REFRIGERATION AND COLD STORAGE PLANT DESIGN FOR TURKEY (CAPACITY 30 TONS CLEAR ICE)

by

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A THESIS

submitted to

OREGON STATE COLLEGE

in partial fulfillment of
the requirements for the
degree of

MASTER OF SCIENCE

June 1948

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DESIGN

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DESIGN

This is a combination ice and cold storage plant to be designed with the following specifications:

Daily capacity of ice plant is to be thirty tons of clear ice. Blocks are to be frozen in 11" x 22" x 48" high ice cans, each having capacity of 315 lbs. of ice.

Shut down -- 4 months

Full capacity -- 4 months

1/2 capacity -- 4 months

Ice storage room is to hold 200 tons of ice with sufficient working space. Temperature is to be maintained at 26° F for 12 months a year.

Fish storage room should be at temperature of 8° F to be maintained for 12 months per year.

Fish freezing room. A fish freezer daily capacity of 500 lbs. is to be operated 12 months per year. Room temperatures are to be maintained at -5° F.

Compressor room is to be located near the ice tank and all compressors are to be driven by electric motors.

Two ice tanks of $\frac{1}{4}$" steel will be used. Each tank will have one accumulator and one brine pump. The tanks are to be insulated with 12" of granulated cork on the sides and 5" of corkboard on the bottom.

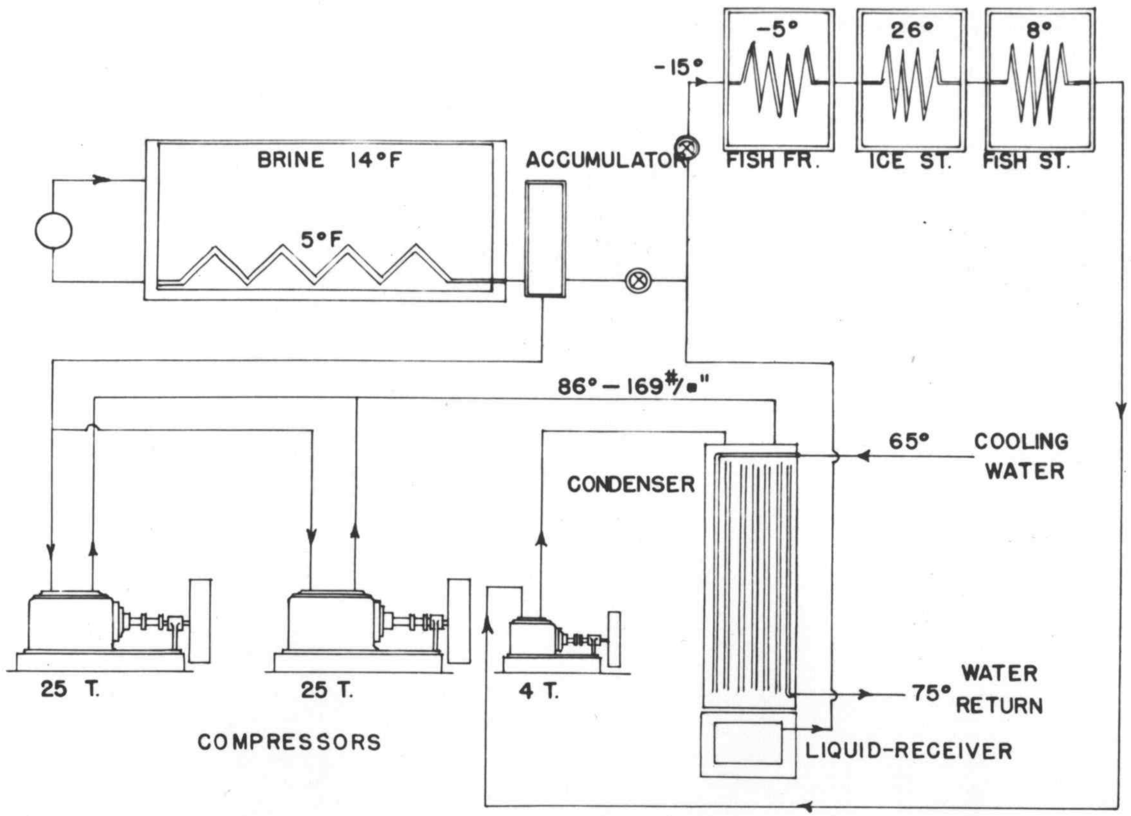
Buildings

All walls are to be 12" brick.

Insulation

Corkboard is to be used throughout, including ice tank insulation.

AMMONIA FLOW DIAGRAM



INSULATION

INSULATION

Insulation is to provide a suitable wall to prevent the penetration of an unreasonable amount of heat. It is not possible to prevent the loss of refrigeration no matter how perfect an insulation is used, and the commercial aspect of the problem must be considered.

For every case there is a maximum amount of insulation that can be practicably installed. When the fixed charges on the additional investment exceed savings, it is unprofitable to add additional insulation.

The selection of a suitable insulating material and its correct application is one of the most important problems in the design of a refrigerating plant.

The most common insulation is cork, and its largest use is for thermal insulation for low temperatures, both in the form of corkboard and cork pipe covering.

The most serious obstacle to the efficiency of low temperature insulation is penetration of water vapor. The thermal conductivity of any material increases with the increase in moisture content.

Excellent vapor stops in the form of asphalt saturated and coated papers are now available. Therefore, in insulation of walls it is used. Asphalt coated papers are easier and more practical to use than any other vapor stops. The vapor stop or vapor barrier is placed on the warm side of the insulation since the flow of vapor is from the warm to the cold side.

Thermal Conductivity Constant K for corkboard and other materials is given in the accompanying table.

Properties of Cork and Other Low-Temperature Insulation

(K is expressed in B.t.u. per sq. ft. per deg. F. per hr. per in. at 65 deg. F., mean temperature)

Material or Form	Density lb.per cu.ft.	Conductivity K	Sp. Heat	Fire Hazard
Corkboard....	$7\frac{1}{2}$ to 8	0.28	0.44	Slow burning
"Rock Cork".	14 3/4 to 17	0.30 to 0.32	0.27	" "
Hair Felt...	8 to 21	0.31 to 0.33	0.32	" "
Wool Felt...	17 to 27	0.27 to 0.29	--	" "
Fibre Board	14.5 to 17	0.33 to 0.36	0.32	" "
Reflective Insulation	Very light	low	--	Fireproof, except when mounted on paper
Glass & Mineral Wool..	1.5 to 7	0.25 to 0.28	0.20	Fireproof
Redwood Bark (Packed)...	2.5 to 6.5	0.23 to 0.30	--	Combustible
Cellular Glass	9.7 to 11.4	0.48	0.16	Fireproof
Sponge Rubber	4.5 to 5.7	0.21 to 0.25	0.30	Combustible
"Unicel"....	5.5 to 7	0.27 to 0.29	--	Slow burning

Compressibility of corkboard insulation has an average compression of 7 to 10 percent under a sustained load of 10 lbs. per sq. in. for approximately 200 hrs. When the load is removed, the greater part of the original thickness of the cork is restored. The rate of heat flow through insulation increases in almost direct proportion to the decrease in thickness.

Economic Thickness of Insulation

The thickness of insulation to be chosen is that which will give the greatest return on the investment. The factors influencing a choice of the economic thickness include

1. The cost of a ton of refrigeration delivered, per 24 hours.

2. The annual investment cost of insulation applied,

3. The annual investment cost of the refrigerating equipment not included in item one, and particularly the room piping, etc.,

4. The value of the space occupied by the insulation, per year, considering the case of a wall, ceiling or floor.

*These losses may be expressed in the following manner:

Let x = the thickness of the insulation, in inches

C = the coefficient of conduction of the insulation, per 1 in. thick, per hour

F = the yearly load factor

B = the cost in dollars for the insulation applied, per 1 in. thick per sq. ft. of surface.

A = the cost in dollars per ton of refrigeration

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*Macintire, H. J., Refrigeration Engineering, Rev. Ed., John Wiley & Sons.

G = the cost in dollars per ton of refrigeration of the machinery, not included in A

I = the interest rate as a percent on the insulation investment

R = the repair cost per year as a percent of the insulation first cost

Y = the life of the insulation, in years

I', R', Y' = similar values applied to machinery, in G

t_a = the temperature of the outside air, degrees F as an average for the period of operations

t_m = the maximum temperature of the outside air in degrees F

t = the temperature of the cold storage room in degrees F

t_p = the temperature of the refrigeration in the piping in degrees F

S = the value per year of 1 cu. ft. of space in the cold storage room

U = the coefficient of heat transfer of the wall per sq. ft. per hour for the materials of construction other than the insulation as given by the usual formula:

$$\frac{1}{U} = \frac{1}{k_1} + \frac{T_1}{C_1} + \frac{T_2}{C_2}$$

1. The cost per year of the heat leakage through the insulation $= \dfrac{t_a - t}{\frac{1}{U} + \frac{X}{C}} \times \dfrac{24 \times 365 \times F \times A}{288000}$

2. The cost of the insulation per year, per square foot per 1 in. thick $= B \times X \left[\dfrac{I}{100} - \dfrac{R}{100} - \dfrac{1}{Y}\right]$

3. The cost per year of the investment required to offset the heat leakage through the insulation

$$= \frac{t_m - t}{\frac{1}{U} + \frac{X}{C}} \times \frac{24}{288000} \times G \left[\frac{I'}{100} - \frac{R'}{100} - \frac{1}{Y'}\right]$$

4. The cost of the space occupied by the insulation per year $= \dfrac{SX}{12}$

As a rule the cost of insulation, applied, can be expressed by the formula

$$B = \frac{C'}{X} + B'$$

where

C' is the cost of finish, plaster, nails, labor and overhead per square foot, and

B' is the cost of the insulation delivered to the job.

