A THESIS

on

THE DESIGN OF A TWENTY TON

ICE PLANT

PRESENTED TO

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## PREFACE.

The writer of this thesis wishes to express his thanks to the York Manufacturing Co., for the many details of plan and construction furnished by their catalogues and blue-prints as sent him on request. He also expresses his thanks to Mr. Geo. Braungart Jr, of Atlanta, Ga., for innumerable suggestions and explanations.

## SCOPE.

In writing this thesis the author has in mind presenting step by step the problems involved in the design of a distilled water ice plant, showing the method of solving same, and the determinations and conclusions reached.

The work has been prepared in the form in which it usually presents itself in modern business and commercial fields, with the idea of showing the practical application of the laws governing mechanical refrigeration in one branch of the arts, viz., that of ice making, which may easile be termed the most important part of the science of mechanical refrigeration.

In the very nature of things, it has been impossible to adhere strictly to theoretical data and principles, as such adherence would not yield the solution of many details which it is thought desirable to fully determine, but recourse was had to the published engineering data, tables of efficiency and performance of several of the largest ice machine builders in this country, of which proper appreciation will be expressed later.

In all cases where tables, or other information were inserted bodily into this treatise, due credit is given to the source of such matter in foot notes, but in many cases the author has used only such parts of different data as seemed to him pertinent.

The author decided to elect for the subject of his inquiry a twenty ton ice making plant, making distilled water ice, in 200# cans, with an abundant supply of water at 70°Fahr., the plant to have in addition to the ice making plant, an ice storage of approximately 100 tons storage capacity. Further, that the machine be driven by a direct connected Corliss Engine, and that steam be furnished by a return-tubular boiler. Further, that the source of water was within easy pumping distance of the plant, without resorting to deep well pumps, or air lifts.

The reasons for selecting this particular type, size, etc., of the plant may be stated as follows:-

20 tons is a fair average installation.
The type selected constitutes by far the majority of the ice plants of the country.

70 degrees Fahr. water is an average for this section of the country, especially if the supply be abundant, as, for instance, it would be from a flowing stream, creek or river, or ample surface well.

The average ice plant contains no more than one storage room for ice, and does not, as a rule, engage in cold storage, wherefore, it was thought unnecessary, in fact, unwise to complicate the problem with a cold storage problem, which only seldom presents itself in practice in conjunction with a straight-out ice plant.

The vast majority of ice plants have direct connected Corliss Engines, and return-tubular boilers, and produce strictly distilled water ice.

Although, in many instances, it is necessary to obtain water from a lower depth than suction lift, this may be considered as a local case, and the equipment necessary to bring this water to the surface of the ground, may readily be considered as a adjunct to the ice plant, being strictly a pumping problem.

The same may be said where the supply of cooling water is limited, necessitating the use of a cooling tower, and its accessories, form an adjunct to the type of plant we are herein describing.

With these remarks, the author believes that he may proceed with the matter of his treatise, after further expressing his appreciation to the many sources of information, particularly,

Gebhart's "Steam Power Plant Engineering."

Siebel's "Compend of Mechanical Refrigeration and Engineering."

Tables published by the York Manufacturing Co.

A. M. Carroll Jr.

# Calculation and Design of a Twenty Ton Distilled-Water Ice Plant.

The calculation and design of a distilled water ice plant present many intricate and interesting problems, the solution of which determine the ultimate efficiency and value of the investment.

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*********

## Calculation of Ammonia Per Minute.

Let $t$ = temp. ammonia in condenser: 85° F.
$t_1$ = " " " exp. coils: 0° F.
$T$ = abs. temp. ammonia in cond.: 545 ft.
$T_1$ = " " " " exp. coils: 460 ft.

$h_1$ = heat of vaporization at $t_1$ = 555.5 B.T.U.
$r$ = refrigeration effect 1 lb. ammonia evap. at $t_1$ from $t$.
$s$ = specific heat of liquid ammonia = 1.1 at $t_1$.
$r = h - (t-t)s$: 555.5-(85-0)1.1 = 462 B.T.U. per lb. ammonia.

6723 B.T.U. refrig. effect req'd. per minute = $\frac{1667}{60} \times 200$

$\frac{6723}{462}$ = 14.56 lbs. ammonia circulated per minute.

