plan for the 14 × 20 cold room**
### Table 8. Contractor estimates for a 14 × 20 walk-in

#### Refrigeration

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>Contractor 1</th>
<th>Contractor 2</th>
<th>Contractor 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>–10</td>
<td>–3.5</td>
<td>–21</td>
<td>–12</td>
</tr>
<tr>
<td>–20</td>
<td>–4.0</td>
<td>–21</td>
<td>–15</td>
</tr>
<tr>
<td>–40</td>
<td>–6.0</td>
<td>–25</td>
<td>–25</td>
</tr>
</tbody>
</table>

#### Cost estimate (2003 dollars)

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>Contractor 1</th>
<th>Contractor 2</th>
<th>Contractor 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Capital expenses</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Box</td>
<td>9,600</td>
<td>15,000</td>
<td>22,300</td>
</tr>
<tr>
<td>Refrigeration equipment</td>
<td>8,600</td>
<td>12,500</td>
<td>50,400</td>
</tr>
<tr>
<td>Miscellaneous: sprinklers, freight, racks, foundation</td>
<td>6,600</td>
<td>6,600</td>
<td>27,700</td>
</tr>
<tr>
<td>Electrical work</td>
<td>2,500</td>
<td>2,500</td>
<td>4,300</td>
</tr>
<tr>
<td>Total + 10%</td>
<td>30,000</td>
<td>40,200</td>
<td>115,000</td>
</tr>
</tbody>
</table>

| Annual operating expenses |              |              |              |
| Energy, $ (kWh)          | 160          | 4,200        | 7,000        |
| Maintenance              | 300          | 300          | 300          |
| Total                    | 460          | 4,500        | 7,300        |

*Data added by authors.*
C. 38 x 40 ROOM

A structure designed to hold about 200,000 pounds of fish (Figure 20). A single door and two aisles enable some sorting, separation, and staging of products. The size of the building will likely require a full foundation, under-floor heating, and a dry-pipe sprinkler system. A full racking system for support of pallets is assumed. Table 9 gives estimates.

Representative Details

<table>
<thead>
<tr>
<th>Type of Store</th>
<th>Large room; outdoor single story that is free standing or adjoined to existing building.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Contents</td>
<td>200,000 pounds of processed salmon, halibut, or albacore tuna.</td>
</tr>
<tr>
<td>Estimated room size</td>
<td></td>
</tr>
<tr>
<td>Floor Space</td>
<td>1,520 square feet.</td>
</tr>
<tr>
<td>Volume</td>
<td>15,200 cubic feet.</td>
</tr>
<tr>
<td>Layout</td>
<td>38 x 40 x 10 foot ceiling; single door at one end; product delivered by lift truck and racks.</td>
</tr>
<tr>
<td>Door</td>
<td>Sliding door, 6-foot minimum width.</td>
</tr>
<tr>
<td>Accessories</td>
<td>Door frame heater, lights, pressure equalizer ports, dial thermometer on exterior wall.</td>
</tr>
<tr>
<td>Foundation</td>
<td>Insulated/ventilated and heated concrete with perimeter and internal-post footings.</td>
</tr>
</tbody>
</table>

Figure 20. Room layout for 38 x 40 building
### Table 9. Contractor estimates for a 38 × 40 room

#### Refrigeration

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>Refrigeration load (TR)</th>
<th>Compressor motor (BHP)</th>
<th>Contractor 1</th>
<th>Contractor 2</th>
<th>Contractor 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>−10</td>
<td>−6</td>
<td>27</td>
<td>18</td>
<td>96</td>
<td></td>
</tr>
<tr>
<td>−20</td>
<td>−8</td>
<td>31</td>
<td>35</td>
<td>124</td>
<td></td>
</tr>
<tr>
<td>−40</td>
<td>−10.5</td>
<td>44</td>
<td>45</td>
<td>60</td>
<td></td>
</tr>
</tbody>
</table>

#### Cost estimate (2003 dollars)

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>−10</th>
<th>−20</th>
<th>−40</th>
</tr>
</thead>
<tbody>
<tr>
<td>Capital expenses</td>
<td>Contractor 1</td>
<td>Contractor 2</td>
<td>Contractor 3</td>
</tr>
<tr>
<td>Box</td>
<td>56,200</td>
<td>37,500</td>
<td>63,800</td>
</tr>
<tr>
<td>Refrigeration equipment</td>
<td>14,500</td>
<td>32,500</td>
<td>136,800</td>
</tr>
<tr>
<td>Miscellaneous: sprinklers, freight, racks, foundation</td>
<td>23,300</td>
<td>23,300</td>
<td>83,600</td>
</tr>
<tr>
<td>Electrical work</td>
<td>8,000</td>
<td>8,000</td>
<td>23,600</td>
</tr>
<tr>
<td>Total + 10%</td>
<td>112,200</td>
<td>111,400</td>
<td>338,600</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Annual operating expenses</th>
<th>Contractor 1</th>
<th>Contractor 2</th>
<th>Contractor 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Energy, $ (kWh)</td>
<td>1,070</td>
<td>6,020</td>
<td>19,000</td>
</tr>
<tr>
<td>Maintenance</td>
<td>600</td>
<td>600</td>
<td>600</td>
</tr>
<tr>
<td>Total</td>
<td>1,670</td>
<td>6,620</td>
<td>19,600</td>
</tr>
</tbody>
</table>

*a* Data added by authors.
**D. 44 × 47 ROOM**

A larger version of the room of Scenario C. With higher ceiling and wider aisles, this room would hold about 500,000 pounds with a low rate of turnover. Room layout appears in Figure 21; one concrete foundation design concept is in Figure 22. An alternative to the warm-air foundation heating (via drainage tiles) would be a grid of tubing carrying warm ethylene glycol. A nominal design might also include a 3 × 3 × 1 foot footing beneath each of 12 support posts in the structure. Table 10 gives estimates.

---

**Figure 21. Layout for 44 × 47 cold store**
Representative Details
Type of Store  Outdoor, free-standing single story building.
Contents  500,000 pounds of mixed seafood products.
Estimated room size
  Floor Space  2,068 square feet.
  Volume  37,224 cubic feet.
Layout  44 × 47 × 18 foot ceiling; single door at one end; product delivered by lift truck and racks.
Door  Sliding door, 6-foot minimum width.
Accessories  Door frame heater, lights, pressure equalizer ports, monitoring thermometers, sprinkler system, racking.
Foundation  Insulated/ventilated and heated concrete with twelve perimeter and internal-post footings

Table 10. Contractor estimates for a 44 × 47 room

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>Refrigeration load (TR)</th>
<th>Compressor motor (BHP)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Contractor 1</td>
<td>Contractor 2</td>
</tr>
<tr>
<td>–10</td>
<td>–10</td>
<td>–36</td>
</tr>
<tr>
<td>–20</td>
<td>–12</td>
<td>54</td>
</tr>
<tr>
<td>–40</td>
<td>–15</td>
<td>86</td>
</tr>
</tbody>
</table>

Cost estimate (2003 dollars)

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>–10</th>
<th>–20</th>
<th>–40</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1</td>
<td>2</td>
<td>3</td>
</tr>
<tr>
<td>Capital expenses</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Box</td>
<td>84,000</td>
<td>56,000</td>
<td>91,600</td>
</tr>
<tr>
<td>Refrigeration equipment</td>
<td>19,700</td>
<td>53,000</td>
<td>186,000</td>
</tr>
<tr>
<td>Miscellaneous: sprinklers, freight, racks, foundation</td>
<td>49,600</td>
<td>49,600</td>
<td>103,500</td>
</tr>
<tr>
<td>Electrical work</td>
<td>10,500</td>
<td>10,500</td>
<td>32,000</td>
</tr>
<tr>
<td>Total + 10%</td>
<td>180,200</td>
<td>186,000</td>
<td>454,400</td>
</tr>
</tbody>
</table>

Annual operating expenses

<table>
<thead>
<tr>
<th>Energy, $ (kWh)</th>
<th>2,600</th>
<th>10,920</th>
<th>26,000</th>
<th>3,130</th>
<th>15,680</th>
<th>34,000</th>
<th>3,900</th>
<th>22,050</th>
<th>45,000</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maintenance</td>
<td>1,500</td>
<td>1,500</td>
<td>1,500</td>
<td>1,650</td>
<td>1,650</td>
<td>1,650</td>
<td>1,950</td>
<td>1,950</td>
<td>1,950</td>
</tr>
<tr>
<td>Total</td>
<td>4,100</td>
<td>12,420</td>
<td>27,500</td>
<td>4,800</td>
<td>17,330</td>
<td>35,650</td>
<td>5,900</td>
<td>24,000</td>
<td>46,950</td>
</tr>
</tbody>
</table>

*Data added by authors.
E. 72 x 74 ROOM

With a ceiling height of 25 feet, this room could be easily configured for a 1 million pound capacity. The layout of Figure 23 allows access to a variety of products on a rack system supporting 4-high pallet loading. Two sliding doors are included in the box design and estimate.

The foundation is constructed as a double insulated slab as shown in Figure 22. In this case, 30 columns and footings are included to support an internal steel frame similar to that shown in Figure 4. Table 11 gives estimates.

Figure 23. Layout plan for 72 x 74 cold store
**Representative Details**

Type of Store  Outdoor, free-standing single story building.
Contents  1,000,000 pounds of mixed seafood products.
Estimated room size

| Floor Space | 5,328 square feet. |
| Volume | 133,200 cubic feet. |

Layout  72 x 74 x 25 foot ceiling; two doors at one end; product delivered by forklift truck.
Door  Sliding doors, 8-foot minimum width.
Accessories  Door frame heater, lights, pressure equalizer ports, monitoring thermometers, sprinkler system, racking.
Foundation  Insulated/ventilated and heated concrete with thirty perimeter and internal-post footings.