Also if P is the cost in dollars per square foot of refrigerating piping installed in the cold room as the equipment represented by G, Then $G = \frac{12000P}{U' \times (t-t_p)}$

where U' = the coefficient of heat transfer for the piping per hour .

Then if Z = total cost per year,

$$Z = \frac{t_a-t}{\frac{1}{U}+\frac{X}{C}} \times \frac{365FA}{12000} + \frac{C'}{X} + B' \times \left[\frac{I}{100} - \frac{R}{100} - \frac{1}{Y}\right]$$

$$+ \frac{SX}{12} + \frac{t_m-t}{\frac{1}{U}+\frac{X}{C}} \times \frac{1}{12000} \times \frac{12000 \times P}{U' \times (t-t_p)} \times \left[\frac{I'}{100} - \frac{R'}{100} - \frac{1}{Y'}\right]$$

For a minimum, $\frac{d_z}{dx} = 0$ so, by differentiating and putting this equal to zero and solving for X, the result becomes

$$X = 1.74 \quad A(t_a-t)F - \frac{0.327P}{k(t-t_p)} \quad I' - R' - \frac{100}{Y'}(t_m-t)xC - \frac{C}{U}$$

Using the following approximate values,

$I = 6\%$

$I' = 6\%$

$R = 3$

$R' = 3$

$Y = 15$

$Y' = 8$

$U' = 1.5$ B.t.u./hr.

$C = 0.35$ B.t.u./hr.

$t_{av} = 95$ deg. F

$(t-t_p) = 12$ deg. F

$F = 1.0$

$A = 2.00$

$S = 0.40$ per cu. ft.

$t_{max} - t = 95$ deg. F.

$P = 4.00$ the cost of piping per square foot installed plus all accessories

$$B = \frac{0.72 - 0.065X}{X}$$

$$G = \frac{288000P}{24 \times k \times (t-t_p)}$$

we get 7" thickness of insulation.

DETERMINATION OF LOADS

DETERMINATION OF LOADS

In determining the loads, the following factors are determined:

a. Losses through walls Q_{wl}

 1. Side walls Q_{wl_1}

 2. Ceiling and floor Q_{wl_2}

b. Losses due to air changes $Q_{inf.}$

c. Miscellaneous losses

 1. Loss due to men working Q_{men}

 2. Loss due to light bulbs Q_{light}

d. Cooling Load.

ICE STORAGE ROOM

a.1. Storage temperature = 26° F

Outside temperature = 95° F

Temperature difference = 69° F

From Table I, Refrigeration Data Book, P. 169, approximate wall heat losses in commercial fixture

For 69° F temperature difference and 7" cork insulation.

Size of the room - 36' x 24' x 25'

Total area of the side walls,

$$A = 2 \times 25(36 + 24)$$

$$= 3000 \text{ ft}^2$$

Wall losses,

$$Q_{wl_1} = 3000 \text{ ft}^2 \times 71 \frac{\text{B.t.u.}}{\text{ft}^2\text{- 24 hrs.}}$$

$$= 213000 \frac{\text{B.t.u.}}{\text{24 hrs.}}$$

2. Ceiling and floor.

$$A = 24' \times 36' \times 2$$

$$= 1730 \text{ ft.}^2$$

$$Q_{wl_2} = 1730 \text{ ft.}^2 \times 71 \frac{\text{B.t.u.}}{\text{ft}^2\text{-24 hrs.}}$$

$$= 127000 \frac{\text{B.t.u.}}{\text{24 hrs.}}$$

$$Q_{wl} = Q_{wl_1} + Q_{wl_2}$$

$$Q_{wl} = 213000 + 127000$$

$$= 340000 \frac{\text{B.t.u.}}{\text{24 hrs.}}$$

b. $Q_{inf.}$

$$\text{Volume} = 36' \times 24' \times 25'$$

$$= 20600 \text{ Cu. ft.}$$

Then from Table 2* for V = 20600 cu. ft. average air changes per 24 hr. for storage rooms due to door opening and infiltration.

$$fa = 2.6 \frac{\text{B.t.u.}}{\text{cu. ft. x 24 hrs.}}$$

From Table 3**, RH = 60% and $t_{outside} = 95^{o}$

Heat removed in cooling to storage conditions or cooling factor from Table 3**.

$$fc = 3 \frac{\text{B.t.u.}}{\text{cu. ft. x 24 hrs.}}$$

$$Q_{inf.} = V \times fa \times fc$$

$$= 20600 \text{ ft}^3 \times (2.6 \times 3) \frac{\text{B.t.u.}}{\text{ft}^3 \times 24 \text{ hrs.}}$$

$$= 161000 \frac{\text{B.t.u.}}{\text{24 hrs.}}$$

c.1. Q_{men}

Assuming that three men working 8 hrs per day,

Heat load for men working = $\frac{\text{B.t.u.*}}{\text{hr. men}}$

$$Q_{men} = 3 \text{ man} \times \frac{750 \text{ B.t.u.}}{\text{hr-man}} \text{ 8 hrs.}$$

$$= 18000 \frac{\text{B.t.u.}}{\text{day}}$$

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* Refrigeration Data Book, p. 170.

** Refrigeration Data Book, p. 171.

2. Q_{light}

Assuming 6 bulbs each 40 watts,

$$Q_{light} = 6 \text{ bulbs} \times 40 \frac{\text{watt}}{\text{bulb}} \times 3.42 \frac{\text{B.t.u.}}{\text{hr-watt}} \times 24\text{hr}$$

$$= 19700 \frac{\text{B.t.u.}}{24 \text{ hr}}$$

$$Q_{Total} = Q_{wl} + Q_{inf} + Q_{men} + Q_{light}$$

Q_{wl} = 340000 $\frac{\text{B.t.u.}}{24 \text{ hr.}}$

Q_{inf} = 161000 "

Q_{men} = 18000 "

Q_{light} = 19700 "

Q_{total} = 538700 $\frac{\text{B.t.u.}}{24 \text{ hr.}}$

FISH FREEZING ROOM

Freezing temperature = -5° F

Outside temperature = 40° F

Thickness = 7"

Room = 15' x 12' x 10'

Temperature difference = 45° F

Wall heat losses = 46 $\frac{\text{B.t.u.}}{\text{ft}^2 \times 24 \text{ hr.}}$

A wall = $2 \times 10(15 + 12)$

= 540 ft^2

a. Wall losses:

1. $Q_{wl_1} = 540\ ft^2 \times 46\ \frac{B.t.u.}{ft^2 \times 24\ hrs.}$

$= 25000\ \frac{B.t.u.}{24\ hrs.}$

2. $Q_{wl_2} = 360\ ft^2 \times \frac{46\ B.t.u.}{ft.^2 \times 24\ hrs.}$

$= 16500\ \frac{B.t.u.}{24\ hrs.}$

$Q_{wl} = Q_{wl_1} + Q_{wl_2}$

$W_{wl} = 41500\ \frac{B.t.u.}{24\ hrs.}$

b. Q_{inf}

Volume $= 15' \times 12' \times 10' = 1800\ ft.^3$

$fa = 9.3\ \frac{B.t.u.}{24\ hrs.}$

$fc = 1.23$ "

$Q_{inf} = 1800 \times [9.3 \times 1.23]$

$= 20600$ B.t.u.

c.1. Q_{men}

one man, 4 hrs. a day

$Q_{men} = \frac{4\ hrs.}{24\ hr} \times 750\ \frac{B.t.u.}{hr\ man} \times 1$ man

$Q_{men} = 3000\ \frac{B.t.u.}{24\ hrs.}$

2. Q_{light}

Assuming 2 40-watt bulbs,

$$Q_{light} = 2 \text{ bulbs} \times 40 \frac{\text{Watt}}{\text{bulbs}} \times 3.42 \frac{\text{B.t.u.}}{\text{hr} \times \text{watt}}$$

$$= 6400 \frac{\text{B.t.u.}}{24 \text{ hrs}} \times 24 \text{ hrs.}$$

d. Cooling Load = Q

$Q_F = q_a + q_b + q$

q_a = Load to cool to freezing point

q_b = Load to freeze

q_c = To cool storage temperature

$q_a = wcp_1 \Delta t$

$w = 500 \frac{\text{lb}}{\text{day}}$

$cp_1 = 0.82^* \frac{\text{B.t.u.}}{\text{lb } ^{\circ}\text{F}}$ (before freeze)

