The Refrigerating Engineer's Pocket Manual

THE REFRIGERATING ENGINEER'S POCKET MANUAL

An Indispensable Companion for Every Engineer and Student Interested in Mechanical Refrigeration

By OSWALD GUETH, M. E.
Member Am. Soc'y Refr. Eng'rs

NEW YORK:
1908

PREFACE

When the author decided to christen his book a "Pocket Manual" he was moved to do so by the words of Kent, that "every engineer should make his own pocketbook." Unfortunately not every engineer has the opportunity or ability to gather useful information without paying dearly for it.

This "long-felt want" is intended to be filled by the "Pocket Manual," a digest of the rules and data of every branch of mechanical refrigeration, embodying the opinions of the foremost men in the field, together with the practical experience of the author, a receptacle for further research and enlargement, a pocketbook in the very sense of the word, which the author trusts will soon find its way into the pocket of every progressive refrigerating engineer.

CONTENTS

Part I.—Principles and Properties.

Part II.—Refrigerating Machinery.

Part III.—Applications of Mechanical Refrigeration.

CONTENTS.

Part IV.—Operation of Compression Plant.

Part V.—The Steam Plant.

Topical Index.

PART I—PRINCIPLES AND PROPERTIES

Thermodynamics

A "British Thermal Unit"

is the heat necessary to raise one pound of water 1° F. at temperature of greatest density which is 39° to 40°.

In mechanical energy or work, a heat unit is equivalent to raising a weight of one pound to a height of 778 feet or 778 pounds to a height of one foot. The *mechanical equivalent of heat* then is 778 foot-pounds.

"Sensible Heat."

is that which is measured by a thermometer or is apparent in change of temperature, and for ordinary calculation each degree that water is heated may be considered one unit of heat for each pound of water, so that the weight of water multiplied by the increase of temperature equals the heat units absorbed.

"Latent Heat."

is that which is absorbed by a body in causing change of structure without increase of temperature. One pound of ice with a temperature of 32°, when melted will give one pound of water at a temperature of 32°, but to melt the ice heat is absorbed; this heat does not increase the temperature, although 142 units are necessary. Water boils at a temperature of 212°. Each pound of water requires 966 units of heat to convert it into steam; the 212° is sensible heat, the 966° latent heat, these added together give the total heat of steam when water is evaporated in an open vessel = 1178 units sufficient to heat 1178 pounds of water 1°.

When water is evaporated under pressure the sensible heat increases while the latent heat decreases. At 100 pounds pressure the boiling water has a temperature of 338°, the latent heat is 879, the total heat 1217 units.

"Specific Heat."

The ratio of heat required to raise the temperature of a given substance one degree to that required to raise the temperature of the same weight of water one degree (from 39.1° Fahr., the temperature of maximum density) is called the *specific heat* of the substance.

Thermodynamic Laws.

The following laws relating to a perfect gas may be safely applied to all gases:

A. The pressure varies inversely as the volume when the *temperature* is constant (Boyle).

$$\frac{V}{V'} = \frac{P'}{P} \qquad VP = \text{Constant.}$$

B. The pressure varies directly as the absolute temperature when the *volume* is constant (Charles).

$$\frac{P}{P'} = \frac{T + 461}{T' + 461}$$

C. The volume varies directly as the absolute temperature when the *pressure* is constant.

$$\frac{V}{V'} = \frac{T + 461}{T' + 461}$$

D. The product of the pressure and volume varies directly as the absolute temperature.

$$\frac{P\ V}{P'\ V'} = \frac{T + 461}{T' + 461}, \text{ or } \frac{P}{P'} = \frac{V'}{V}\ \frac{(T + 461)}{(T' + 461)}$$

Taking the volume of one pound of air at 14.7 lbs. abs. press. and at 32° = 12,387 cb ft., absolute temp. = 32° + 461 = 493°.

$$\frac{12.387 \times 14.7}{493} = .36935 \text{ or } \frac{1}{2{,}7074}$$

This fraction is a constant "a" which when multiplied by the weight and temperature of the gas, and divided by the pressure will give the volume.

$$V\ P = a\ (T + 461)$$

Expansion and Compression.

Under the first law of thermodynamics V P is a constant, that is, the curve which represents the variation of the pressure throughout the stroke of a piston, is a hyperbola and the operation is termed "isothermal" compression or expansion, the curve of equal temperatures.

Under the fourth rule D we have to add to the pressures at every successive stage during compression the heat units which are equivalent to such work, and we obtain instead an isothermal compression an "adiabatic" compression, and instead of V P being constant, V P is raised to such power as is appropriate to the particular gas in question. In the case of ammonia the pressure varies inversely as the volume raised to the 1.298 power

$$\frac{P'}{P} = \left(\frac{V}{V'}\right)^{1.298}$$

(See tables I and II, page 117, by Voorheis.)

SPECIFIC HEAT OF VARIOUS SUBSTANCES.

SOLIDS.

Antimony	0.0508	Steel (soft)	0.1165
Copper	0.0951	Steel (hard)	0.1175
Gold	0.0324	Zinc	0.0956
Wrought iron	0.1138	Brass	0.0939
Glass	0.1937	Ice	0.5040
Cast iron	0.1298	Sulphur	0.2026
Lead	0.0314	Charcoal	0.2410
Platinum	0.0324	Alumina	0.1970
Silver	0.0570	Phosphorus	0.1887
Tin	0.0562		

LIQUIDS.

Water	1.0000	Mercury	0.0333
Lead (melted)	0.0402	Alcohol (absolute)	0.7000
Sulphur "	0.2340	Fusel oil	0.5640
Bismuth "	0.0308	Benzine	0.4500
Tin "	0.0637	Ether	0.5034
Sulphuric acid	0.3350		

GASES.

	Constant Pressure.	Constant Volume.
Air	0.23751	0.16847
Oxygen	0.21751	0.15507
Hydrogen	3.40900	2.41226
Nitrogen	0.24380	0.17273
Superheated steam	0.4805	0.346
Carbonic acid	0.217	0.1535
Olefiant Gas (CH_2)	0.404	0.173
Carbonic oxide	0.2479	0.1758
Ammonia	0.508	0.299
Ether	0.4797	0.3411
Alcohol	0.4534	0.3200
Acetic acid	0.4125	......
Chloroform	0.1567	......

SPECIFIC HEAT OF COLD STORAGE GOODS.

Substance.	Composition. Water.	Composition. Solids.	Specific Heat above Freezing in Heat Units.	Specific Heat below Freezing in Heat Units.	Latent Heat of Freezing in Heat Units.	Substance.	Composition. Water.	Composition. Solids.	Specific Heat above Freezing in Heat Units.	Specific Heat below Freezing in Heat Units	Latent Heat of Freezing in Heat Units.
Beef, lean	72.00	28.00	0.77	0.41	102	Cream ...	59.25	30.75	0.68	0.38	84
Beef, fat..	51.00	49.00	.60	.34	72	Milk. ..	87.50	12.50	.90	.47	124
Veal.....	63.00	37.00	.70	.39	90	Oysters...	80.38	19.62	.84	.44	114
Pork, fat.	39.00	61.00	.51	.30	55	Whitefish..	78.00	22.00	.82	.43	111
Eggs.....	70.00	30.00	.76	.40	100	Eels	62.07	37.93	.69	.38	88
Potatoes..	74.00	26.00	.80	.42	105	Lobster ..	76.62	23.38	.81	.42	108
Cabbage..	91.00	9.00	.93	.48	129	Pigeon ...	72.40	27.60	.78	.41	102
Carrots..	83.00	17.00	.87	.45	118	Chicken..	73.70	26.30	.80	.42	105

The figures in the last column, showing the latent heat of freezing, have been obtained by multiplying the latent heat of freezing water, which is 142 heat units, by the per cent. of water contained in the different materials considered, for as the solid constituents remain in their original condition, only the liquid or watery portion of these materials is concerned in the solidification or freezing of them.

THERMOMETER SCALES.

Cent.	Réau.	Fahr.	Cent.	Réau.	Fahr.	Cent.	Réau.	Fahr.
—40	—32.0	—40.0	21	16.8	69.8	62	49.6	143.6
—38	—30.4	—36.4	22	17.6	71.6	63	50.4	145.4
—36	—28.8	—32.8	23	18.4	73.4	64	51.2	147.2
—34	—27.2	—29.2	24	19.2	75.2	65	52.0	149.0
—32	—25.6	—25.6	25	20.0	77.0	66	52.8	150.8
—30	—24.0	—22.0	26	20.8	78.8	67	53.6	152.6
—28	—22.4	—18.4	27	21.6	80.6	68	54.4	154.4
—26	—20.8	—14.8	28	22.4	82.4	69	55.2	156.2
—24	—19.2	—11.2	29	23.2	84.2	70	56.0	158.0
—22	—17.6	—7.6	30	24.0	86.0	71	56.8	159.8
—20	—16.0	—4.0	31	24.8	87.8	72	57.6	161.6
—18	—14.4	—0.4	32	25.6	89.6	73	58.4	163.4
—16	—12.8	+3.2	33	26.4	91.4	74	59.2	165.2
—14	—11.2	6.8	34	27.2	93.2	75	60.0	167.0
—12	—9.6	10.4	35	28.0	95.0	76	60.8	168.8
—10	—8.0	14.0	36	28.8	96.8	77	61.6	170.6
—8	—6.4	17.6	37	29.6	98.6	78	62.4	172.4
—6	—4.8	21.2	38	30.4	100.4	79	63.2	174.2
—4	—3.2	24.8	39	31.2	102.2	80	64.0	176.0
—2	—1.6	28.4	40	32.0	104.0	81	64.8	177.8
—0	0.0	32.0	41	32.8	105.8	82	65.6	179.6
+1	+0.8	33.8	42	33.6	107.6	83	66.4	181.4
2	1.6	35.6	43	34.4	109.4	84	67.2	183.2
3	2.4	37.4	44	35.2	111.2	85	68.0	185.0
4	3.2	39.2	45	36.0	113.0	86	68.8	186.8
5	4.0	41.0	46	36.8	114.8	87	69.6	188.6
6	4.8	42.8	47	37.6	116.6	88	70.4	190.4
7	5.6	44.6	48	38.4	118.4	89	71.2	192.2
8	6.4	46.4	49	39.2	120.2	90	72.0	194.0
9	7.2	48.2	50	40.0	122.0	91	72.8	195.8
10	8.0	50.0	51	40.8	123.8	92	73.6	197.6
11	8.8	51.8	52	41.6	125.6	93	74.4	199.4
12	9.6	53.6	53	42.4	127.4	94	75.2	201.2
13	10.4	55.5	54	43.2	129.2	95	76.0	203.0
14	11.2	57.2	55	44.0	131.0	96	76.8	204.8
15	12.0	59.0	56	44.8	132.8	97	77.6	206.6
16	12.8	60.8	57	45.6	134.6	98	78.4	208.4
17	13.6	62.6	58	46.4	136.4	99	79.2	210.2
18	14.4	64.4	59	47.2	138.2	100	80.0	212.0
19	15.2	66.2	60	48.0	140.0			
20	16.0	68.0	61	48.8	141.8			

The "Absolute Zero" of temperature denotes that condition of matter at which heat ceases to exist. At this point a body would be wholly deprived of heat and a gas would exert no pressure.

The Absolute Zero on the Fahrenheit scale is about 461° below Zero.
" " " " Centigrade " " 274° " "
" " " " Reamur " " 219° " "

Water

Water (H_2O) is a combination of one atom of oxygen and two atoms of hydrogen.

A gallon of water (U. S. standard) weighs 8 1-3 lbs. and contains 231 cu. inches. A cu. ft. of water weighs 62.4 lbs. and contains 1728 cubic inches, or 7.48 gallons.

A gallon of water evaporated at atmospheric pressure will produce about 200 cu. ft. of steam.

A gallon of water evaporated under a 27-inch vacuum will produce about 2000 cu. ft. of vapor.

Water containing substances in solution has its boiling point raised.

Pure water is of the first importance in an ice factory both for feeding boilers and ice-making.

Water is the greatest natural solvent known, hence is rarely found to be pure. It is capable of absorbing every gas and vapor with which it comes in contact.

Solids in Water.

Animal life, organic matters, such as sewage, decayed vegetable and animal matter, poisonous metals, magnesia, lime, carbonates, sulphates, alkalies, earthy salts, chlorine and bromide combinations, etc., are found in quantity.

Rules for Testing Water.

Water turning *blue litmus paper red* before boiling, which after boiling will not do so; and if the blue color can be restored by warming, *then it is varbonated* (containing carbonic acid).

If it has a sickening odor, giving a black precipitate with acetate of lead, *it is sulphurous* (containing sulphuretted hydrogen).

If it gives a blue precipitate with yellow or red prussiate of potash by adding a few drops of hydrochloric acid, *it is chalybeate* (carbonate of iron).

If it restores blue color to litmus paper after boiling, *it is alkaline.*

If it has none of the above properties in a marked degree and leaves a large residue after boiling, *it is saline water* (containing salts).

Testing by Re-Agents.

Water is not pure if it becomes turbid or opaque by the use of the following agents:

Baryta water indicates the presence of carbonic acid.

Chloride of barium indicates the presence of sulphates.

Nitrate of silver indicates the presence of chlorides.

Oxalate of ammonia indicates the presence of lime salts.

Sulphide of hydrogen slightly acid indicates the presence of either antimony, arsenic, tin, copper, gold, platinum, mercury, silver, lead, bismuth or cadmium.

Sulphide of ammonia, alkaloid by ammonia, indicates the presence of nickel, cobalt, manganese, iron, zinc, alumina or chromium.

Chloride of mercury or gold, or sulphate of zinc, indicates the presence of organic matter.

Water may be found which will pass the tests above described and yet be unfit for use, or, as it is commonly called, "not potable." Distillation is the only method to produce purity in water, whereby all deleterious acids, gases, organic and mineral, and disease germs can be eliminated. The solid and organic matter held in suspense may be removed by filtration.

Condensing Water for Machinery.

Water for use in the ammonia condensing apparatus is preferred when taken from springs or deep wells, for the reason that water from below the surface is much colder than surface water, hence much less is required. Water from considerable depths is almost constant in temperature, and is generally from 50 to 56 degrees the year round, while water from rivers, ponds and streams ranges from 32 degrees in winter to 95 degrees in midsummer. The colder the water used in the condenser, the less power it requires to drive the machinery.

For refrigerating machines allow about 1½ gallons per ton refrigerating capacity, and on ice plants 3 to 4 gallons per ton, dependent upon the temperature.

Air

Air is a mechanical mixture of 20.7 parts oxygen and 79.3 parts nitrogen by volume.

The weight of pure air at 32° F. and atmospheric pressure is 0.081 lbs. per cubic foot. Volume of 1 lb. = 12,387 cu. ft. Air expands 1-491.2 of its volume at 32° F. for every increase of 1° F.

At the sea-level its pressure is 14.7 lbs. per sq. inch. At one mile above 12.02, at 2 miles 9.8 lbs. Roughly, the pressure decreases ½ lb. for every 1,000 feet.

Moisture in Atmosphere.

MOISTURE CONTAINED IN ONE CUB. FT. OF SATURATED AIR.

Temp.	Grains.	Temp.	Grains.	Temp.	Grains.
4	0.5	26	1.69	46	3.6
0	0.55	28	1.83	48	3.85
5	0.73	30	1.97	50	4.12
12	0.91	32	2.13	52	4.4
14	1.05	34	2.32	62	6.17
16	1.14	36	2.51	72	8.55
18	1.23	38	2.7	82	11.67
20	1.32	40	2.89	92	15.75
22	1.41	42	3.08	102	21
24	1.55	44	3.34	112	27.6

RELATIVE HUMIDITY, PER CENT.

Difference between the Dry and Wet Thermometers, Deg. F. — Relative Humidity, Saturation being 100. (Barometer = 30 ins.)

Dry Thermometer, Deg. F.	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	26	28	30
32	89	79	69	59	49	39	30	20	11	2																	
40	92	83	75	68	60	52	45	37	29	23	15	7	0														
50	93	87	80	74	67	61	55	49	43	38	32	27	21	16	11	5	0										
60	94	89	83	78	73	68	63	58	53	48	43	39	34	30	26	21	17	13	9	5	1						
70	95	90	86	81	77	72	68	64	59	55	51	48	44	40	36	33	29	25	22	19	15	12	9	6			
80	96	91	87	83	79	75	72	68	64	61	57	54	50	47	44	41	38	35	32	29	26	23	20	18	12	7	
90	96	92	89	85	81	78	74	71	68	65	61	58	55	52	49	47	44	41	39	36	34	31	29	26	22	17	13
100	96	93	89	86	83	80	77	73	70	68	65	62	59	56	54	51	49	46	44	41	39	37	35	33	28	24	21
110	97	93	90	87	84	81	78	75	73	70	67	65	62	60	57	55	52	50	48	46	44	42	40	38	34	30	26
120	97	94	91	88	85	82	80	77	74	72	69	67	65	62	60	58	55	53	51	49	47	45	43	41	38	34	31
140	97	95	92	89	87	84	82	79	77	75	73	70	68	66	64	62	60	58	56	54	53	51	49	47	44	41	38

The relative humidity of the air is the percentage of moisture contained in it as compared with the amount it is capable of holding at the same temperature. It is determined by the use of the dry and wet bulb thermometer.

Equation of Pipes.

At the same velocity of flow the volume delivered by two pipes of different sizes is proportional to the squares of their diameters; thus, one 4-inch pipe will deliver the same volume as four 2-inch pipes.

With the same head, however, the velocity is less in the smaller pipe, and the volume delivered varies about as the square root of the fifth power. The following table has been calculated on this basis. Thus, one 4-inch pipe is equal to 5.7 pipes of 2-inch diameter.

Diam. in.	1	2	3	4	5	6	7	8	9	10	12	14	16	18	20	24
2	5.7	1														
3	15.6	2.8	1													
4	32	5.7	2.1	1												
5	55.9	9.9	3.6	1.7	1											
6	88.2	15.6	5 7	2.8	1.6	1										
7	130	22.9	8.3	4.1	2.3	1.5	1									
8	181	32	11.7	5.7	3.2	2.1	1.4	1								
9	243	43.	15.6	7.6	4.3	2.8	1.9	1.3	1							
10	316	55.9	20.3	9.9	5.7	3.6	2.4	1.7	1.3	1						
11	401	70.9	25.7	12.5	7.2	4.6	3.1	2.2	1.7	1.3						
12	499	88.2	32	15.6	8.9	5.7	3.8	2.8	2.1	1.6	1					
13	609	108	39.1	19	10.9	7.1	4.7	3.4	2.5	1.9	1.2					
14	733	130	47	22.9	13.1	8.3	5.7	4.1	3.0	2.3	1.5	1				
15	871	154	55.9	27.2	15.6	9.9	6.7	4.8	3.6	2 8	1.7	1.2				
16	..	181	65.7	32	18.3	11.7	7.9	5.7	4.2	3.2	2.1	1.4	1			
17	...	211	76.4	37.2	21.3	13.5	9.2	6.6	4.9	3.8	2.4	1.6	1.2			
18	...	243	88.2	43	24.6	15.6	10 6	7.6	5.7	4.3	2.8	1.9	1.3	1		
19	...	278	101	49.1	28.1	17.8	12.1	8.7	6.5	5	3.2	2.1	1.5	1.1		
20	....	316	115	55.9	32	20.3	13.8	9.9	7.4	5.7	3.6	2.4	1.7	1.3	1	
22	....	401	146	70.9	40.6	25.7	17.5	12.5	9.3	7.2	4.6	3.1	2.2	1.7	1.3	
24	....	499	181	88.2	50.5	32.	21.8	15.6	11.6	8.9	5.7	3.8	2.8	2.1	1.6	1
26	....	609	221	108	61.7	39.1	26.6	19.	14.2	10.9	7.1	4.7	3.4	2.5	1.9	1.2
28	....	733	266	130	74.2	47	32	22.9	17.1	13.1	8.3	5.7	4.1	3	2.3	1.5
30	...	871	316	154	88.2	55.9	38	27.2	20.3	15.6	9.9	6.7	4.8	3.6	2.8	1.7
36	...	...	499	243	130	88.2	60	43	32	24.6	15.6	10.6	7.6	5.7	4.3	2.8
42	...	...	733	357	205	130	88.2	63.2	47	36.2	19	15.6	11.2	8.3	6.4	4.1
48	...	...	...	499	286	181	123	88.2	62.7	50.5	32	21.8	15.6	11.6	8.9	5.7
54	...	...	...	670	383	243	165	118	88.2	67.8	43	29.2	20.9	15.6	12	7.6
60	...	...	...	871	499	316	215	154	115	88.2	55.9	38	27.2	20.3	15.6	9.9

Table of Standard Steam, Gas or Brine Pipe.