**Table 11. Contractor estimates for a 72 x 74 room**

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>Refrigeration load (TR)</th>
<th>Compressor motor (BHP)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Contractor</td>
<td>1</td>
</tr>
<tr>
<td>–10</td>
<td>–</td>
<td>20</td>
</tr>
<tr>
<td>–20</td>
<td>–</td>
<td>22</td>
</tr>
<tr>
<td>–40</td>
<td>–</td>
<td>26</td>
</tr>
</tbody>
</table>

**Cost estimate (2003 dollars)**

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>–10</th>
<th>–20</th>
<th>–40</th>
</tr>
</thead>
<tbody>
<tr>
<td>Capital expenses</td>
<td>Contractor</td>
<td>Contractor</td>
<td>Contractor</td>
</tr>
<tr>
<td>Box</td>
<td>348,000</td>
<td>135,000</td>
<td>196,500</td>
</tr>
<tr>
<td>Refrigeration equipment</td>
<td>75,700</td>
<td>105,000</td>
<td>479,500</td>
</tr>
<tr>
<td>Miscellaneous: sprinklers, freight, racks, foundation</td>
<td>147,900</td>
<td>147,900</td>
<td>212,200</td>
</tr>
<tr>
<td>Electrical work</td>
<td>22,500</td>
<td>22,500</td>
<td>82,600</td>
</tr>
<tr>
<td>Total + 10%</td>
<td>653,500</td>
<td>451,400</td>
<td>1,070,000</td>
</tr>
</tbody>
</table>

**Annual operating expenses**

<table>
<thead>
<tr>
<th>Energy, $ (kWh)</th>
<th>Contractor</th>
</tr>
</thead>
<tbody>
<tr>
<td>(133,200)</td>
<td>(270,000)</td>
</tr>
<tr>
<td>(971,000)</td>
<td>(160,000)</td>
</tr>
<tr>
<td>(430,000)</td>
<td>(1,243,000)</td>
</tr>
<tr>
<td>(199,800)</td>
<td>(550,000)</td>
</tr>
<tr>
<td>(1,657,000)</td>
<td>(7,020)</td>
</tr>
<tr>
<td>Maintenance</td>
<td>5,400</td>
</tr>
<tr>
<td>Total</td>
<td>14,700</td>
</tr>
</tbody>
</table>

*Data added by authors.*
F. 110 × 136 ROOM

This largest of the hypothetical plans falls somewhat beyond the scope of this manual. With a capacity approaching 4 million pounds comes increasing uncertainty in costs. This coastal facility would store and process a variety of seafood products. To maintain storage capacities in off seasons, other products might fill the voids—examples being cranberries or other fruit products, meat, and possibly ice cream if design temperature were −20°F or lower. Figure 24 gives one layout presented by Graham (1984). Although giving possible access to six or eight different products, the dense packing with few aisles suggests a facility with a low turnover rate. Table 12 gives estimates.

**Representative Details**

<table>
<thead>
<tr>
<th>Type of Store</th>
<th>Outdoor, free-standing single story building.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Contents</td>
<td>4,000,000 pounds of mixed seafood products, cranberries, meat, chicken, and other foods.</td>
</tr>
<tr>
<td>Estimated room size</td>
<td></td>
</tr>
<tr>
<td>Floor Space</td>
<td>15,000 square feet.</td>
</tr>
<tr>
<td>Volume</td>
<td>375,150 cubic feet.</td>
</tr>
<tr>
<td>Layout</td>
<td>110 × 136 × 25 foot ceiling; two doors at each end; product delivered by forklift truck.</td>
</tr>
<tr>
<td>Door</td>
<td>Sliding doors, 8-foot minimum width.</td>
</tr>
<tr>
<td>Accessories</td>
<td>Door frame heater, lights, pressure equalizer ports, monitoring thermometers, sprinkler system, racking.</td>
</tr>
<tr>
<td>Foundation</td>
<td>Insulated/ventilated or heated concrete with perimeter and internal-post footings.</td>
</tr>
</tbody>
</table>

**Figure 24. A four-million pound layout with low turnover**
### Table 12. Contractor estimates of a 110 × 136 room

#### Refrigeration

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>Refrigeration load (TR)</th>
<th>Compressor motor (BHP)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Contractor 1</td>
<td>Contractor 2</td>
</tr>
<tr>
<td>−10</td>
<td>–</td>
<td>38</td>
</tr>
<tr>
<td>−20</td>
<td>–</td>
<td>42</td>
</tr>
<tr>
<td>−40</td>
<td>–</td>
<td>48</td>
</tr>
</tbody>
</table>

#### Cost estimate (2003 dollars)

<table>
<thead>
<tr>
<th>Temperature (°F)</th>
<th>Contractor 1</th>
<th>Contractor 2</th>
<th>Contractor 1</th>
<th>Contractor 2</th>
<th>Contractor 1</th>
<th>Contractor 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Capital expenses</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Box</td>
<td>750,000</td>
<td>265,000</td>
<td>750,000</td>
<td>265,000</td>
<td>750,000</td>
<td>302,000</td>
</tr>
<tr>
<td>Refrigeration equipment</td>
<td>86,000</td>
<td>180,000</td>
<td>107,700</td>
<td>230,000</td>
<td>261,000</td>
<td>318,000</td>
</tr>
<tr>
<td>Miscellaneous: sprinklers, freight, racks, foundation</td>
<td>315,000</td>
<td>315,000</td>
<td>346,500</td>
<td>346,500</td>
<td>409,500</td>
<td>409,500</td>
</tr>
<tr>
<td>Electrical work</td>
<td>37,500 (^a)</td>
<td>37,500 (^a)</td>
<td>43,500 (^a)</td>
<td>43,500 (^a)</td>
<td>55,000 (^a)</td>
<td>55,000 (^a)</td>
</tr>
<tr>
<td>Total + 10%</td>
<td>1,307,400</td>
<td>877,300</td>
<td>1,372,500</td>
<td>973,500</td>
<td>1,623,000</td>
<td>1,193,000</td>
</tr>
</tbody>
</table>

#### Annual operating expenses

<table>
<thead>
<tr>
<th></th>
<th>Contractor 1</th>
<th>Contractor 2</th>
<th>Contractor 1</th>
<th>Contractor 2</th>
<th>Contractor 1</th>
<th>Contractor 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electricity cost ($)</td>
<td>26,260</td>
<td>42,000</td>
<td>31,510</td>
<td>55,300</td>
<td>39,400</td>
<td>70,000</td>
</tr>
<tr>
<td>(kWh)</td>
<td>(375,200)</td>
<td>(600,000)</td>
<td>(450,200)</td>
<td>(790,000)</td>
<td>(562,700)</td>
<td>(1,000,000)</td>
</tr>
<tr>
<td>Maintenance</td>
<td>15,200</td>
<td>15,200 (^a)</td>
<td>16,720</td>
<td>16,720 (^a)</td>
<td>19,760</td>
<td>19,760 (^a)</td>
</tr>
<tr>
<td>Total</td>
<td>41,500</td>
<td>57,200</td>
<td>48,230</td>
<td>72,020</td>
<td>59,160</td>
<td>89,760</td>
</tr>
</tbody>
</table>

\(^a\) Data added by authors.
MORE FACTORS AFFECTING COSTS

The above estimates cover the basics—the main pieces making up a cold storage project, and the energy. They attempt to give some “ballpark” figures for representative projects in temperate West Coast locations, i.e., northern California through Southeast Alaska. As building size increases, as design temperatures decrease, and as the given design criteria become more “hypothetical,” the uncertainty of estimates increases. The curves shown in Figure 25 indicate how widely the estimates can vary with project definition. As the amount of information we have to define the project grows (shown by the dotted line), the uncertainty of the estimate, marked by the “Low End” and “High End” curves, diminishes. The scenarios described above are based on estimates that might fall somewhere between the “Concept Screening” category of Class 5, and the “Feasibility” category of Class 4.

As indicated in the graph, much of the uncertainty relates to how well we can define the project and to the factors and components that have been omitted. The following sections describe various factors that will influence the project’s cost and overall uncertainty.

Extras

Although the above scenarios compare units at similar locations, many site-specific costs must be addressed by local engineers and designers. Some things were excluded from the preceding construction-cost tables:

- **Land purchase.**
- **Site preparation**—clearing, leveling, excavating, filling, drainage.
- **Parking lot**—drainage, paving, exterior lighting, landscaping.
- **Support facilities**—offices, restrooms, worker rooms (lockers, lunch), engine and pumping rooms.
- **Permits and fees**—building permits, sales taxes, environmental impact studies and fees.
- **Utility sources**—electrical lines, switching equipment, and transformers to connect with a remote source; water and sewer lines.
- **Backup electrical power generators.**
- **Loading docks and controlled temperature alcoves.**
- **Materials handling equipment** such as forklift trucks, pallets, automatic loading/racking systems.
- **Battery charging equipment** for electric forklift trucks.

---

**Figure 25. Project Definition vs. Estimate Accuracy**

![Graph showing project definition vs. estimate accuracy](image)

Association for the Advancement of Cost Engineering Industrial Design and Construction Inc. Courtesy of Stuart Stevens.
Many other “extras” were also omitted in the category of “annual operating expenses.” These will make up a far larger piece of the total than the electric energy and minor maintenance shown. Cole (2004) reported that electrical energy to power refrigeration makes up 70-75%; the rest goes to lighting and battery charging for lift trucks. And several contractors project total operating costs to be 5-10 times the cost of electrical power. Other operating expenses include the following.

- Labor.
- Depreciation or replacement costs of facilities and equipment.
- Electrical energy or natural gas to power forklift trucks.
- Maintenance of offices and grounds.
- Taxes.
- Other utilities (water, sewer).

**Refrigeration and Capacity**

Contractors will generally select refrigeration to be on the conservative size, i.e., adequately large. Although we regard cold stores to be for storage of already-frozen products, in reality new products brought in will be a bit warmer than the storage temperature. The heat withdrawn while cooling off these products will add to the load on the refrigeration machinery. A difference of 10-15°F is often assumed by designers. If the difference is greater, refrigeration costs increase; frequent door openings will also influence cost (and temperature fluctuation).