$\Delta t = t - t_f$

t = outside temperature

t_f = freezing temperature

$$q_a = 500 \frac{\text{lb}}{24 \text{ hr}} \times 0.82 \frac{\text{B.t.u.}}{\text{lb } ^{\circ}\text{F}} \times 32\ ^{\circ}\text{F}$$

$$= 13{,}100 \frac{\text{B.t.u.}}{24 \text{ hrs.}}$$

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*Refrigeration Data Book, Table 4, p. 171

$$q_b = wL$$

L = Latent heat of fusion

$$q_b = 500\ \frac{lb}{24\ hr} \times 105\ \frac{B.t.u.}{lb}$$

$$= 52500\ \frac{B.t.u.}{24\ hr}$$

$$q_c = wC_{p2}\Delta t$$

C_{p2} = Specific heat after freezing

$$\Delta t = t_f - t_s$$

t_s = Final temperature after freezing

$$q_c = 500\ lbs \times 0.41\ \frac{B.t.u.}{lb\ ^oF} \times 33^o\ F$$

$$= 6{,}750\ \frac{B.t.u.}{24\ hr}$$

q_a = 13,100 $\frac{B.t.u.}{24\ hr}$

q_b = 52,500 "

q_c = 6,750 "

q_F = 72,350 $\frac{B.t.u.}{24\ hr}$

Q_{wall} - 41,500 $\frac{B.t.u.}{24\ hr}$

Q_{inf} - 20,600 "

Q_{men} - 3,000 "

Q_{light} - 6,400 "

Q_{freeze} - 72,350 "

Q_{total} - 123,850 $\frac{B.t.u.}{24\ hr}$

FISH STORAGE ROOM

Storage temperature = 8° F

Outside temperature = 95° F

Room size = 31' x 18' x 15'

Cork thickness = 7"

a.1. Wall heat loss $= 91 \dfrac{\text{B.t.u.}}{\text{ft}^2 \times 24 \text{ hr}}$

Side walls

$$A = 2 \times 15\ (31 + 18)$$

$$= 1470\ \text{ft}^2$$

$$W_{l_1} = 1470\ \text{ft}^2 \times 91 \dfrac{\text{B.t.u.}}{\text{ft}^2\ 24 \text{ hr}}$$

$$= 133500 \dfrac{\text{B.t.u.}}{24 \text{ hr}}$$

2. Ceiling and floor

$$A = 2 \times 31 \times 18\ \text{ft}^2$$

$$= 1115\ \text{ft}^2$$

$$Q_{wl_2} = 1115\ \text{ft}^2 \times 91 \dfrac{\text{B.t.u.}}{\text{ft}^2 \times 24 \text{ hr}}$$

$$= 108500 \dfrac{\text{B.t.u.}}{24 \text{ hr}}$$

b. Q_{inf}

$$V = 31' \times 18' \times 15'$$

$$= 8400\ \text{ft}^3$$

$$f_a = 4 \dfrac{\text{B.t.u.}}{24 \text{ hr}}$$

$$f_c = 3.6$$

$$Q_{inf} = V \times f_a \times f_c$$

$$= 8400 \times [4 \times 3.6]$$

$$= 120000 \frac{\text{B.t.u.}}{\text{24 hr}}$$

c.l. $Q_{men} = 2 \text{ man} \times 750 \frac{\text{B.t.u.}}{\text{hr man}} \times 8 \text{ hr}$

$$= 12000 \frac{\text{B.t.u.}}{\text{24 hr}}$$

2. Q_{light}

Assuming 4 40-watt bulbs,

$$Q_{light} = 4 \times 40 \text{ watt} \times 3.42 \frac{\text{B.t.u.}}{\text{hr-watt}} \times 24 \text{ hr.}$$

$$= 13100 \frac{\text{B.t.u.}}{\text{24 hr.}}$$

$$Q_{wl} = 242000$$

$$Q_{inf} = 120000$$

$$Q_{men} = 12000$$

$$Q_{light} = \underline{13100}$$

$$Q_{total} = 387100 \frac{\text{B.t.u.}}{\text{24 hr.}}$$

ICE MAKING

Heat removed per pound of ice:

To cool the water from 70°F to 32° F	38 $\frac{\text{B.t.u.}}{\text{lb}}$
To freeze the water at 32° F	144 $\frac{\text{B.t.u.}}{\text{lb.}}$
To cool the ice from 32° F to 14° F	9 $\frac{\text{B.t.u.}}{\text{lb.}}$
	191 $\frac{\text{B.t.u.}}{\text{lb.}}$
Add 15 percent for non-computable losses	29 $\frac{\text{b.t.u.}}{\text{lb.}}$
	220

$$\text{Ratio } \frac{220}{144} = 1.6$$

$$Q_{total} = 30 \text{ tons } 1.6 \frac{\text{ton of refrigeration}}{\text{ton of ice}}$$

$$= 48 \text{ tons}$$

$$= 48 \text{ tons} \times 288000 \frac{\text{B.t.u.}}{\text{ton - 24 hr.}}$$

$$= 13.824.000 \frac{\text{B.t.u.}}{\text{24 hr.}}$$

THE COMPRESSORS

THE COMPRESSORS

SELECTION OF COMPRESSORS

The capacity calculated for each room is as follows:

Rooms	Capacity in Tons	Temperature
Ice storage	1.525	15° F
Fish freezing	0.43	-15° F
Fish storage	1.35	-15° F
Ice making	48.00	5° F

We can use one compressor for all loads, but when the ice making is stopped, the capacity will be too large for the evaporators of the rooms. Second alternative will be to use two compressors, one for ice making operating at its condenser and evaporator temperature, and another compressor for the rooms working between lowest evaporator temperature and condenser temperature.

One method is to operate each evaporator at its desired pressure, and throttle the leaving vapor to a pressure corresponding to that of the evaporator in which the lowest is to be maintained.

When the vapor from the ice storage room is throttled to a compressor suction pressure of 20.88 lbs. per sq. in., corresponding to -15^{0} F, as referred to in the figure Page 23, the cycle for operation with a 20.88 lb. per sq. in. suction main is shown in flow diagram Page 23 on P-h Chart. The state points of vapor from the evaporators at pressures greater than 20.88 lb. per sq. in. is in the superheat region after it has been expanded to the lower suction pressure. The state point of vapor in suction line at entrance to the compressor will then correspond to an equilibrium state point resulting from mixing of proper weights of vapor at state points b' and b". Thus it is necessary to determine the weight of refrigerant being circulated through each part of the system. The respective enthalpies at points b' and b" are 606.7 and 616 (assuming subcooling to 70° F and vapor leaves the evaporator at saturated state).

Refrigerating effect in evaporation operating at 20.88 lbs. per sq. in. = 606.7 - 120.5

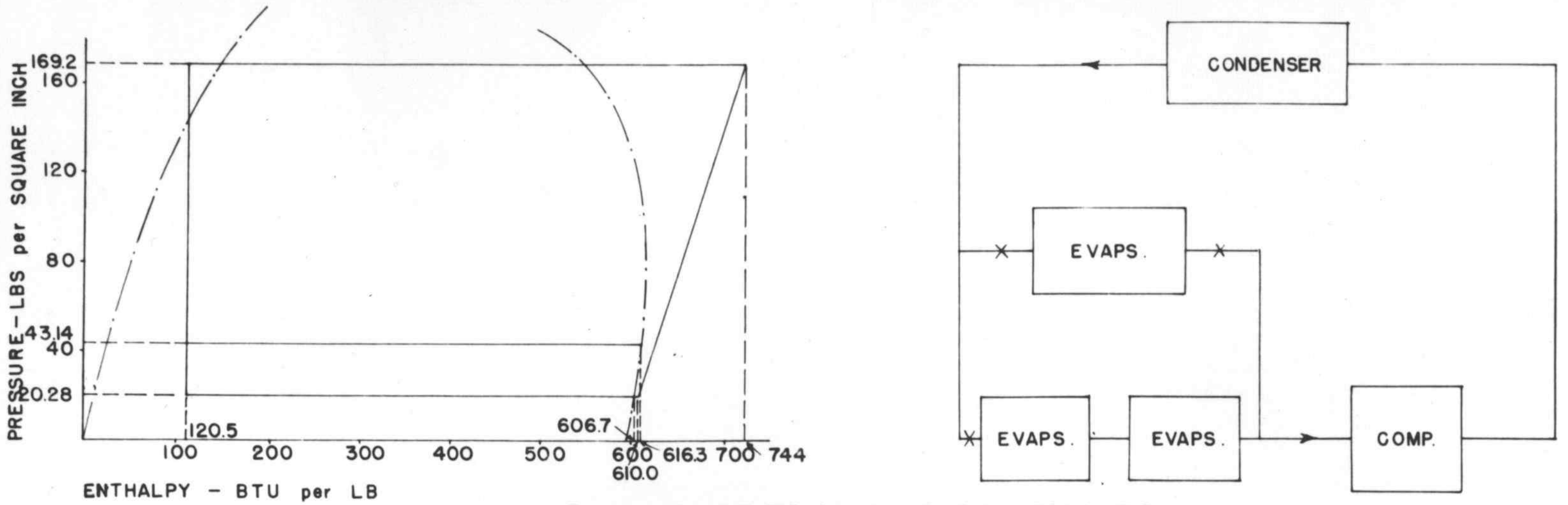
$$= 486.2 \ \frac{\text{B.t.u.}}{\text{lb.}}$$

