Naval Facilities Engineering Command
200 Stovall Street
Alexandria, Virginia 22332-2300

REFRIGERATION SYSTEMS

FOR COLD STORAGE

DESIGN MANUAL 3.04
REVALIDATED BY CHANGE 1 AUGUST 1986

ABSTRACT

Basic criteria are presented on refrigeration systems for cold storage facilities for use by architects and engineers. The contents cover design procedures, refrigeration systems, liquid circulating systems, insulation, automatic control, and ice-making plants.

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FOREWORD

This design manual is one of a series developed from an evaluation of facilities in the shore establishment, from surveys of the availability of new materials and construction methods, and from selection of the best design practices of the Naval Facilities Engineering Command, other Government agencies, and the private sector. This manual uses to the maximum extent feasible, national professional society, association, and institute standards in accordance with NAVFACENGCOM policy. Deviations from these criteria should not be made without prior approval of NAVFACENGCOM Headquarters (Code 04).

Design cannot remain static any more than can the naval functions it serves or the technologies it uses. Accordingly, recommendations for improvement are encouraged from within Navy and from the private sector and should be furnished to Commanding Officer, Southern Division, Code 406, Naval Facilities Engineering Command, P.O. Box 10068, Charleston, SC 29411-0068. As the design manuals are revised, they are being reconstructed. A chapter or a combination of chapters will be issued as a separate design manual for ready reference to specific criteria.

This publication is certified as an official publication of the Naval Facilities Engineering Command and has been reviewed and approved in accordance with SECNAVINST 5600.16. Review of Department of the Navy (DN) Publications; Procedures Governing.

J. P. Jones, JR.
Rear Admiral, CEC, U.S. Navy
Commander
Naval Facilities Engineering Command

MECHANICAL ENGINEERING DESIGN MANUALS

Manual Number	Chapter superseded in Basic DM	Title
DM- 3.01	1	Plumbing Systems
MIL-HDBK-1003/2	2,2A	Incinerators
DM-3.03	3,4,5	Heating, Ventilation, and Air Conditioning Systems and Dehumidifying Systems
DM-3.04	6	Refrigeration Systems for Cold Storage
DM-3.05	7,10	Compressed and Vacuum Air Systems
DM-3.06	8	Central Heating Plants
DM-3.07	9	Fossil Fuel Power Plants
DM-3.08	11	Exterior Distribution of Utility Steam, HTW, CHW, Gas and Compressed Air
DM-3.09	13,14	Elevators, Escalators, Dumbwaiters, Access Lifts, and Pneumatic Tube Systems
DM-3.10	15	Noise and Vibration Control of Mechanical Equipment (Army)
DM-3.11	16	Diesel-Electric Generating Plants
DM-3.12	--	Industrial Controls
MIL-HDBK-1003/13		Solar Heating of Building and Domestic Hot Water
DM-3.14	--	Power Plant Acoustics (Army)
DM-3.15	--	Air Pollution Control System for Boilers and Incinerators (Army)
DM-3.16	--	Thermal Storage
MIL-HDBK-1003/17	--	Industrial Ventilation
DM-3.18	--	Central Building Automation Systems (Army)

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REFRIGERATION SYSTEMS FOR COLD STORAGE

Section 1. INTRODUCTION

1. SCOPE. The basic criteria presented here are applicable to all types of cold storage facilities and automatic ice-making plants used for naval shore activities.

a. Types. The cold storage facilities designed for storage of medical supplies, photographic equipment, perishable food and other uses consists of three main types:

(1) A facility designed to satisfy the cold storage requirements of a subsistence facility (Subsistence Type).

(2) A facility designed to satisfy the requirements of a single installation or activity and any satellite activities (Station Type).

(3) A facility designed to satisfy the requirements of two or more independent installations and/or the requirements of naval vessels (Depot Type).

b. Criteria Sources. For any conditions not covered in this chapter, see ASHRAE criteria. This is a four-volume publication with titles as follows:

(1) ASHRAE Handbook + Product Directory, Fundamentals.

(2) ASHRAE Handbook + Product Directory, Applications.

(3) ASHRAE Handbook + Product Directory, Equipment.

(4) ASHRAE Handbook + Product Directory, Systems.

2. CANCELLATION. This manual on Refrigeration Systems For Cold Storage, NAVFAC DM-3.4, cancels and supersedes Chapter 6 of NAVFAC DM-3, Mechanical Engineering, of September 1972.

3. RELATED CRITERIA. Certain criteria related to the subject matter of this chapter appear elsewhere in the NAVFAC DM series. See the following sources:

Subject	Source
Cold Storage Area Requirements	NAVFAC DM-32.2
Water Treatment	NAVFAC DM-5 Series
Noise and Vibration Control of Mechanical Equipment	NAVFAC DM-3.10

4. POLICY. The minimum cost of owning and operating the system and any standby equipment used are factors for consideration before selection of equipment.

a. Application. Cold storage and ice plant criteria shall conform to the requirements of Department of Defense Construction Criteria Manual (DOD 4270.1-M), Occupational Safety and Health Act (OSHA), Environmental Protection Agency Regulations, National Codes, local and State codes, and Executive Orders as applicable.

b. Standby. The extent of standby equipment, if provided, shall be limited to a practical minimum by taking into consideration the following factors:

(1) Size. Standby equipment shall be sized for normal heat loss due to transmission, defrost load, minimum usage load and minimum or, if possible, no product load.

(2) Standardization. The type and size of equipment, such as compressors, cooling units, and refrigeration accessories, on each installation shall be standardized to minimize the extent of standby equipment and stocking of spare parts.

(3) System flexibility shall be sufficient to assure essential cooling in the event of partial breakdown or partial shutdown for maintenance and economical operation at light loads.

(4) Spare parts. As far as possible, essential spare parts shall be provided instead of complete standby equipment.

(a) Example. In lieu of a compressor, spare parts such as valve plate assembly, shaft seal, and drive belts, may be considered adequate. Instead of a complete spare cooling unit, the installation may have spare fan motor and expansion valve.

(b) Inventory. The inventory of spare parts shall be minimized by standardizing the type and size of equipment at one installation.

1. INFORMATION REQUIRED FOR DESIGN. Obtain information on the following:

a. Location considerations involved in design of refrigeration systems for cold storage. (Table 1.)

b. Information required about cold storage buildings. (Table 2.)

c. Information required about cold storage operation. (Table 3.)

2. HEAT GAIN. Heat load of cold storage rooms depends on the following factors:

a. Heat gain due to transmission through walls, roof, and floor.

b. Heat gain due to solar radiation.

c. Heat gain from products to be stored.

d. Usage heat gain due to infiltration of outside air when cold storage doors are opened and closed.

e. Internal heat gain due to lights, people, and other heat sources inside the storage room.

f. Heat gain from ventilation air.

3. TRANSMISSION HEAT GAIN. Heat gain due to transmission is a function of outside to inside temperature differentials; overall coefficient of heat transfer of building elements such as walls, roof, floor, doors; and area of surface.

a. Outside Design Conditions. Outside design conditions may be obtained as follows:

(1) Outside design temperatures shall be selected from the appropriate columns of the Army, Navy and Air Force Manual, Engineering Weather Data, NAVFAC P-89, as follows:

(a) Dry bulb temperature from 2-1/2 percent column.

(b) Wet bulb temperature from 5 percent column.

(2) For unlisted location, outside design conditions shall be obtained from Naval Facilities Engineering Command Headquarters.

b. Inside Design Conditions. Inside design requirements depend on products needing to be stored as follows:

(1) For the best results, each product should be stored exactly according to conditions recommended in ASHRAE criteria (Section III 1978, Applications).

(2) However, it is not practicable for command applications to provide a separate room for almost each product since several different products are handled at one site. Therefore, the products to be stored are divided into groups requiring similar storage conditions. Storage conditions for many products are listed in Table 4. Unpackaged meats shall not be stored with unpackaged dairy products or fruits or vegetables.

c. Overall Coefficient of Heat Transfer "U" Factor. For additional data, see appropriate tables in ASHRAE criteria (Chapter 27, 1977 Fundamentals).

(1) The "U" factors will depend mainly on type and thickness of insulation material modified according to the type or thickness of wall or roof construction.

(2) Where wood studs are used for support of insulation, "U" factors shall be increased by 10 percent.

(3) Typical "U" factors for walls and roofs. (Table 5.)

d. Surface Area. The area of cold storage walls, roof, and floor shall be calculated from project drawings.

e. Heat Gain Through Floor. Heat gain through floors will depend on the following:

(1) Ground contact. Where cold storage floor slab is in contact with the ground, design shall be based on the following:

(a) For facilities of 5000 SF or less and winter design ambient of 32 deg. F (0 deg. C) or greater, ventilation pipes or heated ethylene glycol solution piping below slab shall be used to prevent slab upheaval. The outside to inside temperature difference for floor heat gain shall be based on the summer design ambient where ventilation pipes are used and on the temperature of the ethylene glycol solution where this system is used.

(b) For facilities of sizes greater than 5000 SF and/or winter design ambients of less than 32 deg. F (0 deg. C), heated ethylene glycol solution piping below slab shall be used to prevent slab upheaval. The outside to inside temperature difference for floor heat gain shall be based on the temperature of the ethylene glycol solution.

TABLE 1

LOCATION CONSIDERATIONS INVOLVED IN DESIGN OF REFRIGERATION SYSTEMS FOR COLD STORAGE

General Consideration	Specific Factor	Detailed Information or Data Required
Location	Altitude	Elevation above sea level
	Latitude (For calculating solar loads)	(1) North or south of equator (2) Degree Line
	Place	(1) Outside design conditions (2) Unusual surroundings
Environment	Water	(1) Corrosion and scaling properties of local water, see Design Manual DM-3, Chapter 9, Section 5 for details of water conditioning
	Atmosphere	Outdoor contaminants which could affect outdoor equipment, air handling equipment filtration
Local Factors	Labor	(1) Availability, skill and costs (2) Design should be based on use of local labor
	Materials	Availability and costs
	Transportation	Shipping, receiving and storage availability of equipment
	Rules, Regulations & Codes	The design shall comply with relevant sections of Naval rules and regulations; local, State and Federal statutes and codes of authorities having jurisdiction over site, with special considerations to building code, plumbing code, sanitary code, electrical code, fire code, life safety and health codes; and applicable standards of Underwriters' and professional societies.

TABLE 2

INFORMATION REQUIRED ABOUT COLD STORAGE BUILDINGS

Category	Factor	Specific Information Required
Architectural Design	Scale Drawings	Plans, elevations, sections orientation
Structural Design	Type of Structure: Columns, Beams Bracing, etc.	Size, depth, location
	Seismic Effects	Record and Pattern
	Expansion and Settlement Joint	Location and expected movement of joints
Type of Construction	Walls, Roof, Floors	Materials, thickness
	Insulation	Type, thickness, "k" or "C" value, "R" factor
Surrounding Condition	Outside	Design conditions, summer and winter
	Adjacent Spaces	Conditioned or Unconditioned temperatures
	Adjacent Buildings	Shading
Access	Doors	(1) Location, type, size and usage (2) Doors for access to and removal of conditioning equipment. (3) Access of lift trucks
	Stairways and Elevators	(1) Location and size (2) Temperature of connecting spaces (3) Equipment horsepower (4) Ventilation requirements

TABLE 3

INFORMATION REQUIRED ABOUT COLD STORAGE OPERATION

Factor	Details
Lighting:	Wattage at Peak
People:	(1) Average number in space (2) Activity (3) Duration of time in space
Equipment:	(1) Heat output of motors and machines used in space (2) Load factor or hours of operation (3) Weight, temperature, specific heat of material handling equipment brought into space
Product	(1) Type, weight, specific heat of product (2) Temperature of product prior to storage (3) Final temperature of product (4) Maximum allowable chilling or freezing time (5) Frequency of loading (6) Type, weight, specific heat of containers
Ventilation:	Air quantity requirements for product
Infiltration:	Size and usage of doors
Purpose of Building:	Chilling, freezing, storage, single temperature, multiple usage
Location of Equipment:	Ductwork, unit coolers, refrigeration, heat rejection equipment

TABLE 4
TYPE OF COLD STORAGE FACILITIES

Room	Design Temp. Deg. F (Deg. C)	RH Percent	Product Stored
Meat	32 to 34 (0 to 1)	85	Fresh meats, fresh fish, chicken, short term storage
Dairy	40 (4)	80	Cheese, eggs, butter
Perishable Foods	35 to 40 (2 to 4)	90	Fruits, vegetables, with storage temperature of 30 to 40 deg. F (-1 to 4 deg. C)
Film (Less than 1 year)	45 to 55 (7 to 12)	40	Unprocessed photographic film
Film (Longer than 1 year, except motion picture film.	0 to -10 (-18 to -23)	85	New film in sealed packages
Freezer (Short Term)	0 (-18)	90	Frozen package goods
Freezer (Long Term)	-10 to -20 (-23 to -29)	90-95	Frozen products: Meat, fish, poultry, ice cream, vegetables, etc.
Dry Storage	50 (10)	55-65	Fruits, vegetables and miscellaneous products with storage temperature of 45 to 60 deg. F (7 to 16 deg. C)
Ice	25 (-4)	-	Bagged ice
Medical (Non Frozen)	40 (4)	50	Antibiotics, blood vaccines
Medical (Frozen)	-4 (-20)	85	Bone and blood
Environmental	-55 (-48) to High Temperature	-	Altitude chambers

SI Units in Parentheses (Deg. C)

TABLE 5

TYPICAL "U" FACTORS FOR WALLS AND ROOFS
BTU/(HR) (SQ. FT) (DEG. F)[*]

Insulation Thickness Inches (mm)	Polyurethane k=.16	Polystyrene k=.20	Polystyrene k=.25	Fiber Glass k=.23	Cellular Glass k=.36
2 (51)	.069(0.392)	.083(.471)	.100(.567)	.093(.528)	.132(.749)
3 (76)	.048(0.272)	.059(.335)	.071(.403)	.066(.375)	.097(.551)
4 (102)	.037(0.210)	.045(.255)	.056(.318)	.052(.295)	.076(.431)
5 (127)	.030(0.170)	.037(.210)	.045(.255)	.042(.238)	.063(.358)
6 (152)	.025(0.142)	.031(.176)	.038(.216)	.036(.204)	.054(.307)

[*] SI Units in Parentheses (W/(m^2)(k))

Notes:

(1) "U" factors are calculated assuming a resistance R of other components including films of 2.0 (hr.) (sq.ft.) (deg. F)/(Btu).

(2) R values for insulation may be found from formula R=X/k R=resistance (hr.) (sq.ft.) (deg F)/BTU

(3) Cellular plastics cannot be used as interior finish materials.

(4) Insulation shall have a flame spread rating not higher than 75, and smoke developed rating not higher than 150 by Standard Test Method for Surface Burning Characteristics of Building Materials ASTM E-84 test. Cellular plastic insulation shall be tested in the same density and thicknesses as the material that will be used in construction applications.

(5) Insulation has no flame spread or smoke limitations:

(a) Where it is contained entirely within panels, and

(b) Where the entire panel assembly as will be used in the actual construction application meets the flame spread and smoke developed limitations cited above in (4), and

(c) Where it is listed by Underwriters' Laboratories or Factory Mutual.

4. HEAT GAIN DUE TO SOLAR RADIATION. Heat gain from solar radiation must be considered according to the following:

a. Heat Gain Through Walls. Usually cold storage walls and ceilings have insulation not less than 2 inches thick which increases the thermal inertia of the structure. The effect of solar radiation on insulated walls is to increase that load only slightly. (See ASHRAE criteria, Chapter 27, 1977, Fundamentals.)

b. Heat Gain Through Roof. The effect of solar heat gain on roofs exposed to solar radiation is somewhat appreciable, and should be considered in heat load calculations. (See ASHRAE criteria, Chapter 25 and 27, 1977, Fundamentals.)

5. PRODUCT HEAT GAIN. Products brought into cold storage in bulk are usually at temperatures higher than inside design temperature.

a. Heat Gain From Products. Heat from products is a function of weight of products loaded in cold storage at each shipment, specific heat of products, temperature of products at time of loading, final product temperature and cooling time which is usually 18 to 24 hours.

b. Respiration Influence. Some Products continuously give off heat as respiration, reaction, or latent heat, but the total amount thus added into the space is usually negligible as compared to the sensible heat of products.

c. Heat Load Factors. The quality of products brought in each shipment for storage will depend on the project criteria. The temperature of the products, if precooled, is usually assumed to be 10 deg. F higher than the storage temperature. The temperature of uncooled products is usually assumed to be the same as the outside design temperature. Refer to ASHRAE criteria for specific heat of various products. (See Chapter 29, 1977, Fundamentals.)

d. Product Freezing Influence. Generally, only frozen products are brought for storage in a freezer room.

(1) If the freezer room is used to freeze the products, the heat load will be a function of weight of material to be frozen, latent heat of fusion of materials and freezing time.

(2) See ASHRAE criteria for latent heat of different materials (Chapter 29, 1977, Fundamentals.)

(3) Some sensible heat load may also be encountered depending on factors such as weight of products loaded, temperature of the product at the time of loading, specific heat above freezing, specific heat below freezing, and final temperature.

e. Product Heat Gain Calculations.

(1) Heat removal in cooling from the initial temperature to some lower temperature above freezing:

EQUATION: $$Q=Mc_1(t_1-t_2) \qquad (2\text{-}1)$$

(2) Heat removal in cooling from the initial temperature to the freezing point of the product:

EQUATION: $$Q=Mc_1(t_1-t_f) \qquad (2\text{-}2)$$

(3) Heat removal to freeze the product:

EQUATION: $$Q=Mh_{if} \qquad (2\text{-}3)$$

(4) Heat removal in cooling from the freezing point to the final temperature below the freezing point:

EQUATION $$Q=Mc_2(t_f-t_3) \qquad (2\text{-}4)$$

Where

Q = heat removed, Btu.
M = mass of the product, pounds.
c_1 = specific heat of the product above freezing, Btu per (pound) (deg. Fahrenheit).
t_1 = initial temperature of the product above freezing, deg. Fahrenheit.
t_2 = lower temperature of the product above freezing, deg. Fahrenheit.
t_f = freezing temperature of the product, deg. Fahrenheit.
h_{if} = latent heat of fusion of product, Btu per pound.
c_2 = specific heat of the product below the freezing point, Btu per (pound) (deg. Fahrenheit).
t_3 = Final temperature of the product below the freezing point, deg. Fahrenheit.

It is usually necessary to remove heat from the product in a given number of hours. The equivalent in Btuh is:

$$q = Q/(\text{hours required for products cooling})$$

6. INFILTRATION HEAT GAIN. Heat gain due to infiltration will depend on frequency of door opening. (See ASHRAE criteria for calculation of heat gain due to infiltration, Chapter 27, 1977, Fundamentals.)