For smaller systems, refrigeration would likely use freon (or HFC) equipment in units mounted on the roof or adjacent wall. For larger systems, particularly those at temperatures as low as –40°, ammonia may be a more likely choice. Higher costs of ammonia systems reflect the higher cost of machinery, use of multistage equipment, need for a separate engine room, safety programs, and personnel.

**Multiple Temperature Options**

The above scenarios and estimates assume free-standing buildings controlled to a single temperature. Often however, a facility may have multiple spaces held at different temperatures. A –40° room for example would likely be housed within a larger and warmer cold store. Common costs and multilevel refrigeration systems will affect actual facility construction and operating costs.

**Location**

For the scenarios defined, we asked contractors to assume representative conditions on the coasts of the Pacific Northwest. The variation of some of those conditions, such as location shifts between Northern California and Western Alaska, can affect the plans and estimates.

**Ambient Temperatures**

Refrigeration capacity and insulation thickness depend upon expected maximum ambient temperatures. Heat leakage load is proportional to the difference between ambient and design room temperatures. Air-cooled condenser performance, too, is dependent upon ambient conditions. Table 13 shows how maximum temperatures vary—probably not enough to make a lot of difference in the refrigeration machinery design size.

Minimum temperature may have an important influence on operation of equipment. An example is the use of evaporative condensers or cooling towers which may freeze if not monitored or controlled. Table 13 gives values of expected minimums along the Pacific coast.

The expected maximum and minimum temperatures are averages. Engineering handbooks and other data give more details on the probability of extreme highs and lows, as well as other factors such as wind velocities and humidity, that will affect decisions at a specific site.
The expected maximum temperatures listed for each location tend to dictate how large the refrigeration equipment needs to be. But it is the number of these warm days per year that will influence the amount of energy needed annually to operate the facility. Engineers sometimes use a “bin method” to anticipate this figure. Climate tables (bin data) for a given location predict the distribution of temperatures throughout the year. For each temperature increment (bin) of 5°F, the tables show how many hours the ambient temperature has typically been in that range in that period of time. By adding these all up over any time period of interest, the refrigeration energy can be accurately predicted.

### Table 13. Influence of location on energy use and costs

<table>
<thead>
<tr>
<th>Location</th>
<th>1% maximum ambient temperature (°F)(^a)</th>
<th>99% Minimum ambient temperature (°F)(^d)</th>
<th>Approx. 2004 electricity cost (cents/kWh)(^a)</th>
</tr>
</thead>
<tbody>
<tr>
<td>San Francisco, CA</td>
<td>79 db; 61 wb(^c)</td>
<td>39</td>
<td>9</td>
</tr>
<tr>
<td>Eureka, CA</td>
<td>68 db; 59 wb</td>
<td>32</td>
<td>9</td>
</tr>
<tr>
<td>Coos Bay, OR</td>
<td>70 db; 61 wb</td>
<td>32</td>
<td>5</td>
</tr>
<tr>
<td>Warranton, OR</td>
<td>72 db; 63 wb</td>
<td>29</td>
<td>5</td>
</tr>
<tr>
<td>Seattle, WA</td>
<td>81 db; 64 wb</td>
<td>28</td>
<td>5</td>
</tr>
<tr>
<td>Bellingham, WA</td>
<td>77 db; 64 wb</td>
<td>21</td>
<td>7</td>
</tr>
<tr>
<td>Sitka, AK</td>
<td>64 db; 57 wb</td>
<td>21</td>
<td>9</td>
</tr>
<tr>
<td>Homer, AK</td>
<td>63 db; 55 wb</td>
<td>4</td>
<td>5</td>
</tr>
<tr>
<td>Kodiak, AK</td>
<td>64 db; 55 wb</td>
<td>11</td>
<td>15</td>
</tr>
<tr>
<td>Bethel, AK</td>
<td>68 db; 57 wb</td>
<td>–24</td>
<td>23</td>
</tr>
</tbody>
</table>

\(^a\)From ASHRAE (1997). This gives the ambient temperature maximum that is not exceeded more than 1% of the time, or 88 hours out of the year.

\(^b\)db is “dry bulb” temperature, measured by a standard thermometer. It influences the rate of heat leaking into the room.

\(^c\)wb is “wet bulb” temperature, a unit of temperature that takes into account the relative humidity. This temperature will influence the performance of an evaporative refrigeration condenser.

\(^d\)From ASHRAE (1997). Ambient temperature will not be colder than this minimum more than 1% of the time, or 88 hours out of the year.

\(^e\)The figures do not include base, demand, and other service charges; these would increase the average per-kWh cost. (e.g., Seattle costs goes from 5 to 6; Warranton from 5 to 8; Homer from 5 to 10.)

**Construction Costs**

The local cost of material, labor, and shipping will all influence the estimates to some minor degree. Means (1998) gives relative industrial costs of materials and installation as a multiplier of the national average:

- California
  - San Francisco: 1.24
  - Eureka: 1.11
- Oregon
  - Eugene: 1.05
  - Portland: 1.07
- Washington
  - Vancouver: 1.04
  - Tacoma: 1.05
  - Seattle: 1.05
- Alaska
  - Ketchikan: 1.30
  - Juneau: 1.25
  - Anchorage: 1.26

The expected maximum temperatures listed for each location tend to dictate how large the refrigeration equipment needs to be. But it is the number of these warm days per year that will influence the amount of energy needed annually to operate the facility. Engineers sometimes use a “bin method” to anticipate this figure. Climate tables (bin data) for a given location predict the distribution of temperatures throughout the year. For each temperature increment (bin) of 5°F, the tables show how many hours the ambient temperature has typically been in that range in that period of time. By adding these all up over any time period of interest, the refrigeration energy can be accurately predicted.

### Electrical Energy Cost

The operational costs given in the scenarios are primarily those due to electrical energy. A value of $0.07 per kWh was used, but Table 13 notes some locational differences that will directly influence the operational cost.
All materials will be delivered from vendors by truck or rail except in Alaska, where shipping or barging will add an additional cost. Past estimates (from 1991), corrected for inflation of 25% (BEA 2003), appear below for barging equipment from Seattle to Alaska (Kolbe and Kramer 1993).

<table>
<thead>
<tr>
<th>Room and machinery shipped</th>
<th>Destination</th>
<th>Barging cost (2003 dollars)</th>
</tr>
</thead>
<tbody>
<tr>
<td>8 × 10</td>
<td>Wrangell</td>
<td>$900</td>
</tr>
<tr>
<td>14 × 20</td>
<td>Sitka</td>
<td>$1,800</td>
</tr>
<tr>
<td>38 × 40</td>
<td>Bethel</td>
<td>$13,400</td>
</tr>
<tr>
<td>47 × 44</td>
<td>Kenai</td>
<td>$24,000</td>
</tr>
<tr>
<td>72 × 74</td>
<td>Kodiak</td>
<td>$27,400</td>
</tr>
</tbody>
</table>

Foundations
We would expect little influence of location on foundation design and cost except in areas of high groundwater, soft soils, or permafrost. A small building on permafrost may require a foundation consisting of a trestle or frame that would “float” the structure on the ground surface. A coastal site may require pilings; for large buildings, two contractors estimated that piling would add $15 per square foot to construction costs.

Snow and Wind Loads
For purposes of estimation, various handbooks such as the Midwest Plan Service handbook (1987) give maximum design snow loads and wind speeds expected in various locations. Although some Alaska locations could have exceptionally high snow loads and therefore a need for additional structure, in fact most buildings would be designed for some wind and snow load. Design and cost differences in much of Alaska would tend to have a minor effect on these estimates.

Earthquake
Requirements in some areas of California and Alaska are to design for a higher earthquake potential. Such requirements will likely have added engineering and construction costs as a result.

---

A RECENT PROJECT
When all the components are added together, the total capital cost will greatly exceed the estimates of the basics given in the scenarios. For the larger room considered above, the cost could double.

As an example, Table 14 describes one firm’s capital and construction cost estimate for a 5 million pound, 12,000 square foot refrigerated warehouse. This was projected for 1999 construction in Juneau, but was not built. The table gives estimated costs for the cold store plus several of the “more factors” described in previous sections. This example would have an overall unit construction cost of $0.64 per pound of storage capacity.

**Table 14. Cost estimate for a 5 million pound refrigerated warehouse in Juneau**

<table>
<thead>
<tr>
<th>Structure</th>
<th>Cost ($)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Office space, 1,600 ft²</td>
<td>$237,800</td>
</tr>
<tr>
<td>Refrigerated dock, 2,400 ft²</td>
<td>$356,700</td>
</tr>
<tr>
<td>Frozen storage, 12,000 ft²</td>
<td>$1,783,700</td>
</tr>
<tr>
<td>Insulated warehouse doors</td>
<td></td>
</tr>
<tr>
<td>2 storage doors</td>
<td>$14,300</td>
</tr>
<tr>
<td>6 truck doors</td>
<td>$26,000</td>
</tr>
<tr>
<td>1 man door</td>
<td>$1,600</td>
</tr>
<tr>
<td>Refrigeration equipment</td>
<td>$70,000</td>
</tr>
<tr>
<td>2 forklift trucks</td>
<td>$100,000</td>
</tr>
<tr>
<td>500 totes</td>
<td>$3,000</td>
</tr>
<tr>
<td>2,000 pallets</td>
<td>$80,500</td>
</tr>
<tr>
<td>Office equipment</td>
<td>$16,000</td>
</tr>
<tr>
<td>2.2 acres of land</td>
<td>$272,800</td>
</tr>
<tr>
<td>Sewage and utilities</td>
<td>$50,000</td>
</tr>
<tr>
<td>Parking lot, 31,200 ft²</td>
<td>$35,000</td>
</tr>
<tr>
<td>Contingency, 5%</td>
<td>$148,200</td>
</tr>
<tr>
<td>Total Estimate</td>
<td>$3,195,500</td>
</tr>
</tbody>
</table>

---

*Six Cold Storage Scenarios*
Chapter 4: Some Rules of Thumb

INTRODUCTION
Let us state up-front that none of the contractors estimating the scenarios of Chapter 3 were comfortable with suggesting “rules of thumb.” That is, no project is straightforward enough for the inexperienced planner to estimate its cost or operating expenses using simple rules or equations. As shown in the scenarios, even the professionals can present a wide range of refrigeration sizes and construction costs to describe the same basic project.