Weight of refrigerant through this evaporator =

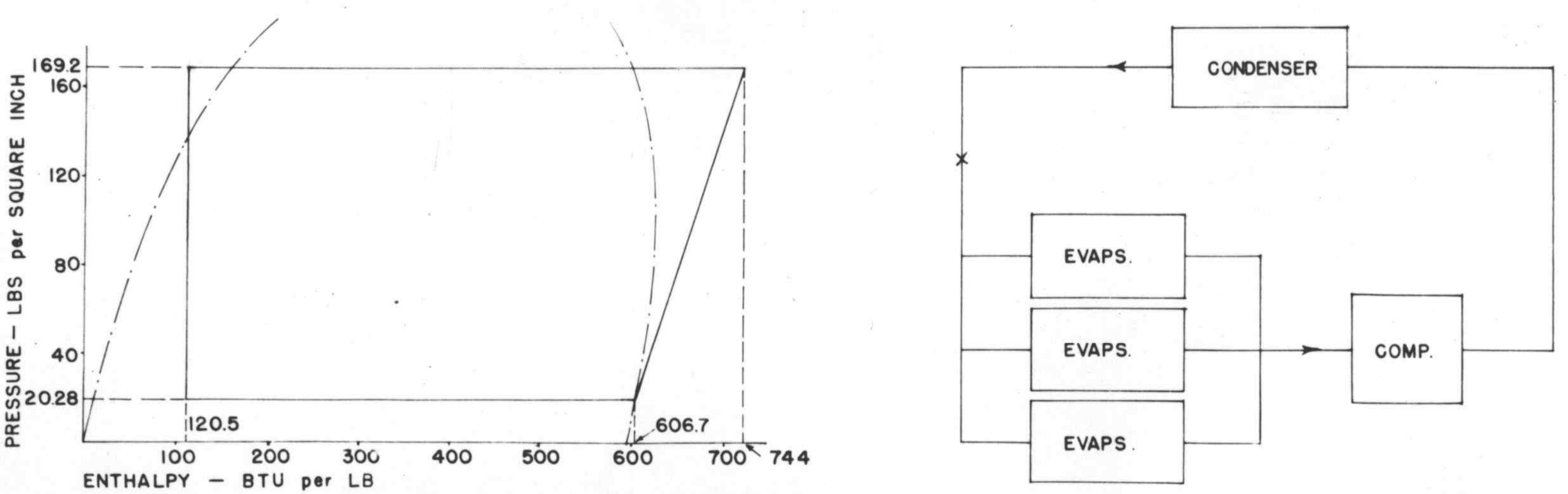
$$\frac{(1.35 + 0.43) \text{ Tons} \times 200 \ \frac{\text{B.t.u.}}{\text{ton min.}}}{486.2 \ \frac{\text{B.t.u.}}{\text{lb.}}}$$

$$= 0.735 \ \frac{\text{lb.}}{\text{min.}}$$

Refrigerating effect in the evaporator operating at 15° F or 43.14 lb. per sq. in.



P-H CHARTS and FLOW CYCLES



$$= \frac{1.525 \text{ tons} \times 200 \frac{\text{B.t.u.}}{\text{min.} \times \text{ton}}}{495.8 \frac{\text{B.t.u.}}{\text{lb}}}$$

$$= 0.617 \frac{\text{lb}}{\text{min.}}$$

Then the state point of the mixed vapor at the compressor entrance is the weighted average of individual state points and is determined from the known pressure and the weighted average enthalpy.

$$h_b = \frac{0.735 \frac{\text{lb}}{\text{min.}} \times 606.7 \frac{\text{B.t.u.}}{\text{lb}} + 0.617 \frac{\text{lb}}{\text{min}} \times 616.3 \frac{\text{Btu}}{\text{lb}}}{0.735 \frac{\text{lb}}{\text{min.}} + 0.617 \frac{\text{lb}}{\text{min.}}}$$

$$= 610 \frac{\text{B.t.u.}}{\text{lb}}$$

The compression line (assumed to be isentropic) is drawn through point b and intersects the condenser pressure at 742 B.t.u. per lb.

Work of compression:

$$= \frac{(742 - 610) \frac{\text{B.t.u.}}{\text{lb}} \times (0.735 + 0.617) \frac{\text{lb}}{\text{min}}}{42.42 \frac{\text{B.t.u.}}{\text{min. H.P.}}}$$

$$= 4.05 \text{ H.P.}$$

The second alternative is operating with all evaporators at 20.88 lb. per sq. in. This will necessitate smaller heat transfer surface in the evaporators to prevent over cooling of the higher temperature cold rooms.

Using the same figures calculated before for 20.88 lb. per sq. in.,

refrigerating effect = 486.2

$$\text{weight of refrigerant} = \frac{3.305 \text{ Tons} \times 200 \ \frac{\text{B.t.u.}}{\text{ton min.}}}{486.2 \ \frac{\text{B.t.u.}}{\text{lb.}}}$$

$$= 1.36 \ \frac{\text{lb.}}{\text{min.}}$$

Following an isentropic compression line from the state point corresponding to saturated vapor at 20.88 lb. per sq. in. to the intersection with condenser pressure 169 lb. per sq. in., we find enthalpy leaving compressor 740 B.t.u. per lb.

Work of compression:

$$= 740 \ \frac{\text{B.t.u.}}{\text{lb.}} - 606 \ \frac{\text{B.t.u.}}{\text{lb.}}$$

$$= 134 \ \frac{\text{B.t.u.}}{\text{lb.}}$$

$$= \frac{134 \ \frac{\text{B.t.u.}}{\text{lb.}} \times 1.36 \ \frac{\text{lb.}}{\text{min.}}}{42.42 \ \frac{\text{B.t.u.}}{\text{min ton}}}$$

$$= 4.1 \text{HP}$$

This shows that the compressor work is nearly same for both cases.

Therefore we shall use second alternative to get cooler evaporator temperature.

COMPRESSOR DESIGN

Capacity	=	3.305
Condenser temperature	=	86° F
Evaporator "	=	-15° F
Refrigerating effect	=	486.2 $\frac{\text{B.t.u.}}{\text{lb.}}$
Compression work	=	134 $\frac{\text{B.t.u.}}{\text{lb.}}$
Refrigerant required	=	1.36 lb./min.
Specific volume	=	12.97 $\frac{\text{cu. ft.}}{\text{lb.}}$
Volume	=	1.36 $\frac{\text{lb.}}{\text{min.}}$ x 12.97 $\frac{\text{cu. ft.}}{\text{lb.}}$
	=	17.5 $\frac{\text{cu. ft.}}{\text{min.}}$
	=	30200 $\frac{\text{cu. in.}}{\text{min.}}$

Volumetric efficiency for different operating conditions are given in Macintire, <u>Refrigeration Engineering</u>:

Ratio $\frac{P_2}{P_1}$	E_s
2.0	0.85
3.0	0.80
4.0	0.75
5.0	0.70
6.0	0.64
7.0	0.58

Where $\frac{P_2}{P_1}$ the ratio of the liquifaction to evaporation pressures.

$$\frac{P_2}{P_1} = \frac{169}{20.88} = 8$$

Volumetric efficiency = .52

Actual volume of compressor

$$= \frac{30,200 \text{ cu. in. per min.}}{.52}$$

$$= 58000 \text{ cu. in.}$$

The maximum speeds and displacements of twin cylinder V.S.A. Ammonia Compressors are given in the following table*:

Bore & Stroke, In.	Max. RPM*	Cu. Ft. per Rev.	Cu. In. per Rev.	Piston Speed Ft. per Min.	Displacement Cu.In.per Min.
4x4	510	0.0581	100.4	340	51,200
4½x4½	495	0.0828	143.1	371	70,900
5x5	480	0.1135	196.3	400	94,300
5½x5½	465	0.1512	261.3	426	121,500

Closest size = 4½ x 4½ in.

$$\text{Volume} = \frac{70,900 \text{ cu. in.}}{\text{min.}}$$

*Taken in part from Equipment Standards formulated by the Refrigerating Machinery Association, 1938.

Although this is larger than the required size, it will be useful during overload period. Required refrigeration can be obtained by adjusting the speed.

Mechanical efficiency, 58%

$$HP = \frac{134 \frac{\text{B.t.u.}}{\text{lb.}} \times 1.36 \frac{\text{lb.}}{\text{min.}}}{42.42 \frac{\text{B.t.u.}}{\text{min.}} \times .58}$$

$$= 7 \text{ HP}$$

$$\frac{HP}{Ton} = \frac{7 \text{ HP}}{3.305 \text{ Ton}}$$

$$= 2.1 \text{ HP}$$

COMPRESSOR FOR ICE MAKING

Total capacity 48 Ton

Use 2-24 Ton compressors.