## Calculation of the Size of Compressor.

Vol. 1 lb. ammo. vapor at $t_1$ (from tables) = 9.03 cu.ft.

14.56 x 9.03 = 131.3 cu.ft.

Volumetric efficiency vert. single acting machine at $t+t_1$ found to be 80%

Therefore. compressor displacement = $\frac{131.3}{80}$ = 164.3 cu.ft.

Assume 80r.p.m.

$\frac{164.3}{80}$ = 2.055 cu.ft. per revolution.

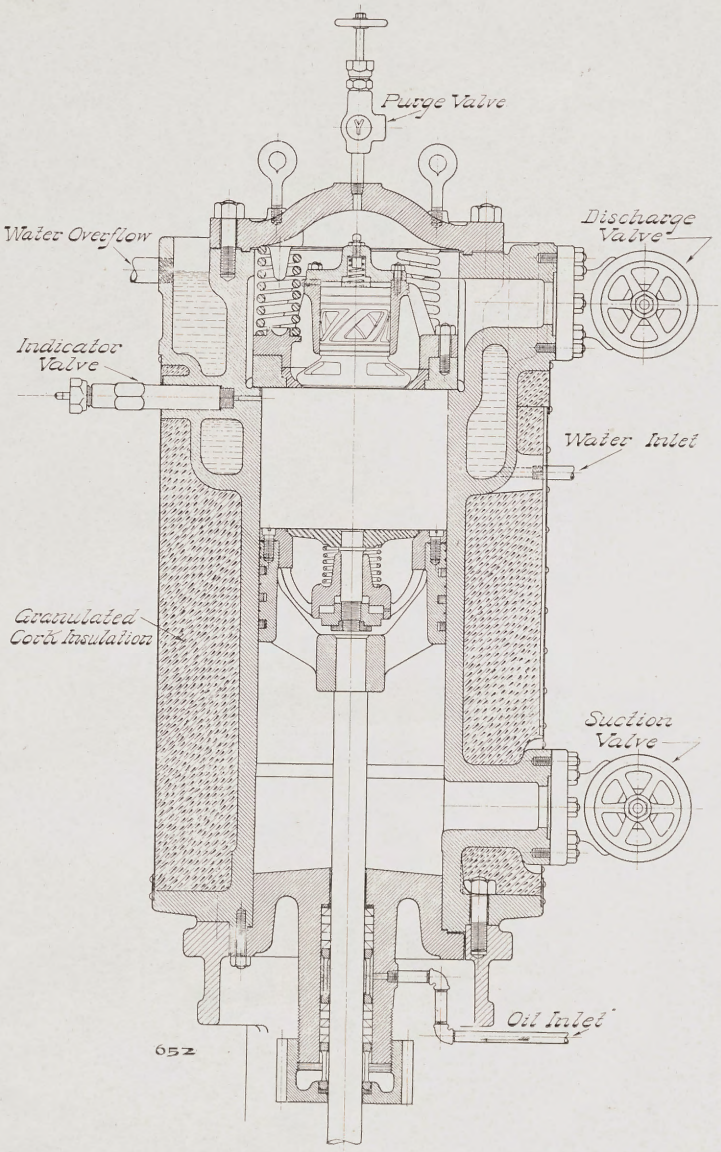
2.055 x 1728 = 3545 cu.in. per revolution.

Assume 20" stroke

$\frac{3545}{20}$ = 177.2 sq.in. area both compressors.

$\frac{177.2}{2}$ = 88.6 " " " One compressor.

= $10\frac{1}{2}$ in. diam. compressor.




Sectional View of Compressor.

## Engine Horse Power Required.

Work required to raise heat vaporization of 1 lb. ammonia at $t_1$ to $t$

$= \frac{t-t_1}{T_1} \times h_1 = \frac{85-0}{460} \times 555.5 = 102.6$ B.T.U. per lb. ammonia.