Inside Diameter, Inches	Outside Diameter, Inches	External Circumference, Inches	Length of Pipe per Square Foot of Outside Surface, Feet	Internal Area, Inches	External Area, Inches	Length of Pipe Containing One Cubic Foot, Feet	Weight per Foot of Length, Pounds	Number of Threads per Inch of Screw	Contents in Gallons* per Foot	Weight of Water per Foot of Length, Pounds
1/8	.40	1.272	9.44	.0572	.129	2500.	.24	27	.0006	.005
1/4	.54	1.696	7 075	.1041	.229	1385.	.42	18	.0026	021
3/8	.67	2.121	5.657	.1916	.358	751.5	.56	18	.0057	.047
1/2	.84	2.652	4 55	.3048	.554	472.4	.84	14	.0102	.085
3/4	1.05	3.299	3.637	.533	.866	270.	1.12	14	.0230	.190
1	1.31	4.134	2.903	.862	1.357	166.9	1.67	11 1/2	.0408	.349
1 1/4	1.66	5.215	2.301	1.496	2.164	96.25	2.25	11 1/2	.0638	.527
1 1/2	1.9	5.969	2.01	2.038	2.835	70.65	2.69	11 1/2	.0918	.760
2	2.37	7.461	1.611	3.355	4.430	42.9	3.66	11 1/2	.1632	1.356
2 1/2	2.87	9 032	1.328	4.783	6.491	30.11	5.77	8	.2550	2.116
3	3.5	10.996	1.091	7.388	9.621	19.49	7 54	8	.3673	3.049
3 1/2	4.	12.566	.955	9.887	12.566	14.56	9.05	8	.4998	4.155
4	4.5	14.137	.849	12.73	15.904	11.31	10 72	8	.6528	5.405
4 1/2	5.	15.708	.765	15.96	19.635	9.03	12.34	8	.8263	6.851
5	5.56	17 475	.69	19.99	24.299	7.20	14.56	8	1 020	8.500
6	6.62	20.813	.577	28.889	34.471	4.98	18.76	8	1.469	12.312
7	7 62	23.954	.505	38.737	45.663	3.72	23.41	8	1 999	16.662
8	8.62	27.096	.444	50.039	58.426	2.88	28.34	8	2.611	21 750
9	9 68	30.433	.394	63.633	73.715	2.26	34.67	8	3.300	27 500
10	10.75	33.772	355	78.838	90.762	1.80	40.64	8	4.081	34.000

* The standard U. S. gallon of 231 cubic inches

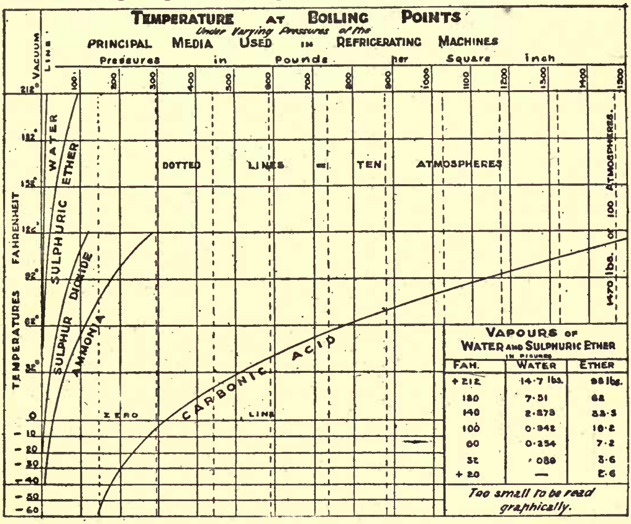
Refrigerating Media

The *efficiency* of a gas depends on three properties:

First, a *low boiling point,* upon which depends the degree of cold that can be produced.

Second, a *high latent heat of evaporation,* upon which depends the total number of heat units, which will be abstracted by the evaporation of a given weight of the medium.

The following diagrams are reproduced from N. Selfe.



Third, a *low specific heat,* upon which depends the amount of refrigeration produced which can be actually utilized.

Ammonia.

Ammonia, H_3N, is composed of one part of nitrogen and three parts hydrogen. It can be obtained from the air, from sal-ammoniac, nitrogenous constituents of plants and animals by process of distillation—as a matter of fact, there are very few substances free from it. At the present day almost all the sal-ammoniac and ammonia liquors are prepared from ammoniacal liquid, a by-product obtained in the manufacture of coal gas.

Pure ammonia liquid is colorless, having a peculiar alkaline odor and caustic taste. It turns red litmus paper blue.

Its boiling point depends on its purity, and is about 28 6-10 degrees below zero at atmospheric pressure.

Compared with water, its weight or specific gravity at 32 degrees F. is about 5-8 of water, or 0.6364.

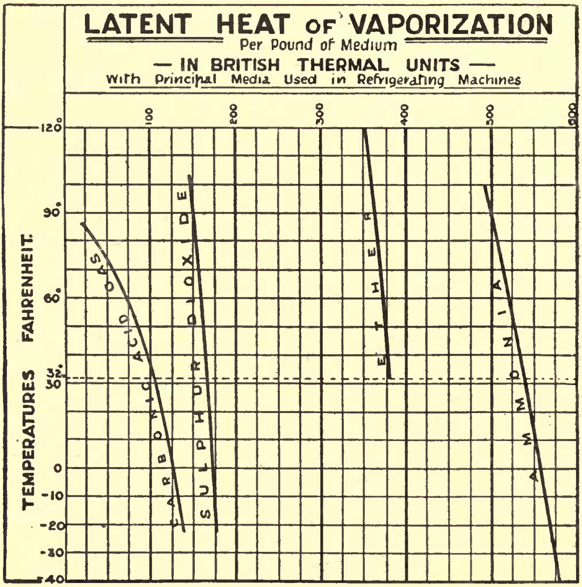
One cubic foot of liquid ammonia, weighing 39.73 pounds, one gallon weighs 5 and 3-10 pounds, one pound of the liquid at 32°, will occupy 21.017 cubic feet of space when evaporated at atmospheric pressure.

Its latent heat of evaporation is not far from 560 thermal units at 32 degrees, at which temperature one pound of liquid, evaporated under a pressure of fifteen pounds per square inch, will occupy twenty-one cubic feet.

Ammonia liquid should be pure. Its purity may be tested by the following simple methods recommended by the Frick Co. and other builders:

Testing for Water by Evaporation.

Screw into the ammonia flask a piece of bent one-quarter inch pipe, which will allow a small bottle to be placed so as to receive the discharge from it. Fill the bottle about one-third



full, and throw sample out in order to purge valve, pipe and bottle. Quickly wipe off the moisture which has accumulated on the pipe, replace the bottle and open valve gently, filling the bottle about half full. This last operation should not occupy more than one minute. Remove the bottle at once and insert in its neck a stopper with a vent hole for the escape of the gas. Procure a piece of solid iron that should weigh not less than 8 or 10 pounds, pour a little water on this and place the bottle on the wet place. The ammonia will at once begin to boil, and in warm weather will ammonia will at once begin to boil, and in warm weather will soon evaporate. If it shows any residuum, pour it out gently, counting the drops carefully. Eighteen drops are about equal to one cubic centimeter, and if the sample taken amounted to 100 cubic centimeters, you can readily approximate the percentage of the liquid remaining.

Test for Inflammable Gases.

Take a pail of water, submerge the bent end of quarter-inch pipe therein, open the valve on flask slightly, and allow a small quantity of gas to flow into the water. If inflammable gases are present, they will rise in bubbles to the surface of the water, and may be proved by igniting the bubbles by means of a lighted candle or match. As water has a strong affinity for ammonia, it will be readily absorbed, while air or other gases will show only in the form of bubbles.

Test for Specific Gravity.

The specific gravities of liquid ammonia by Beaumé scale are given in table below; by drawing off some of the liquid in a tall test tube, the Beaumé Hydrometer (light) may be inserted and the specific gravity read upon the scale. If water is present, the liquid will show a density proportionate to the percentage of the water present.

Specific gravity of pure anhydrous ammonia is .623.

Test for Boiling Point.

By inserting a special low temperature standardized chemical thermometer into liquid drawn into the 8-oz. test jar, readings can be obtained through the side of the jar without removing the instrument. Hold the thermometer in such position that only the bulb is immersed. This test will give you the boiling point of ammonia at atmospheric pressure, and it is well to know that the state of the barometer affects the temperature of the boiling point. With the barometer at 29.92 inches, the boiling point is nearly 28 6-10 degrees below zero. If the ammonia is impure, the boiling point is raised in proportion.

To Test Brine or Water for Ammonia.

"Nessler's Reagent" is used extensively. It is prepared as follows: Dissolve 17 grams of mercuric chloride in cubic centimeters of distilled water; dissolve 35 grams of potassium iodide in 100 cubic centimeters of water; stir the latter solution into the first until a red precipitate is thrown down. Then dissolve 120 grams of potassium hydrate in 200 cubic centimeters of water and allow the solution to cool, then add to the other solution, and add sufficient water to make one litre. Then add mercuric chloride solution until a precipitate forms. Let this settle and decant off a clear solution.

Keep the solution in glass stoppered blue bottles. A few drops of this solution added to a sample of brine or water will cause the brine or water to turn yellow if a small percentage of ammonia is present and turn to a full brown if the percentage of ammonia is large.

Impurities Test.

When testing ammonia for impurities, it is diluted with twice its volume of distilled water. It is then made acid with hydrochloric acid. Then to detect the presence of sulphates, add a solution of chloride of barium. If sulphates are present, a white precipitate will be formed. To detect the presence of chlorides the solution of ammonia and water is acidulated with nitric acid instead of hydrochloric, and the white precipitate is formed by the addition of a nitrate of silver solution. But if, in this case, red appears, there is evidence of organic matter.

Aqua Ammonia.

16° aqua ammonia, often called by druggists F.F.F., containing a little more than 10 per cent of pure anhydrous ammonia, 18° aqua ammonia (F.F.F.F.) containing nearly 14 per cent of anhydrous ammonia. 26° aqua ammonia ("stronger aqua ammonia") containing 29¼ per cent of pure anhydrous ammonia. This is generally used in absorption plants for the start.

PROPERTIES OF SATURATED AMMONIA.

Gauge Pressure, lbs. per Square Inch.	Absolute Pressure, lbs. per Square Inch.	Tem. Drgrees F.	Absolute Temp. Degrees F.	Latent Heat of Evaporation in Thermal Units.	Volume of 1 lb. Vapor in Cubic Feet.	Weight of One Cubic Foot of Vapor in lbs.	Volume of 1 lb. of Liquid in Cubic Feet.	Weight of 1 Cubic Foot of Liquid, in lbs.
—4.01	10.69	—40	420.66	579 67	24.38	0410	.0234	42 589
—2.39	12.31	—35	425.66	576.68	21 32	.0469	.0236	42 337
—0.57	14.13	—30	430.66	573.69	18 69	.0535	0237	42.123
+1.47	16.17	—25	435.66	570.68	16.44	.0608	0238	41 858
3.75	18 45	—20	440.66	567 67	14.51	.0690	0240	41.615
6 29	20.99	—15	445.66	564.64	12.83	.0779	.0241	41.374
9.10	23 80	—10	450.66	561.61	11.38	.0878	.0243	41 135
12.22	26 92	—5	455 66	558.56	10.12	.0988	.0244	40 900
15.67	30 37	0	460.66	555 50	9.03	.1107	.0246	40 650
19 46	34 16	+5	465 66	552.43	8.07	.1240	0247	40 404
23 64	38.34	10	470.66	549 35	7 23	.1383	.0249	40 160
28.24	42 94	15	475.66	546.26	6.49	.1541	.0250	30.920
83.25	47 95	20	480.66	543.15	5 84	.1711	.0252	39 682
38.73	53.43	25	485 66	540.03	5.27	.1897	.0253	39 432
44 72	59.42	30	490.66	536.91	4 76	.2099	.0255	39.200
51.22	65 92	35	495 66	533.78	4 31	.2318	.0256	38 940
58.29	72.99	40	500.66	530.63	3.91	2554	.0258	38 684
65.96	80.66	45	505 66	527 47	3.56	.2809	.0260	38 461
74 26	88.96	50	510.66	524.30	3 24	.3084	.0261	38 226
83.22	97.92	55	515.66	521.12	2.96	.3380	.0263	37 994
92.89	107.59	60	520.66	517 93	2.70	.3697	.0265	37 736
103.33	118.03	65	525.66	514.73	2 48	.4039	.0266	37 481
114 49	129.19	70	530.66	511.52	2 27	4401	.0268	37 230
126.52	141.22	75	535.66	508.29	2 09	4791	.0270	36 995
139.40	154.10	80	540.66	505.05	1 92	.5205	.0272	36 751
153.18	167.88	85	545.66	501 81	1 77	5649	.0273	36 509
167.92	182.62	90	550 66	498.55	1 64	.6120	.0275	36.258
183.65	198.35	95	555.66	495.29	1 51	.6622	0277	36 023
200.42	215.12	100	560.66	492 01	1 39	7153	0279	35 778
218.28	232.98	105	565.66	488.72	1.289	7757	.0281	......
237.27	251.97	110	570.66	485.42	1.203	.8312	.0283	......
258.7	272.14	115	575.66	482.41	1.121	8912	.0285	......
275.79	293.49	120	580.66	478.79	1.041	.9608	.0287	......
301.46	316.16	125	585 66	475.45	.9699	1 0310	0289	......
325.72	340 42	130	590.66	472.11	.9051	1 1048	0291	......
350.46	365.16	135	595 66	468 75	.8457	1.1824	.0293	......
377 52	392.22	140	600.66	465.39	.7910	1 2642	.0295	......
405 79	420.49	145	605.66	462.01	.7408	1 3497	0297	......
435.5	450.20	150	610 66	458 62	.6946	1 4396	.0299	......
466 84	481.54	155	615 66	455.22	.6511	1 5358	0302	......
499.70	514.50	160	620 66	451 81	.6128	1 6318	.0304	......
534 34	549.04	165	625.66	448.39	.5765	1 7344	.0306	......

STRENGTH OF AMMONIA LIQUOR.

% of ammonia by wt.	Sp. grav.	° Beaumé water 10.	° Beaumé water 0.	% of ammonia by wt.	Sp. grav.	° Beaumé water 10.	° Beaumé water 0.
0	1.000	10	0	20	0.925	21.7	11.2
1	0.993	11	1	22	0.919	22.8	12.3
2	0.986	12	2	24	0.913	23.9	13.2
4	0.979	13	3	26	0.907	24.8	14.3
6	0.972	14	4	28	0.902	25.7	15.2
8	0.966	15	5	30	0.897	26.6	16.2
10	0.960	16	6	32	0.892	27.5	17.3
12	0.953	17.1	7	34	0.888	28.4	18.2
14	0.945	18.3	8.2	36	0.884	29.3	19.1
16	0.938	19.5	9.2	38	0.880	30.2	20.0
18	0.931	20.7	10.3				

Carbonic Acid.

Carbonic anhydride, or carbonic acid as it is usually called, has the chemical designation Carbon Dioxide, CO_2, and consists of two atoms of oxygen and one atom of carbon.

The chief characteristics of the gas are absence of odor, neutrality towards materials and food products, the fact that it cannot be decomposed under pressure and its cheapness. It has a specific gravity of 1.529 (air is 1) at atmospheric pressure and becomes a liquid at 124 degrees below zero, Fahr., or 156 degrees below the freezing point at that pressure.

Atmosphere containing 8 per cent of carbonic anhydride can be inhaled without causing inconvenience or leaving any deleterious effects upon the human system. Carbonic anhydride will fall to the floor by reason of its greater specific weight, and even in the event of a serious leak occurring, the air will not become sufficiently saturated to cause any harm.

Fifteen per cent (15%) of carbonic anhydride in the atmosphere will extinguish fire.

Carbonic anhydride is artificially produced in pure form by means of combustion of chalks and magnesite, or by means of the decomposition of marble with sulphuric or nitric acid.

The so-called *Pistet fluid* is a mixture of carbonic acid and sulphur dioxide, which according to Pictet is expressed by the chemical symbol CO_4S. The pressure of this mixture at higher temperature is said to be less than the law of corresponding pressures and temperatures would indicate. According to this there would be less work required of the compressor.

Ethyl chloride (C_2H_5Cl) has been used during the last few years as a refrigerating medium, although to very little extent. Its boiling point at atmospheric pressure is about 54° F. In order, therefore, to produce cold, the machine has to work under vacuum, while the condenser pressure hardly ever exceeds 15 lbs. The gas is neutral towards metals, its critical temperature is at 365° F. It is more expensive than any of the other media, but it is claimed, that on account of the low pressure there will hardly be any loss of gas.

Methyl chloride machines are comparatively new and not in practical use to any extent so far.

Certain hydrocarbons, naphtha, gasoline,, etc., have also been experimented with as refrigerating media. All these liquids possess the same great inflammability as ether, but they are cheaper.

Acetylene (C_2H_2), the once heralded illuminating agent of the future, has also been mentioned as a possible medium. It is highly inflammable. It liquifies at 32° F. under a pressure of 48 atmospheres.

Liquid air has also been prominently spoken of as a refrigerating medium. But under present conditions its production is too expensive to render it available for ordinary refrigeration. Its usefulness is limited to produce extremely low temperatures, which may be required for special purposes in the laboratory.

Brine

Until recent years brine was made by dissolving common salt, NaCl (chloride of sodium) in water. Later chloride of magnesium was used instead. The latter was neutral to iron and did not freeze at extremely low temperatures. Later still, because of the high cost of chloride of magnesium, chloride of calcium, $CaCl_2$, having nearly the same properties as choloride of magnesium, was used either direct or in combination with chloride of magnesium.

Chloride of Sodium.

When using common salt, buy in bags, containing medium ground pure salt. Allow about three lbs. per gallon of water. Continue to dissolve the salt in the brine tank until it reaches a density of 85 to 90 degrees by salt gauge. The stronger the brine the lower temperatures can be obtained without freezing.