Condenser temperature = 86° F

Evaporator temperature = 5° F

Refrigerating effect = $494 \frac{\text{B.t.u.}}{\text{lb.}}$

Compression Work = $99.4 \frac{\text{B.t.u.}}{\text{lb.}}$

Amount of refrigerant required

$$= \frac{24 \text{ Tons} \times 200 \frac{\text{B.t.u.}}{\text{min. ton}}}{494.1 \frac{\text{B.t.u.}}{\text{lb.}}}$$

$$= 9.7 \frac{\text{lb.}}{\text{min.}}$$

$$\text{Displacement} = 9.7 \frac{\text{lb.}}{\text{min.}} \times 8.15 \frac{\text{cu. ft.}}{\text{lb.}}$$

$$= 79 \frac{\text{cu. ft.}}{\text{min.}}$$

$$= 136000 \ \frac{\text{cu. in.}}{\text{min.}}$$

Volumetric efficiency = 70%

$$\text{Displacement} = \frac{136000}{.7}$$

$$= 194000 \ \frac{\text{cu. in.}}{\text{min.}}$$

Size - Bore and stroke $6\frac{1}{2}" \times 6\frac{1}{2}"$

Mechanical efficiency 70%

$$HP = \frac{9.7 \ \frac{\text{lb.}}{\text{min.}} \times 99.4 \ \frac{\text{B.t.u.}}{\text{lb.}}}{42.42 \ \frac{\text{B.t.u.}}{\text{min.}} \times .70}$$

$$= 31.3 \ HP$$

$$HP = \frac{31.3 \ HP}{24 \ \text{Tons}}$$

$$= 1.3 \ \frac{HP}{\text{Tons}}$$

THE EVAPORATORS

THE EVAPORATORS

The evaporators selected for different rooms are:

Fish freezing room	-	Unit cooler
Fish storage room	-	Unit cooler
Ice storage room	-	Expansion coils
Ice tank	-	Verti-Flow coils

All the specifications of unit-coolers are given by the manufacturer*. For the ice storage tank, Verti-flow coils are used. This is a special type evaporating coil with two rows of vertical pipes extending between headers at the top and bottom. This type of cooling surface has been successfully applied to ice-making service. Better efficiency can be obtained by simply increasing the circulation of the brine.

By increasing the velocity in the tank as it flows past the vertical pipes, it is possible to raise the heat transfer of this type coil to such point where the amount of evaporating surface required is considerably less than with any other kind of coil.

The construction of the Verti-flow coil is shown in the picture, and its position is illustrated in the drawing. The bulk heads for directing the flow of brine

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*Jacobs & Gile, Inc., Catalog & Manual

over the coils are placed and are enclosed to form a tunnel.

The vertical centrifugal pumps are used to get better circulation in the tunnel enclosing the Verti-flow coils.

EVAPORATOR FOR FISH FREEZING ROOM

Room temperature	=	$-5^{\circ}F$
NH_3 temperature	=	$-15^{\circ}F$
Refrigeration load	=	$123{,}850\ \frac{\text{B.t.u.}}{24\ \text{hr}}$
Freezing time	=	12 hrs.

For the freezer room, it is highly recommended by W. H. Martin, Professor of Mechanical Engineering, to use low temperature unit cooler. The cooler used has the following specifications:

It is a Freez-E-Fex type forced draft cooling unit, Model F19FRS, capacity 9300 $\frac{\text{B.t.u.}}{\text{hr}}$ for $5^{\circ}F$ temperature difference or equivalent.

$$\text{Freezing room load} = 123{,}850\ \frac{\text{B.t.u.}}{24\ \text{hr}}$$

$$= 5150\ \frac{\text{B.t.u.}}{\text{hr}}$$

Considering the load variation and extension of plant, a higher capacity cooler was selected to use, for the time being, which will decrease the cooling time about one-half.

EVAPORATOR DESIGN FOR FISH STORAGE ROOM

Room temperature = 8° F

NH_3 temperature = -15° F

Load = 387100 $\frac{\text{B.t.u.}}{\text{24 hr.}}$

Temperature difference = 23° F

In this room, select the same kind of forced draft unit cooler as used in the fish freezing room.

Capacity of F19FRS Model cooler for 23° F temperature difference has the cooling capacity of 37.200 $\frac{\text{B.t.u.}}{\text{hr}}$

Load in the room, 16.100 $\frac{\text{B.t.u.}}{\text{hr.}}$. Again, this will reduce the cooling time one-half.

EVAPORATOR DESIGN FOR ICE MAKING

Capacity = 48 Tons

Brine temperature = 14° F

Water temperature supplying cans = 70° F

$Q = WC\Delta T$

Q = Heat removed per hr.

$\Delta t = t_{brine} - t_d$

t_d = temperature drop of NH_3

, $\Delta t = 14^\circ\ F - 4^\circ\ F$

$= 10^\circ\ F$

$C =$ specific heat of brine

$= 0.704^{*}\ \frac{B.t.u.}{lb\ ^\circ F}$

$W = \frac{Q}{C\Delta t}$

$= \frac{48\ Ton \times 12000\ \frac{B.t.u.}{Ton\ hr}}{0.704\ \frac{B.t.u.}{lb - {}^\circ F} \times 4^\circ F}$

$= 205000\ \frac{lb}{hr}$

$= 3420\ \frac{lb}{min}$

GPM $= 385$

EVAPORATOR DESIGN FOR ICE STORAGE ROOM

Assuming direct expansion coils and free convection is used, the design procedure is as follows:

Room temperature $= 26^\circ$ F

NH_3 temperature $= -15^\circ$ F

Refrigeration load $= 34000\ \frac{B.t.u.}{24\ hr}$

$Q = UAT_m$

$T_m = 26^\circ\ F - (-15)^\circ F$ (which is very close for free convection)

$= 41^\circ$ F

---

* Marks Mechanical Engineering Hand Book, P. 2166

$$U^* = 20 \frac{\text{B.t.u.}}{\text{hr } ^\circ\text{F ft}^2}$$

Q = Refrigeration load

$$A = \frac{340{,}000 \dfrac{\text{B.t.u.}}{24 \text{ hr}}}{2 \dfrac{\text{B.t.u.}}{\text{hr } ^\circ\text{F ft}^2} \times 41^\circ \text{ F} \times 24 \text{ hrs}}$$

$$= 170 \text{ ft}^2$$

Introducing the ratio of surface is to length 2.3

$$L = 2.3 \times 170 \text{ ft}$$

$$L = 391 \text{ ft}$$

Say L - 400 ft

Maximum allowable length of pipe corresponding to pipe size for direct expansion coils are:

Pipe Size	Maximum Allowable Length
3/4" D	900'
1" D	1100'
1¼" D	1300'
2" D	1900'

To control the capacity, use 2 200 ft coils supported 8 in. below the ceiling, and located evenly in the room.

Assume length of the coil to be 20 ft. Then each evaporator has 10 - 20 ft. coil distance between the pipe centers 4 in.

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*Adams, M. C., Heat Transmission, Tables VIII, XI, pp. 247, 251.

EVAPORATORS FOR ICE TANK

$Q = UAT_m$

Q = Refrigeration required for ice making

= 48 Tons

U = Conductivity of evaporator coils

$= 100 \frac{\text{B.t.u.}}{\text{hr } {}^{\circ}\text{F ft}^2}$

$$T_m = \frac{\Delta t_1 - \Delta t_2}{\ln \frac{\Delta t_1}{\Delta t_2}}$$

$$= \frac{9^{\circ}\text{ F} - 5^{\circ}\text{ F}}{\ln \frac{9}{5}}$$

$= 6.75^{\circ}$ F

$$A = \frac{Q}{U\ T_m}$$

$$= \frac{48 \text{ Tons} \times 12000 \frac{\text{B.t.u.}}{\text{hr ton}}}{100 \frac{\text{B.t.u.}}{\text{hr } {}^{\circ}\text{F} - \text{ft}^2} \times 6.75^{\circ}\text{F}}$$

$= 810$ ft^2

Taking $1\frac{1}{4}$" Dia. standard steel pipe and also introducing ratio of surface area to length 2.3,

$$L = 810 \text{ ft}^2 \times 2.3 \frac{\text{ft}}{\text{ft}^2}$$

$$L = 1860$$

For each ice tank, we need

$$L = \frac{1860}{2}$$

$$= 930 \text{ ft} - 1\tfrac{1}{4}\text{" D pipe}$$

FREEZING TIME

FREEZING TIME

Time of Freezing Ice:

$$T = \frac{N\,W\,24}{2000}$$

T = Time of freezing, hrs.

W = Weight of ice can, lbs.

w = dimension in inches

t_b = brine temperature, $^{\circ}F$

$$N = \frac{516.6 \times 11^2}{315\,(32-14)}$$

N = 11 cans

$$T = \frac{11 \text{ cans} \times 315\,\frac{\text{lb.}}{\text{can}} \times 24 \text{ hrs.}}{2000\,\frac{\text{lb.}}{\text{tons}}}$$

$$= 41.5 \text{ hrs.}$$

Total cans required per day

$$= \frac{30\,\frac{\text{Ton}}{\text{Day}} \times 2000\,\frac{\text{lb.}}{\text{ton}}}{315\,\frac{\text{lb.}}{\text{can}}} \times \frac{41.5 \text{ hrs.}}{24 \text{ hrs.}}$$

$$= 330\,\frac{\text{cans}}{\text{day}}$$

CONDENSER

CONDENSER

Condensing pressure = 169 lb. per sq. in.

