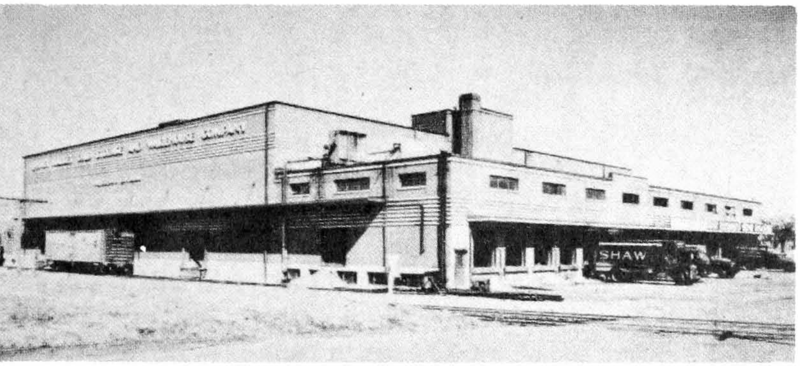
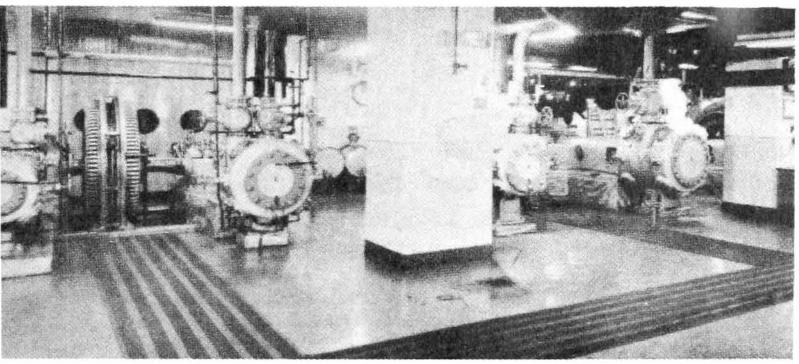
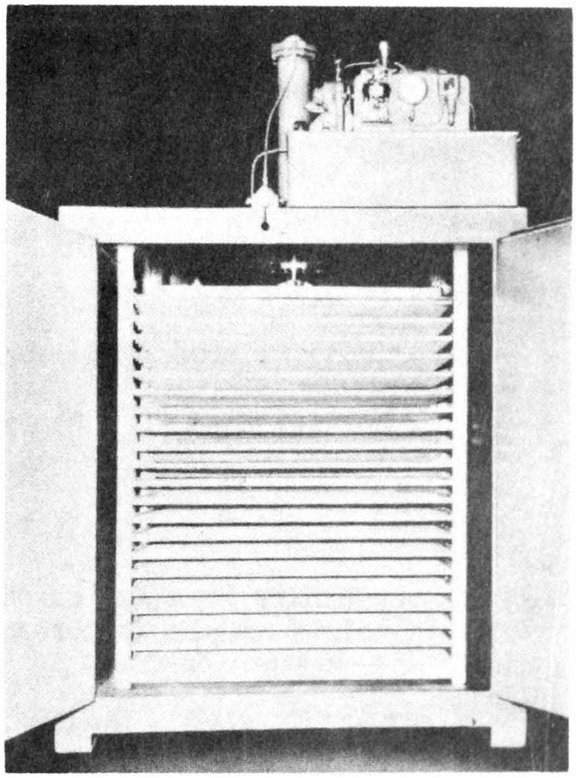
COLD STORAGE DESIGN AND REFRIGERATION EQUIPMENT

REFRIGERATION OF FISH - PART 1







UNITED STATES DEPARTMENT OF THE INTERIOR
Fred A. Seaton, Secretary

FISH AND WILDLIFE SERVICE
John L. Farley, Director

Fishery Leaflet 427

United States Department of the Interior, Fred Seaton, Secretary
Fish and Wildlife Service, John L. Farley, Director

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REFRIGERATION OF FISH - PART ONE

COLD STORAGE DESIGN AND REFRIGERATION EQUIPMENT

By Charles Butler (Section 1), Joseph W. Slavin (Sections 1, 2, and 3), Max Patashnik (Section 1), and F. Bruce Sanford (Section 1)

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This leaflet is part one of a series of five to be prepared within the broader overall subject matter of the refrigeration of fish. Titles of the other four leaflets are:

*Fishery Leaflet 428 - Handling Fresh Fish
Fishery Leaflet 429 - Factors to be Considered in the Freezing and Cold Storage of Fishery Products
Fishery Leaflet 430 - Preparing, Freezing, and Cold Storage of Fish, Shellfish, and Precooked Fishery Products
*Fishery Leaflet 431 - Distribution and Marketing of Frozen Fishery Products

The five leaflets in this series are prepared under the general supervision of Charles Butler, Chief, Technological Section, Branch of Commercial Fisheries, Washington, D. C., and edited by Joseph W. Slavin, Refrigeration Engineer, Fishery Technological Laboratory, East Boston, Massachusetts, and F. Bruce Sanford, Chemist, Fishery Technological Laboratory, Seattle, Washington.

* These leaflets have not yet been published.

SECTION 1

COLD STORAGE DESIGN

By Charles Butler, Chief, Technological Section; Joseph W. Slavin, Refrigeration Engineer; Max Patashnik, Fishery Products Technologist; and F. Bruce Sanford, Chemist *

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INTRODUCTION

Fish are frozen and stored in both public and private cold-storage plants. In large fishing ports such as Gloucester, Boston, and Seattle, there are many public cold-storage plants capable of handling the tremendous daily influx of fish. In small seaports, however, the infrequent and limited supply of fish does not warrant the expense of a large public cold-storage plant. Many of the fish producers in the smaller seaports therefore have to maintain their own freezing and storage facilities.

The many different types of fishery products that are produced from the various species of fish found in the oceans, lakes, and rivers of this country require considerable differences in handling, freezing, and cold-storage techniques. For example, large fish such as salmon, tuna and halibut are usually frozen, glazed, and stored in-the-round and then reglazed at periodic intervals; whereas smaller fish such as cod and haddock are usually filleted, and the fillets are packaged, frozen, and stored in cardboard cartons.

The differences in handling, freezing, and storing various fishery products, the extra labor required in freezing and glazing round fish, and the aversion to storing unpackaged fish with other frozen materials cause considerable reluctance on the part of the operators of public cold-storage plants to freeze and store certain fishery products. This reluctance, in addition to the need for immediate freezing, has made it almost mandatory that individual fish producers have at least limited freezing and cold-storage facilities.

The preparation of complete specifications for a particular installation will require the combined efforts of company personnel, construction and refrigeration engineers, local health and building inspectors, and many others. It will therefore suffice here to mention some of the factors that should be considered in constructing a new cold storage, remodeling an existing plant , or selecting a public cold storage.

The following is a discussion of some of the more important of these factors: (1) plant location, (2) design and construction, and (3) product environment. A discussion dealing with the calculation of refrigeration requirements has also been included.

LOCATION

A cold-storage plant to be used for freezing and storing fishery products should be located so as to have accessibility to the wharves where the fishing boats dock, to fish-processing plants, and to adequate facilities for rail and truck transportation. The unavailability of land or its high cost in some densely populated seaboard towns, however, has resulted in the construction of cold-storage plants in the outlying districts relatively far from the wharves or processing plants. The wide use of refrigerated trucks for local fish shipments, thereby making it possible to transport fish easily from the processing plant to the local freezer, has also been a factor in determining the location of the cold-storage plant.

In some cold-storage plants, particular types of fishery products, such as breaded fish sticks or breaded shrimp are frozen and stored. These plants are usually located adjacent to the processing plant, thereby enabling the processor to convey his product directly into the cold-storage plant for immediate freezing and subsequent storage.

As these examples indicate, the location of a cold-storage plant depends on many factors. Some of the more important of these that should be given consideration when the construction of a cold-storage plant to handle fishery products is contemplated are:

1. The source of fish supply.
2. The type of fishery product to be frozen and stored.
3. The facilities for rail and truck transportation.
4. The available labor supply.
5. The possibility of using sea or river water as cooling agents in the refrigeration condensers.
6. The cost of power.
7. The cost of land.
8. The taxes and miscellaneous costs.

DESIGN AND CONSTRUCTION

A refrigerated warehouse consists of a structure within an envelope of insulating and moisture-proofing material. The inside of this structure is divided into a number of rooms that are maintained at a predetermined temperature by use of adequate refrigeration equipment. To provide maximum operating efficiency, the walls, floors, and ceilings of cold-storage warehouses or of small private cold rooms must be of sturdy, tight construction.

It is beyond the scope of this leaflet to describe the different methods of construction that are used by engineering contractors to obtain a sturdy and tight warehouse or cold-storage room. Only those general requirements of basic design and construction that will acquaint the reader, contemplating such construction, with the subject sufficiently to enable him to discuss his needs intelligently with the engineers who specialize in this field will be discussed here. The discussion will include a description of the general design features as they pertain to refrigerated storage areas. Also, some of the more important specific design features that are common to all types of refrigerated warehouses, both large and small, will be included.

General Design Features

In the planning of a cold-storage plant, sufficient space must be allocated for efficient freezing, handling, and storing of the different types of fishery products peculiar to the geographical area. In addition, auxiliary space is also necessary for the refrigeration equipment, machine shop, elevators, offices, locker rooms, and facilities for loading and unloading. The general arrangement of the space within the plant must allow a smooth flow of the various products from loading platforms to freezer and then to the individual cold-storage rooms. The amount of space actually allocated for handling, freezing, and storing depends to a large extent on the type of products to be handled, whereas the auxiliary space depends on the over-all size and capacity of the warehouse.

The amount of space to be allocated for handling, freezing, and storing of the various types of fishery products varies considerably. A fishery cold-storage warehouse located in Seattle and designed to handle large amounts of round fish such as halibut, salmon, and tuna, for example, should have sufficient space for properly washing and glazing the fish. These fish are usually stored in-the-round one on top of the other in the cold-storage rooms. The additional washing and glazing operations together with the method of stacking in the cold rooms result not only in increased space requirements but also in a slower flow of fish in and out of the freezer than is found with packaged fish.

Fishery cold-storage plants located in the Boston and Gloucester areas are designed to handle large amounts of packaged fishery products and relatively small amounts of round fish. In these areas, the space necessary for washing and glazing the fish is considerably less than that required in a cold-storage plant in the Seattle area. In handling packaged products, however, additional space is necessary to allow a fast movement of the products into and out of the freezer.

With a small cold-storage room located within the processing plant, a fast flow of materials is sometimes sacrificed to obtain maximum utilization of the available space. Also, this particular type of operation does not usually warrant an investment in the equipment necessary for palletization. Products in the small cold-storage room are generally loaded on skids, which are moved into the freezer by hand trucks. The product is then usually unloaded and stacked by hand.

Frozen fish are stored in public cold-storage warehouses of the multistory, one-story, and modified one-story types, and in walk-in freezers located in the local seafood processing plants. Each of these refrigerated storage areas possess certain inherent advantages and disadvantages that should be considered by an operator planning on building a new plant or on making changes and improvements in an old one. The following is a discussion of some of the general design features that apply to refrigerated warehouses and storage areas.

Multistory Warehouse

The first refrigerated warehouses were of multistory construction (figure 1) consisting of a basement and of 3 to 12 floors. Many of these warehouses were designed to freeze and store a particular type of product, such as fish or meat, rather than a diversity of products. The limited refrigerated transportation facilities, the high cost of land within densely populated seaboard towns, the general construction techniques of the period, and the then low cost of labor were all factors that contributed to the design of the multistory building.

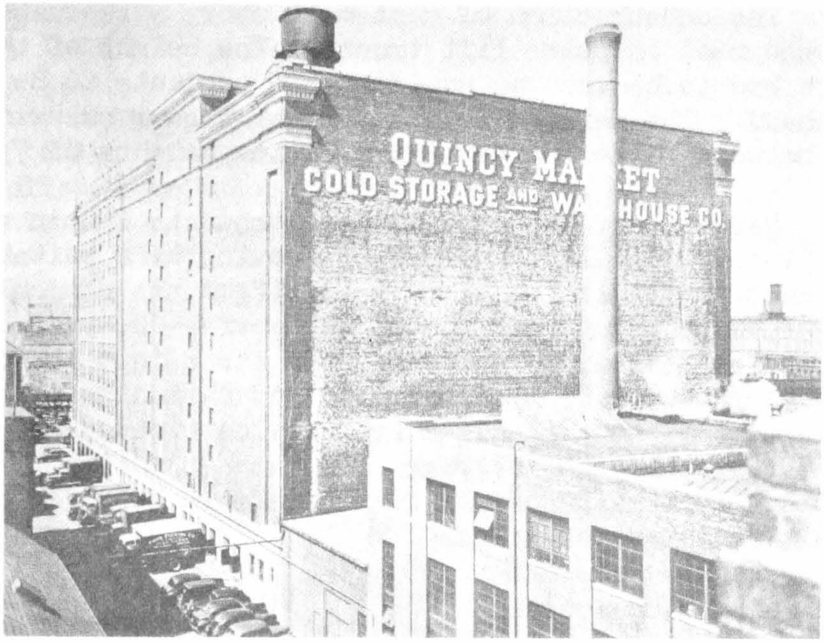

Figure 1.—A multistory warehouse. This warehouse is used to store meat and fishery products. (Photo courtesy of Quincy Market Cold Storage and Warehouse Co.)

Multistory warehouses used to freeze and store fishery products are usually designed with the refrigeration machinery located in the basement and the loading and unloading facilities located on the street and dock level. Freezer rooms of the sharp or air-blast type are generally located on the first floor adjacent to the receiving facilities or on the second floor, though in some plants, the freezers are located on higher floors.

If round fish are handled, the glazing and washing tanks are located in a refrigerated space maintained at temperatures only slightly above freezing. In some plants, the freezer is connected to the glazing and washing room by means of a chute. Thus, a steady flow of frozen fish from the freezer to the glazing room is attained. The various floors are serviced by two or more elevators. A common arrangement is to have the elevator and freezer doors open into a common aisle maintained at temperatures just above freezing. In some plants, the elevators open into non-refrigerated spaces at the street-level loading platform and then into refrigerated spaces at 0° F. in the upper floors. This arrangement can result in frost accumulation on the cables of the elevator thereby impairing its operation. For best design, the elevator shaft should (1) open, on all floors, into refrigerated spaces maintained at temperatures above freezing or (2) open, on all floors, into refrigerator spaces maintained at temperatures below freezing.

The construction of most multistory warehouses came before the development of power lift trucks. The height of the storage rooms therefore had to be such as would enable products to be efficiently stacked by hand. The height of most cold-storage rooms within such warehouses is between 10 to 12 feet, with piling heights of $7\frac{1}{2}$ to 10 feet.

Handling of packaged fishery products within a typical nonpalletized multistory building involves (1) moving by a suitable lift truck the product, which is loaded on skids, from the common carrier to the receiving room on the street floor; (2) moving the product from skids onto blast-freezer carts and moving the carts into the freezer; (3) removing frozen products from the freezer and cartoning packaged products; (4) loading cartoned, packaged products onto skids and moving the skids by a lift truck to the storage room; and (5) piling products within the storage room by hand. If round fish are handled, the operation is much the same as above except that the fish are frozen in a sharp freezer and then sent to a glazing room. Once glazed, they are either packed in wooden boxes or kept separated and transferred to a storage room (figure 2). In the handling of round fish, periodic reglazing is necessary to reduce dehydration during storage.

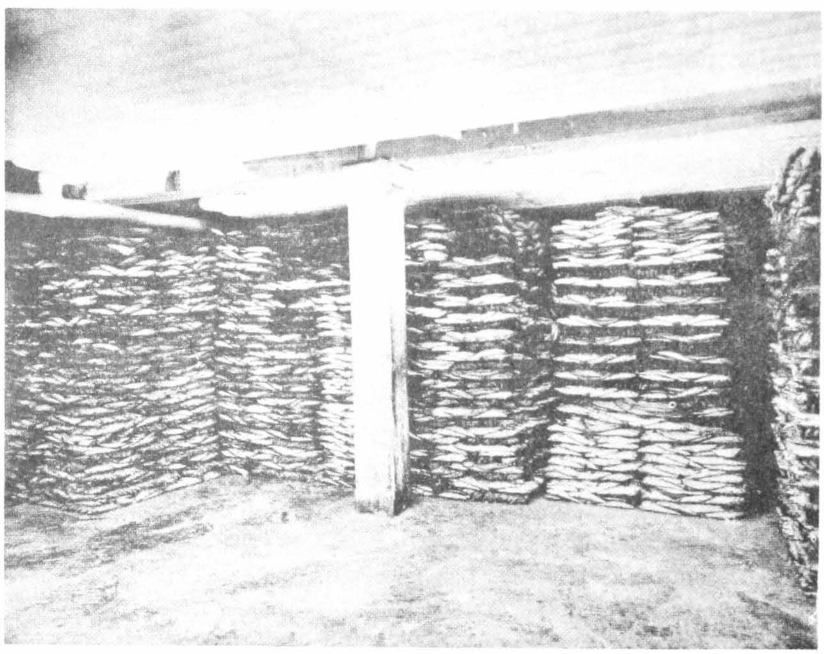

Figure 2.—Storage of mackerel in a multistory warehouse. (Photo courtesy of Quincy Market Cold Storage and Warehouse Co.)

It is obvious from the above that considerable labor is involved in freezing and storing the product in a multistory building. At the inception of refrigerated warehousing, this use of much labor was not an important point because of the relatively low labor costs during that period. The increased cost of labor throughout the years, however, has resulted in a proportional rise in the costs of product handling. This increase in product-handling costs together with the development of mechanical handling equipment has led to the construction of the one-story and modified one-story, completely palletized warehouses. The advantages of a non-palletized multistory warehouse, such as (1) the high ratio of product-storage space to gross-refrigerated space and (2) the low insulation costs, have been greatly outweighed by (1) the high cost of handling and (2) the slow movement of products within the warehouse.

To overcome these disadvantages, some operators have converted their multistory buildings to a palletized-type operation (figure 3). This conversion has been accomplished by using light-weight mechanical lift trucks and stacking products one or two pallets high--usually about $6\frac{1}{2}$ feet. The size of the pallet to be used is governed largely by the spacing of the columns within the building and the size of the door openings both in the storage room and in the common carrier. The conversion to a palletized operation from a non-palletized one reduces the labor necessary to handle the products by about 50 percent.

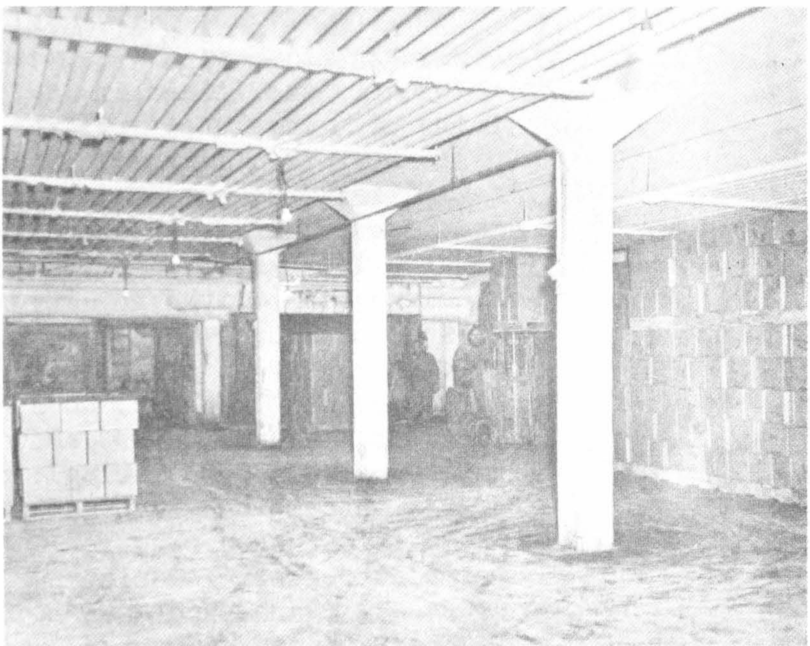

Figure 3.—Handling frozen products in a converted palletized multistory warehouse. Note the low piling heights and the close column spacing. (Photo courtesy of Quincy Market Cold Storage and Warehouse Co.)

The decision to convert a warehouse to palletization is usually arrived at through a study of the initial modification cost and the possible reduction in continued handling costs. In a building of the older type, the cost of reinforcing the floors and elevators to carry the additional load would probably be too high to warrant the conversion. In a multistory building of the newer type, however, the elevators and floors might be strong enough to support the additional load of the mechanical fork trucks and pallets. In this instance, the reduced handling cost might offset the cost of conversion.

Extensive work in the development of smaller and lighter mechanical lift trucks is presently taking place. Such a truck will contribute greatly to the conversion of the nonpalletized multistory warehouses.

One-Story and Modified One-Story Warehouse

The increase in the consumption of frozen foods within recent years has greatly influenced the design of cold-storage warehouses. Whereas many of the first refrigerated warehouses were designed to store a particular type of frozen product, the new-type warehouses are designed to store a wide range of products--such as Pizza pies, canned soups, packaged vegetables, and packaged fish. To facilitate fast handling of these different types of frozen foods at a minimum cost, the new types of cold-storage warehouses employ mechanical handling equipment (figure 4) and automatic labor-saving and safety devices.

Figure 4.--A fork lift truck handling frozen products in a refrigerated warehouse. (Photo courtesy of Quincy Market Cold Storage and Warehouse Co.)

In the planning of such a warehouse, storage space is usually sacrificed in favor of a completely palletized operation. Freezing facilities are in some instances reduced because of (1) the increased cost of handling and (2) the ability of the individual processor to freeze his own products. Also, usually little or no provisions are made for storing products requiring special attention--such as round fish, which require periodic reglazing during storage. The convenient location in the city is often waived in favor of greater transportation facilities, lower cost of land, and lower taxes in the suburban areas. These considerations and many more are reflected in the design of the one-story and modified one-story warehouses of today.

One-story warehouse.--The one-story warehouse is usually constructed on a base slab or on piles extending 3 to 4 feet above the ground. In the first case, heating coils or a series of warm-air ducts are provided under the base slab in order to prevent heaving of the floor due to freezing of the moisture in the ground. In the second case, the ventilated air space carries the cold air away from the ground surface. The buildings are generally made of brick, which forms a complete envelope around the insulation. In some instances, concrete ceilings have been replaced by insulation of the loose or rigid type.

One design of a single-story warehouse features a rectangular building over 400 feet square with a capacity of 2,000,000 cubic feet of refrigerated space at -5° to -10° F., with facilities for unloading 20 railroad cars and 40 large trailers. The railroad and truck unloading platforms are connected by a wide traffic aisle. On each side of the aisle is a large cold-storage room with a capacity of over 1,000,000 cubic feet. Each storage room is serviced through a large vestibule with flapper doors at both ends. Complete mechanical palletization throughout permits the products to be piled 18 feet high in a room that is 20 feet high. Aisles 12 feet wide within the storage rooms are necessary for the large 4,000-pound-capacity fork lift trucks employed. Special crates that clamp over the pallets prevent the crushing of the less dense products when they are being stacked. Other features of this plant are (1) a -40° F. blast freezing room with a capacity of 100,000 pounds of products every 15 hours, (2) a large cooler room, (3) finned-pipe coils located over the aisles to give maximum product piling, (4) elevated truck ramps, (5) warm oil circulating in pipes in the base slab to prevent frost heaving of the floor, (6) an automatic temperature recorder that records temperatures throughout the storage area of the plant at predetermined time intervals, and (7) outside electric plug-in receptacles for the motors driving the refrigeration compressors on trailer trucks.

In a one-story warehouse such as that described above, packaged products are handled almost exclusively. The products to be frozen are usually received loaded on pallets with dunnage between each row of packages. This dunnage insures proper air circulation around products during freezing. The pallets are moved from the trailer truck into the freezer by a fork lift truck. After the products have been frozen, they are removed from the freezer and stacked on another pallet. A reinforced wooden basket is clamped over the pallets to prevent crushing of the less dense products when stacked. The loaded pallet is then moved to the desired location within the cold-storage room.

If the products are already frozen upon arrival at the plant, the fork lift truck transports the loaded pallets from the railroad cars or trailer trucks to within the plant for storage (figure 5). The loaded pallets that are to be located on the lower half of a pile, however, must be equipped with the reinforced wooden baskets previously described. This procedure involves extra handling that would not be necessary in rooms having lower ceilings.

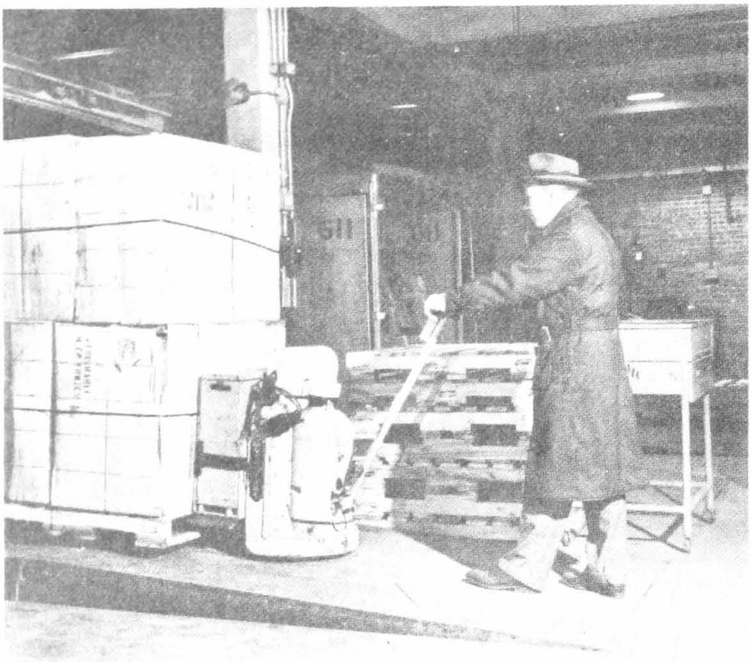

Figure 5.--Unloading frozen products from a refrigerated truck. Note self-leveling dock ramp which meets truck's tail gate. (Photo courtesy of Industrial Refrigeration)

The single-story warehouse, although possessing the advantage of fast handling of products, with small labor costs and large unloading facilities for refrigerated trucks and railroad cars, does have certain disadvantages some of which are (1) long horizontal distances to move products, (2) need for special crates to prevent the less dense products from being crushed because of high piling heights, (3) treatment of floor below freezers to prevent the ground from heaving, (4) lost storage space because of wide aisles necessary for fork lift trucks, (5) high amount of insulation required, and (6) high cost of the land in relation to the capacity of the building.

Modified one-story warehouse.--The disadvantages of the one-story warehouse have led to the design and construction of the modified one-story warehouse (figure 6). The construction techniques used in this

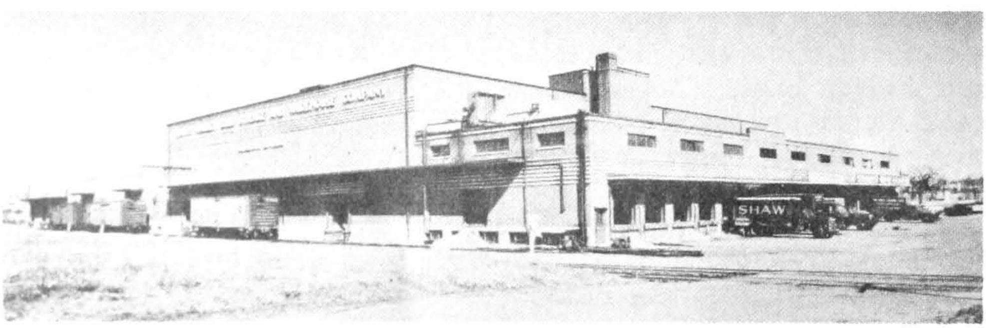

Figure 6.--A modified one-story warehouse. Note the different heights of the buildings and the rail and truck unloading facilities. (Photo courtesy of Quincy Market Cold Storage and Warehouse Co.)

building differ very little from those employed in the one-story warehouse, the principal design differences being in the use of two floors with different piling heights and, in some cases, a mezzanine floor with piling heights only one pallet high. Such an arrangement gives more economical storage and a greater versatility in handling than is possible in the one-story warehouse, where all products are piled to the same height.

One design of a modified one-story warehouse with a total refrigerated space of over 2 million cubic feet features a two-story building with a mezzanine floor connected to a single-story building to form one complete unit. A service building--which provides space for the offices, power plant, and truck receiving and shipping rooms--is joined to the two-story building. The three buildings together comprise one complete warehouse unit.

The first floor of this combination of two-story and single-story buildings accommodates (1) two large cold-storage rooms with a piling height of 16 feet and of $16\frac{1}{2}$ feet, respectively, (2) a cold-storage area under the mezzanine for break-up rooms, and (3) space on the mezzanine with a piling height of $5\frac{1}{2}$ feet for storing small lots of the less dense products (figure 7). The first floor also contains a large refrigerated

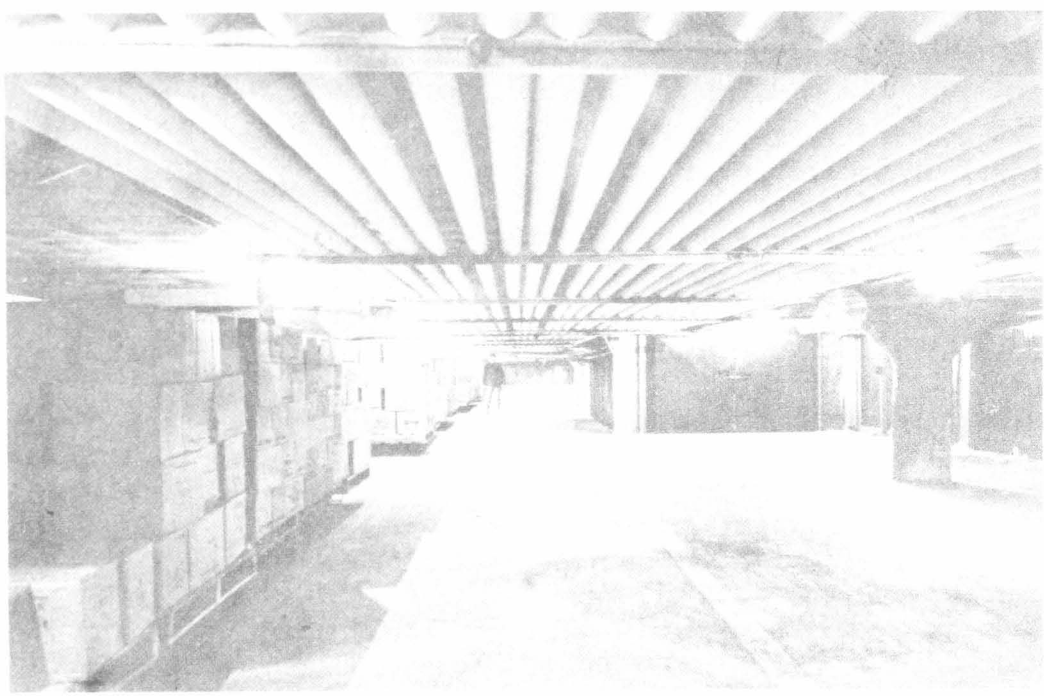

Figure 7.—Product storage on the mezzanine floor of a refrigerated warehouse. Note the low piling heights. (Photo courtesy of Quincy Market Cold Storage and Warehouse Co.)

receiving room running the length of the buildings. The receiving room has freezer doors opening into the cold-storage area, onto the truck unloading platform, and onto the railroad siding. Three blast-freezer rooms with a total capacity of 75,000 pounds of product per 8-hour day

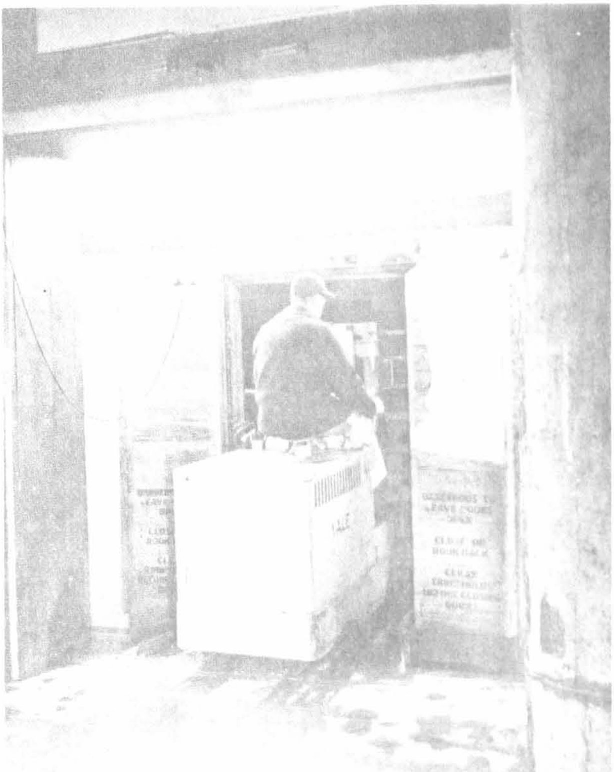

Figure 8.--Unloading frozen products from a refrigerated railroad car. (Photo courtesy of Quincy Market Cold Storage and Warehouse Co.)

The above design, by use of mechanical fork lift trucks (figure 8), allows a smooth flow of material from railroad cars and refrigerated trailers into the receiving room, thence to the blast freezer, and finally to the cold-storage area most suitable for the particular product. The less dense products, such as fish sticks, are received in a frozen condition from the refrigerated carrier and transported by 2,000-pound-capacity fork lift trucks to the mezzanine area (figure 9) for storage; and denser products, such as frozen fish fillets and fish blocks, are carried from the refrigerated carrier to either the first or second floor cold-storage area by 4,000-pound-capacity fork lift trucks. The use are also located on the first floor and can be filled directly from the receiving room.

The second floor of the building consists of one large cold-storage room at -5° F. with a piling height of 16 feet. This room is serviced by an elevator of 20,000-pound capacity, and the mezzanine is serviced by one of 10,000-pound capacity.

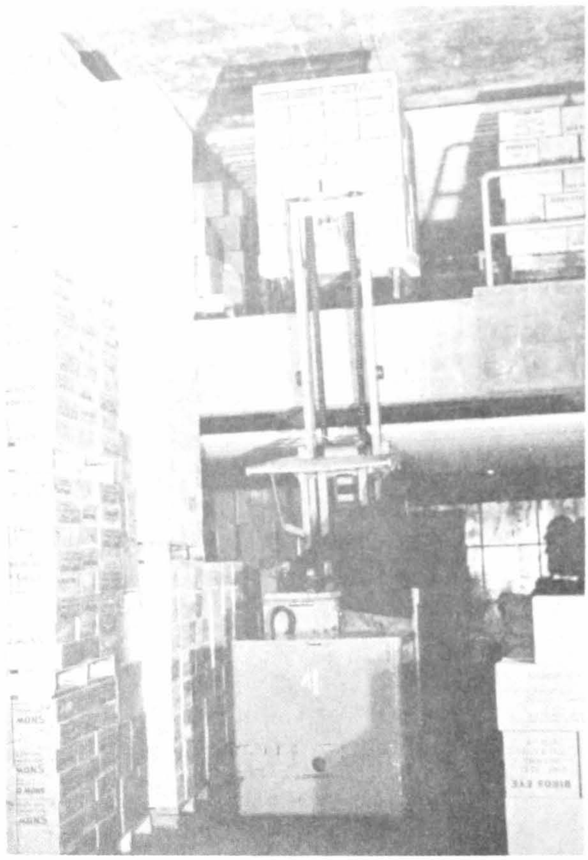

Figure 9.--View showing the first floor and mezzanine floor in a public warehouse. Note the fork lift truck on the first floor stacking products on the mezzanine floor. (Photo courtesy of Quincy Market Cold Storage and Warehouse Co.)

of the lower-capacity trucks in the mezzanine area makes it possible to limit the aisle to a width of 8 feet and the stacks to a height of one pallet. With the 4,000-pound-capacity truck in the other refrigerated areas, products may be stacked three to four pallets high with 12-foot-wide aisles (figure 10). This method of product storage eliminates the need for the special baskets required in the single-story building and gives efficient utilization of the space for product storage.

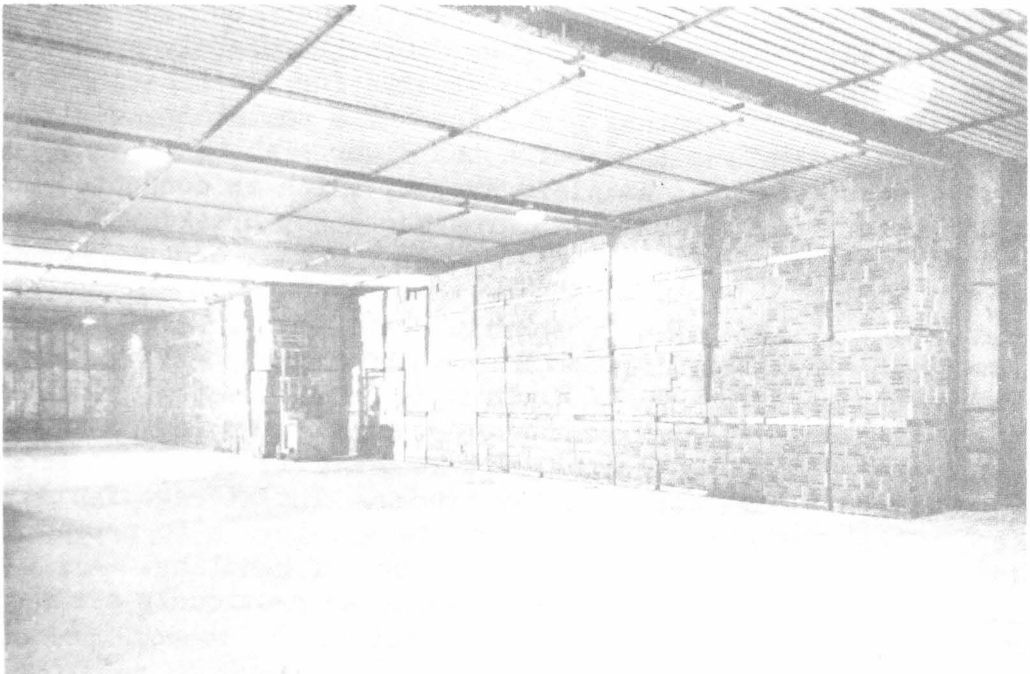

Figure 10.—Product storage on the first floor of a modified one-story warehouse. (Photo courtesy of Quincy Market Cold Storage and Warehouse Co.)