Nevertheless, this is a manual for “thinking and planning”; not for “design.” Results of our scenarios define various size and cost patterns that are interesting and that have early planning value, if the uncertainties and excluded parts of the project are kept firmly in mind.

The following sections list a number of “rules of thumb” or summary curves and tables. Some result from interpreting results of our scenarios; some from the literature; some from comments of the professionals. The manner in which these have been evaluated, filtered, and presented was a decision of the authors.

SIZING
Room
Storage capacity will be 6-8 pounds per cubic foot for small rooms; 10-12 pounds per cubic foot for larger rooms.

Comment: These numbers are from Table 2. Packaging, racks, air circulation, aisles, and requirements for product access will all affect the pounds of product that can be stored in the space.

Floor space used to store product will be 45-50% of the total.

Comment: This is about the best you can do, according to one contractor.

Lighting
Assume 60-100 watts per 100 square feet.

Comment: Assumes fluorescent lights.

Refrigeration
80-100 square feet per TR for small walk-ins.
200-250 square feet per TR for intermediate sized rooms.
300 square feet per TR for large rooms (4 million pound capacity).

Comment: TR is “refrigeration ton.” Results from scenarios and Table 8, Krack (1992) and others. Consider –10 to –20°F storage temperatures. And note that heat loads can vary a lot, as discussed in Chapter 2.

Costs—Capital
Overall refrigerated warehouse
$0.65 per pound of total capacity.

Comment: This figure results from some recent project estimates for buildings on the order of 5 million pounds. Other figures heard: $0.75 per pound for large constructed warehouses; and “a buck a pound” noted for a large complicated facility in a current planning stage.

Building and refrigeration only
$140-$180 per square foot for small walk-ins.
$90-$120 per square foot for mid-sized rooms.
$70-$90 per square foot for rooms on the order of 4 million pound capacity.

Comment: From scenario results and other contractors.
**Building only**

See Figure 26.

Comment: This does not include the cost of foundation, racking, sprinkler system.

**Figure 26. Predicted 2003 cost for constructed envelope**

(Source: Burden et al. 2003)

![Graph showing predicted 2003 cost for constructed envelope](image)

**Refrigeration Machinery**

See Figure 27.

Comment: Based on estimates for buildings smaller than 4 million pound capacity.

**Figure 27. Predicted 2003 cost of installed refrigeration**

(Source: Burden et al. 2003)

![Graph showing predicted 2003 cost of installed refrigeration](image)
**Foundation, Racks, Sprinkler System**

See Figure 28.

Comment: Based on estimations for buildings smaller than 4 million pound capacity.

**Figure 28. Predicted 2003 cost of installed foundation, racks, sprinkler system**

(Source: Burden et al. 2003)

<table>
<thead>
<tr>
<th>Cost/Cubic Foot</th>
<th>0.50</th>
<th>1.00</th>
<th>1.50</th>
<th>2.00</th>
<th>2.50</th>
<th>3.00</th>
<th>3.50</th>
<th>4.00</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cubic Feet</td>
<td>0</td>
<td>50,000</td>
<td>100,000</td>
<td>150,000</td>
<td>200,000</td>
<td>250,000</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**COSTS—OPERATING**

**Power**

See Figure 29.

$200 per HP per year, a low but typical figure.

$375 per HP per year, a high value.

Comments: The above figures given by various contractors assume a rough electrical cost of 7 cents per kWh. Figures would be proportional to electricity cost.

Comments: The curves result from the scenarios and include primarily power to operate refrigeration machinery. Cost would result from multiplying kWh by local electric rates in cost per kWh.

**Total Operating Cost**

This will be 5 to 10 times the cost of power.

Comment: From various contractors; depends on price of electricity.

**Figure 29. Estimated consumption of electric energy**

(Source: Burden et al. 2003)
Part II Storing Frozen Products—A Look at Seafood Temperature and Shelf Life

When chilled storage of seafood does not allow enough holding time, the best way to preserve quality is to freeze it and hold it in frozen storage. Chilled storage provides at best a few weeks of shelf life, whereas frozen storage provides a few months to more than a year for fishery products that have good storage characteristics.

Frozen storage has several important advantages:

- Fishing vessels that freeze at sea can harvest on grounds farther from the port where they will be landing the catch, and they can fish longer when necessary to get a large enough catch to make the trip profitable.
- The large increase in storage life allows processors and wholesalers to hold inventories for a longer time.
- The increased storage life allows products to be shipped greater distances, with the potential of opening up new markets at a seasonally optimum price.
- Frozen storage allows year-round marketing for species such as salmon, shrimp, and crab, which are harvested over a short time each year.
- If freezing is carried out soon after harvest, a higher quality product results than for fish kept in chilled storage until marketed.
- Building internal freezing and cold storage facilities in developing countries provides export opportunities for a wider range of fishery products. The same is true for Alaska, which is a long distance from the markets for its seafood products.

Chapter 5: The Process of Quality Loss

During frozen storage, food is held at a temperature where bacterial changes are stopped and enzyme action is greatly retarded. Freezing and frozen storage conditions are much more critical for seafood than for other foods. Although freezing conditions are not discussed in this book, it is certainly important to use proper freezing procedures, especially to avoid slow freezing rates. A freezing rate at which the product is frozen in 8 hours or less is satisfactory. However, most deteriorative changes occur during frozen storage rather than during freezing, and these changes are the subject of this chapter.

In today’s world market, competition from farmed fish and shellfish as well as competition from other protein foods (in particular poultry products) make it necessary for frozen fishery products to be of the highest quality, especially frozen salmon and frozen shrimp. Storing seafood below 15°F stops the growth and multiplication of bacteria. Enzyme action, however, is not slowed sufficiently to make this a good temperature for storing frozen seafood. For very short storage periods, 0°F or lower is needed. For longer storage times or for fishery products that do not have very good storage characteristics, a holding temperature below –20°F is necessary.

PROTEIN DENATURATION

One of the two major losses of quality in fish muscle during frozen storage is the development of a toughness and dryness that is related to protein denaturation (Sikorski et al. 1976). It is the major cause of quality loss during frozen storage of lean fish. Taste panels describe the cooked fish as tough, fibrous, chewy, and rubbery. The cause of the toughness is protein denaturation, or unfolding, and later aggregation accompanied by loss of water-holding capacity. This results in high thaw drip loss and cook drip loss. Ice crystallization contributes to denaturation of the protein by disrupting the water structure around hydrophobic areas of the protein and by breaking up the water-mediated hydrophobic-hydrophilic interactions (Sikorski...
and Kolakowska 1990). Maintaining this water structure is important to retaining the texture and juiciness of the cooked fish. It is also important in retaining compounds that provide good flavor.

Protein denaturation involves the myofibrillar, or muscle protein, made up of components called the actin-myosin-actomyosin system, tropomyosin, and whole myofibrils. Denaturation consists of an alteration of the secondary and tertiary protein structure, which unravels as protein bonds are broken. Subsequently, there is a cross-linking and some reformation of bonds, leading to protein aggregation.

During freezing and frozen storage of fish, adverse effects leading to protein denaturation and aggregation are caused by several factors. These include the concentration of inorganic salts, free fatty acid formation, reaction with the products of lipid oxidation, and formation of formaldehyde in some members of the family Gadidae (for example cod and pollock) by the enzymic breakdown of trimethylamine oxide (Tokunaga 1974). These relationships appear in Figure 30.

Reducing the rate and amount of protein denaturation depends on good quality raw material, low storage temperatures with minimum temperature fluctuation, and proper packaging to prevent desiccation. The rate of protein denaturation decreases as storage temperature decreases. Seafood studies have shown that a storage temperature of –20°F or lower is needed to provide a long storage time.

**Oxidation Reactions**

The second of the two major causes of quality loss during frozen storage of seafood is lipid oxidation, the most important cause of quality loss in fatty fish. Lipid oxidation causes loss of flavor and nutrition. The result is the unpleasant odor and flavor called rancidity. In chill-stored fish (unfrozen), bacterial spoilage lowers the quality to “poor” before appreciable lipid oxidation occurs. Bacterial spoilage is retarded in fish stored frozen, and very often rancidity becomes a problem. Fatty fish such as herring can become rancid in a month or less if storage conditions are not controlled.

Seafood lipids are healthful, but they are susceptible to oxidation, because the fatty acids are highly unsaturated. In fish, as much as one-third of the fatty acids are unsaturated. Unsaturated refers to an area of a fatty acid where there are not enough hydrogens to fully saturate the carbon atoms. This unsaturation results in carbon-to-carbon double bonds. The oxidation of unsaturated fatty acids proceeds by a free-radical mechanism involving the formation of hydroperoxides and their subsequent breakdown to aldehydes, ketones, alcohols, short-chain fatty acids, and other hydrocarbons. Volatile carbonyl compounds are thought to cause rancid odor and flavor.

The rate of oxidation depends on lipid composition, the presence of catalysts or antioxidants, pH, and storage temperature. Antioxidants can be those that naturally occur in foods or those that are added during processing. Research has shown that free fatty acids formed by the hydrolysis of triglycerides are more susceptible to oxidation, whereas free fatty acids formed by hydrolysis of phospholipids are less susceptible.
Oxidation reactions also cause undesirable color changes. Color in fish skin and flesh is largely due to carotenoid pigments. These are yellow, orange, red, and purple pigments found in both plants and animals. In fish, the most common carotenoid is astaxanthin (others include zeaxanthin, lutein, canthaxanthin, and tunaxanthin.) Oxidation of carotenoid pigments is responsible for fading flesh color in fish such as salmon and of skin color in several species of rockfish and some shellfish. In some fish and shellfish with white or creamy white flesh, oxidation reactions cause yellowing or darkening during long-term cold storage. The term “rust” or “rusting” refers to the migration of oil to the surface of fish during cold storage, giving a yellow or light brown discoloration. The rust discoloration has been attributed to reactions of amino acids or free amino groups of proteins with reducing sugars or with some lipid oxidation products.