Total tons of refrigeration = 51.305 Tons

Entering temperature of water = 65^{o}

Leaving temperature of water = 75^{o}

Weight of ammonia in the compressor for

ice making = 20.2 lb/min

$Q = W (h_a - h_d)$

h_a = heat content of NH_3 entering condenser

$= 714 \frac{\text{B.t.u.}}{\text{lb.}}$

h_d = heat content of NH_3 leaving condenser

$= 120.5 \frac{\text{B.t.u.}}{\text{lb.}}$

W = weight of ammonia circulating

Q = heat lost by NH_3

$Q_1 = 20.2 \frac{\text{lb}}{\text{min}} (714 - 120.5) \frac{\text{B.t.u.}}{\text{lb.}}$

$= 12000 \frac{\text{B.t.u.}}{\text{min.}}$

Weight of NH_3 circulating in storage rooms

small compressor = 1.355 lb./min.

$$Q_2 = 870 \ \frac{\text{B.t.u.}}{\text{min}}$$

Total heat absorbed by condenser:

$$Q = Q_1 + Q_2 = 12870 \ \frac{\text{B.t.u.}}{\text{min}}$$

Total amount of water required:

$$Q = C_p \ W \Delta T_m$$

$$C_p = 1 \ \frac{\text{B.t.u.}}{\text{lb. }^\circ\text{F}}$$

$$T_m = 14^\circ \text{ F} - 10^\circ \text{ F}$$

$$T_m = 10^\circ \text{ F}$$

$$W = \frac{12870 \ \frac{\text{B.t.u.}}{\text{min}}}{1 \ \frac{\text{B.t.u.}}{16^\circ \text{ F}} \times 10^\circ \text{ F}}$$

$$W = 915 \ \frac{\text{lb.}}{\text{min.}}$$

$$\text{GPM} = 110$$

Select the condenser Vertical-Shell and tube type, making use of Figure 4, Refrigeration Data Book, p. 259.

For a discharge pressure 169 - 14.7 = 154.3 psi and inlet condenser water temperature of 75°, it requires 8 sq. ft. per ton.

Take capacity 54 ton, to allow losses:

$$\text{Total surface required} = 54 \text{ ton} \times 8 \ \frac{\text{ft}^2}{\text{ton}}$$

$$= 432 \text{ ft}^2$$

To this total surface some correction factors* should be introduced.

Total surface,

$$S = 432 \times fsm \times fsc$$

fsm = Surface multiplier

fsc = Surface correction factor

Average suction pressure of two compressors,

$$= \frac{20.88 \frac{\text{lb.}}{\text{sq. in.}} - 34.27 \frac{\text{lb.}}{\text{sq. in.}}}{2}$$

$$= 27.5 \frac{\text{lb.}}{\text{sq.in.}}$$

fsm = .975

In determining the surface correction factor, fsc, assume N - 2" tubes with a length of 12 ft,

O.D. = 2.375"

But the total surface,

$$S = 432 \times .975 \times fsc \text{ ft}^2$$

$$= 420 \; fsc \text{ ft}^2$$

---

*Refrigeration Data Book, p. 259.

can also be expressed:

$$S = N\,l \times D$$

$$S = N \times 12\text{ ft} \times \frac{2.375\text{ in.}}{12\ \frac{\text{in.}}{\text{ft.}}}$$

$$= 7.46\ N$$

$$N = \frac{GPM}{\frac{GPM}{Tube}}$$

$$= \frac{110\ GPM}{GPM}$$

By substituting above relations for areas

$$420\ fsc\ ft^2 = \frac{110\ GPM \times 7.46}{GPM}\ \frac{GPM}{Tube}$$

$$fsc \times GPM = 1.95$$

Assume

3 GPM gives fsc = .738

3 x .738 = 1.95 GPM

2.2 = 1.95 GPM

This value is close enough.

$$S = 432\ ft^2 \times .975 \times 7.38$$

$$= 310\ ft^2$$

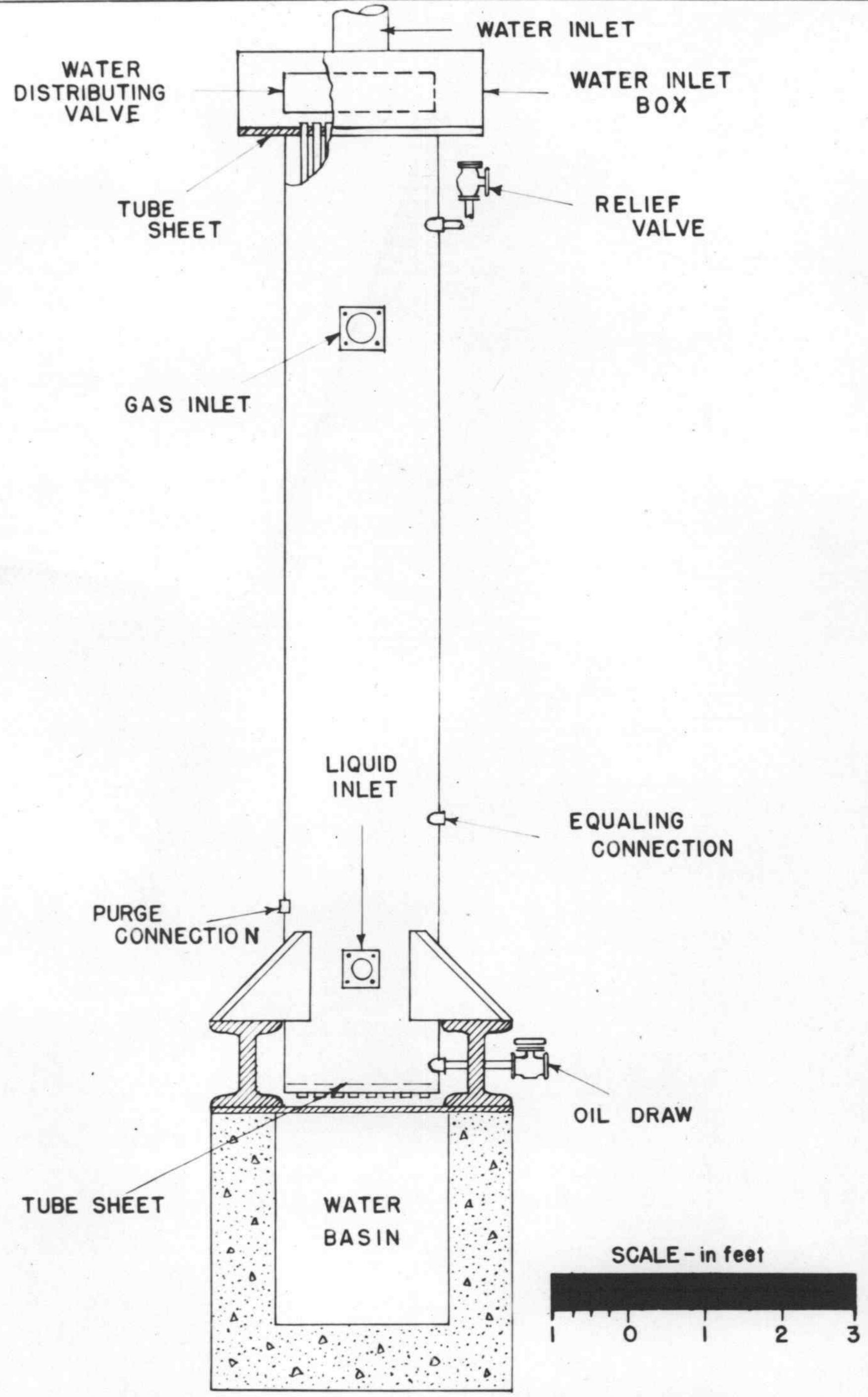
Number of tubes required

$$= \frac{310}{3} = 104$$

Assuming a shell diameter 24",

Number of 2" tubes = 51 = 20%

Therefor, we can use 2 - 24" tube shells.



VERTICAL OPEN SHELL-and-TUBE AMMONIA CONDENSER

AIR COMPRESSOR AND PUMPS

AIR TREATMENT FOR ICE

The diagram shows the low pressure system of air treatment for raw water ice. In this system there is a continuous recirculation of air one time after another. The air first goes through the water in the cans, then goes through the spaces in the framework between the can surface and covering block, then to the air headers, and finally to the blower suction.

With this method, very low pressure air can be used, and the percentage of undergrade ice reduced very much. It is recommended by A. H. Boer in the paper "Air Treatment for Raw Ice" to supply to each can about .5 cubic feet of air per minute and lifting pressure by the blower 28 in. water

Air Compressor

As stated before, supply of air to each can-- 0.5 cubic feet per minute, and pressure -- 28 in. of water. We have 330 cans.

$$Q = 0.5 \frac{\text{cu. ft.}}{\text{min-can}} \times 330 \text{ cans}$$

$$Q = 165 \frac{\text{cu. ft.}}{\text{min.}}$$

The power required to circulate air can be found from the following method:

Work = weight of air x total lift = $Q P_a h_a$ ft.lb.