Another feature of this plant is the use of reflectant insulation material in the walls and ceiling of one of the buildings. This insulation is built up of 8 layers of aluminum sheets with a 5/8-inch dead air space between each layer. These sheets are stapled to treated wood screeds running in both vertical and horizontal directions. The outside sheet is sealed with metallic lead, which provides an effective vapor seal. A multistage potentiometer is used to measure the temperatures in the air space between each successive layer of reflectant insulation material.

The inside sheet of insulation is a heavy-gauge steel sheet, which is galvanized and treated with an aluminum lacquer for added protection. Additional structural members are provided on the walls to prevent any possible damage from fork lift trucks. The roof consists of a concrete slab with the same type of reflectant insulation as is used in the walls. The design of the roof is such that it will support and retain a 6-inch-deep layer of water when required. This layer of water reduces the effective refrigeration load in hot weather.

The temperature within the cold-storage rooms is held at -5° F. by Diesel-driven compressors of variable speed. These compressors maintain a constant ammonia suction pressure, making it possible to keep the room temperature from rising more than 5° F. above that of the cold brine circulating through the overhead pipe coils. This small temperature differential results in a high relative humidity, which contributes greatly to the storage life of the products. Other features of this plant are its wide, light-weight flapper doors, completely automatic refrigeration machinery, and hydraulic platforms for facilitating the unloading of the trailer trucks. Heating coils containing a glycol solution at 50° F. are installed below the floor slab to prevent frost heaving. The plant also has an alarm system, which is connected to the local telegraph office, to indicate fire, machinery breakdown, or forced entry of the premises.

A modified single-story cold-storage plant of over a million cubic feet to be used exclusively for the storage of frozen fishery products has recently been constructed in Gloucester, Massachusetts. This is one of the most modern fishery cold-storage warehouses in the world. The site of construction is such that it is possible to unload frozen fish directly from vessels and trailer trucks. The storage facilities consist of two large rooms at -5° F. on separate levels to provide maximum utilization of space with a minimum amount of handling. All of the modern labor-saving and safety devices mentioned previously are being utilized.

Walk-in Freezers and Coolers

Walk-in freezers and coolers are found in fish-processing plants, fish markets, and many other places handling fresh or frozen fishery products. The rooms comprising the freezers and coolers are similar to those used in large public warehouses, inasmuch as they are completely enclosed by suitable insulation and moisture-proofing material and are cooled by bare pipe coils, finned pipe coils, refrigerated plates, or blower-type unit coolers. The walk-in cooler or freezer, however, differs somewhat from the large public cold-storage warehouses because of its size and method of product storage and handling.

Fishery products usually are stored in the walk-in freezer (1) on shelves, (2) in cardboard or wooden boxes, or (3) in the round, stacked individually. The exact method of storage employed depends on the specific application. A small dealer selling fish chowder and other frozen specialty items, for example, might have the various products arranged on shelves within the freezer. This arrangement will enable him to select immediately the particular product desired by the customer. A fish processor located in New England, as another example, might store iced fresh fish in wooden boxes in the cooler at temperatures of 35° to

40° F. prior to processing. The fish, after being processed in the form of packaged fillets, would be frozen and then packed into cartons. These cartons would then be loaded on to skids and moved into the freezer by a small hand truck. The cartons, once in the freezer, would be unloaded and stacked by hand. If round fish were to be handled, they would be glazed and stored on shelves, in wooden boxes, or merely stacked one on top of the other.

In the walk-in freezer, the piling height for packaged products should be between $7\frac{1}{2}$ and 10 feet, with at least 1 foot clearance between the cooling coils and the top of the products. Piling heights in excess of 10 feet will result in extreme difficulty in product handling. In large cold-storage rooms, however, piling heights of 20 feet can be utilized if palletization and fork lift trucks are employed.

A designer, in allocating the proper amount of space for the storage of a product, must know its density. Such products as fish sticks and fish fillets have densities of approximately 25 to 30 pounds per cubic foot and of 55 to 60 pounds per cubic foot, respectively. Round fish stored in wooden boxes or stacked individually within the freezer have densities ranging from 30 to 35 pounds per cubic foot. If round fish of an average weight of 10 pounds are stored on shelves, about 1 square foot of shelf space should be allocated for each $7\frac{1}{2}$ pounds of fish. In the storage of 10-pound $2\frac{1}{2}$-inch-thick packages of fish fillets, about 1 square foot of shelf space should be allocated for each 10 to 11 pounds of fish.

The location of the cooler and freezer should be such that the cooler provides an anteroom for the freezer. With such an arrangement, the opening of the freezer door will result in the infiltration into the freezer of air at 35° to 40° F. from the cooler, rather than of the warm relatively humid air from the processing room. The prevention of the entrance of the warmer air when personnel enter and leave the freezer minimizes (1) frost accumulation on the coils and (2) temperature rise within the freezer.

Specific Design Features

Refrigerated Surfaces

The refrigerated surfaces consist of bare pipe coils, finned pipe coils, refrigerated plates, or blower-type unit coolers. The following gives a description of these refrigerated surfaces and a discussion of their relative merits.

Bare pipe coils.--Bare pipe coils consist of steel pipe that is fabricated to form continuous coils. These coils are suspended from the ceiling of the freezer by means of suitable hangers. Air space from 3 to 6 inches should be left between the top of the pipe coil and the ceiling to permit proper air circulation. The particular size of pipe used varies from 1 to 2 inches in diameter, depending on the desired capacity of the specific installation.

A refrigerant, such as ammonia or Freon-12, is allowed to expand through the pipes, in the direct expansion system; and a prechilled calcium-chloride brine solution is circulated through the pipes, in the indirect-expansion system (see "Evaporators," section 2). The advantages of expanding the refrigerant directly in the pipe coils are (1) a high rate of heat transfer and (2) a lower initial cost. The principal disadvantage of the direct-expansion system employing ammonia is the possible spoilage of product and the danger to personnel in event of pipe leakage or rupture. The use of an indirect-expansion system employing calcium-chloride brine circulating through the coils results in a slower rate of heat transfer and a higher initial cost but has the advantage of maintaining a more constant temperature within the storage room than would normally be possible with a direct-expansion system.

Bare pipe coils have the advantages of being durable, relatively inexpensive, and easily available but have the major disadvantages of small amount of cooling surface per linear foot of pipe, high cost of installation, and high weight per square foot of actual cooling surface.

<u>Finned pipe coils</u>.--Finned pipe coils have been used to a large extent within the last decade in cold-storage warehouses. Ammonia, Freon-12, or a calcium-chloride brine solution is usually circulated through the coils in order to provide the necessary refrigeration effect. A finned pipe coil has considerably more cooling surface per linear foot than has a bare pipe coil. One type of commercial finned pipe coil consisting of fins 7 inches square and 1 inch apart and bonded on a 2-inch-diameter steel pipe, for example, has 8.1 square feet of cooling surface per linear foot as compared with 0.62 square feet of cooling surface per linear foot for standard 2-inch-diameter steel pipe. Such a finned pipe coil has a heat transfer rate of 1.2 B.t.u. per hour per °F. per square foot of cooling surface as compared with a heat transfer rate of 2.0 B.t.u. per hour per °F. per square foot of cooling surface for standard 2-inch pipe. Since, however, the finned pipe coil has such a large cooling surface per linear foot, it is obvious that, for a particular installation, the total linear feet of finned pipe coils required to accomplish the necessary refrigeration effect would be considerably less than that required with bare pipe coils.

Inasmuch as the number of linear feet of finned pipe coil required is relatively small, most of the finned pipe coils can be located over the aisles of the cold-storage room. This location permits higher stacking of products in the storage area and prevents the dripping of water on the products when the pipes are defrosted. The fins of the pipe coils should be at least 1-inch apart because closer spacing will result in excessive frost build-up, necessitating frequent defrosting. The finned pipe coil loses more efficiency due to frost build-up than does either the bare pipe coil or the refrigerated plate.

Refrigerated plates.--Refrigerated plates are essentially a modification of the bare pipe coil. The simplest construction involves the welding of a suitable metal plate to one side of the coil. A better method, however, is to weld or solder two metal plates on each side. These plates are then crimped together on all sides and welded or soldered. The result, essentially, is a flattened pipe coil with a large amount of cooling surface.

A predetermined number of plates are connected together in parallel with common inlet and outlet pipes so as to form a complete bank of plates. These plates are suspended vertically from the ceiling of the refrigerated space in much the same manner as are the bare pipe coils. A refrigerant such as ammonia, Freon 12, or calcium-chloride brine is used to provide the necessary refrigeration effect. The heat-transfer rate per square foot of cooling surface is similar to that of bare pipe coils. The size of the plate is such, however, that there is a large amount of cooling surface per linear foot. In addition, such plates possess the advantage of losing very little efficiency due to frost accumulation, and of being very easy to defrost and to install. Refrigerated plates are used to a large extent in walk-in freezers and to some extent in large cold-storage plants.

Blower-type unit coolers.--The blower-type unit cooler consists essentially of a bank of bare or finned pipe coils with a fan to circulate the air within the storage room. This type of cooler is suitable for use in either a direct- or indirect-expansion system. In most installations, a direct-expansion system employing ammonia or Freon 12 is used because of its high efficiency and low initial cost.

Commercial blower-type unit coolers are of either the dry-pipe type or the spray type. The main difference between these two types is in the method of defrosting. In the dry-pipe type, defrosting is accomplished by the use of (1) water sprays, (2) circulation of hot refrigerant gas in the coils, or (3) electric heaters; whereas in the spray type, salt or glycol solutions are sprayed on the coils at periodic intervals to dissolve the frost. The salt or glycol solutions, after picking up the moisture from the dissolved frost on the coils, drain to a basin at the bottom of the cooling unit. These solutions are maintained at their proper strength by the addition of salt to the salt solution or by use of a concentrator with the glycol solution.

Since the unit cooler employs forced-air circulation, it is very important that the cooling surfaces of the units be of sufficient area so that the temperature differential between the circulating air and the refrigerant can be kept to 10° F. or less. The circulation of the air within a freezer using a unit cooler has a greater drying effect on the products than has the natural convection currents that are present in a freezer using bare pipe coils, finned pipe coils, or refrigerated plates. It is therefore necessary to maintain a higher relative humidity in the air in a forced-air circulating system than in a natural-air circulating system. The actual amount of moisture withdrawn with each type of refrigerated surface, however, varies greatly with the particular installation. In all instances, adequate consideration should be given to the selection of the refrigerated surfaces to be used so that it will be possible to maintain a relatively high humidity within the freezer, thereby keeping the dehydration of the product at the absolute minimum.

Insulation

Function and properties of insulation.--The function of insulation is to restrict the flow of heat into the refrigerated room from the outside surroundings. The ability of a particular type of insulation to resist the flow of heat by conduction from the warm side to the cold side is measured by the thermal conductivity of the material. (A discussion of thermal conductivities of insulating materials and of methods of calculating the heat gained through the insulation in refrigerated rooms is given later in this section.)

The three types of insulation suitable for use in refrigerated warehouses are (1) loose or fill material, (2) rigid board or block materials, and (3) reflectant materials. Within each of these three types, the available commercial products are numerous and offer a wide selection for a specific installation.

An insulation material for use in a cold-storage room should possess the following properties:

1. Low thermal conductivity.
2. Low water absorption.
3. Low water-vapor transmission.
4. Low inflammability.
5. Good mechanical properties.
6. Resistance to fungi and vermin.
7. Be easily cut, shaped, and glued, etc., in installing.

Selection of insulation.--Some of the factors to be considered in selecting insulating materials are (1) initial cost; (2) installation cost; (3) operating, maintenance, repair, and depreciation costs; (4) local weather conditions, temperature, and humidity; (5) type of product to be stored; (6) permanence of structure; and (7) temperature in cold-storage rooms.

The most satisfactory insulating material is the one that offers the best combination of characteristics desirable for a specific set of local conditions, at the most economical cost both initially and operations-wise. No one insulating material is superior in all respects. For example, one rigid type of insulation material--although offering the advantages of low thermal conductivity, high structural strength, and ease of installation--possesses the disadvantages of high density and the absorption of water. On the other hand, a competitive product that offers a higher resistance to heat transfer, a lower density, and imperviousness to water requires the use of special construction techniques for installation.

Application of insulation.--After the proper insulating material has been selected, careful consideration must be given to the method of applying it to the walls, ceiling, and floor of the refrigerated enclosure. The main object is to obtain a tight insulated envelope around the periphery of the cold-storage room. In applying the insulation, an adequate vapor seal must be used between the outer wall and the first layer of insulation. This vapor seal is necessary to restrict the passage of water vapor from the outside through the walls, ceiling, and floor 1/ of the refrigerated room.

If the insulation is not properly applied or if an ineffective vapor seal is used, water vapor will permeate the vapor barrier and, condensing to water, will freeze within the insulation. This formation of ice greatly decreases the insulating value of the material. In addition, some of the water vapor continues to migrate through the walls and deposits on the refrigerated coils within the room, necessitating frequent defrosting of the coils.

The thickness of insulation used is closely related to tax structure. In a given installation, for example, it may be less expensive to adopt the "minimum" safe insulation for temperatures and products and to expect larger power and maintenance bills, since these latter expenses come out of revenues earned before taxes are calculated, whereas the amortization of the building is after taxes. A table at the end of this section gives recommended insulation thicknesses for refrigerated warehouses located in various areas of the country.

Freezer Floors

In the older multistory warehouses, cooler rooms or nonrefrigerated rooms were located on the ground floor. Precautions against frost heaving of the floor were therefore not necessary. In recent years, however, the utilization of the ground floor for low-temperature storage has resulted in floor heaving and buckling in many warehouses.

Floor heaving is the result of the cumulative effects of ice pockets formed under the insulated floor as the ground is chilled below freezing. This formation of ice exerts an upward force on the floor slab, resulting in cracking and bulging of the floor (figure 11). The rate of ice

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1/ A vapor seal is not used if the cold storage room is on the ground floor. (See "Freezer Floors.")

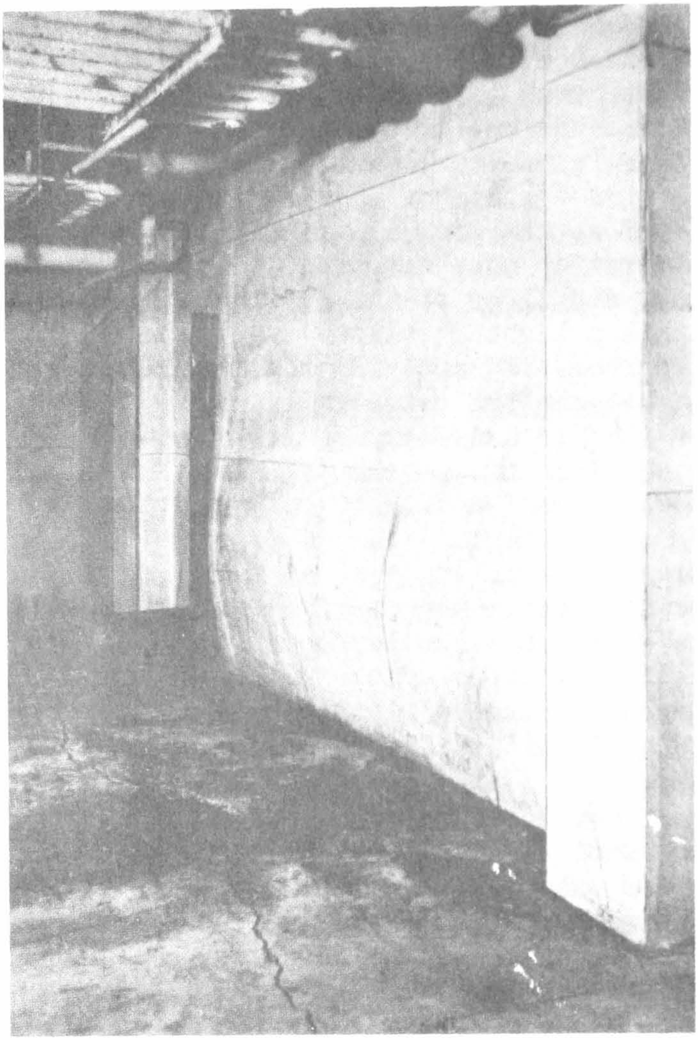

Figure 11.--Cracking and bulging of a freezer floor. (Photo courtesy of Armstrong Cork Co.)

formation in the ground depends on (1) the temperature of the cold-storage room, (2) the insulation thickness, (3) the characteristics of the soil, and (4) the height of the water level in the particular area.

The usual design of a freezer floor consists of a layer of gravel, on the surface of the ground, supporting (1) a concrete slab covered with a thick layer of insulation and (2) a concrete wearing-slab. The cold surface of the freezer floor causes a continuous migration of moisture from the lower layer of ground to the upper layer of ground and into the cold-storage room. A moisture-vapor-proof material should not be used between the floor slab and the insulation because the use of this material will result in trapping moisture in the surface layer of ground and will greatly accelerate frost heaving. A thick insulation material, on the other hand, will reduce the depth of the freezing zone and is therefore to be recommended.

In areas where the water table is low and the soil is very dry, use of sufficient amount of fill and suitable insulation might prevent frost heaving of the ground. To provide a positive method of protection, however, heat must be added to the ground by any one of several methods. A heated oil solution may be circulated through pipe coils in the slab below the insulation, for example, or electric heating elements may be enclosed in a conduit in the bottom slab. In another system, flues are formed in the ground by hollow tile, and the use of natural or forced air circulation through these flues furnishes the necessary heat to prevent the ground from freezing. A thermostatic control employed in the above systems permits maintenance of proper temperatures in the ground at all times.

Some cold-storage plants are constructed on piles, with the floor being located 4 to 5 feet above the ground level. If the floor area is relatively small, natural air circulation will give adequate protection against ground freezing. For large floor areas, however, forced air circulation utilizing one or more fans is necessary.

In the selection of a particular method of preventing frost heaving, consideration must be given to (1) initial cost, (2) operating cost, and (3) dependability of each particular system. The characteristics of the soil and the climatic conditions also affect this selection. Until adequate data are obtained on the causes of and the remedies for frost heaving, the owner must weigh the possible damage due to frost heaving against the increased cost of providing adequate protection.

Refrigerator Doors

Refrigerator doors are of the infitting, overlap, or sliding types. The thickness of insulation, hardware, and protective covering for each door varies with the particular installation.

Infitting or plug-type doors (figure 12) are relatively compact and

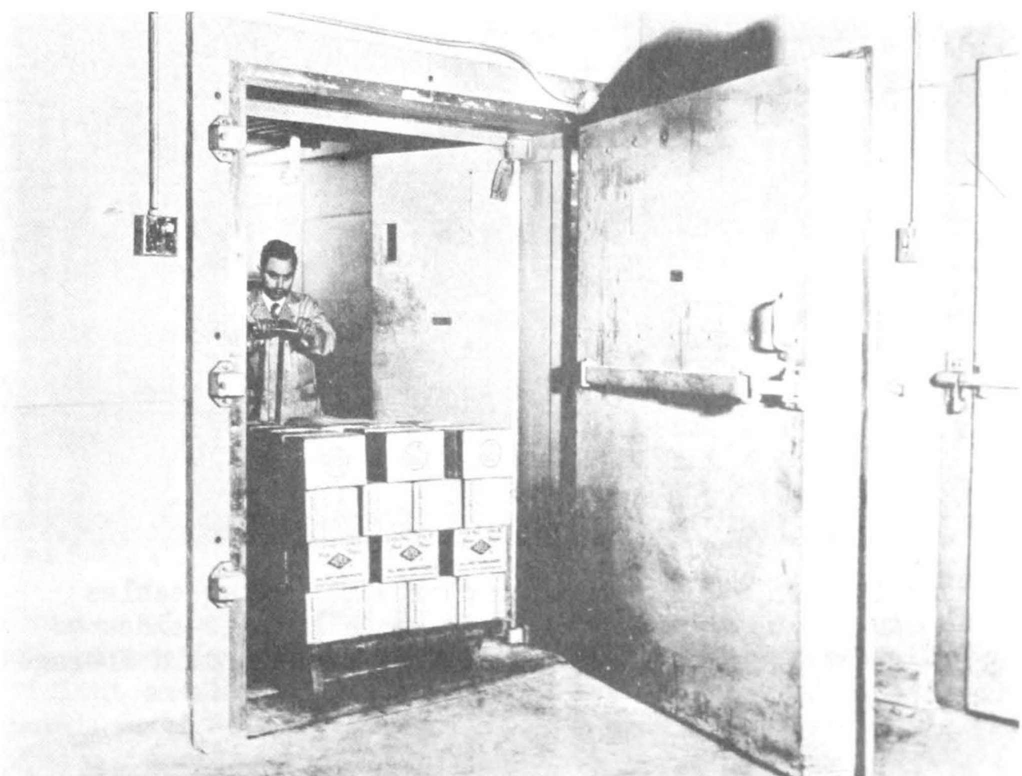

Figure 12.—An infitting freezer door equipped with a set of flapper doors to minimize loss of cold air while the storage room is being loaded and unloaded. (Photo courtesy of Jamison Cold Storage Door Co.)

light. Air leakage, because of the small amount of sealing area and of the build up of frost on the sides of the door, has been a problem in the past. In recent years, however, the construction of doors with a double seal, on inside and outside, and the use of suitable electric heating elements have overcome these difficulties. The infitting door is widely used for cooler rooms and is also suitable for 0° F. freezer rooms.

The overlap door (figure 13) was designed to eliminate some of the inherent disadvantages of the infitting door when used for low-temperature applications (0° to -60° F.). This door has a wide gasket to insure a tight seal. The use of an electric heating element in the door frame is recommended to prevent frost accumulation on the area in contact with the gasket. This door is widely used in commercial cold-storage warehouses and in quick freezers operating at temperatures as low as -60°F.

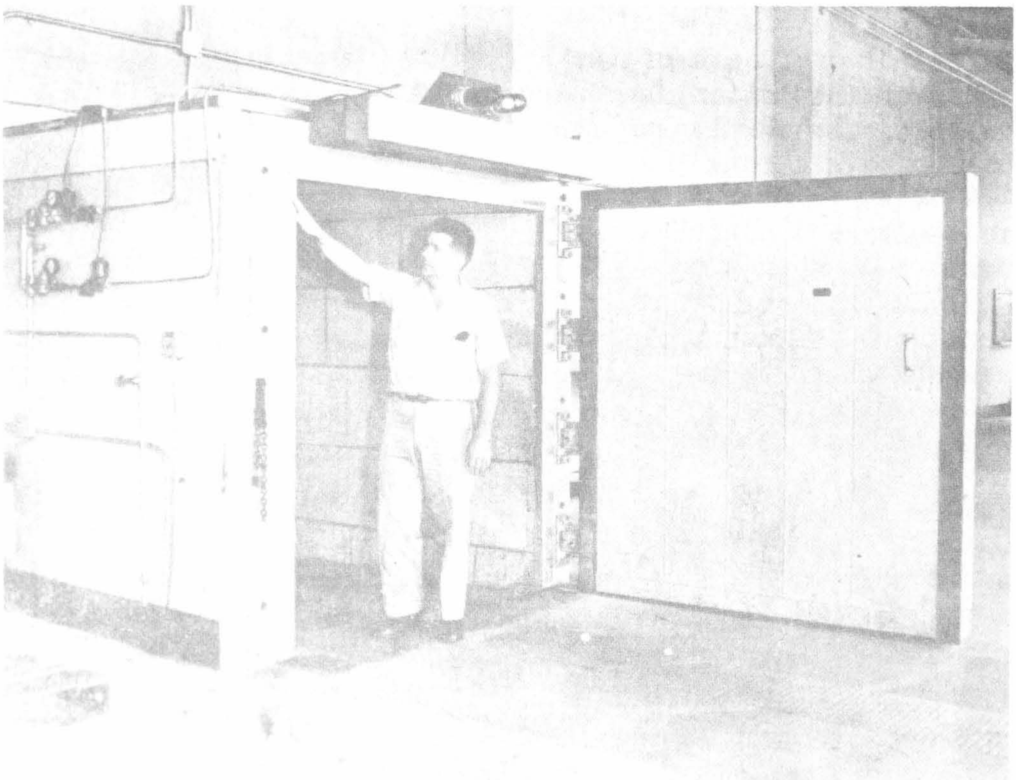

Figure 13.--An overlap freezer door with heater cables embedded beneath the cover plate on all three sides of the door frame. (Photo courtesy of Jamison Cold Storage Door Co.)

The sliding door is basically an overlap-type door suspended by pulleys, which ride on a track located above the door. A suitable linkage arrangement permits the door to move sidewards whenever it is desired. This door offers the advantage of more usable floor space, elimination of heavy load on the door frame, and ease of operation. The quick opening and closing action of these doors makes a double vestibule door less necessary.

Most vestibules have a refrigerator door and, in addition, a set of flapper doors at the entrance of the vestibule and another set of flapper doors between the vestibule and the refrigerated storage area. These flapper doors are available in plywood or rubber.

The plywood doors are usually covered with protective galvanized metal sheathing to minimize possible damage. These doors, although having a relatively low initial cost, are subject to damage from fork lift trucks. Frost accumulation on the doors is also a problem.

The rubber doors have a higher initial cost than do the plywood doors. This increase in initial cost, however, is offset by the ability of the doors to withstand the heavy abuse of fork lift trucks and to resist the accumulation of frost.

PRODUCT ENVIRONMENT

The term product environment refers to the atmosphere surrounding the product during storage. In the storage of fishery products, the air within the cold-storage room must have (1) a low temperature, (2) a high relative humidity, and (3) a low velocity. The effects of the temperature, humidity, and velocity of the air on the storage life of fishery products is described in Fishery Leaflet 429, section 2. The following will therefore be limited to a discussion of some of the factors that should be considered in the design of a warehouse in order to obtain the optimum air temperature, relative humidity, and air circulation.

Temperature

For satisfactory storage of frozen fishery products, the temperature of the storage room should be kept at 0° F. or lower and should undergo as little fluctuation as is possible. Temperatures below 0° F., although greatly increasing the storage life of the product, do require more costly construction and equipment and may increase operating expenses as well.

The number of square feet of coil cooling surface largely determines the equipment-operating costs. A cold-storage room at 0° F. with an insufficient amount of cooling coils, for example, might require a coil temperature of -20° F. to furnish the necessary refrigeration effect, whereas another installation with a large cooling surface might require a coil temperature of only -5° F. to accomplish the same purpose. In the first example, the refrigeration system would have to operate at evaporator temperatures of -20° F., which would result in (1) higher initial cost of refrigeration machinery, (2) higher cost of operation, and (3) lower cost of pipe coils. In the second example, the refrigeration system would have to operate at an evaporator temperature of -5° F., which would result in (1) lower initial cost of refrigeration machinery, (2) lower cost of operation, and (3) higher cost of pipe coils.

As the temperature of the air decreases, the ability of the air to hold water also decreases. Air at 0° F. and at a relative humidity of 100 percent, for example, contains about three times more water than does air at -20° F. and at the same relative humidity. Therefore, for a given relative humidity, air at a lower temperature will absorb less moisture from the product than will air at a higher temperature.

Recent developments in the design of refrigeration equipment have made it possible to obtain cold-storage-room temperatures of -10° F. far more efficiently than was ever before thought possible. Anyone contemplating the construction of a refrigerated warehouse should therefore carefully weigh the increased refrigeration costs for lower temperatures against the return from the pronounced increase in storage life of the product.

Relative Humidity

When air containing moisture undergoes a drop in temperature at a constant pressure, a temperature will be reached at which condensation and precipitation of the moisture will occur. This temperature is the dew point. If the dew point is equal to the temperature as indicated by a standard thermometer (dry-bulb temperature), the air is completely saturated and has a relative humidity of 100 percent. If, however, the dew point is lower than the dry-bulb temperature, the air is only partially saturated and has a correspondingly lower relative humidity. These definitions are necessary for the following considerations of the factors affecting humidity levels in the refrigerated warehouse.

The moisture contained within the air exerts a certain vapor pressure, which is ordinarily measured in pounds per square inch (p.s.i.). Other factors being equal, completely saturated air at a high temperature will have a higher vapor pressure and will contain more moisture than will completely saturated air at a lower temperature.

In an empty refrigerated room in which the coil temperature is say at -10° F. and the room temperature is at 0° F., condensation, on the coils, of the moisture in the air will occur until the dew point of the 0° F. air (dry-bulb temperature) within the freezer reaches -10° F. Air at 0° F. having a dew point of -10° F. exerts a vapor pressure of 0.0108 p.s.i. Correspondingly, if the air at 0° F. were completely saturated, it would have a dew point of 0° F. and exert a vapor pressure of 0.0188 p.s.i. The percent relative humidity is equal to the vapor pressure of the air within the room at its dew-point temperature (-10° F.) divided by the vapor pressure of completely saturated air at the dry-bulb temperature (0° F.).

This relationship is expressed by the formula:

$$H = \frac{P_1(100)}{P_2}$$

where H is the relative humidity, P_1 is the vapor pressure of water at the dew-point temperature, and P_2 is the vapor pressure of water at the dry-bulb temperature.

The use of this formula is illustrated in the following problem:

Sample Problem

In designing a cold-storage plant, a producer wishes to maintain the room at 0° F. and the cooling coils at -10° F. Assuming that the room is void of products, determine the percent relative humidity.

Because the room is empty, it can be assumed that the dew point of the air at 0° F. will be the same as the coil temperature, which is -10° F. From psychrometric tables (American Society of Refrigerating Engineers 1954), we find that P_1 is 0.0108 p.s.i. and P_2 is 0.0188 p.s.i. The relative humidity therefore is:

$$\frac{(0.0108)(100)}{0.0188} = 57.5 \text{ percent.}$$

If, however, the coil temperature were maintained at -5° F. instead of at -10° F., the dew point of the air would be -5° F. In this example, the relative humidity would be:

$$\frac{(0.0141)(100)}{0.0188} = 75 \text{ percent.}$$

The relative humidity in commercial freezers is 10 to 20 percent higher than are those indicated by the calculated figures above because of the evaporation of moisture from the product. In a freezer at 0° F. with a relative humidity of 75 percent and a pipe-coil temperature of -15° F., the moisture-vapor pressure within the package and in contact with the frozen fish would be 0.0188 p.s.i. The air in the freezer, however, would have a vapor pressure of 0.0141 p.s.i. and the moisture-vapor pressure at the coils would be 0.00825 p.s.i. These differences in vapor pressure result in migration of moisture from the air surrounding the product to the air of the room, thereby causing an increase in the humidity level. At the cold coils, the dew point drops, and the excess vapor condenses on the coils and deposits as frost.

A high relative humidity can be obtained by (1) adding moisture to the room, (2) increasing the cooling surface, or (3) employing the jacketed principle.

Adding Moisture to the Room

The relative humidity can be greatly increased by spraying steam into the atmosphere within the refrigerated room. This method, however, is not very suitable because of (1) the accumulation of ice on the freezer floor due to the difficulty in saturating the air with all the water vapor in the steam and (2) the excessive accumulation of frost on the cooling coils. Cooke (1939) reported that it was possible to obtain a relative humidity of 95 percent by adding moisture to the room in such a manner that it would not accumulate on the cooling coils. In this method, the steam was atomized and injected into the atmosphere so as almost to saturate the air completely. The problem of frost accumulation on the cooling coils was overcome by continually spraying an ethylene glycol solution over the coils to absorb any moisture condensed on them. The moisture absorbed was then removed by heating the ethylene glycol and was returned to the freezer in the form of steam. Recent tests on a system of this type indicate that a doubling of the power consumption is necessary to raise the relative humidity of the room from 60 to 100 percent.

Increasing Surface Area of the Coil

As was mentioned previously, the temperature of the cooling coil and of the room largely determines the relative humidity. The sample problem illustrated that a high humidity can be obtained by maintaining a minimum difference between the temperature of the cooling coil and that of the air. The method of determining the amount of cooling-coil surface required for a particular installation is described later in this section in "How to Calculate Cold-Storage Requirements."

Cold-storage plants constructed within recent years have greatly increased the amount of cooling-coil surface over the once widely used ratio of 1 linear foot of 2-inch pipe coil per 6 cubic feet of storage space. The increase in the surface area of the coil together with the utilization of special insulation materials and construction techniques to minimize air infiltration has made it possible to obtain very high humidities in the modern cold-storage warehouse.

Jacketed Principle

In the jacketed system, cold air circulates through an enclosed space or jacket that completely surrounds the room. The outside surface of the jacket is lined with a suitable insulation that is 8 to 10 inches thick. The inside surface consists of the materials that comprise the walls, floor, and ceiling of the storage room. Vertical and longitudinal wooden members in the jacket are arranged so as to provide ducts that distribute cold air in the proper amounts through various parts of the jacket. The jacket is a closed-duct system containing a set of finned-pipe or bare-pipe cooling coils to maintain the air at the proper temperature and a fan to provide the necessary air circulation through the ducts. The cold air circulating through these ducts maintains the inner storage room at a predetermined temperature by removing the heat that migrates through the insulated walls, ceiling, and floor before it can enter the refrigerated storage area.

With proper air circulation throughout the inner room, the difference between the temperature of the jacket walls and that of the room can be kept as small as 2° F. Owing to the small magnitude of this temperature differential, it is possible to maintain relative humidites as high as 98 percent in the storage space. Also, the air in the ducts does not enter the inner room; therefore, it cannot draw moisture away from the product. During operation, the moisture contained in the air within the duct is withdrawn by defrosting the cooling coils. Consequently, this air becomes very dry, resulting in very little frost accumulation on these coils.

A jacketed-type freezer was recently constructed at the Fish and Wildlife Service laboratory at East Boston, Massachusetts. Tests are presently being conducted to determine the effect of the jacketed principle on the storage life of fishery products and to gather engineering data in regard to the operation of the equipment.

Air Circulation

Rapid circulation of air within the cold-storage room will result in the withdrawal of moisture from the product. If a blower-type unit cooler is used to furnish the refrigeration effect, the air circulation must be kept at a minimum. This desired amount of circulation is accomplished by properly balancing the amount of cooling surface against the rate at which the air is moved by the fan. To minimize the drying effect due to the air circulation, a freezer employing a unit cooler with forced-air circulation must have a higher relative humidity than that required in a conventional still-air room.

In such a conventional still-air room, the temperature difference between the coil and the room determines the amount of circulation due to natural convection currents resulting from the difference in density between warm and cold air. Furthermore, an increase in this temperature difference causes a proportional increase in the moisture-vapor pressure differences, resulting in an additional increase in the natural circulation of air within the room. Conversely, a lowering of the temperature difference will result in a decrease in the differential between the moisture-vapor pressures, thereby causing slower movement of air. By proper proportioning of the amount of cooling surface area to the refrigerator load, the air circulation can thus be kept at a minimum.

CALCULATION OF THE COLD-STORAGE REQUIREMENTS OF FISH

In the calculation of the cold-storage requirements of fish, three of the basic considerations are (1) the rate of heat flow into the cold-storage room (heat-gain load), (2) the size of the evaporator needed to remove this heat, and (3) the size of the compressor needed to keep the system in operation.

The calculated heat-gain load of the cold-storage room determines the size of the compressor; and the size of the compressor selected, in turn, determines the size of the evaporator.

To enable the prospective builder to make a preliminary estimation of the cold-storage requirements for a particular service and thereby help him to discuss his needs intelligently with the refrigeration engineer or contractor upon whose recommendations he will ultimately rely, the remaining part of this section gives information on (1) how to calculate the heat-gain load in the cold-storage room, (2) how to calculate the size of the compressor, and (3) how to calculate the size of the evaporator. In addition, to show the interrelationships among these three sets of calculations, an overall illustrative problem is given.

HOW TO CALCULATE THE HEAT-GAIN LOAD IN THE COLD-STORAGE ROOM

In the calculation of the total heat-gain load (q') in the cold-storage room, the following four basic sources of heat flow into the room are considered: (1) wall-heat-gain load (q'_w), (2) air change or service load (q'_s), (3) product load (q'_p), and (4) miscellaneous load (q'_m). The total heat load is the sum of these four sources. Conventionally, this sum is increased by 10 percent to allow for a factor of safety.

In the first of the following subsections, the relationships between the various symbols representing heat (the quantity of heat and the rate of heat flow) are explained. In the remaining subsections, the four basic sources of heat flow are considered.

Relationships between Symbols Representing Quantity of Heat and Rate of Heat Flow

Symbols Employed

Inasmuch as the removal of heat is the basic problem treated in refrigeration, it is convenient to have symbols for the representation of heat. The following are the ones used in this leaflet:

Q = quantity of heat, in B.t.u.

q = rate of heat flow, in $\frac{\text{B.t.u.}}{\text{hr.}}$ or B.t.u./hr.

q' = rate of heat flow, in $\frac{\text{B.t.u.}}{\text{24 hr.}}$ or B.t.u./24 hr.

If "t" represents the time, in hours, the relationships between the symbols are as follows:

$$q = \frac{Q}{t} \qquad \text{in} \qquad \frac{\text{B.t.u.}}{\text{hr.}}$$

$$q' = 24q \; \frac{\text{hr.}}{24\text{ hr.}} \qquad \text{in} \qquad \frac{\text{B.t.u.}}{\cancel{\text{hr.}}} \times \frac{\cancel{\text{hr.}}}{24\text{ hr.}} = \frac{\text{B.t.u.}}{24\text{ hr.}}$$

$$= 24\,\frac{Q}{t}\;\frac{\text{hr.}}{24\text{ hr.}} \qquad \text{in} \qquad \frac{\text{B.t.u.}}{\cancel{\text{hr.}}} \times \frac{\cancel{\text{hr.}}}{24\text{ hr.}} = \frac{\text{B.t.u.}}{24\text{ hr.}}$$

Example of Typical Use

The following example shows a typical use of these symbols:

if Q = 32,000 B.t.u. (quantity of heat removed from product) and
t = 16 hr. (time needed to remove heat),

then $q = \frac{Q}{t} = \frac{32{,}000}{16}\,\frac{\text{B.t.u.}}{\text{hr.}} = 2{,}000\,\frac{\text{B.t.u.}}{\text{hr.}}$ and

$$q' = 24q\;\frac{\text{hr.}}{24\text{ hr.}} = 24\left(2{,}000\,\frac{\text{B.t.u.}}{\cancel{\text{hr.}}}\right)\frac{\cancel{\text{hr.}}}{24\text{ hr.}} = 48{,}000\,\frac{\text{B.t.u.}}{24\text{ hr.}} \quad \text{or}$$

$$= 24\,\frac{Q}{t}\;\frac{\text{hr.}}{24\text{ hr.}} = 24\left(\frac{32{,}000}{16}\,\frac{\text{B.t.u.}}{\cancel{\text{hr.}}}\right)\frac{\cancel{\text{hr.}}}{24\text{ hr.}} = 48{,}000\,\frac{\text{B.t.u.}}{24\text{ hr.}}$$

In the discussions that follow, the details of these conversions will be omitted to save space.