Amount ammonia circulated = 164.3 cu.ft. vapor at t

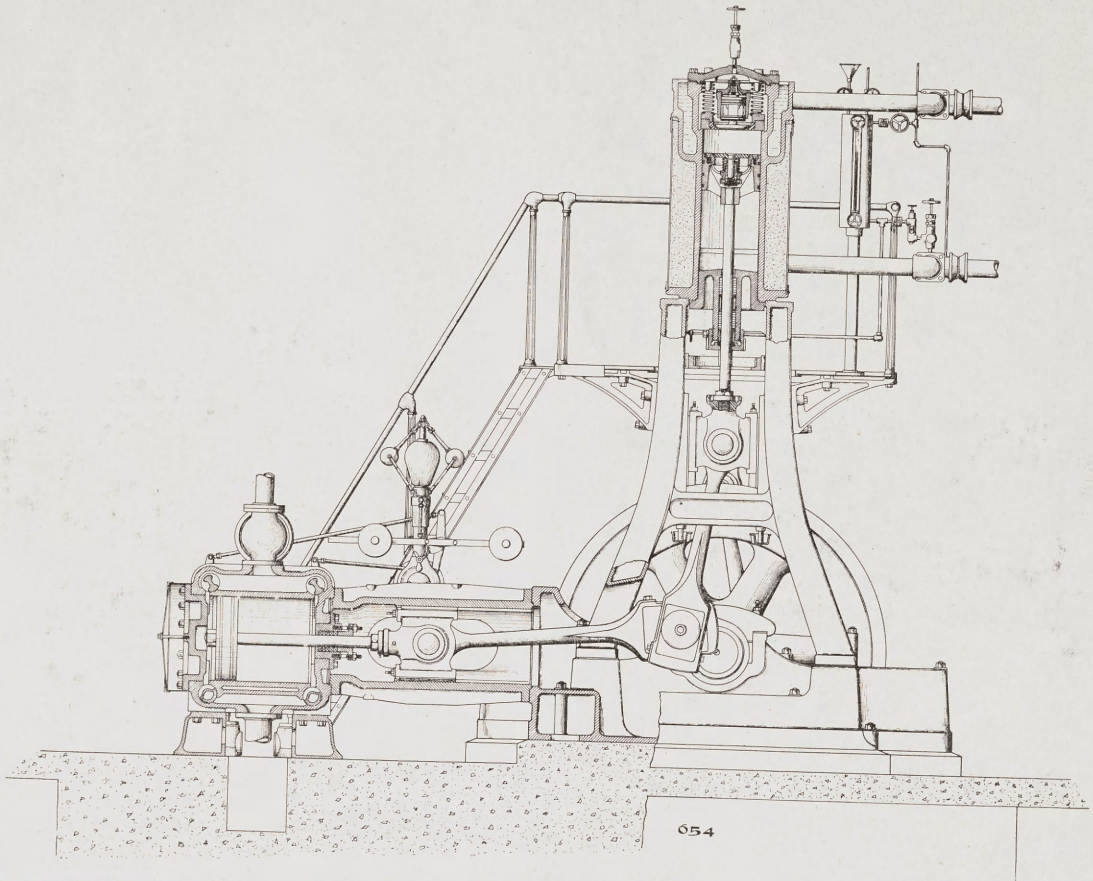
= 164.3 x.1107#: 18.2#

$\frac{18.2\# \times 102.6 \text{ B.T.U.} \times 778}{33000} = 44$ H.P.

Mechanical efficiency engine and compressor: 85%

$\frac{44}{.85} = 51.8$ engine horse power required.

Use 52 H.P.




Longitudinal Sectional View of Machine.

## Boiler Horse Power Required.

Using simple condensing Corliss engine of 52H.P.. ;low speed-

Use steam pressure: 80# per sq. in.
Feed water at 210 F.
Steam consumption of this engine is based on 26# steam per H.P. hour.

Let F: factor of evaporation
$\lambda$ : total heat of steam: 1185.3
q2: temp. of feed water-32°: 178

From Gebhart, we have

$$F = \frac{\lambda - q2}{970.4}$$

Substituting in the above formula-

$$F = \frac{1185.3-178}{970.4} = \frac{1007.3}{970.4} = 1.037$$

Total steam required by engine= 52 x 26 = 1350 #.
Steam required by auxiliaries = 400#.
Total steam actually required: 1350 + 400 : 1750#.
Allowing 150# for other possible auxiliaries, we have
Total steam required: 1750 + 150: 1900 #.
1900 lbs. steam under given conditions is equivalent to 1900 x F : 1900 x 1.037: 1970#, from and at 212 F.