Q = Volume of air cu. ft.

d_a = Density of air

h_a = the pumping head

$$h_a d_a = \frac{h \text{ in. of water}}{12} \, d \text{ of water}$$

$$h_a = \frac{h_{water} \times d_w}{12\, d_a}$$

$$= \frac{5.2\, Q\, h_w}{33000\, e}$$

h_w = 28" total head inches of water

e = efficiency of fan

d_w = density of water

$$HP = \frac{5.2\, Q\, h_w}{33000\, e}$$

$$= \frac{5.2 \times 165 \times 28}{33000 \times .60}$$

$$HP = 0.120$$

PUMPS

Condenser Pump

For pumping the condenser water, we install two centrifugal single stage pumps each of 55 GPM.

Then specifications of the pump are*:

efficiency = 70%

specific speed n_{sq} = 2000

$$RPM = n_{sq} \times \frac{h^{3/4}}{\sqrt{GPM}}$$

*A. Church, Centrifugal Pump, P. 64, fig. 4.12

Assuming a head of 30 ft,

$$RPM = \frac{(30)^{3/4}}{\sqrt{55}} \quad 2000$$

$$= \frac{12.7}{7.45} \times 2000$$

$$= 3400 \text{ RPM}$$

$$HP = \frac{8.33\ Q\ h}{33000 \times \text{eff.}}$$

$$Q = G\ P\ M$$

$$HP = \frac{8.33 \times 55 \times 30}{33000 \times .70}$$

Brine Pumps

Vertical centrifugal pump

Capacity = 378 G P M

H = 30 ft. (in the case I need more circulation)

Say two brine pumps each has capacity 190 GPM,

Efficiency = 72

Specific speed = 2000

$$n_s = \frac{RPM \times \sqrt{GPM}}{H^{3/4}}$$

n_s = specific speed

H = head in ft.

H = 30 ft.

$$RPM = \frac{n_s \times H^{3/4}}{\sqrt{G\ P\ M}}$$

$$RPM = \frac{2000 \times (30)^{3/4}}{13.8}$$

R P M = 1850

$$HP = \frac{W\,H}{33000 \times eff.}$$

$$W = \text{capacity in } \frac{lb.}{min.}$$

H = ft.

eff. = .90

$$HP = \frac{190\ GPM \times 89\ \frac{lb.}{gal.} \times 30}{33000\ \frac{ft.lb.}{min.HP} \times .9}$$

$$= 15.5\ HP$$

PIPES

PIPES

a. For ice compressors.

1. Suction pipe.

$$NH_3 \text{ volume} = 136000 \ \frac{\text{cu.in.}}{\text{min.comp.}} \times 2 \text{ comp.}$$

$$= 273000 \ \frac{\text{cu. in.}}{\text{min.}}$$

Let NH_3 vapor velocity $= 2500 \ \frac{\text{ft.}}{\text{min.}}$

$Q = AV$

$V = NH_3$ velocity

$A =$ cross sectional area

$Q =$ quantity flowing

$$A = \frac{273000 \ \frac{\text{cu. in.}}{\text{min}}}{1728 \ \frac{\text{cu. in.}}{\text{cu. ft.}} \times 2500 \ \frac{\text{ft.}}{\text{min.}}}$$

$= .0634$ sq. ft.

$= 9.15$ sq. in.

$$\text{Pipe diameter} = \sqrt{\frac{9.15 \text{ in.}^2}{0.7854}}$$

$= 3.4$ in.

Use steel pipe of diameter $3\frac{1}{2}$".

2. From compressor to condenser valve:

$$NH_3 \text{ volume} = 9.7 \ \frac{\text{lb.}}{\text{min. comp.}} \times 2 \text{ comp.} \times 2.5 \ \frac{\text{cu.ft.}}{\text{lb.}}$$

$$= 48.4 \ \frac{\text{cu. ft.}}{\text{min.}}$$

$$\text{Pipe area} = \frac{48.4\ \frac{\text{cu. ft.}}{\text{min.}} \times 144\ \frac{\text{sq. in.}}{\text{sq. ft.}}}{2500\ \frac{\text{ft.}}{\text{min.}}}$$

$$= 2.77 \text{ sq. in.}$$

Pipe diameter = 1.88"

Use steel pipe diameter 2"

3. From condenser to expansion valve. In this line ammonia is in liquid form:

$$\text{Density} = 0.026\ \frac{\text{ft.}^3}{\text{lb.}}$$

$$= 19.4\ \frac{\text{lb.}}{\text{min.}} \times 0.026\ \frac{\text{ft.}^3}{\text{lb.}}$$

$$= .505\ \frac{\text{ft.}^3}{\text{min.}}$$

Velocity of liquid = 200 ft./min.

$$\text{Pipe area} = \frac{0.505\ \frac{\text{ft.}^3}{\text{min.}}}{200\ \frac{\text{ft.}}{\text{min.}}} \times 144\ \frac{\text{sq. in.}}{\text{sq. ft.}}$$

$$= .363 \text{ sq. in.}$$

Pipe diameter = .66"

Use steel pipe of diameter 3/4".

b. 1. Suction pipe for the storage rooms:

$$NH_3 \text{ volume} = 1.36\ \frac{\text{lb.}}{\text{min.}} \times 12.97\ \frac{\text{cu. ft.}}{\text{lb.}}$$

$$= 17.6\ \frac{\text{cu. ft.}}{\text{min.}}$$

$$\text{Pipe area} = \frac{17.6\ \frac{\text{cu.ft.}}{\text{min.}} \times 144\ \frac{\text{sq. in.}}{\text{sq. ft.}}}{2500\ \frac{\text{ft.}}{\text{min.}}}$$

$$= 1.01 \text{ sq. in.}$$

$$\text{Pipe diameter} = \sqrt{\frac{1.01 \text{ sq. in.}}{0.7854}}$$

$$= 1.14''$$

Use steel pipe of diameter $1\frac{1}{4}''$

2. From compressor to condenser:

$$NH_3 \text{ Volume} = 1.36\ \frac{\text{lb.}}{\text{min.}} \times 2.5\ \frac{\text{cu. ft.}}{\text{lb.}}$$

$$= 3.4\ \frac{\text{cu. ft.}}{\text{min.}}$$

$$\text{Pipe area} = \frac{3.4\ \frac{\text{cu.ft.}}{\text{min.}} \times 144\ \frac{\text{sq. in.}}{\text{sq. ft.}}}{2500\ \frac{\text{ft.}}{\text{min.}}}$$

$$= .195 \text{ sq. in.}$$

$$\text{Pipe diameter} = \sqrt{.195\ \frac{\text{sq. in.}}{0.7854}}$$

$$= 0.5''$$

Use steel pipe of diameter $\frac{1}{2}''$.

3. From condenser to expansion valve:

$$\text{Density} = 0.026\ \frac{\text{ft.}^3}{\text{lb.}}$$

$$= 1.36\ \frac{\text{lb.}}{\text{min.}} \times 0.026\ \frac{\text{ft.}^3}{\text{lb.}}$$

$$= .0354\ \frac{\text{ft.}^3}{\text{min.}}$$

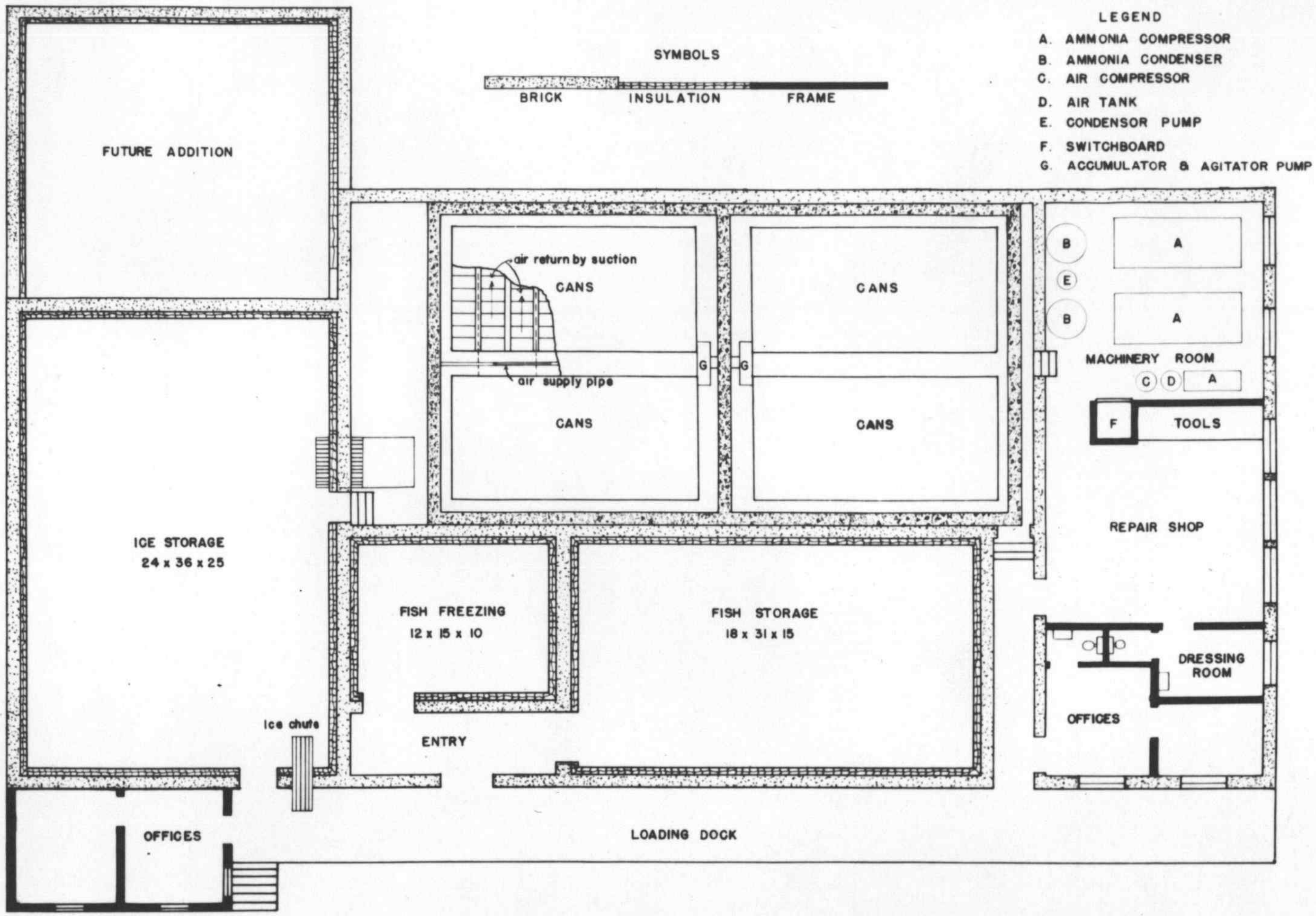
$$\text{Velocity} = 200 \ \frac{\text{ft.}}{\text{min.}}$$