To prevent undesirable oxidative changes, keep oxygen away from the seafood product. This can be done by glazing to provide a covering of ice, packaging with an oxygen-impermeable material, and using an antioxidant in a dip or glaze. The most effective protection is vacuum packaging, using a film with low permeability to oxygen in combination with an antioxidant dip such as sodium erythorbate.

**ENZYMES ACTIVITY**

Freezing does not inactivate the enzymes in food. In fact, enzymes are responsible for destructive changes that take place during freezing and frozen storage. An example is soft texture caused by enzyme breakdown of muscle connective tissue. For vegetables, blanching or applying enough heat to inactivate enzymes provides longer frozen storage life. Except for crab and shrimp, however, most seafood is frozen without heat treatment.

Low temperature is important in reducing unwanted enzyme activity. Below about 15°F, reaction rates due to enzymes decrease. At 0°F, they are slow enough to allow short storage times for fishery products with good-to-excellent frozen storage characteristics. For longer frozen storage times and for fishery products with fair-to-poor storage characteristics, a temperature of -20°F or lower is needed. Species with fair-to-poor storage characteristics include those with high lipid content and those with enzymes that cause the breakdown of trimethylamine oxide to dimethylamine and formaldehyde.

The usual effect of lower temperature on any chemical reaction is to decrease the reaction rate. Experimental work with several different enzymes in fish and shellfish, however, shows that reaction rates are higher when fish muscle is partially frozen in the 15 to 30°F temperature range (Nowlan and Dyer 1969, Toyomizu and Shono 1972, French et al. 1988). For example, protein hydrolysis is fastest at 26 to 28°F, phospholipid hydrolysis is most rapid at about 25°F, adenosine triphosphate (ATP) breakdown is highest at about 29°F, and glycoside breakdown at 25 to 26°F. These high rates are due to a concentration of enzymes and reactants as water molecules freeze into pure-water crystals. Also, large ice crystals disrupt cells, releasing enzymes and increasing the susceptibility of the tissue to enzyme breakdown.

The temperature zone from 30° down to 23-24°F has been called the “critical zone.” Avoid long, slow freezing conditions that hold the product in the critical zone longer than absolutely necessary. There can be some trade-offs—bacterial action is greatly reduced at these temperatures and is very slow at the low end of this range. Thus storage at partial freezing or superchilling temperatures has been used successfully to transport salmon intended for canning. Fish intended to be frozen and stored, however, should never be kept in the critical zone.

**LIPID HYDROLYSIS**

During frozen storage of fish, lipids hydrolyze to free fatty acids. Both phospholipids and triglycerides are hydrolyzed, with the phospholipids being the more susceptible of the two. Cholesterol esters and waxes undergo very little change during frozen storage. The rates of phospholipid and triglyceride hydrolysis vary with species and storage temperature.

The fish enzymes called lipases that hydrolyze lipids resist low temperatures and continue to be active in frozen tissue. In some cases, there is an increase in the amount of active enzymes released during freezing. However, the usual effect of temperature on chemical reactions, including those
catalyzed by enzymes, is a slowing of the reaction rate as the temperature is lowered. An exception is the temperature zone a few degrees below the freezing point of water, where the reaction rate of enzyme-catalyzed reactions like lipid hydrolysis increases rather than decreases. Below 15°F the reaction rate decreases gradually, although lipid hydrolysis doesn’t stop completely even at –20°F.

The free fatty acids formed in seafoods by lipid hydrolysis do not themselves lower product quality. The increase in free fatty acids is associated with the denaturation of myofibrillar protein and the resulting undesirable texture. Free fatty acids catalyze (speed up) the conversion of native myofibrillar protein to denatured myofibrillar protein (see Figure 30). The increase of free fatty acids also affects the rate of oxidative lipid changes.

**DESICCATION**

Desiccation is the dehydration or drying up of a substance, in this case food product. In frozen storage, desiccation occurs by sublimation (passage of solid or ice directly to gas or water vapor without going through the liquid phase or water). It is surprising that sublimation occurs so readily as much heat must be absorbed to enable this process. The latent heat of sublimation (1,114 Btus per pound) is a sum of the heat of fusion (144 Btus per pound) and the heat of vaporization (97 Btus per pound).

This dehydration of frozen seafood products is a common cause of quality loss, but it is much more easily prevented than protein denaturation and lipid oxidation. Moisture loss shows up as frost on freezer walls, cooling coils, and the inside of packages prepared for retail markets.

Moisture loss results in undesirable changes in texture and flavor loss due to loss of volatile components. Weight loss is also an economic loss. It has been found that the loss in weight of fish fillets during poorly controlled frozen storage can exceed 10% (Strasdine et al. 1978). For example, if fish fillets are worth $8.00 per pound, a retailer could lose as much as $80.00 for every 100 pounds of fish in frozen storage. In products prepared for retail market, this weight loss can cause packages to fall below the declared weights. Desiccation in an advanced state is called freezer burn. In extreme freezer burn, the skin of the fish has a dry, wrinkled look, the surface takes on a dull, chalky-white color, and the flesh becomes dry and tough. Desiccation of seafood accelerates the rate of both protein denaturation and lipid oxidation.

Prevent desiccation by glazing or by packaging with material having low permeability to water vapor. Fluctuating storage temperatures increase moisture loss, and should be minimized. The operator must avoid fluctuations in cold storage temperature caused by (1) adding warmer products, (2) leaving doors open, (3) starting and stopping refrigeration machinery, (4) lack of air curtains, and (5) small package size. To avoid dehydration of unwrapped, unglazed fish or fish products, keep the relative humidity as high as possible. A relative humidity of 85-90% usually is adequate; maintaining higher values is difficult. Avoid the need for high humidity by storing seafood wrapped or glazed.
Chapter 6: The Causes and Control of Quality Loss

Most quality deterioration in frozen seafood occurs after freezing, unless unusually slow freezing rates are used and the product is held for a long time in the critical zone (23-30°F). The rate of quality loss of frozen seafood depends on the storage temperature and amount of temperature fluctuation.

The quality of any frozen food also depends on the “PPP” factors, or product-processing-packaging. “Product” refers to quality of the raw material, “processing” refers to the product form, and “packaging” refers to the protection used to avoid the problems of desiccation and oxidation.

RAW MATERIAL QUALITY

Maintaining the high quality of frozen seafood products is critical to meet the standards imposed by producing companies, by customers, by country of origin, and by importing countries. Freezing does not improve fish quality. If freezing and cold storage are properly carried out, however, the quality of the fish will be maintained very close to what it was at the time of freezing. Consequently, good quality frozen seafood requires good quality raw material. There are two aspects of raw material quality: (1) intrinsic quality, and (2) prefreezing treatment.

Intrinsic quality is the quality or value of the fish when it is harvested. The harvester can’t change or control intrinsic quality. Important factors that affect intrinsic quality are species, sex, fishing ground, nutritional condition, size, maturity, season, food consumed, parasites, and environmental contaminants. They all affect the raw material quality and the price the fisherman gets for the catch.

Prefreezing treatment is the handling and storage of raw material between catching and freezing. The fisherman and fish processor control the prefreezing treatment, which includes:

- Length of time between harvesting and freezing.
- Holding temperature.
- Stage of rigor mortis when freezing.
- Processing procedures (product form).

These factors are just as important as intrinsic quality factors. Fish subjected to improper prefreezing treatment will not be of high quality even if they get excellent treatment during freezing and frozen storage.

Seafood treatment should be as good for fish intended to be frozen as for fish intended for the fresh market. All fish and shellfish should be chilled quickly and completely immediately after catching, using ice or chilled or refrigerated seawater. Once the product is chilled, the allowable holding time will vary with the species. For example, hold raw herring and other small or fatty fish at 32°F for no longer than a day. Hold raw halibut and other species with good-to-excellent frozen characteristics at 32°F for no longer than 3-5 days.

It is important to keep the chilled storage temperature as close as possible to 32°F, and to keep the chilled storage time as short as possible. Doyle (1989) has summarized research data to show how storage temperatures exceeding 32°F will affect spoilage. Table 15 shows how spoilage rate (r) is expected to vary with temperature, compared to an assigned rate of 1.0 at 32°F. The right-hand columns in Table 15 express this in terms of “equiva-
lent days on ice.” For example, holding seafood at 32.0°F for 24 hours is 1.0 day on ice; holding the product at 50.0°F for 18 hours would produce the same spoilage as if it were held on ice for 3.0 days.

Only the highest quality raw material should be used to prepare frozen products. Poor quality frozen seafood is too often caused by freezing stale fish. The practice of freezing seafood near the end of its chilled storage life must be eliminated. When lower quality frozen seafood is the result of a delay in freezing, the product should be labeled and sold as a cheaper brand in markets where lower quality is acceptable.

### Holding Temperature

Holding temperature affects the storage life of frozen seafood more than any other factor. The holding temperature must be low enough to stop or greatly retard bacterial action, enzyme changes, and non-enzyme chemical reactions.

Lowering the temperature of frozen seafood decreases the rate of deterioration at all temperatures except in the critical freezing zone (23-30°F), where enzyme-catalyzed reactions increase. Below about 15°F, bacterial growth and replication stops. At 0°F, enzyme and other chemical reactions are slow enough to allow storing frozen seafood for short periods. A storage temperature of −20°F is needed to store frozen seafood for longer periods, and to store fish or shellfish with high lipid content or high enzyme activity. Higher temperatures cause flavor loss and nutritional loss.