In making the brine it is well to use a water-tight box, say 4ft. wide, 8 ft. long, and 2 ft. high, with a perforated false bottom and compartment at end.

Locate the brine maker at a point above the brine tank. Con-

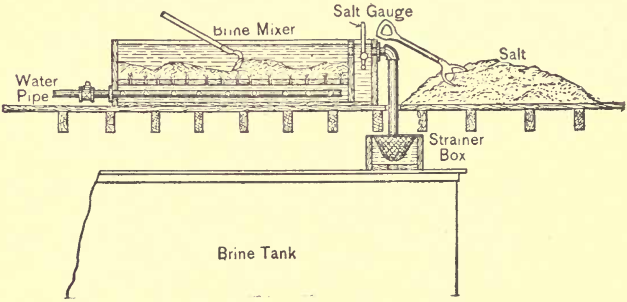


FIG. 1—METHOD OF MAKING BRINE.

nect the space under the false bottom with your water supply, extending the pipe lengthways of the box and perforated at each side to insure an equal distribution of water over the entire bottom surface, use a valve in water supply pipe. Near the top of the brine maker at end compartment, put in an overflow with large strainer to keep back the dirt and salt, and connect with this a pipe, say 3 ins. diameter, with salt catcher at bottom leading into the brine tank. Use a hoe or shovel to stir the contents. When all is ready partly fill the box with water, dump the salt from the bags on the floor alongside and shovel into brine maker, or dump direct from the bags into brine maker as fast as it will dissolve; regulate the water supply to always insure the brine being of the right strength as it runs into the brine tank: this point must be carefully noticed.

Filling the brine tank with water and attempting to dissolve the salt directly therein is not satisfactory, as quantities of salt settle on the tank bottom coils, forming a hard cake.

When desired to strengthen the brine, suspend bags of salt in the tank, the salt dissolving from the bags as fast as required, or the return brine from the pumps may be allowed to circulate through the brine maker, keeping same supplied with salt.

Chloride of Calcium.

Fused Calcium. Commercial calcium is made by melting the crystals at 400° F., thus driving off the water of crystallization, leaving the remainder 75 per cent calcium and 25 per cent water. This solution, while hot, is poured into iron drums and sealed up air tight. This calcium comes in 600 to 700 pound iron drums, which should be painted with asphalt varnish, so that they can be stored away in damp and cold rooms without danger of rusting.

When making brine, the calcium should be broken up into pieces and placed in a barrel or tank with perforated bottom. Then the water or brine should be pumped over it until the brine is of the required strength. To break up the calcium, hit the drums a number of heavy blows with a sledge hammer, the iron cover can then be removed with a cold chisel and the calcium will be found to be broken up as desired.

Heat is generated as the calcium dissolves and, if possible to do so, it will be found more convenient to dissolve the calcium when the brine is not being refrigerated. It dissolves more rapidly in warm or hot than in cold brine. Steam can be used to advantage for the rapid dissolving of large quantities of chloride of calcium.

Fluid Calcium: This is of 1400 specific gravity (weighing 11.66 pounds per gallon), and contains about 40 per cent of anhydrous chloride of calcium in solution; it is water white and clear. It is shipped in tank cars of 4,500 gallons. When diluted with an equal volume of water, it gives a solution of 1,200 specific gravity, which is strong enough for most purposes.. Calcium fluid of specific gravity of 1600 (weighing 13.32 pounds per gallon), containing up to 60 per cent of anhydrous chloride of calcium in solution crystallizes into a semi-solid mass in cool weather, and it is necessary to warm it up to 60° Fahr., which makes it rather difficult to handle during cool weather, unless steam is conveniently at hand. The 1,600 specific gravity, or 60 per cent solution, when diluted with two parts of water, gives a brine of 1,200 specific gravity.

A solution of chloride of sodium brine, twenty-five per cent by weight, is saturated and will freeze at 0° F., but will tend to separate the salt and begin to freeze at 5° F. A solution of chloride of calcium, twenty-five per cent by weight, freezes at —22° F. In can ice making the brine is usually carried at 10° to 16° F., which requires ammonia at from—5° F. to 5° F. At these temperatures salt will separate out and ice will form on the expansion coils, thereby insulating them and requiring a lower back pressure.

Chloride of calcium brine of 1.22 specific gravity has twenty-four per cent of calcium chloride by weight, or four pounds to the gallon. This brine freezes at—17° F., and in can ice making can be diluted with thirty per cent of water before it will freeze, as will a saturated salt brine solution. Chloride of calcium brine having two and one-half to three pounds to the gallon is all right for ice making. In brine tanks the salt brine freezes on the coils and insulates them, or in brine coolers freezes in the coils and breaks them. Salt brine loses in evaporation, some of the salt being carried away, while calcium brine does not.

Aside from its stability to stand lower temperatures, calcium chloride has the advantage over sodium chloride or salt brine of having absolutely no action upon iron, thus materially increasing the life of brine tanks and brine coils. While the cost of calcium chloride is somewhat greater than salt, this is offset to some extent by the fact that 25 per cent less calcium than salt is required.

Table of Calcium Brine Solution.

Deg. Baumé 60° F.	Deg. Salom-eter. 60° F.	Per Cent Calcium by Weight	Lbs. per Cu. Ft. Sol.	Lbs. per Gallon	Specific Gravity	Specific Heat	Freezing Point F.	Amm. Gauge Pressure
0	0	0	0	0	1	1	32	47.31
1	4	.943	1.25	$\frac{1}{6}$	1.007	.996	31.1	46.14
2	8	1.886	2.5	$\frac{1}{3}$	1.014	.988	30.33	45.14
3	12	2.829	3 75	$\frac{1}{2}$	1.021	.98	29.48	44.06
4	16	3.772	5	$\frac{2}{3}$	1.028	.972	28.58	43
5	20	4.715	6.25	$\frac{5}{6}$	1.036	.964	27.82	42.08
6	24	5.658	7.5	1	1.043	.955	27.05	41.17
7	28	6.601	8.75	$1\frac{1}{6}$	1.051	.946	26.28	40 25
8	32	7.544	10	$1\frac{1}{3}$	1.058	.936	25.52	39.35
9	36	8.487	11 25	$1\frac{1}{2}$	1.066	.925	24.26	37.9
10	40	9.43	12.5	$1\frac{2}{3}$	1.074	.911	22.8	36.3
11	44	10.373	13.75	$1\frac{5}{6}$	1.082	.896	21.3	34.67
12	48	11 316	15	2	1.09	.89	19.7	32.93
13	52	12.259	16.25	$2\frac{1}{6}$	1.098	.884	18.1	31.33
14	56	13 202	17.5	$2\frac{1}{3}$	1.107	.878	16.61	29.63
15	60	14.145	18.75	$2\frac{1}{2}$	1.115	.872	15.14	28.35
16	64	15.088	20	$2\frac{2}{3}$	1.124	.866	13.67	27.04
17	68	16 031	21.25	$2\frac{5}{6}$	1.133	.86	12.2	25.76
18	72	16.974	22.5	3	1.142	.854	10	23 85
19	76	17 917	23.75	$3\frac{1}{6}$	1.151	.849	7.5	21.8
20	80	18.86	25	$3\frac{1}{3}$	1.16	.844	4.6	19.43
21	84	19 803	26.25	$3\frac{1}{2}$	1.169	.839	1.7	17.06
22	88	20.746	27.5	$3\frac{2}{3}$	1.179	.834	— 1.4	14.7
23	92	21.689	28.75	$3\frac{5}{6}$	1.188	.825	— 4.9	12.2
24	96	22.632	30	4	1.198	.817	— 8.6	9.96
25	100	23.575	31.25	$4\frac{1}{6}$	1.208	.808	—11.6	8.19
26		24.518	32.5	$4\frac{1}{3}$	1.218	.799	—17.1	5.22
27		25.461	33.75	$4\frac{1}{2}$	1.229	.79	—21.8	2.94
28		26.404	35	$4\frac{2}{3}$	1.239	.778	—27.	.65
29		27 347	36.25	$4\frac{5}{6}$	1.25	.769	—32.6	1" Vac.
30		28.29	37.5	5	1.261	.757	—39.2	8.5" "

Table of Chloride of Sodium (Salt) Brine.

Degrees on Salom.	Percent-age Salt by Weight	Pounds Salt per Cu. Ft.	Pounds Salt per Gallon	Specific Gravity	Specific Heat	Freezing Point F.	Ammonia Gauge Pressure
0	0	0	0	1	1	32	47.32
5	1.25	.785	.105	1.009	.99	30.3	45.1
10	2.5	1.586	.212	1.0181	.98	28.6	43.03
15	3.75	2.401	.321	1.0271	.97	26.9	41
20	5	3.239	.433	1.0362	.96	25.2	38.96
25	6.25	4.099	.548	1.0455	.943	23.6	37.19
30	7.5	4.967	.664	1 0547	.926	22	35.44
35	8 75	5.834	.78	1.064	.909	20.4	33.69
40	10	6.709	.897	1.0733	.892	18.7	31.93
45	11.25	7.622	1.019	1.0828	.883	17.1	30.33
50	12 5	8.542	1.142	1 0923	.874	15.5	28.73
55	13.75	9.462	1.265	1 1018	.864	13.9	27.24
60	15	10.389	1.389	1.1114	.855	12 2	25.76
65	16.25	11.384	1.522	1.1213	.848	10.7	24.46
70	17 5	12 387	1.656	1.1312	.842	9.2	23.16
75	18.75	13.396	1 791	1.1411	.835	7.7	21 82
80	20	14.421	1 928	1.1511	.829	6.1	20.43
85	21 25	15.461	2.067	1 1614	.818	4.6	19 16
90	22.5	16 508	2.207	1 1717	.806	3.1	18 2
95	23.75	17 555	2.347	1.182	795	1.6	16.88
100	25	18.61	2.488	1 1923	.783	0	15.67

PART II—REFRIGERATING MACHINERY

Looking back in history we read in the Songs of Solomon that in ancient times snow was used for the cooling of food and drink. The Kalif Mahdi (775) is said to have received shipments of snow by camels at Mecca, also the Sultan in the year 1000 had ice shipped continuously from Syria for his kitchen.

The cooling of water by means of mixtures of snow and salpeter was known to the Chinese already in the twelfth century.

Freezing mixtures of different salts with ice or snow appeared in Europe in the year 1550 in various compositions. This method of producing cold, however old, is still in every day use for such purposes as freezing ice cream.

FREEZING MIXTURES.

Ammonium nitrate...	1	From +40° to +4°
Water	1	
Ammonium chloride..	5	From +50° to +10°
Potassium nitrate....	5	
Water	16	
Ammonium chloride..	5	From +50° to +4°
Potassium nitrate....	5	
Sodium sulphate.....	8	
Water	16	
Sodium nitrate.......	3	From +50° to —3°
Nitric acid, diluted...	2	
Ammonium nitrate...	1	From +50° to —7°
Sodium carbonate....	1	
Water	1	
Sodium phosphate....	9	From +50° to —12°
Nitric acid, diluted...	4	
Sodium sulphate.....	5	From +50° to +3°
Sulphuric acid, diluted	4	
Sodium sulphate.....	6	From +50° to —10°
Ammonium chloride..	4	
Potassium nitrate....	2	
Nitric acid, diluted...	4	
Sodium sulphate.....	6	From +50° to —40°
Ammonium nitrate...	5	
Nitric acid, diluted...	4	

Snow or pounded ice..	2	From +50° to —5°
Sodium chloride......	1	
Snow or pounded ice..	5	From +50° to —12°
Sodium chloride......	2	
Ammonium chloride..	1	
Snow or pounded ice..	24	From +50° to —18°
Sodium chloride......	10	
Ammonium chloride..	5	
Potassium nitrate....	5	
Snow or pounded ice.	12	From +50° to —25°
Sodium chloride......	5	
Ammonium nitrate...	5	
Snow	8	From +32° to —23°
Sulphuric acid, diluted	2	
Snow	8	From +32° to —27°
Hydrochloric acid....	5	
Snow	7	From +32° to —30°
Nitric acid, diluted...	4	
Snow	4	From +32° to —40°
Calcium chloride.....	5	
Snow	2	From +32° to —50°
Calcium chloride, cryst	3	
Snow	3	From +32° to —51°
Potash	4	

In India it has been the custom from ancient times to make ice by the quick evaporation of water, for which purpose the Indian puts flat dishes filled one-half inch with water in a box twenty inches deep filled with straw. In dry nights part of the water evaporates, and being well insulated against the outer air, causes the rest of the water to freeze. The Compression and Absorption Machines are based on this principle of evaporation.

Ice made under vacuum was first done by Leslie, born 1766, at Largo, in Scotland. Leslie placed a shallow dish filled with concentrated sulphuric acid, and a few inches above that a small glass dish with water under the receptacle of an air pump. Under the vacuum water vapors were formed, which, however, were quickly absorbed by the acid, so that the evaporation of the water proceeded very rapidly. Through this quick evaporation on the surface of the water the heat of the water below was removed, until it was frozen. This is the principle of the Vacuum Machine.

At the beginning of the last century Hutton constructed a special machine in which compressed air was cooled and allowed to expand. He obtained in this way such low temperatures that alcohol was made to freeze. This is the principle of the Cold Air Machine.

These different methods of producing cold have passed through various stages of development and have led to constructions of types of machines, of which the compression machine has become the most prominent one. A description of these systems will be given in the following order:

A. Cold Air Machines.
B. Vacuum Machines.
C. Absorption Machines.
D. Compression Machines.

Cold Air Machines

The cold air machine has long been regarded as a thing of the past on account of its low efficiency and enormous size, and no machine of this type can be found any more in use on terra firma. But, strange to say, the cold air machine is still being built and has been installed in a large number of vessels. The specifications for bids for several U. S. warships provide that the refrigerating apparatus shall be of the "Cold Air Machine" type.

Principle of Cold Air Machine.—When air is compressed in a cylinder by mechanical means, its temperature rises. The heat of compression can be removed by injecting a spray of cold water into the cylinder or by passing the compressed air through a heat exchanger, where the temperature of the air will be lowered to nearly that of the cooling water.

When the air is now allowed to expand while doing work in an air engine, the temperature will be reduced considerably below

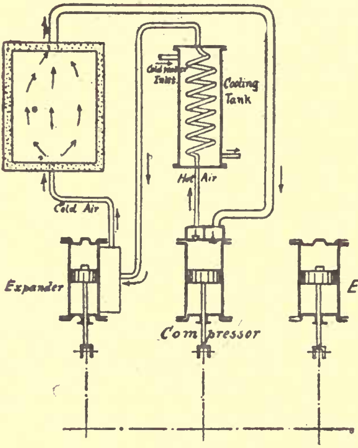


FIG. 2—DIAGRAM OF COLD AIR MACHINE.

the initial temperature and the expanding air is capable of absorbing the heat of the rooms to be cooled. For example: air of 68° F. under atmospheric pressure will be heated up to 185° F. when subjected to a pressure of two atmospheres. If we cool this hot air down to about 86° F. by means of cooling water and let it then expand to its initial pressure, its temperature will be lowered to 13° below zero. After the air has done the work of cooling it may reenter the compressor, thus performing a continuous cycle of operation.

This operation is illustrated in Fig. 2.

The air enters the compressor, is compressed and forced through a cooling coil submerged in cold water, where the heat of compression is removed. So cooled, it enters the expander. By expanding, its temperature is again lowered and the now cold air is discharged into the rooms to be cooled.

Historical Facts.—In 1850 Dr. Gorrie, an American, constructed the first cold air machine. In his machine the heat of compression was removed by a spray of cold water which was injected into the compressor. By expanding the cooled air a second spray of water was turned into ice.

A similar machine was constructed two years later by Nesmond. The compressor was provided with a water jacket and the air was compressed to twenty atmospheres. In a second cylinder the air was allowed to expand, whereby liquids were cooled or water was frozen.

About this time the Windhausen cold air machine was brought into the market and met with some success. About one hundred of these machines were built and several were in active operation up to the year 1883.

The Bell-Coleman machine found undoubtedly the largest market, although the machine did not differ in principle from Windhausen's design, but it was superior in the construction.

Of later constructions we only mention those by Menck and Hambrock, Lightfoot, Haslam Foundry Co., and the Leicester Allen machine.

Quite a number of government vessels, private yachts and steamers plying in South American waters are fitted with this latter type of machine.

The "Allen" Machine.

The Allen cold air machine, Fig. 3, is working on a continuous cycle of operation. The air is taken in by the air compressor B, under 60 to 70 pounds pressure and compressed to 210 to 240 pounds. The hot air is passed through a copper coil C immersed

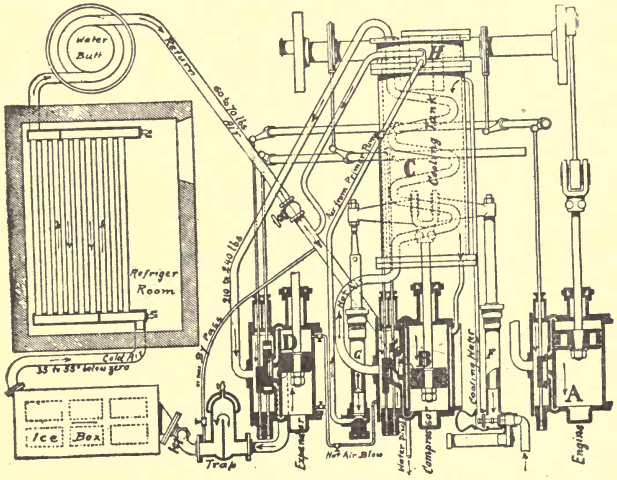


FIG. 3—DIAGRAM OF ALLEN MACHINE.

in circulating cold water, where the temperature is reduced to nearly that of the water.

The now cooled air enters the valve-chest of the expander D, which is constructed like a steam engine with a cut off valve. The valves admit the highly compressed air upon the piston to a certain point of the stroke and then shut it off. The piston continues to travel to the end of the stroke under the expanding force of the compressed air, assisting in this way the engine in doing the work of compression.

The result of the expansion is a very low temperature of the air at the end of the stroke. By the return stroke of the piston the air is pushed out to such places as are to be refrigerated.

On its way to the ice-making box the air passes through a trap, where the oil is separated which is used in the compressor and expanded. The trap contains a steam jacket in order to melt the frozen contents when they are to be blown out.

Pump F circulates the cooling water through the cooling tank and through the water jacket around the compressor B.

A small air compressing pump, G, takes air from the atmosphere and charges the system with the required air pressure, which it maintains.

This air contains the usual atmospheric moisture and to expel this the air is first forced through the trap H, where the air is cooled by coming in close contact with the cold head of the reservoir. It is claimed that about 80 per cent of the moisture is in this way deposited out of the air and drained off by pet-cocks. This is of great importance, as the large amounts of latent heat in the water vapor would produce serious losses in the result of the machine if the air contained water, this being subject to the heating and freezing processes which occur in the machine.

By comparing the cold air machine with compression machines, it is evident that machines which do not liquefy the refrigerating medium cannot be as economical as those which do. The compression and expansion cylinders of the cold air machine have to be very large, which increases the friction considerably. Besides this there is excessive clearance and this together with the unavoidable moisture contained in the air reduces the actual efficiency to less than 33 per cent of the theoretical efficiency.

The reason for still using the cold air machine on board ship is all and alone the harmless character of the refrigerating medium air.