Wall-Heat-Gain Load

Basic Heat-Flow Equation

The heat entering the cold-storage room through the walls, ceiling, and floor depends upon (1) the type and thickness of the insulation and of the other construction materials, (2) the outside surface area of the cold-storage room, and (3) the difference between the temperature of the air inside the cold-storage room and that of the air outside. These factors may be combined into a basic heat-flow equation:

$$q = \frac{kA}{x}(T_2 - T_1) \qquad \text{where,}$$

q = rate of heat leakage into the cold region, in B.t.u. per hour.

A = surface area, in square feet (based on outside dimensions).

T_2 = temperature on the warm side (outside air temperature), in °F.

T_1 = temperature on the cold side (refrigerator air temperature, in °F.

x = thickness, in inches, of material bounding the cold region.

k = conductivity constant of material bounding cold region, in the units 2/ $\frac{(B.t.u./hr.)}{(ft.^2)(°F./in.)}$

Conductivity Constant (k)

The significance of the conductivity constant k may be more readily understood from figure 14. This constant is defined as the rate of heat flow, in B.t.u. per hour, through a 1-square-foot surface of 1-inch thickness with a 1° F. temperature difference between the two sides of the surface.

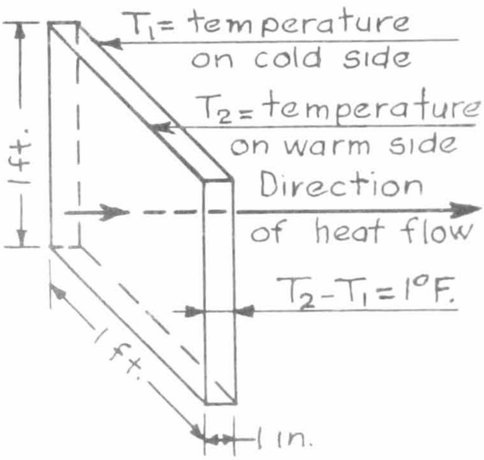

Figure 14.--Standard conditions for determination of conductivity constant.

If:

$$A = 1 \text{ sq. ft.}$$
$$T_2 - T_1 = 1° \text{ F.}$$
$$x = 1 \text{ in.}$$
$$(\text{time} = 1 \text{ hr.})$$

Then:

$$q = \frac{kA}{x}(T_2 - T_1)$$
$$= \frac{k(1)(1)}{(1)}, \quad \text{or}$$
$$q = k \text{ (numerically)}$$

Data on k factors for some common industrial materials used for insulation and construction are listed in table 1. Such values are usually given in "per inch of thickness" and are expressed in the units:

$$\frac{(B.t.u/hr.)}{(ft.^2)(°F./in.)}$$

but occasionally they will be given in "per foot of thickness" and will be expressed in the units:

$$\frac{(B.t.u./hr.)}{(ft.^2)(°F./ft.)} \quad \text{or, simplifying} \quad \frac{(B.t.u./hr.)}{(ft.)(°F.)}$$

To convert either of the latter two k values to the former one, multiply either of these values by 12.

2/ To those not accustomed to working with refrigeration, units such as this one may seem to be somewhat complex. The careful use of these units, however, clarifies the problems and aids in the avoidance of errors. The method of handling the units will be shown in the numerous illustrative problems that follow.

Table 1.—Thermal conductivities of some common insulating materials

Description	Density	Mean temp.	Conductivity coefficients k	Conductivity coefficients U
	Lb. per ft.^{3}	°F.	1/	2/
Air space, 1-in. thick bounded by ordinary materials		60		1.07
		30		0.98
		0		0.89
Air space, 1-in. thick bounded on warm side by aluminum paint		60		0.62
		30		0.59
		0		0.57
Air space, 1-in. thick bounded on warm side by aluminum foil		60		0.46
		30		0.46
		0		0.45
Aluminum foil, crumpled	0.2		0.28	
Aluminum foil, spaced	2.4		0.22	
Asbestos, packed	43.8		1.52	
Asbestos, loose	29.3		0.94	
Asbestos, corrugated	16.2		0.52	
Asphalt	132	68	5.2	
Concrete	150		12.0	
Corkboard	7.07	0	0.27	
	7.07	-14	0.23	
	10.5		0.28	
Felt—wool	20.6	86	0.38	
Fiberglas	3.0		0.24	
Fir, Douglas, 0% moisture	34	75	0.67	
Glass wool	1.65	60	0.27	
	1.65	30	0.25	
	1.65	0	0.23	
Mineral wool	3.5	60	0.27	
	3.5	30	0.24	
	3.5	0	0.22	
Pine, Oregon	37.0		0.80	
Pine, white	31.2	86	0.78	
Pine, yellow			1.00	
Rock wool	6.0		0.26	
	10.0	90	0.27	
	18.0		0.29	
Rubber board, expanded	4.5	60	0.22	
	4.5	30	0.22	
	4.5	0	0.21	
Wood fiberboard, 3/4-in. thick (moisture as received)	17.0	72		0.33

1/ $\frac{(\text{B.t.u./hr.})}{(\text{ft.}^2)(\text{°F./in.})}$ 2/ $\frac{(\text{B.t.u./hr.})}{(\text{ft.}^2)(\text{°F.})}$

Calculation for Single-Wall Construction

Calculation by use of basic heat-flow equation.—The following problem shows how to calculate the wall-heat-gain load by use of the basic heat-flow equation. In this problem, only the insulation material is considered. For walls made up of thick insulation, the small error that results from neglecting the other wall-construction materials is on the safe side.

Problem 1a

Given:
1. 50 ft. x 50 ft. x 10 ft. outside dimensions of cold-storage room.
2. 6 in. corkboard insulation (walls, ceiling, floor) of density 10.5 lb./ ft.^{3}
3. 0° F. air temperature inside refrigerator.
4. 95° F. air temperature outside refrigerator.

To find: The wall-heat-gain load in B.t.u./24 hr. (q'_w).

Solution:

$$q = \frac{kA}{x}(T_2 - T_1)$$

$$= \frac{(0.28)(7,000)(95)}{6}$$

$$= 31,000 \text{ B.t.u./hr.}$$

$$q'_w = (24)(31,000)$$

$$= 744,000 \text{ B.t.u./24 hr.}$$

$k = 0.28 \dfrac{(\text{B.t.u./hr.})}{(\text{ft.}^2)(\text{°F./in.})}$ (from table 1).

$A = 50 \times 50 \times 2 + 50 \times 10 \times 4$

$= 7,000 \text{ ft.}^2$

$T_2 = 95° \text{ F.}$

$T_1 = 0° \text{ F.}$

$T_2 - T_1 = 95° \text{ F.}$

$x = 6 \text{ in.}$

Calculation by use of table.--Instead of using the basic-heat-flow formula, the designer can estimate the wall-heat-gain load by using table 2. The heat leakages through different thicknesses of cork or cork-equivalent insulation are listed in table 2 for various temperature differences (outside temperature minus cold-storage-room temperature). The use of this table may be more readily understood by repeating the calculation for the conditions given in problem 1a:

Problem 1b

Given:
1. Outside surface area = 7,000 ft.^{2}
2. Cork insulation = 6 in.
3. Temperature difference = 95° F.

To find: The wall-heat-gain load in B.t.u./24 hr. (q'_w).

Table 2.--Wall-heat gain for various temperature differences and thicknesses of cork or equivalent insulation (based on thermal conductivity, k = 0.30) 1/

Insulation (cork or cork equivalent 2/)	Wall-heat gain for temperature differences (outside design, or ambient, temperature minus refrigerator temperature in °F.) of:																	
	1°	40°	45°	50°	55°	60°	65°	70°	75°	80°	85°	90°	95°	100°	105°	110°	115°	120°
Inches	B.t.u. per 24 hr. per sq. ft.																	
1	7.3	292	329	365	402	438	475	511	548	584	621	657	694	730	767	803	840	876
2	3.6	144	162	180	198	216	234	252	270	288	306	324	342	360	378	396	414	432
3	2.4	96	108	120	132	144	156	168	180	192	204	216	228	240	252	264	276	288
4	1.8	72	81	90	99	108	117	126	135	144	153	162	171	180	189	198	207	216
5	1.44	58	65	72	79	86	94	101	108	115	122	130	137	144	151	158	166	173
6	1.2	48	54	60	66	72	78	84	90	96	102	108	114	120	126	132	138	144
7	1.03	41	46	52	57	62	67	72	77	82	88	93	98	103	108	113	118	124
8	.90	36	41	45	50	54	59	63	68	72	77	81	86	90	95	99	104	108
9	.80	32	36	40	44	48	52	56	60	64	68	72	76	80	84	88	92	96
10	.72	29	32	36	40	43	47	50	54	58	61	65	68	72	76	79	83	86
11	.66	26	30	33	36	40	43	46	50	53	56	59	63	66	69	73	76	79
12	.60	24	27	30	33	36	39	42	45	48	51	54	57	60	63	66	69	72
13	0.55	22	25	28	30	33	36	39	41	44	47	50	52	55	58	61	63	66
14	0.51	20	23	26	28	31	33	36	38	41	43	46	49	51	54	56	59	61

1/ Method of using values in table 2 to convert to wall-heat-gain load:
Table 2 value (B.t.u./24 hr./sq.ft.) x outside surface area of cold-storage room (sq. ft.) = total wall-heat-gain load (B.t.u./24 hr.).

2/ The equivalent thickness of an insulation material in terms of cork is obtained by multiplying the ratio, $\frac{\text{thermal conductivity of material}}{\text{thermal conductivity of cork}} = \frac{k_m}{k_c}$, by the thickness of cork in this table. Thus, the equivalent thickness of expanded rubber board (k = 0.21; table 1) in terms of 6 inches of corkboard (k = 0.28; table 1) for 0° F. usage is: $\frac{k_m}{k_c} \times 6$ in. $= \frac{0.21}{0.28} \times 6 = 4.5$ in. The cork equivalents of a series of materials are additive.

Solution: For values of 6 in. and 95° F., enter table 2 and obtain $\frac{q'_w}{ft.^2} = 114 \frac{(B.t.u.)}{(24\ hr.)(ft.^2)}$. This value can be converted to B.t.u./24 hr. by the following calculation:

$$q'_w = 114 \frac{(B.t.u.)}{(24\ hr.)(ft.^2)} \times 7{,}000\ ft.^2$$

$$= 798{,}000\ B.t.u./24\ hr.$$ 3/

Calculation for Multiple-Wall Construction

Generally, insulated walls are made up of two or more materials; and the resistance to the flow of heat through the main insulating material is usually so much greater than that through the other materials that the flow of heat through these other materials may ordinarily be neglected, as was done in problems 1a and 1b. When, however, it is desirable to evaluate the additional insulating effect of the other materials, the basic heat-flow equation $q = \frac{kA}{x}(T_2 - T_1)$ must be modified. The term $\frac{k}{x}$ instead of representing a single material must now represent the series of materials being considered. This overall term is called the coefficient of transmittance or overall coefficient of heat transfer and is commonly designated by the symbol U.

The heat-flow equation for a multiple-wall material may thus be expressed in the form:

$$q = UA(T_2 - T_1)$$

where,

$$U = \frac{1}{\frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \dots}, \text{ in } \frac{(B.t.u./hr.)}{(ft.^2)(°F.)}$$

x = thickness of each layer of material, in in.
k = conductivity of each layer of material, in $\frac{(B.t.u./hr.)}{(ft.^2)(°F./in.)}$

q = rate of heat leakage into cold region, in B.t.u./hr.
A = surface area, in sq. ft. (based on outside dimensions).
T_2 = temperature on the warm side (outside air temperature), in °F.
T_1 = temperature on the cold side (refrigerator air temperature), in °F.

3/ This value (798,000) is slightly higher than that obtained in problem 1a (744,000), because table 2 is based upon a conductivity of k = 0.30. If a k value of 0.28 more accurately reflects the particular insulation used, 798,000 can be multiplied by 0.28/0.30 to obtain 745,000, which is in close agreement with 744,000.

Calculation neglecting air films.—The following problem shows how to calculate the wall-heat-gain load for a wall of multiple construction, neglecting air films:

Problem 2

Given:
1. 50 ft. x 50 ft. x 10 ft. cold-storage room
2. 0° F. inside temperature.
3. 95° F. outside temperature.
4. Wall construction as shown in figure 15.

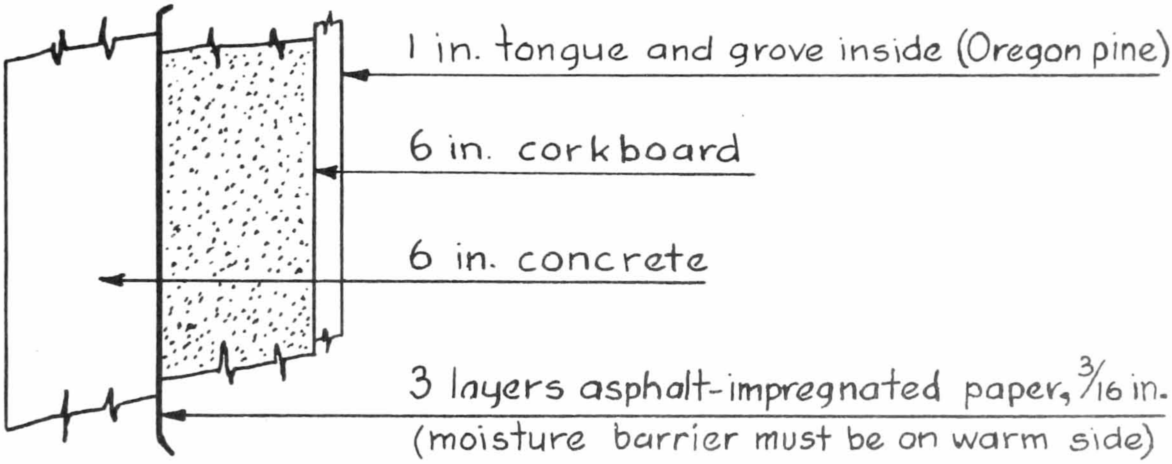

Figure 15.--Cross section of wall of cold-storage room.

To find: The wall-heat-gain load in B.t.u./24 hr. (q'_w).

Solution:

$$U = \frac{1}{\frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \frac{x_4}{k_4}}$$

$$= \frac{1}{\frac{6}{12} + \frac{3/16}{5.2} + \frac{6}{0.28} + \frac{1}{0.80}}$$

$$= \frac{1}{0.50 + 0.04 + 21.43 + 1.25}$$

$$= \frac{1}{23.22}$$

$$= 0.0431 \frac{(\text{B.t.u./hr.})}{(\text{ft.}^2)(^\circ\text{F.})}$$

$k_1 = 12 \frac{(\text{B.t.u./hr.})}{(\text{ft.}^2)(^\circ\text{F./in.})}$

$k_2 = 5.2$ "

$k_3 = 0.28$ "

$k_4 = 0.80$ "

(all from table 1).

$x_1 = 6$ in.

$x_2 = 3/16$ in.

$x_3 = 6$ in.

$x_4 = 1$ in.

$$q = UA(T_2 - T_1)$$

$$= (0.0431)(7{,}000)(95)$$

$$= 28{,}700 \text{ B.t.u./hr.}$$

$A = 7{,}000$ sq. ft. (from problem 1).

$$q'_w = 24 \times 28{,}700$$

$$= 689{,}000 \text{ B.t.u./24 hr.}$$

$T_2 - T_1 = 95°$ F.

The calculated wall-heat-gain load in problem 2 is 7.4 percent less than that in problems 1a and 1b, owing to the insulating effect in problem 2 of the concrete, asphalt vapor barrier, and Oregon pine. For small installations, such as this one, it is common practice in the calculation of the wall-heat-gain load to ignore the effect of the building materials other than the insulation. This practice provides an extra factor of safety. For large installations, however, the insulating effect of all the building materials must be considered because of the large surface area involved. Failure to include the insulating effect of these other materials would result in the purchase of unnecessary refrigeration equipment.

The insulating effect of air films has been neglected in problem 2 because this effect is usually insignificant in the case of small installations with well-insulated walls. If, however, the effects of air films are to be included in the calculation, as in that for a large cold storage, the symbol U in the basic heat flow equation $q = UA(T_2 - T_1)$ takes the form:

$$U = \frac{1}{\frac{1}{f_o} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + \ldots \frac{1}{f_i}}$$ where,

x = thickness of each layer of material in in.

k = thermal conductivity of each layer of material.

f_o = factor for outside air film = 6.5. This value of 6.5 is for an outside surface exposed to the weather and is the estimated insulation effect of a surface air film with a 15-mile-per-hour wind blowing. If this outside surface is on a refrigerator located inside of a building or in an otherwise protected location where the wind does not blow upon it, then $f_o = 1.65$.

f_i = factor for inside air film = 1.65. This value is for still air and is the estimated insulation effect of the stationary air film that clings to the surface.

<u>Calculation including air films</u>.--The following problem shows how to include the effects of air films in calculating the wall-heat-gain load for a wall of multiple construction:

<u>Problem 3</u>

Given: The same data as those in problem 2 except that the air films on the outside and inside walls are included.

To find: The wall-heat-gain load in B.t.u./24 hr. (q'_w).

$$\text{Solution: } U = \cfrac{1}{\cfrac{1}{f_o} + \cfrac{x_1}{k_1} + \cfrac{x_2}{k_2} + \cfrac{x_3}{k_3} + \cfrac{x_4}{k_4} + \cfrac{1}{f_i}}$$

$$= \cfrac{1}{\cfrac{1}{6.5} + 0.50 + 0.04 + 21.43 + 1.25 + \cfrac{1}{1.65}}$$

$$= \frac{1}{23.98}$$

$$= 0.0417 \frac{(\text{B.t.u./hr.})}{(\text{ft.}^2)(^\circ\text{F.})}$$

$$q = (0.0417)(7{,}000)(95)$$
$$= 27{,}700 \text{ B.t.u./hr.}$$

$$q'_w = 24 \times 27{,}700$$
$$= 665{,}000 \text{ B.t.u./24 hr.}$$

The calculated wall-heat-gain load in problem 3 is 10.6 percent less than that in problems 1a and 1b, owing to the insulating effect of the construction materials plus the air films on the outside and inside walls.

Inasmuch as the wall-heat-gain is only one part of the total heat load, it is customary to omit the insulating effect of the air films and of the materials other than the main insulation, especially if the installation is a small one with insulated walls of the recommended thickness.

Thickness of Insulation

To determine the proper thickness of insulation, the designer must consider such factors as the type of insulation and its cost, the type of construction, the temperature differentials, and the operating costs of the refrigeration equipment. There is an optimum point where the fixed charges for insulation, construction, and equipment are in approximate balance with operating costs, and the total costs are close to a minimum.

Economical thicknesses have been worked out for the range of temperature differentials commonly encountered. A list of these economical thicknesses is given in table 3. It is not always practical, however, to employ the recommended thickness. Domestic freezer boxes, for example, must be able to go through a 2-foot 8-inch door; and commercial boxes, through a 3-foot door. To install the recommended insulation thickness in such boxes would make the usable space too small. A compromise must therefore be made between the standard of insulation and the limitations of space. Some of the newer synthetic insulations with low k factors are helping to meet these needs.

Table 3.--Recommended minimum thicknesses of insulation for refrigerators located in the northern or southern part of the United States, based on standards of the refrigeration industry

Temperature of refrigerator	Cork or equivalent thickness 1/	
	In northern U. S.	In southern U. S.
°F.	Inches	Inches
50 to 60	2	3
40 to 50	3	4
25 to 40	4	5
15 to 25	5	6
0 to 15	6	7
-15 to 0	7	8
-40 to -15	9	10

1/ Method of calculating cork-equivalent thickness is given in footnote 2 of table 2.

The choice of insulation thickness is sometimes closely related to the fabrication costs or to the special conditions of installation. For example, in insulating the hold of a small boat equipped with steel tanks, stiffened with 2-inch angles, for holding sea water at 30° F., the maximum insulation would probably be 2 inch (standard calls for 4 inch; table 3). To go to a thicker insulation, however, might unduly increase the fabrication costs. Furthermore, since by far the largest portion of the refrigeration costs go into precooling the sea water and cooling the incoming fish and since this type of installation is operated only during a small fraction of the year, it would be advisable to keep fixed capital costs of insulation to a minimum.

Outside Design Temperature

The "outside design temperature" is the basic temperature from which heat loads are calculated. It is defined as the temperature, in the locality of the refrigerator, that is not exceeded more than a given percentage of the time. An average temperature is not used because on the hottest days--when the refrigeration is most needed--the equipment would be too small. On the other hand, the maximum recorded temperature is not used because the total effect of this higher-than-design temperature does not manifest itself immediately. Thus, before this higher temperature begins seriously to affect the load on the refrigeration equipment, it ordinarily begins to drop off, owing to a change in conditions, such as the setting of the sun. Inasmuch as the equipment is selected with an adequate factor of safety, it is reasonable to work on the basis of less than the worst temperature conditions.

Determination of outside design temperature.—Outside design temperatures for a given locality can be determined from data obtained from the local United States Weather Bureau office, from the local newspaper offic or from some standard reference such as the American Society of Refrigerating Engineers Data Book, Design Volume.

Corrections to outside design temperature.—If the refrigerator is exposed to direct sunlight, additional heat must be added to the heat load. This correction can conveniently be made by adding to the design temperature the proper values found in table 4 for the walls and roof. The design temperature for floors that are set directly on the ground without a ventilation space underneath should be lowered by 20° F. (the ground temperature is considered to be 20° F. below the outside air temperature).

Table 4.—Correction to outside design temperature for solar radiation

Type of surface	Amount[1/] to be added to design temperature for:			
	East wall	South wall	West wall	Flat roof
	°F.	°F.	°F.	°F.
Dark-colored surfaces, such as slate roofing, tar roofing, and black paints	8	5	8	20
Medium-colored surfaces, such as unpainted wood, brick, red tile, dark cement, and paints (red, gray, green)	6	4	6	15
Light-colored surfaces, such as white stone, light-colored cement, and white paint	4	2	4	9

1/ These values are applicable for the full 24-hour day.

If walls are common with other storage areas, the design temperature should be appropriately modified.

Air Change or Service Load

The air change or service load accounts for the flow of heat into the refrigerator caused by door openings and other similar leakages. The effect of door openings and other leakage sources is difficult to estimate accuratel number of different methods of estimation have consequently been developed.

Service Load Estimated on Basis of Wall-Heat-Gain Load

(1) A common method used by some designers is to estimate the air change or service load as 15 percent of the wall-heat-gain load. Under some conditions, this estimate will be too low.

(2) Other designers use an estimate of 20 percent for light service, 33-1/3 percent for normal or average service, and 50 percent for heavy service.

Service Load Estimated on Basis of Volume of Refrigerator

Still other designers reason that the number of door openings and resulting air changes is related to the volume of the refrigerator. Data on air changes per 24 hours due to door openings and air infiltration are given in table 5. Experience has shown that this table is practical.

Table 5.—Data for estimation of air change or service load: average[1/] air changes per 24 hours due to door openings and air infiltration for cold-storage rooms of various volumes

Net volume of room	Average air changes per 24 hr.	Net volume of room	Average air changes per 24 hr.	Net volume of room	Average air changes per 24 hr.
Cubic feet	Number	Cubic feet	Number	Cubic feet	Number
200	44.0	2,000	12.0	20,000	3.5
300	34.5	3.000	9.5	25,000	3.0
400	29.5	4,000	8.2	30,000	2.7
500	26.0	5,000	7.2	40,000	2.3
600	23.0	6,000	6.5	50,000	2.0
800	20.0	8,000	5.5	75,000	1.6
1,000	17.5	10,000	4.9	100,000	1.4
1,500	14.0	15,000	3.9		

1/ For heavy usage, multiply these values by 2. For light usage or long storage, multiply the values by 0.6.

The heat given up to the refrigerator for each cubic foot of air taken from outside conditions to refrigerator conditions is given in table 6.

The air change or service load in B.t.u./24 hr.(q'_s) is calculated from tables 5 and 6 by use of the equation:

$$q'_s = VNQ_{cft}$$ where,

V = volume of air per air change (usually taken as the inside volume of the refrigerator), in ft.^{3}/air change.

N = number of air changes per 24 hours, in air changes/24 hr.

Q_{cft} = heat extracted in cooling outside air to temperature of the refrigerator, in B.t.u./ft.^{3}

Table 6.--Data[1/] for estimation of air change or service load: heat given up by outside air in cooling to cold-storage temperatures

Temperature of cold-storage room	Heat given up by outside air at temperatures and relative humidities of:							
	85° F.		90° F.		95° F.		100° F.	
	50%	60%	50%	60%	50%	60%	50%	60%
°F.	B.t.u. per cu. ft.							
50	1.32	1.54	1.62	1.87	1.93	2.22	2.28	2.65
45	1.50	1.73	1.80	2.06	2.12	2.42	2.47	2.85
40	1.69	1.92	2.00	2.26	2.31	2.62	2.67	3.06
35	1.86	2.09	2.17	2.43	2.49	2.79	2.85	3.24
30	2.00	2.24	2.26	2.53	2.64	2.94	2.95	3.35
	40° F.		50° F.		90° F.		100° F.	
	70%	80%	70%	80%	50%	60%	50%	60%
	B.t.u. per cu. ft.							
30	0.24	0.29	0.58	0.66	2.26	2.53	2.95	3.35
25	0.41	0.45	0.75	0.83	2.44	2.71	3.14	3.54
20	0.56	0.61	0.91	0.99	2.62	2.90	3.33	3.73
15	0.71	0.75	1.06	1.14	2.80	3.07	3.51	3.92
10	0.85	0.89	1.19	1.27	2.93	3.20	3.64	4.04
5	0.98	1.03	1.34	1.42	3.12	3.40	3.84	4.27
0	1.12	1.17	1.48	1.56	3.28	3.56	4.01	4.43
- 5	1.23	1.28	1.59	1.67	3.41	3.69	4.15	4.57
-10	1.35	1.41	1.73	1.81	3.56	3.85	4.31	4.74
-15	1.50	1.53	1.85	1.92	3.67	3.96	4.42	4.86
-20	1.63	1.68	2.01	2.09	3.88	4.18	4.66	5.10
-25	1.77	1.80	2.12	2.21	4.00	4.30	4.78	5.21
-30	1.90	1.95	2.29	2.38	4.21	4.51	4.90	5.44

1/ These data are used in conjunction with those in table 5.

The following problem illustrates how tables 5 and 6 and this equation are employed:

Problem 4

Given: 1. 50 ft. x 50 ft. x 10 ft. cold-storage room.
2. 6 in. cork insulation; 12 in. overall wall thickness.
3. 90° F. outside temperature.
4. 60% relative humidity.
5. 0° F. temperature of cold-storage room.
6. 1 door in room.

To find: Service load in B.t.u./24 hr. (q'_s).

Solution:

$$V = 48 \times 48 \times 8 = 18{,}400 \text{ ft.}^3$$

N = 3.63 air changes per 24 hr. (obtained from table 5 by interpolating for 18,400 ft.^{3}).

Q_{cft} = 3.56 B.t.u./ft.^{3} (obtained from table 6 for heat extracted per ft.^{3} to cool air from 90° F. and 60% relative humidity to 0° F.).

$$q'_s = VNQ_{cft} = (18{,}400)(3.63)(3.56) = 238{,}000 \text{ B.t.u./24 hr.}$$

Product Load

A product placed in a cold storage at a temperature above that of the storage will give up heat until it reaches the storage temperature. The amount of heat given up by the product can be calculated from (1) the weight of product, (2) the initial temperature, (3) the specific heat of product above freezing, (4) the freezing temperature of the product, (5) the latent heat of the product, (6) the specific heat of the product below freezing, and (7) the final temperature.

Formulae for Calculation

Quantity of heat removed (Q).—When a given weight of product is cooled from one temperature to a lower one, some or all of the calculations outlined below must be made:

(a) In cooling the product from the initial temperature (T_1) to a lower temperature (T_2) above freezing, such as in a chill room, the heat removed in B.t.u. (Q_{12}) is given by the formula:

$$Q_{12} = Wc(T_1 - T_2)$$

(b) In cooling the product from the lower temperature (T_2) above freezing to the freezing temperature (T_f)[4], the heat removed

[4] For calculation purposes, the freezing point of fishery products is usually assumed to be 28° F.

in B.t.u. (Q_{2f}) is given by the formula:

$$Q_{2f} = Wc(T_2 - T_f)$$

(c) In freezing the product at the freezing temperature (T_f), the heat removed in B.t.u. (Q_f) is given by the formula:

$$Q_f = WL$$

(d) In cooling the product from the freezing temperature (T_f) to the storage temperature (T_3), the heat removed in B.t.u. (Q_{f3}) is given by the formula:

$$Q_{f3} = Wc_i(T_f - T_3)$$

(e) In cooling the product from the initial temperature (T_1) directly to the storage temperature (T_3), the total heat removed in B.t.u. (Q) is given by the formula:

$$Q = Q_{12} + Q_{2f} + Q_f + Q_{f3} = W[c(T_1 - T_f) + L + c_i(T_f - T_3)]$$

where,

- W = weight of product, in pounds.
- c = specific heat of fishery product above freezing, in B.t.u./(lb.)(°F.)
- c_i = specific heat of fishery product below freezing, in B.t.u./(lb.)(°F.)
- L = latent heat of fusion of fishery product, in B.t.u./lb.

Product load (q'_p).—To convert the total amount of heat (Q) removed in the time "t" to the product load (q'_p), divide "Q" by "t" and multiply by 24, as was shown in "Relationship Between Symbols Representing Quantity of Heat and Rate of Heat Flow."

Sample calculation.—The following problem shows how to determine the quantity of heat removed (Q) and the product load (q'_p).

Problem 5a

Given:
1. One ton of fish at 50° F. to be frozen to 0° F. every 24 hours by (a) prechilling to 40° F. in a chill room and (b) then freezing on shelf plates at -10° F.
2. c = 0.8; c_i = 0.4; and L = 115.

To find:
(1) The quantity of heat removed, in B.t.u. (Q).
(2) The product load in B.t.u./24 hr. (q'_p).

Solution: To cool from 50° F. to 40° F. in chill room:

$$\begin{aligned} Q_{12} &= Wc(T_1 - T_2) \\ &= (2{,}000)(0.8)(50-40) \\ &= 16{,}000 \text{ B.t.u.} \end{aligned}$$

To cool from 40° F. to freezing temperature:

$$\begin{aligned} Q_{2f} &= Wc(T_2 - T_f) \\ &= (2{,}000)(0.8)(40-28) \\ &= 19{,}200 \text{ B.t.u.} \end{aligned}$$

To freeze at 28° F.:

$$\begin{aligned} Q_f &= WL \\ &= (2{,}000)(115) \\ &= 230{,}000 \text{ B.t.u.} \end{aligned}$$

To cool from freezing point to storage temperature:

$$\begin{aligned} Q_{f3} &= Wc_i(T_f - T_3) \\ &= (2{,}000)(0.4)(28-0) \\ &= 22{,}400 \text{ B.t.u.} \end{aligned}$$

To cool from 50° F. to 0° F. (total):

$$\begin{aligned} Q &= Q_{12} + Q_{2f} + Q_f + Q_{f3} \\ &= 16{,}000 + 19{,}200 + 230{,}000 + 22{,}400 \\ &= 288{,}000 \text{ B.t.u.} \end{aligned}$$

To convert to a B.t.u./hr. basis:

$$\begin{aligned} q &= \frac{Q}{t} \\ &= \frac{288{,}000}{24} \frac{\text{B.t.u.}}{\text{hr.}} \end{aligned}$$

To convert to a B.t.u./24 hr. basis:

$$\begin{aligned} q'_p &= 24\left(\frac{288{,}000}{24}\right) \frac{\text{B.t.u.}}{24 \text{ hr.}} \\ &= 288{,}000 \frac{\text{B.t.u.}}{24 \text{ hr.}} \end{aligned}$$

Specific Heats and Latent Heat of Fusion

The specific heats and the latent heat of fusion of a fishery product depends, among other factors, upon the relative amounts of moisture, oil, and solids in the product and are difficult to determine precisely. For most fishery products, however, the following approximations will be found satisfactory:

c = specific heat above freezing = 0.8 B.t.u./(lb.)(°F.)
c_i = specific heat below freezing = 0.4 B.t.u./(lb.)(°F.)
L = latent heat of fusion = 115 B.t.u./lb.

The latent heat of fusion, in problem 5a, constitutes about 80 percent of the total product load $\left(\frac{230,000 \times 100}{288,000}\right)$. In product freezing calculations, such as this one, the latent heat of fusion will always constitute a high percentage of the product load.

Rate of Cooling or of Freezing

If problem 5a had been such that 1 ton of fish at 50° F. was to be frozen to 0° F. every 16 hours instead of every 24 hours, the calculations for the product load (q'_p) would have been as follows:

Problem 5b

Given: One ton of fish at 50° F. to be frozen to 0° F. every 16 hours by (a) prechilling to 40° F. in a chill room and (b) then freezing on shelf plates at -10° F.

To find: Product load in B.t.u./24 hr. (q'_p).

Solution: Q = 288,000 B.t.u. (the same as in problem 5a).
t = 16 hr.

$$q = \frac{Q}{t}$$
$$= \frac{288,000}{16} \text{ B.t.u./hr.}$$

$$q'_p = 24\left(\frac{288,000}{16}\right)$$
$$= 432,000 \text{ B.t.u./24 hr.}$$

Note that the product load in problem 5b is 50 percent larger than that in problem 5a. Thus close estimation of "t," the time required to cool or to freeze the product load, is essential.

The time "t" is determined by many factors—such as thickness of the product, type and thickness of the packaging material, freezing temperature employed, and method of freezing. In this connection, it is important to note that the freezing time is not necessarily decreased by an increase in refrigerator capacity. (This subject is discussed further in section 3.)

Miscellaneous Load

All of the energy dissipated in the refrigerator must be included in the heat load. This energy comes originally from electric motors, electric lights, and men working in the area.

Electric Motors

Heat-load equivalent.—The useful energy output of the motor (h.p. rating) and the motor losses (friction and resistance) are given in table 7.

Table 7.—Heat equivalent of electric motors 1/ under various conditions

Motor size	Heat equivalent under conditions of:		
	1 Load inside refrigerator box and motor outside 2/	2 Motor inside refrigerator box and load outside 3/	3 Motor and load inside refrigerator box 4/
H.p.	B.t.u./hr./h.p.	B.t.u./hr./h.p.	B.t.u./hr./h.p.
1/8 to 1/2	2545	1700	4250
1/2 to 3	2545	1150	3700
3 to 20	2545	400	2950

1/ The heat equivalent of electric motors is made up of two components: column 1, the useful horsepower output of the motor, and column 2, the frictional and electrical losses of the motor. Column 3 is the sum of these two components.

2/ Use values in column 1 if the driving motor is outside the refrigerator and the load is inside the box: for example, if a pump motor outside the box is circulating brine or chilled water within the box.

3/ Use values in column 2 if the driving motor is inside the refrigerator box and the load is outside the box: for example, if a motor within the box is driving a pump or fan outside the box or in another space.

4/ Use values in column 3 if the driving motor and its load are in the refrigerator box: for example, if a motor is driving the fan of a forced-circulation unit cooler.

Sample calculation.—The following example shows how to calculate the heat load due to a combination electric motor and fan located in the refrigerated storage area.

Problem 6a

Given: 1/3 h.p. motor driving fan for forced circulation of air through a unit cooler.

To find: Heat load in B.t.u./24 hr. (q'_{em}) due to motor and fan in refrigerated area.

Solution: 1 h.p. = 4,250 B.t.u./hr. (from table 7)
1/3 h.p. = 1,417 B.t.u./hr.

$$q'_{em} = (1,417)(24)$$
$$= 34,000 \text{ B.t.u./24 hr.}$$

Electric Lights

Heat-load equivalent.—1 watt = 3.42 B.t.u./hr.

Sample calculation.—The following example shows how to calculate the heat load due to electric lights:

Problem 6b

Given: Two 100-watt lights in use from 8:00 a.m. to 4:00 p.m. in a cold-storage room.

To find: Heat load in B.t.u./24 hr. (q'_{el}) due to lights.

Solution: 1 watt = 3.42 B.t.u./hr.
200 watts = 200 x 3.42 = 684 B.t.u./hr.

$$q'_{el} = (684)(24)$$
$$= 16,400 \text{ B.t.u./24 hr.}$$

Occupancy by People

Heat-load equivalent.—Table 8 gives the average hourly load due to occupancy at different temperatures of the cold-storage room. At best, however, this source of heat load is difficult to estimate accurately. People give up heat at varying rates, depending on the temperature of the working area, the type of clothing worn, the size of man, and the exertion put forth in doing the work. People going into the refrigerated area for short duration carry with them heat substantially above that given in table 8. If traffic of this type is heavy, additional allowances must therefore be made.

Table 8.--Heat equivalent per person working in cold-storage room

Temperature of cold-storage room	Heat equivalent 1/
°F.	B.t.u. per hr. per person
50	720
40	840
30	950
20	1,050
10	1,200
0	1,300
-10	1,400

1/ These values are applicable for the full 24-hour day.

Sample calculation.--The following example shows how to calculate the heat load due to men working in the refrigerator.

Problem 6c

Given: Two men are stacking a 0° F. storage area with fish for 6 hours per day.

To find: Occupancy load in B.t.u./24 hr. (q'_o) for the two men.

Solution: For one man working at 0° F.:

$$q = 1{,}300 \text{ B.t.u./hr. (from table 8)}$$

For two men working at 0° F.:

$$q = 2 \times 1{,}300$$
$$= 2{,}600 \text{ B.t.u./hr.}$$

$$q'_o = (24)(2{,}600)$$
$$= 62{,}000 \text{ B.t.u./24 hr.}$$

Note that if the men worked more than 6 hours--say 16 or 20 hours--the occupancy load would still be the same because the rate of 1,300 B.t.u. per hour given in table 8 is figured for the full 24-hour day. Although in the above problem the load lasts for only 6 hours, sufficient refrigeration capacity must be provided for the full 24 hours to handle this maximum demand even though it means having a surplus of refrigeration capacity during the period when there is no heat load due to occupancy. This same principle applies in calculating the heat load for lights and motors in the cold-storage area.

Total Miscellaneous Load

The total miscellaneous load is equal to the sum of the loads due to electric motors, electric lights, and men working in the area:

$$q'_m = q'_{em} + q'_{el} + q'_o$$

Total Heat Load

Method of Calculation

The total heat load (q') is calculated as the sum of the four basic sources of heat flow plus 10 percent (for a safety factor):

$$q = 1.10(q'_w + q'_s + q'_p + q'_m)$$

With hypothetical data being assumed for each of the four basic sources of heat flow, the example given in table 9 illustrates how the total heat load can be calculated (hypothetical data for a freezer room are included to show how such data compare with those for a cold-storage room).