7. INTERNAL HEAT GAIN. Internal heat gain for small cold storages is usually negligible because people are present only during loading and unloading periods, lights are on only during these periods, and other appliances are seldom provided. For special applications having internal heat loads, see NAVFAC DM-3.5.

8. VENTILATION HEAT GAIN. Ventilation is seldom supplied to cold storage facilities since people are usually only present for loading and unloading. If ventilation is required for people or product, load is calculated the same as infiltration.

9. CAPACITY OF REFRIGERATION EQUIPMENT. The capacity of refrigeration equipment shall be based on satisfying the total maximum heat load of cold storage for 24 hours with operation of refrigeration equipment for 16 hours because of air defrost requirements. If electric, hot gas or water defrost is used, capacity could be calculated on 20 hour operation.

Section 3. REFRIGERATION SYSTEM

Part 1. SYSTEM SELECTION

1. REFERENCE. Refer to ASHRAE criteria. (Chapter 24, 1976, Systems.)

2. TYPE OF SYSTEMS. The type of refrigeration system may be either direct expansion or indirect. Normally, direct expansion systems should be used. Indirect systems may be preferred if the installation has a large number of rooms and each room is maintained at a different temperature, or if refrigerant piping is very long and complicated due to interconnections. The final selection shall be based on a system of having the lowest life cycle cost.

a. Direct Expansion System. In this system, the air in cold storage is cooled directly by an evaporator or evaporators connected to a compressor.

b. Indirect System. These systems may have one or two compressors with shell and tube evaporators for cooling brine. The evaporator and compressor with or without condenser are usually arranged in a factory-assembled package. In this system, brine is circulated through the unit coolers in various cold storage rooms. It is cooled to a temperature required to satisfy the heat load of the room designed for the lowest temperature.

3. REFRIGERANTS. Refrigerant selection affects both first costs and operating costs. The most common refrigerants used in cold storage refrigeration systems is one of the halocarbon compounds R-12, R-22, R-502 or ammonia (R-717).

a. Halocarbon Refrigerants.

(1) Halocarbon refrigerants are rated as group 1 according to ANSI B9.1-1971/ASHRAE 15-1978 Safety Code for Mechanical Refrigeration and are considered nonflammable. Refrigerants R-22 and R-502 are classified as group 5a according to Underwriters' Laboratories classification of comparative hazard to life of gases and vapor while R-12 is classified as group 6. These ratings indicate the halocarbons are safer to use than ammonia and are generally preferred for safety reasons.

(2) The halocarbon refrigerants can be used satisfactorily under normal conditions with most of the common metals such as steel, cast iron, brass, copper, tin, lead, and aluminum.

(3) Because of their physical properties, the halocarbons are better suited to air-cooled condensing. The required lower compression ratios allow the use of lighter equipment.

(4) Types of halocarbon refrigerants:

(i) Refrigerants 12, 22, and 502 should be used within their saturated suction temperature range for single stage compressors, but they may be used down to -80 deg. F (-62 deg. C) with compound or cascade systems.

(ii) Refrigerant 22 is preferred over R-12 in single stage systems because only approximately 60 percent of the compressor displacement is required. Additional advantages are: less refrigerant circulated per ton, smaller pipe sizes, and higher suction pressures.

(iii) Refrigerant 12 is used by compressor manufacturers to increase the number of available units without adding sizes of compressors. It does have a slightly lower brake horsepower per ton than Refrigerant 22.

(iv) The main disadvantage with Refrigerant 22 is oil return, and therefore, an efficient oil separator must be used.

(v) R-502 has lower brake horsepower per ton than R-22.

b. Ammonia Refrigerant.

(1) Ammonia is rated group 2 according to ASHRAE 15-1978 Safety Code for Mechanical Refrigeration and is considered explosive when present in a range of 16 to 25 percent by volume in air. It is rated as group 2 according to Underwriters' Laboratories classification of comparative hazard to life of gases and vapors. Although ammonia is a toxic material, it is considered a self-alarming refrigerant. Its smell makes leaks quickly detectable.

(2) Most of the common metals can be used with ammonia with the exception of copper, brass, bronze, and zinc.

c. Refrigerant Performance.

(1) Table 6 lists comparative refrigerant performance for four common refrigerants based on 5 deg. F (-15 deg. C) evaporation and 86 deg. F (30 deg. C) condensation.

TABLE 6
COMPARATIVE REFRIGERANT PERFORMANCE[*]

Refrigerant	BHP/Ton (W/W)	Compression Ratio	Refrigerant Circulated lb./min./ton (kg/s/W)	Typical Saturated Suction Temperature Deg. F (Deg. C)
R-717	0.989 (0.210)	4.94	0.422 (11)	-10 to 40 (-23 to 4)
R-12	1.002 (0.212)	4.08	4.00 (108)	30 to 50 (-1 to 10)
R-22	1.011 (0.214)	4.03	2.86 (77)	30 to 50 (-1 to 10)
R-502	1.079 (0.229)	3.75	4.38 (118)	-40 to 0 (-40 to -18)

[*] SI Units are in parentheses

(2) Because of the higher required compression ratio, ammonia compressors are heavier in construction than halocarbon compressors; but because of the greater refrigeration effect, the compressor size is smaller or the operating speed is less. The heavy duty machinery generally has a long life and low operating costs.

d. Leak Detection of Halocarbon Refrigerants.

(1) There are several methods of leak detection, the most common being the electronic detector and the halide torch. The operation of the electronic detector depends on the variation in current flow due to ionization of decomposed refrigerant between two oppositely charged platinum electrodes.

(2) The halide torch is a fast and reliable method of detecting leaks of halocarbon refrigerants. Air is drawn over a copper element heated by a methyl alcohol or hydrocarbon flame. If halocarbon vapors are present, they will be decomposed and the color of the flame will change to bluish green. The electronic detector is the most sensitive although the halide torch is suitable for most purposes.

e. Leak Detection of Ammonia. Ammonia leaks are quickly detected by smell. Location can be found by burning a sulfur candle in the vicinity of the suspected leak or by bringing a solution of hydrochloric acid near the object. If ammonia vapor is present, a white cloud or smoke of ammonia sulfite or ammonium chloride will be formed. Ammonia can also be detected with an indicating paper which changes color in the presence of a base.

4. COMPRESSOR EQUIPMENT. Positive displacement compressors are used for these refrigeration systems.

a. The types of positive displacement compressors used are reciprocating, rotary, and helical rotary (screw). (See ASHRAE criteria, Chapter 12, 1979, Equipment.)

(1) In general, reciprocating or helical rotary compressors with factory-assembled condensing equipment are used.

(2) For rooms with high product loads, it is generally required to have some form of capacity control such as suction valve lift unloading, cylinder head bypass, or multiple units. On compressor systems with saturated suction temperatures lower than 20 deg. F (-7 deg. C), cylinder unloading should not be used; individual compressors shall be cycled.

(3) The compressor shall be selected for suction pressure corresponding to design compressor suction temperature. Allowance shall be made for pressure drop in suction piping from the evaporator to the compressor.

For unitary equipment, allowance must be made for compressor running time versus defrost time. For off cycle defrosting, the net refrigeration effect of the compressor based on 16 hours of operation shall not be less than the design heat of the cold storage room for 24 hours. For hot gas or electric defrost, 20 hours running time is acceptable. For central station system with multiple evaporators, the compressor can run all the time.

(4) If a compressor is connected to more than one evaporator and the evaporators are operating at different saturated suction temperatures, then the compressor capacity shall be based on the saturated suction temperature corresponding to the evaporator with the lowest saturated suction temperature.

b. Compressor Drives. Electric motor-driven compressors are generally used for stationary installations. (See Electrical criteria in Section 8 for sizing motors.) If electricity is not available, compressors may be driven by gasoline or diesel engines.

(1) Types of electric motor drives.

(a) Open-type. These motors are usually squirrel cage type induction motors and may be connected directly to or by belts to the compressor. These motors shall have normal starting torque characteristics unless the compressor has no unloading capability. The high starting torque motors are then required. This type motor is always used on ammonia systems and parallel type operation of halocarbon refrigerants.

(b) Hermetic type and semi-hermetic type. These motors are in a separate class since they are cooled by liquid or vapor refrigerant at temperatures much lower than air-cooled motors. They are not rated on a horsepower basis, but generally by full load and locked rotor currents. Their use is limited to less than 25-ton (88 kW) systems. Hermetic or semi-hermetic systems shall not be used with ammonia systems.

(2) Gasoline and diesel engines. Gasoline or diesel engines may be used if electric power is not available.

(a) Starting. In order to start the engine unloaded, provide a clutch between the engine and compressor. The clutch may be magnetic slip, hydraulic, or mechanical type.

(3) Vibration Isolation. Compressor motor assemblies shall be vibration isolated in accordance with DM-3.10.

5. HIGH SIDE EQUIPMENT, CONDENSERS, AND CONDENSER WATER SYSTEMS.

a. Air Cooled Condensers. Economy of operation, installation, availability of, and noise level are factors indicated below that should be considered in selecting an air-cooled condenser unit.

(1) Application. Air-cooled condensers are usually used as an integral component of a factory-assembled unit or in factory-matched components. Larger air cooled condensers may also be applied where it is not economical to provide a water-cooled system due to high cost, unavailability or unsuitability of water, where 12 month operation is required, or where an economic analysis favors air-cooled condensing. The most common available sizes range from 2 to 100 tons (7 to 352 kW) capacity.

(2) Economical selection. For a given compressor and entering air temperature, the condensing pressure increases and cooling capacity decreases as the condenser size is reduced. An entering air temperature increase also produces these effects for a given condenser. Generally, for economical selection, the condensing temperature should be about 15 deg. F(-8 deg. C) above the entering air temperature. The condensing temperature shall not be lower than 10 deg. F (-12 deg. C) and not more than 30 deg. F (-1 degree C) above entering air temperature. To obtain compressor systems that operate more efficiently and with longer compressor life, the saturated condensing temperature should not be more than the following:

(a) 115 deg. F (46 deg. C) when the saturated suction temperature is zero deg. F (-18 deg. C) or above.

(b) 110 deg. F (43 deg. C) when the saturated suction temperature is below zero deg. F (-18 deg. C).

The ultimate maximum condensing temperature shall not exceed 130 deg. F (54 deg. C).

(3) Grade installations. Units should be elevated above the ground to prevent dirt accumulation on the coil.

(4) Roof installations. Design for roof-mounted units should compensate for higher roof ambients by increased air flow and location of air intake.

(5) Noise. Due to the large quantity of air handled, air-cooled condensers are usually noisy. While locating air-cooled condensers, consideration shall be given to transmission of objectionable noise to the adjoining area. For acoustical control, see NAVFAC DM-1, DM-3.10, and DM-3.15.

(6) Air intake. The air intake side shall be free from any obstructions that restrict air flow. Air intakes shall be located away from the air discharge side to avoid bypassing of warm discharge air.

(7) Head pressure control. In systems using thermal expansion valves, a relatively constant pressure of the high side needs to be maintained. Cold ambient air temperatures across the condensers can cause a condensing pressure so low that expansion valves on the evaporator will not operate properly. In mild climates, cycling of the condenser fans can maintain near constant head pressure.

Units scheduled to operate at low ambient temperatures shall be provided with automatic head pressure control systems such as the following:

(a) Pressure controlled dampers to regulate condenser cooling air.

(b) Throttling valve that maintains high side pressure by reducing liquid flow from the condenser, thus flooding part of the condenser and reducing active condenser surface.

The system selected should minimize operating horsepower.

b. Water-Cooled Condensers. Three types of water-cooled condensers are discussed below. The use of each type and the most economical types are also discussed. Water-cooled condensers, when clean and properly sized, generally operate more efficiently than air-cooled condensers especially in periods of high elevated air temperatures. Water is usually supplied from cooling towers, but a closed water circulating system is also used extensively.

(1) Types. Types of water-cooled condensers are as follows:

(a) In the double pipe condenser, the tube carrying the water is placed within a larger tube, and the refrigerant gas is passed between inner and outer tubes. This type is usually used only with small self-contained units.

(b) The shell-and-coil type has a coil of pipe placed inside a closed shell. The cooling water flows through the coil while the refrigerant gas to be condensed is discharged by the compressor into the shell. This type is used on small and medium-sized condensing units.

(c) The shell-and-tube condenser has a number of tubes which are assembled in a large shell. The cooling water is circulated through the tubes and discharge gas from the compressor is condensed in the shell. Shell-and-tube condensers can be cleaned and serviced very conveniently. Large sizes are often provided with marine-type boxes through which all the tubes can be reached without disconnecting the water pipes.

(2) Economical selection. The following factors which affect initial and operating costs, shall be considered carefully to select the most economical condenser:

(a) Increasing the size of a condenser will increase the compressor efficiency, but at the same time the initial cost of the condenser will also increase.

(b) Increasing the cooling water flow will increase the condenser capacity, but will also increase the pumping cost or the water consumption.

(c) Reducing the shell diameter and increasing the tube length will reduce the initial cost, but will increase frictional pressure drop of the water circuit.

(d) The fouling factor, sometimes termed the scaling factor, depends on the quality of water. Generally, new condenser ratings are based on a fouling factor of 0.0005, and where water with higher scaling characteristics will be used, a higher fouling factor shall be selected. The condensers selected for a higher fouling factor will be more expensive. (See ASHRAE criteria for fouling factors, Chapter 16, 1979, Equipment.)

(3) Closed system. Water is force circulated through condenser and closed coil in a cooling tower. Water is permanently treated in the system and does not deteriorate. Condenser efficiency is kept high. The system is different from the evaporative condenser system because water or brine is circulated in lieu of refrigerant. Rate of flow of water is about 4 gallons per minute per ton of refrigeration.

c. Evaporative Condensers.

(1) Operation. The evaporative condenser is an air induced-draft cooling tower with a bare pipe refrigerant coil wetted by a water spray located in the air stream. The water evaporating over the tubes extracts heat from them and causes condensation of the hot discharge gas circulating through the tubes. The water is recirculated by a pump and the water level is maintained by a float control.

(2) Water consumption. The water consumption, inclusive of evaporation, drift, and other losses, will be 2 to 5 gallons per hour per ton of refrigeration.

(3) Economic selection. Generally, evaporative condensers are selected on the basis of 20 to 30 deg. F (-7 to -1 deg. C) difference between the entering wet bulb and condensing temperatures. Selections for the lower differences will result in lower power consumption.

(4) Subcooling. In some instances, evaporative condensers are provided with liquid subcooling coils to reduce the temperature of liquid below the temperature equivalent to the condensing pressure. While subcooling of the liquid increases the total refrigeration capacity, a more beneficial effect is the elimination of expansion valve and liquid line troubles from gas flash.

(5) Limitations.

(a) Evaporative condensers may be used only when the water available for make-up does not have scaling properties and with suitable water treatment. Evaporative condensers are more prone to corrosion and scaling than any other type of condenser due to high surface temperature of tubes, presence of air together with water, and electrolytic action of dissimilar metals.

d. Receivers

(1) Application. In shell-and-coil and shell-and-tube condensers, the shell may also be used as the receiver for the condensed refrigerant. Separate receivers are provided for air-cooled, double-pipe and evaporative condensers.

(2) Size. Every refrigeration system shall have a receiver large enough to hold the entire refrigerant charge of the system during seasonal shutdown or maintenance work. The refrigerant charge shall have a volume no more than 90 percent of the receiver volume for storage temperature of not more than 105 deg. F (41 deg. C); for storage temperatures higher than 105 deg. F, the charge shall have a volume of not more than 80 percent of the receiver volume. If the receiver of a factory-built system is not large enough, provide an additional receiver to supplement it.

(3) Gas binding at receiver. For data concerning gas binding, see ASHRAE criteria. (Chapter 26, 1976, Systems.)

e. Condenser Water System.

(1) Heat rejection. Quantity of heat to be rejected is dependent on the evaporator load plus the heat equivalent of the power required for the compressor.

(a) Quantity of heat rejected. This can be obtained from the compressor ratings or estimated by multiplying a heat rejection factor by the evaporator load or by adding the equivalent heat of the driver power to the evaporator load.

(i) Heat rejection factor. Typical heat rejection factor (HRF) (ratio of condenser heat rejected to heat absorbed by evaporator) for compressor driver refrigeration systems using refrigerant 12 can be selected from Tables 7 or 8.

(ii) Total heat rejection. The quantity of heat to be rejected to the condenser water is a function of the type of compressor, type of refrigerant and the operating temperatures. The total heat rejection ratio can vary from 15,000 BTUH/TON to 36,000 BTUH/TON (1.25 to 3 W/W).

(b) Quantity of condenser water. The quantity of water required by a condenser depends on the following:

(i) Heat rejection of the refrigeration system (q).

(ii) Entering condenser water temperature t_{ew}; generally 75 deg. F (24 deg. C) for the domestic water and 85 deg. F (29 deg. C) for the cooling tower water.

(iii) Leaving condenser water temperature t_{1w}; generally 95 deg. F (35 deg. C) for compressor driven refrigeration system.

(iv) Quantity of condenser water can be calculated from the following formula:

$$GPM = \frac{q}{500 \times (t_{1w} - t_{ew})}$$

GPM = Quantity of condenser water, GPM
q = heat rejection, BTUH
t_{1w} = Temperature of the leaving water deg. F
t_{ew} = Temperature of the entering water, deg. F

TABLE 7

HEAT REJECTION FACTORS - OPEN COMPRESSORS

REFRIGERANT 12

Evaporator Suction Temperature Deg. F (Deg. C)	Condensing Temperature Deg. F (Deg. C)					
	80 (27)	90 (32)	100 (38)	100 (43)	120 (49)	130 (54)
-30 (-34)	1.42	1.46	1.52	1.57	1.63	1.71
-20 (-29)	1.38	1.42	1.47	1.51	1.57	1.64
-10 (-23)	1.34	1.37	1.42	1.46	1.51	1.57
0 (-18)	1.30	1.33	1.37	1.41	1.46	1.51
10 (-12)	1.26	1.28	1.31	1.35	1.40	1.45
20 (-7)	1.22	1.24	1.27	1.30	1.34	1.38
30 (-1)	1.18	1.20	1.23	1.26	1.29	1.33
40 (4)	1.14	1.16	1.18	1.21	1.25	1.28
50 (10)	1.10	1.12	1.15	1.18	1.21	1.24

TABLE 8

HEAT REJECTION FACTORS - HERMETIC OR SEMI-HERMETIC COMPRESSORS

REFRIGERANT 12

Evaporator Suction Temperature Deg. F (Deg. C)	Condensing Temperature Deg. F (Deg. C)					
	80 (27)	90 (32)	100 (38)	100 (43)	120 (49)	130 (54)
-30 (-34)	1.47	1.51	1.57	1.64	1.70	1.77
-20 (-29)	1.43	1.46	1.52	1.58	1.64	1.71
-10 (-23)	1.38	1.42	1.48	1.53	1.59	1.65
0 (-18)	1.34	1.38	1.43	1.48	1.54	1.60
10 (-12)	1.30	1.34	1.39	1.44	1.50	1.56
20 (-7)	1.26	1.29	1.34	1.38	1.43	1.49
30 (-1)	1.22	1.25	1.29	1.33	1.38	1.43
40 (4)	1.17	1.21	1.25	1.29	1.33	1.38
50 (10)	1.13	1.17	1.20	1.24	1.29	1.33

(2) Once-through system. In a once-through system, the warm water leaving the condenser is usually wasted to a drain. The cooling water supply shall be from raw water sources only, such as wells, lakes, ponds, rivers, or the sea. Once-through systems are usually designed for a flow rate of 1.5 to 2 gallons per minute per ton (0.332 to 00.45 cubic metres per second per watt (m^3/s/W)) when the temperature of water entering the condenser is maintained at 75 deg. F (24 deg. C) or higher or throttled to give the effect of 75 deg. F (24 deg. C) water.