NOTES ON COLD AIR MACHINES:

Vacuum Machines

The vacuum machine is, strictly speaking, based on the same principle as the absorption machine, which we will discuss in our next article. Water is the evaporating medium and sulphuric acid is used for absorbing the vapors.

Principle of Vacuum Machine.—The evaporation of the water at a low temperature in order to produce refrigeration is brought about by forming a vacuum by means of a vacuum pump. Such a vacuum is now produced in a closed vessel. In this the water is injected, part of which quickly evaporates, whereby the necessary latent heat is removed from the remaining water, which will be cooled and finally frozen. Theoretically about six times the amount of water can thus be frozen by the evaporation of one part of water, as the latent heat of the water is about 940, that is, about six times the latent heat of ice, viz., 142.

If the vacuum should be maintained solely by a pump, this pump would have to be of an enormous size on account of the low tension of the water vapor at the temperature of the refrigerator. In order to avoid excessively large pumps an absorbent was looked for to release the work of the air pump, and this has led to the introduction of sulphuric acid, by which the vapors are quickly absorbed and removed by the air pump.

The acid in the course of time becomes weak and has to be concentrated again by distillation.

The operation is illustrated in Fig. 4. The vacuum pump is connected to the absorber, a long cylindrical vessel filled to two-

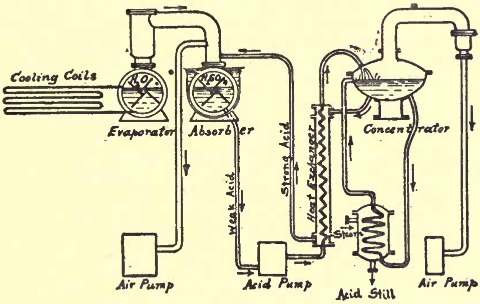


FIG. 4—DIAGRAM OF VACUUM MACHINE.

thirds with concentrated sulphuric acid, which is kept in motion by paddles to facilitate the absorption of the water vapors coming from the water. The absorber is encased in a cold water jacket. In the cooler, which is well insulated, the refrigerating work takes place, whereupon it is connected to coils through which the cold liquid circulates.

The other apparatus shown in the illustration serves for the concentration of the sulphuric acid. The cold weak acid is pumped through an exchanger into the distiller, where part of the absorbed water is evaporated and removed by a small air pump. The strong acid leaves at the bottom and flows through the still back to the distiller in a superheated state. When concentrated the acid leaves at the highest points, parts with its heat and re-enters the absorber.

Historical facts.—In 1810 Leslie constructed a small vacuum machine. He was followed by quite a number of others, among whom was Carré, whose machine was exhibited at the World's Fair in Paris in 1867. Windhausen was the first to build a vacuum machine in Germany (1878). His machine is illustrated in diagramatic form in Fig. 4. The vacuum maintained by the pump is 1-1500 atm. = 1-50 inch abs. press.

Of later inventions those by Lange, Southby and Blyth and Patten may be mentioned.

The latter type is of American origin and of recent date.

Patten Vacuum Machine.

The apparatus starts with the evaporator or freezing chamber, as it is called here, Fig. 5, as only ice is produced. A vacuum of about 30 inches is maintained in the freezing chamber by the air pump, which will cause the temperature to drop down to

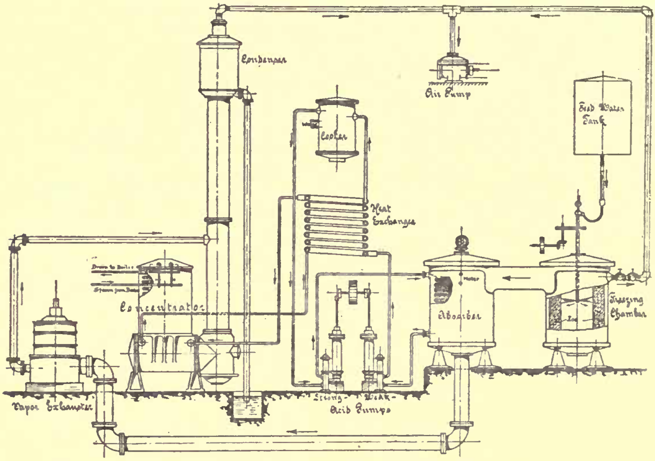


FIG. 5—DIAGRAM OF PATTEN MACHINE.

26° F. The water, generally city water, which has previously been filtered, is fed by a hose from the feed water tank to a spraying device, by means of which it is sprayed against the ice-forms in the freezing chamber. By means of special mechanism a rotary reciprocating motion is imparted to the sprayer. In this way cylinders of ice are formed, having an outside diameter of six to eight feet, and a height of four to eight feet. The thickness may be, of course, varied, and depends on the quantity of water fed. A cylinder of about seven feet outside diameter and thirteen inches thick, having a length of three feet and over, weighs about 3,200 pounds and takes about one hour to freeze.

When harvesting the ice, the cover is raised and the cylinder is withdrawn from the freezing chamber and transferred to the cutting table, where it is reduced to blocks of commercial size.

It is claimed that about 86 per cent of the water is instantly frozen in touching the sides of the ice forms. The other 14 per cent of vapor from the freezing chamber are led to the absorber, where they come in contact with the sulphuric acid which is trickling over lead coils, through which cold water is circulated. The vapors are drawn through the absorber by means of the vapor exhauster, where they are compressed and forced into a large pipe leading to the vapor condenser.

The weak acid leaves the absorber and is pumped through a double pipe heat-exchanger in counter current, where it takes up part of the heat of the strong acid before entering the concentrator. Steam from the boiler is supplied to the lead-lined steam pipes of the concentrator and the weak acid of about 45° Beaumé is converted to strong acid of about 60° Beaumé.

The strong acid leaves the concentrator, gives up part of its heat to the weak acid in the heat exchanger and in a special cooler receives a final cooling, sufficient to be used again in the absorber.

The vacuum in the concentrator being about 27 inches, the overflow of the condenser must have a head of at least thirty-three feet above the hot well.

The first plant, which Patten erected, did not use any chemical absorber. It was erected in Baltimore at a cost of over three hundred thousand dollars, but has proved a failure. Other plants using sulphuric acid have successively been erected in Baltimore, New York, San Francisco and Porto Rico.

There are many reasons why the vacuum machine is prevented from being more adapted. The ice frozen by this process is not transparent, but opaque and resembles chalk. The vessels and pipes containing the sulphuric acid must be of lead or lead-lined on account of the corrosive properties of the acid. The necessity for distilling the sulphuric acid represents one of the principle expenses, while the handling of this liquid is of considerable inconvenience. These reasons besides the difficulties to keep the system perfectly tight will necessarily put the vacuum machine behind other systems, or at least will confine its use to special cases.

NOTES ON VACUUM MACHINES:

Absorption Machines

The absorption machine is operated in a similar manner as the vacuum machine, only that ammonia is used instead of water. Ammonia has a great affinity for water, so much in fact that one part of water at 32° F. will absorb about 1,000 parts of ammonia at atmospheric pressure. This fact is utilized in the following way:

Principle of Absorption Machine.—Liquid ammonia under an average pressure of 150 lbs. per square inch is admitted to the expansion coils, where it rapidly evaporates. In doing this it produces a refrigerating effect equal to its latent heat of vaporization. The expanded gas is subjected to a stream of cold water in the absorber, where it is quickly absorbed, forming aqua ammonia. This liquor is pumped through a heat exchanger into the liquor still, commonly called the generator, where it is heated up by means of steam coils and the ammonia driven off as gas. The hot gas being confined produces pressure much as steam does in a boiler. It passes from the still to the condenser, where it is reduced to a liquid again under the influence of pressure and cold water.

The weak hot liquor leaves at the bottom of the still and gives up part of its heat in the exchanger to the incoming strong liquor, before being able to absorb anew the ammonia vapors in the absorber.

Historical Facts.—The inventor of the absorption machine with a continuous cycle of operation is F. Carré, of Paris (1860). His machine was improved by many others, notably Vass and Littman,

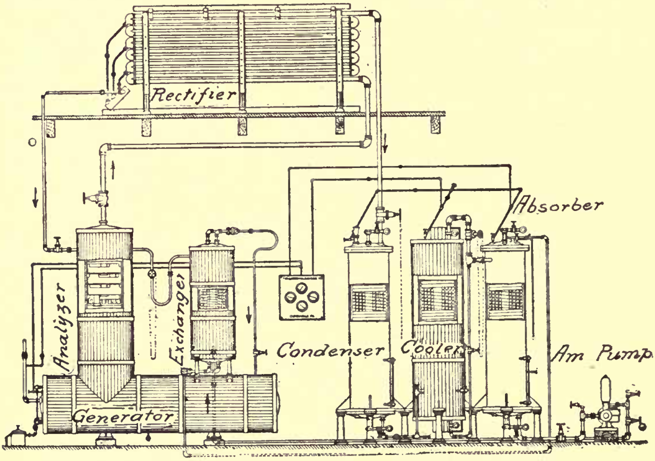


FIG. 6—PONTIFEX (CARBONDALE) ABSORPTION MACHINE.

Nicolle and Pontifex. The latter type is of English origin, but is, with slight alterations, extensively built in this country, where it has become one of the leading absorption systems.

Pontifex (Carbondale) Absorption Machine.

The illustration, Fig. 6, shows the generator with the analyzer and exchanger mounted on top. The first charge of aqua ammonia is placed in the generator, where it is heated by means of steam coils in the usual manner. The liberated gas passes upward through the analyzer where some of the water still left in suspension in the gas is removed by a series of baffle plates. Thence the gas enters the lower coil of the rectifier, where the remaining water is condensed, much in the same way, as ammonia is liquefied in the De La Vergne counter current ammonia condenser. The condensed water collects in a manifold and returns automatically to the generator.

Thence the gas passes to the condenser, where it is liquefied. The condenser serves also as a liquid receiver, from where the liquid is fed to the expansion coils in the brine cooler.

The expanding gas is absorbed in the absorber by the weak liquor coming from the exchanger and the resulting strong liquor is returned by the ammonia pump through the coils in the exchanger to the generator.

Condenser, cooler and absorber are of the coil and shell type, the coils are wound concentrically and project through stuffing boxes in the heads and are manifolded outside of the shells.

Vogt Absorption Machine.

The generator, Fig. 7, consists of a main casting, divided into four compartments, communicating with each other, and four horizontal pipes, connected to the main casting, which contain the

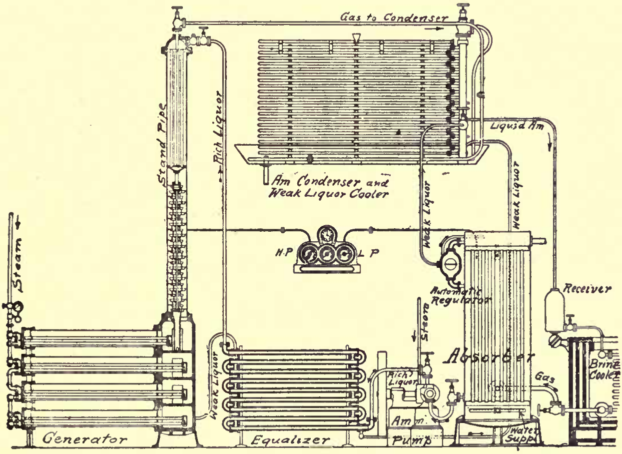


FIG. 7—VOGT ABSORPTION MACHINE.

steam heating coils. On top of the main casting is mounted a stand pipe containing an analyzer and rectifying coil for drying the gas before leaving the still. The strong liquor is admitted to the top of the stand pipe, passes through the rectifying coils and analyzer to the upper compartment, flowing thence over the steam coil in the horizontal pipes from one to the other until the lower compartment is reached.

The gas generated passes through the opening in each compartment to the stand pipe, where the moisture is deposited, and the dry gas passes to the condenser, which is of the atmospheric horizontal zig-zag coil pattern.

The absorber is constructed like an upright tubular boiler open at the top. Tubes are distributed uniformly and arranged in such manner that they can be cleaned while the machine is in operation. The cooling water enters at the bottom and discharges at the top. The return gas from the expansion coils enters at the bottom and the weak liquor at the top, the flow of the latter being controlled by an automatic regulator.

The ammonia pump is of the double-acting horizontal fly-wheel pattern, its speed is 25 revolutions per minute.

The exchanger is of the double pipe pattern. The strong liquor enters at the bottom, while the weak liquor from the still enters the exchanger at the top.

Management of Absorption Machine.

The first thing to be looked after in a new plant is that the apparatus is thoroughly freed from air before it is charged and that it is properly tested. The manufacturers are generally supposed to do this, but even if they do, the process should be carefully looked after by the engineer in order to avoid complications. Two ways are recommended for forcing out the air, the most effective of which is to use a vacuum pump. If the pump is not available, the apparatus may be filled with steam, all valves being open, one being open to the atmosphere. The steam forces the air out and then when the valve is closed and the machine cools down, the steam condenses, leaving a vacuum in the apparatus. The pumping method is much more desirable, since the steam method sometimes softens the joints, if they are made up with rubber especially, and it is seldom that the boiler pump is not available.

When the air has been expelled, the apparatus is ready to receive the ammonia and the charge pipe is connected to a drum of ammonia and then with another until the ammonia ceases to flow in because the vacuum has been destroyed, as shown by the vacuum gauge. Nearly all the ammonia can be put in in this way, but an amount nearly sufficient to make up the proper charge will be put in by the ammonia pump. In making the connections to the ammonia drums and to the pump, particular care must be taken to not allow any air to enter the machine along with the ammonia. The ammonia is now warmed up by allowing steam to flow through the coils of the heater, and this is continued until the pressure on the system rises to about 100 pounds in most cases. A piece of hose is then attached to the purge cock, which is opened, and the end of the hose placed in some vessel containing water. This allows any remaining air to come out, appearing in the form of bubbles on the surface of the water, but preventing any flow of the ammonia. The condensing water is then turned on, and also the steam, until the liquid ammonia shows in the gauge. Then turn on the cooling water wherever it is used and let the steam into the generator coils, and open up the connection to let the poor liquor into the absorber. When the liquid shows in the receiver gauge, open up the expansion valve a little and the valve on the pipe between absorber and cooler. The ammonia pump will have to be started directly, if everything works all right. If air develops, it must be eliminated through the purge cock on the absorber. If insufficient pressure develops, the charge must be increased by connecting a drum of liquid ammonia to the cooler and allowing it to flow in. Before doing this the expansion valve should be shut.

The ammonia pump should be lower than the supply when pumping ammonia. The proportionate strength of the weak to the strong liquor should be about 17 to 28. When this is not the case it is probably due to leaks.

Ammonia will cause the rubber packing on pump rods to swell, therefore the glands must not be screwed down too tight.

"Priming" has been a frequent cause of shut-downs. This is a case of all the ammonia going over into the condenser, including the aqua ammonia. It may even get into the expansion coils if they are not protected by a check valve. This is indicated by the height of the liquid in the still, by a drop in pressure on the cooler, and the melting off of the ice on the expansion valve air pipe. The liquor in the still should always cover the steam coils. The "boiling over" may not extend further than from the generator to the absorber, but may extend to the condenser, as stated above. If the liquid is at the right level in the liquid receiver, the proper level is likely to be maintained in the generator unless too much is coming from the absorber. The pressure behind the expansion valve should maintain the proper height of liquid in the generator. To provide against this trouble, a valve is placed on the poor liquor line at the absorber, so that the ammonia can be kept at the proper height. When the ammonia has gone over into the expansion coils, the expansion valve can be almost closed and a vacuum pumped on the absorber. The gas is then blown through the coils and this will generally take it all back to the absorber. This trouble may be avoided when the expansion coils are built in sections connected to manifolds with separate valves. In such case each section can be cleared separately.

James Cooper, in *Power*, recommends in a case of priming that the pump be kept going to get a good vacuum on the absorber. Then to open the expansion valve so as to get all the weak liquor out of the receiver and condenser into the cooler, and if the pressure is still below that of the absorber, and they both show a vacuum at this time, shut the expansion valve and open the anhydrous charging valve. This will let the air run in from outside and cause the cooler to show atmospheric pressure, which will be greater than the pressure in the absorber, and then be pumped to the generator again. This operation to be kept up until the machine is normal. The cause of this condition may be that the charge is too weak or the machine is working too fast and the generator is dirty. The weak liquor will have to go through the purge line at the bottom of the cooler, and to keep a greater pressure on the cooler than on the absorber the gas line will have to be closed between the cooler and the absorber. This will force the liquid out faster. This is recommended in case there is no pipe from the receiver to the cooler.

The management of an absorption system mainly depends on the regulation of pressures and temperatures. If, for instance, there is too high a pressure in the absorber and consequently too high a temperature in the cooler, the cause may be either too little or too warm cooling water or too much liquid in the system or the presence of foreign gases and air in the system. These latter are eliminated through the purge cock at the top of the absorber.

One reason for the failure of an absorption machine not to work to its full capacity at times is because the steam coils in the generator become air locked. By putting on a small vacuum pump the efficiency of the still may be considerably increased.

Leaks in rectifying pans are indicated when a sample of liquid from the liquid receiver shows a high percentage of water.

A leak in the exchanger is indicated by the cooling of the pipe connecting the exchanger with the weak liquor at the bottom of the still. There is also likely to be a hissing sound produced by the leak. The leak can usually be traced by noting the temperature of the pipe.

Economy of Absorption Machine.

The absorption machine, once a favorite, was largely replaced by the compression system, but is now coming into considerable use under certain conditions. The economy has been greatly increased since the manufacturers are able to produce an almost perfect anhydrous gas from the generator and since it is possible to use the exhaust steam from the auxiliary machinery to evaporate the ammonia in the generator.

According to Torrance, in a paper before the Eastern Ice Association the best absorption machines of the present time use, in the generator, about 30 pounds of steam per hour per ton of refrigerating effect under can ice conditions, some use 35, and many machines recently erected, but of poor design, use 50 pounds or more. A theoretically perfect absorption machine would require for the generator about 24 pounds per hour per ton with 10 pounds steam pressure for can ice conditions, this quantity being practically independent of the temperature of the condensing water.

If a machine uses 26 pounds of steam per hour per ton, then we could freeze ice on the can system out of 60° F. raw water with the following steam consumption per hour per ton of ice:

	Pounds.
Cooling water from 60° to 32° F.	5
Freezing water at 32° F.	26
Cooling ice from 32° to 15° F.	1.5
Cooling 300-lb. cans from 60° to 15° F.	.2
Radiation and losses	7.3
Meltage loss 3% of total	1.2
Total pounds steam per hour	41.2

A horizontal tubular boiler, semi-bituminous coal, under careful firing will evaporate 10.3 pounds of water per pound of coal from and at 212° or 10 pounds into steam at 70 pounds pressure with 212° feed water. Hence, coal per hour would be 4.12 pounds per ton of ice, or 99 pounds per day per ton of ice, or 20 pounds of ice per pound of coal.

Practical Ice Plants of the Present.—If we have a horizontal tubular boiler with above mentioned evaporation, from feed water at 212° (which is quite easily obtained with a slight pressure on the exhaust), we should be able to make 10 pounds of ice per pound of coal provided we have no losses.

If the plant is designed properly there would be five losses.

(1) Condensed steam caused by radiation of pipes and pump cylinders which forms an emulsion with the lubricating oil and is trapped out in the oil separator. There is no cut-off on these pumps and the condensation is practically limited to the radiation of the exposed surfaces and should not exceed 5 per cent.