$$\text{Pipe area} = \frac{0.0354 \ \frac{\text{ft.}^3}{\text{min.}}}{200 \ \frac{\text{ft.}}{\text{min.}}} \times 144 \ \frac{\text{sq. in.}}{\text{sq. ft.}}$$

$$= .0255 \text{ sq. in.}$$

Pipe diameter = .18"

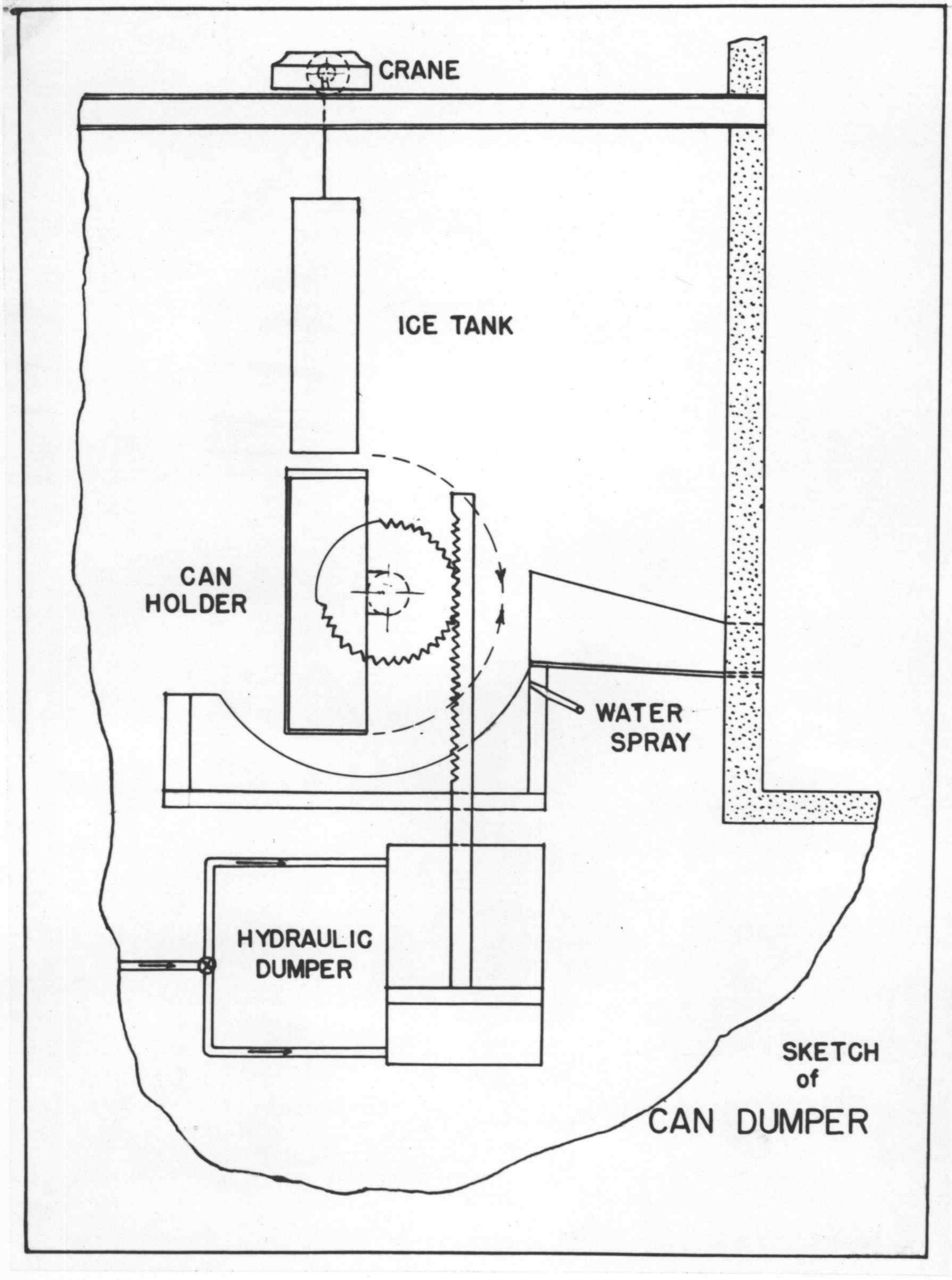
Use steel pipe of diameter $\frac{1}{4}$".



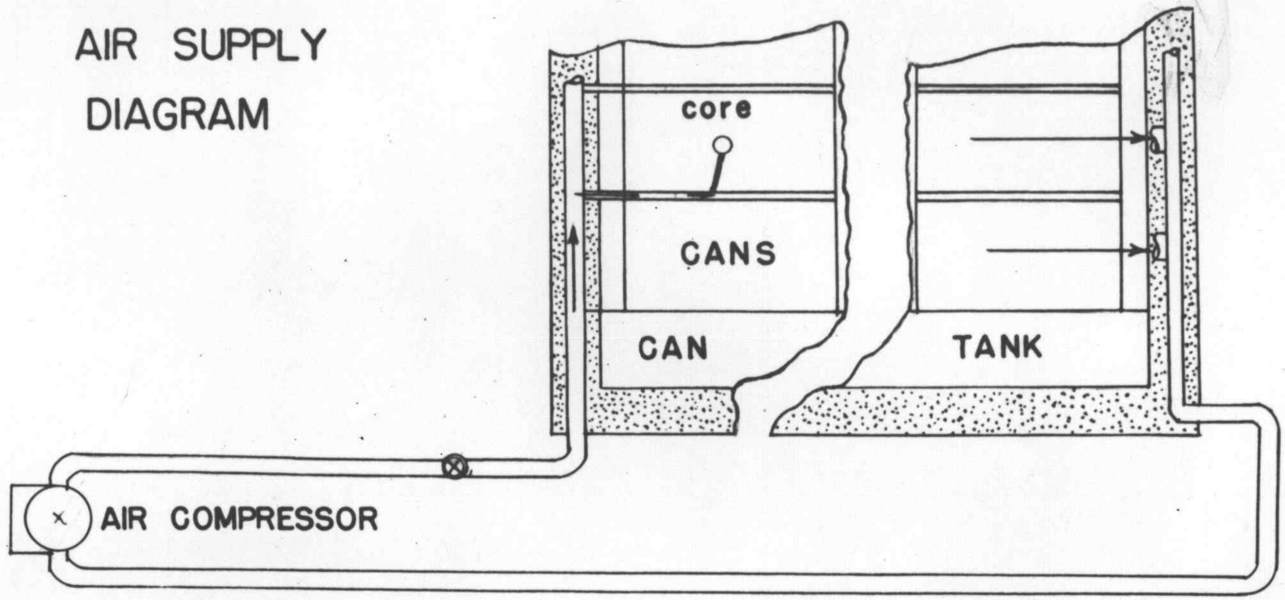
SKETCH of ICE PRODUCTION and STORAGE PLANT

SCALE in feet

1 0 1 2 3 4 5 10 20


CRANE
ICE TANK
CAN
HOLDER
WATER
SPRAY
HYDRAULIC
DUMPER
SKETCH
of
CAN DUMPER

AIR SUPPLY
DIAGRAM



BIBLIOGRAPHIES

American Society of Refrigerating Engineers. Refrigerating Data Book and Catalogue for 1934-1936. American Society of Refrigerating Engineers, New York, N. Y., 1934.

Macintire, H. J., Refrigeration Engineering, Rev. Ed., John Wiley & Sons.

Marks, L. S., Mechanical Engineers' Handbook, Mc-Graw-Hill Book Company, New York, 1932.

Moyer, J. A. and Fittz, R. U., Refrigeration, Mc-Graw-Hill Book Company, New York, 1932.

Product Engineering, March, 1942, periodical.