Hence B.H.P. actually required

$$= \frac{1970}{34.5} = 57 \text{ H.P.}$$

However, since possible defects in the engine or auxiliaries must be taken into consideration, and since it is not wise to "push" a boiler, the condensed steam from which is to be used for ice making, it is advisable to equip with greater boiler capacity than is actually required.
We will, therefore, select one 80 H.P. boiler of the return-tubular type.

80 H.P. Boiler.
(Chimney)

Heating surface = 80 x 10 = 800 sq.ft.

Grate area = $\frac{1}{45}$ x 800 = 17.8 sq.ft.

Width grate = 4½' Length grate = 4'

Outside air 60°F. Chimney gases 500°F.

Draught.

| | |
|---|---|
| Loss in furnace (from Gebhart's curves) | .125 |
| Loss in boiler | .300 |
| Total | .425 |

Since loss in chimney alone approximates 20% of total

$\frac{.425}{.80}$ : .532 will be max. pressure diff.

From Gebhart

$$D_1 = H\left(\frac{7.64}{T_2} - \frac{7.95}{T_1}\right)$$

$$.532 = H\left(\frac{7.64}{520} - \frac{7.95}{960}\right)$$

$$H = \frac{.532}{.0064} = 83\text{ft.} = \text{height of chimney.}$$

Area of Chimney.

Kent's formula

$$A - .6\sqrt{A} = \frac{.3 \times H.P.}{\sqrt{H}}$$

$$A - .6\sqrt{A} = \frac{.3 \times 80}{\sqrt{83}}$$

$$A - .6\sqrt{A} - 2.53 = 0$$

$$\sqrt{A} = \frac{.6 \pm \sqrt{.36+10.1}}{2} = \frac{.6+3.24}{2} = 1.92$$

$$A = \overline{1.92}^2 = 3.69 \text{ sq.ft.}$$

$$D = \sqrt{\frac{14.8}{3.1416}} = 4.715 = 2.17 \text{ ft.} = 26.04\text{in.} \text{ (approx.)}$$

Use 80' high by 27" diameter.

## Auxiliaries.

### Feed Water Pump.

The type of pump in general use for this installation is a duplex horizontal pump.

The steam consumption of such a pump is found as follows:

Let

W = weight of water to be pumped per hour.

H = head to be pumped against, which is equivalent to boiler pressure.

P = weight of steam required per Horse Power hr. by pump.

Steam consumption

$$S = \frac{W \times H \times P}{33000 \times 60} = \frac{2000 \times 178 \times 125}{33000 \times 60} = 24\# \text{ steam pr. hr.}$$

### Water Pump.

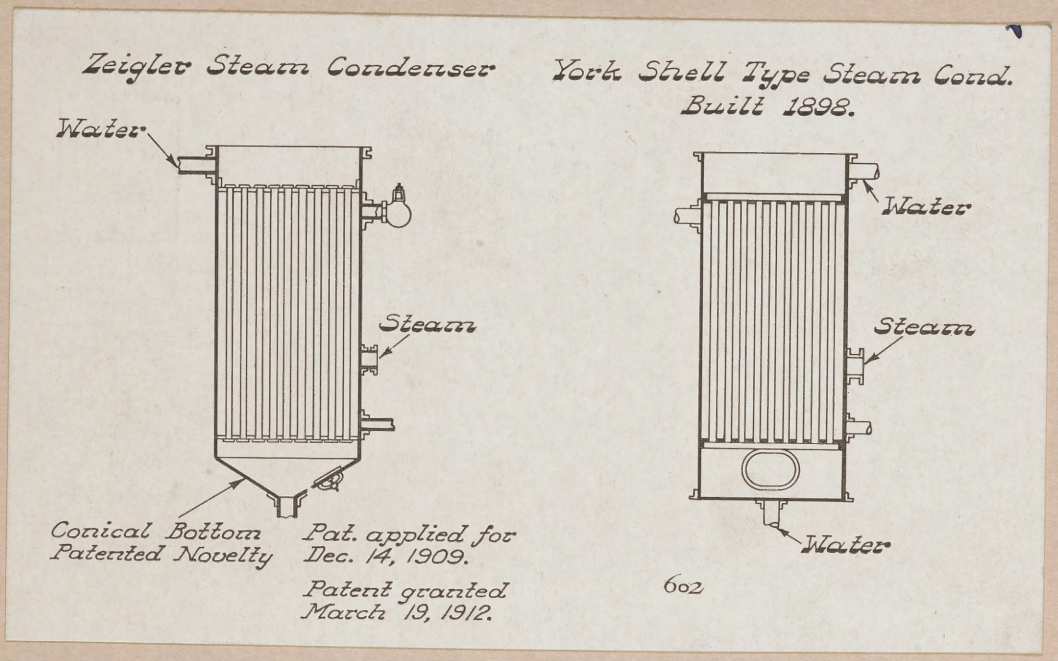
The total water to be pumped includes much condensing water which cannot be utilized as feed water. This pump is a very essential feature of the plant and should be well designed and installed.