(2) Direct leakage of steam from stuffing boxes and joints. This is too small to be considered.

(3) Reboiling loss. The condensed steam from the generator discharges at 10 pounds pressure into the reboiler and immediately drops in temperature from 240° to 212° F., causing 1 per cent to evaporate, which produces all the reboiling generally necessary.

(4) Skimming loss under these conditions should not exceed ½ per cent.

(5) Meltage at ice cans, 3 per cent.

Total losses 9½ per cent.

The boiler evaporation being 10:1 under the above conditions this would make the economy 9 pounds of ice per pound of coal, which is about the result actually obtained in practice.

NOTES ON ABSORPTION MACHINES:

Compression Machines

Principle of Compression Machines.—The compression machine is based on the evaporation of liquids, which have a low boiling point. The latent heat of evaporation represents the amount of cold that can be produced in precisely the same way as in the absorption machine. The former system, however, differs from the latter in so far, as the expanded gas after having done the work of cooling in the expansion coil, instead of being absorbed, enters the suction of a strong air compressor, where the necessary pressure is applied to reduce the gas to a liquid again.

The principal refrigerating media used in the compression machine are ether, sulphur dioxide, carbonic acid and ammonia.

The systems are all based on the same principle and the machines differ only in points of construction.

A compression machine comprises the three fundamental parts:

(1) *The compressor,* which withdraws the gas from the refrigerator coil and compresses it into the condenser.

(2) *The condenser,* where the heat of compression is removed by cooling water and the gas becomes liquefied.

(3) *The refrigerator,* where the liquid evaporates into a gas and does the refrigerating work.

These principles are generally the same for the various liquids employed, amplified, of course, by different appliances for lubricating the piston and stuffing box, by special devices for separating oil and foreign matters from the medium, etc.

Ether Machines.

In 1834, Perkins employed already the vapors of Ether (Ethyl Ether) whose boiling point is at above 100 degs. F., for his compression machines and the construction and arrangement of his system were similar to the modern compression machines.

It consisted principally of a compressor, refrigerator and condenser with regulating valve between the two last mentioned.

In 1867, Teller used first Methyl Ether, which has a lower boiling point, and in 1878 Vincent employed Chlormethyl Ether.

Ether machines were never very popular, chiefly on account of their great danger in case of fire and the relative large compressors, for which reason we do not want to go any deeper into the constructive details of this type of machine.

Sulphur Dioxide Machines.

These machines have lately come more and more into the foreground. Though the latent heat of the medium is lower than ammonia besides having a higher boiling point which requires larger compressors, this machine has certain advantages. The pressures corresponding to the required temperatures are low; they go up to sixty pounds at the highest during compression and down to seven to fifteen pounds in the refrigerator.

Lubrication is entirely superfluous, as the liquid SO_2 is a first-class lubricating medium. Another advantage is its non-corrosive action toward metals, which allows the use of brass, copper and other metals besides iron. But great care has to be taken to maintain tight joints as any leakage might produce sulphuric acid, which would become detrimental to any metal.

Teltier was the first in 1865, to recognize the importance of sulphur dioxide as a refrigerating medium, and in 1876, Pictet made use of the same in his machine. His machines have since then been built extensively.

The principles of the compression machines are also applied to the sulphur dioxide machines, although the whole arrangement is simpler, as the apparatus for separating the oil from the gas and everything herewith connected are not needed.

Carbonic Acid Machines.

Carbonic acid (CO_2) has besides ammonia and sulphur dioxide found the greatest use in compression machines. This machine was first built in 1883, by the Maschinenfabrik Augsburg, but became more known through Windhaussen in 1889, who succeeded in bringing an efficient design in the market.

In his machine the clearance was filled out with glycerine. This brought some disadvantages. Part of the glycerine could pass through the valves into the pipes and apparatus and reduce the efficiency. This loss again increased the clearance.

Sedlacek built his machine so, that the sealing liquid was kept under pressure and the loss made up automatically by a small pump. Later constructions have done away with glycerine and use oil instead.

It will be found that machines working with dry gas are capable of performing a refrigerating duty which exceeds that of the wet system by about ten per cent. (Goosmann, A. S. R. E. Trans., 1906.) When manufacturers, nevertheless, adhere to the wet system in preference, it is simply the logical outcome of practical considerations. The packing of the piston consists of leather cups; this material does not withstand temperatures above 200° F. and in order to keep them pliable, it is necessary to remove the heat of compression by means of wet gases from the evaporator. Metallic packing with its consequent greater piston leakage and dry gas compression, offers no gain in comparison with the wet system and its slight loss of evaporation which is offset by the advantage of using a tight piston packed with cupped leathers.

The fact that during compression the gas is in a superheated state, occasioning considerable changes in its entropy with temperatures and pressures above the critical, explains the peculiarity that the refrigerating work of this system does not cease with high condenser temperatures.

Constructional Details.—The cylinders are made of soft forged steel, as it seems impossible, here as well as in England, to secure sound castings that will withstand the high internal pressures. These cylinders require considerable lathe and drill work for the bore, canals and other openings. When finished, however, it is hardly necessary to subject them to tests.

The bore should be about one-fourth of the stroke, for instance, a machine of 20 tons capacity having a bore of four inches should have a stroke not less than sixteen inches. A machine of five-inch bore by 20-inch stroke will easily have a capacity of 40 tons, which shows the influence of a slight increase in the size upon the capacity.

A long piston is of great advantage. The relation of diameter and length of piston is about 1:2.5. These valves are usually placed in the horizontal position, but as they are comparatively small and of light weight, it does not require a very heavy spring to close them. The discharge valves are placed vertically and are therefore always in the centrical position. The area of the discharge and of the suction valve is one-seventh of the piston area for the former and one-half for the latter. On the piston rod end two suction valves are frequently used, as there is hardly sufficient room for one valve having the required area. The width of the seat should not exceed 0.1 to 0.12 of the valve disc diameter, and an angle of 70° to 90° for the discharge valve seat and 60° to 75° for the suction valve are considered good practice. A valve life of 0.33 diameter for the suction valve and 0.28 diameter for the discharge valve are the right proportions. A spring tension of

8 to 9 lbs. for the suction valve and 10 to 11 lbs. for the discharge valve will be found ample.

The most essential point is the stuffing box. Owing to the high internal pressure, as well as to the comparatively large piston rod, it is necessary to divide the stuffing box into several chambers, consisting of removable lanterns, which are so arranged that the pressure is reduced by steps. The chamber next to the cylinder bore takes care of the leakage; a controlling device is usually connected to this chamber by means of which the gas is returned to the suction side at a pressure higher than that of the evaporation and lower than the condenser pressure. The next chamber is kept under oil by a force pump, which forces the oil into it at a pressure slightly above that of the suction. An oil outlet, controlled by a ball valve, leads from this chamber to the suction canal of the compressor, so that a small amount of oil together with an occasional bubble of gas enters the compressor at this point. Garlock or any other soft packing is used at the outer end merely as a wiper of the lubricating material, preventing oil leakage at that point.

Leather cups are used almost exclusively as the packing material, they having given much better satisfaction than any other known method of packing. In packing the stuffing box with this material, the glands must be drawn up tight, as no provision for expansion of the material need be made in this case; only the outer nut, which holds the Garlock packing in place, is left comparatively loose. The life time of this packing is a season or more with ordinary care. A trap to separate the oil from the gas is connected in the discharge pipe between compressor and condenser.

Safety valves are always used. The location of this valve on the compressor is in the discharge canal. They also serve the purpose of protecting the compressor in the case of careless starting, without opening the delivery stop valve. This valve is usually provided with a cast iron disc, proportioned to break at a pressure of about 150 atmospheres.

When condenser water of temperatures above 74° F. is used it is advisable to provide a special liquid cooler for the purpose of reducing the temperature of the liquid before it passes the expansion valve. Submerged, atmospheric and double-pipe condensers are used; the customary rules prevail regarding the surface of the evaporator pipe, with this difference, that the evaporating temperatures may readily be dropped much below zero F. without changing materially the ratio of compression, which ordinarily is 1 :3.

While it is true that the theoretical efficiency of the carbonic acid system is not equal to that of the ammonia machine, owing to the greater percentage which the specific heat of the liquid carbonic acid bears to the latent heat of evaporation, yet the practical efficiency of the machine, owing to compensating features, makes up for the above loss. These consist in less piston leakage, a smaller depression of the suction line, and slightly smaller losses through clearance.

Ammonia Compression Machines

In 1870 Linde built the first ammonia compression machine, which has become the standard for modern refrigerating machines.

About the same time Boyle constructed a similar machine.

The Linde machine in its principle is operated on the compression cycle, which we have described above. Almost all later designers have constructed their machines after the Linde and Boyle patterns with slight variation.

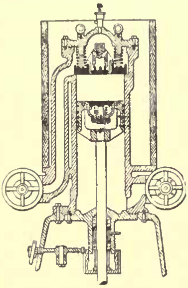


FIG. 8—"SAFETY HEAD" COMPRESSOR.

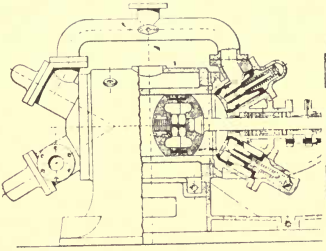


FIG. 9—"LINDE."

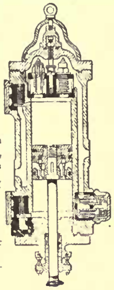


FIG. 10—"OIL" COMPRESSOR.

The leading compressor types as built in this country are illustrated in Figs. 8 to 10, and may be briefly enumerated here.

The *Linde* compressor, Fig. 9, is worth careful study by both the student and engineer, as it is a good example of how efficiency may be combined with simplicity. The cylinder is one plain cylindrical bushing. Both heads, holding the valves, as well as the piston, are turned spherical and fit snugly against each other. There is hardly any clearance, the piston at extreme end of the

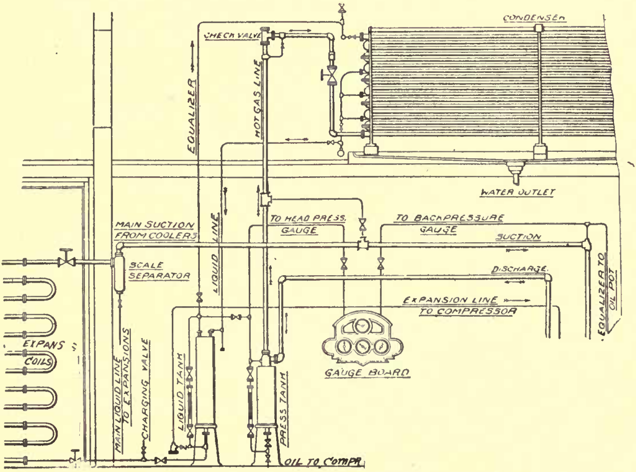


FIG. 11—"DE LA VERGNE" COMPRESSION SYSTEM.

stroke being only 1-32 inch from the cylinder head. The compressor is double-acting and may be horizontal or vertical.

The *safety-head compressor,* Fig. 8, is also put on the market by a great number of builders.

The advantage of the safety head is the security it guarantees against the breaking of the head in case of accidental breaking of valves or any other part of the machine, as well as an overcharge of liquid ammonia getting in the compressor, in which case the head lifts and allows the obstruction to pass through.

The oil *compressor,* Fig. 10, was, some ten years ago, considered the foremost machine in the market, and is still one of the most efficient ones; but owing to its expensive construction it is only built when there is a special demand for it.

Cycle of Operation.

The cycle of operation is illustrated in Figs. 11 and 12. These cuts show plainly every detail, and as drawings sometimes speak plainer than words, especially to the trained engineer, we will try to save space by omitting the descriptions.

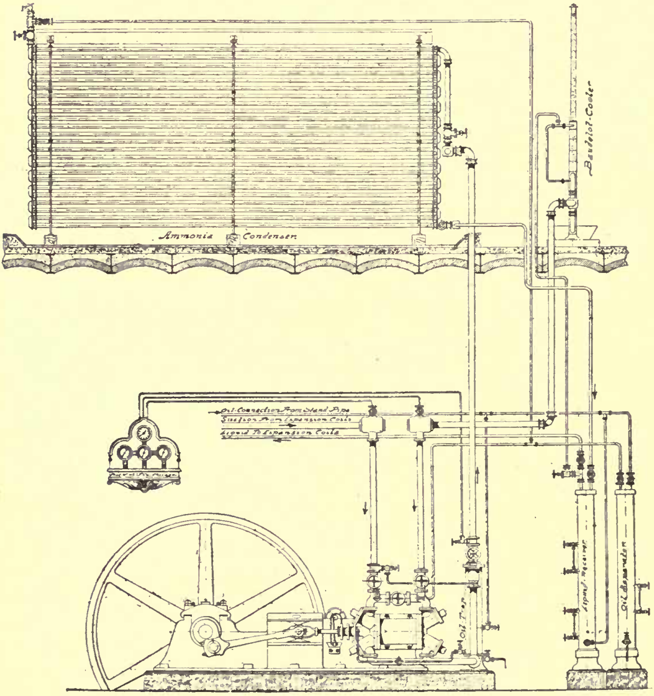


FIG. 12—"LINDE" COMPRESSION SYSTEM.

Compressor

Capacity of Compressor.

The refrigerating capacity of a compressor per minute is the product of the number of cubic feet that can be discharged by the compressor per minute and the refrigerating effect of one cubic foot of gas. Thus we have to consider the following two points:

1. *The cubic capacity of the compressor.*
2. *The refrigerating effect of the medium employed.*

Cubic Capacity.

The *theoretical displacement* is ascertained by multiplying the *piston area* by the *stroke,* and the *number of revolutions per minute,* and, in case of a double-acting compressor, by *doubling the result* (deduct area of piston rod).

$$C = 2 \frac{3.14d^2}{4} ln$$

where d = dia. of piston, l = stroke, n = number of rev. p. min.

The *actual displacement* depends on the efficiency of the compressor. The greater the ratio of compression, the greater is the loss with a given amount of clearance. Assuming a condenser pressure of 160 pounds and a back pressure of 20 pounds, or a compression ratio of 1:8, with a clearance of ⅛ inch, the gas would re-expand from 160 pounds to 20 pounds, and occupy 1 inch space, before fresh gas could be admitted into the compressor. This 1 inch would be deducted from the effective stroke and by assuming a compressor having a 10-inch stroke, would mean a loss of 10 per cent.

Refrigerating Effect of Medium.

The refrigerating effect of 1 cb. ft. of gas is represented by the latent heat of 1 lb. of gas, divided by the volume of 1 lb. of gas.

From the latent heat, however, we have to deduct the amount of refrigeration, which is required to reduce the temperature of the liquid from the condensor temperature to the refrigerator temperature. This amount is the difference in temp. multiplied by the spec. heat of the medium.

$$r = \frac{h_1 - (t - t_1)\, s}{v}$$

t = condens. temp., t_1 = refr. temp., s = spec. heat of medium, h_1 = latent heat at temp. t_1, v = volume of 1 lb. of gas in cub. ft. at refr. temp. (See ammonia table.)

Example.—What is the refr. capacity of a double-acting ammonia compressor 9 × 15, 70 rev. p. min., temp. in refr. = 0°, temp. in condenser = 85°.

By assuming an efficiency of 90%, the actual displacement would be $2 \frac{3.14 \times 0.75^2}{4} \times 1.25 \times 70 \times 0.9 = 69.3$ cb. ft. p. min.

The refr. effect per cb. ft. $= \frac{555.5 - (85 - 0)\, 1}{9.1} = 52.3$ units p. min.

Capacity of compressor = 69.3 × 52.3 = 3624.4 units per min., or in tons of refr. $= \frac{3624.4 \times 60 \times 24}{284{,}000} = 18.4$ tons in 24 hrs.

Cubic capacity of compressors per ton per min. =

$$\frac{69.3}{0.9 \times 18.4} = 4.18 \text{ cub. ft.}$$

REFRIGERATING EFFECT (B. T. U.) OF ONE CU. FT. OF AMMONIA GAS PER MIN.

Temperature of Gas in Degrees F.	Corresponding Suction Pressure, Lbs. per sq. in.	Temperature of the Liquid in Degrees F.								
		65°	70°	75°	80°	85°	90°	95°	100	105
		Corresponding Condenser Pressure (gauge), lbs. per sq. in								
		103	115	127	139	153	168	184	200	218
	G. Pres.									
-27°	1	27.30	27.01	26.73	26.44	26.16	25.87	25.59	25.30	25.02
-20°	4	33.74	33.40	33.04	32.70	32.34	31.99	31.64	31.30	30.94
-15°	6	36.36	36.48	36.10	35.72	35.34	34.96	34.58	34.20	33.82
-10°	9	42.28	41.84	41.41	40.97	40.54	40.10	39.67	39.23	38.80
- 5°	13	48.31	47.81	47.32	46.82	46.33	45.83	45.34	44.84	44.35
0°	16	54.88	54.32	53.76	53.20	52.64	52.08	51.52	50.96	50.40
5°	20	61.50	60.87	60.25	59.62	59.00	58.37	57.75	57.12	56.50
10°	24	68.66	67.97	67.27	66.58	65.88	65.19	64.49	63.80	63.10
15°	28	75.88	75.12	74.35	73.59	72.82	72.06	71.29	70.53	69.76
20°	33	85.15	84.30	83.44	82.59	81.73	80.88	80.02	79.17	78.31
25°	39	95.50	94.54	93.59	92.63	91.68	90.72	89.97	88.81	87.86
30°	45	106.21	105.15	104.09	103.03	101.97	100.91	99.85	98.79	97.73
35°	51	115.69	114.54	123.39	112.24	111.09	109.94	108.79	107.64	106.49

CUBIC CAPACITY OF COMPRESSOR (PER MIN.) PER TON OF REFR. (IN 24 HRS.)

Temperature of Gas in Degree F.	Corresponding Suction Pressure, Lbs. per sq. in.	Temperature of the Gas in Degrees F.								
		65°	70°	75°	80°	**85°**	90°	95°	100°	105°
		Corresponding Condenser Pressure (gauge), lbs. per sq. in.								
		103	115	127	139	**153**	168	184	200	218
	G. Pres.									
-27°	1	7.22	7.3	7.37	7.46	7.54	7.62	7.70	7.79	7.88
-20°	4	5.84	5.9	5.96	6.03	6.09	6.16	6.23	6.30	6.43
-15°	6	5.35	5.4	5.46	5.52	5.58	5.64	5.70	5.77	5.83
-10°	9	4.66	4.73	4.76	4.81	4.86	4.91	4.97	5.05	5.08
- 5°	13	4.09	4.12	4.17	4.21	4.25	4.30	4.35	4.40	4.44
0°	**16**	3.59	3.63	3.66	3.70	**3.74**	3.78	3.83	3.87	3.91
5°	20	3.20	3.24	3.27	3.30	3.34	3.38	3.41	3.45	3.49
10°	24	2.87	2.9	2.93	2.96	2.99	3.02	3.06	3.09	3.12
15°	28	2.59	2.61	2.65	2.68	2.71	2.73	2.76	2.80	2.82
20°	33	2.31	2.34	2 36	2.38	2.41	2.44	2.46	2.49	2.51
25°	39	2.06	2.08	2.10	2.12	2.15	2.17	2.20	2.22	2.24
30°	45	1.85	1.87	1.89	1.91	1.93	1.95	1.97	2.00	2.01
35°	51	1.70	1.72	1.74	1.76	1 77	1.79	1.81	1.83	1.85

Horse Power Required.