Table 9.—Calculation of total heat load

Heat load	Hypothetical cold-storage-room calculation data	Hypothetical freezing-room calculation data
	B.t.u./24 hr.	B.t.u./24 hr.
(1) Wall-heat-gain load	170,000	44,000
(2) Air-change or service load	101,000	21,000
(3) Product load	52,000	255,000
(4) Miscellaneous load	67,000	67,000
Sum	400,000	407,000
Ten percent factor of safety	40,000	41,000
Calculated total heat load	440,000	448,000

Note that, in cold-storage rooms, the product load is but a small part of the total heat load, whereas in freezing rooms, the product load is the principal part of the total heat load.

Rule-of-Thumb Methods

There are numerous short rule-of-thumb methods for estimating the total heat load. Some are based, fallaciously, only on the outside surface area of the refrigerator. The more acceptable methods, however, are based on a combination of the surface area and the net volume.

In the latter procedure, the four heat-source loads (wall-heat-gain, air-change, product, and miscellaneous) are estimated as follows: (1) the wall-heat-gain load is evaluated in terms of the outside surface area of the refrigerator (table 2), and (2) the remaining three heat-gain loads are combined in an estimate based on the net volume of the refrigerator. Simplified tables for making this latter estimate are available in manufacturers' catalogues and in the American Society of Refrigerating Engineers Design Data Book. Although the data in these tables are based upon experience, they apply only under prescribed conditions. Consequently, use of these tables without a knowledge of the basic assumptions and limitations can lead to a very poor estimate. It is therefore suggested that these shortcut methods be relied upon only by the experienced estimator.

For freezing fishery products, one can obtain a very rough approximation of the required refrigeration capacity by (1) calculating the product load and (2) then adding additional capacity to take care of the other three sources of heat gain. The product load required to lower the temperature of 1 ton of fish from 50° to 0° in 24 hours is about 1 ton of refrigeration. The additional capacity added for the other three sources of heat gain depends upon the type of freezer used and has been variously estimated. The American Society of Refrigerating Engineers Applications Data Book gives a range of 25 to 100 percent for this additional capacity.

HOW TO CALCULATE THE SIZE OF THE COMPRESSOR

Method of Rating Compressors

The capacity of a refrigerating compressor is generally rated in B.t.u. per hour or in tons of refrigeration[5] at the desired evaporative temperature[6]. Small compressors used on domestic or commercial refrigerators may be rated (1) in B.t.u. per hour or (2) by the horsepower of

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5/ This term is frequently abbreviated, and the rate of refrigerating effect is stated simply in tons. The standard commercial ton of refrigeration is arbitrarily defined as the removal of heat energy from the cold region at the rate of 288,000 B.t.u. per 24 hours or of 12,000 B.t.u. per hour. This unit derives from the fact that the latent heat of fusion of ice is approximately 144 B.t.u. per pound; thus, for a 24-hour period, 144 B.t.u. per pound x 2,000 pounds per ton equals 288,000 B.t.u. per 24 hours per ton. A refrigerating machine that is operating at a capacity of 1 ton is therefore absorbing heat energy at the same rate as 1 ton of ice would if it were melting in a 24-hour period.

6/ The evaporative temperature is the temperature of the refrigerant inside the evaporator.

the motor required to drive them. Large commercial or small industrial compressors may be rated (1) in tons of refrigeration, (2) in B.t.u. per hour, or (3) by the horsepower required to drive the unit. The ratings of B.t.u. per hour and tons of refrigeration, however, are the ones most commonly used in refrigeration terminology.

Method of Determining Required Size of Compressor

The selection of the proper size of compressor involves the consideration of five factors:

1. The calculated total heat load.
2. The actual hours of operation of the compressor.
3. The calculated capacity of the compressor.
4. The temperature of the refrigerant in the evaporator (evaporative temperature).
5. The available sizes of compressors.

Calculated Total Heat Load (q')

The calculated total heat load in B.t.u. per 24 hours (q') is used as the basis for determining the compressor capacity in B.t.u. per hour or in tons of refrigeration. The method of using q' to determine the compressor capacity is illustrated in problem 7.

Hours of Operation of Compressor (t_c)

If it were possible to operate a compressor 24 hours per day, the calculated total heat load in B.t.u. per 24 hours (q') could be expressed directly as the required compressor capacity (after conversion to B.t.u. per hour). Thus, a calculated total heat load (q') of 240,000 B.t.u. per 24 hours would require a compressor capacity of 10,000 B.t.u. per hour. Compressors, however, are usually not operated continuously for 24 hours per day, as is assumed in the calculation of the total heat load. Rather, they are operated noncontinuously in order to obtain a maximum of trouble-free service, to allow for a defrost cycle, and to provide for normal maintenance and repair. Conventionally, 16 hours of operation of the compressor per 24 hours is considered good practice, but other operating times are used. The most common of these are listed in the following

<u>No defrost cycle</u>.--If the refrigerant temperature is above 30° F., ice will not form on the coils. Under these conditions, general practice favors basing the size of the equipment on an operating time of 18 or 20 hours.

<u>Natural defrost cycle</u>.--If a natural defrost cycle using the heat from the refrigerator air at 35° F. and above is employed, general practice favors basing the size of the equipment on an operating time of 16 hours.

Artificial defrost cycle.—Artificial defrost cycles are of two kinds: (1) automatic and (2) manual.

If an automatic defrost cycle is employed (refrigerator temperatures below 35° F.), the amount of heat added during defrosting depends on the method of operation. The manufacturer therefore ordinarily furnishes data on the heat gain due to defrosting and recommends the operating time to be used in calculations. In the absence of specific data from the manufacturer, an operating time of 16 hours can be assumed safely.

If a manual defrost cycle is employed (the defrosting is ordinarily done once a year), the operation is equivalent to no defrost because the time of defrosting can be chosen when the refrigeration load is the lightest. At such a time, the reserve refrigeration capacity is ample. An operating time of 18 or 20 hours (the same as that employed with no defrost) is therefore used.

Calculated Capacity of Compressor (q_c or q_{ct})

To correct for this less-than-24-hour-per-day operation of the compressor, the required compressor capacity (q_c in B.t.u. per hour, or q_{ct} in ton of refrigeration) is determined as follows:

$$q_c = \frac{q'}{t_c} \quad \text{or} \quad q_{ct} = \frac{q_c}{(12{,}000 \text{ B.t.u./hr.})/(\text{ton of refrigeration})}$$

where, q' is the total heat load in B.t.u. per 24 hours and t_c is the actual time of operation of the compressor per 24 hours.

The following problem shows how the required size of the compressor is calculated:

Problem 7

Given:
1. 480,000 B.t.u./24 hr., calculated total heat load, q'.
2. 16 hr., time (t_c) of compressor operation per 24-hr. day.
3. 0° F., temperature of cold-storage space.
4. -10° F., evaporative temperature (specified when compressor is ordered, but does not otherwise enter into the calculation).

To find: The capacity or size of the required compressor (1) in B.t.u./hr. (q_c) and (2) in ton of refrigeration (q_{ct}).

Solution: (1) B.t.u.-per-hour basis

$$q_c = \frac{480{,}000 \text{ B.t.u./24 hr.}}{16 \text{ hr./24 hr.}}$$

$= 30{,}000$ B.t.u./hr. at an evaporative temperature of -10° F. (basis for selection of compressor).

(2) Ton-of-refrigeration basis

As previously defined, 1 ton of refrigeration equals 288,000 B.t.u. per 24 hours or 12,000 B.t.u. per hour.

Therefore,

$$q_{ct} = \frac{q_c}{(12{,}000\ \text{B.t.u./hr.})/(\text{ton of refrigeration})}$$

$$= \frac{(30{,}000\ \text{B.t.u./hr.})}{(12{,}000\ \text{B.t.u./hr.})/(\text{ton of refrigeration})}$$

= 2.5 ton of refrigeration at an evaporative temperature of -10° F. (basis for selection of compressor) 7/.

Evaporative Temperature

Inasmuch as the output of the compressor depends upon the evaporative temperature, this temperature must be specified to the compressor manufacturer to assure that the selected equipment will be of a suitable size. The evaporative temperature, however, does not otherwise enter into the calculations for determining the desired size of the compressor.

(It should be pointed out that general practice favors a difference of not more than 10° F. between the temperature of the refrigerant in the evaporator and that of the refrigerated storage space in order to prevent dessication of the product.)

Selection of Compressor Size

The size of the compressor is selected on the basis of the calculated capacity requirements (at the evaporative temperature) in B.t.u. per hour (q_c) or in tons of refrigeration (q_{ct}) as was illustrated in problem 7. Inasmuch as compressors are manufactured in standard sizes, probably none of them will be exactly equal to the size indicated by the calculations. The next larger available size rather than the next smaller size should be the one chosen and is symbolically represented by q_c^* or q_{ct}^*. (Other considerations in the selection of a compressor are given in section 2.)

7/ An alternate way of making this calculation would be:

$$q_{ct} = \frac{480{,}000\ \text{B.t.u./24 hr.}}{(288{,}000\ \text{B.t.u./24 hr.})/(\text{ton of refrigeration})} \times \frac{1}{16\ \text{hr./24 hr.}}$$

$$= (1.67\ \text{ton refrigeration}) \times \frac{24}{16} = 2.50\ \text{ton of refrigeration at an}$$

evaporative temperature of -10° F.

HOW TO CALCULATE THE SIZE OF THE EVAPORATOR

Formula for Calculation

Heat required for evaporation of the refrigerant is supplied to the evaporator surface by the surrounding medium that is being cooled. This medium might be a gas, such as the air in the refrigerator; a liquid, such as brine; or a solid, such as a fish product. The rates of heat flow into the evaporator are controlled by a number of factors, which include the type of evaporator (bare pipe or tubing, finned pipe coils, or refrigerated plates), the material and type of construction of the evaporator, the exposed area of the evaporator, and the temperature difference between the surrounding medium and the refrigerant. The same principles employed in the calculation of the flow of heat through walls are generally applicable in the calculation of the size of evaporators. The heat-flow equation may be expressed in the form:

$$q_c^* = UA(T_2 - T_1) \quad \text{or} \quad A = \frac{q_c^*}{U(T_2 - T_1)} \quad \text{where,}$$

A = surface area required for the evaporator, in sq. ft.

[Note: For finned surfaces, it is not practical to calculate the effective coil area because the transfer of heat is variably affected by (1) the size of the fins with relation to the size of the tubing, (2) the number of fins per inch, and (3) the type of contact between the fins and tubing.]

q_c^* = the capacity of the compressor in B.t.u./hr. [Note: This value (q_c^*) is for the capacity of the compressor actually selected rather than for the calculated capacity of the compressor (q_c). See "Selection of Compressor Size." If the capacity of the compressor is given in ton of refrigeration (q_{ct}^*), q_{ct}^* may be converted to q_c^* by means of the following equation:

$$q_c^* = (q_{ct}^*)(12{,}000 \text{ B.t.u./hr.})/(\text{ton of refrigeration})$$

as is illustrated in problem 8.]

U = overall coefficient of heat transfer, in $\frac{(\text{B.t.u./hr.})}{(\text{ft.}^2)(°\text{F.})}$

T_2 = temperature of surrounding medium, such as the air in the refrigerator, in °F.

T_1 = temperature of the refrigerant in the evaporator (evaporative temperature), in °F.

The following discussions and calculations pertain to evaporators used in cold-storage rooms and are limited to (1) bare pipe coils or tubing, (2) finned pipe coils, (3) refrigerated plates, and (4) blower-type unit coolers.

Bare Pipe Coils or Tubing

Bare pipe coils and tubing are most generally applied in liquid cooling, where the coils are submerged, as in an immersion freezer, or in installations in which air temperatures are to be maintained below 34° F., as in cold-storage rooms.

The following is a discussion of the overall-heat-transfer coefficients of cold-storage-room coils and the method of determining the amount of coil surface necessary for a particular installation.

Overall-Heat-Transfer-Coefficient Values

Overall-heat-transfer-coefficient values for pipe coils[8/] with gravity air circulation range from 1.6 to 2.3 $\frac{\text{(B.t.u./hr.)}}{(\text{ft.}^2)(°\text{F.})}$, according to most manufacturers' data. If the temperature differences are quite wide, these heat-transfer values are found to be somewhat larger, owing to the increased circulation of air.

For storage rooms maintained at 0° F., by the use of a direct-expansion system and bare pipe coils at -10° F., a coefficient of heat transfer $U = 1.6 \frac{\text{(B.t.u./hr.)}}{(\text{ft.}^2)(°\text{F.})}$ may be assumed.

Determination of Required Coil Area

The coil area is determined by solving for A in the basic heat flow equation:

$$A = \frac{q_c^*}{U(T_2 - T_1)}$$

The following problem shows how the required area of coil surface and length of pipe is determined:

8/ If these coils are in the form of shelves in direct contact with the fishery product, heat transfer is generally considered to be increased by 15 to 25 percent. If the coils are used in liquid-cooling applications, the heat-transfer coefficients may be increased tenfold over those in air-cooling applications, depending upon the precise conditions of application.

Problem 8

Given:
1. Refrigerator room at 0° F.
2. $1\frac{1}{4}$-inch-pipe-coil evaporator.
3. Evaporative temperature of -10° F.
4. 2-ton ammonia compressor (q^*_{ct} = 2 tons of refrigeration)

To find:
(1) The square feet of evaporator surface (A) required.

(2) The total length of $1\frac{1}{4}$-inch pipe required.

Solution: (1) Square feet

$$A = \frac{q^*_c}{U(T_2 - T_1)}$$

$$= \frac{24,000}{(1.6)(10)}$$

= 1,500 sq. ft. of evaporator surface required

q^*_{ct} = 2 ton of refrigeration.

1 ton of refrigeration = 12,000 B.t.u./hr.

q^*_c = 2 x 12,000
= 24,000 B.t.u./hr.

$U = 1.6 \frac{(\text{B.t.u./hr.})}{(\text{ft.}^2)(°\text{F.})}$

T_1 = -10° F.

T_2 = 0° F.

$T_2 - T_1$ = 10° F.

(2) Total length

$1\frac{1}{4}$-in. pipe = 2.30 linear ft./sq.ft. (from table 10)

required length = 1,500 x 2.3 = 3,650 ft.

Finned Pipe Coils

Finned pipe coils are used to provide the necessary refrigeration effect in chill rooms (34° to 40° F.) and in cold-storage rooms (-20° to 0° F.). At temperatures above 34° F., frost does not accumulate on the coils, and coils with 6 to 8 fins per inch of length can be used. In low-temperature applications (temperatures below 0° F.), however, the number of fins is generally reduced to one to three fins per inch of length in order to allow for the build-up of frost. Finned coils used in low-temperature applications are designed with built-in artificial defrosting systems.

Table 10.—Outside surface areas of copper tubing and of steel and wrought-iron pipe

Nominal size of copper tube	Outside surface areas of copper tube
Inches	Linear ft. per sq. ft.
3/8	7.64
1/2	6.11
5/8	5.09
3/4	4.36
7/8	3.40
Nominal size of steel and wrought-iron pipe	**Outside surface area of steel and wrought-iron pipe**
Inches	Linear ft. per sq. ft.
1/2	4.55
3/4	3.64
1	2.90
$1\frac{1}{4}$	2.30
$1\frac{1}{2}$	2.01
2	1.61

Overall-Heat-Transfer-Coefficient Values

The overall-heat-transfer-coefficient values for direct-expansion finned pipe coils vary greatly with the coil design and with the specific application. Heat-transfer coefficients for various applications of finned-pipe coils have been established by each coil manufacturer for his products. The manufacturer's data should therefore be employed in the selection of a coil for a particular use. The data presented by a coil manufacturer is similar to that shown in table 11.

The following problem shows how to select from the manufacturer's catalogue the coil of proper size for a small walk-in cooler:

Problem 9

Given:
1. Load requirement of 6,000 B.t.u./hr. (q^*_c).
2. Room temperature of 35° F.
3. Fin coil, 120 in. long with 3 fins per in.

To find: Type of coil in table 11 for installations with temperature difference of (1) 15° F. and (2) 17° F.

Table 11.--Partial listing of typical capacity ratings and data for finned tube coils

Designation of coil[1/]	Tubes in coil	Capacity ratings for temperature differences and fin spacing of: [2/]			
		1° F.		15° F.	
		1/2 inch	1/3 inch	1/2 inch	1/3 inch
Type	Number	B.t.u. per hr. per inch of finned length per 1°F.		B.t.u. per hr. per inch of finned length per 15°F.	
A	1	0.27	0.33	4.1	5
B	2	0.55	0.67	8.2	10
C	3	0.82	1.00	12.3	15
D	4	1.09	1.33	16.4	20
E	5	1.37	1.67	20.5	25
F	6	1.64	2.00	24.6	30
G	7	1.91	2.33	28.7	35
H	8	2.18	2.67	32.8	40
I	9	2.46	3.00	36.9	45
J	10	2.73	3.33	41.0	50

1/ Length of coil to be specified after designation of type.

2/ Temperature difference equals temperature of refrigerator box minus temperature of refrigerant in coil.

Solution: (1) To meet the 15° F. requirement, convert the given load data (q_c^*) to a per-inch-of-fin-length basis:

$$\frac{(6{,}000 \text{ B.t.u./hr.})}{(120 \text{ in.})(15^\circ \text{ F.})} = 50 \frac{(\text{B.t.u./hr.})}{(\text{in.})(15^\circ \text{ F.})}$$

Select coil J (1/3-inch fin spacing, 120-inch length) from table 11 to meet the requirements of 50 $\frac{(\text{B.t.u./hr.})}{(\text{in.})(15^\circ\text{F.})}$ for the temperature difference of 15° F.

(2) To meet the 17° F. requirement, which is not listed in the table, express the B.t.u. load on a per-inch-of-fin-length-per-1-degree basis:

Thus, $$\frac{(6{,}000 \text{ B.t.u./hr.})}{(120 \text{ in.})(17^\circ \text{ F.})} = 2.94 \frac{(\text{B.t.u./hr.})}{(\text{in.})(1^\circ \text{ F.})}$$

Now enter table 11 and select coil I (1/3-inch fin spacing, 120-inch length) in the 1° F. column for a fin spacing of 1/3 inch, since 3.00 $\frac{(\text{B.t.u./hr.})}{(\text{in.})(1^\circ \text{ F.})}$ is close to 2.94.

Note: Coil I (1/3-inch fin spacing, 120-inch length) if used in an installation with a temperature difference of 15° F., has a capacity of 3.0 x 15 = 45 $\frac{\text{(B.t.u./hr.)}}{\text{(in.)(15° F.)}}$. This same coil, if used in an installation with a temperature difference of 17° F., has a capacity of 3.00 x 17.0 = 51 $\frac{\text{(B.t.u./hr.)}}{\text{(in.)(17° F.)}}$ or

$$= 6{,}120 \ \frac{\text{(B.t.u./hr.)}}{\text{(120 in.)(17° F.)}}$$

Refrigerated Plates

Refrigerated plates of various types are used for shelf surfaces in freezer rooms, in display cases, in low-temperature storage rooms, and in liquid-chilling tanks.

Overall-Heat-Transfer-Coefficient Values

According to most manufacturers' data, overall-heat-transfer-coefficient values of 2.0 to 2.5 $\frac{\text{(B.t.u./hr.)}}{\text{(ft.}^2\text{)(°F.)}}$ have been found for plates used in still air at 40° F., considering both sides of the plates as being effective and assuming them to be free of frost. Ratings of 2.0 and lower have been given for plates used in air at 0° F., assuming some frost accumulation[9].

For rooms at 0° F., the average overall-heat-transfer coefficient for plates is assumed to be 2.0 $\frac{\text{(B.t.u./hr.)}}{\text{(ft.}^2\text{)(°F.)}}$

Determination of Required Plate Area

The required plate area is determined by solving for A in the basic heat-flow equation:

$$A = \frac{q_c^*}{U(T_2 - T_1)}$$

The following problem shows how to calculate the number of square feet of evaporator surface required:

9/ If the plates are utilized as freezing-shelf surfaces for freezing food products, the ratings are increased by 15 to 25 percent.

Problem 10

Given: 1. Load requirement of 24,000 B.t.u./hr. (q_c^*).
2. Refrigeration room at 0° F.
3. Refrigeration effect to be furnished by overhead refrigerator plates at -16° F.

To find: The number of square feet of refrigerator plates (A) required (both sides of plates are effective).

Solution: Substitute given data in equation:

$$A = \frac{q_c^*}{U(T_2 - T_1)} = \frac{24{,}000}{2(16)} = 750 \text{ sq. ft. (both sides effective)}$$

$q_c^* = 24{,}000$ B.t.u./hr.

$U = 2.0 \dfrac{\text{(B.t.u./hr.)}}{(\text{ft.}^2)(°\text{F.})}$

$T_1 = -16°$ F.

$T_2 = 0°$ F.

$T_2 - T_1 = 16°$ F.

Blower-type Unit Coolers

The blower-type unit coolers are principally used in maintaining temperatures above 34° F. on a natural defrost cycle, but they may be used in low-temperature applications if a method of artificial defrosting is incorporated in their design. Owing to the more rapid air circulation with this type of cooler, there is danger of product dehydration if the difference between the temperature of the air and that of the coil is too great. For ice-glazed fish, most manufacturers list the maximum permissible difference in temperature as 16° F. and the maximum air velocity as 60 feet per minute (American Society of Refrigerating Engineers Design Data Book). For unglazed fish, the temperature difference should not exceed 10° F.

Method of Rating

It is not practical to calculate heat-transfer rates for unit coolers because of the many variables involved. Instead, the coolers are rated in B.t.u. per hour at a specified temperature difference. Thus, one manufacturer lists data for his unit coolers in a manner similar to that used in table 12.

Selection of Suitable Unit Cooler

In the selection of a suitable unit cooler, the following factors should be considered:

Table 12.--Typical partial listing of capacity ratings for unit coolers 1/

Designation of cooler	Unit cooler capacity ratings for specified temperature differences of: 2/		
	10° F.	15° F.	20° F.
Type	B.t.u. per hr.	B.t.u. per hr.	B.t.u. per hr.
A	900	1,350	1,800
B	3,000	4,500	6,000
C	6,000	9,000	12,000
D	9,000	13,500	18,000
E	10,000	15,000	20,000
F	12,000	18,000	24,000
G	24,000	36,000	48,000

1/ A manufacturer's table shows a greater number of in-between sizes than are shown in this sample table.

2/ Temperature difference equals temperature of refrigerator minus temperature of refrigerant in coil.

1. The temperature of installation (high temperature versus low temperature).

2. If the installation is for low temperature, the method of defrost (hot gas, water, or electric).

3. The temperature difference between the air in the refrigerator room and the refrigerant in the unit cooler.

4. The load requirement in B.t.u. per hour.

The following problem shows how to select a unit cooler from manufacturer's data:

Problem 11

Given:
1. Refrigerator room at 0° F.
2. Hot-gas defrost.
3. Unit cooler with fin-coil temperature (a) of -10° F. or (b) of -12° F.
4. Load requirement of 12,000 B.t.u. per hour (q_c^*).

To find: (1) Applicable type unit in table 12 for fin-coil temperature difference of 10° F.

(2) Applicable type unit in table 12 for fin-coil temperature difference of 12° F.

Solution: (1) For a fin-coil temperature difference of 10° F., the type F unit in table 12 has the proper requirements (12,000 B.t.u./hr.) and would therefore be the unit selected.

(2) For a fin-coil temperature difference of 12° F., the capacity requirement would be calculated as follows: The capacity needed on the basis of a 12° F. temperature difference would be: $\frac{12{,}000 \text{ B.t.u./hr.}}{12^\circ \text{ F. difference}}$

This capacity, on the basis of a 1° F. temperature difference, would be: $\frac{1{,}000 \text{ B.t.u./hr.}}{1^\circ \text{ F. difference}}$

And this capacity, on the basis of a 10° F. temperature difference, would be: $\frac{10{,}000 \text{ B.t.u./hr.}}{10^\circ \text{ F. difference}}$

The type-E unit (capacity of 10,000 B.t.u. per hour at a temperature difference of 10° F.) would therefore be the one selected from table 12.

OVERALL ILLUSTRATIVE PROBLEM

The following overall illustrative problem shows how the preliminary calculations required in the construction of a cold-storage room might be made.

The Problem

A fish processor in the New England area had need to construct a nonpallatized 0° F. cold-storage room with a capacity for 25,000 pounds of packaged fillets in cartons. Two thousand pounds of the product was to be cooled from 10° to 0° F. every 24 hours.

Information Required

The processor had to determine:

1. Thickness of insulation.
2. Size of room.
3. Area of insulation.
4. Size of compressor.
5. Size and number of refrigerated plates.

Determination of Thickness of Insulation

The processor decided to insulate his cold-storage room with cork.

Since the cold-storage temperature was 0° F. and the plant was located in the northern part of the United States, table 3 showed that the required thickness of insulation was 6 inches.

Determination of Size of Room

Length of Room

The space situation fixed the outside length of the room at 15 feet. Inasmuch as the processor decided to line the room inside and outside with 1/2-inch-thick plywood, the inside length of the room was fixed at 13.8 feet (15 feet minus 1 foot 2 inches).

Width of Room

The processor found that the density of his product was 55 pounds per cubic foot (see "Walk-in Freezers and Coolers" in this section). He decided to limit the height of the product to 6 feet, for easy stacking, and to have an aisle 4.5 feet wide running the length of the room. To determine the width of the room, he made the following calculations:

$$\text{Volume of product} = \frac{25{,}000}{55} = 455 \text{ ft.}^3$$

$$\text{Area of floor occupied by product} = \frac{455}{6} = 75.8 \text{ ft.}^2$$

$$\text{Area of floor occupied by aisle} = (4.5)(13.8) = 62.1 \text{ ft.}^2$$

$$\text{Total area of floor} = 75.8 + 62.1 = 137.9 \text{ ft.}^2$$

$$\text{Width of room (inside dimension)} = \frac{137.9}{13.8} = 9.99 = 10 \text{ ft. (in round numbers)}$$

$$\text{Width of room (outside dimension)} = 10 + \frac{2(\frac{1}{2} + 6 + \frac{1}{2})}{12} = 11.2 \text{ ft.}$$

Height of Room

Refrigerated plates 12 inches high were selected to provide the necessary refrigeration effect. The inside height of the room therefore was equal to the product-piling height (6 feet) plus the clearance between plates and product (6 inches) plus the clearance between plates and ceiling (6 inches) or to a total of 8 feet.

Floor construction--consisting of $\frac{1}{2}$-inch-thick plywood, 6-inch-thick corkboard, $\frac{1}{2}$-inch-thick plywood, and 4-inch-thick concrete slab--and ceiling construction--consisting of $\frac{1}{2}$-inch-thick plywood, 6-inch-thick corkboard, and $\frac{1}{2}$-inch-thick plywood--when added to the inside height, gave an outside height of 9.5 feet.

Volume of Room

The inside volume of the room was thus fixed at 1,104 cubic feet (13.8 x 10 x 8).

Determination of Area of Insulation

Inasmuch as the required area of insulation is very nearly equal to the outside surface of the room, the processor made the following calculations:

Area of front and rear walls	= 2 x 9.5 x 11.2 = 213 $ft.^2$
Area of side walls	= 2 x 9.5 x 15 = 285 $ft.^2$
Area of roof and floor	= 2 x 11.2 x 15 = 336 $ft.^2$
Area of outside surface of room	= 213 + 285 + 336 = 834 $ft.^2$
Area of insulation required	= 834 $ft.^2$

Determination of Size of Compressor

In the selection of the proper size of compressor, five factors had to be considered:

1. Calculated total heat-gain load.
2. Hours of operation of compressor.
3. Calculated capacity of compressor.
4. Evaporative temperature.
5. Available size of compressor.

Calculated Total Heat-Gain Load (q')

The formula the processor used for determining the total heat-gain load was:

$$q' = 1.10(q'_w + q'_p + q'_s + q'_m)$$ where,

q'_w is the wall-heat-gain load, q'_p is the product-heat-gain load, q'_s is the service-heat-gain load, and q'_m is the miscellaneous-heat-gain load. The figure 1.10 provides a factor of safety of 10 percent.

Wall-heat-gain load (q'_w).--From a study of the weather reports in the files of the local newspaper, the processor decided that 90° F. would be a reasonable outside design temperature. Using the resulting temperature difference of 90° F. (90° F. outside design temperature minus 0° F. cold-storage temperature) and the specified 6 inches of coil insulation, he obtained 108 B.t.u. per square foot per 24 hours from table 2. He then calculated the wall-heat-gain load as:

$$\begin{aligned} q'_w &= (108)(834) \\ &= 90{,}100 \text{ B.t.u./24 hr.} \end{aligned}$$

Product-heat-gain load (q'_p).--Taking 0.4 as the specific heat of the frozen packaged fillets, he calculated the product-heat-gain load as:

$$\begin{aligned} q'_p &= (2{,}000)(0.4)(10) \\ &= 8{,}000 \text{ B.t.u./24 hr.} \end{aligned}$$

Service-heat-gain load (q'_s).--By interpolation in table 5 for a cold-storage room with a volume of 1,104 cubic feet (one door to the room), the processor obtained a value of 16.8 air changes per 24 hours. Then assuming 90 percent relative humidity for the air outside and 60 percent relative humidity for the air inside of the cold-storage room, he obtained 3.56 B.t.u. per cubic foot from table 6, and calculated the service-heat-gain load as:

$$\begin{aligned} q'_s &= (16.8)(1{,}104)(3.56) \\ &= 66{,}100 \text{ B.t.u./24 hr.} \end{aligned}$$

Miscellaneous-heat-gain load.--To insure adequate light, the processor decided to use two 100-watt bulbs. Taking 1 watt as being equal to 3.42 B.t.u. per hour, he calculated the heat-gain-load from the lights as:

$$\begin{aligned} q'_{el} &= (2)(100)(3.42)(24) \\ &= 16{,}400 \text{ B.t.u./24 hr.} \end{aligned}$$

Estimating that one man working intermmittently over an 8-hour day could handle the product, he obtained the value of 1,300 B.t.u. per hour from table 8, and calculated the heat-gain load for the workman as:

$$\begin{aligned} q'_o &= (1{,}300)(24) \\ &= 31{,}200 \text{ B.t.u./24 hr.} \end{aligned}$$

The total miscellaneous heat load was then calculated as:

$$\begin{aligned} q'_m &= 16,400 + 31,200 \\ &= 47,600 \text{ B.t.u./24 hr.} \end{aligned}$$

Calculated value of q'.--Having determined the four separate heat-gain loads, the processor calculated the total heat-gain load as:

$$\begin{aligned} q' &= 1.10(90,100 + 8,000 + 66,100 + 47,600) \\ &= 233,000 \text{ B.t.u./24 hr.} \end{aligned}$$

Hours of Operation of Compressor (t_c)

Inasmuch as the installation was a small one in which a manual defrost cycle was to be used, the compressor would operate 18 hours per day [see "Hours of Operation of Compressor (t_c)" in this section].

Calculated Capacity of Compressor (q_c)

Using the formula $q_c = \frac{q'}{t_c}$, the processor calculated the required capacity of the compressor as:

$$\begin{aligned} q_c &= \frac{233,000}{18} \\ &= 12,900 \text{ B.t.u./hr.} \end{aligned}$$

Evaporative Temperature

Since the product was adequately packaged, it was decided that an evaporative temperature of -15° F. could be used.

Selected Size of Compressor (q_c^*)

The processor, on looking through the various manufacturer's catalogues, found that a typical catalogue listed a model E as having a capacity rating of 11,500 B.t.u. per hour (water cooled; Freon 12; evaporative temperature -16° F.) and a model F as having a rating of 13,200 B.t.u. per hour (under the same conditions). He therefore chose model F.

He found that this model was rated at 14,800 B.t.u. per hour at -12° F. By interpolating (between -12° F. and -16° F.) he found the capacity of the selected compressor (-15° F. evaporative temperature) to be:

$$\begin{aligned} q_c^* &= 13,200 + \frac{(14,800 - 13,200)(16-15)}{(16-12)} \\ &= 13,600 \text{ B.t.u./hr.} \end{aligned}$$

(Note that owing to the fact that compressors are manufactured only in certain sizes, the capacity of compressor selected was about 5.4 percent greater than the required capacity of 12,900 B.t.u. per hour. Thus the room had an additional reserve of refrigeration capacity.)

Determination of the Size and Quantity of Refrigerated Plates

By consulting a catalogue listing refrigerated plates, the processor found that the heat-transfer value (U) of one manufacturer's plates under the conditions of the problem was 2.0 B.t.u. per square foot per hour per degree Fahrenheit. Since the temperature difference between the room and the refrigerant was 15 degrees Fahrenheit, he calculated the total cooling surface required (A) from the basic equation:

$$q_c^* = UA(T_2 - T_1) \qquad \text{or} \qquad A = \frac{q_c^*}{(U)(T_2 - T_1)}$$

$$= \frac{13,600}{(2.0)(15)}$$

$$= 454 \text{ ft.}^2$$

Inasmuch as the length of the room was 13.8 feet, plates 12 feet long could be used.[10] The processor therefore made the following calculations:

Minimum number of plates required $= \dfrac{454}{1 \times 12 \times 2}$ (plates 12 in. deep, 12 ft. long, effective both sides.)

$= 18.9$ plates

Actual number of plates required = 20 plates (to give a balanced system).

In selecting the plates, the processor found that four banks of plates with five plates per bank would be satisfactory.

10/ Refrigerated plates are generally furnished in lengths of 5, 6, 7, 9, or 12 feet and in banks of four, five, or six plates per bank. The manufacturer's recommendations must be followed in the selection of the number of plates per bank so that the pressure drop through the system will not be too great.

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SECTION 2

REFRIGERATION EQUIPMENT

By Joseph W. Slavin, Refrigeration Engineer *

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INTRODUCTION

Refrigeration is the act of producing and maintaining in a substance or space a temperature below that of the surrounding atmosphere.

The withdrawal of heat from the substance or space to accomplish the desired degree of refrigeration below 32° F. requires the use of any one of several refrigerating processes. Each of these depends upon the use of a substance, called the refrigerant, that can be readily converted from a liquid into a vapor or gas, and also from a vapor or gas back into a liquid within a reasonably narrow range of pressures.

The refrigerant, if first stored as a liquid under pressure, then allowed to flow at reduced pressure through a set of pipe coils in the closed system, will withdraw heat from the medium surrounding the coils during the vaporizing stage. The heat so absorbed is removed from the refrigerated area when the vapor or gas returns to that portion of refrigerating equipment designed to cool and compress it again to the liquid state for reuse.

Although a number of refrigerating processes have been developed, the two in commercial use today are the compression system and the absorption system. In this section the two systems will be discussed, and their principal machinery components described. There will also be a discussion of commercial evaporators and of refrigerants.

COMPRESSION SYSTEM

The compression system was first developed for practical use in the middle 1880's, and it became widely accepted for mechanical refrigeration purposes in the early 1900's. In this system, a continuous refrigeration cycle takes place that alternately evaporates the liquid refrigerant at low pressure and temperature, and condenses the vapor at a high pressure and temperature. This continuous refrigeration cycle was made possible by years of hard work and research, which resulted in the development of equipment that would produce a practical, efficient system.

In the early stages of development, large, bulky, and inefficient pieces of equipment were used, resulting in breakdowns and poor operation. As the years progressed, however, small, compact, highly efficient equipment was produced, which resulted in increased use of the compression system for cold-storage plants, quick freezers, home freezers, air conditioning, and many other purposes.

Principles of Operation

The refrigerants commonly employed in the modern compression system are ammonia, Freon 12, Freon 22, carbon dioxide, and methyl chloride.

Of these, ammonia and Freon 12 are the most widely used in commercial plants, because of their many operational and economic advantages.

Compression systems may be either single stage or multiple stage. In the single-stage system, the refrigerant is pumped by means of a single-compression process, whereas in the multiple-stage system, two or more compression processes are used.

Single-stage compression system

The basic equipment in a single-stage compression system consists of a receiver, expansion valve, evaporator, compressor, condenser, and the necessary interconnecting piping and valves. The cycle of operation is as follows (figure 1): The liquid refrigerant, which for purposes of illustration we will assume to be Freon 12, is stored under a pressure of approximately 120 pounds per square inch gauge (p.s.i.g.) in the receiver. From the receiver, it flows through a thermostatically controlled expansion valve to the evaporator coils. The expansion valve, by suitably throttling the refrigerant flow, reduces the pressure from 120 p.s.i.g. to a pressure corresponding to the desired evaporator temperature, thereby producing a mixture of liquid and vapor. The

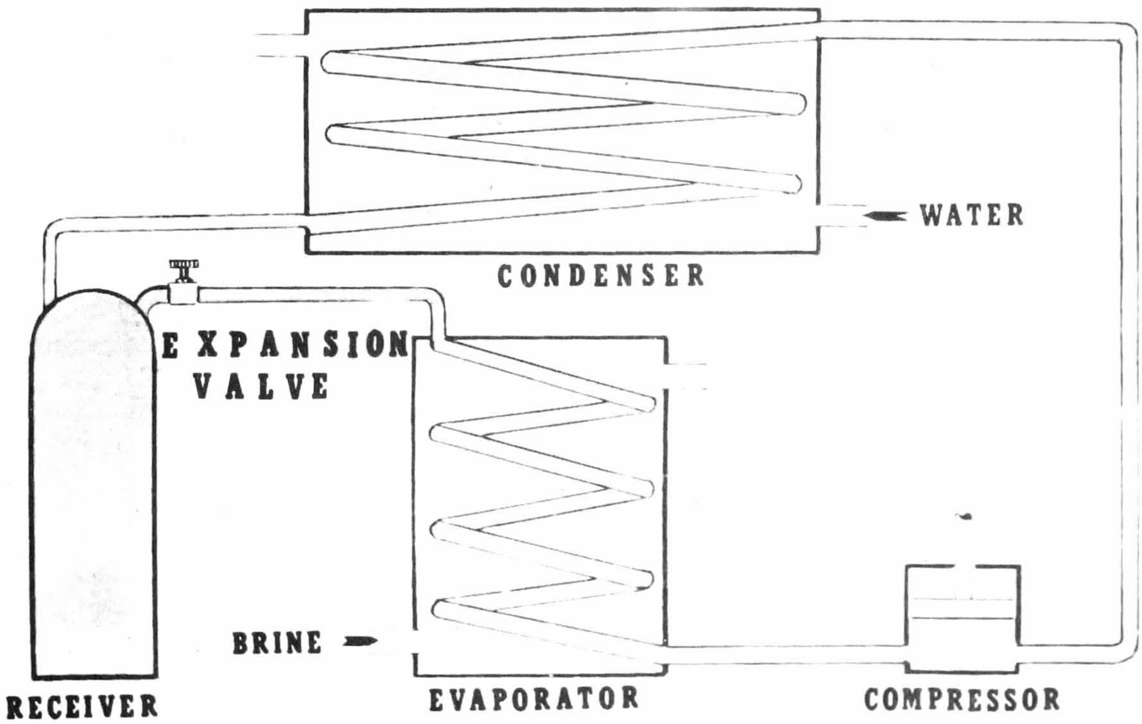

Figure 1.--Flow diagram of the basic single-stage compression system.

liquid then flows through the evaporator coils, where it extracts heat from the products being cooled--brine, in the illustration--and changes to a vapor at approximately the same pressure and temperature as that at which it left the expansion valve.