Although cold storage temperature is the most important influence on storage life of frozen seafood, other factors (described below) can also be very important.

#### Temperature Fluctuation

Storage temperature fluctuation is another factor that decreases frozen shelf life. It is impossible to keep food in frozen storage at a constant temperature, because of the cyclical operation of the refrigeration equipment and movement of product in and out of storage during transport from processor to wholesaler to retailer.

For fish or fish fillets stored at 0°F, fluctuations in temperature to 15°F (as may occur during transport) are very detrimental (Dyer et al. 1957). Considerable deterioration takes place, reducing shelf life greatly. For seafood products held at −20°F, fluctuations to −15 or −10°F cut storage life in half, and fluctuations to 0°F cut shelf life to one-third. The effects of temperature fluctuation are cumulative. Shelf life loss can be estimated if a record of storage temperatures is kept.

### Table 15. Relative rates of seafood spoilage for different temperatures and times

<table>
<thead>
<tr>
<th>Holding temperature (°F)</th>
<th>Relative rate of spoilage = r</th>
<th>Equivalent days on ice</th>
<th>Time at elevated holding temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>4 hr</td>
<td>8 hr</td>
<td>12 hr</td>
</tr>
<tr>
<td>28.4</td>
<td>0.6</td>
<td>0.1</td>
<td>0.2</td>
</tr>
<tr>
<td>32.0</td>
<td>1.0</td>
<td>0.7</td>
<td>0.3</td>
</tr>
<tr>
<td>35.6</td>
<td>1.4</td>
<td>0.4</td>
<td>0.6</td>
</tr>
<tr>
<td>39.2</td>
<td>2.0</td>
<td>0.6</td>
<td>1.0</td>
</tr>
<tr>
<td>42.8</td>
<td>2.6</td>
<td>0.8</td>
<td>1.0</td>
</tr>
<tr>
<td>46.4</td>
<td>3.2</td>
<td>1.1</td>
<td>1.6</td>
</tr>
<tr>
<td>50.0</td>
<td>4.0</td>
<td>1.3</td>
<td>2.0</td>
</tr>
<tr>
<td>53.6</td>
<td>4.8</td>
<td>1.6</td>
<td>2.4</td>
</tr>
<tr>
<td>59.0</td>
<td>6.3</td>
<td>2.1</td>
<td>3.1</td>
</tr>
</tbody>
</table>

A relative spoilage rate (r) is a comparison to the rate for a product held at 32°F (where r is equal to 1). Because of biological variability within a species, numbers are approximate.
temperature is kept and storage characteristics of the species are known.

The problems with quality loss caused by fluctuating storage temperature are mainly due to ice recrystallization, moisture loss, and enzyme or other deteriorative chemical reactions. Reforma-
tion of ice crystals damages tissue structure and may lead to undesirable texture. The tissue damage increases the rates of dehydration and deteriorative chemical changes. Moisture loss due to sublimation of the ice at or near the product surface can be a serious problem in unpackaged raw material and in products with poor packaging. Frost or ice may form on the inside of seafood retail packages as the result of temperature fluctuation, and consumers often won't buy these products. Fluctuating tempera-
ture accelerates a number of chemical changes in seafood. The most important are oxidative changes leading to rancidity (lipid oxidation) and discoloration.

PACKAGING

Almost as important as temperature control in obtaining maximum shelf life is protection by some type of packaging. The functions of packaging include the following:

- **Envelop the product to assist in moving it to market.**
- **Protect the product from dehydration, oxidation, contamination, and mechanical damage.**
- **Identify the product to assist in storage management and to provide consumer information.**

Frozen product not packaged prior to freezing should be packaged as soon as possible after freeze-
ing. The packaging should be strong, waterproof, and stain-resistant, and should not contaminate the product. Closeness of fit is also an important property. The packaging material should be impermeable to fats and oils. To reduce the rate of dehydration, the material should have a low permeability to water vapor. To reduce the rate of oxidation, the material should have low oxygen permeability.

REPROCESSING

Double freezing describes a storage and processing operation in which the product is frozen, then thawed or partly thawed to facilitate processing, then refrozen. Seafood processors often store fish and shellfish in bulk as frozen raw material, then thaw it to create the final products, and then refreeze some of those products. An example is frozen, dressed halibut, which is sometimes par-
tially thawed for steaking, then frozen for storage.

There is some quality loss during freezing, even with quick freezing. However, if the initial quality is very high (that is, if the first freezing is done with very fresh fish), double freezing can still result in very good quality. Fish frozen at sea are suitable for processing methods in which a second freezing is necessary.

Studies on cod, flounder, and rockfish show that refreezing can result in a very good quality prod-
uct, especially if the first freezing is done before rigor mortis sets in (Peters et al. 1968; MacCallum et al. 1969; Tomlinson et al. 1969, 1973). Experi-
mental work with pollock indicates that conditions for the first freezing and for thawing are more important than conditions for the second freezing to obtain a good quality product (Choe et al. 1975).
Chapter 7: Recommended Frozen Storage Conditions

Damage to frozen food from high or fluctuating temperatures cannot be corrected. Temperature abuse fluctuations, even if they are not extreme, can cause a serious damage problem. The following guidelines are important to maintain the quality of frozen seafood.

**TIME-TEMPERATURE-TOLERANCE**

The time-temperature-tolerance (TTT) concept was introduced very early in the study of frozen food stability. TTT uses the time-temperature history of frozen food during storage to describe shelf life (Edhborg 1965). TTT is based on the following assumptions:

- For every frozen product, there is a relationship between storage temperature and time stored at this temperature during which the product may change in quality.
- Quality loss at different frozen storage temperatures is irreversible and cumulative.
- The sequence of quality changes does not influence the total quality change.

Data from frozen storage studies can be used to produce quality maintenance diagrams, referred to as TTT diagrams (see Figure 31). TTT diagrams show the relationship between storage life and storage temperature and have been developed for many food products. If the three assumptions above are correct, TTT diagrams can be used to evaluate the effect of fluctuating temperature on the quality of a frozen product and to estimate remaining shelf life.

Application of the TTT concept depends first on accurate time and temperature data. Many temperature recording devices are available that can give the needed data.

Second, a satisfactory method of determining the end of shelf life is needed. Storage life must be measured by overall acceptability, which has several limiting factors. The importance of these factors varies greatly depending on species, storage temperature, type of packaging, and others. Taste of the product is usually the limiting indicator, but other indicators (texture, for example) can become more important in overall desirability. Consequently, no single measure of quality, whether physical, chemical, or sensory, can be used as a standard measure for all seafood. Because of this, frozen storage stability estimates vary depending on the quality indicator.

Most seafood researchers and processors believe that temperature fluctuation itself is detrimental to the quality of frozen seafood (see discussion of temperature fluctuation and temperature control). This means that for seafood, quality loss during temperature fluctuation is more than simply a total of the loss at each holding temperature.

**RECOMMENDED HOLDING TEMPERATURE**

When designing a cold storage facility, you should consider all products that will be stored there and the length of time they will be held. Cold store operators should use the lowest practical storage temperature, since they may not know about all the species that will be stored, and some products may stay in storage longer than the time intended.

Temperature of the incoming product should be as close as possible to the temperature of the cold storage room to minimize temperature fluctuation.
in all products. Cold store operators should check temperatures frequently to be sure everything is operating properly, and to have a continuous record of product temperature.

The American Frozen Food Institute, the National Fisheries Institute, and the National Frozen Food Association all recommend 0°F as a storage temperature for frozen food warehouses and retail display cases. This may be fine for most foods, but not for seafood. A storage temperature of 0°F is suitable only for some fish and shellfish species held for brief periods. For most species, storage at 0°F results in appreciably lower quality. And for long-term storage of all species, a lower temperature is needed. Any frozen storage temperature above 0°F is unacceptable for seafood.

Tables 16, 17, and 18 give frozen storage characteristics of some Pacific fish and shellfish. Species differ considerably in the length of time they will retain high quality in frozen storage. For many species, an extremely wide range of frozen storage times is reported in the literature. This is due to variation in intrinsic quality and handling of the raw material, and variation in how the end point of storage life is determined. Learson and Licciardello (1986) found literature values for the shelf life of haddock at 0°F to range from 24 to 72 weeks, a three-fold difference.

---

**Figure 31. Time - Temperature - Tolerance (TTT) of frozen haddock and herring**

(from Sikorski and Kolakowska 1990)

*HQL (high quality life) is defined as the time of storage of the initially high quality product to the moment when the first statistically significant (p < 0.01) difference in quality appears.