The *work required* from the compressor for every lb. of liquid consists in lifting the latent heat through the range of refr. temp. to condens. temp.

$$W = \frac{t - t_1}{T} h_1 \quad (T = \text{abs. refr. temp.} = t_1 + 460)$$

The amount of liquid per minute is the product of the cubic capacity and the weight of 1 cb. ft. of gas at refr. temp.

Example continued: The work for above compressor would be

$$= \frac{t - t_1}{T} h_1 C_1 a = \frac{85 \times 555.5 \times 69.3 \times 0.11}{460} = 782.5 \text{ units per min.}$$

$$= \frac{782.5 \times 778}{33{,}000} = 18.5 \text{ H. P.}$$

The actual horse-power required to operate the compressor must necessarily be larger on account of the friction of piston, stuffing box, etc., which varies with the size of the compressor and the method of transmission of power. For safe calculations assume the actual horse-power to be at least 1.4 times the theoretical.

$$18.5 \times 1.4 = \text{rd. } 26 \text{ h. p.}$$

H. P. BASED ON 27 LBS. BACK PRESS. AND 156 LBS. CONDENSING PRESS.

Tons refr. ..	5	10	15	20	30	50	75	100	150	200	300	500
H. P.	10	15	20	25	37	60	90	120	180	240	350	580

HORSE POWER PER CU. FT. OF AMMONIA PER MINUTE.

CONDENSER PRESSURE AND TEMPERATURE.

REFRIGERATOR PRESSURE & TEMP.	p	103	115	127	139	153	168	184	200	218
p	Temp. F	65°	70°	75°	80°	85°	90°	95°	100°	105°
4	—20°	.1809	.1916	.2022	.2128	2235	.2342	2448	.2554	2661
6	—15°	.1864	.1980	.2097	.2214	2330	2447	2563	2679	.2796
9	—10°	.1937	.2067	.2196	.2325	2454	2583	2712	2842	.2971
13	— 5°	.2001	.2144	.2287	.2430	2573	.2716	2859	.3002	.3145
16	0°	.2048	.2206	.2363	.2521	.2679	.2836	.2994	.3151	.3309
20	5°	.2083	.2257	.2430	2604	.2778	.2952	.3125	.3299	.3473
24	10°	.2096	.2286	2477	.2667	.2858	.3048	.3239	.3429	3620
28	15°	.2089	.2298	.2506	.2715	.2924	.3133	.3342	.3551	.3760
33	20°	.2054	.2282	.2510	2738	.2966	.3195	.3423	.3651	3879
39	25°	.1992	.2240	.2489	.2738	.2987	.3236	.3485	.3734	3983
45	30°	.1897	.2169	.2440	.2711	.2982	.3253	.3524	.3795	.4066
51	35°	.1768	.2062	.2357	.2651	.2946	.3241	.3535	.3830	.4124

Economy of Compression Machine.

The economy depends mainly upon the back pressure. Maximum economy is obtained at 28 lbs. suction pressure and about 150 lbs. condensing pressure. Under these conditions, for a non-

CAPACITY OF COMPRESSOR IN TONS OF REFR. UNDER DIFFERENT BACK PRESSURES.

Diameter Compressors Inches	Diameter Engine Inches	Stroke Inches	Suction or Back Pressure—Gauge					
			5 Pounds	10 Pounds	15.67 Pounds	20 Pounds	25 Pounds	30 Pounds
7½	11½	10	6	8	10	11	13	15
9	13½	12	13	16	20	23	26	29
11	16	15	19	24	30	34	39	44
12½	18	18	26	33	40	46	52	59
14	20	21	39	49	60	69	78	88
16	24	24	58	73	90	103	118	132
18	26	28	81	102	125	143	163	184
20	28½	32	114	142	175	200	229	258
22½	32	36	146	183	225	257	294	331
25	36	42	194	244	300	343	392	442
30	44	48	324	407	500	571	654	736

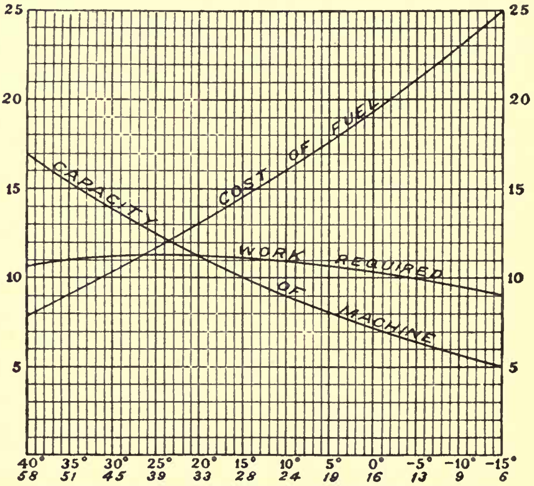
condensing steam engine, consuming coal at the rate of 3 lbs. per hour per I. H. P. of steam cylinders, 24 lbs. of ice-refrigerating effect are obtained per lb. of coal consumed. For the same condensing pressure, and with 7 lbs. suction pressure, which affords temperatures of 0 degrees F., the possible economy falls to about 14 lbs. of "refrigerating effect" per lb. of coal consumed.

The above table, compiled by the York Mfg. Co., gives the sizes of compressors and their capacity under different back pressures, based on 60° condensing water. The condensing pressure is determined by the amount of condensing water supplied to liquefy the ammonia in the condenser. If the latter is about 1 gallon per minute per ton of refrigerating effect per 24 hours, a condensing pressure of 150 results, if the initial temperature of the water is about 56 degrees F. Twenty-five per cent. less water causes the condensing pressure to increase to 190 lbs.

The work of compression is thereby increased about 20 per cent., and the resulting "economy" is reduced to about 181 lbs. of "ice effect" per lb. of coal at 28 lbs. suction pressure, and 11.5 at 71 lbs. If, on the other hand, the supply of water is made 3 gallons per minute, the condensing pressure may be confined to about 105 lbs. The work of compression is thereby reduced about 25 per cent., and a proportional increase of economy results.

If the engine may use a *condenser to secure a vacuum* an increase of economy of 25 per cent. is available over the above figures, making the lbs. of "ice effect" per lb. of coal for 150 lbs. condensing pressure and 28 lbs. suction pressure 30.0, and for 71 lbs. suction pressure, 17.5. In this case it may be assumed that water will also be available for condensing the ammonia to obtain as low a condensing pressure as about 100 lbs., and the economy of the refrigerating machine becomes for 28 lbs. back pressure, 43.0 lbs. of "ice effect" per lb. of coal, or for 71 lbs. back pressure, 27.5 lbs. of ice effect per lb. of coal. If a *compound condensing engine* can be used with a steam con-

DIAGRAM SHOWING ECONOMY AT DIFFERENT BACK PRESSURES.



REFRIGERATOR PRESSURE & TEMPERATURE.

sumption per hour per horse-power of 161 lbs. of water, the economy of the refrigerating machine may be 25 per cent. higher than the figures last named, making for 28 lbs. back pressure a refrigerating effect of 54.0 lbs. per lb. of coal, and for 7 lbs. back pressure a refrigerating effect of 34.0 lbs. per lb. of coal. (Prof. J. A. Denton.)

In the above diagram the line marked *capacity of machine* shows the diminished capacity as the back pressure is reduced. If the machine has a capacity of 10 tons at a return pressure of 28 pounds, as shown by vertical height of the curve, it has a capacity of 5 tons only with a return pressure of 6 pounds. Under the same circumstances the cost of fuel per ton is increased in the ratio of the vertical heights to the curve marked *cost of fuel,* namely, from 14.5 to 25. In other words the cost per ton is nearly doubled while the capacity is halved. The work as seen by the curve marked *work required* diminishes very slowly. (De La Vergne Co.)

Dry vs. Wet Compression.

A dry compression plant will need, with an expansion evaporating system: A medium size compressor; a large size evaporating system; a small amount of ammonia.

A dry compression plant will need, with a flooded evaporating system: A small size compressor; a small size evaporating system; a large amount of ammonia.

A wet compression plant will need, with a wet compression evaporating system: A large size compressor; a medium size evaporating system; a medium amount of ammonia.

According to C. Vollmann, the wet compression system has the following advantages over the dry compression system:

First. By letting the ammonia vapors return to the compressor in a partially wet state, we are enabled to work with a higher back pressure, thereby having the ammonia gas in the refrigerator pipes of a higher density than if the vapors were perfectly dry. Furthermore, we are enabled to keep the refrigerator pipes partially filled with liquid ammonia, in consequence of which the surface of the refrigerator can be materially reduced.

Second. By keeping the compressor parts at a cool temperature, the compressor draws in a greater amount of vapors than where the parts are highly overheated. With a dry compressor, although the cylinder is water jacketed, the internal parts are kept at a very high temperature, and when the dry ammonia vapors are drawn into the compressor, they immediately get heated up, and by expanding prevent the compressor from drawing in its full amount of vapors.

Third. By keeping the compressor at a cool temperature, the compressor oil which is taken into the compressor through the stuffing box cannot evaporate, but is kept in its liquid state, and as such deposited in the oil collector.

Fourth. With the wet compression system, the engineer in charge knows if sufficient ammonia is circulated through the system or not, by placing his hand on the delivery pipe. If this keeps fairly warm, a sufficient amount of ammonia is passed through the system.

In regard to Vollmann's theory (No. 2) that a larger volume of vapor could be handled by the wet compressor at each stroke, we must not overlook the fact that the interchange of heat between the ammonia and the walls of the compressor cylinder is evidently much greater than anticipated by many, as was proved in the tests made, at the test plant of the York Mfg. Co. Sixteen of these tests were made in four series of four runs each, the speeds used being 40, 60, 80 and 100 revolutions per minute in each series. The results proved that while the liquid handled is slightly less with dry compression, the cool'ng done was about fifteen per cent. more with dry than with wet compression, and further that the cooling decreases rapidly toward the lower speeds with wet compression.

Tests made with the horizontal double-acting compressor indicated that the results were even more in favor of the dry compression than those obtained previously with the vertical compressor. All the tests were made at the standard head pressure of 185 pounds, gauge, and it was observed that in comparing the tonnage made at a given back pressure for the two conditions that the difference increases rapidly as the suction pressure decreases. The tonnage made with five pounds suction pressure was nearly three times that made with wet compression at the same suction pressure, while at twenty-five pounds the difference was only about one-half more in favor of dry compression.

In a series of tests made in 1904, the results showed that the higher the temperature of the discharge gas, the more cooling was done per unit of piston displacement and per unit of power expended.

In tables I and II a comparison is made between three machines. The vertical single-acting machine of 100 tons refrigerating capacity is taken as the basis.

The wet compression machines are assumed to have 70% volumetric efficiency when operating under dry compression conditions.

TABLE NO. I.

Comparative Amount of Work that can be gotten out of 18-inch by 28-inch Compressors, under the conditions stated, and the Size and Horse Power of the Engine needed to drive each machine.

Condition	Type Machine	Compressor				Engine		
		No.	Size	Volumetric Efficiency	Tons Refrig.	Size	I. H. P.	H. P. per Ton
Dry Comp.	Vertical S. A.	2	18x28	80%	100	26 x28	170	1 7
Dry Comp.	Horiz. D. A.	1	18x28	70%	88	26 x28	171.6	1 95
Wet Comp.	Horiz. D. A.	1	18x28	...	64	25½x28	167	2 61

TABLE NO. II.

Comparative Size of Compressor required to do 100 tons refrigation under the conditions stated, also the Size and Horse Power of Engine needed to drive each machine.

Condition	Type Machine	Compressor				Engine		
		No.	Size	Volumetric Efficiency	Tons Refrig.	Size	I. H. P.	H. P. per Ton
Dry Comp.	Vertical S. A.	2	18 x28	80%	100	26 x28	170	1.7
Dry Comp.	Horiz. D. A.	1	19½x28	70%	100	28 x28	195	1.95
Wet Comp.	Horiz. D. A.	1	22½x28		100	32½x28	261	2.61

Conditions:—15.67 lbs. suction pressure; 185 lbs. discharge pressure; no liquid cooling; one-quarter cut-off in steam cylinder; 90 lbs. steam pressure; and 59 revolutions per minute.

The Condenser

A large condenser surface will greatly assist the economical working of the machine. The amount of pipe depends on the temperature of the cooling water, as with warmer water a higher latent heat of the medium has to be transferred to the cooling water.

Condenser Surface.

The condenser surface equals the product of the latent heat and the amount of liquid passing the compressor per minute, divided by the heat transmission.

Example continued: How large is the surface of an atmospheric condenser for an 18-ton refrigerating machine?

$$F = \frac{h\,k}{m\,(t - t_1)}$$

Where h = latent heat of ammonia at 85° = 500; k = amount of ammonia passing the compressor p. min. (which is the product of the cubic capacity of the compressor and the weight of 1 cb. ft. of gas at the refr. temp. = $69.3 \times 0.11 = 7.6$); m = number of heat units transferred per minute per sq. ft. of iron pipe per degree of difference (m = 1 for atm. condensers, 0.8 for submerged condensers) ; t = temp. of ammonia in coils = 85° F.; t_1 = temp. of water (mean between initial of 70° and final of 80° = 75° F.).

$$F = \frac{500 \times 7.6}{1\,(85 - 75)} = 380 \text{ sq. ft.}$$

= 21 sq. ft. per ton of refrigeration.

For safe calculations employ for atm. condensers the following values:

Initial temp. of water.....	50°	55°	60°	65°	70°	75°	80°	85°
Condensing surface in sq. ft. per ton of refr..........	19	20.5	22	24	26	28	30.5	34.5

In case of *submerged condensers* we have to add 20 per cent. to the above amount of surface, as the heat transmission is 0.8 instead of 1.

Amount of Cooling Water.

By calculating the amount of cooling for above condenser we have to divide the latent heat of the liquid passing the compressor per minute (which is 7.6 lbs.) by the amount of heat which has been taken up by the cooling water (difference between the final and initial temperature of the water).

$$A = \frac{500 \times 7.6}{80 - 70} = 380 \text{ lbs. per minute.}$$

= 2.6 gal. per minute per ton of refr.

For safe calculations use the values given in the following table, based on a final temperature of water of 95° F.:

COOLING WATER PER TON OF REFRIGERATION.

Initial temperature of water	50°	⅝ gal. per minute.
" " " "	55°	¾ " " "
" " " "	60°	⅞ " " "
" " " "	65°	1 " " "
" " " "	70°	1⅕ " " "
" " " "	75°	1½ " " "
" " " "	80°	2 " " "
" " " "	85°	2¾ " " "

For submerged condensers allow at least 20 per cent. more water.

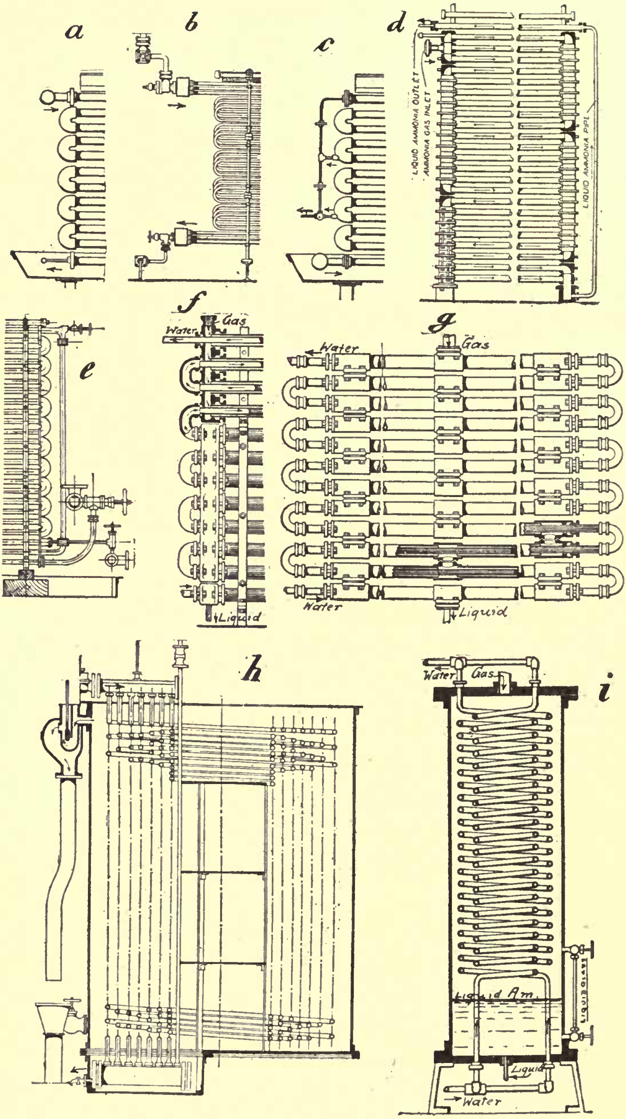


FIG. 13—VARIOUS TYPES OF AMMONIA CONDENSERS.

a, Standard top fed. c, top fed, continuous wound coil. c, bottom fed ("De La Vergne"). d, "American Linde." e, "Frick." f and g, double pipe. h, submerged condenser. i, shell and coil condenser.

Where local conditions are favorable to allow the condenser to be put on the roof and exposed to the winds, the same cooling water may be used over and over again, provided the atmospheric condenser is built sufficiently high, as it is done in Germany.

Another method to economize is by employing a cooling tower. (See notes on cooling towers.)

Builders of refrigerating machines rate the atmospheric ammonia condensers for average conditions as follows:

The Fred W. Wolf Co.: 22.5 sq. ft. per ton of refrigeration; condensers are 24—2″ pipes high by 20 fet. long.

The De La Vergne Machine Co.: 13 sq. ft. per ton of refrigeration; condensers are 18—2″ pipes high by 20 ft. long.

The Linde Co. of Germany: Submerged condensers have 32 sq. ft. for small machines of 10 to 25 tons down to 19.5 sq. ft. for machines of 100-ton refr. capacity; atmospheric condensers are 48—1¼″ pipes high (2″ centers) by 16′—7″ long.

Double pipe condensers have of late come more to the foreground. Their high efficiency is due to the perfect heat exchange, which is obtained through observing the counter-current principle. They are rated on a basis of about 14½ foot of pipe per ton of refrigeration.

Most commonly we find 2-in. pipe inside of 3-in. pipe or 1¼-in. pipe inside of 2-in. pipe. Some manufacturers prefer to circulate the cooling water through the inner pipe, some through the outer pipe. The double pipe condensers are built 18 ft. long and from 2 to 12 pipes high. For large machines take several sections, but not over 12 pipes high.

Tables No. III and No. IV give the capacities and horse power per ton refrigeration of one section counter-current double-pipe condenser. 1¼-inch and 2-inch pipe, 12 pipes high, 19 feet outside water bends, for water velocities 100 feet to 400 feet per minute: initial temperature of condensing water 70 degrees.