This vapor then enters the compressor, where it is compressed to 120 p.s.i.g. The compressed gas, after leaving the compressor, is discharged into a condenser, which uses water or air as the cooling medium. The gaseous refrigerant in the condenser is cooled to a liquid, and this liquid then enters the receiver, thereby completing the cycle.

When using refrigerants other than Freon 12 in the compression system--such as ammonia, Freon 22, carbon dioxide, and methyl chloride--the cycle of operation is similar to that described above. However, the operating pressures and specific equipment design for each compression system will vary with the type of refrigerant employed.

Multiple-stage compression system

The use of the single-stage compression system for low-temperature applications is limited by (1) the compression ratio of the compressor, (2) the difference between the evaporator temperature and the discharge temperature, and (3) the capacity of the system in tons of ice melting equivalent1/.

To keep the power requirements and displacement of the compressor at a minimum, two- and three-stage compression systems are used.

Because of the many factors involved in the design of a compression system--such as capacity, evaporative temperature, type of compressor, and refrigerant--there is no clear-cut dividing point as to where a two-stage system should be used instead of a single-stage system, or a three-stage system instead of a two-stage system. However, generally the single-stage system is used for evaporative temperatures of above -20° F., the two-stage system is used for evaporative temperatures of -20° to -75° F., and the three-stage system is used for evaporative temperatures below -75° F.

Two-stage Compression System

In the two-stage system, there are two compressors connected together in series (figure 2). The compressor nearest the evaporator is

1/ The refrigeration capacity of a system is measured in tons of ice melting equivalent (i.m.e.). One ton i.m.e. is the rate of cooling afforded by 1 ton of ice at 32° F. melting in 1 day, which is equal to the extraction of 288,000 B.t.u. per day.

referred to as the low-stage compressor, whereas the other one is referred to as the high-stage compressor. Each compressor serves to compress the gas partially so that, upon leaving the high-stage compressor, the gas pressure is the same as it would be upon exit from a single-stage system. The gas from the evaporator, after passing through the low-stage compressor, must be cooled before entering the high-stage compressor; otherwise the excessive heat will damage the high-stage compressor. This cooling of the gas is referred to as "interstage cooling" and is accomplished by either a water-cooled gas cooler or by a flash-type cooler.

In the water-cooled gas cooler, water circulating within tubes cools the gas, which passes around the outside of the tubes.

In the flash-type cooler, part of the liquid refrigerant from the receiver passes through a thermal expansion valve into a liquid cooler, where it (1) cools the liquid going from the receiver to the evaporator and (2) then passes into the discharge line of the low-pressure compressor, where it mixes with the gas entering the high-stage compressor and cools this gas.

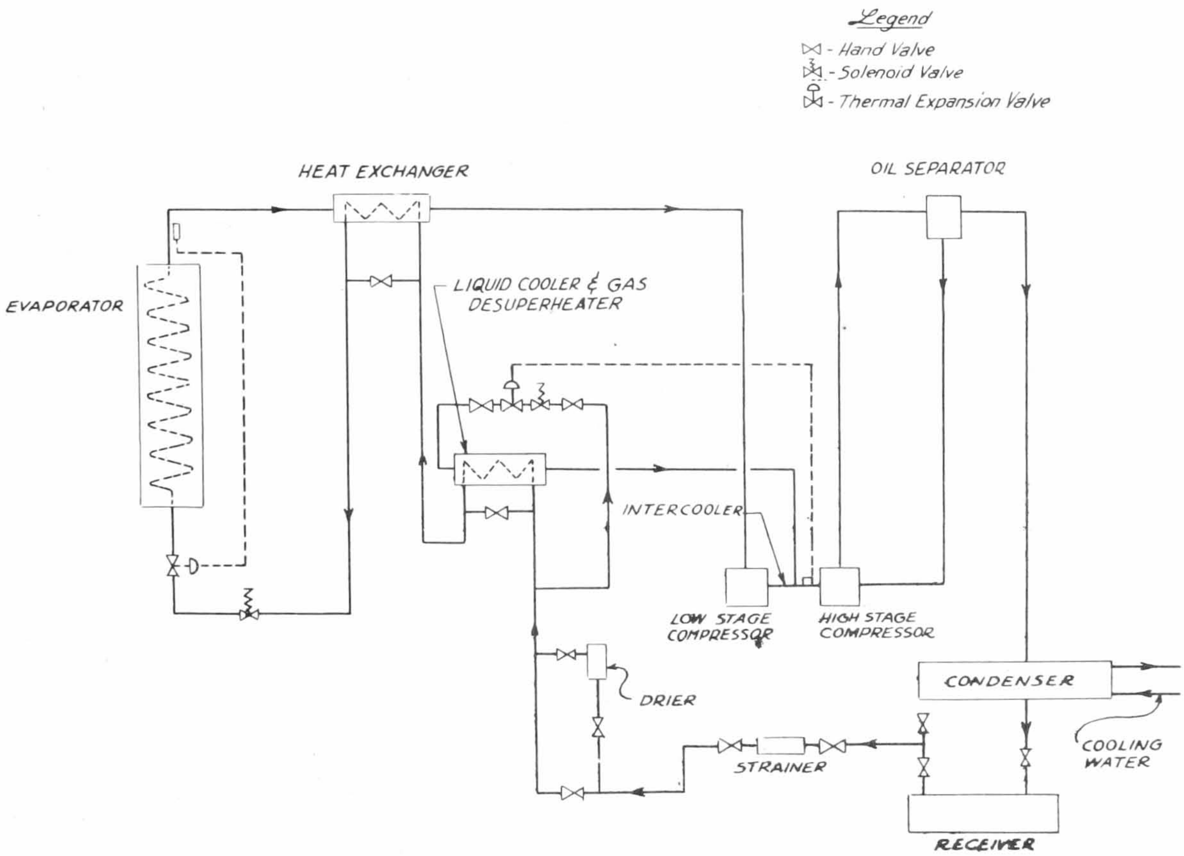

Figure 2.—Schematic diagram of a Freon 12 two-stage compression system, showing all the auxiliary equipment.

The refrigerant, after leaving the high-stage compressor in a two-stage compression system, follows the same path as does the refrigerant leaving the compressor in a single-stage system. The basic equipment used in the two-stage system is also similar to that used in the single-stage system, with the exception of the intercooling assembly and the additional compressor.

Refrigeration compressors are now being manufactured with two stages built into one machine. This unification is accomplished by means of a certain number of the cylinders (usually three) operating on the low stage and the rest (usually one) operating on the high stage. The result is a compact, efficient compressor for low-temperature application.

Three-stage Compression System

The three-stage compression system is similar to the two-stage system except that the compression of the refrigerant is accomplished by three compressors connected in series rather than by two. The three compressors are referred to as low-, intermediate-, and high-stage compressors. The gas between the low and intermediate stage and between the intermediate and high stage is cooled by use of methods similar to those described for the two-stage system.

In the selection of a compression system for low-temperature applications, a refrigeration engineer should be consulted in order to determine the most satisfactory and economical method of handling the specific application involved.

Equipment

In addition to the compressor, condenser, receiver, expansion valve, and evaporator, which form the basic compression system, there are other pieces of machinery that have been added throughout the years in order to improve the performance of the system. These additional pieces have been grouped together under the general classification of "Auxiliary Equipment," which is described later in this section. A description of the specific equipment design employed with each refrigerant would be too lengthy for this publication. Therefore, the following discussion will be limited to a general description of the basic equipment contained in the compression system. Additional information can be obtained from the references at the end of this section.

Compressors

The compressor is referred to as the heart of the compression system. Its principal function is similar to that of a positive-acting pump being used to circulate water from a low-pressure supply to a tank under high pressure. In the compression system, the compressor draws the refrigerant, which is in the form of a vapor, out of the evaporator at a low pressure and temperature and compresses it to a gas at a higher pressure and temperature. The high pressure enables the refrigerant to flow through the other component pieces of equipment that then put it in a state ready for reuse in the evaporator.

The first compressors used were of the reciprocating type. They were large and uneconomical and ran at the very low speed of 50 revolutions per minute. Many improvements in this type of machine were made in the years following its inception, resulting in a compact, efficient, high-speed machine capable of automatic operation. During the periods when the reciprocating compressor was being improved upon, other machines known as the centrifugal compressor and the rotary compressor were developed for refrigeration purposes. The following is a discussion of these three types of compressors, their uses, and their relative merits.

Reciprocating Compressors

The reciprocating compressor is the most common type of compressor. It is used to supply refrigeration for cold-storage plants, sharp freezers, plate freezers, blast freezers, and many other kinds of freezing apparatus. All reciprocating compressors are basically alike, consisting of pistons, crankshaft, valves, and other component parts. There are many different commercial makes of reciprocating compressors, each employing a slightly different design. As these various designs are too numerous to mention here, this discussion will be concerned with the general design features that are incorporated in one manner or another in the present-day machines.

Classification.—A reciprocating compressor is classified either as single-acting or double-acting. In the single-acting compressor (figure 3), the gas is compressed and discharged at the top of the piston only, whereas in the double-acting compressor (figure 4), both the bottom and top of the piston are used to compress and discharge the gas. In the double-acting compressor, a crosshead and other gear is used to absorb the thrust on the under side of the piston. It is obvious that the double-acting compressor will give a much higher output of gas for the same size unit operating at the same speed. Owing to the extra gear involved, consisting of the crosshead and component parts, it is cheaper to produce the single-acting compressor in the small and medium sizes, and the double-acting compressor in the larger sizes.

Compressor speed.--The speed of a compressor is determined in the original design by the manufacturer, and therefore we will touch on the subject only enough to show the effect of the speed on the size and weight of the unit. If higher rotative speeds are used, compressors can be built that occupy less space and that are lighter in weight per unit of horsepower. The speed depends on the size of the cylinders,

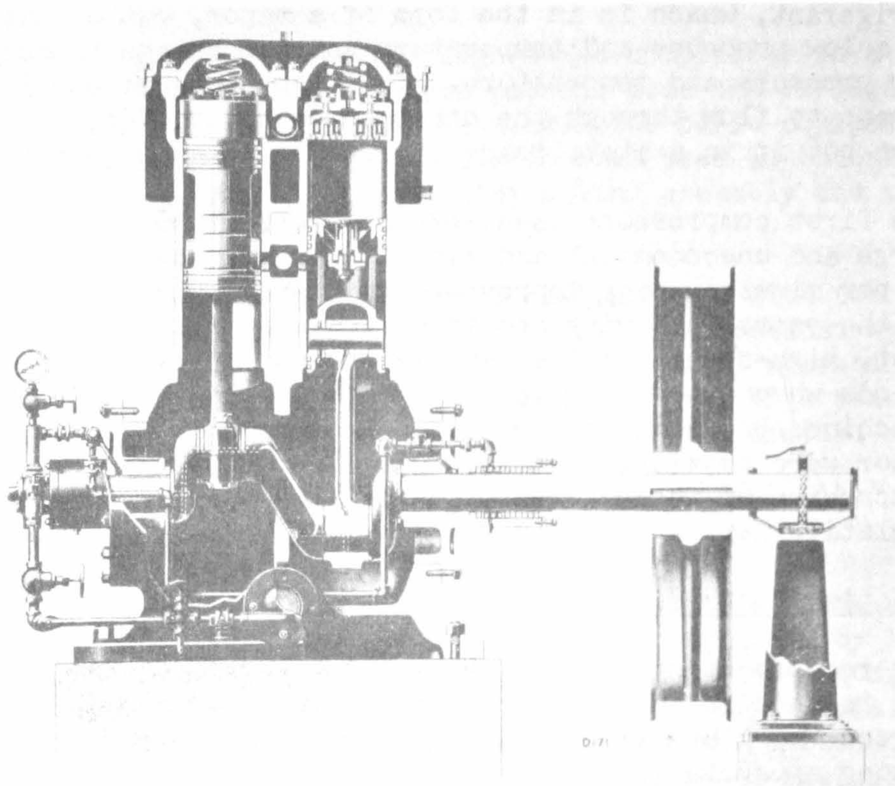

Figure 3.--An ammonia, single-acting reciprocating compressor. In this compressor, the suction gas enters the cylinders through valves in the top of the pistons and is discharged through plate-type valves in the cylinder heads. (Photo courtesy of Frick Company)

the number of cylinders, and the valve action. The cylinders may be arranged in line, radially, or in a v-formation, depending on the manufacturer's design. The trend in compressor design today is toward higher rotative speeds, ranging from 300 to 1,750 r.p.m.

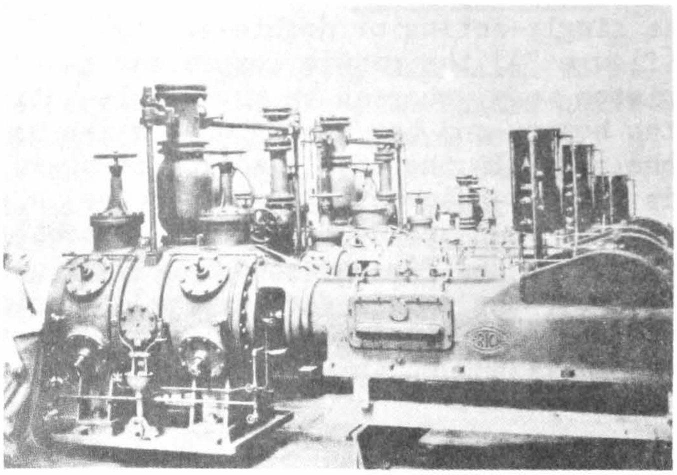

Figure 4.--Ammonia double-acting reciprocating compressors. (Photo courtesy of Frick Company)

Capacity regulation.--If a compressor is used in a plant where the refrigeration load varies continually, a method of capacity regulation should be employed; otherwise continuous starting and stopping of the machine, which is referred to as "cycling," will occur when the load is small, and long running periods will occur when the load is large. To make possible the operation of a single compressor at reduced loads, three different methods known as (1) cylinder bypass, (2) clearance pockets, and (3) speed regulation have been employed. In addition, if more than one compressor is available, a system of compressors in parallel can be used. The following is a description of the above mentioned methods of providing capacity regulation along with a discussion of their related merits.

Cylinder bypass.--This method can be used (1) by holding the suction valves open on some of the cylinders, (2) by opening a valve between cylinders where the crank bends are 180 degrees apart, (3) by arranging a bypass at the bottom half of the cylinder so only the upper half is used (double-acting only), or (4) by installing a solenoid valve at the discharge end of some of the cylinders. In the last procedure, some of the gas is made to flow back to the suction line, thereby bypassing the other cylinders.

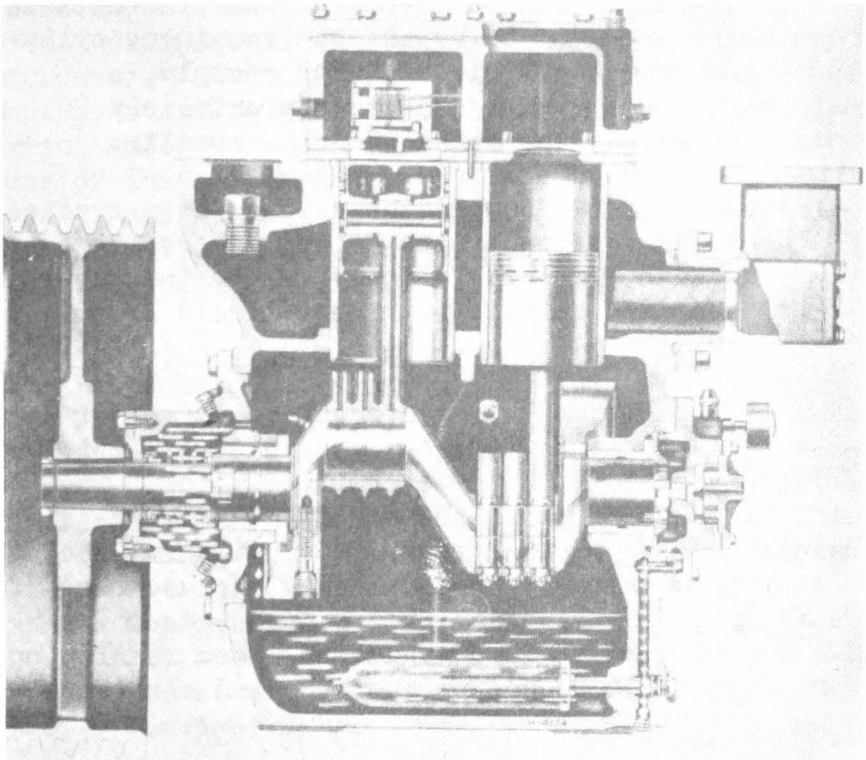

Figure 5.--An ammonia, single-acting reciprocating compressor with variable capacity control. When the solenoid located in the compressor head is de-energized, fingers actuated by a power spring hold open the suction valve strips, reducing the output capacity of the machine. (Photo courtesy of the Worthington Corporation)

The most common type of cylinder bypass is a solenoid valve in a line connected to the discharge of 2, 3, or more cylinders. As the load decreases, the solenoid valve opens, bypassing the hot gas back to the suction line. Thus, the number of cylinders bypassed depends on the load.

Clearance pockets.--Operation of a machine with clearance pockets is similar to that with cylinder bypass, the only difference being in the method of application; the results are the same.

Clearance is the space in the cylinder between the piston, at the end of the compression stroke, and the suction and discharge valves, when seated. If the clearance is increased at a constant pressure, the amount of vapor circulated will be reduced, thus reducing also the cylinder power and efficiency. This is the principle on which the clearance pocket operates.

Clearance pockets are of two major designs. The first design consists of 2 or 3 chambers connected to the compressor cylinder by means of a valve in each chamber. When the valve is opened, a passageway is opened between the cylinder and chamber, thereby increasing the clearance by the net volume of the clearance pocket, resulting in a lower output of the machine. The second design consists of an auxiliary cylinder and piston connected to the compressor cylinder. The amount of clearance is controlled by moving the auxiliary piston in and out.

The following are some of the disadvantages resulting from the use of cylinder bypasses and clearance pockets:

1. The compressor is operated with only some of its cylinders in use. Power is therefore being expended to bypass the remaining cylinders, resulting in a lower compressor efficiency. For example, a compressor operating at half capacity will consume approximately 70 percent of the power of a compressor operating at full capacity.

2. The lowered efficiency of the compressor results in a greater degree of "superheat"2/ upon compression. With Freon 12, the degree of superheat may not be excessive, but with ammonia, where the superheat is excessive, a provision must be made to inject liquid ammonia into the suction line after starting.

Speed regulation.--The capacity of a reciprocating compressor may be controlled by the use of a two-speed, three-phase induction motor or of a rheostat controlling the speed of a direct-current motor. The slowest speed permissible is that at which the oil film in the bearings can be sufficiently maintained to provide proper lubrication. Speed regulation results in a certain degree of inefficiency because the compressor reaches its maximum efficiency at the load and speed for which it was designed. The loss in efficiency due to speed regulation, however, is much less than that due to cylinder bypass and clearance pockets.

Multiple compressors in parallel.--As was just pointed out, the maximum capacity of a reciprocating compressor is delivered when the

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2/ When a liquid is raised to the boiling point corresponding to the pressure on the liquid, the resulting gas is said to be "saturated." If this gas is removed from the liquid and is heated to undergo an increase in temperature without an increase in pressure, it is said to be "superheated." The degree of superheat is the difference between the temperature of the gas at its saturated temperature and its temperature after being superheated.

compressor is operated at its designed load; therefore, in a plant where the load varies appreciably, the most economical arrangement is to have two or more compressors connected in parallel. Each compressor will then operate at its designed capacity, and as the load varies, the compressors will start up or shut off by means of automatic switches. This method requires a higher initial investment, a greater number of parts in the system, and a larger space. The increase in efficiency, however, will in time offset the higher initial cost. Thus, in spite of the initial cost, the use of multiple compressors in parallel is the method most widely employed for obtaining capacity regulation.

Compressor size and method of drive.--Reciprocating compressors are available in sizes of 1/4 hp. to over 350 hp. They are suitable for pumping efficient refrigerants such as ammonia, methyl chloride, carbon dioxide, Freon 12, and Freon 22. If the compressor is of 100 hp. or less, the compressor, the driving motor, the condenser, and the receiver usually form an integral unit. If compressors of over 100 hp. are used, the condenser and receiver are located in the machinery space adjacent to the compressors. The most common method of driving reciprocating compressors is by means of an alternating-current induction motor. The compressor is, however, readily adaptable for drive by means of a direct-current motor, steam engine, diesel engine, or gasoline engine. Because of its adaptability for these various drives, it has a very wide range of use in refrigeration applications. Its high efficiency, low initial cost, low maintenance cost, ruggedness, and compactness are reasons for its wide use.

Selection of a reciprocating compressor.--There are many factors that should be considered when a reciprocating compressor is selected for a particular application. The following is a list of some of the more important of these factors:

1. The horsepower and tonnage rating at the evaporator temperatures required.

2. The amount of floor space taken up by the compressor and drive.

3. The size of the component equipment such as condenser and receiver.

4. The ability of the compressor to operate efficiently at variable loads.

5. The initial cost, operating cost, and over-all efficiency.

6. The advantages and disadvantages of the refrigerant to be used.

7. The ruggedness, durability, and ability to operate automatically.

Centrifugal Compressors

The centrifugal compressor is well suited for high-tonnage and low-temperature requirements. This type of compressor is made in sizes of 100- to 3,000-tons capacity. The refrigerants most widely used in it are Freon 11 and Freon 113.

The centrifugal compressor (figure 6) resembles, in principle, a multiple-stage, centrifugal water pump consisting of 2 to 4 impellers in series, with a stationary set of diffuser plates for each impeller. The total pressure produced is the sum of the pressures produced by the individual impellers. The pressure produced by the centrifugal compressor is also related to the molecular weight of the refrigerant, those refrigerants having a high molecular weight being the most suitable for use. The compressor rotates at speeds from 3,500 to 4,000 r.p.m. for units of 1,000- to 2,000-tons capacity, and from 7,000 to 8,000 r.p.m. for units of 100- to 200-tons capacity. Power may be provided by electric motors equipped with gear-type speed increasers or by a steam turbine. The capacity of the machine can be efficiently controlled by varying the speed of the drive or by using a suction damper that controls the flow of gas to the compressor.

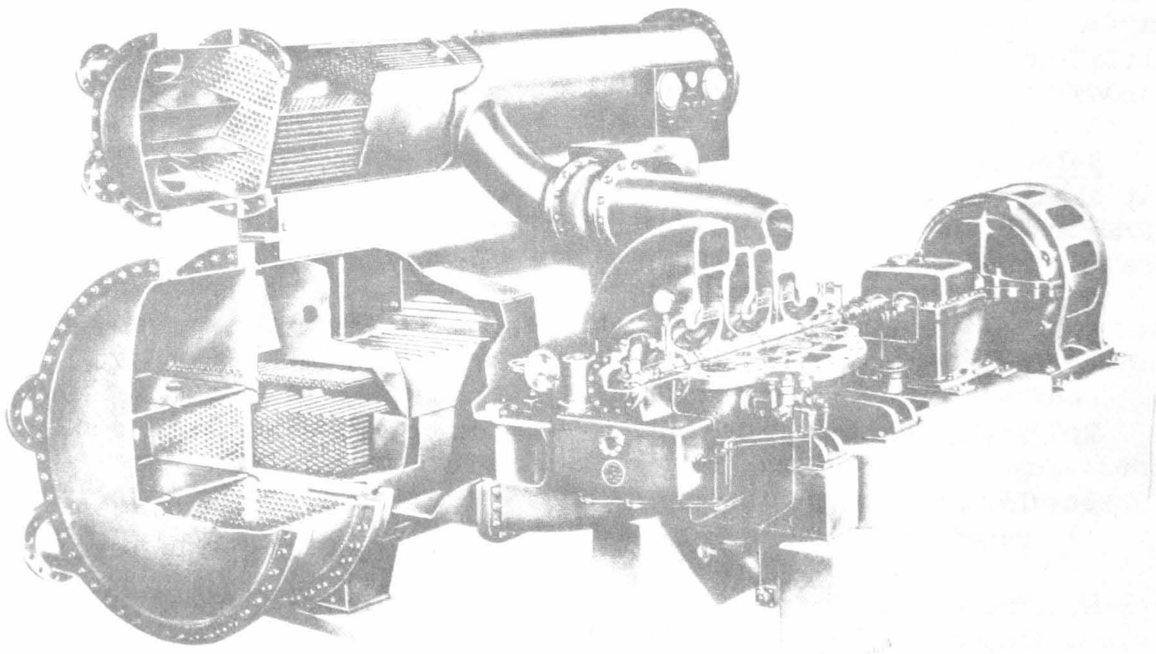

Figure 6.--A centrifugal compressor with condenser and brine cooler. (Photo courtesy of Worthington Corporation)

Because of the large volume of gas required to produce the necessary refrigeration effect, an indirect system with a heat exchanger must be employed. The heat exchanger is of the flooded type, with the Freon 11 flowing around the tubes and the brine circulating through the tubes. The condenser, brine cooler, interstage liquid cooler, compressor, and drive are usually located on a common base plate. This arrangement results in less heat losses and makes a more compact unit.

Commercial use of the centrifugal machine has largely been limited to large refrigerator ships and to air conditioning.

Rotary Compressors

The rotary compressor has recently been adapted for use as a booster or low-stage compressor in many cold storage and freezing plants. The machine is well suited for this application because of its high efficiency in handling large volumes of gas at low pressure.

In a two-stage system, the difference between the suction and discharge pressure of the low-stage compressor is relatively small (5 to 30 pounds). If this pressure difference decreases to about 5 to 10 pounds, difficulty in valve operation of a reciprocating compressor might be experienced, whereas the operation of a rotary compressor, which has no valves, will not be affected. However, the rotary compressor is not suitable for use as a high-stage compressor because of the limited allowable pressure difference across the machine.

The rotary compressor consists essentially of a cylinder within which is located a rotor fitted with blades. The gas--usually Freon 12 or ammonia--enters the compressor through a series of suction ports and is compressed by the rotating blades and forced out of a discharge port located in the casing. In many machines, a secondary "coolant" is circulated through a jacket to prevent freeze up of the cylinder due to the low suction-temperature encountered.

Condenser

The function of the condenser is to cool the gaseous refrigerant after it leaves the compressor, thereby changing it to a liquid. The two principal types of condensers are the water-cooled type and the air-cooled type.

The water-cooled condenser consists of a shell with tubes running longitudinally through it. Salt water or fresh water circulating through these tubes cools the refrigerant, which surrounds the outside of the tubes. The water, in flowing through the tubes, makes from one to four passes, depending on the particular design of the equipment.

The air-cooled condenser (figure 7) is similar to a radiator in an automobile in that it consists of rows of finned tubing. Two fans--one attached to the drive pulley on the compressor and the other to the drive pulley on the motor--draw the air in from the surrounding atmosphere and over the finned coils, thereby cooling the gaseous refrigerant circulating through the finned coils.

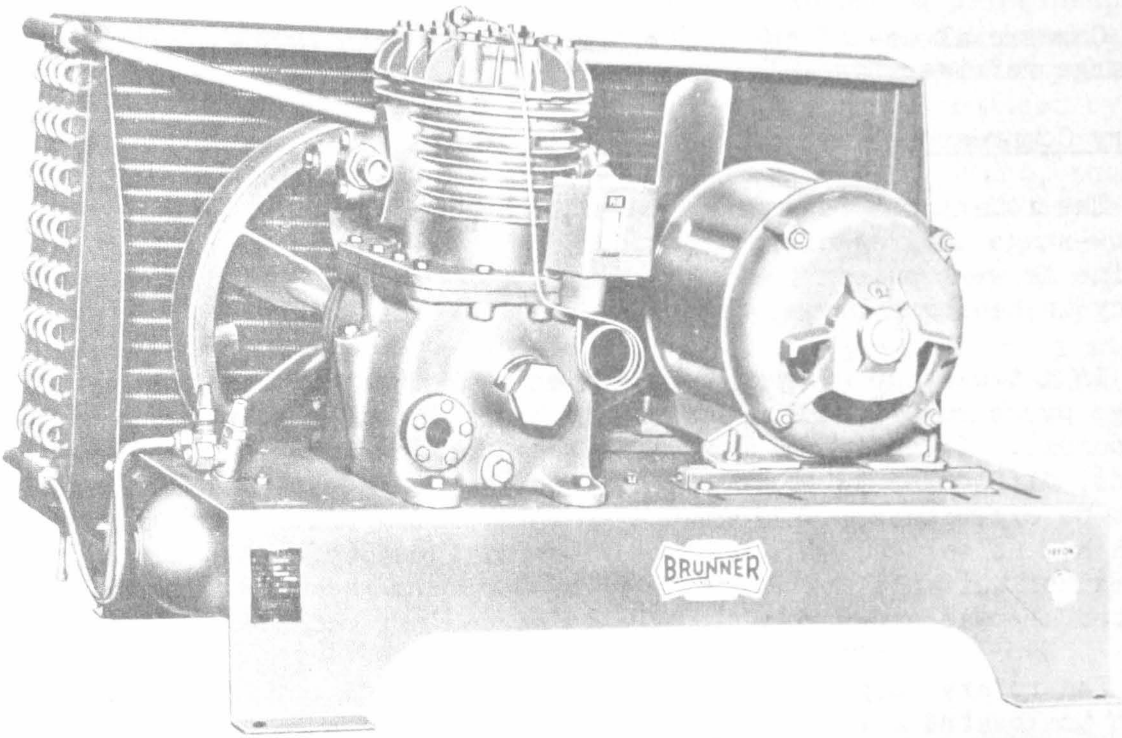

Figure 7.--A 3-hp. Freon 12 compressor with an air-cooled condenser. (Photo courtesy of Brunner Manufacturing Company)

Of the two types of condensers mentioned above, the water-cooled type is used the more extensively. This is because of its adaptability for both large and small loads. The air-cooled condenser is used largely with small refrigerating compressors, from 1/4 hp. to 3 hp. It has found wide usage in small Freon installations and in refrigerated railroad cars and trucks.

Receiver

The receiver is a cylindrical vessel used to store additional refrigerant. It is located between the condenser and the expansion valve (figure 1). In providing a storage place for the refrigerant, the receiver performs the following two functions:

1. It furnishes additional refrigerant in the event of leakage.

2. It provides a storage place for the refrigerant when the system is shut down for maintenance.

Expansion valve

The function of the expansion valve is to control the flow of refrigerant through the evaporator. This valve makes it possible to obtain a high heat transfer within the evaporator and also prevents liquid refrigerant from entering the compressor. The first expansion valves were operated manually and had to be adjusted whenever the load varied. The hand-operated expansion valves have now been replaced by automatic-type expansion valves that are of the thermostatic type, the constant-pressure type, or the float-operated type.

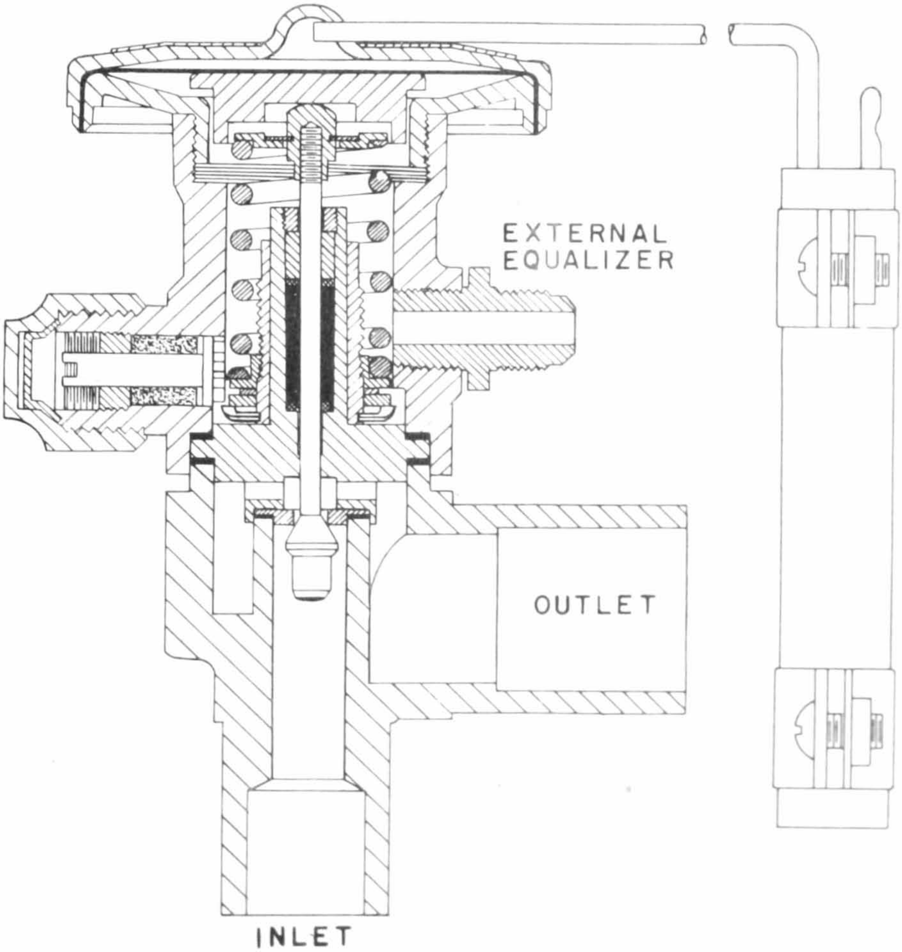

Figure 8.—Cross-sectional view of a single-outlet thermo-expansion valve for Freon 12. (Photo courtesy of Alco Valve Company)

In the thermostatic type (figure 8), the valve opening is controlled by the temperature of the gas leaving the evaporator. In the constant-pressure type, the valve opening is controlled by the evaporator pressure. In the float-operated type, the valve opening is controlled by the level of liquid refrigerant in the evaporator. The float-operated type is used only on flooded-evaporator systems, whereas the thermostatic and the constant-pressure types are used on both flooded and dry-expansion systems. (For a description of flooded and dry-expansion systems, see "Evaporators.")

Evaporator

The design of a particular evaporator depends upon the specific cooling application involved, rather than upon the nature of the refrigeration system employed, such as the compression or absorption system. Therefore, a detailed discussion of evaporators will be given later in this section. At this point, it will suffice to say that the evaporator is primarily a piece of heat-transfer equipment, using refrigerant as a cooling medium, thereby cooling air, water, brine, or foodstuffs.

Auxiliary equipment and controls

In addition to the basic equipment required in the compression system, auxiliary equipment and controls are necessary to insure satisfactory operation.

Auxiliary Equipment

The auxiliary equipment required consists of the following: (1) oil separator, (2) drier, (3) sight glass, and (4) heat exchanger. The following discussion will be concerned with the functions of these pieces of equipment in regard to the compression system.

Oil separator.--A small amount of oil mixes with the refrigerant during the compression cycle. If this oil is allowed to circulate through the system, sticking of the expansion valves and a drop in operating capacity will result.

The oil separator consists of a cylindrical tank equipped with baffles and a sight glass, located in the hot-gas discharge line between the compressor and condenser. The oil contained in the refrigerant-discharge gas, after being mechanically trapped in the oil separator, flows back to the compressor crankcase.

Drier.--The drier is used to absorb the small amounts of moisture that may infiltrate into the system and is located in the liquid line between the receiver and the expansion valve. Commercial driers consist of a tubular vessel containing a removable cartridge made of a moisture-absorbing material—such as silica gel, activated alumina, anhydrous calcium sulfate, or calcium oxide.

Sight glass.--The sight glass is located in the liquid discharge line, usually after the drier, and it enables the operator to observe when there is an insufficient amount of refrigerant in the system or when the refrigerant flow stops as the result of a frozen expansion valve or of other causes.

Heat exchanger.--The heat exchanger is located between the evaporator and the low side of the compressor. The mixture of liquid and vapor leaving the evaporator is superheated when passing through the heat exchanger, by the liquid refrigerant flowing in coils within the heat exchanger. The liquid refrigerant, after being chilled in the heat exchanger, then enters the expansion valve. The precooling of the liquid refrigerant by the superheating of the compression suction gas increases the capacity output of the compressor considerably. Heat exchangers should be used in all commercial applications where the evaporator temperature is below 20° F.

Controls

To obtain continuous automatic operation of the compression system, certain controls are necessary. The ones generally used consist of a thermostat, a solenoid valve, a low-pressure switch, and a high-pressure switch. (For ultra-low temperature applications, other controls are used, which, however, will not be treated here.)

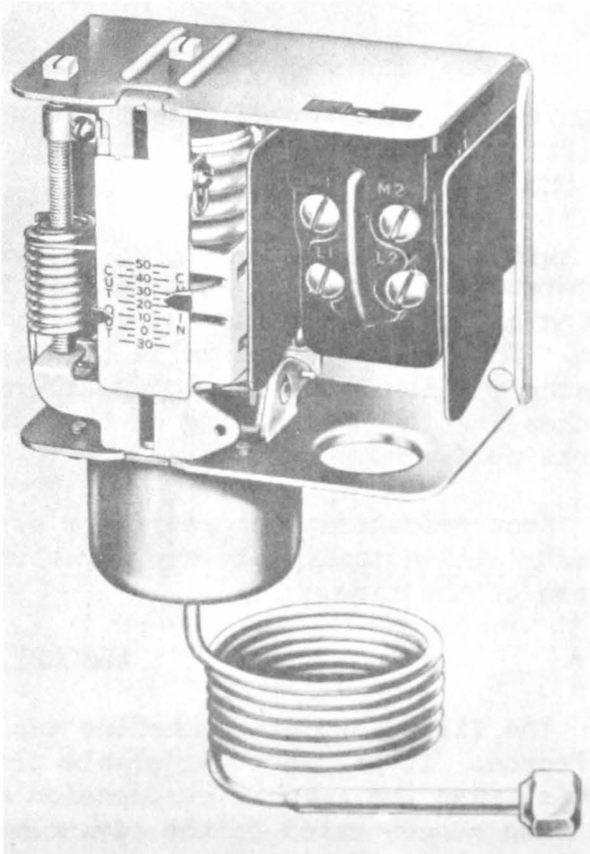

Figure 9.—A low-temperature thermostat. (Photo courtesy of Penn Controls, Inc.)