(a) Well water system. Systems shall be designed to use well water only when wells of sufficient capacity are available. Wells may be drilled for refrigeration if economically or technically feasible. Existing wells may be used for condenser cooling if the water is of acceptable quality and temperature.

(i) Disposal of waste water. Waste water is usually returned to dispersion wells or leaching fields to make up for the water drawn from the cooling wells.

(ii) Condenser construction. Condenser design should be adaptable for mechanical cleaning. Materials used in construction should be suitable for quality of water available.

(b) Lake, river, or sea water system. Economic considerations involved in selection of this system shall include the cost of pumping installation, filtration equipment, and special construction materials required by the quality of the water.

(3) Condensing water cooling devices. Cooling towers or spray ponds used to cool condenser water to eliminate uneconomical domestic water waste. These cooling devices shall be located to minimize problems of drift, fogging, air recirculation, noise, and/or entry of contaminants into the device such as foliage, fly ash, and earth.

(a) Performance. Recirculation systems are usually designed for a flow rate of 2.5 to 4 gallons per minute per ton (0.55 to 0.88 m^3/s/W) which will correspond to a cooling range of 8 to 12 deg. F (-13 to -11 deg. C) . Actual flow rates will depend on evaporating temperature, condensing temperature and design wet bulb temperature.

(b) Cooling pond or spray pond. Condenser water will be cooled by cooling pond or spray pond only if the architectural scheme of the site so permits.

(i) Spray nuisance. If a spray pond is used, consideration shall be given to the effect of moisture carryover due to wind.

(ii) Water treatment. Water treatment for control of algae shall be provided. Controlled bleed off of water shall be provided to control salts build-up in system.

(iii) Water consumption. Water consumption will depend on spray design and prevailing wind velocities at the site. For average conditions, total water consumption due to pond evaporation and other losses may be 10 percent of the quantity circulated.

(c) Atmospheric cooling tower. The following disadvantages should be considered when selecting atmospheric cooling towers:

(i) Space. Atmospheric cooling towers need more space than mechanical draft cooling towers.

(ii) Water consumption. The total water consumption due to evaporation and other losses will be 3 to 5 percent of quantity circulated.

(iii) Reliability of performance. The performance is not reliable, since it is affected by wind velocity and direction. The selection should be based on zero miles per hour wind velocity.

(d) Mechanical draft cooling tower. Mechanical draft cooling towers may be induced-draft or forced-draft type. Induced-draft type towers are preferred because they are less susceptible to recirculating moist air discharged from the unit and they consume less energy due to lower hp requirement.

(i) Recirculation of air. Towers shall be so installed as to avoid any recirculation of discharge air through them. If there is any chance of recirculation, towers shall be oversized by selecting them for a wet bulb temperature higher than normal design wet bulb temperature.

(ii) Water consumption. The water consumption due to evaporation and other losses including bleed-off to prevent scaling or buildup of deposits will be approximately from 1-1/2 to 5 percent of the quantity circulated.

(e) Cooling tower construction. The cooling tower should be constructed from materials that will give a reasonable service life for the specific service conditions. Construction shall meet fire and safety codes. (See Water Cooling Towers, NFPA Standard No. 214.)

(f) Capacity control. Capacity control shall be provided on cooling towers designed to work the year around, with a variable load so that the water temperature will not fall below 75 deg. F (24 deg. C). Control types include:

(i) Small indoor cooling towers. Use modulating dampers on outlet.

(ii) Large outdoor cooling towers. Provide multispeed fan motors and modulating dampers on air inlets.

(iii) By-pass condenser water into cooling tower basin or interior storage tank or into condenser water supply line. A modulating three-way diverting control valve shall be used upstream of cooling tower or two-way valve shall be installed in the bypass line.

(g) Winter operation. These precautions shall be followed if condenser water systems are expected to work in freezing weather:

(i) Outdoor piping. All outdoor piping exposed to freezing weather shall be arranged to drain into the cooling tower pan during off cycle. As an alternate, outdoor piping shall be traced either with steam lines or with electric heating cables to prevent freeze-ups.

(ii) Water in pan. During the off cycle, water in the pan shall be drained into a storage tank located in a heated area. Alternatively, water in the pan shall be kept from freezing by heating it with an immersion type heater system (steam or electric). In large water pans, water may be kept warm by circulating it through a steam or electric water heater.

(h) Water treatment. A suitable water treatment shall be included to prevent corrosion and algae growth. Bleed-off to control salts buildup shall be discharged from the basin into the drain which shall be connected either to a sanitary sewer or industrial waste treatment system.

6. LOW SIDE EQUIPMENT, EVAPORATORS. Three different types of evaporators, their applications and characteristics are:

a. Types of Evaporators. Three types of evaporators and the conditions under which they are used are as follows:

(1) Bare pipe evaporators. Bare pipe tubing or pipe is generally used for cooling of liquids in shell-and-tube coolers or in applications for air cooling below the freezing point of water.

(2) Plate evaporators. In cold storage work, plate evaporators are used in freezer rooms. Plates may be used as shelves, in which case the flat surface gives better contact with the product than is obtained with pipe coils, and therefore, rapid chilling of products is achieved.

(3) Finned coils.

(a) Finned coils shall have a fin spacing from about 4 fins per inch to about 8 fins per inch.

(b) The unit coolers used for cold storage shall have finned surfaces with forced circulation of air over them.

b. Selection Consideration. When selecting equipment, the following criteria must be considered:

(1) Refrigerant temperatures. Generally, temperature differences between room and cooling medium shall be as follows:

Freezers, 5 to 10 deg. F (-15 to -12 deg. C)
Medium Temperature such as dairy, 10 to 15 deg. F (-12 to -9 deg. C)
Meat and produce storage, 5 to 10 deg. F (-15 to -12 deg. C)
Meat and produce preparation rooms, 15 to 20 deg. F (-9 to -7 deg. C)

(a) Increasing temperature difference accelerates frost accumulations on evaporator coils and produces dehydration.

(b) Reducing temperature difference produces a high relative humidity which results in an increased rate of spoilage in unfrozen meats and some vegetables.

(c) Actual temperature difference shall be based on the unit cooler performance data to give the optimum humidity conditions for the products to be stored. (See Section 2 for inside design conditions.)

(2) Number of evaporators. The right number of evaporators must be enough to distribute the cool air uniformly over the area to be cooled.

(a) The distribution of cool air shall be uniform in every part of the storage area.

(b) Irregular and/or large rooms may need more than one evaporator to produce uniform coverage of the entire area.

(c) As an alternate, one cooling unit with ductwork may be provided to distribute cooling air uniformly.

(3) Air velocity. The velocity of air for forced circulation units must not be more than 100 feet per minute (0.51 meters per second) as measured over the stored products. Higher velocities increase dehydration of the product.

c. Evaporator Location Considerations. Evaporator or cooling units should be located inside cold storage rooms.

(1) Cooling units should not be suspended in front of doors as the warm air from outside will be drawn directly into the unit when a door is opened. Air flow should be either towards the door or sideways against the door to reduce infiltration when the door is opened.

(2) For large narrow rooms, cooling units shall be arranged to blow across the room width to reduce the air velocity over the stored products.

(3) There shall be workable headroom below unit coolers and drain fittings.

7. REFRIGERANT PIPING. Refer to detailed design procedures in ASHRAE criteria (Chapters 26, 27, and 29, 1976, Systems).

a. "Refrigerant piping systems should be designed and operated to generally accomplish the following:

(1) Assure proper refrigerant feed to the evaporators.

(2) Provide practical refrigerant line sizes without excessive pressure drop.

(3) Prevent excessive amounts of lubricating oil from being trapped in any part of the system.

(4) Protect the compressor from loss of lubrication at all times.

(5) Prevent liquid refrigerant or slugs of oil from entering the compressor, either during operating or idle time.

(6) Maintain a clean and dry system.

(7) Prevent excess suction gas superheat at the inlet of the compressor during normal operation."*

(8) Provide for expansion and contraction of piping.

b. Items to consider to prevent liquids from entering the compressor:

(1) Provide a suction accumulator in the suction piping to trap liquids and meter the liquids into the suction line at a controlled rate.

(2) Pitch hot gas piping away from the compressor. Provide loops in the hot gas line immediately after the compressor to minimize the possibility that condensed refrigerant will drain back to the compressor head.

(3) A tightly closing check valve should be installed in the hot gas line near the condenser to prevent refrigerant that may boil off from the condenser from condensing in the hot gas line or on the compressor heads.

c. Two-pipe refrigeration system. Navy Resale and Services Support Office Commissary projects will only be designed as two-pipe systems. Common main method (three-pipe system) shall not be used.

8. REFRIGERATION SYSTEM ACCESSORIES AND SAFETY CONTROLS. Table 9 lists the required accessories for the refrigeration system.

a. When applying a single compressor to multiple evaporators, control accessories are required to maintain constant evaporator pressure regardless of changes in load.

(1) Multiple evaporators operating at the same temperature.

(a) Install an evaporator suction pressure regulator in the suction line to the compressor to maintain a constant evaporator pressure. For multiple evaporators in same room, temperature can be controlled by thermostat and liquid line solenoid in lieu of the pressure regulator.

(2) Multiple evaporators operating at different temperatures.

(a) Install an evaporator suction pressure regulator in the branch suction line or in the suction line of each evaporator to maintain a constant evaporator pressure.

(b) A check valve may be installed in the evaporator operating at the lowest pressure.

b. Safety controls are used to stop the refrigeration systems until a fault has been located and corrected. Paragraph (2) below contains a listing of minimum controls necessary.

* (Reprinted with permission from the 1976 systems volume, ASHRAE Handbook & Product Directory.)

(1) Operation. Safety controls shall be wired so that when the system stops under the action of any safety control, it will not restart automatically until the fault has been corrected.

(a) On ammonia cold storage plants, the control system shall shutdown the compressor, room units and main liquid line stop valve upon a power reduction/outage and set off an audible and visible alarm locally and remotely in a monitoring office if necessary. The system would be energized manually after normal (or emergency) power is restored.

(2) Controls. The following is a list of minimum controls for safe operation of refrigeration systems:

(a) High discharge pressure cutout.

(b) Low evaporating pressure cutout.

(c) Low oil pressure cutout.

(d) Motor high temperature cutout.

(e) Flow switch for condenser water.

(f) Flow switch for chilled brine.

(g) Low temperature cutout for brine chiller.

(h) Receiver high pressure relief valve.

(i) Motor overloads.

(j) Time delay relays to prevent short cycling.

TABLE 9

LIST OF REQUIRED ACCESSORIES FOR REFRIGERATION SYSTEMS

Accessories	Application
Discharge Line:	
(1) Muffler	Usually for compressors having spread above 1000 rpm.
(2) Oil Separator	For discharge or suction line if oil is likely to be trapped. Recommended for flooded type systems. For system operating lower than 0 deg. F (-18 deg. C) saturated suction temperature.
Liquid Line:	
(1) Sight glass	Standard for all applications.
(2) Strainer	Standard for all applications.
(3) Dehydrator with three-valve bypass	Shall be used for systems having hermetic compressors or below freezing evaporator temperature.
(4) Solenoid Valve ...	Standard for all applications.
Suction Line:	
(1) Suction-to-liquid heat exchanger	Required for each evaporator especially for flooded system or if system has vertical lift for liquid line.
(2) Suction strainer .	When steel piping is used for suction line. Ahead of evaporator pressure regulators.
Isolating Valves .... (Ball or plug valves)	For condenser and evaporator when these are installed at location remote from compressor.

9. SYSTEM SELECTION AND EQUIPMENT LOCATION.

a. Refrigeration System for Small Cold Storage Buildings (Up to 25 tons) (88 kW)). The small plant refrigeration system is usually the single package unit containing evaporator, compressor, air-cooled condenser, receiver, halocarbon refrigerant, and control devices if suitable outside walls are available for the through-the-wall condenser section. Split system package units with remote air-cooled condenser should be used if suitable outside wall is not available. One condenser should be used with each compressor. The condenser can be located outside at ground level, on the building roof, or in a common equipment room with adequate forced ventilation. Compressors in package units are reciprocating hermetic or semi-hermetic type. Refrigerant system is the direct expansion halocarbon type; refrigerants R-12, R-22 and R-502 are all used depending on application. Evaporator fans are the direct mounted propeller type either blowing through a coil bank or pulling air through the coil bank. Larger than 25 ton compressors may be used if so dictated by an economic and energy analysis.

The smallest rooms will usually consist of a single air-cooled condensing unit with a single direct expansion coil in the cold room. Control will be off and on from a thermostat either starting and stopping the compressor or operating a liquid line solenoid valve which will allow the compressor to pump down or shut off on a pressure control. Various means of automatic defrost may be used.

As larger rooms are encountered and also a multiplicity of rooms, a number of condensing units may be employed with multiple evaporators to each unit. The unit package system for wall or the split package system for roof or ground mount of the condensing unit of the factory fabricated type may be used, singly or in multiple on a single room. Life cycle cost analysis should be used to determine most economical choice.

A typical arrangement for the refrigeration equipment applied to a small cold storage facility is shown on Figure 1.

b. Refrigeration System for Intermediate and Large Cold Storage Buildings (Over 25 tons (88 kW)).

(1) Recommended system. The recommended system for intermediate and large cold storage buildings is the recirculated ammonia type. This system should utilize reciprocating compressors, evaporative condensers, fan coil evaporators and necessary accumulators, intercoolers, and recirculators.

Evaporators in all spaces that operate at temperatures above freezing with chill spaces 33 to 50 deg. F (1 to 10 deg. C) shall be on one set of compressors in one system, and evaporators in spaces below freezing with freezer spaces 32 to -20 deg. F (0 to -29 deg. C) shall be on a second set of compressors in another system.

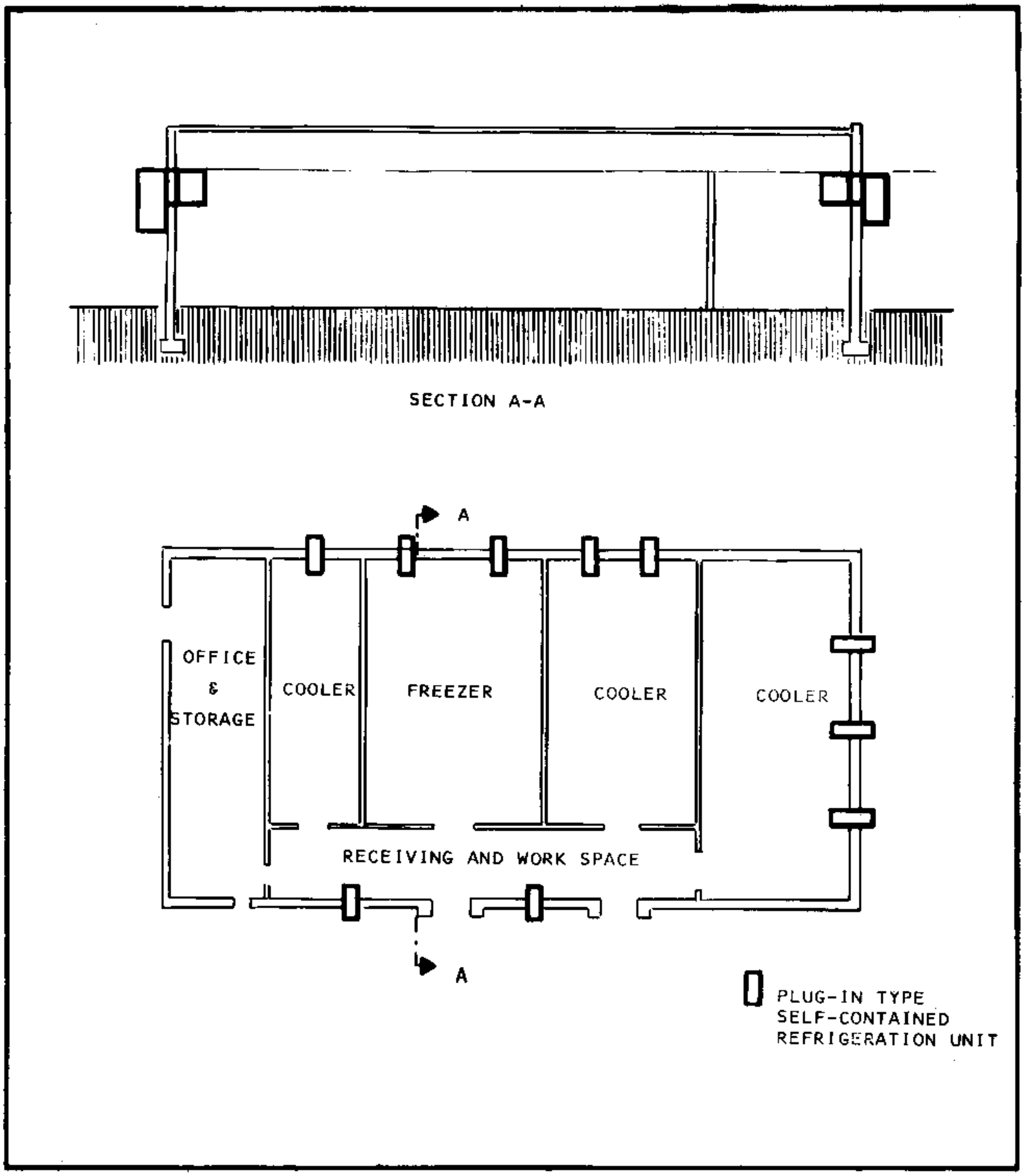


FIGURE 1
Small Cold Storage Facility

A standby compressor shall be provided in both the freezer system and in the chill temperature system for pull-down after unloading and for emergency operation during outage of a compressor unit. For parallel operation, piping shall be provided to equalize crankcase oil levels. In addition to oil equalizers, parallel systems often use an oil reservoir and oil level floats on each compressor.