In any plant to be built, the exact head to be pumped against will be known. For our purpose, a head of 65ft. will be assumed, and by substituting in the formula given above we have:

Since total water is to be pumped is to be about 34,800# per hour,

$$S = \frac{34,800 \times 125}{33,000 \times 60} = 140\# \text{ steam per hour.}$$

## Steam Condenser.




Steam to be condensed per hr. = 1900#
Latent of steam = 965.6 B.T.U.
Water inlet to condenser = 84°F.
" outlet from " = 138°F.

Mean water temp. = $\frac{84+138}{2}$ = 111°F.

Temp. steam in condenser = 212 F. (atmos. pressure)

Mean diff. temp. water & steam = 212-111 = 101° F.
K, for transfer water to steam thru tubes = 100 B.T.U. per sq.ft.per hr. per degree difference temperature.

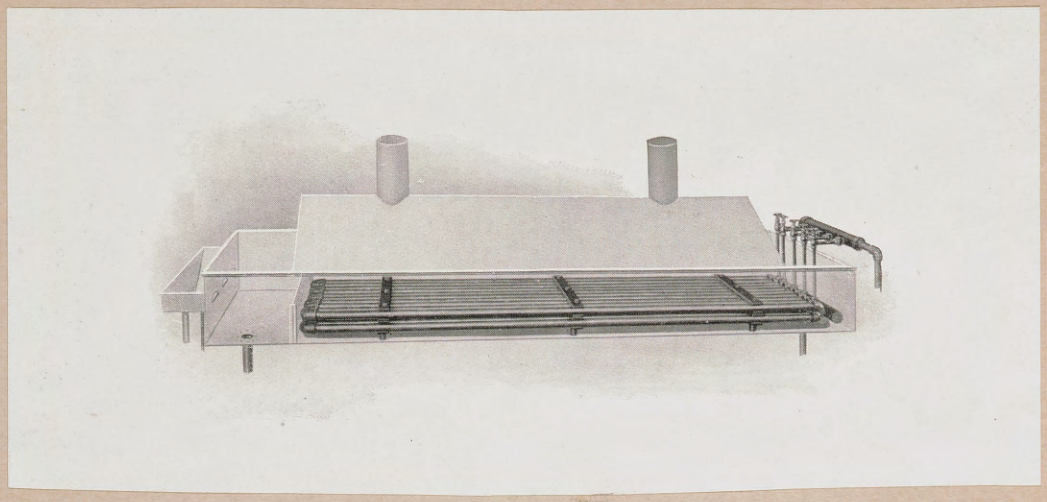
Amt. condenser surf.required =

$$= \frac{\text{lbs.to be condensed x B.T.U. to be extracted}}{\text{diff, temp. x K}}$$

$$= \frac{1900 \times 965.6}{101 \times 100} = 185 \text{ sq.ft.-Use 41-2" tubes (outside)}$$

Internal diameter of shell = 21" length 9' (about)

## Reboiler.



Use a reboiler made by the York Mfg. Co. This combimes reboiling and skimming by allowing an overflow at one end so that the foam and impurities are allowed to pass over this and thence be drained off.

Let
T= temp. range of water to be reboiled.
= 212-180= 32°
W= wt. of water to be reboiled= 1900#.
X= factor due to radiation losses= 2.

We have the steam consumption

$$S = \frac{W \times T \times X}{970.4} = \frac{1900 \times 32 \times 2}{970.4} = 126\#.$$

Water to be reboiled per hr. = 1900#
Water entering reboiler = 180°F.(assumed)
Water heated 212°F. (boiling point)
Factor for radiation losses= 2 (assumed)
K, transfer steam to water= 100 B.T.U. per hr. per sq.ft. per degree diff. temp.
Temp. steam entering reboiler= 323°F. at 80# press.
" " leaving " = 212°F. at atmos. press.