The thermostat (figure 9) is essentially an electric switch that is opened or closed by a thermo bulb filled with mercury. Thermostats are located in the cold-storage room and are available for the control of temperatures from 40° to -100° F. The electric switch in the thermostat is connected in series with the electric coil in the solenoid valve. The solenoid valve is located in the liquid refrigerant line after the receiver and just before the expansion valve. The low- and high-pressure switches are on the compressor. The low-pressure

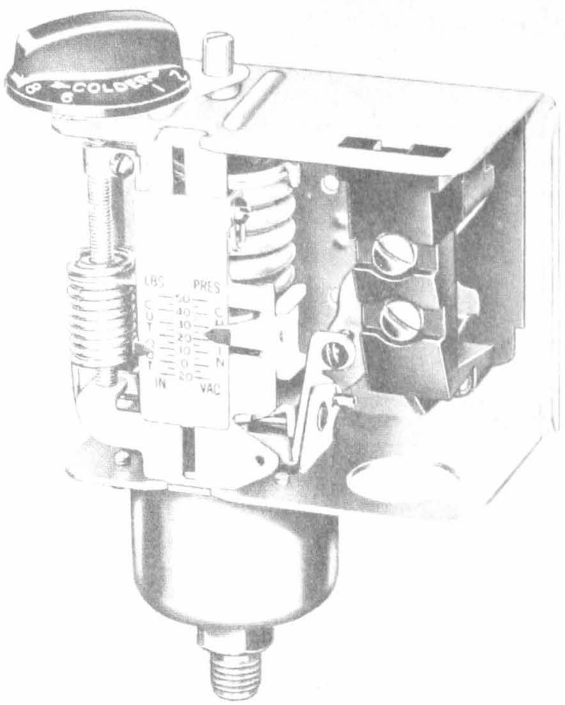

Figure 10.--A low-pressure switch. (Photo courtesy of Penn Controls, Inc.)

switch (figure 10) shuts off the power to the compressor when the suction pressure reaches the predetermined lower limit, and starts it up again when the suction pressure rises to the predetermined upper limit. The high-pressure switch shuts off the compressor when the discharge pressure exceeds the normal operating pressure, thereby preventing damage to the machinery.

The operation cycle of these controls is as follows: When the cold-storage room reaches its operating temperature, the switch in the thermostat opens, causing the solenoid valve to close, thereby stopping the flow of refrigerant to the evaporator. The compressor then pumps the vapor out of the evaporator, thereby reducing the suction pressure. When the suction pressure reaches that of the setting on the low-pressure switch, the switch opens and shuts off the power to the compressor. When the temperature in the cold-storage room rises $2\frac{1}{2}°$ F., the thermostatic switch closes, energizing the coil in the solenoid valve, causing it to open. The open solenoid valve then admits refrigerant to the evaporator, causing a rise in the suction pressure. When the suction pressure reaches that of the setting on the low-pressure switch, the compressor starts up again.

Most refrigeration compressor systems are designed so that the compressor will actually operate for about 16 hours a day--or about two-thirds of the time.

ABSORPTION SYSTEM

The first absorption machine was invented in 1860 by Ferdinand Carre, in France. It gained considerable prominence from 1860 to the early 1900's, when the ammonia compression machine became widely accepted. Work was then concentrated on the advancement of the compression machine until the late 1930's, when companies began to introduce absorption machines, ranging from 25- to 3,600-tons capacity, for industrial use. The following discussion will deal with the principle of operation and the equipment used in the modern absorption system.

Principle of Operation

Ammonia is the only economically suitable refrigerant for use in the absorption system. The principle of the absorption system is based on the absorption of ammonia vapor by water at cooling-water temperature and the release of ammonia vapor when the water is heated. The following is a description of the operation:

Liquid anhydrous ammonia is stored under a pressure of approximately 160 p.s.i.g. in an ammonia receiver (figure 11). From the receiver, the ammonia is fed through an expansion valve into the

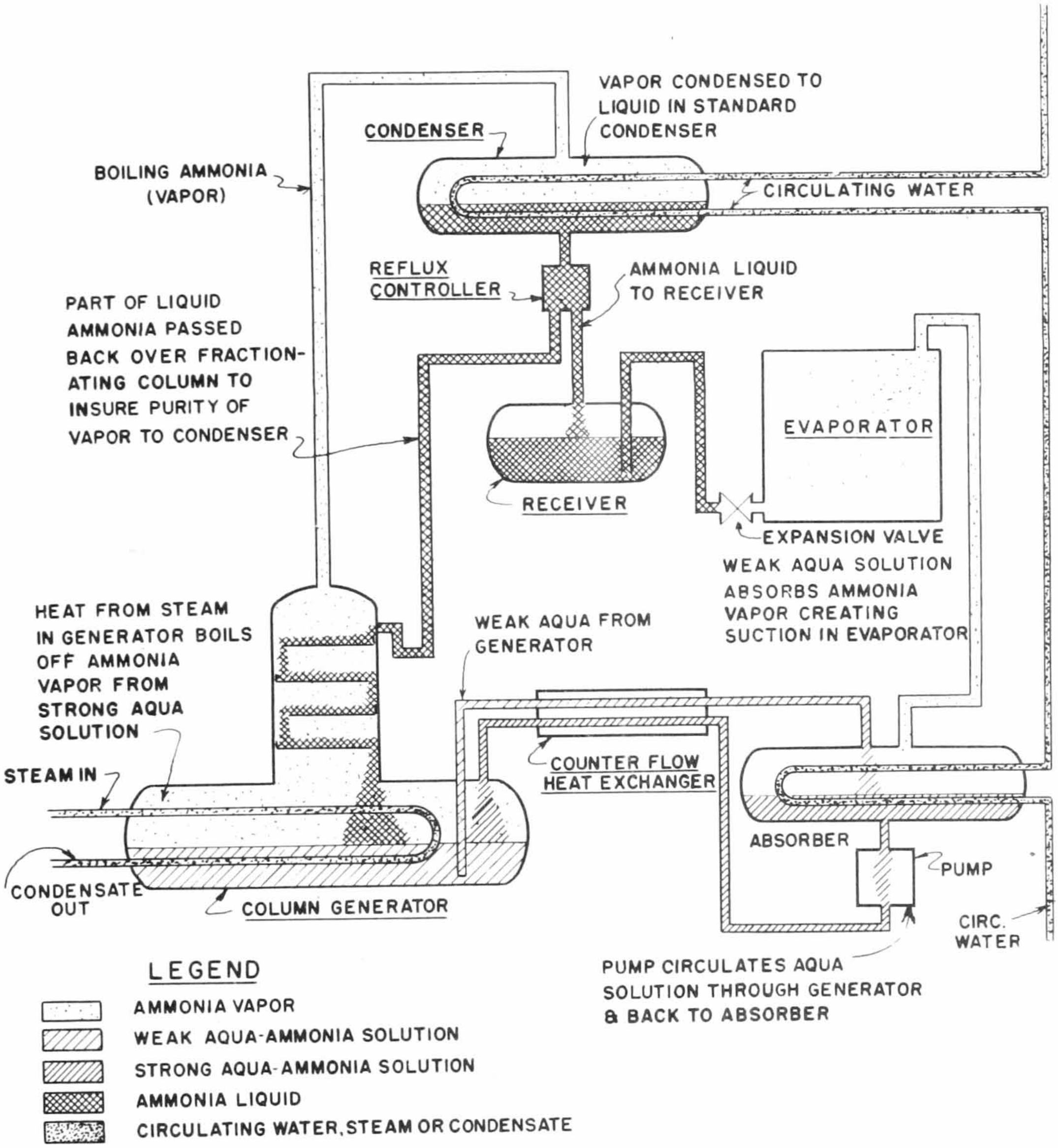

Figure 11.--Flow diagram of the absorption system. (Photo courtesy of Thermofreeze Company)

evaporating coils, if used in a direct system, or into a flooded or direct expansion-type brine cooler, if used in an indirect system (see Evaporators). The ammonia vapor that is formed by the extraction of heat in the evaporating coils flows from the evaporator into the absorber at a pressure corresponding to that of the refrigerant at the evaporator temperature. The absorber is similar to a shell-and-tube-type condenser, with water at temperatures of 50° to 70° F. circulating through the tubes. In the absorber, the ammonia vapor mixes with the "weak liquor", which flows from the generator through a heat exchanger to the absorber. The mixture of ammonia and water formed in the absorber is referred to as "strong liquor." The strong liquor is then pumped from the absorber through a heat exchanger to the generator by an aqua-ammonia pump. The heat exchanger serves two purposes: (1) to cool the weak liquor and (2) to add heat to the strong liquor. The generator consists of a cylindrical shell with heating coils immersed in the strong-liquor solution (aqua ammonia). Steam flows through the heating coils, adding heat to the strong liquor and causing a mixture of ammonia and water vapor to rise into the distilling column. As this mixture rises through the column, it passes through a series of baffles and is cooled by liquid ammonia that is admitted from the condenser to the top of the column. This liquid ammonia condenses the water contained in the mixture of ammonia and water vapor, causing it to return to the generator, where it mixes with the strong liquor. The weak liquor formed in the generator when the ammonia is driven off then flows to the absorber. The ammonia vapor leaving the distilling column goes to the condenser, where it is condensed to a liquid. The liquid ammonia then flows from the condenser to the receiver. A continuous supply of liquid ammonia also flows from the condenser through a reflux meter to the distilling column. The portion of ammonia entering the receiver now goes to the evaporator, and the cycle begins again.

To operate the absorption system efficiently, a perfect balance must be obtained at all times: that is, there must be the proper amount of weak and strong liquor flowing for each pound of ammonia going into the evaporator, the proper amount of cooling water flowing to the condenser and absorber, and the proper amount of steam flowing to the generator. The absorption machine is best suited for large, steady loads because of the difficulty in maintaining a perfect balance at variable loads.

Equipment

The equipment contained in the absorption system consists of a receiver, expansion valve, evaporator, absorber, aqua pump, generator, distillation column, reflux meter, and condenser. The function and design of the receiver, expansion valve, evaporator, and condenser are the same as that described in the compression system. The following discussion will deal with the functions and general design features of the equipment that is peculiar to the absorption system--the absorber, aqua pump, heat exchanger, generator, distillation column, and reflux meter.

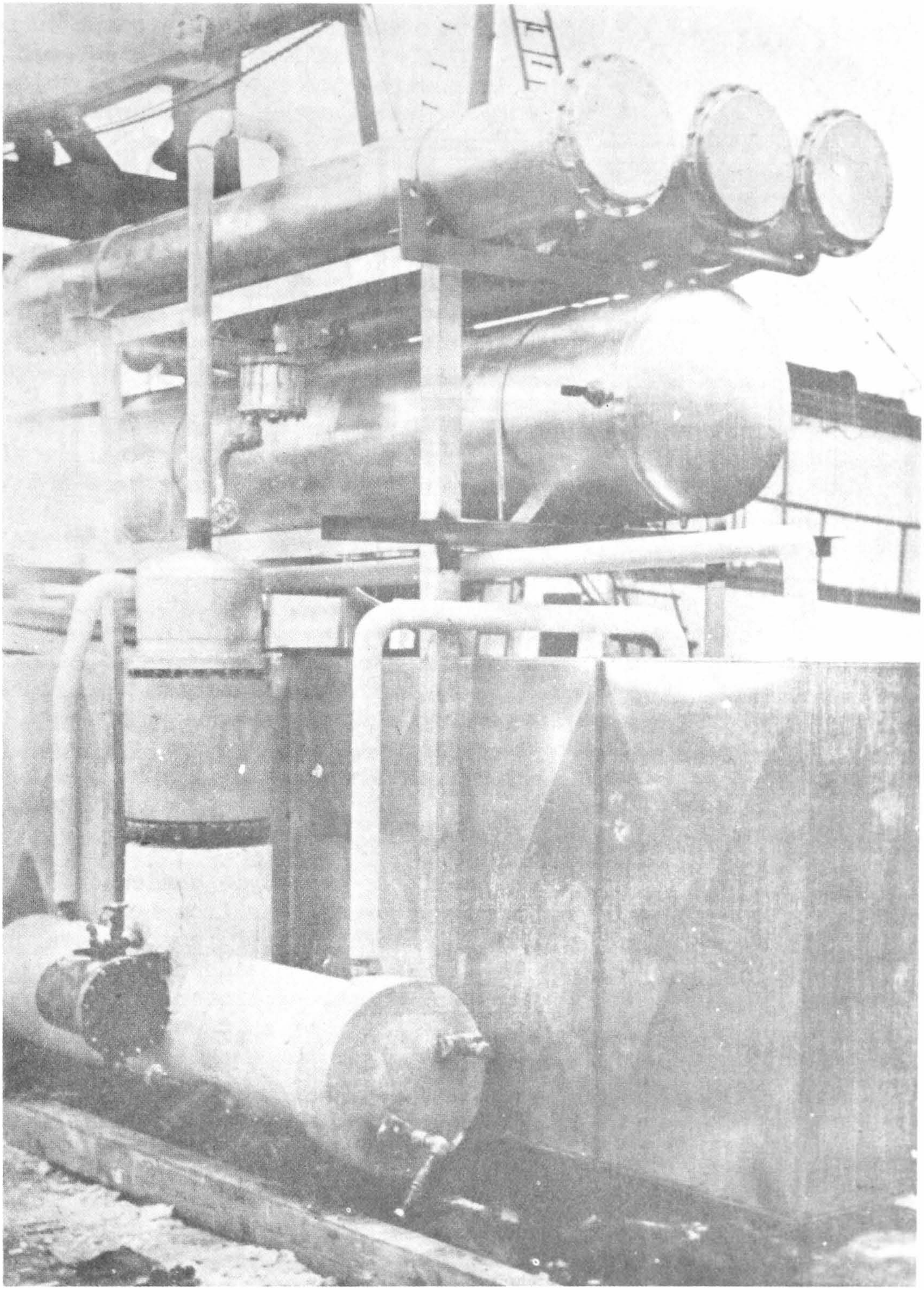

Figure 12.--View of an absorption refrigeration unit. (Photo courtesy of Thermofreeze Company)

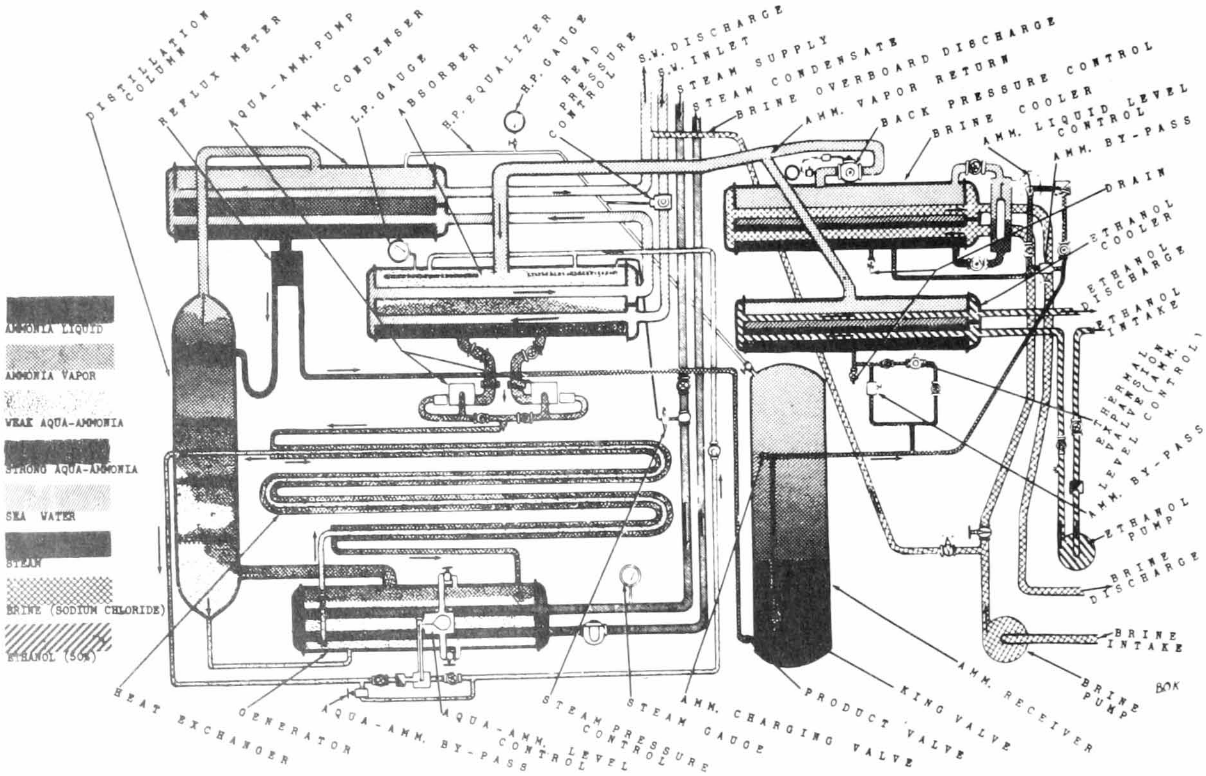

Figure 13.—Diagrammatic view of an absorption system as installed on a fishing vessel. (Drawn by Boris Knake, Fish and Wildlife Service)

Absorber

The absorber is similar to a shell-and-tube-type condenser. Fresh or salt water is circulated within the tubes making from 1 to 4 passes, depending on the specific design. The ammonia vapor from the evaporator mixes with weak liquor, which flows around the outside of the tubes within the vessel. The resulting solution, or strong liquor, is then pumped to the generator by the aqua pump. The fresh or salt water used as a cooling medium takes away the waste heat from the weak liquor, so that the proper absorption of ammonia from the evaporator will take place.

Aqua pump

The aqua pump can be of the reciprocating or rotary type. One reciprocating type in use is hermetically sealed, thereby preventing any ammonia leakage by the drive shaft. The aqua pump is used to pump the strong liquor from the absorber through the heat exchanger into the evaporator. It has been found that to obtain maximum performance, the aqua pump should pump approximately 8 pounds of strong liquor for each pound of ammonia passing through the evaporator. Some pumps have a variable-speed drive so that the speed of the pump can be reduced when operating at small loads and increased when operating at large loads. Either a direct- or alternating-current motor is used for driving the pump.

Heat exchanger

The function of the heat exchanger is to heat the strong liquor before it enters the generator, and at the same time, cool the weak liquor before it enters the absorber. This exchange of heat permits a substantial saving in steam input to the generator and also saves on the amount of cooling water required in the absorber.

The heat exchanger is usually of the counterflow type, which is merely an insulated pipe with a smaller pipe running inside the larger one. The weak aqua flows in one direction through one pipe, while the strong aqua flows in the opposite direction through the other. This type of heat exchanger is used because of its high rate of heat transfer.

Generator

The generator is a heat exchanger, usually of the shell-and-tube type. Steam flowing through the tubes at a pressure of 15 to 25 p.s.i.g. heats the strong liquor surrounding the tubes, causing the ammonia to vaporize and rise into the distillation column at a pressure of 120 to 160 p.s.i.g. The amount of steam supplied to the generator varies with the operating load.

Distillation column or analyzer

The distillation column--sometimes referred to as an analyzer--consists of a chamber containing, at spaced levels, a series of plates with small valves through which pass the vapors leaving the generator. From the condenser, cold liquid ammonia enters the top of the distillation column and drops in countercurrent flow to the vapors leaving the generator. In so doing, the cold liquid ammonia--owing to its affinity for water--strips the water vapor from the hot ammonia vapor. The result is that approximately 98-percent-pure ammonia vapor leaves the column and flows to the condenser.

Reflux meter

The reflux meter consists of an orifice in a pipe connecting the bottom of the condenser to the top of the distillation column. This orifice permits a steady flow of liquid ammonia to the distillation column, thereby enabling it to function properly.

COMPARISON BETWEEN THE ABSORPTION AND COMPRESSION SYSTEMS

The selection of an absorption or a compression machine depends largely on the type of service desired and on the conditions that prevail where the machine is to be used. The following summary of the advantages of the absorption system over the compression system gives an indication of the types of applications for which the absorption system should be considered.

1. The only moving part of the absorption system is the aqua pump; therefore, the operation is relatively quiet and subject to little mechanical wear.

2. Waste or exhaust steam can economically be utilized to supply the energy necessary to operate the absorption system.

3. For low-temperature applications, the absorption machine can operate with little decrease in capacity, whereas the capacity of a compression machine decreases greatly at low evaporator temperatures.

4. At reduced loads, the absorption unit is almost as efficient as at full capacity.

5. Liquid refrigerant leaving the evaporator will slightly unbalance the absorption system; whereas in a compression system, this liquid refrigerant will cause serious damage to the compressor.

6. The absorption system will operate at capacities of well over 1,000 tons, whereas the largest single-compression unit does not exceed 1,000 tons.

7. When used in large plants, the absorption equipment can be located outside the cold-storage building.

The following summary of the advantages of the compression system over the absorptión system gives an indication of the types of applications for which the compression system should be considered:

1. At high evaporating temperatures, a compression machine will operate more efficiently than will an absorption machine.

2. For evaporator temperatures of 0° F. and above, the initial cost of the compression machine is cheaper.

3. The compression machine requires less maintenance because of the type of equipment used.

4. The compression machine is positive acting, with the performance depending on the compressor, condenser, and evaporator; whereas the

performance of the absorption machine depends on the proper proportioning of all its parts, the condition of each piece of equipment, and the skill of the operating engineer.

5. The compression machine can economically use refrigerants such as ammonia, carbon dioxide, Freon 12, Freon 22, and methyl chloride; whereas the absorption machine is limited to the use of ammonia, which has many undesirable properties.

6. The compression machine takes up considerably less space.

7. The compression machine is manufactured in sizes from 1/4-ton to 1,000-tons capacity, whereas there are very few companies that make absorption units smaller than 100-tons capacity.

8. The compression machine can be driven by an electric motor, diesel engine, gasoline engine, or steam engine; whereas only steam can be used with the absorption machine.

9. The compression machine has a wide range of applicability that has been proven hundreds of thousands of times in such installations as refrigerated vessels, sharp freezers, blast freezers, and plate freezers.

EVAPORATORS

The evaporator is designed to effect an efficient heat transfer between the cooling medium and the material to be cooled. Evaporators have been designed to cool many materials; however, this discussion will be concerned only with the principal uses of the evaporator in relation to the cold storage and freezing of fishery products.

Classification

There are two general types of evaporators: the direct-expansion evaporator, and the indirect-expansion evaporator.

In the direct-expansion type, the refrigerant itself, expanding through the evaporator, absorbs the heat from the material to be cooled, whereas in the indirect-expansion type, refrigerated brine, circulating through the "evaporator" tubes, absorbs the heat from the material to be cooled.

The following is a discussion of these two types of evaporators, with emphasis on their relative merits.

Direct-expansion evaporators

In the direct-expansion evaporator, the refrigerant is used as a cooling medium to cool air, water, brine, or food product. An example of a direct-expansion evaporator is a cold-storage room with pipe coils attached to the ceiling. The refrigerant flowing through the pipes absorbs the heat from the air, which is circulated around the outside of the pipe coils by natural convection currents. Thus, the air in the room is cooled as a result of the mixture of liquid and gaseous refrigerant in the pipes changing to a vapor.

There are two general types of systems employed in direct-expansion evaporators: the dry-expansion system, and the flooded system.

In the so-called dry-expansion system, the evaporator coils are only partially filled with refrigerant. In the flooded system, a cylindrical vessel, which is referred to as an accumulator, is located at the end of the evaporator coils. The accumulator is maintained with a sufficient amount of refrigerant to keep the evaporator coils flooded with "boiling" refrigerant at all times. The top of the accumulator is connected to the compressor suction, in the compression system, or to the absorber, in the absorption system.

The flooded system is used in preference to the dry-expansion system in large commercial installations, where excessive pressure drops are incurred because of the length of the evaporator coils. The dry-expansion system is used for small evaporators, where a large pressure drop is not encountered. The rate of heat transfer is higher in the flooded system than in the dry-expansion system.

Indirect-expansion evaporators

In the indirect-expansion evaporator, a refrigerated brine solution is pumped through the so-called evaporator coils, thereby cooling the surrounding atmosphere or product. Calcium-chloride brine is generally used in a closed system, where the brine is circulated within the evaporator coils on which the fish are frozen, while a sodium-chloride brine is used in an open system, where the fish are frozen by direct contact with the brine.

A brine cooler (the true evaporator) is used to refrigerate the brine solution. There are two types of commercially available brine coolers: the direct-expansion brine cooler, and the flooded brine cooler.

In the direct-expansion brine cooler, the brine is cooled by expansion of the refrigerant through a series of pipe coils located within the shell of the cooler. The brine is circulated around the outside of the pipe coils by a brine pump. The baffles in the cooler are arranged in such a manner that the brine can make from 1 to 6 passes.

In the flooded-type brine cooler, the brine is circulated through a series of tubes located within a cylindrical vessel. The brine usually makes from 1 to 4 passes. The exact number of passes varies with the specific type of cooler design. The brine is cooled by "boiling" liquid refrigerant surrounding the outside of the tubes. The refrigerant is maintained at a level of approximately 2/3 the diameter of the cooler, by a float-type expansion valve.

Application

The specific design of the evaporator depends to a large extent on the types of application for which it is to be used. In the freezing of fishery products, we are concerned with only those evaporators designed for use in cold-storage plants and quick freezers. The following is a general description of some of the commercial evaporators used for the cold storage and freezing of fishery products.

Cold-storage rooms

There are three commercial types of evaporators used in cold-storage plants. The first two types consist of pipe coils or finned pipe coils suspended from the ceiling of the room (figure 14). Refrigerant or brine is circulated through the coils, thereby cooling the surrounding atmosphere to the proper temperature. The third type is a blower-type cooling unit. This evaporator consists of a housing that contains coils of finned pipe and one or more fans located so as to force the air in the room around the coils (figure 15). A refrigerant such as ammonia or Freon 12 is generally used as the cooling medium, but in some units, cold brine has also been used. A more detailed description of the commercial evaporators used in cold-storage plants is given in the section entitled "Cold Storage Design."

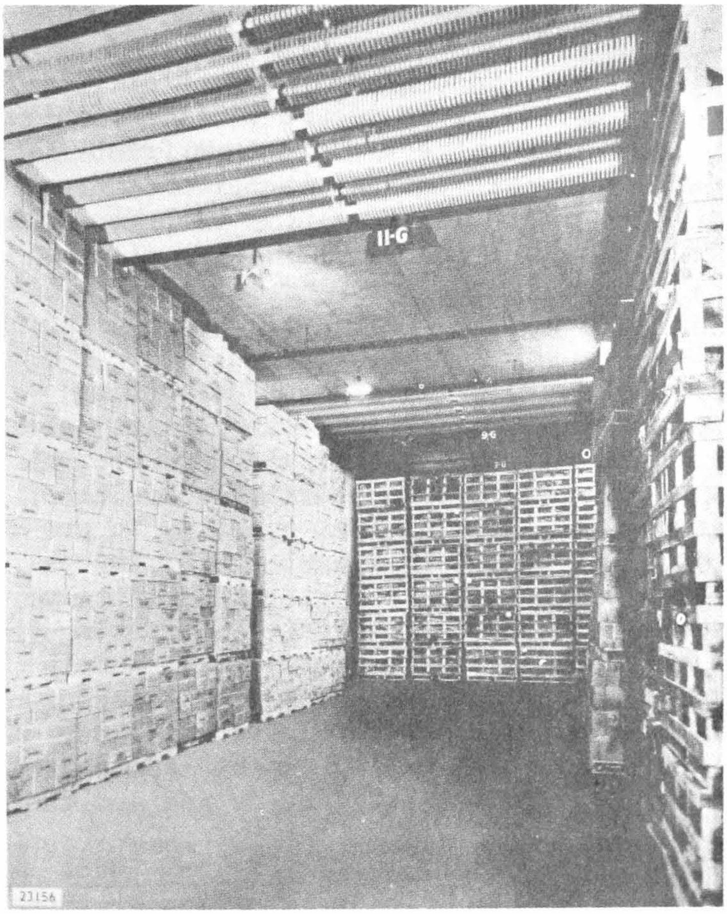

Figure 14.--Finned pipe coils in a cold-storage room. (Photo courtesy of York Corporation)

Quick freezers

Fishery products are frozen in evaporators referred to as sharp, plate, blast, or immersion freezers. In the sharp, plate, or blast freezers, either a refrigerant or a brine solution can be used in the evaporator, whereas in the immersion freezer, a brine solution is generally used. These freezers are described in the section entitled "Refrigeration Requirements and Freezing Methods" (section 3).

Figure 15.—Blower-type cooling units in a refrigerated room. (Photo courtesy of Blount Seafood Corporation)

Defrosting of Evaporators

The accumulation of frost on evaporator coil surfaces will result in a reduction of heat transfer between the refrigerant and the air or product being cooled. Defrosting has sometimes been accomplished by shutting off the flow of refrigerant to the evaporator and opening the cold-storage-room door, thereby admitting warm air to the inside of the room. This procedure is not recommended because of (1) the possibility of product thawing, (2) the large amount of time required, and (3) the excessive amount of moisture admitted. To remove frost accumulation from evaporators quickly and yet prevent the product from changing temperature, two types of defrosting systems have been developed: water defrosting, and hot-gas defrosting.

Water defrosting

Water defrosting is a cheap and efficient method for defrosting blower-type cooling units that produce evaporator temperatures from above freezing to -40° F. Defrosting is accomplished by circulating water at a temperature of about 50° F. through spray nozzles located above the evaporator coils. The nozzles should be in sufficient quantity to provide adequate water to defrost the coils completely in 5 to 10 minutes. At temperatures below 0° F., a flow of 3 gallons of water per minute per square foot of coil surface area is required for effective defrosting. When defrosting, the fan should be stopped, and the refrigerant supply to the coils should be closed off. To prevent the water from freezing, a suitable drain, with a trap located outside the refrigerated space, must be installed to carry off the water. Defrosting can be accomplished manually, or automatically by means of a suitable timing device and the proper arrangement of valves and piping.

Water defrosting is not suitable for use with coils having gravity circulation of air, since the spray-header system would be too extensive.

Hot-gas defrosting

Hot-gas defrosting is accomplished by causing the hot gas discharged from the compressor to flow through the evaporator coils. If two or more evaporator coils are served by one compressor, the piping can be arranged so that the hot gas from the compressor discharge will flow through a first set of evaporator coils that are connected, in turn, to a second set of evaporator coils. The hot gas, in flowing through the first set of coils, provides the necessary heat for defrosting and, as a result, changes from a hot gas to a cold liquid. This liquid, in flowing through the second set of coils, absorbs heat from the medium surrounding them and, in so doing, changes back to a gas. On systems in which only one evaporator is used, an additional heat exchanger must be installed to prevent liquid refrigerant from entering the compressor. Hot-gas defrosting can be accomplished manually or automatically by means of a suitable timing device and the proper arrangement of valves and piping.

REFRIGERANTS

A gas, to be suitable for use as an efficient refrigerant, must have a low boiling temperature and a high latent heat of vaporization[3/] at atmospheric pressure, in addition to many other properties. All other factors being equal, a lower boiling point and a higher latent heat of vaporization will result in a greater absorption of heat per pound of refrigerant flowing through the evaporator. However, good thermodynamic properties do not necessarily determine the ideal refrigerant. Such a refrigerant, in addition to having good thermodynamic properties, must be nonexplosive, nontoxic, noninflammable, noncorrosive, nonirritating, noninjurious to foods, suitable for mechanical application, practically odorless, easily obtainable, inexpensive, efficient, and economical to use.

Many types of refrigerants have been tried since the inception of the mechanical refrigeration system. In recent years, new refrigerants closely resembling the ideal refrigerant have been developed. The refrigerants used in commercial refrigeration installations today are ammonia, Freon 12, carbon dioxide, methyl chloride, Freon 11, and Freon 22. The following discussion will be concerned with the advantages and disadvantages that should be considered in the selection of these refrigerants for use in a commercial refrigeration plant.

Ammonia

Ammonia is a very economical and efficient refrigerant because of its low boiling point and high latent heat of vaporization at atmospheric pressure. It is used extensively in large commercial and

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3/ The latent heat of vaporization, expressed in British thermal units (B.t.u.'s), is the heat necessary to cause 1 pound of refrigerant to change from a liquid to a gas at its boiling temperature.

industrial refrigeration plants. In small- and medium-sized commercial plants, ammonia is being replaced to a great extent by Freon 12 and Freon 22 because of their many advantageous physical properties.

Advantages of Ammonia

Ammonia (1) has excellent thermodynamic properties, (2) is a very efficient refrigerant, (3) is neutral to iron and steel, (4) is ideally suited for use in compression and absorption systems, and (5) has low initial cost.

Disadvantages of Ammonia

Ammonia (1) requires high pressure, (2) attacks most alloys, (3) diminishes in efficiency with overheating, (4) is highly toxic, and (5) adversely affects food, water, and plant life.

Freon 12 (Dichlorodifluoromethane)

Freon 12 is an economical and efficient refrigerant. It is not, however, as efficient as ammonia, because of its higher boiling point and its lower latent heat of vaporization at atmospheric pressure. The boiling temperature of ammonia at atmospheric pressure is -28° F., and its latent heat of vaporization is 589. B.t.u. per pound. In comparison, the boiling temperature of Freon 12 at atmospheric pressure is -21° F., and its latent heat of vaporization is about 72 B.t.u. per pound.

Freon 12 is used in lieu of ammonia in many small commercial refrigeration plants because of its nontoxic properties and the compactness of the condensing unit required. It is as near a perfect refrigerant as is presently obtainable. The following are the advantages and disadvantages of Freon 12 that should be considered in its use in a commercial refrigeration plant:

Advantages of Freon 12

Freon 12 (1) has good thermodynamic properties, (2) has moderate operating pressures, (3) is noncorrosive, (4) is nontoxic, (5) is noninflammable, (6) is nonexplosive, (7) is practically odorless, and (8) does not affect food or plant life.

Disadvantages of Freon 12

Freon 12 (1) requires that care be taken to remove all air and moisture, as otherwise, corrosion and freezing of the expansion valve might result; (2) has high solvent action on rubber; (3) is not as economical to use as is ammonia; and (4) has a high initial cost.

Carbon Dioxide

Carbon dioxide is used extensively in refrigeration systems in large industrial plants. It operates under extremely high pressures, requiring close attention. The following are the advantages and disadvantages that should be considered in the use of carbon dioxide in a commercial refrigeration plant:

Advantages of Carbon Dioxide

Carbon dioxide is (1) cheap, (2) easily obtainable, (3) noninflammable, (4) nonpoisonous, (5) noncorrosive, (6) odorless, (7) requires small compressor and small piping, and (8) does not affect food or plant life.

Disadvantages of Carbon Dioxide

Carbon dioxide (1) operates under extremely high pressure, which may cause leakage at joints and stuffing boxes; (2) requires high motor power; and (3) requires a large continuous supply of cold condensing water for efficient operation.

Methyl Chloride

Methyl chloride is extensively used in commercial reciprocating and rotary compressors up to 25 hp. The following are the advantages and disadvantages that should be considered in the use of methyl chloride in a commercial plant:

Advantages of Methyl Chloride

Methyl chloride (1) can be used at low operating pressures; (2) does not affect metals; (3) can readily be used in air-cooled condensers; and (4) is odorless, but can be tainted with a warning agent to give notification of leaks.

Disadvantages of Methyl Chloride

Methyl chloride (1) is inflammable; (2) dissolves lubricants; (3) absorbs moisture from the air, resulting in hindering or stopping of the action of the expansion valve; (4) adversely affects food and plant life; (5) is highly toxic; and (6) is explosive.

Freon 11 (Trichloromonofluoromethane)

The physical properties of Freon 11 are similar to those of Freon 12. Freon 11 is used in large commercial centrifugal compressors where a large volume of refrigerant is delivered and a refrigerant of high molecular weight is required.

Freon 22 (Monochlorodifluoromethane)

The physical properties of Freon 22 are also similar to those of Freon 12. Freon 22, however, has a lower boiling temperature at atmospheric pressure, and it operates at higher discharge pressures than does Freon 12. It is used largely in place of Freon 12 for low-temperature applications.

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SECTION 3

REFRIGERATION REQUIREMENTS AND FREEZING METHODS

By Joseph W. Slavin, Refrigeration Engineer *

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This section will discuss (1) the refrigeration requirements and (2) the methods used in the freezing of fish.

REFRIGERATION REQUIREMENTS

Numerous factors control the freezing of fishery products. These factors may be divided into two classifications: (A) those that are controlled by the product and that should be considered in the preparation and packaging of a product for freezing and (B) those that are controlled by the refrigeration system and that should be considered in the installation and application of a refrigeration system for freezing the fishery product.

Package and Product Factors in Freezing

In the production of a quick-frozen product, such factors as (a) packaging, (b) thickness of product, and (c) temperature of product prior to freezing are important.

Packaging

Sufficient study is not always given to the type and size of package for a particular type of product. It is quite common for a contract to be signed for the delivery of a large amount of packages, even though a definite method of freezing has not been decided on.

To freeze packaged fishery products quickly and efficiently, thought should be given to the thickness of the package, the insulating effect of the package, and the surface area of the package. The following is a brief discussion of these items and how they effect the freezing of packaged fishery products.

Thickness of package.--The optimum thickness of package for a particular product is that size which will enable the production of a high-quality product at a minimum production cost. The use of a thin package will result in (1) faster product freezing, (2) lower freezing-power costs, (3) higher handling costs, and (4) higher packaging-material costs; the use of a thick package will result in (1) slower product freezing, (2) higher freezing-power costs, (3) lower handling costs, and (4) lower packaging-material costs.

Tests conducted at the U. S. Fish and Wildlife Service laboratory, East Boston, Massachusetts, showed that, by freezing packaged fish fillets in the thinner packages, (1) a faster freezing rate and (2) a larger volume of frozen-product output is obtained (Slavin 1955). These tests also showed that less electrical energy is required to freeze packaged fish fillets in the thinner packages than in the thicker ones.

It is very important that the top of the product comes in contact with the package, thereby eliminating a dead-air space, which would increase the freezing time and refrigeration costs considerably.

Insulating effect of package.—A packaging material with a low moisture-vapor permeability is used in the manufacture of most packages for fishery products. This material protects the product while it is in cold storage by preventing excessive dehydration. However, the material also has an insulating effect, which increases the freezing time and the freezing costs. The rate of heat transfer through the packaging material is inversely proportional to its thickness. Therefore, the ideal packaging material is thin enough to produce the maximum possible freezing rate, and yet thick enough to withstand the abuse that the package will encounter.

Surface area of package.--For two packaged products of the same weight, the package with the larger surface area will freeze the faster (see "Thickness of package"). The surface area of a package is also important in the freezing of the product, because of its relation to the size of the freezer. Therefore, the package surface area should be so determined that there will not be any unused product space in the freezer after it has been loaded to its maximum capacity. This determination of proper surface area is very important in the operation of a multiplate compression freezer.