(2) Direct and indirect systems discussion. There are essentially two types of cold storage refrigeration plants. These are the direct and indirect systems. In the direct system, direct expansion coils are used in the cold rooms with the compressor and high side equipment concentrated in a central machine room. The indirect system utilizes this same type of machine room, and in addition, adds brine chillers. Brine is chilled in the machine room and is circulated by pumps through pipe lines to the cold rooms where it is circulated through the cooling units in the various rooms.

The indirect brine system has advantages in simplicity of operation, ability of system to absorb short load peaks, ease of control, avoidance of possible leakage of refrigerant into storage areas, and flexibility of piping system design. Brine is particularly desirable for convection coil installation and for large spread out systems. The higher initial cost due to the need for pumps, motors, valves, and control equipment; the maintenance costs required for this equipment, and monitoring the condition and strength of the brine; and higher power cost due to added pumps and lower compressor suction temperatures more than offset the advantages.

Plants can be either or both types. One type or the other will usually predominate.

The normal cold storage plant will have at least two and sometimes more refrigerant temperature levels available with one refrigerant piping system at a proper temperature level for cooler storage and another for freezer storage.

The large machine room shall contain a minimum of 2 and possibly up to 4 or 5 refrigerant lines to the storage space. One of the chief advantages of the central machine room is flexibility. By cross connections and the use of two stages, it is almost impossible for a breakdown to occur that will seriously affect the overall operation of the plant. This is particularly true with a multiplicity of machines. Freezer storage is most often handled by booster compressors, with the boosters normally discharging into an intermediate pressure that is also used for cooler storage. The optimum booster compressor discharge may be at variance from the suction pressure required for the coolers, but usually is so close that it is impractical, from an efficiency standpoint, to carry the optimum intermediate pressure plus that required for cooler storage. All high stage machines in the machine room shall be valved so that any machine can operate on any of the high or intermediate pressures. Boosters serving freezers should be interconnected so that they can be valved into whatever suction duty is required. Variations will be found in valving of machines depending on the individual plant design.

(a) Compressors. Three types of compressors are generally used; these are the open reciprocating, screw, and centrifugal type.

Open reciprocating compressors are used in the 25 through 250 ton (88 to 880 kW) range. Screw machines may be the best choice in a range of 200 through 800 tons (700 through 2800 kW). Centrifugal units are usually used for the largest installations.

Many cold storage operations use the medium speed multicylinder compressor, either direct driven or belt driven at operating speeds of 900 to 1200 RPM. The first cost is less than the slower speed machines and maintenance is low and machine life would be long when these machines are used in a properly designed plant. Adequate safeguards are a necessity in the plant, with large suction accumulators and intercoolers being required for proper operation.

Booster compressors most commonly used are the reciprocating type and the screw type. Both render excellent service. In a constant suction pressure plant where the low, or booster, suction is kept at a relatively constant level, the screw and reciprocating types of compressors will both operate efficiently. This type of service is used in freezer storage rooms and in continuous freezers. Where suction pressure can vary widely, as in some batch freezing plants, the screw compressor can cause some problems when compression ratios become too high, but the reciprocating compressor is not particularly bothered by this condition. The reciprocating machine with internal unloading, can more efficiently match compressor capacity with system requirements than the screw compressor. Both types of compressors are used with success in cold storage operations.

Automation of the cold storage facility is required because the cold storage plant must operate continuously 24 hours. Heavy duty equipment, although having a greater first cost, is justified by the longer life and longer maintenance free periods. In sizing electric motors for compressor drives, the horsepower required should be checked for all possible conditions of load that may be encountered in the operation of the plant and a motor selected that will not be overloaded under any conditions that may be imposed upon it by the compressor it is driving.

(b) Condensers. The larger cold storage plants will use cooling towers or evaporative coolers for condensing.

(i) Evaporative condensers. Condensing temperatures and their corresponding pressures should be kept to the practical minimum for long equipment life and economical operation. If evaporative condensers are used, desuperheater coils for the incoming refrigerant gas can be used to good advantage, especially in an ammonia installation. A good desuperheater coil (for ammonia) and proper bleed-off of water, will minimize scaling problems. With parallel operation of evaporative condensers, proper trapping of the outlets and sufficient height of the condenser outlet above the receiver should be observed to prevent liquid backup in the condensers which reduce capacity.

(ii) Cooling tower with water-cooled shell and tube condensers. Condensers shall be oversized sufficiently to assure adequate heat transfer.

(c) Receivers for refrigerant should be sized generously. It may not be necessary to install receiver capacity for the pump-down of the entire plant. Most large plants are somewhat sectionalized so that various sections of the plant may be pumped down and adequate receiver capacity should be installed to hold the charge from the largest section. It is also good practice to use a standby receiver capable of handling a bulk truck shipment of several thousand pounds of refrigerant, since cost can be lessened by buying refrigerant in larger bulk quantities. Receivers shall be installed adjacent to one another if they are to be operated in parallel for best operation. Parallel receivers shall have equalizing lines between them.

(d) Accumulators and intercoolers. Large accumulators and intercoolers should be used in any ammonia plant and are useful in Halocarbon Refrigeration Systems. Suction line accumulators shall be provided with liquid refrigerant return systems. Accumulators should be large enough to keep refrigerant velocity below a point where any liquid carry over to the suction line will occur. Adequate baffling should be built into the accumulator to prevent splashing or turbulence of the liquid refrigerant from causing liquid refrigerant to enter the suction line. Liquid return systems may be either powered by pressure or by liquid refrigerant pumps. From the accumulators, some systems return the excess refrigerant directly to the plant receiver, while some return it to the liquid line periodically after shutting off the main flow from the liquid receivers.

(e) Refrigerant pumps. When using a liquid refrigerant recirculating system, liquid flow is accomplished with mechanical pumps or by a gas pressure pumping system. The mechanical pumps include open, semi-hermetic, magnetic clutch, and "canned rotor" arrangements with either positive rotary, centrifugal or turbine vane construction. Cavitation and Net Positive Suction Head (NPSH) are considerations when selecting the pump. Sealing of shafts usually requires double mechanical seals with an oil feed from an oil reservoir. Motors are selected with a service factor to take care of operation with cold, stiff oil. Surrounding temperatures, heat gains, operating pressures, internal bypasses, operation of automatic valves, and evaporation of refrigerant are all to be considered in selecting a mechanical pump. (See ASHRAE Criteria, Chapter 25, 1976, Systems.)

(f) Two-Stage refrigeration systems. The main operating economy in two-stage plants is obtained by prechilling the liquid refrigerant at the intermediate pressure before using it in the low stage evaporators. This requires the use of some type of intercooler. This intercooler also serves the function of chilling the booster discharge gas to a saturated condition. For efficient and economical operation, the liquid chilling feature should not be eliminated from an intercooler. Intercoolers should be generous in size and with some reserve for future plant growth.

(g) Evaporators. Evaporator equipment in the various storage rooms is mostly confined to some type of forced air evaporator unit, either floor or ceiling mounted. Pipe coils should not be used because their first cost is high and defrosting is difficult.

The typical ceiling type evaporator consists of a cooling coil with fins at various spacings depending on the temperature of the room and manufacturer of the coil. Sizes of coils will usually vary from 2 to 20 tons (7 to 70 kW) refrigeration capacity. Air circulation is obtained by a propeller or squirrel cage fan, either blowing through the coil or pulling air through the coil. Drain pans under the coil are used to catch drip from condensation or defrost as the case may be. Drain pans are sometimes insulated to prevent external drip from a cold pan. A number of ceiling fan units placed in a line and blowing out from one wall of a cold storage room can cover wide rooms without duct work and even temperatures and uniform air flow can be maintained. The more units in the line, the wider the room that can be spanned. With high ceilings in a cold storage room, a distance of 100 to 150 feet (30 to 46 m) may be spanned by the blower coils along one wall of the room with the blower units evenly spaced. The multiplicity of units all blowing in the same direction tends to get the entire mass of room air circulating in a parallel pattern so that the entire room is well covered with adequate air circulation.

Care should be exercised when using ceiling type blowers that maintenance considerations be included in the design. These coils are up and out of the way and there is sometimes a tendency to forget about them until trouble develops. A maintenance inspection schedule shall be posted in a conspicuous place to prevent breakdown of equipment. In cooler and freezer storage rooms, propeller fans are provided when no duct work is involved, since the propeller fan is more efficient than the centrifugal fan when very small pressures are needed. Fans may be direct connected to the shaft of the driving motor or belt driven depending on the size and horsepower required to drive the fan. The larger units will employ slow speed belt driven fans with standard motors. The smaller units will utilize smaller higher speed direct motor mounted fans. Many times the smaller fan units will not have replacement parts when they need replacement after a few years and will require new unit purchases. Good practice in freezer operation is to use units with the fan or fans mounted to pull air through the coil and discharge it out into the room. The coil defrosts the air and the fan is less likely to get frosted. In most instances the floor-type unit consists of a coil and fan or fans mounted above a drain pan and all encased in a suitable housing. Centrifugal fans are normally used since air must normally be conducted up to the ceiling level of the cold room and turned to spread out in the room. This imposes some resistance and more horsepower is usually required for the floor-type unit than for a comparable ceiling unit. Air entering a floor-type unit also makes a 90 degree turn to flow through the coil and in a standard ceiling unit passes straight through the unit without turns. The main advantage of the floor unit is ready accessibility for maintenance and repair. The disadvantages are that it takes up floor space that could otherwise be used for merchandise storage and that it, if not heavily guarded, is subject to damage from materials handling equipment. Since floor-type units are usually larger than the ceiling type, fewer are used per room and piping costs will normally be less in a total installation. This will about offset the normally higher cost of the floor-type unit so that the total installation cost and equipment cost will not vary significantly regardless of the type units used.

(3) Summary for intermediate and large cold storage buildings. In the warehouse system, reliability and low operation cost resulting from efficient design and application of machinery should be the first consideration of the owner. Year round operation and maintenance of temperatures and conditions in the warehouse rooms within narrow limits must be maintained. For this reason, reliability is paramount, which means heavy duty machines and non-over-loaded motors and equipment. Operating costs are also important. These costs consist of power cost and maintenance, replacement, and repair costs. For best results, all of these add up to obtaining a system containing the best and most efficient components available. A typical arrangement for the central refrigeration equipment is shown for an intermediate cold storage facility on Figure 2. This arrangement would also apply to a large cold storage building.

c. Equipment Location. The refrigerating equipment for large refrigerated rooms should be located in a separate machine room which should include ample space for the equipment and its maintenance. It should have adequate ventilation, be segregated from other areas, and be located on a outside wall and have separate exits. Small and medium size prefabricated rooms may have refrigeration equipment mounted on top or alongside.

(1) Air-cooled condensers, evaporative condensers or water cooling towers may be located on the roof or at grade adjacent to the machine room.

(2) The evaporator equipment may be located in the conditioned space or in a penthouse over the refrigerated rooms. The penthouse offers many advantages:

(a) Storage area is more fully utilized.

(b) Defrost water drains can be piped through penthouse walls to discharge on the main storage roof.

(c) Equipment is not subjected to physical damage by stocking trucks.

(d) Service on cooling equipment and controls can be handled by a single individual from floor or roof deck location.

(e) Maintenance and service costs are minimized.

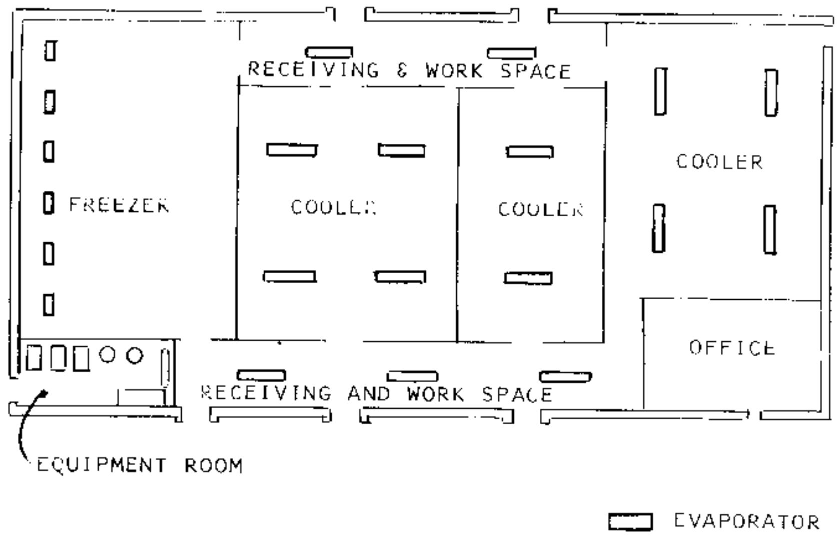


FIGURE 2
Intermediate Cold Storage Facility

Part 2. SYSTEMS OPERATING

FROM 40 Deg. F to -80 Deg. F

1. DEFROSTING. Evaporators operating on refrigerant temperatures below freezing will accumulate frost which causes reduction in capacity. The effect of frost is most serious in finned-tube coils as the frost blocks the air passages, retarding both air flow and heat transfer. Therefore, it is necessary to defrost evaporators regularly before the frost coating seriously affects the air flow and heat transfer. The accumulation of frost increases with a reduction in refrigerant temperature and an increase in usage load.

a. Air Defrosting. Air defrosting may be used only for cold storage rooms maintained above freezing temperatures. The evaporator defrosts during off-cycle, that is, when the room does not need cooling and the supply of cooling medium is stopped. The air circulating fan continues to run melting the frost. Systems requiring long off-cycle periods for defrosting produce objectionable variations in the room temperature. A time clock may also be used to operate the defrost cycle. When defrosting, some of the moisture on the coil will be put back into the room. This is not desirable for low humidity applications.

b. Defrosting by Heat. The small unit coolers used for rooms maintained at temperatures below freezing may be defrosted by supplying heat. During the defrosting cycle, heat is supplied to melt the frost and the air circulation fan is stopped to prevent carry over of defrosted moisture. The drain pan for collection of melted frost and drain pipes are usually heated by electric resistance cables. The defrosting cycle is controlled automatically by a time switch or program switch which should have adjustment for frequency and duration of the defrosting cycle. The heat may be supplied by any of the following means:

(1) Water spray defrosting. In this system, a water spray header is provided over the finned-tube coil through which water is sprayed over the coil until all frost is melted.

(a) The sensible heat of water is used to melt the frost. The supply of refrigerant to the evaporator is stopped and the drain pan and the drain pipes are usually heated electrically during the defrosting cycle.

(b) Water drains. Arrangements shall be provided to drain off all the water from the spray header and the water supply pipe within the cold storage room before normal cooling operation is started.

(c) Limitations. The water spray defrosting system is most economical, but the defrosting cycle is usually prolonged and may be used only when a continuous supply of water is available. Water temperature should be above 40 deg. F (4 deg. C). This method of defrost may be used for rooms operating at 0 deg. (-18 deg. C) or above.

(2) Electric defrosting. In this system, the finned-tube coil is built with electric heating elements placed either between the fins or inside the tubes to supply heat during the defrosting cycle. This system is simple to operate and maintain, but usually both initial and operating costs are high. This can be an efficient and relatively rapid method. It can be used for very low temperature applications and is a simple system to operate and maintain.

(3) Hot gas defrosting. This can be used for low temperature applications and provides a rapid method of defrosting. This method utilizes compressed vapor from the compressor to apply heat directly to the evaporator, and in some systems, also to the drain pan. Most systems use the latent heat of condensation of the compressed vapor as the heat source, but some use only sensible heat of highly superheated vapor. (Refer to manufacturer's recommendations.)

"To assure an adequate supply of hot gas, it is recommended that not over one- third of the evaporator load on the system be defrosted at one time. The remaining two-thirds of the evaporator load must continue to operate to supply the necessary gas. Some evaporators require three or more times their equivalent in refrigerating evaporators on the line to provide sufficient heat for defrost in allowable time."[*]

"The hot gas should be piped into the drain pan coil first to prevent freezing the water which runs off the coil. The hot gas leaving the drain pan coil passes through a spring-loaded check valve or inverted trap, and enters the evaporator downstream from the expansion valve (Figure 3). Some manufacturers introduce the hot gas at the suction connection, and bypass the expansion valve through a relief valve into the suction line downstream from a suction solenoid valve which closes during defrost (Figure 4)."[*]

"Without the check valve or inverted trap, liquid refrigerant will boil in the hot gas line and drain pan coil and cause frosting of drain pans and hot gas lines. This is particularly important in multiple evaporator systems."[*]

"After the hot gas solenoid valve is closed, residual hot gas may condense in the hot gas line and pan coil. Spring-loading the check valve forces the condensed hot gas to re-evaporate at a saturation temperature above the room dew point. Condensate and frost cannot form on the hot gas line or pan coil."[*]

[*] (Reprinted with Permission from the 1976 System Volume, ASHRAE Handbook & Product Directory.)