Mean temp. difference= 257.5-201. = 56.5°

$$\text{Amt. heating surface required} = \frac{\text{lbs. water to be heated} \times \text{B.T.U. to be added} \times \text{radiation factor}}{K \times \text{difference in temperature}}$$

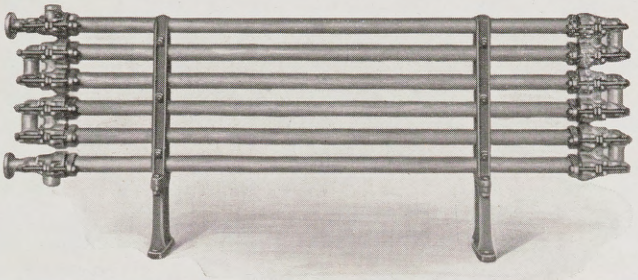
$$= \frac{1900 \times (212-180) X^2}{100 \times 56.5} = 21.5 \text{ sq. ft.}$$

21.5 sq ft. = 21.5 x 2.9 = 62.4 lineal ft. 1"pipe.

In practice, 75 ft. 1" pipe is used.

## Cooler.

Double Pipe Type



Distilled water enters at 200° F.
" " leaves at 80° F.

Mean temp. distilled water: $\frac{200 \quad 80}{2}$ : 140° F.

Cooling water enters at 70° F.
" " leaves at 138° F.

Mean temp. cooling water: $\frac{70 \quad 138}{2}$ : 104° F.

Cooling surface required =

$$\frac{\text{Heat to be extracted per hour}}{\text{Temp. diff. x rate of flow of heat per sq.ft. per hr. per degree difference.}}$$

Heat to be extracted = lbs. water to be used per hr. x temp.range

= 1750 x (200-80) = 210,000 B.T.U. per hour.

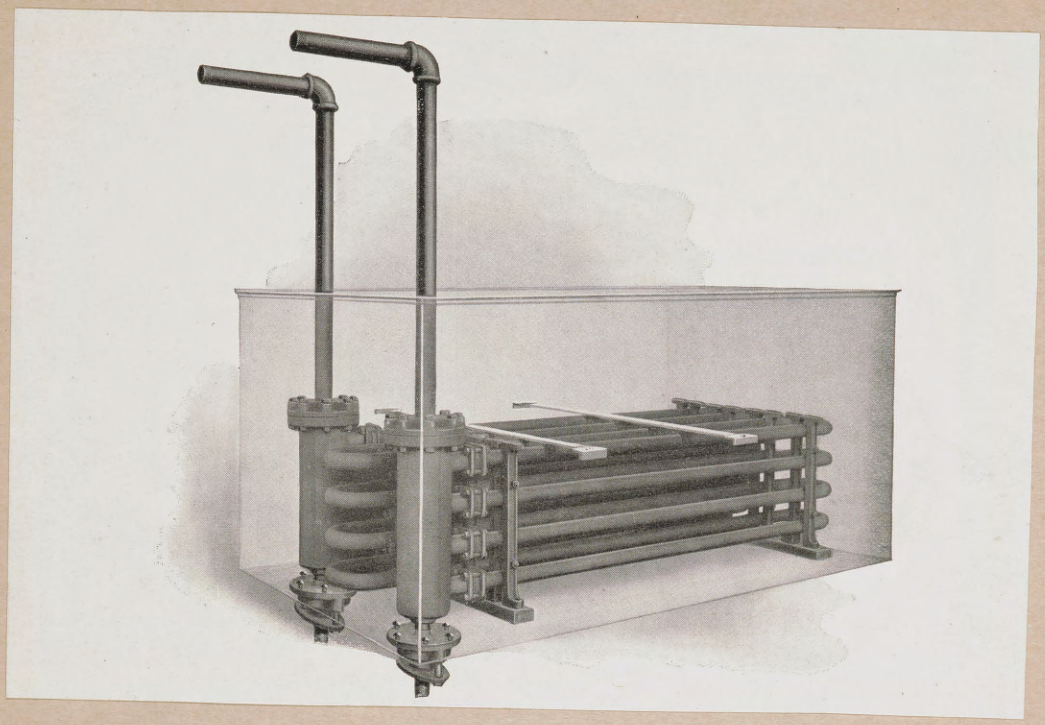
K is assumed at 100 B.T.U. per sq.ft. per degree per hour.