*PSL (practical storage life) is defined as the period of storage during which the product retains its characteristic properties and suitability for human consumption or intended process.*

---

Recommended Frozen Storage Conditions
Table 16. Frozen storage temperature and storage life for marine fish and fish products that are low to moderate in lipid concentration

<table>
<thead>
<tr>
<th>Product</th>
<th>For highest quality&lt;sup&gt;a&lt;/sup&gt;</th>
<th>For good quality&lt;sup&gt;b&lt;/sup&gt;</th>
<th>Maximum storage temperature (°F)</th>
<th>Storage life (months)</th>
<th>References</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>2 months to consumption</td>
<td>6 months to consumption</td>
<td>0°F</td>
<td>−20°F</td>
<td></td>
</tr>
<tr>
<td>H&amp;G salmon</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Chum</td>
<td>−15</td>
<td>−20</td>
<td>4</td>
<td>8</td>
<td></td>
</tr>
<tr>
<td>Coho</td>
<td>−10</td>
<td>−20</td>
<td>6</td>
<td>10</td>
<td></td>
</tr>
<tr>
<td>Pink</td>
<td>−15</td>
<td>−30</td>
<td>3</td>
<td>6</td>
<td></td>
</tr>
<tr>
<td>Sockeye</td>
<td>−10</td>
<td>−20</td>
<td>7</td>
<td>12</td>
<td></td>
</tr>
<tr>
<td>Farmed Atlantic salmon</td>
<td>−10</td>
<td>−20</td>
<td>7</td>
<td>12</td>
<td></td>
</tr>
<tr>
<td>Pacific cod</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>IQF fillets</td>
<td>−10</td>
<td>−20</td>
<td>7</td>
<td>12</td>
<td></td>
</tr>
<tr>
<td>Fillet blocks</td>
<td>−10</td>
<td>−20</td>
<td>8</td>
<td>15</td>
<td></td>
</tr>
<tr>
<td>Pacific whiting</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>IQF fillets</td>
<td>−20</td>
<td>−30</td>
<td>6</td>
<td>10</td>
<td></td>
</tr>
<tr>
<td>Fillet blocks</td>
<td>−10</td>
<td>−20</td>
<td>7</td>
<td>14</td>
<td></td>
</tr>
<tr>
<td>Surimi</td>
<td>−15</td>
<td>−25</td>
<td>6</td>
<td>12</td>
<td></td>
</tr>
<tr>
<td>Greenlings</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Lingcod</td>
<td>0</td>
<td>−10</td>
<td>10</td>
<td>20</td>
<td></td>
</tr>
<tr>
<td>Alaska pollock</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>IQF fillets</td>
<td>−15</td>
<td>−20</td>
<td>6</td>
<td>10</td>
<td></td>
</tr>
<tr>
<td>Fillet blocks</td>
<td>−10</td>
<td>−20</td>
<td>7</td>
<td>12</td>
<td></td>
</tr>
<tr>
<td>Surimi</td>
<td>−22</td>
<td>−22</td>
<td>6</td>
<td>12</td>
<td></td>
</tr>
<tr>
<td>Pacific halibut</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fletsches</td>
<td>−10</td>
<td>−20</td>
<td>8</td>
<td>16</td>
<td></td>
</tr>
</tbody>
</table>

<sup>a</sup>Storage temperatures given are the highest that will still allow minimal loss of quality. At the end of the frozen storage period, the product will be almost as good as fresh fish that has been held properly chilled and is frozen within a few days after harvest. This level of quality is termed HQL (high or highest quality life). No quality changes will be detected by a trained sensory panel.

<sup>b</sup>Practical storage life is defined as the length of time the product remains in a condition that can be described as good quality. At the end of good quality life (practical storage life or PSL), quality has diminished but consumers purchasing this product would be likely to purchase it again. At longer storage times, the product will be of fair to poor quality; although it may still be edible, it is not the quality of seafood that the industry should be marketing.
### Table 16 continued

<table>
<thead>
<tr>
<th>Product</th>
<th>For highest quality</th>
<th>For good quality</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Maximum storage temperature (°F)</td>
<td>Storage life (months)</td>
</tr>
<tr>
<td></td>
<td>2 months to consumption</td>
<td>6 months to consumption</td>
</tr>
</tbody>
</table>
Table 17. Frozen storage temperature and storage life for marine fish and fish products that are moderate to high in lipid concentration

<table>
<thead>
<tr>
<th>Product</th>
<th>For highest quality(^a)</th>
<th>For good quality(^b)</th>
<th>References</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Maximum storage temperature (°F)</td>
<td>Storage life (months)</td>
<td>References</td>
</tr>
<tr>
<td></td>
<td>2 months to consumption</td>
<td>6 months to consumption</td>
<td>0°F</td>
</tr>
<tr>
<td></td>
<td>Albacore steaks</td>
<td>–10</td>
<td>7</td>
</tr>
<tr>
<td></td>
<td>Albacore loins</td>
<td>–10</td>
<td>8</td>
</tr>
<tr>
<td></td>
<td>Skipjack gutted head on</td>
<td>–15</td>
<td>5</td>
</tr>
<tr>
<td></td>
<td>Yellowfin gutted head on</td>
<td>–15</td>
<td>6</td>
</tr>
<tr>
<td>Herring roe</td>
<td>–10</td>
<td>–20</td>
<td>8</td>
</tr>
<tr>
<td></td>
<td>Eulachon</td>
<td>–20</td>
<td>4</td>
</tr>
<tr>
<td></td>
<td>Rainbow smelt</td>
<td>–10</td>
<td>6</td>
</tr>
<tr>
<td></td>
<td>Surf smelt</td>
<td>–10</td>
<td>6</td>
</tr>
<tr>
<td></td>
<td>Pacific sardine</td>
<td>–15</td>
<td>5</td>
</tr>
<tr>
<td>Shark</td>
<td>Salmon</td>
<td>–10</td>
<td>9</td>
</tr>
<tr>
<td></td>
<td>Thresher</td>
<td>–10</td>
<td>10</td>
</tr>
<tr>
<td>Skate</td>
<td>Whole wings</td>
<td>–10</td>
<td>2</td>
</tr>
</tbody>
</table>

\(^a\)Storage temperatures given are the highest that will still allow minimal loss of quality. At the end of the frozen storage period, the product will be almost as good as fresh fish that has been held properly chilled and is frozen within a few days after harvest. This level of quality is termed HQL (high or highest quality life). No quality changes will be detected by a trained sensory panel.

\(^b\)Practical storage life is defined as the length of time the product remains in a condition that can be described as good quality. At the end of good quality life (practical storage life or PSL), quality has diminished but consumers purchasing this product would be likely to purchase it again. At longer storage times, the product will be of fair to poor quality; although it may still be edible, it is not the quality of seafood that the industry should be marketing.
### Table 18. Frozen storage temperature and storage life for shellfish and other marine invertebrates

<table>
<thead>
<tr>
<th>Product</th>
<th>For highest quality&lt;sup&gt;a&lt;/sup&gt;</th>
<th>For good quality&lt;sup&gt;b&lt;/sup&gt;</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Maximum storage temperature (ºF)</td>
<td>Storage life (months)</td>
</tr>
<tr>
<td></td>
<td>2 months to consumption</td>
<td>6 months to consumption</td>
</tr>
<tr>
<td><strong>Spot shrimp</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cooked meat</td>
<td>−20</td>
<td>−30</td>
</tr>
<tr>
<td><strong>Pink shrimp</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Raw in shell</td>
<td>−10</td>
<td>−15</td>
</tr>
<tr>
<td>Cooked meat</td>
<td>−20</td>
<td>−30</td>
</tr>
<tr>
<td><strong>Sidestripe shrimp</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Raw in shell</td>
<td>0</td>
<td>−10</td>
</tr>
<tr>
<td>Cooked meat</td>
<td>−20</td>
<td>−30</td>
</tr>
<tr>
<td><strong>King crab</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cooked meat</td>
<td>−10</td>
<td>−25</td>
</tr>
<tr>
<td><strong>Dungeness crab</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cooked in shell</td>
<td>−15</td>
<td>−20</td>
</tr>
<tr>
<td>Cooked meat</td>
<td>−20</td>
<td>−30</td>
</tr>
<tr>
<td><strong>Tanner/snow crab (bairdi)</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cooked in shell</td>
<td>−15</td>
<td>−20</td>
</tr>
<tr>
<td>Cooked meat</td>
<td>−15</td>
<td>−30</td>
</tr>
<tr>
<td><strong>Tanner/snow crab (opilio)</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Cooked in shell</td>
<td>−15</td>
<td>−20</td>
</tr>
<tr>
<td>Cooked meat</td>
<td>−15</td>
<td>−30</td>
</tr>
<tr>
<td><strong>Pacific oyster</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Shucked meats</td>
<td>−15</td>
<td>−25</td>
</tr>
<tr>
<td><strong>Manilla clam</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Shucked meats</td>
<td>−10</td>
<td>−15</td>
</tr>
<tr>
<td><strong>Razor clam</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Whole in shell</td>
<td>−10</td>
<td>−15</td>
</tr>
<tr>
<td>Shucked meats</td>
<td>−15</td>
<td>−20</td>
</tr>
<tr>
<td><strong>Geoduck</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Neck meats</td>
<td>0</td>
<td>−10</td>
</tr>
<tr>
<td>Breast meats</td>
<td>−5</td>
<td>−15</td>
</tr>
<tr>
<td><strong>Mussel</strong></td>
<td>Cooked meats</td>
<td>−10</td>
</tr>
<tr>
<td>Cooked meats</td>
<td>−10</td>
<td>−20</td>
</tr>
<tr>
<td><strong>Scallop</strong></td>
<td>Raw meats</td>
<td>0</td>
</tr>
<tr>
<td><strong>California squid</strong></td>
<td>Raw</td>
<td>0</td>
</tr>
<tr>
<td>Dressed</td>
<td>0</td>
<td>−10</td>
</tr>
<tr>
<td><strong>Sea cucumber</strong></td>
<td>Raw</td>
<td>0</td>
</tr>
<tr>
<td>Boiled</td>
<td>0</td>
<td>−10</td>
</tr>
</tbody>
</table>

<sup>a</sup>Storage temperatures given are the highest that will still allow minimal loss of quality. At the end of the frozen storage period, the product will be almost as good as fresh shellfish that has been held properly chilled and is frozen within a few days after harvest. This level of quality is termed HQL (high or highest quality life). No quality changes will be detected by a trained sensory panel.

<sup>b</sup>Practical storage life is defined as the length of time the product remains in a condition that can be described as good quality. At the end of good quality life (practical storage life or PSL), quality has diminished but consumers purchasing this product would be likely to purchase it again. At longer storage times, the product will be of fair to poor quality; although it may still be edible, it is not the quality of seafood that the industry should be marketing.
When using Tables 16–18, it is important to consider the following points:

1. The time and temperature values in the tables are from scientific literature, and from fish processors and fisheries technologists involved in handling and storage research. Few references give values that can be put directly into the tables, since a variety of methods were used to determine end of shelf life. Some researchers used sensory methods, while others used chemical methods. Those who used sensory methods evaluated raw fish in some cases and cooked fish in other cases. Consequently, literature values were used only as a starting point and were modified after input from processors and researchers. The modified values appear in the tables.