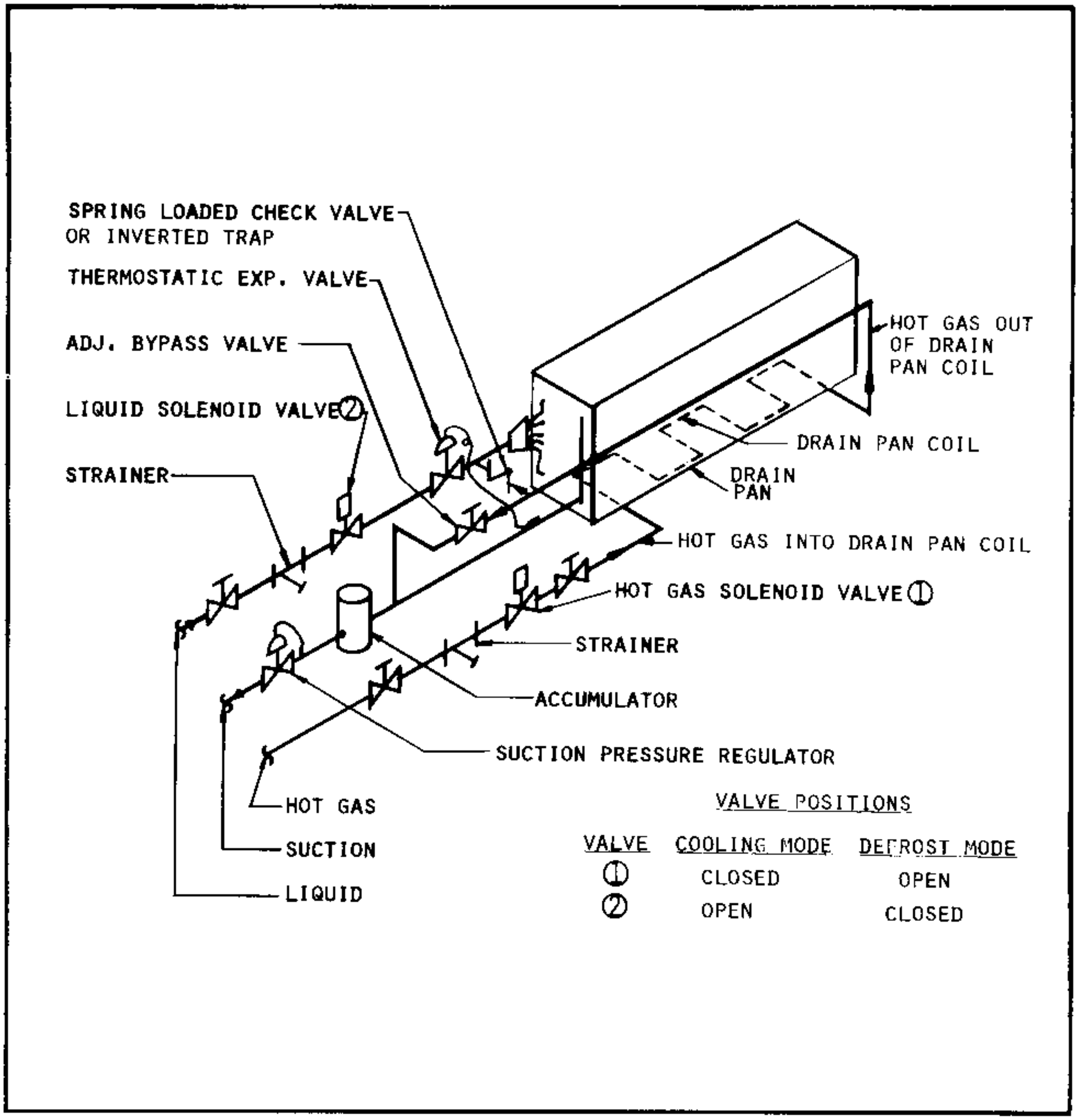
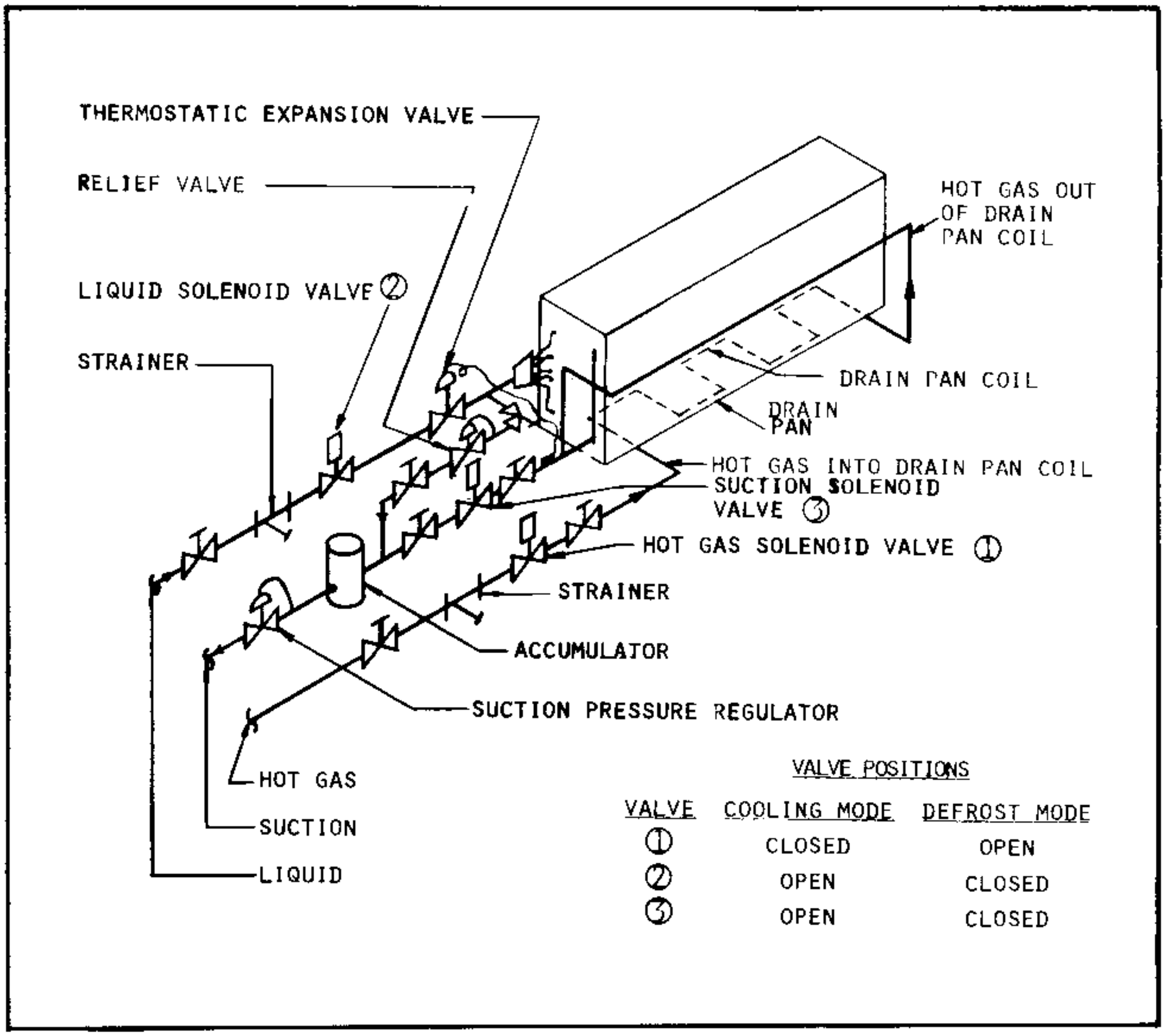


FIGURE 3
Hot Gas Defrost Piping Schematic
(Hot Gas Into Liquid Connection)



VALVE POSITIONS

VALVE	COOLING MODE	DEFROST MODE
①	CLOSED	OPEN
②	OPEN	CLOSED
③	OPEN	CLOSED

FIGURE 4
Hot Gas Defrost Piping Schematic
(Hot Gas Into Suction Connection)

When hot gas enters the evaporators, the pressure begins to rise, and liquid refrigerant in the coil together with fresh condensed refrigerant, formed during defrost, enter the suction line. A re-evaporator or accumulator sufficient to trap and prevent slugs of liquid must be provided.

To prevent excessive suction pressures and consequent motor overload during defrost, a suction pressure regulator (holdback valve) at the compressor suction is often used. This valve is adjusted to remain wide open during refrigeration and to throttle during defrost.

c. Brine Spray System. By continuously spraying brine on the evaporator coil, the frost is prevented from forming. Brine which may be used in food storage rooms includes sodium chloride and propylene glycol. Sodium chloride may be used in systems cooling rooms to 10 deg. F (-12 deg. C) or above. Propylene glycol may be used for rooms down to -35 deg. F (-37 deg. C). The brine concentration decreases with constant moisture pick-up and therefore arrangements must be provided to keep the concentration constant either by boiling away the moisture or by regularly adding the salt or glycol.

d. Defrosting of Plate or Pipe Coils. The capacity of the pipe or plate coils is not affected seriously by frost, especially if the coils are selected to give required cooling with normal frost thickness.

(1) Frequency of defrosting. It is not necessary to provide for frequent defrosting of pipe or plate coils. The coils are occasionally defrosted by stopping the refrigeration compressor when the room happens to be completely empty or by emptying the room for a short period.

(2) Expediting defrosting. It is possible to expedite defrosting by brushing away the frost or by blowing warm air over it from a portable blower.

e. Defrosting Controls.

(1) Initiation of the defrost cycle should be automatic, either by means of a running time monitor, time clock, or air pressure differential control.

(2) Defrost cycle should be as short as possible and is terminated by sensing the coil temperature in which 40 deg. F (4 deg. C) would indicate the removal of frost. Restart of fan is delayed until coil temperature has returned to normal operating temperature.

2. DRAIN PIPING. Drain piping shall be as follows:

a. Size. Drain pipes shall be large enough and shall have sufficient pitch to allow free flow of water and small particles of frost removed during defrosting. It should be no less than 1 inch (25 mm) in diameter and shall have pitch of not less than 1/2 inch (13 mm) per foot (300 mm).

b. Length. The length of the drain piping in the cold storage room should be as short as possible. Drain lines should be trapped outside the refrigerated space to prevent infiltration of moisture.

c. Protection of Drain Piping from Freezing.

(1) Drain piping within a cold storage room operating below 32 deg. F (0 deg. C) requires some means of heating during defrost periods. Depending on the method of defrost, electric heat or hot gas tracing of pipes should be used.

3. LUBRICATION. As the evaporating temperature is decreased, the following problems are encountered: (a) trapping of oil in the evaporator, and (b) freezing of water and oil wax in the expansion valve. The following precautions are recommended to eliminate oil problems:

a. Oil Separator. Each compressor shall be provided with a discharge oil separator to reduce carry over of oil to the evaporator. (See ASHRAE criteria, Chapters 26 and 29, 1976, Systems.)

b. Type of Oil. Oils shall be wax-free napthenic base type, with viscosity 280 to 300 SSU (Saybolt Seconds Universal) (60-65 CS) at 100 deg. F (38 deg. C). The moisture content of oil shall not be above 20 parts per million. (See ASHRAE criteria, Chapter 32, 1976, Systems.)

4. TYPE OF SYSTEM. System type is generally dependent on the evaporator temperature. As the suction pressure of the compressor is reduced, there is a drastic reduction in refrigerating effect and an increase in horsepower per ton.

a. Single Stage System. Single stage compressors are usually limited to evaporator temperatures down to -40 deg. F (-40 deg.).

b. Multistage System. Multistage systems consist of more than one stage of compression. (See ASHRAE criteria Chapters 24 and 29, 1976, Systems.)

(1) Compound system. Compound compressors are usually used for producing temperatures below -40 deg. F (-40 deg. C).

(a) In a compound system, compressors are connected in series using a single refrigerant, the discharge of low stage compressor being the suction of an intermediate or high stage compressor.

(b) The choice of two or three stage compression system is indicated by the limitation of compression ratio of the compressors involved, as well as by owning and operating costs.

(2) Cascade system. A cascade system utilizes a series of refrigerants of progressively lower boiling points. Each refrigerant is used as a cooling medium to condense, under pressure, the refrigerant of the next lower boiling point.

(a) Cascade systems are used for temperatures below -40 deg. F and high-pressure low-volume refrigerants permit the use of small compressors in factory-assembled packages.

(3) Multistage systems are mostly used for larger field erected plants.

5. EVAPORATORS. Materials and type of construction used for evaporators shall not be susceptible to temperature shock and thermal fatigue.

6. HEAT EXCHANGER. Liquid to suction heat exchangers shall be selected to dry the suction vapor. Superheating of suction vapor shall be avoided to prevent overheating of compressor.

7. PRESSURE DROP. Refrigerant lines shall be sized generously to prevent excessive pressure drops and resultant drastic reduction in efficiency. Suction and hot gas lines should be sized for pressure drops equivalent to 2 deg. F (1 degree C) change in saturation temperature for halocarbon refrigerants and 1 degree F (0.6 degree C) for ammonia. High pressure drops reduce capacity and increase horsepower requirements.

8. SPECIAL CONSIDERATION. For systems operating from -20 to -80 deg. F (-29 to -62 deg. C), use detailed design procedure in ASHRAE criteria. (See Chapters 24 and 29, 1976, Systems.)

SECTION 4. BRINE CIRCULATING SYSTEMS AND LIQUID REFRIGERANT RECIRCULATING SYSTEMS

1. BRINES. In an indirect refrigeration system, water is generally used as a secondary cooling medium for temperatures down to 40 deg. F (4 deg. C). Applications require cooling medium temperatures below 40 deg. F (4 deg. C) employ chemical solutions of water having freezing temperatures substantially below the operating temperature. (See ASHRAE criteria for properties and selection considerations for the brines (water solutions of calcium and sodium chloride and propylene glycol), Chapter 17, 1977, Fundamentals.)

a. Calcium Chloride. Calcium chloride brine is the most common secondary refrigerant down to -40 deg. F (-40 deg. C). Corrosion is the principal problem for which chromate treatment is recommended.

b. Sodium Chloride. Sodium chloride brine is used for applications where, due to hygienic reasons, contact with calcium chloride brine may not be permitted. It is also preferred for spray-type unit coolers for cold storage rooms. Sodium chloride brine should not be used below 10 deg. F (-12 deg. C).

c. Propylene Glycol. Propylene glycol solution can be used for temperatures down to -35 deg. F (-37 deg. C). This brine may be more expensive than the calcium or sodium chloride brines and shall be inhibited to neutralize corrosive properties.

d. Usage. Brines are used in larger systems where safety and easy piping is considered of prime importance. Brines are desirable for transmitting refrigeration because: (1) in the event of leakage, brine is less objectionable than refrigerant gases; (2) individual temperature control of each space or fixture may be simpler than with a direct expansion refrigerant; (3) sharp but short load peaks may be absorbed in a brine system, particularly if it is designed as a storage system; and (4) if brine sprays are used, a defrost system is not required because any moisture condensed in the cooling is dissolved in the brine.

e. Undesirable Features. Brine systems have certain undesirable features including: (1) corrosion of equipment is often possible due to chemical or electrical action; (2) additional equipment and maintenance due to the need for pumps, motors, valves, and control equipment, and the required attention to correcting the condition and strength of brine; (3) possibility of equipment damage due to freezing if brine condition and refrigerant plant operation are not properly supervised; and (4) normally higher power cost due to added pumps and lower compressor suction temperature.

2. REFRIGERANTS USED AS BRINES. Where circulating brine systems are required below -35 deg. F (-37 deg. C), the usual brines are unusable. Below this temperature, the following brines have been used: trichloroethylene, methylene chloride, Refrigerant 11, methanol, ethanol, and acetone. These are specialized systems and require pressurization of the refrigerant to prevent evaporation. (See information in ASHRAE criteria, Chapter 28, 1976, Systems and Chapter 17, 1977, Fundamentals.)

3. BRINE PIPING SYSTEMS.

a. Piping, Pumps, and Valves. The piping, pumps, and valves shall be of materials and sizing to suit the brine used.

b. System Frictional Pressure Drop. For charts and tables on viscosity, specific gravity and other required information, see ASHRAE criteria, Chapter 17, 1977, Fundamentals; and Carrier System Design Manual, Part 4.

c. Make-Up. Make-up for brine system shall be mixed in a tank to proper proportions and pumped into the system. Brine systems shall not be connected directly to potable water supplies.

4. LIQUID REFRIGERANT RECIRCULATING SYSTEMS. A very excellent method of feeding refrigerant where direct expansion is used is by means of the liquid recirculation method. In this system refrigerant liquid is fed into a low pressure receiver connected to the suction of the load being worked upon. The liquid refrigerant flashes down to the temperature corresponding to the suction pressure and the chilled liquid is then pumped to the evaporator units in the cold storage rooms. Instead of boiling off all of the liquid refrigerant in the evaporator unit as is done with flooded or expansion valve operation, an excess of liquid is fed in to the evaporator unit. In ammonia plants, this flow will be as high as 4 or 5 times more refrigerant pumped through the evaporator unit than is evaporated. In the case of the halocarbon refrigerants, slightly less liquid is normally pumped. With the increased flow of liquid, the liquid refrigerant becomes, in part, a brine flowing through the evaporator and giving very high performance by keeping the entire inner surfaces of the refrigerant tubes wet with refrigerant which increases the heat transfer ability of the tubes. The excess liquid refrigerant flows back to the low side receiver along with gas from evaporation. At the receiver, the gas separates and is pumped back to the compressor and on to the high side. The return cold liquid drops into the liquid pool in the low side receiver and is again circulated through the system.

Liquid flow may be accomplished either with mechanical pumps designed for liquid refrigerant flow or by patented pressure pumping systems in which regulated high side pressure is used to force the liquid refrigerant through the low side units. In some of these systems the chilled liquid is isolated in relatively small drums or vessels, and the high side pressure applied to force the liquid into the system. The high side pressure is normally reduced so that no more pressure than that required to force the liquid is applied. Alternating drums can be used so that a continuous flow of liquid can be assured by allowing one drum to fill while the other is feeding. In either system, pump or pressure feed, the end result is to overfeed the low side units. Liquid recirculation has a number of advantages over either flooded operation or direct expansion. Control is simplified in that all of the refrigerant flow controls are outside of the chilled areas and at one location at the low pressure receiver. A simple solenoid valve in the liquid inlet to low side unit is sufficient to shut off the flow of liquid for temperature control or for defrost application. Other controls can be used on the low side unit if more sophisticated control is desired, but the basic flow controls are concentrated at one vessel which is of some advantage in any type plant. Close temperature differences between refrigerant and room temperature can be maintained by this system. Evaporator surface is used more efficiently than with other methods of direct refrigerant cooling and a minimum amount of surface is required for good results. Since more liquid refrigerant is circulated than in the conventional refrigeration system, larger liquid and suction lines are necessary. In a large plant, the first cost of a recirculated liquid system will be very little, if any more than with any other good refrigerant cycle, direct expansion or flooded. In any recirculating system utilizing refrigerant pumps, a spare pump should always be included as insurance with each system. Any refrigerant system can successfully use a liquid recirculation system, but the most commonly used refrigerant with these systems is ammonia. (See ASHRAE criteria, Chapter 25, 1976, Systems.)

Section 5. INSULATION

1. GENERAL INSULATION PRACTICE.

a. Requirements. Insulation is required on piping and duct work to conserve energy and prevent condensation. (See ASHRAE criteria, Chapter 19, 1977, Fundamentals for charts and tables for thickness of insulation required to prevent condensation at various operating temperatures and ambient dew point temperatures.) Wherever insulation is used to prevent condensation, a suitable vapor barrier shall also be used on the warm side to prevent absorption of moisture by insulation and corrosion of equipment. The surface of the insulation shall not be exposed in the refrigerated spaces. Outdoor installation requires a waterproof protective covering.

(1) Piping to be insulated. The following types of piping are to be insulated:

(a) All brine supply and return piping.

(b) Refrigeration suction piping.

(c) Drain piping.

(d) Condenser water piping where insulation is required to protect outdoor piping from freezing or where cooling towers are used year round.

(e) Refrigerant discharge piping where people can come in contact.

(f) Refrigerant liquid piping when temperature of surrounding space is higher than condensing temperature.

(2) Equipment to be insulated. The following is a list of equipment that is normally insulated:

(a) Brine pumps.

(b) Unit cooler fans and casings located external to cooled space.

(c) Brine chillers.