Therefore, Surface S = $\frac{210,000}{(140-104) \times 100}$ = 58.3 sq.ft. cooling surf.

58.3 x 2.3 = 134 lineal ft. 1¼"(inner) pipe.

Use 1, 8 pipe high cooler about 17'0" effective length.

## Storage Tank or Pre-Cooler.



Should be large enough for about 4 hrs. supply.
4hrs. supply = 4 x 1750 = 7000# water.

This is equivalent to $\frac{7000}{62.5}$ = 112 cu.ft.

Use tank 3ft. x 3ft.9in. and 10ft, high.

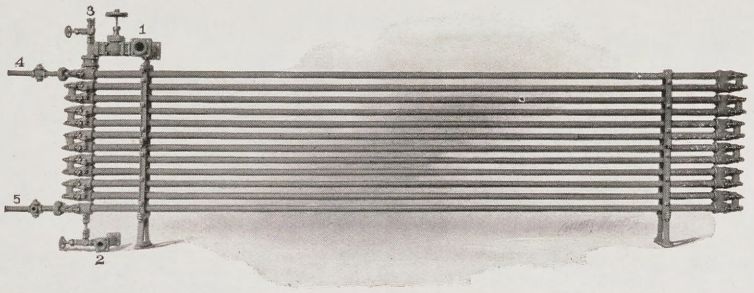
To find cooling surface required-
Water enters at 80°F.
" leaves at 40°F.
Temp. range, water = 80-40 = 40°
Heat to be extracted per hr. = 1750 x 40 = 70,000 B.T.U.
Ammonia in coils at 15.67# press. & 0°F. temp.
K is assumed 10 B.T.U. per sq.ft. per degree diff. per hr.
But surface

$$S = \frac{\text{Heat to be extracted}}{K \times \text{temperature diff.}} = \frac{70,000}{10 \times (\frac{40+80}{2} - 0)} = 116.8 \text{ sq.ft.}$$

This is equivalent to 116.8 x 2.3: 268 lineal ft. $1\frac{1}{4}$" pipe.

## Ammonia Condenser.

"York" Counter Current Type




Use a double pipe Ammonia Condenser, since this is found to be the condenser in most common use in plants of this size.

Assume 10.5 sq.ft. per ton refrigeration for cooling surface in accordance with American practice.

But total refrigeration tonnage= 20 x 1.70= 34

Total cooling surface= 34 x 10.5= 362 sq.ft.

Use condenser with $1\frac{1}{4}$" inner pipe x 2" outer pipe.

2.3 lineal ft. of $1\frac{1}{4}$"pipe contains 1 sq.ft. of surface.

Hence, total length= 2.3 x 362= 833 lineal ft.

Use 4 stands of pipes, 12 pipes high,with 17ft.4in. effective length of pipes.

Overall length 19 ft.

## Accumulator (Shell & Coil Type)

Ammonia Temp. inlet 84° F.
" in coil 0° F.
Mean 42° F.

Outside ammonia at 15.67# = 0°F
Mean difference 42°F

Assume K = 100 B.T.U. per sq. ft. per hr. per degree diff.

Ammonia used per min. per ton = 3.74 cu.ft.

Ammonia used per hour = 3.74 x 34 x 60 = 7638.0 cu.ft.

Wt. ammonia used per hour = $\frac{7638.0}{9.36}$ = 815.4 lbs.

Radiation factor = 2

Specific heat ammonia = 1.1

Hence, surface required

$$S = \frac{815.4 \times 84 \times 1.1 \times 2}{42 \times 100}$$

= 35.8 sq. ft.