TABLE NO. III —*High Pressure Constant.*

Condensing Water				Capacity in tons. Refrigeration per 24 hours	Condensing Pressure. Lbs. per square inch	Horse Power per Ton Refrigeration		
Velocity through 1¼-inch pipe. Feet per min.	Total Gallons used per min.	Gallons per min. per ton refrigeration	Friction through coil. Lbs. per square inch			Engine driving compressor	Circulating water through condenser	Total engine and water Circulation
100	7.77	1.16	2.28	6.7	185	1.71	0.0016	1 7116
150	11.65	1.165	5.75	10.	185	1.71	0.004	1.714
200	15.54	1.165	9.98	13.4	185	1.71	0.007	1.717
250	19.42	1.18	15.	16.4	185	1.71	0.011	1.721
300	23.31	1.24	21.6	18.8	185	1.71	0.016	1.726
400	31.08	1.30	37.8	24.	185	1.71	0.030	1.74

Tests made at York determined the value of a square foot of condensing surface under different conditions.

The data relate only to 70° condensing water, and the values given will not be true for any other temperature or condition than those stated.

The following tables show the effect of increasing the condensing water passing through a double-pipe condenser, to do certain work. If "capacity" is the requirement, table No. III shows what can be done and what the cost in power will be. If a "re-duction in horse-power" is the requirement, table No. IV shows how to obtain it and at what expense.

TABLE NO. IV—*Capacity Constant.*

100	7.77	0.777	2.28	10.	225	2.04	0.001	2.041
150	11.65	1.165	5.75	10.	185	1.71	0.004	1.714
200	15.54	1.554	9.98	10.	165	1.54	0.009	1.549
250	19.42	1.942	15.	10.	155	1.46	0.018	1.478
300	23.31	2.331	21.6	10.	148	1.40	0.030	1.43
400	31.08	3.108	37.8	10.	140	1.33	0.071	1.401

NOTES—Above tables are based on the heat transmission obtained for various velocities of water, as averaged up from York Manufacturing Company's tests on double-pipe condensers.

The horse power per ton is for single-acting compressor and 15.67 lbs. suction pressure.

The friction in water pump and connections should be added to water horse power and to total horse power.

HORIZONTAL DOUBLE ACTING COMPRESSORS AND CONDENSERS.

COMPRESSOR.								ENGINE.			AMMONIA CONDENSER.							
									Floor Space Required for Compressor and Engine.					Space Required if Sections are placed 20-inch Centers.				
New Number.	Old Number.	Tons Refrigerating Capacity per 24 Hours.	Diameter of Cylinder, in.	Stroke of Cylinder, in.	Size of Connections, inches.	Horse Power Required.	Revolutions per Minute.	Diameter Stroke, inches.	Width.	Length.	Number of Sections.	Number of Pipes.	Length of Pipes in feet.	Length.	Width.	Height.	Liquid Receiver and Oil Separator, Length in feet.	Liquid Receiver and Oil Separator, Diameter in inches.
8	III	6	5⅞	12⅝	2	12	70	9x14	6 ft. 6 in.	x 11 ft. 9 in.	1	18	15	18 x	2 ft. 0 in.	x 9 ft. 3 in.	4	10
10	IV	12	8 1/16	12⅝	2	18	70	11x16	8 " 0 "	x 12 " 6 "	2	18	15	18 x	3 " 9 "	x 9 " 3 "	4	10
10A		15	8¼	15	2	23	70	11x16	8 " 0 "	x 12 " 6 "	2	18	17½	21 x	3 " 9 "	x 9 " 3 "	4	10
11	IV½	18	9	15	2½	27	70	10x30	9 " 0 "	x 17 " 0 "	2	18	20	24 x	3 " 9 "	x 9 " 3 "	4	10
11A		20	9½	15	2½	30	70	10x30	9 " 6 "	x 17 " 0 "	2	20	20	24 x	3 " 9 "	x 10 " 6 "	4	10
12	V	25	9⅞	16⅝	2½	38	70	12x30	9 " 6 "	x 18 " 0 "	2	24	20	24 x	5 " 0 "	x 12 " 0 "	6	10
12A		25	10¼	17	2½	38	70	12x30	10 " 0 "	x 18 " 6 "	2	24	20	24 x	5 " 0 "	x 12 " 0 "	6	10
13	V⅓	33	11	17⅝	3	50	70	12x36	10 " 0 "	x 20 " 6 "	3	24	20	24 x	6 " 8 "	x 12 " 0 "	6	10
14	V⅔	40	11	21¼	3	55	70	14x36	10 " 0 "	x 20 " 6 "	3	24	20	24 x	6 " 8 "	x 12 " 0 "	6	10
14A		40	11	21¼	3	55	70	14x36	11 " 6 "	x 20 " 6 "	3	24	20	24 x	6 " 8 "	x 12 " 0 "	6	10
15	VI	50	12 13/16	21¼	3	63	66	16x36	11 " 6 "	x 21 " 9 "	4	24	20	24 x	8 " 4 "	x 12 " 0 "	6	10
15B		50	12 13/16	21¼	3	63	66	16x36	12 " 3 "	x 21 " 9 "	4	24	20	24 x	8 " 4 "	x 12 " 0 "	6	10
15A		65	13¾	25	3½	81	63	16x42	12 " 6 "	x 23 " 9 "	5	24	20	24 x	10 " 0 "	x 12 " 0 "	8	10
16A		75	14	30	3½	94	60	18x42	13 " 0 "	x 25 " 0 "	6	24	20	24 x	11 " 8 "	x 12 " 0 "	8	10
17	VII	75	15	25	3½	94	60	18x42	12 " 3 "	x 25 " 0 "	6	24	20	24 x	11 " 8 "	x 12 " 0 "	8	10
17A		85	15	30	4	106	57	20x42	13 " 9 "	x 26 " 0 "	7	24	20	24 x	13 " 4 "	x 12 " 0 "	8	10
18	VII½	90	15½	30	4	115	57	20x42	12 " 3 "	x 26 " 0 "	7	24	20	24 x	13 " 4 "	x 12 " 0 "	8	10
18A		90	15½	30	4	115	57	20x42	14 " 0 "	x 26 " 0 "	7	24	20	24 x	13 " 4 "	x 12 " 0 "	8	10
19	VIII	100	18½	27 3/16	5	125	50	22x42	13 " 6 "	x 26 " 6 "	8	24	20	24 x	15 " 0 "	x 12 " 0 "	8	10
19A		100	16½	30	5	125	60	20x42	14 " 6 "	x 26 " 6 "	8	24	20	24 x	15 " 0 "	x 12 " 0 "	8	10
20A		120	18	30	5	150	57	22x42	15 " 0 "	x 26 " 6 "	10	24	20	24 x	18 " 4 "	x 12 " 0 "	8	10
20C		150	20	30	5	188	57	24x42	17 " 6 "	x 31 " 0 "	12	24	20	24 x	21 " 8 "	x 12 " 0 "	8	10
20B		175	21	36	5	220	50	26x48	15 " 6 "	x 29 " 0 "	14	24	20	24 x	25 " 0 "	x 12 " 0 "	8	10
21A	X	225	24	36	6	280	50	30x48	15 " 6 "	x 30 " 9 "	18	24	20	24 x	31 " 8 "	x 12 " 0 "	10	15
21B		225	21	48	6	280	45	32x48	16 " 6 "	x 31 " 0 "	18	24	20	24 x	31 " 8 "	x 12 " 0 "	10	15
15C		125	13 & 13	30 & 30	3½	156	60	18x30x30	18 " 0 "	x 30 " 0 "	10	24	20	24 x	18 " 4 "	x 12 " 0 "	8	10
19E		300	18 & 18	42 & 42	5	375	50	22x44x42	20 " 0 "	x 42 " 6 "	24	24	20	24 x	41 " 8 "	x 12 " 0 "	12	18
22A		450	21 & 21	48 & 48	6	560	45	26x48x48	18 " 0 "	x 44 " 3 "	36	24	20	24 x	61 " 8 "	x 12 " 0 "	12	20
23A		500	22 & 22	48 & 48	6	625	45	26x48x48	19 " 0 "	x 44 " 3 "	40	24	20	24 x	68 " 4 "	x 12 " 0 "	12	20
24A		600	25 & 25	48 & 48	7	750	45	32x60x48	20 " 0 "	x 44 " 6 "	48	24	20	24 x	81 " 8 "	x 12 " 0 "	12	20

The Horse Power required is based on condensing water at a temperature not exceeding 60° Fahrenheit and will vary according to temperature of condensing water and back pressure.

Slide valve engines are used with No. 8, No. 10 and No. 10A Compressors. With all other sizes we use Standard Corliss Engine.

Compressors No. 15C, No 19E, No. 22A, No. 23A and No. 24A each have two cylinders, operated by Cross Compound Corliss Engine.

CAPACITY OF SMALL COMPRESSORS. (VERTICAL SINGLE ACTING.)

Size of Compression Cylinder.	No. of Cylinders.	Revolutions per minute.	Horse Power required.	Refrigerating capacity.	Ice Making capacity.	Length over Crank Shaft.	Engine attached	Width of Bed.	Height from floor to top of Machine.
2½" x 6	1	140	1	½ Ton	¼ Ton	2' 5"	3'	2' 6"	4' 6"
3½" x 7	1	120	2	1 "	½ "	2' 5"	3'	2' 6"	4' 8"
4 " x 8	1	110	3	1½ "	¾ "	3'	3' 6"	3' 2"	5' 8"
4½" x 8	1	110	4	2 "	1 "	3' 2"	3' 6"	3' 2"	5' 8"
5½" x 10	1	100	6	3 "	1½ "	3' 6"	4' 4"	4' 8"	6' 3"
6 " x 10	1	100	8	4 "	2 "	3' 6"	4' 4"	4' 8"	6' 3"
5½" x 10	2	90	10	6 "	3 "	4' 4"	5' 6"	4' 8"	6' 3"
6 " x 10	2	90	12	8 "	4 "	4' 4"	5' 6"	4' 8"	6' 3"

NOTES ON COMPRESSION MACHINES:

PART III—APPLICATION OF MECHANICAL REFRIGERATION

Insulation

The insulation of cold storage rooms is a matter of vital importance when viewed from an economic standpoint. A large percentage of the actual work of a refrigerating machine is required to make up for transfer of heat through the walls, floors and ceilings occasioned by improper insulation.

The general rule applied to all insulation is: An air-tight surface towards the source of heat and insulating strata towards the cold side of the wall.

Attention may be called to the following points:

(1) Air is one of the best non-conductors of heat, but it must be kept still; if it is allowed room to form currents it will convey a large quantity of heat from the outer wall to the inner wall by convection, since rapid currents are formed when air is free to move between walls differing only a few degrees in temperature.

(2) Filling in with loose non-conducting material must be done with great care, since it is liable to settle in places.

(3 The penetration of air and moisture are to be specially guarded against by the use of pitch in connection with brick or stone, or paper when wood is used.

(4) Materials should be selected for insulation that are free from unpleasant odor and non-absorbent; in wood, spruce is preferred, since it is free from knots, has little or no odor, and is, at the same time, comparatively cheap.

(5) In applying wooden insulation all the joints between the boards should be laid in white lead, and triangular wooden strips with paper behind should be put in every corner in the room. The paper between the layers of boards must be carefully folded in the corners so as not to break, and laid so that the edges of the paper overlap each other.

(6) The flow of heat is nearly proportional to the difference of temperature between the inside and the outside wall; this circumstance must be taken into consideration in arranging insulation; what would be sufficient in a cold storage room to be kept at 36 degrees would be totally inadequate in a case of a freezing room to have a temperature of 5 to 10 degrees. It is a good plan to locate a freezing room inside of a cold storage room so that the difference of temperature between its inside and outside walls may be more moderate.

(7) The best insulation is none too good, and is by far the cheapest in the end.

Fireproof Cold Storage Warehouse Construction.

(J. E. Starr, A. S. R. E. Trans. 1907. Abridged.)

Three classes of fireproof construction:

Class A. Cold storage buildings erected with outer and inner walls of tile, the outer wall not carrying any weight but its own, and the floors a combination of concrete and tile, weights carried on the inner walls and partitions. Insulation between inside and outside wall a continuous fill.

Class B. Cold storage warehouse containing an inside building, with reinforced concrete columns and girders, and with floors of either reinforced concrete or combination of reinforced concrete and tile, all weights carried on columns. Outside walls either of brick or tile, or a combination of both. Inside walls of vitrified tile. Insulation between inside and outside walls a continuous fill.

Class C. Cold storage building with iron framework with weights carried partially on columns and partially on outside brick walls, all ironwork covered with fireproofing. Inside wall of vitrified tile Insulation between inside and outside walls a continuous fill.

Of *Class A* (all tile) one example may be quoted of a three-story house in Washington Court House, Ohio.

This house consisted of an outside wall of two 4-inch hollow vitrified tile, an inside wall of one course of 4-inch vitrified tile standing eight inches away from the outside wall. The floors rested on the inside wall and on the partitions which later divided the house into three sections.

The space between inner and outer walls was filled with granulated cork, making an unbroken fill from bottom to the small garret, or a circulating air space between the top floor of the cold rooms and the roof. The top of this filled space was closed with tile which could be easily taken off, so that if any settling occurred it might be observed and filled in.

Experience of four years has shown, however, that little, if any, settling occurs. Experience in filling an 8-inch space showed that the cork would not "bridge" and leave voids in the 8-inch space even when filled from a height of twenty or thirty feet.

The inside wall was therefore entirely surrounded by insulation and no heat could pass through it without first passing through the cork, except at the very small areas where the inside and outside wall were tied by extending the partions through to the outside wall.

The tile was laid up in cement mortar and panels of outside wall surface 25 feet wide and 33 feet high have successfully withstood wind pressure and all outside influence.

In this particular building the floors were of the well known Johnson type. This consists of a reinforced concrete tension member, about one inch thick, covering the entire span or "bay." On top of these two courses of 6-inch tile was laid a finished cement wearing floor.

It will be observed that this method of construction places the tile in compression while the thin concrete with its strengthening rods and web are in tension.

Long spans can thus be successfully built to carry far in excess of the maximum cold storage load of 400 pounds per square foot.

Partitions were made with double 4-inch tile with from six to eight inches of cork filled space between.

The first building of *Class B* was nine stories high and was built in St. Paul, Minn.

The building proper was entirely carried on columns very much as our present skyscrapers are built, excepting that the columns were all of reinforced concrete, and the outer skin was not carried on the outside girders as in the case of office buildings, but was entirely independent of the main structure and standing about eight feet away from it at all points.

The outside wall was only 12 inches thick from bottom to top, but was reinforced by an imbedded "I" beam framework.

There was only about a square inch of conducting material between the outside wall and the inside structure at the head of each column and its conducting effect practically nil as compared to the total.

As the floors and outside columns and girders were thus about eight inches from the outside wall it was only necessary to build from floor to ceiling a 4-inch vitrified tile wall and fill the 8-inch space with the non-conducting material giving the same continuous insulation as at first described in case of *Class A*.

The outside wall was thoroughly waterproofed by a thick odorless coating on the inside (which may be in time followed up by an outside water proofing).

The floors in this building were 6-inch reinforced concrete or reinforced concrete girders and beams in spans.

The insulation of floors was made on top, using either Lith or cork board from two to four inches thick, depending on conditions.

These insulating boards were laid on the floor slap, well "doped," with odorless pitch and waterproofed on top. Over this a two-inch concrete floor was laid, reinforced with a wire web and the whole finished off with a ½-inch wearing floor of cement and sand rendered waterproof.

Partitions were of double 4-inch hollow tile with insulating filled space between from four to eight inches.

Under the *Class C* of construction comes the cold storage building of the Murphy Storage & Ice Co., of Detroit. This was a ten-story building constructed with built-up steel columns and with steel girders running longitudinally with the greatest dimension of the building, the end of girders resting on the walls, and with "I" beams running between the girders and from the girders to the walls on a spacing of a little over four feet. The walls therefore carried their share of the weight of the outside spans. The floors were of a combination tile and concrete.

Four-inch tile walls were built from floor to ceiling flush with the edge of this floor, leaving, therefore, a continuous fill from top to bottom eight inches thick, excepting where the "I" beams ran into the wall at each story on centers of a little over four feet.

The ends of the "I" beams which projected through the 8-inch space between the edge of the floor and the outside wall were carefully wrapped with hair felt dipped in an odorless compound and made a tight joint with the outside wall.

The inner surfaces of the outside wall were coated continuously from top to bottom with a thick coat of odorless waterproofing material and the inside 4-inch wall was built up in the same manner as described for *Classes A* and *B* and the space between filled with granulated cork.

The columns and "I" beams, wherever exposed, were covered with a hollow tile fireproofing, plastered on the outside. The partitions were constructed of double walls of hollow tile with a fill of from four to eight inches of insulating material between, as in the case of the other houses described. The floors were also insulated, as before described, by laying from two to four inches of lith board on the floors, thoroughly "doped" and waterproofed with a 2-inch course of concrete on top, reinforced with wire netting and a finishing course of ½-inch of well troweled cement and sand.

The floors on all three classes of these buildings were finally waterproofed by a concrete filler and a concrete paint presenting a glassy surface, and impervious to water.

All of the storage rooms in these buildings were singularly free from odor, and the air was unusually keen and sweet as compared with buildings constructed with wooden insulation, as all of the surfaces were either of vitrified tile or waterproofed concrete, neither of which absorb or give out odors. It may also be pointed out that the continual passing of the air over the calcium brine surfacers greatly purified the air, as it has been proven that chloride of calcium is quite effective as a germicide. The researches on this subject conducted by Dr. O. Profe, Dr. Hesse and other German authorities show conclusive results on this point.

All doors throughout all of these buildings were covered with either galvanized iron or tin in accordance with the underwriters' specifications.

It was ascertained that where buildings were divided into separate fire risks, the conduction from one floor risk to the other through the continuous girders could be best avoided by placing the skeleton framework of each fire risk entirely on its own column, instead of using a common column between the two fire risks. This allows a continuous fill of insulating material between each fire risk.

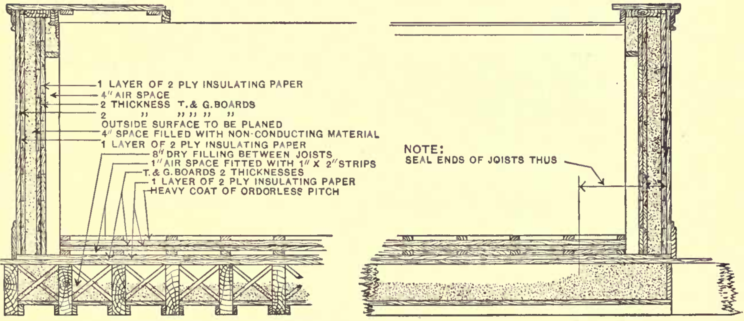
It has been proven conclusively that almost any of the insulating materials in common use when put up between fireproof walls of tile or brick do not contain sufficient air to support combustion in case of fire playing on the inner or outer wall. Tests have been made by making an opening of good size in outer wall, exposing the insulation, and building a hot bonfire on the outside immediately against the opening, and continuing the test for several hours. At the end of the test it was found that the insulation was only charred a few inches back from the opening.

In a general way it may be stated that the cost of the buildings per cubic foot, fully insulated, will run, if anything, less than the cost of a wooden building whether of the ordinary girder or floor beam type, or of mill construction, or of a combination of iron and wood, and that the general method here described of practically constructing the inside of a building with a continuous course of insulation all around has entirely obviated many of the difficulties which might be apprehended in the use of these materials.