(3) Connections. Piping connections to equipment coils and drain pans, valves, and unions shall be fully insulated and arranged so that all condensation flows to and through the drain pan.

2. COLD STORAGE INSULATION. Economic factors, minimum thickness of insulation, and installation of insulation are as follows:

a. Economic Considerations. Factors such as cost of insulation and amount of insulation must be carefully considered.

(1) Cost of insulation. In cold storage plants, the cost of insulation of cold storage rooms amounts to a substantial part of the total cost of the installation. As the thickness of the insulation is increased, its cost goes up, but the cost of refrigeration decreases.

(2) Selection. The thickness and type of insulation shall result in minimum total life cycle costs comparing the operational expenses over the 25 year life of the structure with first cost. (For method of determining the economic thickness, see ASHRAE Handbook, Chapter 19, Fundamentals.) The Economic Analysis shall be prepared in accordance with NAVFAC P-442, Economic Analysis Handbook, to determine optimum insulation thickness for cold storage buildings.

b. Minimum Thickness of Insulation. Walls, floors, and doors will be insulated. Problems of conductivity, solar radiation, and heat gain must be given adequate consideration.

(1) Conductivity. The insulation material generally used for cold storage applications should have conductivity between 0.16 and 0.36 British thermal unit hour per square foot per degree F per inch of thickness.

(2) Heat gain. Using the recommended insulation material, the heat gain should be calculated based on an optimum insulation thickness that has been determined by life cycle costing including energy costs, maintenance costs, and construction costs of installation. (See charts 1 through 8 in Appendix B.) (See Table 10 for general recommendations for R-values.)

(3) Solar radiation. Solar radiation shall be considered in heat gain calculations and in determining insulation thickness and equipment sizing.

(4) Smoke and fire safety. Interior finish and insulation shall comply with the requirements of DOD 4270.1-M (June 1978), fire protection criteria, and other criteria required to assure satisfaction in service performance.

(5) Shading. Solar heat gain to the storage facilities should be reduced by providing external shading wherever possible. This shading can be obtained from nearby buildings, trees, or the external construction of overhangs or projections on the buildings.

c. Installation. Insulation shall be installed in a minimum of two layers with staggered joints.

(1) One side vapor barrier. A vapor barrier is essential on the warm side to prevent moisture condensation in the insulation.

(2) Two side vapor barrier. The enclosures subjected to alternately warmer and cooler temperatures than the surrounding temperatures may require a vapor barrier on both sides of the insulation, particularly if high humidities inside the enclosure are encountered.

(3) Leakage of moisture at junctions of floors, walls and ceilings of cold storage warehouse shall be prevented with vapor barriers with adequate expansion provisions.

3. FREEZER ROOM FLOORS. Insulation shall be provided under floor slabs for all cold storage rooms maintained at temperatures below freezing with an auxiliary heating system below the floor insulation to keep the soil temperature above 32 deg. F (0 deg. C).

a. Risk. Freezing of the ground under the freezer space may cause heaving of the floor or column and wall footings. The risk becomes considerable with storage temperatures below 15 deg. F (-9 deg. C) and in rooms over 10 to 15 feet (3.3 to 4.9 m) in width. Lower temperatures increase the risk. The problem may be quite acute with unfavorable fine grained soils such as fine sands, silts, and clays. The problem is lessened with dry fill provided at least 6 feet (2 m) below the slab.

b. Drainage. Positive drainage of dry fill is essential, especially if there is any possibility of seepage of the ground water.

c. Heating Systems. The heating of the floor may be done by several means.

(1) Panel heating. The floor can be constructed with appropriate heating means installed to keep the temperatures below the floor above the freezing point.

(2) Electric coils. Electric heating coils may be installed in conduit in the floor slab and controlled from floor sensors installed in the slab.

(b) Brine piping. Hot glycol brine may be circulated through piping installed in the floor. This piping should be installed in conduits buried in the floor to facilitate the repair of the piping.

(c) Air heating. Ducts built into the floor can be used to circulate air to heat the floor. It is possible that the available outdoor air could be circulated unheated when the outdoor temperatures permit.

TABLE 10

RECOMMENDED INSULATION R-VALUES

Type of Facility	Temp. Range (Deg.)	RECOMMENDED R-VALUES[*] (HR>)(SQ. FT.)(deg F/(BTU)		
		Floors	Walls/ Suspended Ceilings	Roofs
Cooler[***]	40 to 50 F (4 to 10 C)	Perimeter Insulation Only	17 to 30 (3 to 5.28)	25 to 35 (4.4 to 6.16)
Chill Cooler[***]	28 to 35 F (-2 to 2 C)	15 to 20 (2.64 to 3.52)	25 to 35 (4.4 to 6.16)	30 to 40 (5.28 to 7.04)
Holding Freezer	-10 to 20 F (-23 to -7 C)	30 to 35 (5.28 to 6.16)	35 to 40 (6.16 to 7.04)	40 to 50 (7.04 to 8.8)
Blast Freezer[**]	-40 F (-40 C)	35 to 40 (6.16 to 7.04)	50 to 60 (8.8 to 10.56)	50 to 60 (Suspended Ceilings) (8.8 to 10.56)

[*] Due to the wide range in the cost of energy, and the even wider range in the cost of insulation materials on a thermal performance basis, a recommended R-value is given as a guide in each of the respective areas of construction. (SI Units are in parenthesis; the R-values are (m^2)(K)/W).

[**] R-values shown are for a blast freezer built within an unconditioned space.

[***] If a cooler has the possibility of being converted to a freezer in the future, consideration should be given to insulating the facility with the higher R-values from the freezer section.

1. GENERAL PURPOSE COLD STORAGE CONTROL. Operation of the entire refrigeration plant shall be completely automatic.

a. Type of Control. Generally, it is only necessary to control the room temperature for above freezing temperatures; defrost control shall be added for storage temperatures below 32 deg. F (0 deg. C). The room shall be provided with a room type or remote bulb type, electric thermostat with adjustable differential 3 to 5 deg. F (2 to 3 deg. C).

b. Control Arrangement. Controls for a single compressor will be as follows:

(1) One compressor for one room. A room or remote bulb thermostat shall control the compressor motor. The refrigerant liquid line solenoid valve should cycle with the motor.

(2) One compressor for more than one room. A thermostat shall control the liquid line solenoid valve of the respective room, while the compressor shall be under control of a low pressure switch. If rooms are at different temperatures, evaporator pressure regulators shall be provided.

c. Thermostat Location. Thermostats or sensors shall sense average room temperature and shall be located in a place having good air movement. For example, on the intake side of circulating fans avoid locating thermostats or sensors in a direct stream of supply air.

d. High Limit Thermostat. An alarm to indicate excessive temperature in cold storage space or a compressor fault shall be located locally and also located in the office or other similar supervisory area. One alarm light shall be provided for each room and for each compressor on panels with audible alarm with manual silencing switch.

e. Relative Humidity Control.

(1) When maintaining high relative humidities, the temperature rise from the coil leaving temperature to the room temperature must be at a minimum. The temperature difference between the coil leaving temperature and the refrigerant temperature must also be small.

(2) For rooms requiring conditions such as 60 deg. F (16 deg. C) and 40 percent RH or 50 deg. F (10 deg. C) and 60 percent RH, the relative humidity is controlled by maintaining a proper coil leaving temperature or dew point and maintaining the space temperature with reheat. Since this is not energy efficient, it is necessary to establish the design operating conditions as realistically as possible.

(3) For rooms with relative humidity design of 75 percent to 90 percent, control is obtained by controlling the coil refrigerant temperatures with an evaporator pressure regulator and maintaining the room temperature by cycling a coil solenoid valve.

f. Control Diagram. A control diagram with sequence of operation shall be framed under glass and mounted on wall of mechanical equipment room.

g. Programmable Controller. Considerations shall be given to providing a programmable controller when dictated by an economic and energy analysis.

Section 7. AUTOMATIC ICE-MAKING PLANTS UP TO 10 TONS PER DAY

1. PLANT SIZE. The size of an ice-making plant depends on the population of the area to be served. For details of ice consumption allowance for normal use, see Table 11.

a. Hospital Use. The ice required per bed per day for hospital use shall be three times the amount specified for normal use.

b. Expansion. Unless otherwise specified, a minimum of 20 percent allowance shall be made for future expansion.

c. Size Definition. The plant size is generally specified in tons of ice (2,000 pounds per ton) (907 kg) produced in 24 hours.

d. Adjustment for Site Conditions. Manufacturers usually rate their ice-making plants for supply water temperature of 75 deg. F (24 deg. C) and condensing water leaving temperature of 95 deg. F (35 deg. C). Any increase in supply water temperature or condensing water leaving temperature will result in reduced output and/or high brake horsepower per ton. Therefore, plant capacity shall be based on the conditions encountered at the site.

TABLE 11

ICE CONSUMPTION ALLOWANCE FOR NORMAL USE

Location	Pounds per Day per Person
1. North of 35 deg. latitude in northern hemisphere	1.2
2. South of 35 deg. latitude in southern hemisphere	1.2
3. Temperate Zone	2.0
4. Semi-tropical, tropical and desert area.	2.5

2. TYPE OF ICE. The uses of the various types of ice and the quantity per day per person must be known when selecting type.

a.   Beverage Ice. Tube or cube is generally used for direct consumption in beverages and drinking water.

b.   Packing Ice. Some crushed, flake, or chip ice is also used for direct consumption in beverages, but mostly this type of ice is used for chilling packed products and transport perishables.

c.   Special Application Ice. Unless crushed, flake, or chip ice is required in large quantity for special applications, the plant should be selected to produce tube or cube ice and should have an arrangement to produce some crushed ice when desired.

3. TYPE OF PLANT. The ice plant should be completely factory-assembled package, direct expansion type, suitable for continuous and automatic operation. Plants larger than 5-ton (18 kW) capacity may have the high side as a separate package which can be piped in the field to the low side section.

a.   Refrigerant. Refrigerants from the halocarbon group are preferred.

(1) Refrigerant 12. This refrigerant generally should be used for plants up for 10 tons (35 kW) as it has low condensing pressure, its evaporating pressure for ice-making is above atmospheric pressure, and replacement costs are moderate.

(2) Ammonia. Ammonia systems are more economical for larger plants, usually above 10-ton capacity (35 kW).

b.   Condenser. For additional data on condensers, see Section 3.

4. STORAGE OF ICE. The storage room, where used for day storage only, should accommodate approximately three days full production based on 13 square feet per ton (46 m^2/kW). A smaller storage room may be provided if ice production is handled by more than one machine.

a.   Storage Room Design. The following is a list of criteria that must be met in the design of a storage room:

(1) Volume. Tube or cube ice requires about 62 cubic feet (6.1 m^3/kW) of net storage space per ton. Crushed ice requires about 57 cubic feet (5.6 m^3/kW) of net storage space per ton.

(2) Height. Ice storage rooms should not be over 12 feet (3.8 m) high and ice should not be stored in piles higher than 8 feet (2.5 m).

(3) Division walls. Where storage facilities for two types of ices are required (tube or cube, and crushed), uninsulated partitions shall be provided to divide rooms into two equal volumes or otherwise if desired.

(4) Floors. The floor of storage bins may be inclined if the ice is discharged through floor or side wall hatches for loading trucks or conveyors. The inclination of such a floor shall not be less than 30 deg. (0.52 radians) from horizontal.

(5) Access. Access to storage bins shall be provided in the ceiling or wall to permit occasional entrance of an attendant.

(6) Door. The door, if in a wall, shall have a removable barricade to retain the ice when opened.

(7) Drains. A floor drain with trap shall be provided to drain away water of melted ice.

(8) Insulation. For insulation requirements, see Section 5.

b. Refrigeration for Storage Room. Ice produced by most automatic plants is subcooled to a temperature of 25 deg. F (-4 deg. C) or less. It may be stored without refrigeration up to about 7 days.

(1) Adhesion. With subcooled ice, there is hardly any adhesion between the pieces of ice and the ice will not form one large mass.

(2) Storage without refrigeration. The storage of ice without refrigeration will result in meltage of relatively small amount from the surfaces of the pile in contact with the walls and the floor of the storage room and the upper surfaces below the ceiling. The interior of the pile will not be subject to the same degree of melting and may not melt at all. Ice cakes shall be stored about 6 inches (150 mm) off the floor and at least 8 inches (200 mm) from the walls to permit air circulation around the ice.

(3) Long storage. If it is desired to store the ice for long periods, it should be packed in bags or other containers which should be stored in a refrigerated room.

(a) Before packing, the ice should be "dried" by conveying it slowly for about 5 minutes in an open top, helicoid conveyor through unrefrigerated space to drain off the excess moisture.

(b) Next, it should be conveyed for about 10 minutes through a room or tunnel maintained at about 10 deg. F (-12 deg. C).

(c) Ice "dried" in this manner may be bagged and held indefinitely in a refrigerated room maintained at a temperature no higher than 25 deg. F (-4 deg. C).

(4) Refrigeration. Separate compressors shall be provided for ice storage rooms. The evaporator shall be either plate bank or pipe coil selected to give required cooling without frequent defrosting.

c. Storage Room Insulation. R-values of insulation should be as recommended for a room operating at 24 to 28 deg. F (-4 to -2 deg. C) (Table 10).

5. ICE HANDLING. If possible, ice plants shall be located above the storage room to permit gravity loading of ice. Where gravity loading cannot be arranged, conveying equipment may be required to remove the ice to the storage room.

a. Conveyor Type. The conveyor may be helicoid, bucket, slat, or belt type, either horizontal or vertical as individual cases may require.

b. Conveyor Capacity. The capacity of the conveying equipment shall be based on the amount of ice harvested from the machine periodically.

c. Operation. Operation of the conveyor shall be synchronized with the harvesting cycle of the plant.

d. Ice Removal. Ice, when stored in a pile without refrigeration, may be scooped out with a shovel. It may also be removed through the floor or wall hatches if the floor of the storage room is inclined. Piles may be loosened in large quantities with minimum effort by prodding with a pole or light rod. Any other equipment for ice handling will depend on the purpose for which the ice is to be used.

e. Bagging. Machines for weighing and filling bags and other ice containers may be required if ice is to be stored for long periods. Container may also be required for distribution of ice for automatic vending stations.

6. WATER REQUIREMENTS. Water requirements for ice-making are as follows:

a. Quantity of Water for Ice-Making. In addition to the water required for ice production, allow 10 to 15 percent for extra consumption due to wastage and blow-down for control of salt concentration.

b. Quality of Water for Ice-Making. The following criteria must be satisfied to obtain quality of water desired:

(1) Minimum standard. The quality of water used for ice-making shall be as high as that required for drinking water.

(2) Water hardness. Clear and hard ice will be produced if water hardness (in terms of total sulfates, carbonates, and chlorides) does not exceed approximately 200 milligrams per liter (litre). Bleed-off of a portion of water used for production of ice may be necessary to remove excess dissolved salts.

(3) Water analysis. The chemical compositions of the water should be the determining factor in deciding the treatment of water to be used for producing ice.

7. ICE PLANT OPERATION. Operation of ice-making plants shall be controlled automatically. The production of ice shall proceed until the storage space is filled to a predetermined capacity.

8. REFERENCE. See ASHRAE, Chapter 54, 1978, Applications, for additional design information.

1. INSTRUMENTATION. Instrumentation for refrigeration systems shall be as follows:

a. Indicating Type Instrument. A measuring instrument shall be provided near each equipment unit and each automatic control device (thermostat, humidistat, and pressure switch) to facilitate adjustments and testing of the control device. Such instruments are usually of the indicating type unless a permanent record of operation is desired. Typical application of these instruments other than at control devices are listed hereinafter:

(1) Thermometers with thermometer wells.

(a) Brine inlet and outlet at coils and chiller.

(b) Condenser water inlet and outlet at condenser.

(c) Direct expansion coil refrigerant suction connection.

(d) Bearings of large compressors and motors.

(2) Pressure gages with gage cocks and snubbers when required.

(a) Suction and discharge of pumps.

(b) Inlet and outlet of strainers.

(c) Suction and discharge of compressors.

(d) Lubrication system of compressors.

(e) Brine inlet and outlet of coils and chiller.

(f) Condenser water inlet and outlet at condenser.

(g) Before spray nozzles at cooling tower, evaporative condenser, or spray coil evaporator.

(h) Upstream of evaporator pressure regulator.

b. Recording Instruments. Recording thermometers, hygrometers, and other special recording instruments shall be provided for large depot-type facilities and where permanent records of conditions are critical for storage or required to analyze operating costs.

c. Combination Instruments. Recording and indicating instruments should be combined with the control device to measure conditions at the point of control.

2. SAFETY PROTECTION AND SECURITY PRECAUTIONS.

a. Protection Against Hazards. Protective devices on hazardous equipment shall be in accordance with OSHA requirements and as specified herein.

(1) Moving equipment. Provide protective guards to eliminate accident hazards from equipment in motion such as:

(a) Belt drives.

(b) Couplings.

(c) Propeller fans.

(d) Centrifugal fans with free discharge or free suction.

(2) Equipment with high operating temperatures. If surface temperatures are high enough to cause burns, the equipment shall be either insulated or shielded and shall have a notice warning about the hazard.

(3) Electric motors. All electric motors not within sight of their controllers and without cutout devices near the motor location shall be provided with a lockout switch.

(4) Drips. To prevent slippery floors, pipe all drips of water or oil to the nearest floor drain.

(5) Ventilation. Refrigeration equipment rooms shall be ventilated to remove the refrigerant from the room space in case of accidental leakage. (See ASHRAE 15-1978 Safety Code for Mechanical Refrigeration.)

(6) Emergency switch. Mechanical equipment rooms that house large refrigeration equipment shall have break-glass emergency switches located near the exits. These switches shall be wired to stop the power supply to all equipment except exhaust fan or fans.

b. Personnel Safety and Security.

(1) Door. Cold storage doors shall open from inside as well as outside.

(2) Alarm. Each room shall have a push button wired to alarm bell outside the room, and to a location constantly manned or monitored on a 24- hour basis, so that any person who gets locked inside the room can call for help.

(3) Security alarm. If there is a security system, alarms should be installed to indicate unauthorized entry.

3. ELECTRICAL CRITERIA. Electrical criteria for refrigeration systems shall be as follows:

a. Standards. All the electrical equipment shall comply with standards of the National Electrical Manufacturers Association (NEMA), Institute of Electrical and Electronics Engineers (IEEE), American National Standards Institute (ANSI), and the American Society of Heating, Refrigerating and Air Conditioning Engineers (ASHRAE). (For additional data, see NAVFAC DM-4 series.)

b. Motor Starting Characteristics. Selection of motor starting characteristics shall be based on inrush current limitations of the electric installation.

c. Motor Controllers. Motor controllers for equipment shall include:

(1) Overload protection.

(2) Momentary contact type, start-stop, push button control for manually operated controller.