2. Experimental values for storage life of frozen seafood are often not comparable. Chemical analysis data often does not correlate with sensory analysis data. In addition, sample size and control samples are sometimes not adequate to measure quality loss. The seafood industry has a great need for the manual proposed by Learson and Licciardello (1986), which would define standardized methods of measuring frozen storage life of seafood.

3. The storage times given in the tables should be regarded as the best attainable. The raw material is top quality, handling and processing are as good as possible, and protection is provided during storage by glazing or by packaging in materials with low permeabilities to oxygen and water vapor. Under normal commercial conditions, seafood may not store as well as indicated by the tables.

4. The maximum storage temperature values in the first two columns of Table 16 and 17 are not comparable to the storage life values in the second two columns (see the two footnotes for these tables). Temperatures in the first two columns provide what Sikorski and Kolakowska (1990) called “high quality life” (HQL). HQL is the storage time of initially high quality product to the moment when the first statistically significant difference in quality appears. The product is maintained very close to the quality of a properly chilled fresh product no more than a few days after harvest.

   The shelf life values in the second two columns give what Sikorski and Kolakowska called “practical storage life” (PSL). PSL is the storage time during which the product retains its suitability for human consumption. The ratio of PSL over HQL is usually between 2 and 5. The storage life values in the second two columns of the tables may be slightly lower than the PSL of Sikorski and Kolakowska, as the definition of practical shelf life used here presumes the quality is not simply suitable for consumption but is high enough to be considered “good” and a consumer would likely purchase it again.

5. Although we give single values in the two tables rather than a range of temperatures or storage times, the variation within species is large enough so that in some cases a range may be much more appropriate. For example, chinook salmon from the Yukon River may have very different storage characteristics from chinook salmon from Cook Inlet.

   It is strongly recommended that all seafood be held at –20°F or lower and that cold storage facilities be designed using this as a guideline. Although –20°F is considered a very good holding temperature for most seafood, even lower temperatures are better. Holding at temperatures below –20°F reduces quality loss and promotes longer shelf life, but it may not always be economical. For extra storage life or for very demanding products (like fatty fish to be held for up to a year, crab or shrimp, fish to be used for sushi or sashimi), a holding temperature of –30 to –40°F is needed. For fish or shellfish to be used as top grade sushi or sashimi, a storage temperature of –60 to –70°F is recommended.

**TEMPERATURE CONTROL: LIMITS OF FLUCTUATION**

Temperature fluctuation during frozen storage of seafood is detrimental to quality. It results in tissue damage caused by ice recrystallization and it promotes dehydration. If a frozen storage temperature fluctuates 5 or 10°F, the product storage life is reduced to half of what it could be.

Under commercial conditions, a 5 or 10°F fluctuation in frozen storage temperature is not unusual. To avoid temperature fluctuations:

- Do not leave a door open longer than necessary when adding or removing products from the cold store.
- Do not put products into the cold store before they are frozen completely to or below the cold store temperature.

To reduce moisture loss, minimize fluctuations in seafood storage as much as possible. Keep temperature fluctuation to 5°F or less. With a recording thermometer, monitor the cold store and all transit
Recommended Frozen Storage Conditions

Packaging Recommendations

Packaging to protect frozen seafood from dehydration, oxidation, and contamination. A good packaging material is strong, tight-fitting, low in permeability to water vapor, low in permeability to oxygen, and inexpensive (low labor and material costs). The following types of packaging are discussed here:

- Glazing
- Hydrocarbon polymer films
- Films made from cellulose
- Chlorinated polyvinyl chloride films
- Hermetically sealed metal cans

Glazing consists of adding a layer of ice to a product by dipping, spraying, or brushing. It is inexpensive, provides a tight fit, and has low permeability both to water vapor and oxygen. Consequently, it provides good protection against dehydration and oxidation. However, glazing is lost by sublimation, and periodic inspection and reglazing are necessary. The glaze is not very strong and will crack off if the product is mishandled. Additives such as sugar, cornstarch, salt, sodium alginate, and carboxymethylcellulose can be added to the glaze water to strengthen the glaze. Antioxidants such as ascorbic acid and sodium erythorbate are sometimes added. Glazing is now used primarily for individually quick frozen (IQF) fillets and whole fish. Fish or fish fillets packed as a shatter pack are protected with an interweaving of waxed paper or plastic sheeting between fish or fillets so they can be taken apart without thawing.

Hydrocarbon polymer films such as polyethylene and polypropylene are strong and inexpensive, but they do not provide a good barrier to water vapor and oxygen. Therefore, they don’t prevent dehydration and oxidation. Polyethylene bags are often used to hold glazed seafood products like IQF fillets or shrimp. Also, small whole fish, fish fillets, and shrimp can be frozen with added water in polyethylene bags.

Films made from cellulose, such as cellophane, are not good packaging material because of their very high permeability to water vapor and relatively high permeability to oxygen.

Chlorinated polyvinyl chloride films (like polyvinylidene chloride or Saran wrap) provide very good protection against dehydration and oxidation, because their permeability to water vapor and oxygen is very low. These are shrink films and can be used in vacuum-packaging processes.

Packing seafood in a metal can that will be evacuated, hermetically sealed, and frozen is an excellent way of protecting against dehydration and oxidation. This process has been used commercially to pack picked crab and shrimp. It is not widely used except for institutional packs, possibly because consumers, mistaking the product for one that has been retorted, may believe it can be stored at room temperature.
References


Appendix A

Manufacturers and Suppliers of Small Cold Storages

This incomplete list includes primarily U.S. companies that manufacture or supply small walk-in cold storage rooms. Many have a product range that also includes larger facilities, refrigeration equipment, and individual components. Contact these businesses for more details.

ADVANCE ENERGY TECHNOLOGIES, INC.
One Solar Drive
Clifton Park, NY 12065-1387
(800) 724-0198
www.advanceet.com

ALCHEM
3617 Strawberry Rd.
Anchorage, AK 99502-7111
(907) 243-2177
(888) 485-5002

AMERICAN PANEL
5800 SE 78th St.
Ocala, FL 34472-3412
(800) 327-3015
(352) 245-7055
www.americanpanel.com

AMERICAN WHOLESALIE CO. (USED EQUIPMENT SUPPLIER)
4519 Hamilton Ave
Cleveland, OH 44114
(216) 426-8882
www.awrco.com

BALLY REFRIGERATED BOXES
135 Little Nine Drive
Morehead City, NC 28557
(800) 24BALLY
www.ballyrefboxes.com
BARR, INC. (USED EQUIPMENT SUPPLIER)
1423 Planeview Dr.
Oshkosh, WI 54904
(920) 231-1701
www.barrinc.com

W.A. BROWN & SON, INC.
209 Long Meadow Dr.
Salisbury, NC 28147
(800) 438-2316
www.wabrown.com

ELLIOTT-WILLIAMS CO.
2900 E. 20th St.
Indianapolis, IN 46218
(317) 453-2295
www.elliott-williams.com

IMPERIAL MFG.
2271 NE 194th
Portland, OR 97230
(800) 238-4093
(503) 665-5539
www.imperialmfg.com

INTERNATIONAL COLD STORAGE CO. INC.
215 E. 13th
Andover, KS 67002
(800) 835-0001
(316) 733-4088
www.icsco.com

JOHNSON BUILDING SYSTEMS, INC.
975 Jaymor Road #4
Southampton, PA 18966
(800) 445-7249
www.johnsonbuildingsystems.com

KELLY CONTAINER, INC. (USED EQUIPMENT SUPPLIER)
911 South St. Rte. 75
Suffield, CT
(800) 628-0497
www.kellycontainer.com

KOLPAK
2915 Tennessee Ave. N
P.O. Box 550
Parsons, TN 38363
(731) 847-5306
(800) 826-7036
www.kolpak.com
KYSOR PANEL SYSTEMS
3201 NE Loop 820, Suite 150
Fort Worth, TX 76137
(800) 633-3426
www.kysorpanel.com

LINGLE REFRIGERATION
(DAVID A. LINGLE AND SON MFG.)
104 S. Knoxville Ave.
P.O. Box 519
Russellville, AR 72811
(479) 968-2500

MASTER-BILT
908 Highway 15 North
New Albany, MS 38652
(662) 534-9061
(800) 647-1284
www.master-bilt.com

RAM FREEZERS AND COOLERS, INC.
1395 E. 11th Ave.
Hialeah, FL 33010
P.O. Box 1664
Miami Springs, FL 33266-1664
(305) 887-1000

U.S. COOLER CO.
325 Payson Ave.
Quincy, IL 62301
(800) 521-2665
www.uscooler.com

ZER-O-LOC INC.
9757 Juanita Drive NE, Suite 119
Kirkland, WA 98033
(425) 823-4588
www.zeroloc.com
The Oregon Cold Storage Simulation Model

This Excel-based simulator, OCSSim, was developed at Oregon State University to provide preliminary guidance to managers, planners, members of the fishing and fish processing industries, and others who wish to select or plan a cold storage facility. Although the concepts used to develop this model are applicable to many food industries, OCSSim is designed for and geared toward the fishing and fish processing industries. Incorporating historic monthly seafood landings in Oregon, the linkage of seasonal inventories with regional catch rates is thus focused on Oregon. However, other inventory schedules can be entered to see how decisions on inventory control, combined with facility design, can influence annual and cumulative profitability for 20 years of proposed cold store operation. The model allows the user to adjust parameters (e.g., storage temperature, labor costs, product dissipation) based on their expectations of specific business operations.

The model is designed for use by individuals having only a basic understanding of computers and spreadsheets. Using a few user-provided parameters, the model will furnish estimates of costs and revenues associated with a particular cold storage facility design, as well as other outputs. It allows consideration of a range of complexity and includes three scenarios as examples and potential starting points.

A complete description with background information is presented by Burden et al. (2003). Download the report: www.uaf.edu/MAP/workshops/cold-storage/Merrick-revised.pdf. A copy of the model itself, with user manual, can be downloaded from the Web site of the Coastal Oregon Marine Experiment Station: marineresearch.oregon-state.edu.
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  Layout 28

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