= 35.8 x 2.3 = 82.4 lineal ft. 1 1/4" pipe in coil

## Freezing Tank.
## Flooded, or wet, surface exp. coils.

Heat to be extracted

| | | |
|---|---|---|
| 1750 x (40-32) | = | 14,000 B.T.U. |
| 1750 x 144 | = | 2,520,000 B.T.U. |
| 1750 x (32-15) | = | 14,850 B.T.U. |
| Total | | 2,548,850 B.T.U. |
| Rad. losses = 15% = | | 386,150 B.T.U. |
| Aggregate | | 2,935,000 B.T.U. |

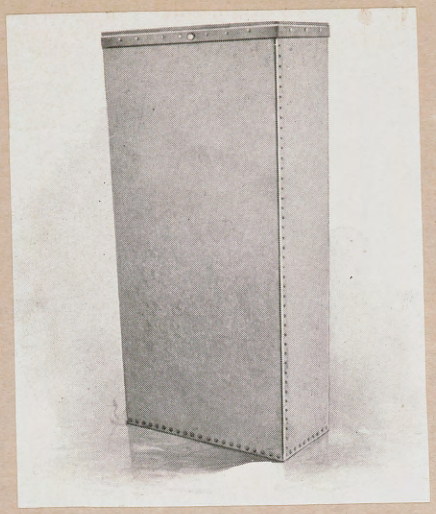
Take K = 100 B.T.U. per degree per sq.ft. per hr. at 15.67# & 0°

Mean diff. temp. = 15 -0 = 15°

$$\frac{2,935,000}{100 \times 15} = 1956 \times 2.3$$

= 4500 lineal ft. 1¼" pipe.

## Freezing Cans.



Weight of block: 200# net- 210# gross each.

48 hrs. freezing time on account flooded system.
Use 40° water, 15° brine, 0° ammonia.

Size can $11\frac{1}{2}$"x $22\frac{1}{2}$" (top)
$10\frac{1}{2}$"x $21\frac{1}{2}$" (bottom)
32"long (inside)
33" " (outside)
Ice block 28" long.

No. cans =

$$\frac{\text{wt. ice per day x no. days to freeze}}{\text{wt. block of ice.}}$$

$$= \frac{40{,}000 \times 2}{200} = 400 \text{ cans} = \text{total number.}$$

## Agitator.

Use 1 propellor type agitator as designed by G. Braunngart Jr., for the York Mfg. Co. This agitator is 1½ h.p.

The steam sonsumption of engine to drive this is assumed at 100# steam per horse power hour.

Hence, steam sonsumption

$S = 100 \times 1\frac{1}{2} = 150$# steam per hour.

Agitator designed by Geo. Braunngart Jr.,
and
Built by the York Manufacturing Co.
(See drawing on next page)

D = diam. 11.75" = .98'

L = width 3" = .25'

H = Hub 2.75":

$\phi$ = angle at periphery 60° ( cos = .5, cot = .57735

R = revolutions = 300 per min.

P = Pitch = $\pi D \cot \phi$ = 21.3" = 1.775'

V = velocity of stream from propellor = PR = 532 ft. per min.

A = area " " : area D - area H = 102.5 sq.in. = .7125 sq.ft.

A.V. = 379 cu.ft. of water displaced per min. = 2835 gallons.

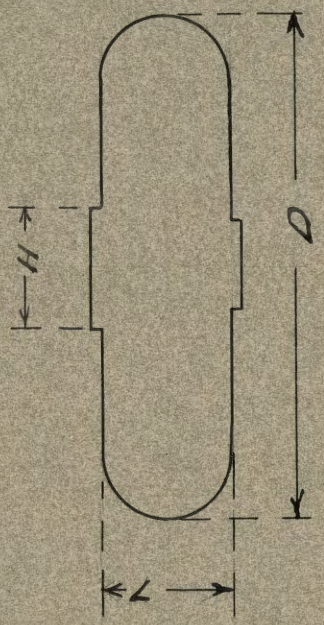
E = Effieiency 70%. S = app. Slip = 8%

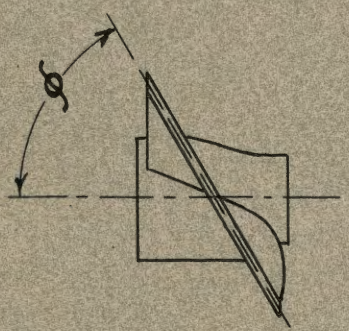
$$\begin{array}{r} 2835 \\ \times .7 \\ \hline = 1984.5 \end{array}$$

1984.5 x .92 = 1825 gallons actual.

YORK STANDARD

PROPELLOR AGITATOR



# PLAN VIEW OF
# TWENTY-TON DISTILLED-WATER
# ICE-PLANT

Designed by A.M. Carroll Jr.

Scale- ¼" = 1'