(3) Hand-Off-Automatic selector for automatically operated controller with safety controls common to "Hand" and "Automatic" operation.

(4) Low voltage release.

(5) Auxiliary contacts for interlocking and control circuit.

(6) Control circuit transformer, when control voltage is different from the line voltage.

(7) Control circuit fusing in all controllers, NEMA size 2 and larger.

(8) Time-delay relay where several motors are interlocked to start automatically in a predetermined sequence.

(9) Pilot light on cover when motor is installed out of sight of the controller.

d. Emergency Operation. Check and prepare a list of the equipment requiring operation during emergency. All such equipment shall be wired from the emergency supply panel.

e. Special Construction. All electrical switches, controls, and wiring wherever possible shall be installed outside the cold storage space so that the amount of special moisture-proof equipment and wiring will be kept to an absolute minimum.

(1) Motor. Compressor motors shall be selected for pull-down periods when the load on the motor will be more than the normal load, such as higher suction temperature. Motors inside cold storage spaces shall be totally enclosed type.

(2) Type of fittings. All electrical fittings inside cold storage rooms shall be vapor-proof type.

(3) Wiring. For wiring inside cold storage rooms, use synthetic rubber-sheathed multiple conductor cable or other cable suitable for wet location and 75 deg. C rise operations. When cable is run inside a conduit, the ends of the conduit shall be sealed off to prevent condensation inside.

4. OTHER ITEMS TO CONSIDER IN DESIGN.

a. Door Opening Sizes. Door opening is dependent upon the type of material handling equipment used and size of pallets that storage items are stacked on. Double acting doors are generally sized as 8 feet(2.6 m) wide by 10 feet (3.3 m) high.

b. Door Seals. Doors used at truck platforms or for railroad cars require a cushion seal at the sides and top on the outside of the opening to reduce the intake of ambient air when closed. Heating is required to prevent ice formation at doors to low temperature rooms. Split plastic door curtains shall be used where cost effective.

c. Door Protection. Bumpers are required around openings of doors as protection for doors from materials handling equipment.

d. Use of Forklift Trucks. Turning radius of trucks must be considered when laying out aisles. Storage height may be up to 18 feet (5.9 m).

e. Monorail. Some storage areas may use a monorail for material handling. This will limit storage heights and space between storage items.

f. Security. Door alarms should be provided if unauthorized access is restricted.

g. Fire and Smoke Alarms. Proper fire alarms and smoke alarms are required as protection for occupants. Local codes may require smoke vents near the roof to release smoke to aid fire fighting crews.

h. Space Requirements. Storage configuration, space requirements, weight-handling, and storage heights are covered in NAVFAC DM-32.2.

5. ENERGY CONSERVATION CRITERIA.

a. Use hot gas from refrigeration for heating. Install a heat reclaim condenser for use in heating water or brine. (See Figure 5.)

(1) Heat brine for the underfloor heating system.

(2) Heat water for heating adjacent office spaces.

b. Control Infiltration to Space. Where material handling equipment must enter from outdoors, install vestibule areas to reduce infiltration.

c. Control Internal Loads in Space.

(1) Keep lights off unless building is occupied.

(2) Minimize the use of material handling equipment which is stored outside and used in the storage facility. Material handling equipment will add a load to the space each time it enters if it is above the storage temperature.

(3) Control the product temperature of new products brought into storage to be close to design temperature.

d. Select temperature of storage facilities to be no lower than necessary. Control temperatures at design to minimize operating time.

6. STRUCTURAL CRITERIA.

a. Items for Structural Coordination. Structural coordination must include consideration of all of the following. (For additional data, see NAVFAC DM-2 series.)

(1) Foundations. Verify the following factors:

(a) Dead weight of equipment and storage items in operating condition.

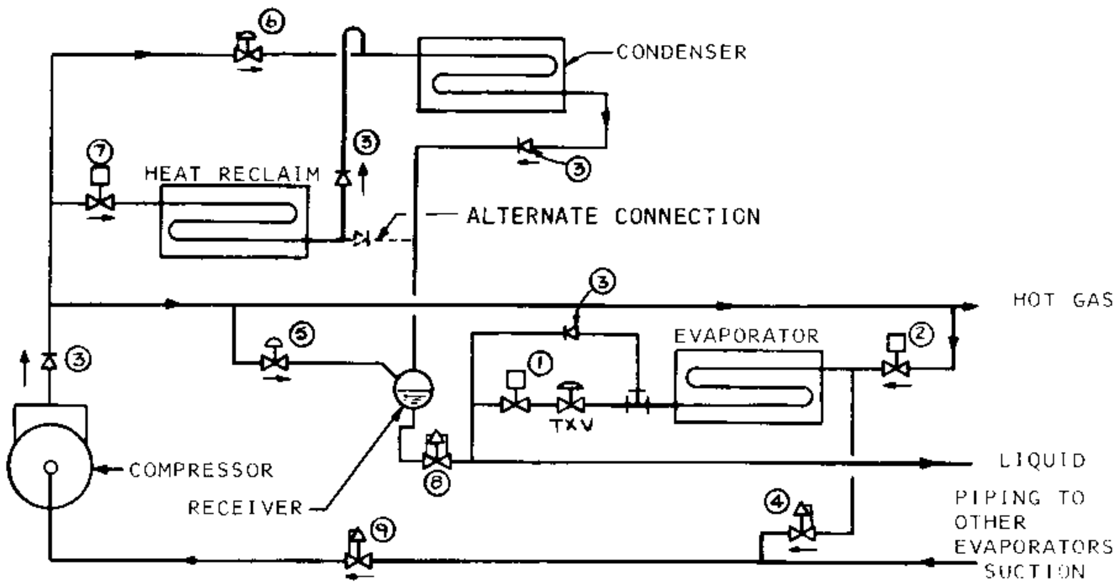
(b) Dynamic reactions and weight of reciprocating or vibrating equipment.

(c) Size and type of equipment base.

(d) Load distribution on base.

(e) Spacing between units of equipment shall be not less than 4 feet for maintenance. Accessibility to condenser must be provided and maintained for cleaning.

(2) Location. Insofar as possible, equipment shall be located to transfer the weight directly to the main structural members, girders, or columns, preferably columns.



NOTES

1 LIQUID SOLENOID VALVE.

2 HOT GAS DEFROST SOLENOID VALVE.

3 CHECK VALVE.

4 EVAPORATOR PRESSURE REGULATOR WITH ELECTRIC SHUT-OFF.

5 CONDENSER BYPASS REGULATOR PART OF CONDENSER PRESSURE CONTROL SYSTEM.

6 DISCHARGE PRESSURE REGULATOR PART OF CONDENSER PRESSURE CONTROL SYSTEM.

7 HOT GAS HEAT RECLAIM SOLENOID VALVE.

8 LIQUID DIFFERENTIAL PRESSURE REGULATOR.

9 CRANKCASE PRESSURE REGULATOR.

VALVE	COOLING MODE	HEAT RECLAIM MODE	DEFROST MODE
1	OPEN	OPEN	CLOSED
2	CLOSED	CLOSED	OPEN
7	CLOSED	OPEN	CLOSED

FIGURE 5
Typical Heat Reclaim System

(3) Pipe supports. Verify the following points:

(a) Weight of risers: include water column and unsupported pipe.

(b) Lateral thrust due to expansion joints.

(c) Dynamic forces at bends.

(d) Location and type of support.

(4) Opening through structural members. Size and location of such openings shall be based on the following factors:

(a) Overall outer size of pipe or duct inclusive of flange and insulation, if any.

(b) Standard length of pipe or duct between joints.

(c) Direction and extent of pitch for pipe.

(5) Access to equipment must be provided; stairs are preferred. Check size of the largest and weight of the heaviest pieces of equipment which have to be brought in for installation or removed for service. Verify facilities for transport and hoisting of such equipment. Determine if existing platforms, stairs, and elevators are adequate.

(6) Miscellaneous considerations. In determination of structural coordination, the following factors must be considered:

(a) Wind forces. The design of outdoor equipment (cooling tower, stack, and their supports) shall be based on the maximum wind velocities prevalent at the site.

(b) Seismic considerations. If the site is in seismic Zone 2, 3, or 4 and is subject to earthquakes, the design of equipment and piping shall be in accordance with NAVFAC P-355 and piping shall be tightly secured to structures.

7. AIR CURTAIN FAN SYSTEMS.

a. Doorway design, application, and operation is one of the most critical parts of the cold storage facility in the conservation of energy. They are as important as the walls, roofs, and refrigeration machinery in regard to energy. Doors that are left open too frequently and for too long a time may be properly installed, but do not contribute towards conserving energy. From observations in existing facilities it is concluded that unnecessary open door time is far greater than management realizes or is necessary.

b. An air curtain can be 70 to 80 percent effective in reducing heat inflow to a freezer and in reducing moisture infiltration. The moisture infiltration is easily noticed since eventually it will result in frost formation, usually on the cooling coils. An air curtain cannot protect against moisture infiltration due to vapor pressure difference between inside and the outside, however, gains due to this phenomena are insignificant; less than one percent of the total heat transfer.

c. An air curtain efficiency is realistically in order of 30 to 40 percent.

d. The ideal application requires that the room be tightly sealed with only one opening or doorway. The air stream should be of the proper velocity, directed at a specific angle and of a certain thickness and the entire system should reach equilibrium conditions.

e. An air curtain installed at a freezer with only one door is far more effective than on a freezer with two active doorways.

f. Recent research has shown that air curtains can be effective at doorways to retard the flow of heat and moisture between the outside and an enclosed space where a solid door is opened either continuously or intermittently.

(1) Air curtains should be mounted on the warm side of the doorway to avoid frosting conditions within the air curtain system of ducts and fans and to prevent cold air from blowing on personnel.

(2) The air curtain outlet nozzle should cover at least the entire width of the doorway.

(3) The air curtain nozzle should, for the best effect, be angled out from the wall between 15 to 30 deg. (0.3 to 0.5 radians), a little more than 15 deg. (0.3 radians) seems to be most practical.

(4) The velocity of the air leaving the air curtain is important and should be accurately determined. High velocities can increase the rate of heat transfer and low velocities can cause the air curtain to break contact with the floor and bend backwards toward the cold side.

(5) The thickness or depth of the air stream has a very minor effect on the effectiveness of an air curtain.

(6) The study assumes that the air curtain works on a doorway to an absolutely sealed room thus after operating for a short period builds up a slight over pressure in the room to balance itself.

g. One way to increase the effectiveness of an air curtain is to baffle it at the sides. Such baffles can normally be added relatively inexpensively as part of the necessary protective barrier system.

h. The use of an air curtain must be made recognizing the problems involved including fogging, ice and frost formation and the effect of external wind forces. High curtain velocities up to 6,000 FPM (68 MPH) (30 metres per second) have been proposed to resist external wind forces. The high curtain velocities must consider the effects of much higher noise levels and probable intermixing of curtain air and room air. The application of the air curtain to interior locations can utilize much lower air velocities.

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NAVFACENGCOM Design Manuals and P-Publications.

Government agencies may obtain Design Manuals and P-Publications from the U.S. Naval Publications and Forms Center, 5801 Tabor Ave., Philadelphia, PA 19120. TWX 710-670-1685, AUTOVON: 442-3321. The stock number is necessary for ordering these documents and should be requested from the NAVFACENGCOM Division in your area.

Non-Government organizations may obtain Design Manuals and P-Publications from the Superintendent of Documents, U.S. Government Printing Office, Washington, DC 20402.

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DM-2 Series	Structural Engineering
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DM-5 Series	Civil Engineering
DM-32 Series	Supply Facilities

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APPENDIX A

LIFE CYCLE CHARTS

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CHART 1

EXAMPLE OF LIFE CYCLE COSTING AT MIAMI, FLORIDA TO DETERMINE OPTIMUM ROOF INSULATION THICKNESS - COOLER at 35° F

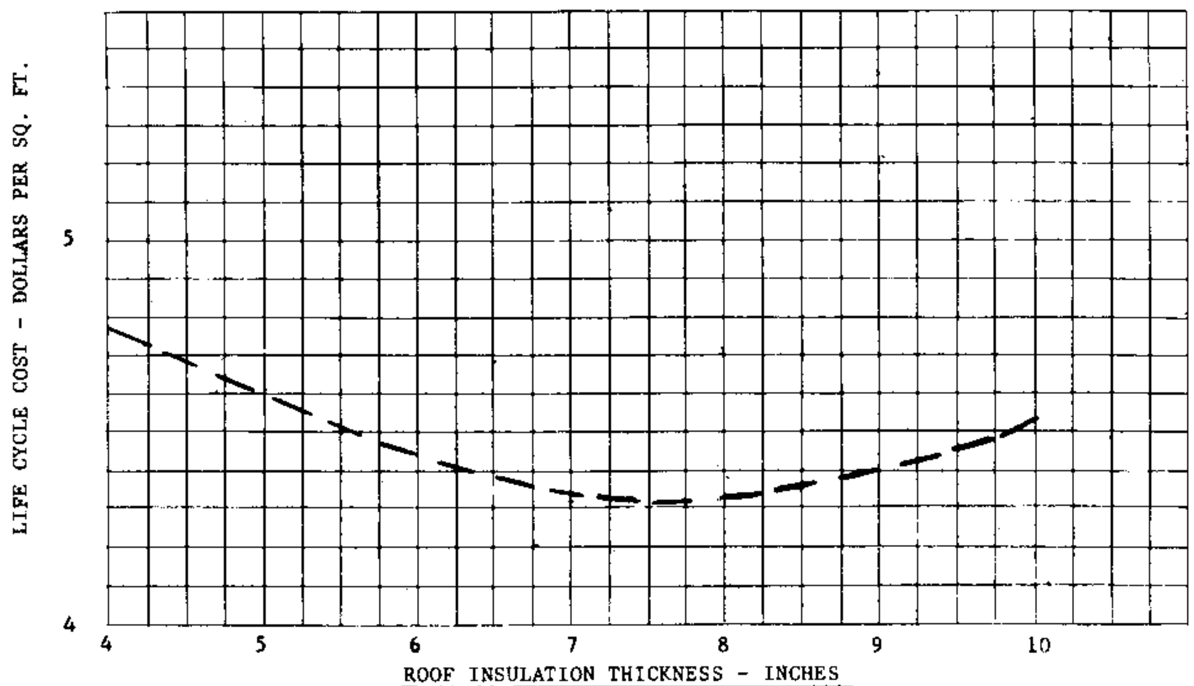
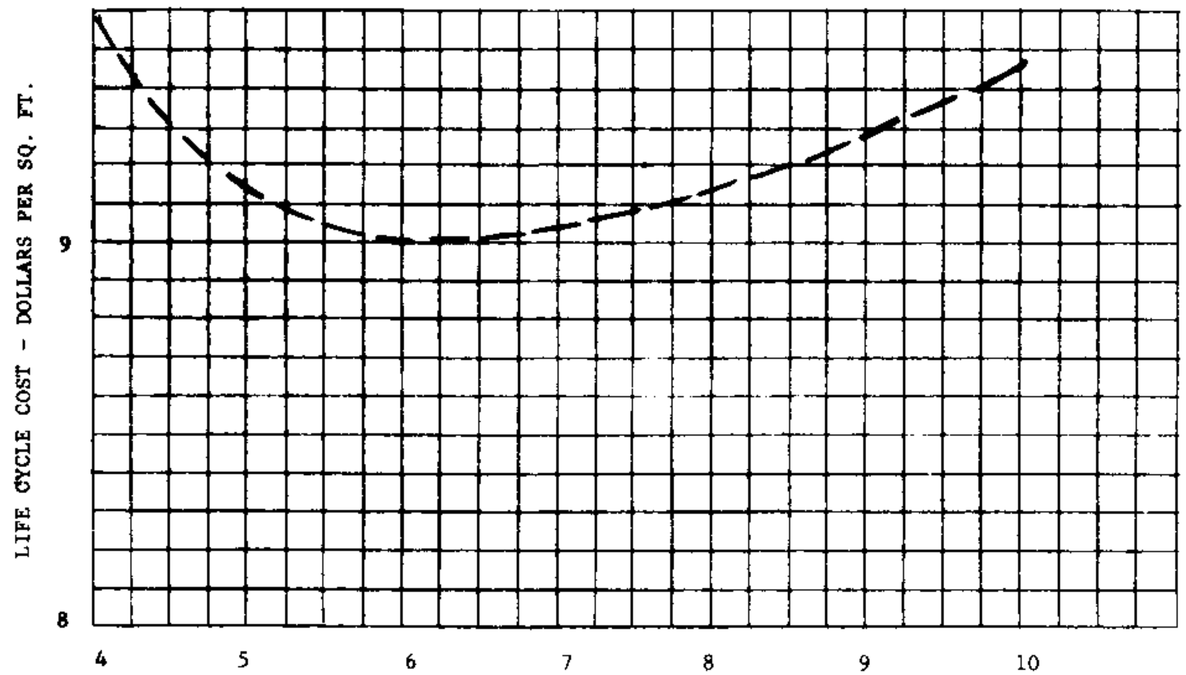


CHART IS BASED ON COMPUTER ENERGY ANALYSIS TO DETERMINF EFFECT OF INSULATION THICKNESS ON LIFE CYCLE COSTING ENERGY SAVING FOR POLYSTYRENE INSULATION SYSTEM WITH 25 - YEAR LIFE SPAN.

JAN 1979

CHART 2

EXAMPLE OF LIFE CYCLE COSTING AT MIAMI, FLORIDA TO DETERMINE OPTIMUM WALL INSULATION THICKNESS - COOLER AT 35° F



WALL INSULATION THICKNESS - INCHES

CHART IS BASED ON COMPUTER ENERGY ANALYSIS TO DETERMINE EFFECT OF INSULATION THICKNESS ON LIFE CYCLE COSTING & ENERGY SAVING FOR POLYSTYRENE INSULATION SYSTEM WITH 25 - YEAR LIFE SPAN.