Thickness of product

The greater the thickness of a product in relation to the surface area in contact with the cooling medium, the greater is the time required for the center to freeze. The time required to freeze, however, is not directly proportional to the product thickness, as might be assumed. Tests conducted by the U. S. Fish and Wildlife Service laboratory, East Boston, Massachusetts, indicated that the time required to freeze packaged fish fillets in a plate freezer is approximately directly proportional to the square of the package thickness, in accordance with the following equation:

$$\frac{t_x}{t} = \frac{d_x^2}{d^2} \quad \text{or} \quad t_x = \frac{t d_x^2}{d^2}$$

where t_x is the time required to freeze a package of thickness d_x, if t is the time required to freeze a package of thickness d.

Thus, if it takes 180 minutes (t) to freeze packaged fish fillets 2 inches thick (d), it will take about 281 minutes (t_x) to freeze packaged fillets 2½ inches thick (d_x), since

$$t_x = \frac{(180)(2.5)^2}{2^2}$$

$$t_x = 281 \text{ minutes}$$

The actual freezing time, in experimental tests, varied from 260 to 280 minutes, depending on the air space in the package and the plate temperature.

To apply the foregoing formula, the user should keep all other factors the same--such as plate temperature, packaging material, and air space in the package: the thickness of the packaged product should be the only variable. Increased air space within the package or a higher plate temperature will increase the freezing time considerably.

Temperature of Product

The heat contained within a product is commonly given in British thermal units (B.t.u.), 1 B.t.u. being the amount of heat required to raise the temperature of 1 pound of water 1 degree Fahrenheit. The amount of heat, expressed in B.t.u., required to raise the temperature of 1 pound of a material 1 degree Fahrenheit, is called the specific heat of the material.

Since the specific heat of water and of ice differ and since the flesh of fish is composed of about 80 percent water, fish have two different specific heats: the one, before freezing, and the other, after freezing 1/. The specific heat, expressed in B.t.u. per pound of most fishery products before freezing is 0.8; the specific heat after freezing is 0.4.

The freezing process involves the removal of two types of heat: sensible heat and latent heat. Sensible heat involves a change of product temperature without a change of state (as when the temperature of water is lowered from 40° F. to 32° F.), whereas latent heat involves a change of product state without a change of temperature (as when water freezes to ice at 32° F. or when ice melts to water at 32° F.). The latent heat is largely removed from most fishery products between the temperatures of 29° and 27.5° F. The latent heat of most fishery products has been calculated to be approximately 115 B.t.u. per pound. The formula given below is used to determine the amount of heat that must be extracted to freeze a certain quantity of fish.

If:

Q = total heat to be extracted from product (B.t.u.).

T_1 = temperature of product prior to freezing (degrees Fahrenheit).

T_2 = temperature at which the latent heat is removed from product (degrees Fahrenheit) 2/.

1/ This simplified concept of a constant heat above and below freezing, although not in complete accord with experimental observations, is sufficiently accurate for most freezing calculations.

2/ Fish does not freeze at a definite temperature, but rather over a range of temperatures (see Fishery Leaflet 429, section 1). However, T_2 can usually be taken as 28° F., for most calculations.

T_3 = final temperature of frozen product (degrees Fahrenheit).
c = specific heat of product prior to freezing (B.t.u. per pound per degree Fahrenheit).
c_i = specific heat of product, after freezing (B.t.u. per pound per degree Fahrenheit).
L = latent heat of product (B.t.u. per pound).
W = weight of product being frozen (pounds).

Then:

$$Q = W(T_1 - T_2)c + WL + W(T_2 - T_3)c_i$$
$$= W[(T_1 - T_2)c + L + (T_2 - T_3)c_i]$$

The following problem and solution show how this formula is applied.

Problem 1:

Determine how much heat, expressed as B.t.u., must be extracted to lower the temperature of 1,000 pounds of fish from 40° to 0° F.

Solution to problem 1:

$$Q = W[(T_1 - T_2)c + L + (T_2 - T_3)c_i]$$
$$= 1{,}000\,[(40-28)0.8 + 115 + (28-0)(0.4)]$$
$$= 136{,}000 \text{ B.t.u. (rounded off)}$$

W = 1,000 lbs.
T_1 = 40° F.
T_2 = 28° F.
T_3 = 0° F.
c = 0.8 B.t.u./lb./°F.
c_i = 0.4 B.t.u./lb./°F.
L = 115 B.t.u./lb.

Effect of higher product temperature on refrigeration capacity.--The following problem and its solution show the effect of a high product temperature on refrigeration capacity.

Problem 2:

Assuming that 136,000 B.t.u. must be extracted to lower the temperature of 1,000 pounds of fish from 40° to 0° F., how much more heat would have to be extracted if the initial temperature had been 60° instead of 40° F.?

Solution to problem 2:

$$Q_{12} = W(T_1 - T_2)c$$
$$= (1{,}000)(60-40)(0.8)$$
$$= 16{,}000 \text{ B.t.u.}$$
$$\frac{(16{,}000)(100)}{136{,}000} = 11.8\%$$

$W = 1{,}000$ lbs.
$T_1 = 60°$ F.
$T_2 = 40°$ F.
$c = 0.8$ B.t.u./lb./°F.

This calculation indicates that by prechilling the fish from 60° to 40° F. in a chill room or in ice immediately prior to freezing, 11.8 percent less refrigeration capacity will be required.

Effect of higher product temperature on time required to freeze product.--The freezing time of a particular product is determined by a combination of many factors in addition to the product temperature, such as packaging, refrigeration capacity, method of heat transfer between product and cooling medium, and temperature of cooling medium. These factors are more or less interrelated, and it would be very difficult to state exactly how much any one factor would affect the freezing time. However, with other factors being equal, the product with a higher initial temperature will take longer to freeze. This fact can be illustrated by referring to the solution of problems 1 and 2. Assuming that the refrigeration machine used to freeze the fish in problem 1 was designed to lower the temperature of 1,000 pounds of 1-inch-thick packaged fillets of fish from 40° F. to 0° F. in 1 hour (the required refrigeration capacity is 136,000 B.t.u. per hour), it would take at least 1 hour and 7 minutes to freeze the fish in problem 2, which had an initial temperature of 60° F.

Thus, both the refrigeration capacity requirements and freezing time can be decreased by precooling the product before freezing.

Refrigeration-System Factors in Freezing

The factors that are controlled by the refrigeration system and that affect the freezing time and cost of freezing are (1) the refrigeration capacity, (2) the temperature of the cooling medium, and (3) the method of heat transfer between product and cooling medium. As these factors are interrelated, a composite study of the exact effect of each one on the freezing of fish would be lengthy. Therefore, the following discussion will deal with only the general effects of these factors.

Refrigeration Capacity

The capacity of a refrigeration system is expressed in tons of refrigeration. One ton of refrigeration, given in B.t.u., is equal to the heat that must be removed in freezing 2,000 pounds of water at 32°F.

to ice at 32° F., in a 24-hour period. Expressed in B.t.u., 1 ton of refrigeration is equal to the removal of 288,000 B.t.u. per 24 hours (2,000 x 144), 12,000 B.t.u. per hour, or 200 B.t.u. per minute.

A refrigeration system designed to freeze a certain quantity of fish must have sufficient capacity, in tons of refrigeration, to equal the heat in B.t.u. that must be withdrawn from the product within a predetermined time, to cool it from its temperature prior to freezing to a temperature suitable for preservation (about 0° F.).

For example: A 1-pound package of 1-inch-thick packaged fillets can be cooled from 40° to 0° F. in a period of 1 hour in a plate freezer with refrigerant at a temperature of -24° F. flowing through the plates. To cool 1,000 1-pound packages of this product within a period of 1 hour, the refrigeration system must have a heat-withdrawal capacity equal to 1,000 times the B.t.u. required to cool the single 1-pound package.

A producer who, by experimentation, finds that a particular product will freeze within a certain period of time with a refrigerant at a certain temperature flowing through the evaporator can readily determine the refrigeration capacity requirement in B.t.u. per hour necessary to freeze any predetermined amount of this product, by the formula given below, where:

- q = heat per unit of time to be extracted from product (B.t.u. per hour).
- t = time required to freeze product (hours).
- T_1 = original temperature of product prior to freezing (degrees Fahrenheit).
- T_2 = temperature at which the latent heat is removed from the product (degrees Fahrenheit).
- T_3 = final temperature of frozen product (degrees Fahrenheit).
- c = specific heat of product prior to freezing (B.t.u. per pound per degree Fahrenheit).
- c_i = specific heat of product, after freezing (B.t.u. per pound per degree Fahrenheit).
- L = latent heat of product (B.t.u. per pound).
- W = weight of product to be frozen (pounds).

$$q = \frac{1}{t}\left[W(T_1 - T_2)c + WL + W(T_2 - T_3)c_i\right]$$

or

$$q = \frac{W}{t}\left[(T_1 - T_2)c + L + (T_2 - T_3)c_i\right]$$

The following two problems illustrate how to determine the necessary refrigeration capacity requirement in B.t.u. per hour if the freezing time, the product temperature before and after freezing, the temperature of the refrigerant in the evaporator, and the pounds of fish to be frozen are known:

Problem 1:

Experimental data show that a 1-inch-thick 1-pound package of fish fillets can be cooled from 40° to 0° F. in a period of 1 hour with refrigerant at a temperature of -24° F. circulating through the plates in a plate freezer. What is the capacity requirement of the refrigeration system in B.t.u. per hour necessary to freeze 1,000 pounds of this product?

Solution to problem 1:

$$q = \frac{W}{t}\left[(T_1 - T_2)c + L + (T_2 - T_3)c_i\right]$$

$$= \frac{1,000}{1}\left[(40-28)0.8 + 115 + (28-0)0.4\right]$$

$$= 136,000 \text{ B.t.u./hr,}$$

q = B.t.u./hr.
t = 1 hour
T_1 = 40° F.
T_2 = 28° F.
T_3 = 0° F.
c = 0.8 B.t.u./lb./°F.
c_i = 0.4 B.t.u./lb./°F.
L = 115 B.t.u./lb.
W = 1,000 lbs.

Problem 2:

Experimental data show that a 2-inch-thick 10-pound package of fish fillets can be cooled from 40° to 0° F. in 3 hours with refrigerant at a temperature of -24° F. circulating through the plates in a plate freezer. What is the capacity requirement of the refrigeration system in B.t.u. per hour necessary to freeze 1,000 pounds of this product?

Solution to problem 2:

$$q = \frac{W}{t}\left[(T_1 - T_2)c + L + (T_2 - T_3)c_i\right]$$

$$= \frac{1,000}{3}\left[(40-28)0.8 + 115 + (28-0)0.4\right]$$

$$= 45,300 \text{ B.t.u./hr.}$$

q = B.t.u./hr.
t = 3 hours
W = 1,000 lbs.
T_1 = 40° F.
T_2 = 28° F.
T_3 = 0° F.
c = 0.8 B.t.u./lb./°F.
c_i = 0.4 B.t.u./lb./°F.
L = 115 B.t.u./lb.

In addition to the refrigeration capacity requirement necessary for product freezing, as expressed in problems 1 and 2, additional refrigeration capacity is necessary to compensate for wall-heat gain, air changes, and miscellaneous heat gains. For the calculation of these additional heat gains, see section 1 of this leaflet.

The capacity of a refrigeration system to be used in product freezing is equal to the product-freezing requirement in B.t.u. per hour plus the additional heat gains mentioned above. To select a suitable compression or absorption system for freezing a predetermined amount of fish within a certain period of time, the refrigeration-capacity requirement in B.t.u. per hour at the required evaporator temperature must be known.

Temperature of the Cooling Medium

The rate of heat transfer between a particular product and the cooling medium is proportional to the difference between the temperature of the product and that of the cooling medium. Therefore a lower cooling-medium temperature will result in removal of the heat from the product at a faster rate, thereby decreasing the product-freezing time. The actual decrease in product-freezing time depends on the particular type of product and the freezing method employed.

To produce a lower cooling-medium temperature, the refrigeration machine must operate at a lower suction pressure, resulting in increased compressor horsepower per ton of refrigeration produced. As the cooling-medium temperature becomes progressively lower, the ratio of compressor horsepower to tons of refrigeration produced increases considerably, resulting in higher operating costs.

As mentioned above, a low cooling-medium temperature results in (1) faster freezing and (2) increased refrigeration costs. Therefore, to freeze a product quickly and economically, a proper balance is necessary between the cooling-medium temperature, the freezing rate, and the cost of refrigeration per pound of product frozen. In the selection of a suitable cooling-medium temperature, the type of product to be frozen and the specific design features of the freezer must also be given the utmost consideration. In planning an installation to obtain fast product freezing economically, a refrigeration engineer should be consulted.

Method of Heat Transfer between Product and Cooling Medium

To obtain rapid product freezing, a high rate of heat transfer [3/] must exist between the product being cooled and the cooling medium. In addition, the size of the evaporator and the available refrigeration capacity must be adequate so as to maintain the cooling medium at its proper temperature.

The four principal methods of freezing fish are (1) placing the product on pipe-coil shelves that have either brine or refrigerant flowing through the pipes, as in a sharp freezer, (2) placing the product between two refrigerated plates, as in a multiplate freezer, (3) circulating cold air around the product, as in a blast freezer, or (4) immersing the product in a cold brine solution, as in an immersion freezer.

If, in these four principal methods of freezing, it were possible to keep all other factors such as packaging, thickness of product, temperature of product, and the temperature of the cooling medium constant, the rate of freezing would depend on the ability of the cooling medium to withdraw the necessary heat out of the product. The ability of a particular cooling medium, when cooled to a given temperature, to absorb the heat from the product is largely dependent on whether or not the medium is a good or poor conductor.

The following discussion will be concerned with the effect of the various cooling media used in sharp, multiplate, blast, and immersion freezers on the product-freezing time.

Sharp freezer.--In the sharp freezer, the fish to be frozen are placed on shelves comprised of pipe coils. In most cases, metal pans, large sections of galvanized sheet iron, or iron plates are placed on the coils, to hold the fish. A refrigerant such as ammonia, Freon 12, or cold brine is circulated through the pipes to provide the necessary refrigeration effect. The pans, sheets, or plates on which the fish are placed act as a conductor, thereby allowing the heat to flow between the points of contact of the fish and pan and the pan and pipes. The heat transfer between the refrigerant in the pipes and the fish, at the point of contact, is relatively high; however, the over-all heat-transfer rate is quite low because of the extensive surface area of the product that is not in contact with the pipes and that is cooled largely by the natural circulation of cold air within the freezer room. The result is slow and uneven product-freezing. If sufficient refrigeration capacity is available, fans can be used to circulate the cold air over the products, thereby increasing the freezing rate.

3/ The factors affecting heat transfer in an evaporator used to cool air or brine are too numerous to mention here. References listed at the end of the section give further information on this problem.

Multiplate freezer.--In the plate freezer, the upper and lower surfaces of the products are in contact with refrigerated aluminum plates. These plates act as a conductor that allows the heat to flow freely between the plates and the product. This ready transfer of heat results in fast and efficient freezing. However, if a positive contact between plates and product is not maintained, owing to air space in the package, or to other causes, slow and inefficient freezing will result. In freezing packages that are over 3 inches thick, the surface area not in contact with the refrigerated plates becomes appreciably large relative to the surface in contact with the plates, thereby greatly increasing the freezing time of the product. A large bulky product would be frozen more suitably in a blast freezer, where all exposed surfaces are scrubbed by circulating cold air.

Blast freezer.--In the blast freezer, air at a low temperature is circulated, by means of a fan, around the products to be frozen. To achieve rapid freezing, the air should be circulated over the products at velocities of over 500 feet per minute and should be of sufficient volume so that it does not rise more than 10 degrees in temperature while cooling the product. The velocity of the air circulating over the products can be controlled by varying the quantity of air delivered in cubic feet per minute and the air space between each row of products. A producer considering the construction of a blast freezer should consult a refrigeration engineer, to obtain a minimum product-freezing time economically.

Immersion freezer.--In brine freezing fish a very high heat transfer takes place between the fish and the brine. This is because the refrigerated brine, being a good conductor, is in contact with the exposed surface area of the entire fish. In order to obtain a high heat transfer rate between the fish and the brine, it is necessary that sufficient circulation of the brine be maintained so as to allow it to flow freely around the fish. This type of freezing is one of the quickest and most efficient methods of freezing round fish.

FREEZING METHODS

A concern that is establishing a plant for the freezing of fishery products should, in the selection of a freezer suitable for its requirements, consider the type of fishery product to be frozen, the amount of product to be frozen within a certain specified time, the cost of freezing, the handling costs, the equipment maintenance costs, and the quality and appearance of the frozen product.

These considerations have been reflected, to a large extent, in the design of the sharp, multiplate, blast, and immersion freezers that are used for the freezing of fishery products today. These freezers have been improved throughout the years, resulting in the development of conveyor-type sharp freezers, continuous multiplate freezers, and tunnel-type blast freezers.

The following will discuss the sharp, multiplate, blast, and immersion freezers, with particular emphasis on handling methods, product-freezing times, and relative merits.

Sharp Freezers

The sharp freezer was one of the first devices used for the freezing of fishery products. The freezer consists of a series of pipe-coil shelves, which are maintained at temperatures of -20° to -40° F. by refrigerant flowing through the pipes. The fish to be frozen are placed on these coils, in the round or in packages.

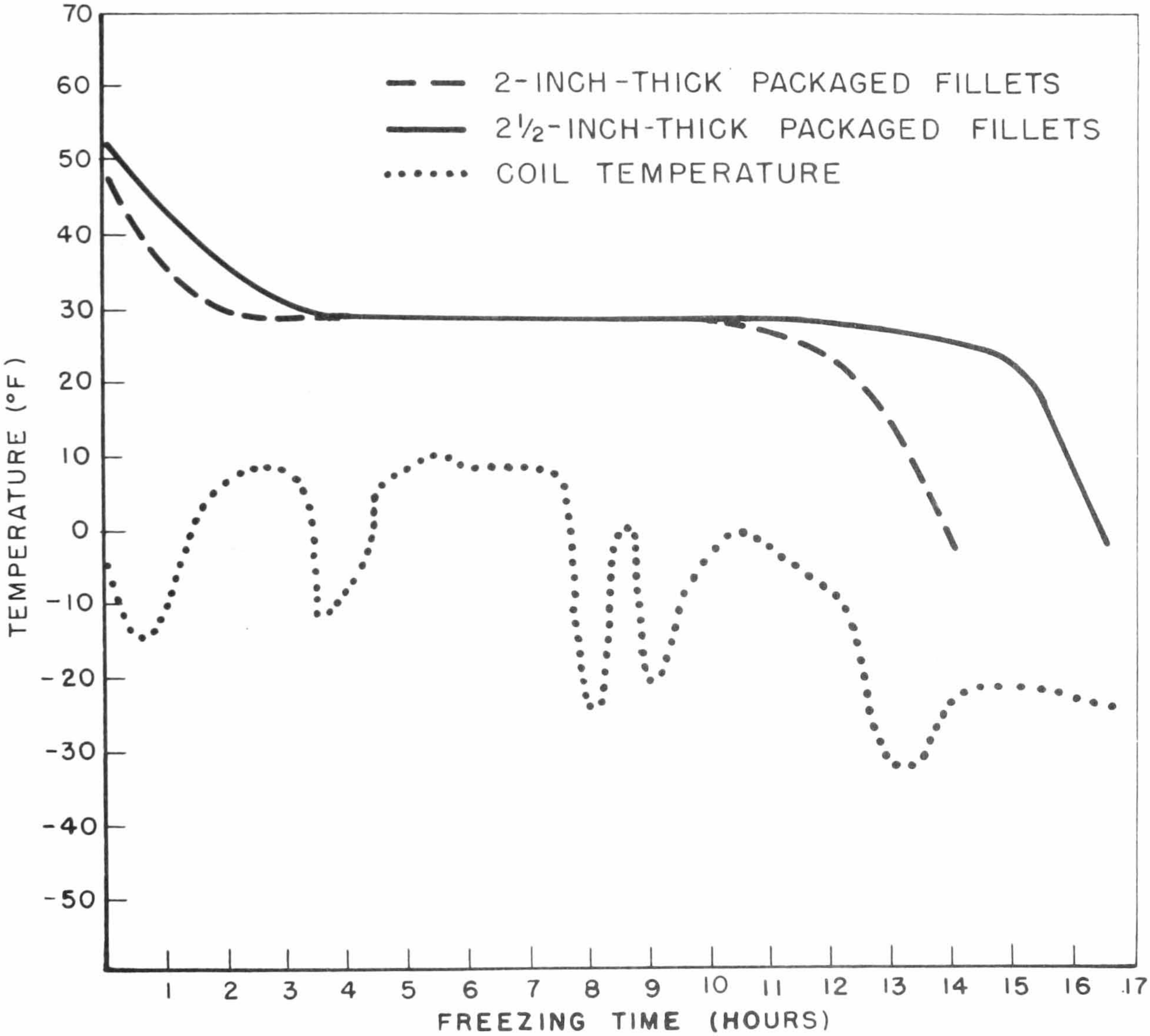

Figure 1.—Freezing packaged fish fillets in a sharp freezer.

Sharp freezers are referred to as either the conventional type or as the continuous-conveyor type.

In the conventional type, the products to be frozen are loaded onto skids. These skids are then moved into the freezer by a hand truck, and the products are unloaded from the skids and placed on the freezer shelves.

In the continuous-conveyor type, the products are conveyed from the processing room, by means of a continuous conveyor, into the freezer. The products are then removed from the conveyor belt by hand and placed on the freezer shelves.

The following is a description of typical commercial installations of these types of sharp freezers.

Conventional-Type Sharp Freezer

A large sharp freezer with a capacity of 40,000 pounds of $2\frac{1}{2}$-inch-thick 10-pound packaged fish fillets consists of a room approximately 60 feet long, 24 feet wide, and 7 feet high located between two cold-storage rooms. The freezing is accomplished by means of two banks of coil shelves, 50 feet in length, located along each wall. There are 9 shelves in each bank of coils, with approximately 9 inches between each shelf. The coils consist of $1\frac{1}{4}$-inch-diameter steel pipes, 18 coils to a shelf, with 5 inches between each coil. Ammonia is expanded through the coils to furnish the low coil-temperature necessary for freezing. In some sharp freezers, cold brine is circulated in the coils. This practice, however, is more costly, owing to (1) the lower heat-transfer rate and (2) the need for extra machinery and maintenance. Two or more low-velocity fans are used to provide adequate air circulation within the freezer. The coils are defrosted every 6 months, or when needed, by means of a hot-gas defrost.

Figure 2.—Freezing halibut in a conventional-type sharp freezer.

The sharp freezer is suitable for freezing round fish in pans, or fish fillets packaged in 5- and 10-pound boxes. Packaged 5- and 10-pound boxes of fish fillets are received at the loading platform on skids from the processor. The skids are then moved into the freezer by a hand truck, where the products are loaded on the pipe coils. Two men are used to unload each skid, and approximately two or three skids can be unloaded at once. After the freezer is loaded, the fans are started up, to provide the necessary air circulation. Four men can load or unload 40,000 pounds of packaged fish fillets in about 3 hours.

The time required to freeze packaged fish fillets in a sharp freezer is shown in figure 1.

The advantages of the sharp freezer are:

(1) The cost of freezing is lower than with the blast or plate freezer.

(2) The sharp freezer will give a fairly high output of frozen fish.

(3) It has a low maintenance cost.

The disadvantages of the sharp freezer are:

(1) It freezes products much slower than do blast, plate, or immersion freezers.

(2) It requires considerable handling of products.

(3) It requires that the coils be defrosted at least once every 6 months.

(4) It requires that the 1-pound packages of fish fillets have weight on them to prevent them from bulging, owing to the expansion resulting from their being frozen.

(5) The loading and unloading of the products results in coil frosting and increased freezing time.

<u>Continuous-Conveyor-Type Sharp Freezer</u>

This type of freezer is similar to the sharp freezer previously described, except for a continuous conveyor belt that is built into the freezer.

A freezer 110 feet long, 29 feet wide, and 10 feet high has capacity for 120,000 pounds of 10-pound boxes of packaged fish fillets. The freezing is accomplished by $1\frac{1}{4}$-inch-diameter steel coils, which are flooded with ammonia. The coils are arranged in 4 banks: one bank is placed against each wall, and 2 banks are located in the center of the room. Each coil bank has 9 shelves (8 coils per shelf), with 7 inches between each shelf. Ammonia refrigeration is used to maintain the coils at temperatures of -10° to -20° F.

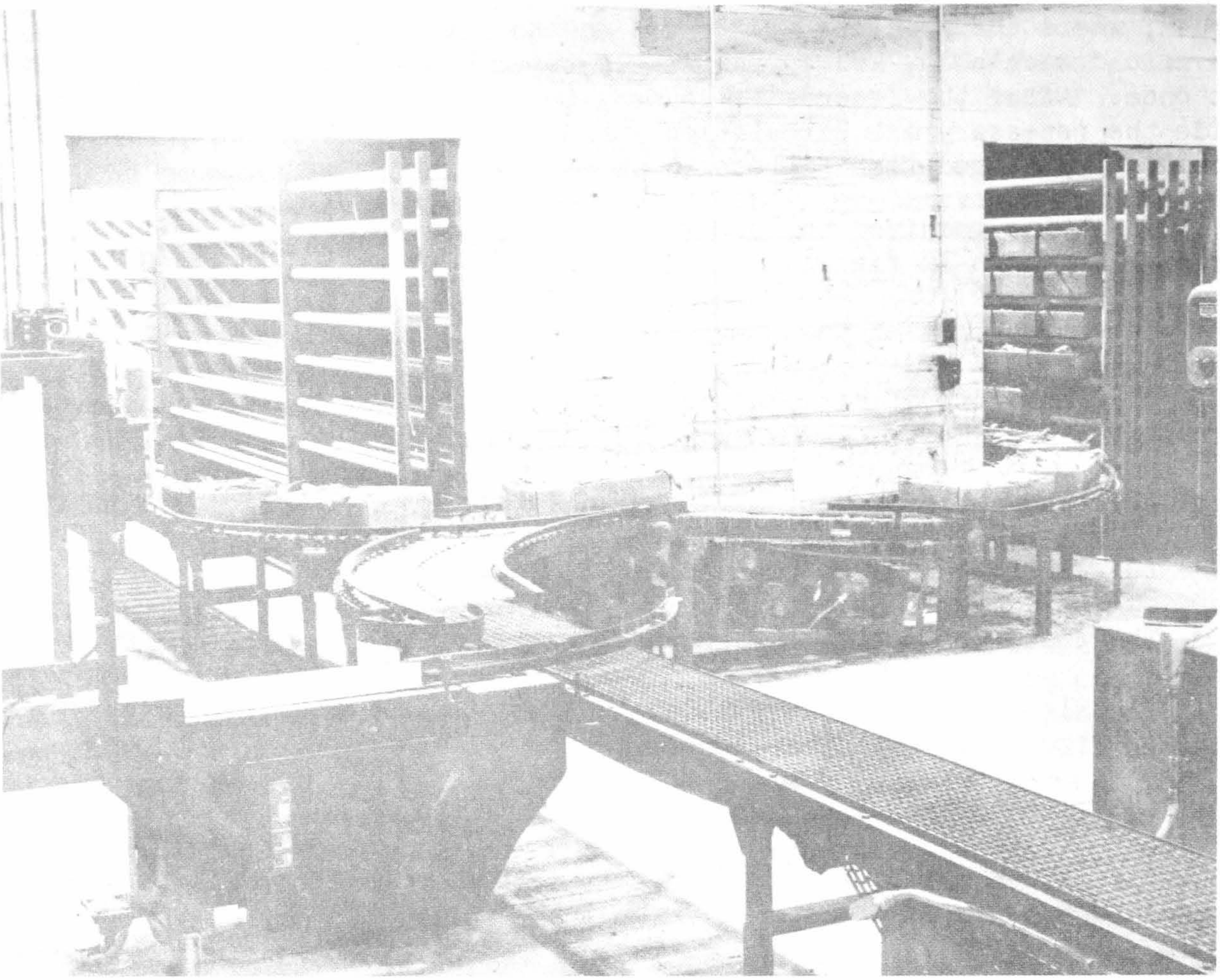

Figure 3.—A sharp freezer with a continuous conveyor. (Photo courtesy of Conveyor Specialty Company, Inc., and Cape Cod Cranberry Company)

A galvanized metal-mesh conveyor belt (12 inches wide and 335 feet long) runs from the processing table through an opening in the freezer wall into the freezer between the bank of the wall and center coils on one side of the freezer, around a 180-degree turn at the far end of the freezer, and back between the banks of wall coils and the center coils on the other side to the front of the freezer (figure 3). Another conveyor belt is connected to the previous one at the 180-degree turn. This conveyor runs from the back end of the freezer into an adjacent cold-storage room and out the end wall of the cold-storage room to a truck-loading platform.

The 10-pound commercial boxes of fish are placed on the conveyor belt in the processing room. The conveyor carries the fish into the freezer, where one man loads the boxes on the coil shelves. After the products are frozen, the fish are loaded onto the conveyor, which carries them to the cold-storage room. The products for shipment are placed on a conveyor that carries them from the cold-storage room to the loading platform, where a portable conveyor is used to transport them into a refrigerated truck.

The advantages of this freezer over the conventional-type sharp freezer are (1) handling costs are reduced, (2) less defrosting is required, and (3) freezing time of products is but little affected by loading and unloading of the freezer.

Multiplate Freezers

The multiplate freezer lowers the product temperature by direct contact of the product with movable refrigerated aluminum plates. The first multiplate freezers were of the batch type and had to be loaded and unloaded by hand. However, recently, continuous multiplate freezers have been developed that can be loaded and unloaded automatically.

Batch-Type Multiplate Freezer

In the batch-type multiplate freezer, the freezing is accomplished by expanding a refrigerant through horizontal movable aluminum plates, which are stacked vertically within an insulated cabinet (figure 4). The plates are raised and lowered by a hydraulic-pressure system, thereby ensuring a positive contact between the plates and the packaged products during freezing. This type of freezer is available in capacities of from 360 to 2,400 pounds of 2-inch-thick 5-pound packaged fish fillets. In the small models, the compressor and accessory equipment are located under the freezing cabinet, whereas in the larger models, the refrigeration machinery is separate from the freezing cabinet. The refrigeration machinery is similar to

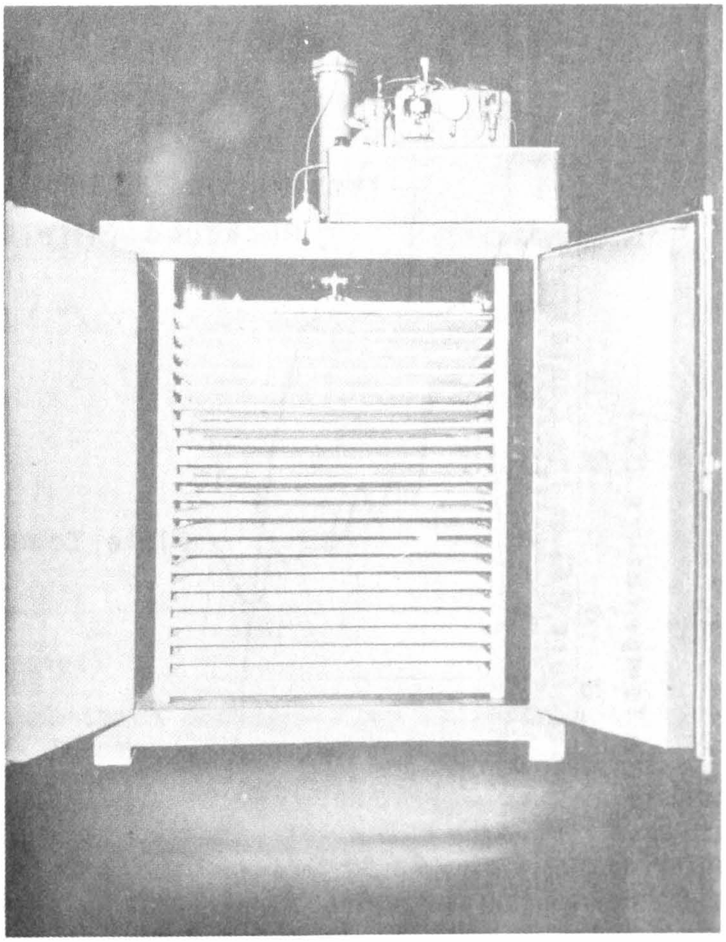

Figure 4.—A multiplate compression freezer. (Photo courtesy of American Plate Freezer Corporation)

that of any compression system, consisting of a motor-driven compressor, condenser, receiver, expansion valve, and heat exchanger. The refrigerant inlet to the movable aluminum plates is connected to the expansion-valve outlet by means of flexible hoses. The outlet of the plates is connected to the compressor suction manifold also by flexible hoses.

Before the packaged products are placed in the freezer, the plates are cooled to about -15° F. These products are then laid on trays, which are placed on the movable aluminum freezer plates. Wooden spacers of the same length as the plates and of the same height as the packages are placed between each set of plates to prevent crushing of the packages. All the packages on the same plate must be of the same thickness, to ensure proper contact for freezing.

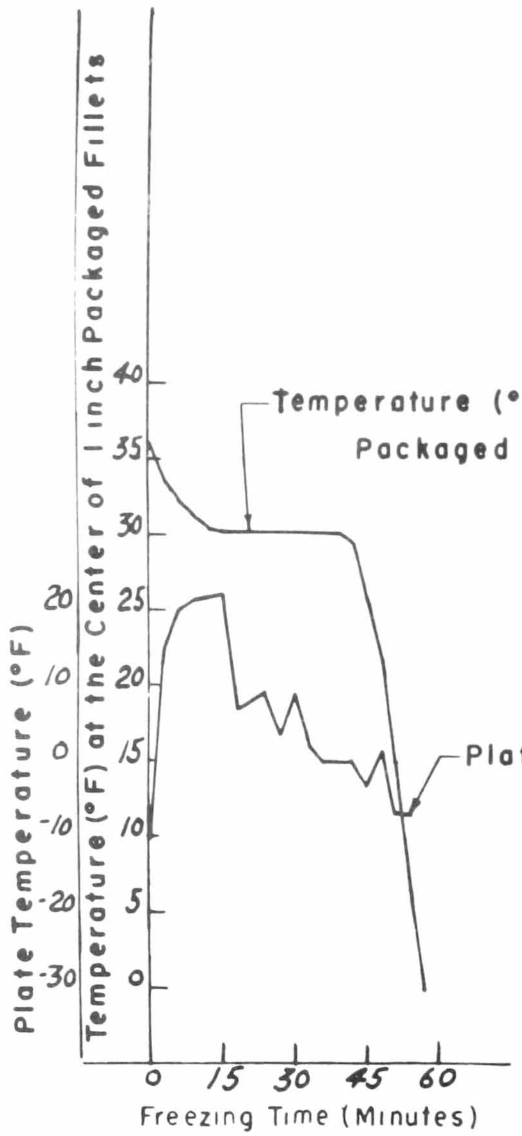

Figure 5.—Freezing 1-inch-thick packaged fish fillets in a multiplate compression freezer.

When the freezer is loaded, the cabinet doors are closed, and a lever is activated that supplies hydraulic pressure to the plates, causing them to move down on the spacers, thereby ensuring good contact between the packages and the plates. When the packaged products are frozen, the cabinet door is opened, the plates are raised, and the trays containing the packages are removed from the freezer.

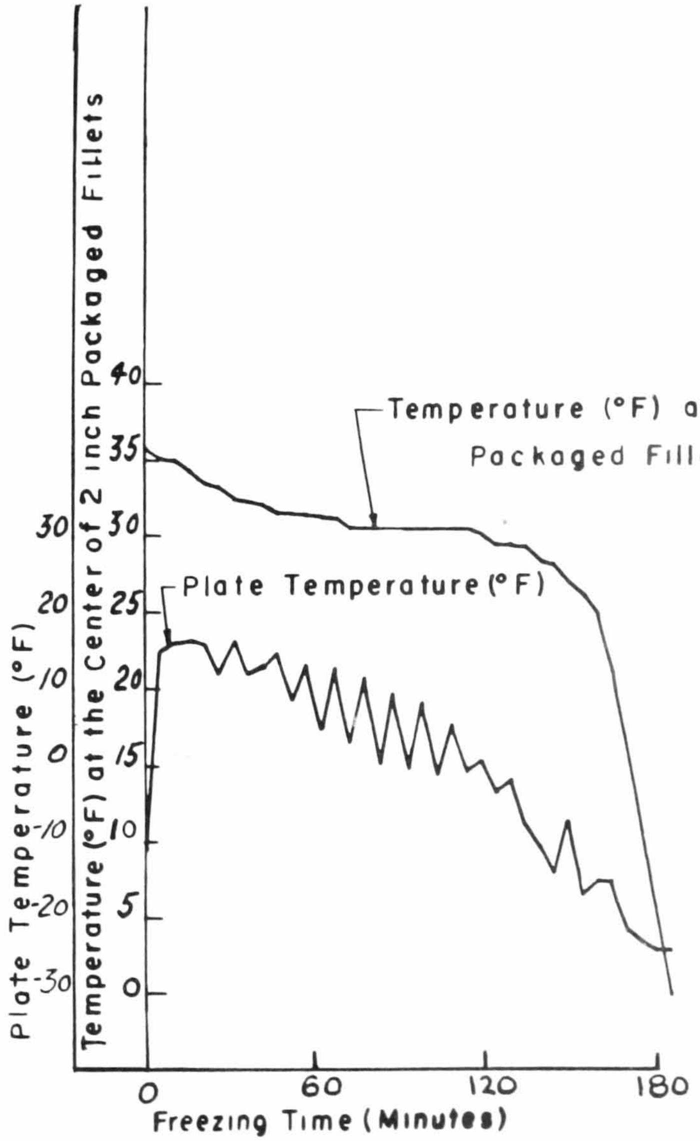

Figure 6.—Freezing 2-inch-thick packaged fish fillets in a multiplate compression freezer.

The multiplate freezer is very suitable for quick freezing fish fillets, packaged in 1-pound (1-inch-thick), 5-pound (2-inch-thick), and 10-pound ($2\frac{1}{2}$-inch-thick) commercial boxes. This freezer will also freeze packaged precooked fishery products very satisfactorily, producing a uniformly shaped package. Tests conducted at the Fish and Wildlife Service laboratory, East Boston, Massachusetts, (Slavin 1955) show that the freezing time and energy required for freezing packaged precooked fish sticks is greater than that required for freezing packaged fish fillets, because of the slower rate of heat transfer due to the air space within the package. The freezing times required for fish fillets packaged in 1-pound (1-inch-thick), 2-pound (2-inch-thick), and 10-pound ($2\frac{1}{2}$-inch-thick) commercial boxes, and breaded precooked fish sticks packaged in 10-ounce ($1\frac{1}{2}$-inch thick) commercial boxes are shown in figures 5, 6, 7, and 8, respectively.