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CHART 3

EXAMPLE OF LIFE CYCLE COSTING AT MIAMI, FLORIDA TO
DETERMINE OPTIMUM ROOF INSULATION THICKNESS - FREEZER AT -10° F

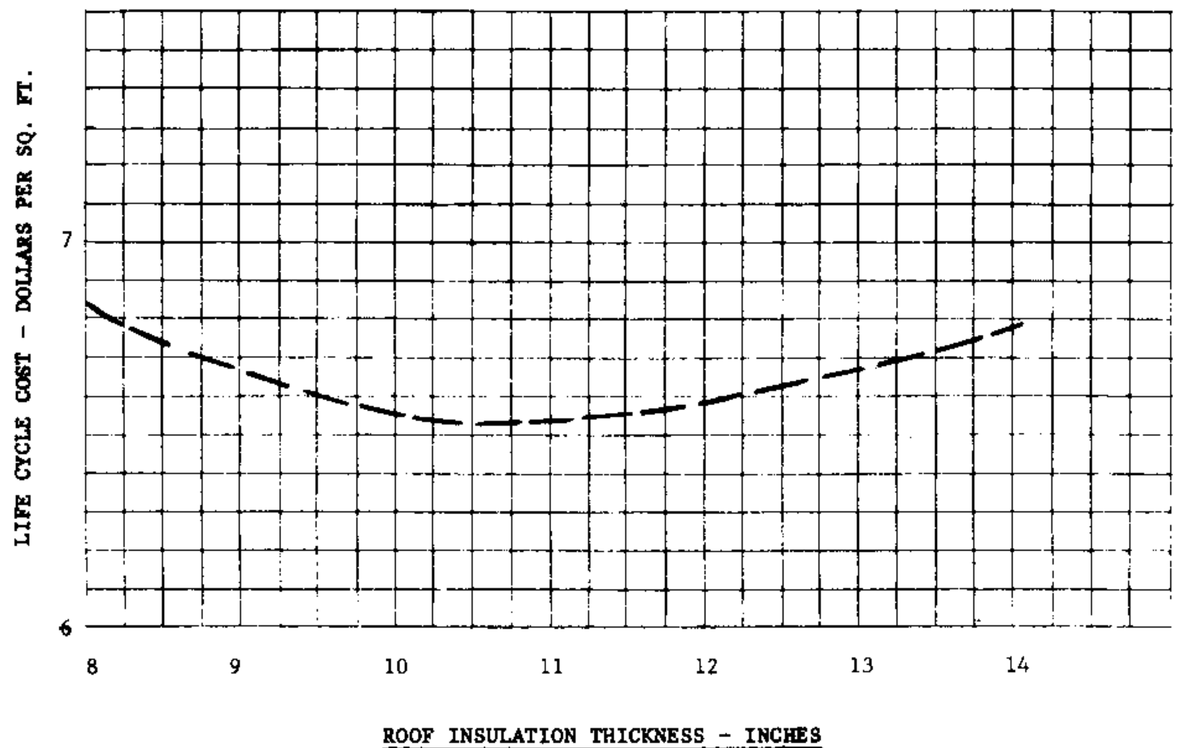
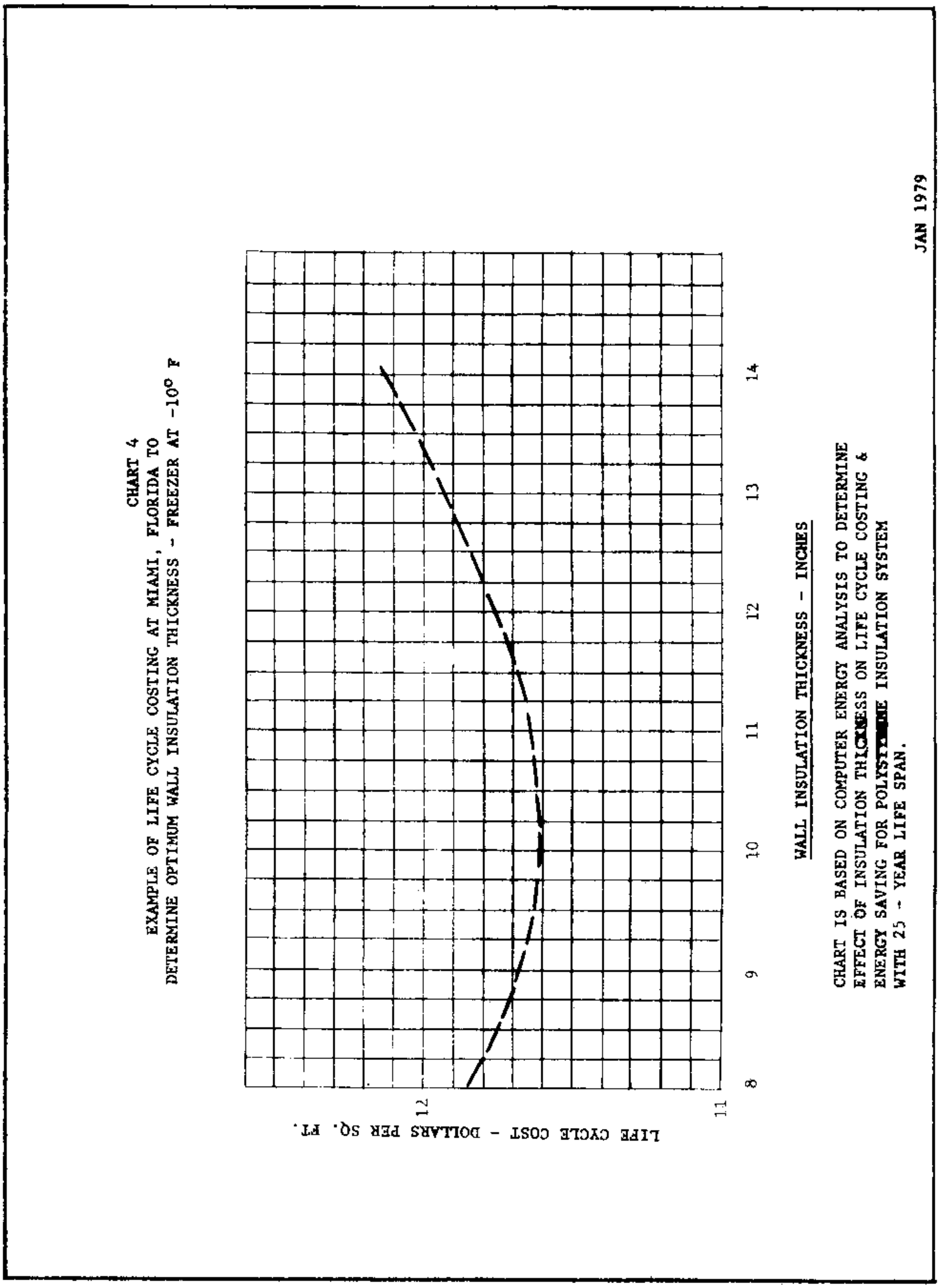


CHART IS BASED ON COMPUTER ENERGY ANALYSIS TO DETERMINE
EFFECT OF INSULATION THICKNESS ON LIFE CYCLE COSTING &
ENERGY SAVING FOR POLYSTYRENE INSULATION SYSTEM
WITH 25 - YEAR LIFE SPAN.

JAN 1979



Determine Optimum Wall Insulation Thickness -
Cooler at -10 deg. F]

CHART 5

EXAMPLE OF LIFE CYCLE COSTING AT BOSTON, MASS. TO DETERMINE OPTIMUM ROOF INSULATION THICKNESS - COOLER AT 35° F

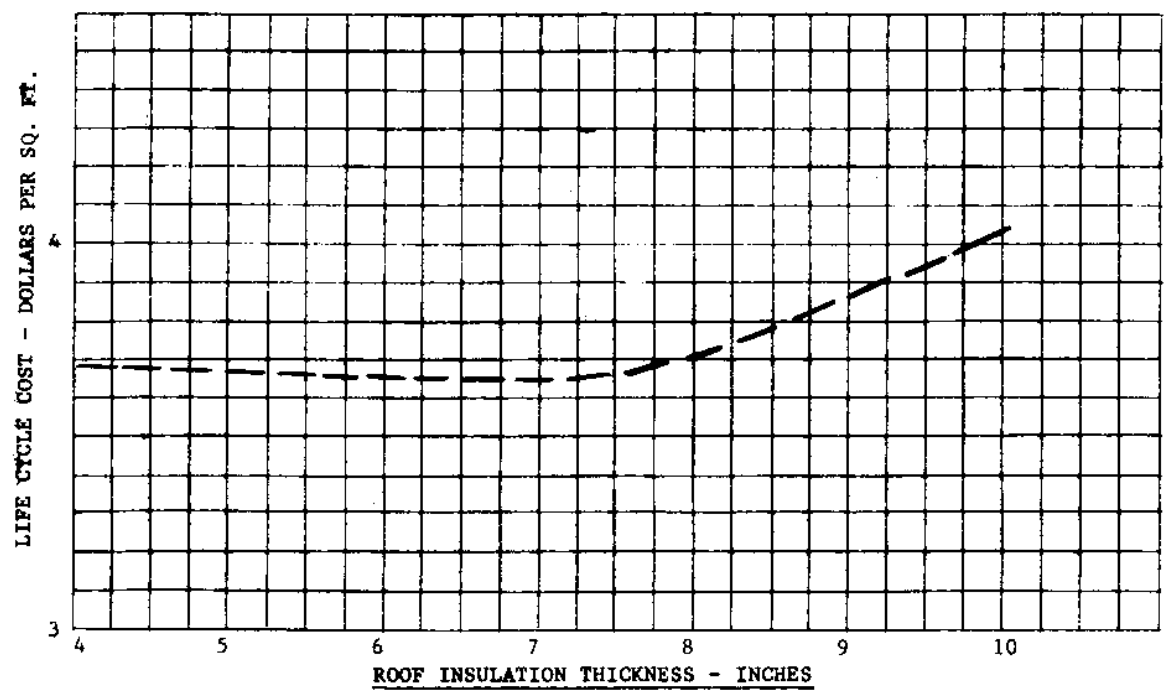


CHART IS BASED ON COMPUTER ENERGY ANALYSIS TO DETERMINE EFFECT OF INSULATION THICKNESS ON LIFE CYCLE COSTING & ENERGY SAVING FOR POLYSTYRENE INSULATION SYSTEM WITH 25 - YEAR LIFE SPAN.

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CHART 6

EXAMPLE OF LIFE CYCLE COSTING AT BOSTON, MASS. TO
DETERMINE OPTIMUM WALL INSULATION THICKNESS - COOLER AT 35° F

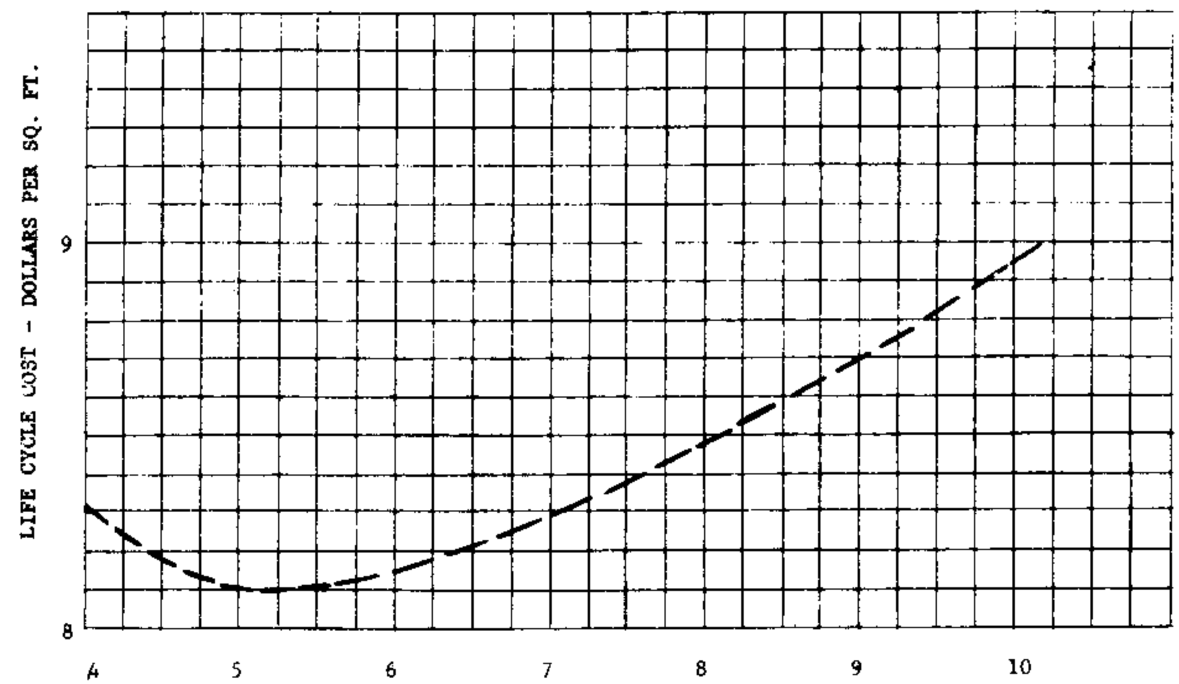

CHART IS BASED ON COMPUTER ENERGY ANALYSIS TO DETERMINE
EFFECT OF INSULATION THICKNESS ON LIFE CYCLE COSTING &
ENERGY SAVING FOR POLYSTYRENE INSULATION SYSTEM WITH
25 - YEAR LIFE SPAN.

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CHART 7

EXAMPLE OF LIFE CYCLE COSTING AT BOSTON, MASS. TO DETERMINE OPTIMUM ROOF INSULATION - FREEZER AT -10° F

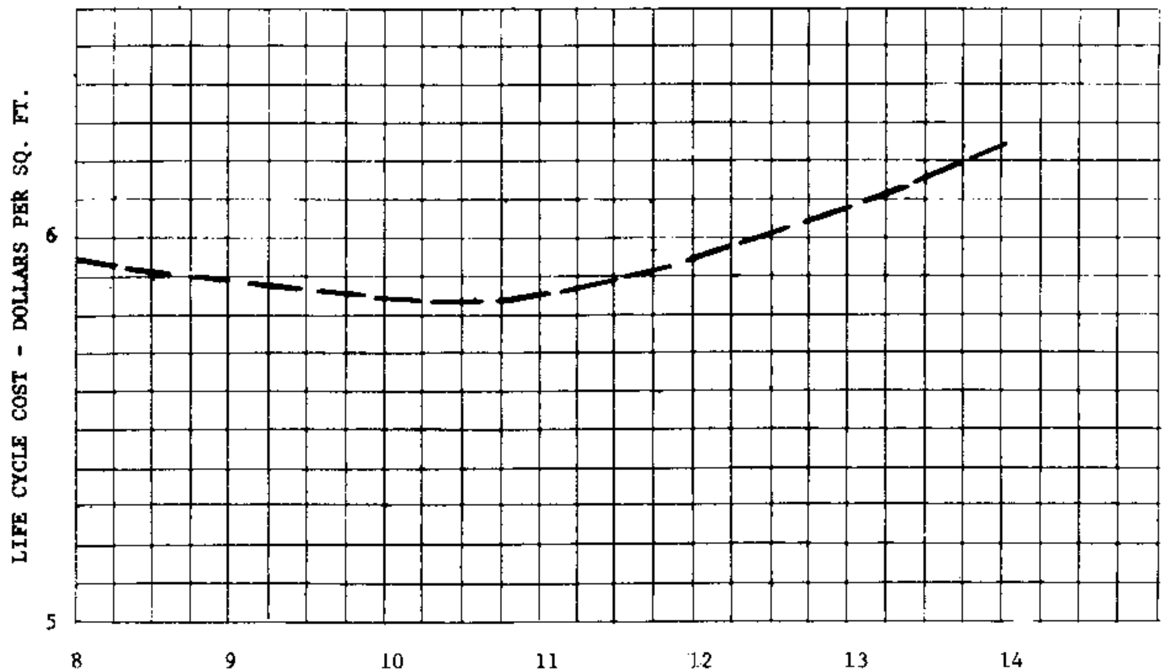

CHART IS BASED ON COMPUTER ENERGY ANALYSIS TO DETERMINE EFFECT OF INSULATION THICKNESS ON LIFE CYCLE COSTING & ENERGY SAVING FOR POLYSTYRENE INSULATION SYSTEM WITH 25 YEAR LIFE SPAN.

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CHART 8

EXAMPLE OF LIFE CYCLE COSTING AT BOSTON, MASS. TO DETERMINE OPTIMUM WALL INSULATION - FREEZER AT -10° F

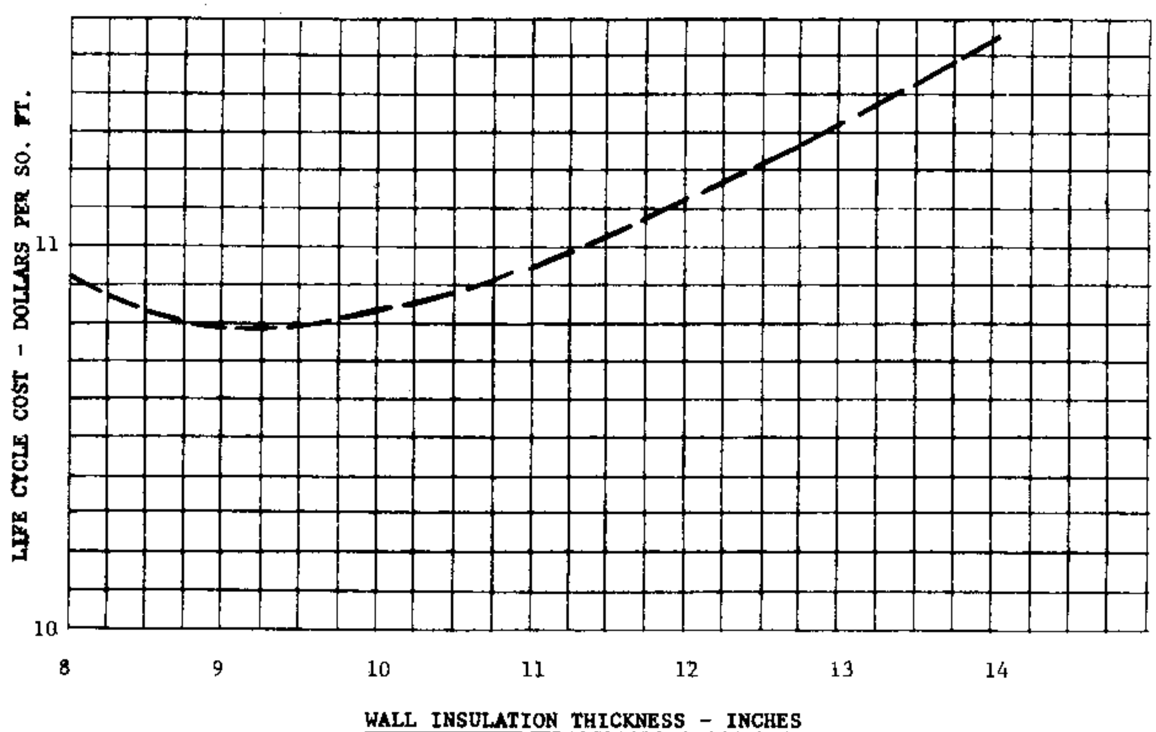

CHART IS BASED ON COMPUTER ENERGY ANALYSIS TO DETERMINE EFFECT OF INSULATION THICKNESS ON LIFE CYCLE COSTING & ENERGY SAVING FOR POLYSTYRENE INSULATION SYSTEM WITH 25 - YEAR LIFE SPAN.

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