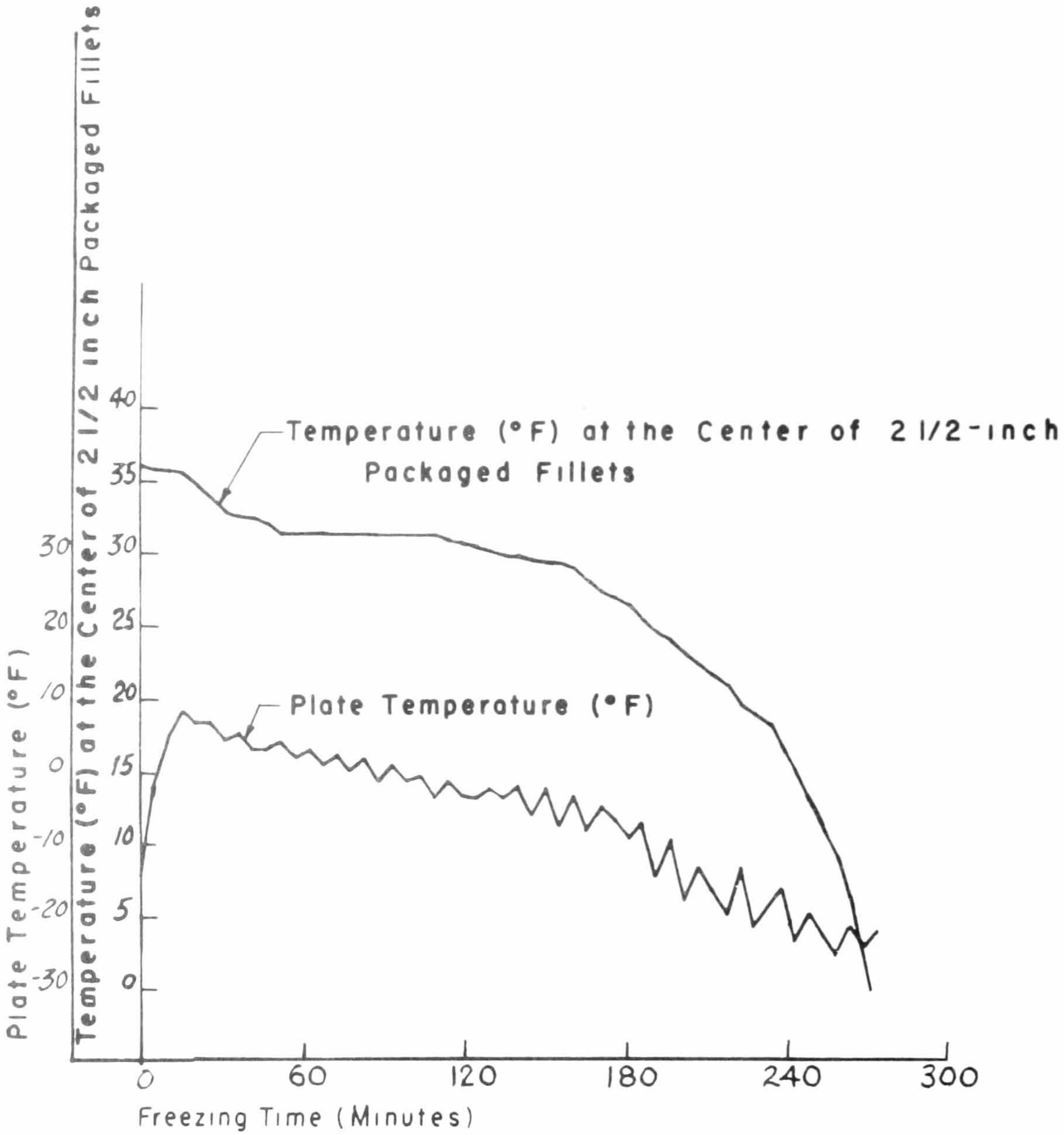

Figure 7.--Freezing $2\frac{1}{2}$-inch-thick packaged fish fillets in a multiplate compression freezer.

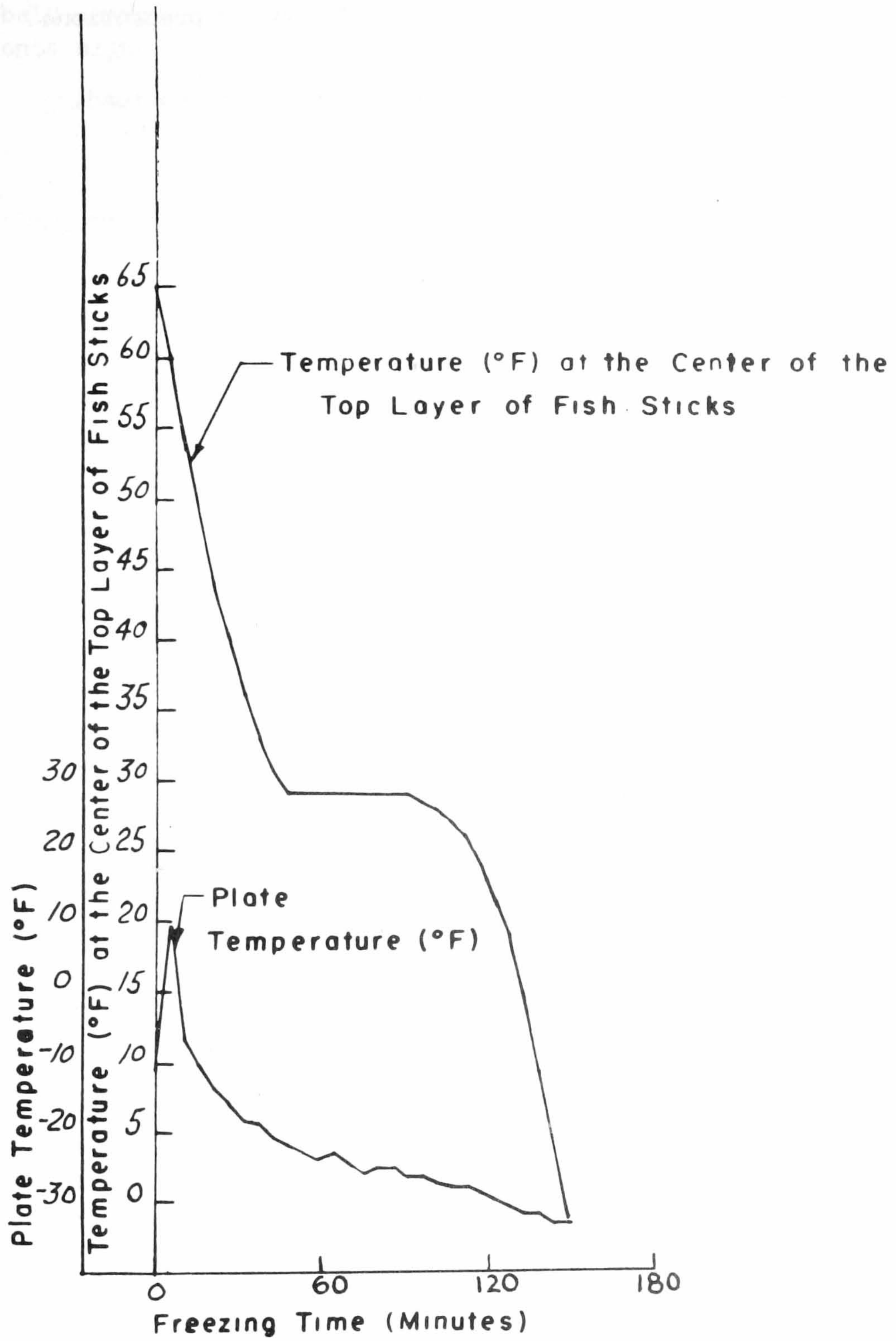

Figure 8.—Freezing 1½-inch-thick (10-ounce) packages of fish sticks in a multiplate compression freezer.

To obtain fast, efficient freezing, the following precautions should be taken:

1. Maintain the plates at a low temperature prior to loading the freezer.

2. Keep the freezer doors closed, except when loading or unloading, thereby preventing excessive frost from building up on the plates.

3. Use proper-sized spacers.

4. Package fish fillets properly, leaving as few voids as possible.

The advantages of the multiplate freezer are:

1. It produces a uniform well-shaped package with a minimum of voids.

2. It requires a minimum amount of floor space.

3. It freezes packaged fish fillets quickly and economically.

4. It does not require defrosting of the plates, if the freezer is operated properly.

The disadvantages of the multiplate freezer are:

1. It requires much handling in loading and unloading the products.

2. It freezes very slowly those products with a dead-air space in the package.

3. It requires large storage space for pans and spacers.

Continuous-Type Multiplate Freezer

The continuous-type multiplate freezer is similar to the batch-type multiplate freezer in that the same method of freezing is employed but is different in that the products are loaded and unloaded automatically. The freezer is available in 1 or in 8 stations, with respective capacities of 1,750 or 14,000 10-ounce packages of fish sticks. The larger-sized freezer consists of 8 stations and can be enclosed in a refrigerated room approximately 34 feet long, 20 feet wide, and 12 feet high (figure 9). This freezer will handle packages ranging in thickness from 7/8 inch to $2\frac{1}{2}$ inches. The adjustment to new package size can be made by means of four exposed screws.

Packaged fish products are fed directly from the wrapping machine, by a belt conveyor, through an opening in the walls of the refrigerated room onto eight roller-type loading stations at the front of the freezer.

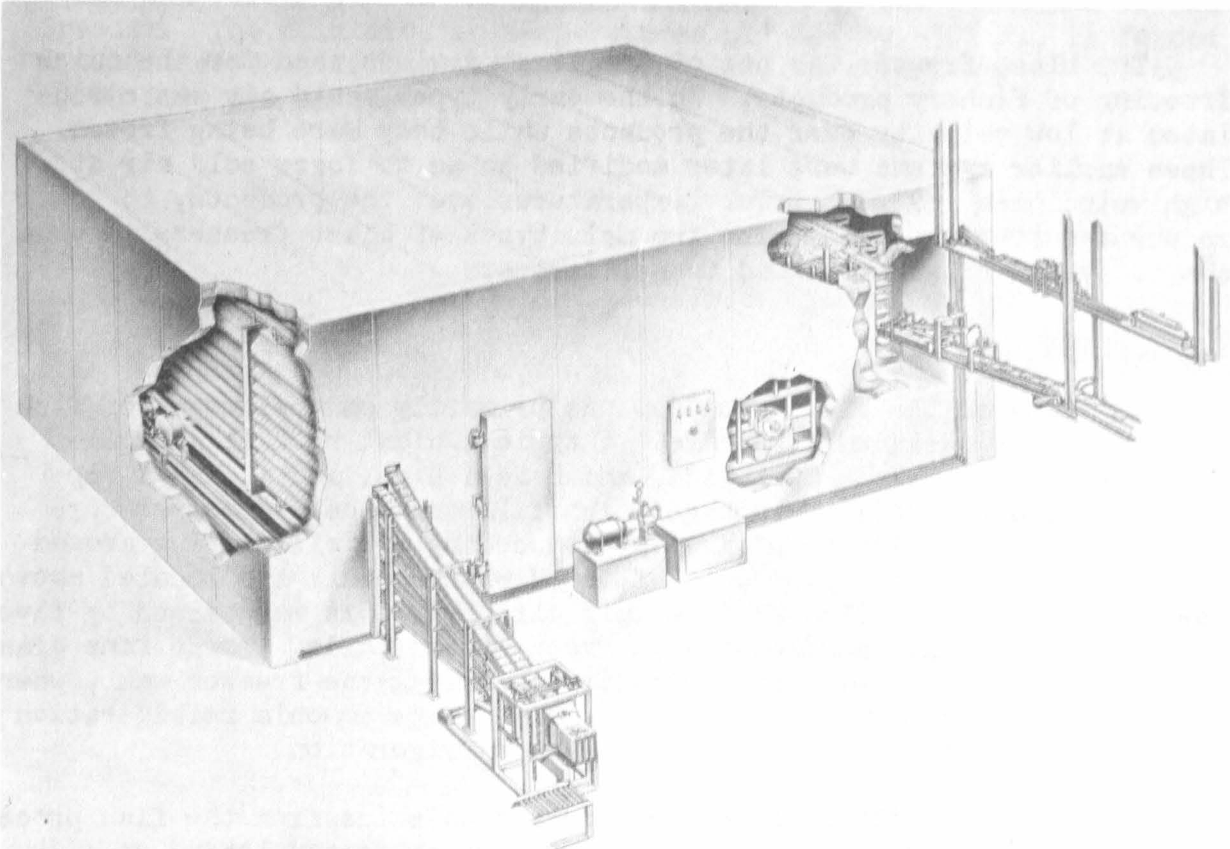

Figure 9.—A continuous multiplate compression freezer. (Photo courtesy of Patterson Freezer Corporation)

Electronic devices control the loading of each station so that, when one station is loaded, the conveyor moves up to the next station, until all eight loading stations are filled. When loading, each station receives one full row of packages at a time. The stations consist of upper and lower aluminum freezing plates. During each loading cycle, a movable frame causes the lower freezing plate of each station to move forward and down, thereby opening the station. The forward and back action of the lower plate forces the rows of packages to move toward the rear of the freezer. The loading cycle continues until all eight stations are completely filled. When the freezer is filled, each incoming row of packages forces out a frozen row of packages from the other end of the machine on each of the eight levels. The frozen packages are then conveyed to an automatic cartoning machine located outside the refrigerated room.

This freezer is ideally suited for the freezing of packaged, precooked fishery products and fish fillets packaged in 1-pound commercial boxes. The main advantage of this freezer over the conventional multiplate freezer is a large-capacity frozen-product output with a minimum amount of handling.

Blast Freezers

The blast freezer was one of the first devices used for the quick freezing of fishery products. In the early types, cold air was circulated at low velocity over the products while they were being frozen. These earlier systems were later modified so as to force cold air at high velocities and much lower temperatures over the products, to reduce the freezing time. The two main types of blast freezers in wide use today are blast rooms and tunnel freezers.

Blast Rooms

The size of the blast room depends primarily on the amount of fish the freezer is designed to handle. A typical blast room is approximately 60 feet long, 12 feet wide, and 8 feet high; and it has a capacity of 40,000 pounds of packaged fish fillets. The low temperature necessary for freezing (-40° F.) is produced by air circulating around 1½-inch-diameter steel pipe coils flooded with ammonia and located above the freezer ceiling. The required air circulation is maintained by five 5-hp. squirrel-cage fans located in front of the coils. These fans draw the air through the coils and force it down along the freezer wall, where baffles direct it across the products. Two-stage ammonia refrigeration equipment is used to supply the necessary refrigeration.

The packaged fish fillets are received on skids from the fish processors. The products are then removed from the skids and loaded on buggies. The buggies consist of a series of galvanized steel shelves 4 inches apart, supported by an angle-iron frame on each side. The base of the buggies is also of angle iron and has 4 small wheels to facilitate movement. Each buggy will hold 1,000 pounds of fish fillets packaged in 10-pound commercial boxes. The buggies,after being loaded, are pushed into the freezer. When the freezer is full (10 buggies deep and 4 across), the freezer door is closed, and the fans are turned on. The time required to freeze 2½-inch-thick 10-pound boxes of fish fillets is approximately 16 hours. The blast room freezes products much slower than does the tunnel-type freezer, because of the considerable decrease in air velocity due to the large number of buggies in the freezer. This freezer is ideally suited for freezing a large quantity of fish over a comparatively long period of time. The time required to load and unload the freezer is much less than that required for the sharp or plate freezer.

Tunnel-Type Freezer

A typical tunnel-type freezer (figure 10) consists of a room approximately 26 feet long, 5 feet high, and 3½ feet wide, with a capacity of 9,900 pounds of packaged fish fillets. Finned pipe coils flooded with ammonia provide the low temperature necessary for freezing. A 25-hp. squirrel-cage fan, located in front of the coils, draws air through the coils and forces it out through a duct placed at the front end of the freezer. The cold air, at temperatures of -40° to -50° F., is forced around the products located on the buggies, at a velocity of 500 to 1,000 feet per minute.

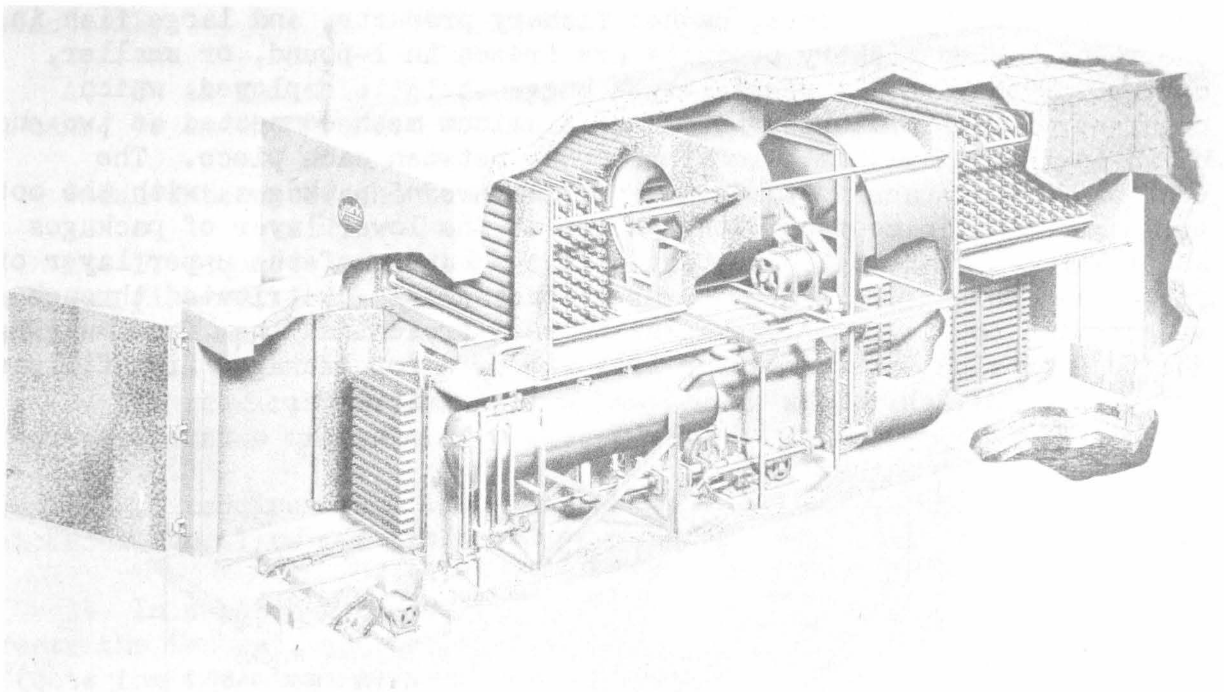

Figure 10.--A tunnel-type blast freezer. (Photo courtesy of York Corporation, York, Pa.)

An overlap freezer door is located at each end of the freezer, and a pair of vestibule-type doors are located approximately 6 feet in front of it. The space between the vestibule doors and the rear of the freezer is called the precooling chamber. An endless-chain conveyor, located at the rear of the freezer, engages a pawl on the buggies and carries them onto a pair of angle-iron tracks into the freezer. A button operates the conveyor and controls the rate of progress.

This type freezer is designed so that either a continuous process or a batch process can be employed. The packaged fish fillets are received from the processors and loaded on the shelves of buggies, which are approximately 3 feet wide, 3 feet long, and 5 feet high. Each buggy will hold approximately 1,000 pounds of 10-pound boxes of fish fillets. The buggies, when loaded, are pushed into the precooling chamber, where the continuous-chain conveyor carries them into the freezer. The freezer will hold 10 buggies: 9 in the freezer chamber, and 1 in the precooling chamber.

When the tunnel is used as a continuous freezer, the buggies are put into and removed from the freezer at predetermined time intervals, depending on production output and the freezing time of the product being handled. When the tunnel is used as a batch freezer, it is loaded to its full capacity of 9 buggies; after the freezing time has elapsed, these buggies are then removed to cold storage.

This type freezer is suitable for freezing packaged fish fillets, precooked fishery products, canned fishery products, and large fish in the round. When fishery products are frozen in 1-pound, or smaller, commercial packages, a special-type buggy shelf is employed, which consists of two reinforced pieces of aluminum mesh connected at two ends by an angle iron with a 1-inch air space between each piece. The shelves form a spacer between adjacent layers of packages, with the bottom part being in contact with the top of the lower layer of packages and the top part being in contact with the bottom of the upper layer of packages. Weight on the top layer of packages is distributed through the shelves to the lower packages, thereby preventing them from bulging due to expansion. The freezing time required for packaged fish fillets and precooked fish sticks are shown in figure 11.

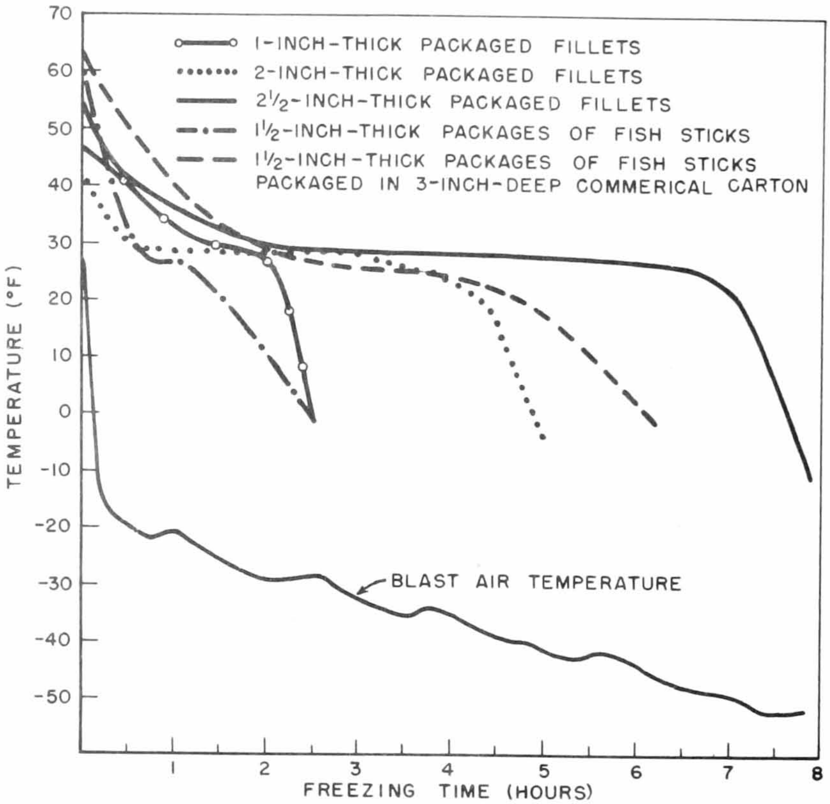

Figure 11.--Freezing packaged fish fillets and fish sticks in a tunnel-type blast freezer (air velocity 500-1000 feet per minute).

Advantages of the tunnel-type blast freezer:

1. It requires only a minimum amount of product handling.

2. It has a high frozen-product-capacity output per square foot of space.

3. It can freeze large bulky packages of fishery products satisfactorily.

4. It is suitable for freezing fishery products in packages and in cans, or in the round.

Disadvantages of the tunnel-type blast freezer:

1. It requires special buggy shelves to be used when freezing packaged fish fillets and fish sticks in the 1-pound and smaller-size packages.

2. It requires more electrical energy than do the plate or sharp freezers, because of the energy needed to operate the fan.

3. It requires that the packaged products be all of the same thickness, to allow for the proper flow of air.

4. In a batch process, the products at the front end of the tunnel freeze the fastest, and the variation of air flow in different buggies affects the freezing rate.

5. It must be defrosted periodically, resulting in lost production.

6. It requires more time for freezing packaged fish fillets than does the plate freezer.

7. It has higher maintenance costs than have plate or sharp freezers.

8. It cannot freeze small loads economically.

Immersion Freezers

The immersion-type freezer is used largely for the freezing aboard a fishing vessel of fish—such as haddock, cod, and tuna—or shellfish—such as shrimp. The product is frozen by immersing it in an agitated cold brine solution of a fixed concentration and temperature.

The type of immersion freezer employed depends on the nature of the product being frozen. Although sharp, blast, or plate freezers are somewhat versatile inasmuch as they will freeze a variety of different fish species in various forms, the application of the immersion freezer is restricted largely to the specific type of fish for which it was designed.

The following factors should be considered in the selection of a commercial immersion freezer aboard a fishing vessel: (1) type of fish to be frozen, (2) freezing time for average weight fish, (3) handling requirements, (4) source of available power, (5) refrigeration-equipment space requirements, (6) average catch, (7) space required to hold a portion of the fish prior to freezing, (8) cost and dependability of refrigeration equipment, (9) effect of brine on the product, (10) freezing temperature of the brine, and (11) cost of maintaining a clean brine supply. The following describes various types of immersion freezers employed on vessels.

An Immersion Freezer Designed to Freeze New England Groundfish Aboard a Fishing Vessel

An immersion-freezing system designed to freeze New England groundfish has proven successful in use on the experimental trawler Delaware, which is operated by the Fish and Wildlife Service laboratory, East Boston, Massachusetts. This system will freeze approximately 850 pounds of fish per hour. The system is of the indirect type, with sodium chloride brine (23 percent salt by weight) being used as the cooling medium. The freezing is accomplished within a rectangular-shaped brine tank (figure 12) located in fish hold number 2. This tank is 8 feet long,

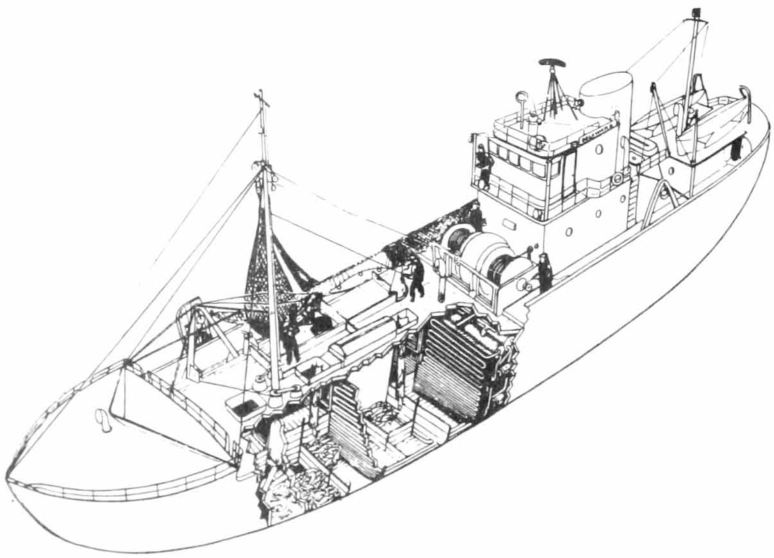

Figure 12.--Cutaway view of the Delaware showing brine freezing tanks and fish storage bins.

14 feet deep, and 5 feet wide, with a capacity of 35,000 pounds of brine. The tank extends from the floor of the fish hold to about 30 inches above the deck. The tank cover has two doors, one on each side, so that the fish can be loaded and unloaded simultaneously. There are 11, cylindrical, galvanized, metal-mesh baskets connected at each end by a continuous chain drive, located within the freezer tank. An electric motor placed in a watertight enclosure on deck adjacent to the tank is used to provide the power for the clockwise and counterclockwise rotation of the baskets through the brine.

The brine in the freezer tank is circulated by a centrifugal-type pump through a flooded-type brine cooler, where it is cooled to the proper temperature. A 25-ton absorption refrigeration machine located in a watertight compartment forward of the engine room is used to provide the necessary refrigeration. An ethanol solution, at a temperature of -10° F., circulating through steel coils located on the walls and ceilings of each of two holds, maintains these holds at 0° F. An additional small fish hold is used for icing fish for experimental purposes.

The freezing operation is as follows: Fish, after being caught, are put, in the round, into the cylindrical baskets located in the brine tank. Each basket will hold 500 pounds of fish. After the baskets are loaded from the deck, the freezer-tank doors are closed, and the basket-drive motor is started, causing the baskets to rotate through the brine. The movement of the baskets through the brine provides adequate brine circulation around each fish, thereby insuring uniform, quick, and efficient freezing. Each basket is numbered. Fish of approximately the same weight are put together in the same basket. After the proper freezing time elapses (table 1), the fish are removed from their respective baskets, glazed and conveyed by aluminum chutes to the cold-storage hold. The fish are then stored in cold storage until arrival of the vessel at the laboratory, at which time the frozen fish are discharged, thawed, filleted, and refrozen.

The advantages of this type of freezer are:

1. It freezes products quickly and efficiently.

2. It is versatile, inasmuch as it can also freeze tuna or shrimp, quickly and efficiently.

3. It requires a minimum amount of handling.

4. It uses a sodium chloride brine, which is relatively inexpensive.

5. It produces a high-quality frozen fish.

6. It has low maintenance costs.

Table 1.--Freezing time for whole round cod and haddock, of various thicknesses, in sodium-chloride brine at 10° and 0° F.

Thickness 1/ of fish	Approximate 2/ round weight of fish	Freezing time at:	
		10° F.	0° F.
Inches	Pounds	Minutes	Minutes
1½	1 - 1½	55	35
2	1½ - 2½	85	55
2½	3 - 5	125	80
3	4½ - 7½	170	110
3½	7 - 10	220	145
4	9 - 12	280	185

1/ Side to side thickness (smallest diameter of a cross section) at the point of maximum girth.

2/ Round weight is generally 10 to 15 percent higher than is dressed weight.

The disadvantages of this freezer are:

1. Careful temperature regulation is required in the brine cooler to eliminate the possibility of the brine freezing out at -6° F., which might result in bursting tubes within the brine cooler.

2. The penetration of salt into the fish will be excessive if they are left in the brine considerably longer than the required freezing time.

An Immersion Freezer Designed for the Freezing of Shrimp Aboard a Fishing Vessel

The limited storage life of iced shrimp has recently led to the design and installation of commercial immersion freezing plants aboard several shrimp fishing vessels. A sodium chloride brine solution--its use having been proven in other types of immersion freezers--was used first (Dassow 1954).

The shrimp frozen in this solution were of good quality; however, they sometimes fused together into a solid mass upon freezing, and subsequently dehydrated in cold storage unless protected by spray glazing.

Mingledorff (1954) found that, by using a brine solution composed of salt and sugar, the shrimp would not fuse together upon freezing and that they could be stored for a period of 60 days or more without dehydration. This solution is presently being employed in the immersion freezers aboard several shrimp fishing vessels.

The freezing is done in a stainless-steel tank approximately 5 feet wide, 7 feet long, and 4 feet high, which is located on the deck of the vessel (figure 13). The brine temperature is maintained at 0° F. by Freon 12 circulating through plates placed within the tank. A hydraulically driven propeller inside the tank provides the necessary brine agitation.

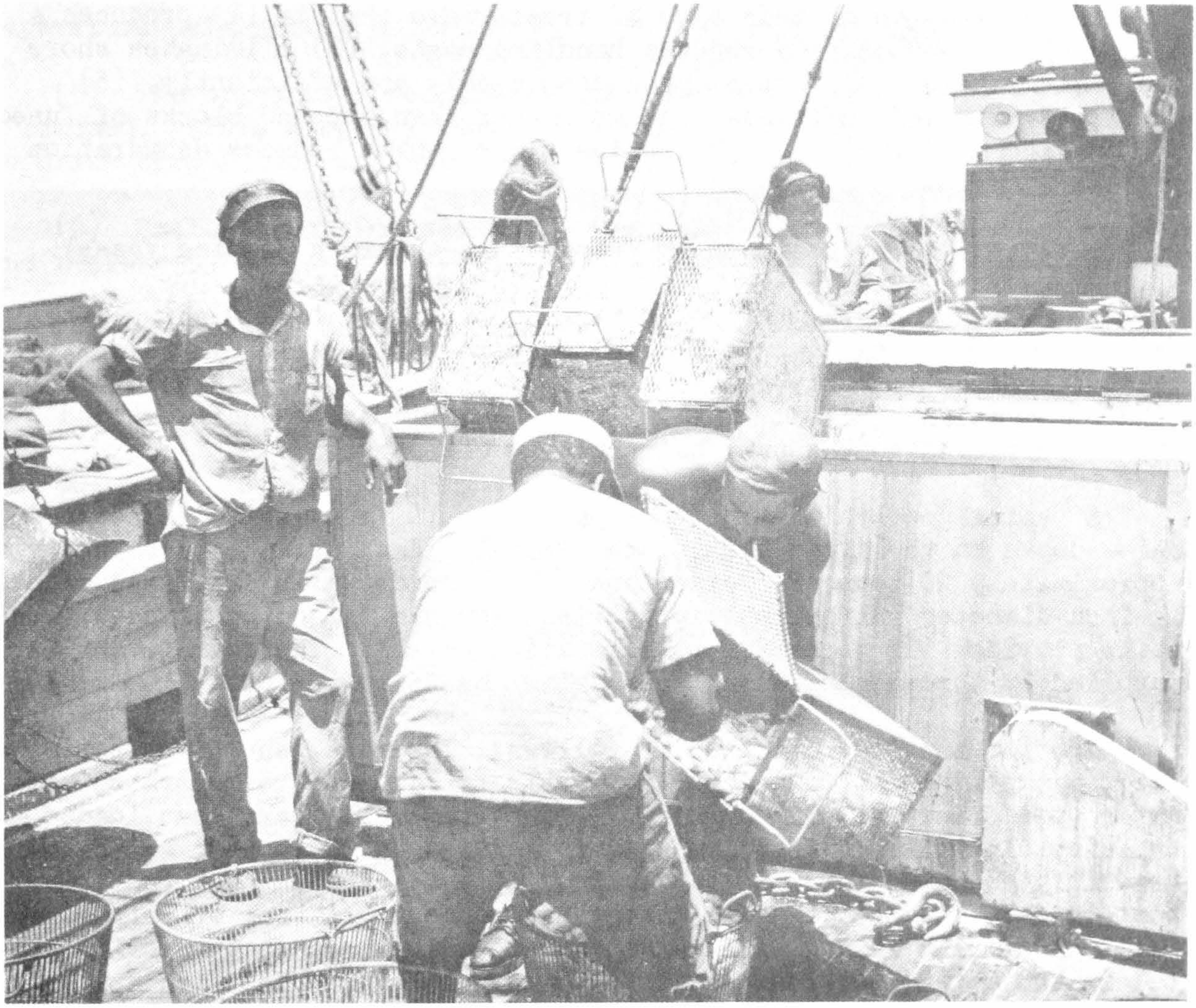

Figure 13.--Fresh shrimp entering the freezing tank aboard the shrimp trawler Prince Charming. (Photo courtesy of Mingledorffs, Inc.)

The refrigeration is supplied by a diesel-driven Freon 12 compressor. The frozen shrimp are maintained at 0° F. by an additional diesel-driven compressor supplying refrigeration to overhead plates located within the holds.

The freezing operation is as follows: Shrimp, after being caught, are headed and washed. They are weighed in 50-pound lots and put into stainless-steel wire baskets. These baskets, in turn, are set in the freezing tank (figure 13). When 15 minutes elapses, the baskets are picked up, and the shrimp are dumped into containers, which are then placed in a cold-storage room. At the end of each day, the shrimp are put into 50-pound master cartons. By this method, it is possible to process and freeze more than 3,000 pounds of shrimp in 10 hours.

The advantages of this type of freezer are that it (1) produces a high-quality product, (2) reduces handling costs, (3) eliminates shore processing costs, (4) freezes products quickly and efficiently, (5) permits thawing of individual shrimp rather than 5-pound blocks of fused shrimp, and (6) provides a protective glaze, which reduces dehydration in cold storage.

An Immersion Freezer Designed to Freeze Tuna Aboard a Fishing Vessel

To obtain a constant flow of high-quality tuna from fishing vessel to cannery, the tuna must be frozen on board the vessel. The freezing method employed is quite complicated because of the large size of the fish often being handled. The following describes the equipment and methods employed on the tuna bait boat or clipper.

A typical modern tuna clipper has 10 to 14 freezing wells located below decks on the after part of the vessel. Each well has a capacity of approximately 20 tons of frozen tuna. The inside walls are lined with 1¼-inch-diameter galvanized pipe coils. Ammonia expanding through these coils provides the necessary refrigeration effect. The refrigerant is supplied by three compressors each driven by 25-hp. motors.

The freezing operation is as follows: The tuna, shortly after being caught, are put into freezing wells filled with refrigerated sea water (preferably at 30° to 32° F.). When the wells have been loaded to capacity with fish and the temperature lowered to 30° to 32° F., salt is gradually added to the water and mixed by means of a brine-circulating pump to form a dense brine solution, using about 3/4 sack of salt for each ton of fish in the well. Ammonia expanding through the coils located in the wells cools the brine to 20° F. or lower. The fish are held in the well until their internal temperature is lowered to at least 23° F. (preferably 10° to 15° F.), at which time the brine is pumped overboard. The well is then used as a still-air cold-storage room, with the refrigeration effect being furnished by the same coils that were used to chill the brine. This well is known as a dry well. The temperature of the ammonia within the coils is maintained at -10° to 0° F., thereby eventually producing a dry-well temperature of 10° to 20° F.

Before the arrival of the vessel at the cannery, the fish are started to thaw in order that they will be ready for unloading as soon as the ship docks. The procedure used is as follows:

(1) The refrigeration in the dry well is shut off.

(2) Approximately 8 hours later, the dry well is flooded with sea water.

(3) The brine circulator is started up.

(4) The same amount of salt is added as was added to lower the well temperature originally.

(5) The brine is circulated until a brine temperature of 28° F. is reached. (This step may take approximately 48 hours, depending on the brine circulation and hold capacity.)

(6) Small amounts of sea water are added to the wells until the brine reaches 33° F.

(7) The 33° F. brine is circulated until the fish reach an internal temperature of 25° F.

(8) The fish are unloaded into the cannery.

The advantages of this type of freezer are:

1. It requires a minimum amount of product handling.

2. It has low maintenance costs.

3. At rated capacity, it produces a good-quality frozen fish, in large volume.

4. It requires a minimum amount of space because fish are frozen, stored, and thawed in the same tank.

The disadvantages of this type of freezer are:

1. It freezes products very slowly.

2. It requires careful control of temperature because, if proper temperatures are not maintained within very close limits, spoilage of the product may result. This control requires careful loading so as not to overload the well and thus exceed its freezing capacity.

3. It is not versatile, inasmuch as this freezer is not suitable

for freezing groundfish, mackerel, or shellfish.

4. It requires large amounts of salt for both freezing and thawing.

5. It requires a high-capacity refrigeration system and considerable auxiliary power for large-volume brine pumps.

6. With small tuna, like skipjack, a slow rate of freezing in brine often leads to excessive absorption of salt by the flesh.

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