The fire risk is also a very important feature as the first asking rate on these buildings was only 40c. on contents, which is only about 1-3 the average rate on wooden or mill constructed buildings, and in some cases ¼ the rate. As to the buildings themselves, the owners as a rule feel that they are practically indestructible and carry their own insurance.

A comparison of the fire risk in a fireproof cold storage warehouse with the average so-called fireproof building is not a fair one on account of the fact that there are practically no openings into the main part of the warehouse, while the average fireproof office building is vulnerable in a general conflagration, owing to the fact that a very large percentage of its outside surface is made up of window openings, and that it is divided into small rooms containing in the doors, trim and other woodwork a large amount of inflammable material.

TANK INSULATION.



LONGITUDINAL SECTION. TRANSVERSE SECTION.

FIG. 14.

TRANSMISSION OF HEAT THROUGH $1\frac{1}{2}''$ TO $2''$ IRON PIPES PER SQ. FT. PER HOUR PER DEGREE OF DIFF. IN TEMP.

Mode of Operation	B. T. U.	Example.
Ammonia gas inside, water outside	50	*Submerged Condenser.*
Ammonia gas inside, running water outside	60	*Atmospheric Condenser.*
Ammonia gas inside, brine outside	25	*Brine Tank.*
Ammonia gas inside, wort outside (counter current)	60	*Dir. exp. beer cooler.*
Ammonia gas inside, air outside	2-8	*Direct expansion.*
Cold brine inside, water outside	80	*Water Cooler.*
Cold brine inside, water outside	60	*Distilled Water Cooler.*
Cold brine inside, wort outside	70	*Brine Beer Cooler.*
Cold brine inside, wort outside (counter current)	75	*Baudelot Cooler with brine.*
Am. liquor inside, water outside (counter current)	60	*Absorber.*
Am. liquor inside and outside (counter current)	50	*Exchanger.*
Water inside and outside (counter current)	50	*Exchanger.*
Steam inside, water outside (counter current)	500	*Steam Condenser.*
Steam inside, water or am. liquor outside	300	*Am. Liquor Still.*
Steam inside, air outside	2-3	*Steam pipes.*

TRANSMISSION OF HEAT THROUGH VARIOUS INSULATIONS PER SQ. FT. IN 24 HOURS PER DEGREE OF DIFF. IN TEMP.

	B.T.U.
2 boards with paper, 1 inch air space, 5 inches Nonpareil sheet cork, paper, board	0.9
1 board with paper 3 inches Nonpareil sheet cork, paper, board	2.1
1 board with paper, 2 inches Nonpareil sheet cork, 2 boards with paper	3
2 boards with paper, 4 inches granulated cork, 2 boards with paper	1.7
1 board, $2\frac{1}{2}$ inches mineral wool, paper, board	3.62
1 board, paper, 1 inch mineral wool, paper, board	4.6
2 boards with paper, 8 inches mill shavings, paper, 2 boards with paper, dry	1.35
Same, damp	2.1
1 board, 2 inches air space, board, 2 inches "Lith," paper, board	1.8
4 boards, 1 inch flax sheet lining, 2 papers	2.3
1 board, 6 inches silicated strawboard (air cell), layer of cement	2.5
4 boards, 4 quilts of hair	2.52
2 double boards with 2 papers, 1 inch hair felt	3.32
1 board, paper, 2 inches calcined pumice, paper, board	3.4
1 board, 2 inches pitch, board	4.25
4 double boards with paper (8 boards) and three ⅜ inches air spaces	2.7
2 double boards with paper (4 boards) and 1 inch air space	3.71
4 boards with 2 papers, solid, no air space	4.28
Brickwall, 3 inches, hollow tile, 4 inches mineral wool, 3 inches hollow tile, cement plaster	0.7
Concrete floor, 3 inches book tiles, 6 inches dry underpiling, double space hollow tile arches, cement plaster	0.8

TABLE OF RELATIVE VALUE OF NON-CONDUCTING MATERIALS.

Loose Wool	3.35
Loose Lampblack	1.12
Geese Feathers	1.08
Felt, Hair or Wool	1.
Carded Cotton	1.
Charcoal from Cork	.87
Mineral Wool	.83
Fossil Meal	.79
Straw Rope, wound spirally	.77
Rice Chaff, loose	.76
Carbonate Magnesia	.76
Charcoal from Wood	.75
Paper	.74
Cork	.71
Sawdust	.68
Paste of Fossil Meal and Hair	.63
Wood Ashes	.61
Wood, across grain	.55
Loam, dry and open	.55
Chalk, ground, Spanish white	.51
Coal Ashes	.49
Gas-house Carbon	.47
Asbestos Paper	.47
Paste of Fossil Meal and Asbestos	.47
Asbestos, fibrous	.36
Plaster of Paris, dry	.34
Clay, with vegetable fibre	.34
Anthracite Coal, powdered	.29
Coke, in lumps	.27
Air Space, undivided	.22
Sand	.17
Baked Clay, Brick	.07
Glass	.05
Stone	.02

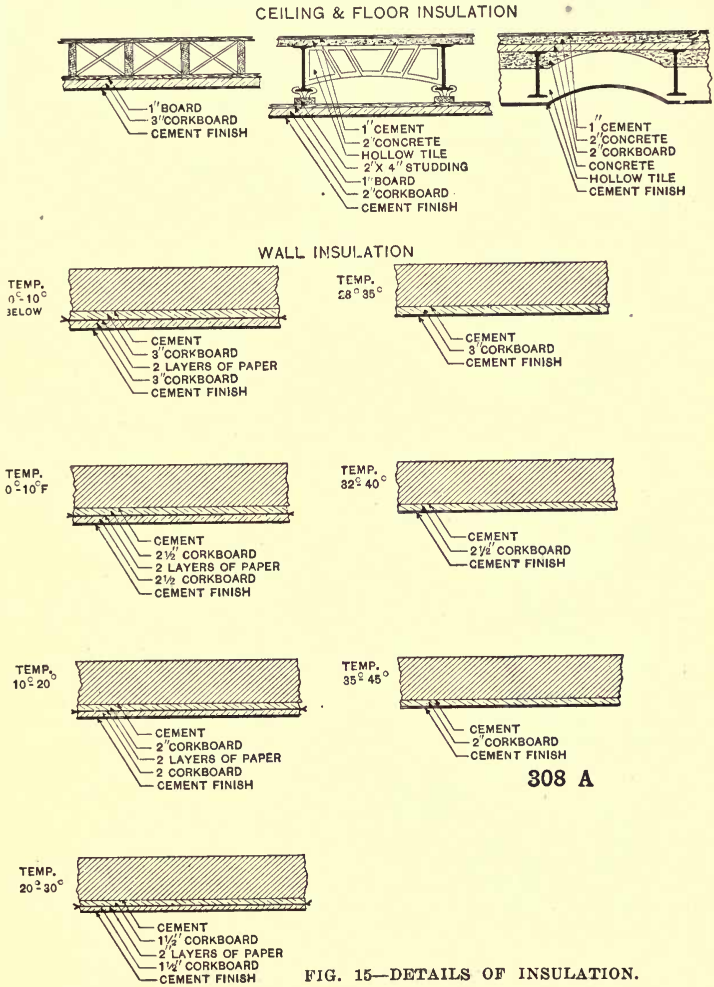


FIG. 15—DETAILS OF INSULATION.

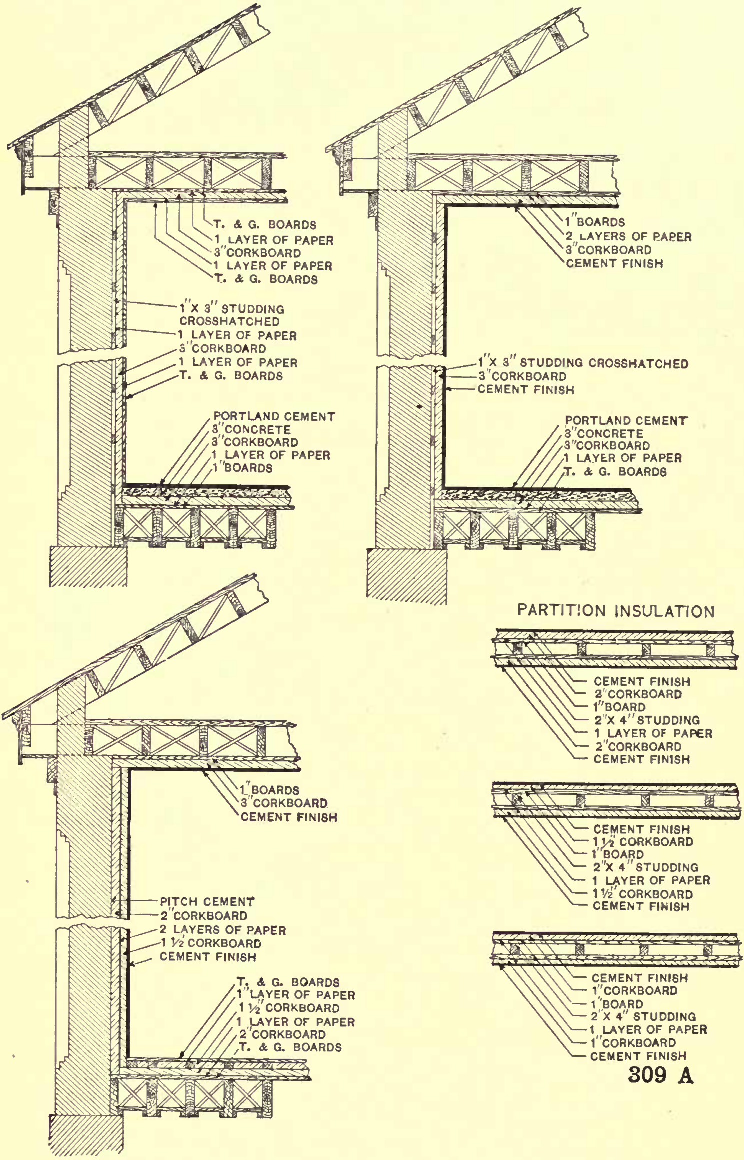


FIG. 16—DETAILS OF INSULATION.

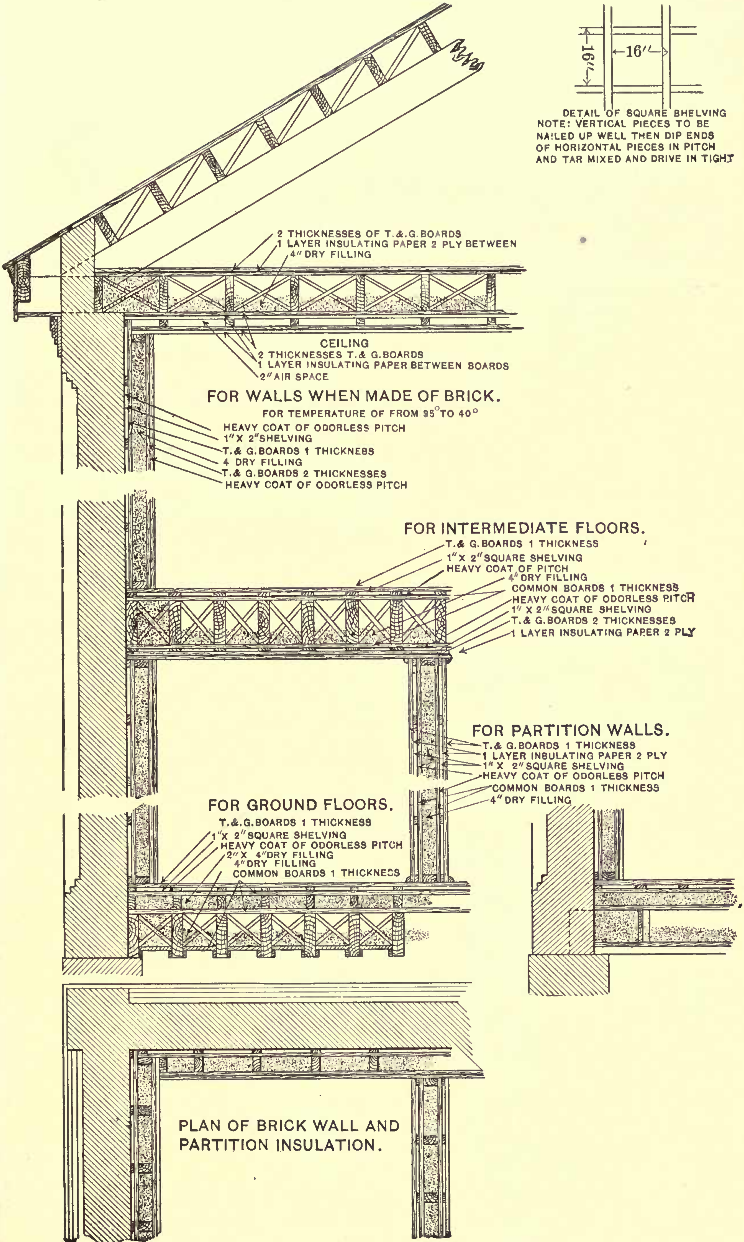


FIG. 17—DETAILS OF INSULATION.

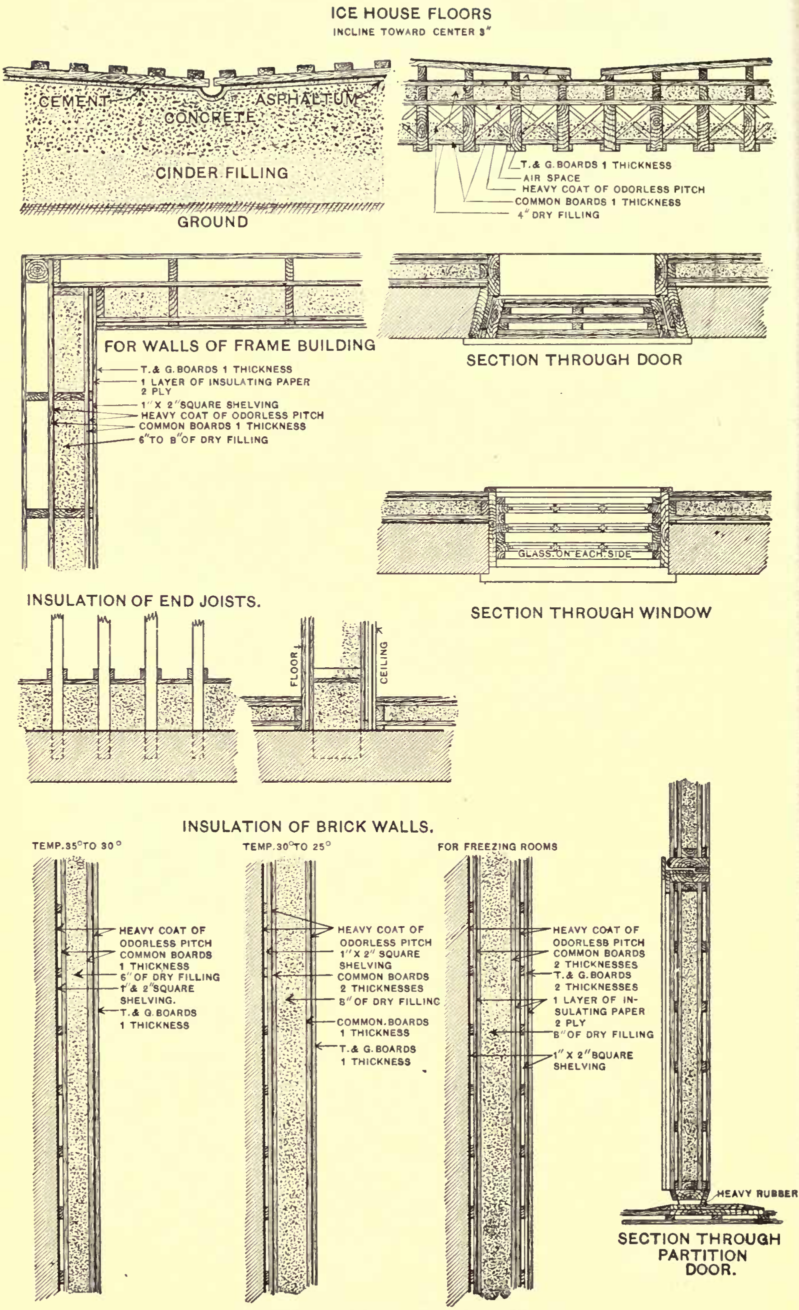


FIG. 18—DETAILS OF INSULATION.

NOTES ON INSULATION:

General Cold Storage

Cold storage comprises the preservation of perishable articles by means of low temperature. Refrigeration is produced by *direct* or *indirect* expansion or *forced air circulation.*

COLD STORAGE TEMPERATURES.

ARTICLES	°Fahr	ARTICLES	°Fahr	ARTICLES	°Fahr
FRUIT		**FISH**		**VEGETABLES**	
Apples	30-32	Fresh Fish	25-30	Asparagus	33-35
Bananas	34-36	After Freezing	18-20	Cabbage	32-34
Berries, fresh	36-40	Dried Fish	35-40	Carrots	33-34
Cranberries	33-34	Oysters in shell	35-40	Celery	32-34
Cantaloupes	33-40	Oysters in tubs	30-35	Dried Beans	35-45
Dates, Figs, etc.	50-55	**CANNED GOODS**		Dried Corn	35-45
Fruits, dried	35-40	Sardines	35-40	Dried Peas	35-45
Grapes	34-36	Fruits	35-40	Onions	32-33
Lemons	36-40	Meats	35-40	Parsnips	32-33
Oranges	34-36	**BUTTER, EGGS, ETC.**		Potatoes	34-36
Peaches	32-33	Butter	12-18	Sauerkraut	35-38
Pears	32-33	Butterine	20-32	**MISCELLANEOUS**	
Watermelons	32-36	Cheese	30-34	Cigars, Tobacco	35-42
MEATS		Eggs	29-32	Furs, Woolens, etc	25-35
Brined	35-40	**LIQUIDS**		Honey	36-40
Beef, fresh	33-35	Beer, Ale, etc., bbl'd	33-42	Hops	32-34
Beef, dried	36-40	Beer, etc., bottled	45-50	Maple Syrup, Sugar	40-45
Calves	32-34	Cider	30-40	Oils	35-45
Hams, Ribs, Shoulders, (not brined)	30-34	Ginger Ale	35-40	Poultry dressed, iced	28-30
Hogs	29-32	Wines	40-45	Poultry, dry picked	26-28
Lard	35-40	**FLOUR AND MEAL**		Poultry, scalded	20-26
Livers	33-34	Buckwheat Flour	36-40	Game, to freeze	0-5
Sheep, Lambs	32-33	Corn Meal	36-40	Game after frozen	10-28
Ox-tails	30-32	Oat Meal	36-40	Poultry to freeze	0-5
Sausage casings	38-45	Wheat Flour	36-40	Poultry after frozen	10-28
Tenderloins, Butts, etc.	33-34			Nuts, in shell	35-40
				Chestnuts	33-35

Refrigeration Required.

For *rough estimates* the following table by Siebel, based on an outside temperature of 80 to 90° F., is of good practical use:

CUBIC FEET PER TON OF REFR. IN 24 HOURS.

Size of Building in Cubic Feet.	Temperature. 0°	10°	20°	30°	40°	50°
100	150	600	800	1,000	1,600	3,000
1,000	500	2,500	3,000	4,000	6,000	12,000
10,000	700	3,000	4,000	6,000	9,000	18,000
30,000	1,000	5,000	6,000	8,000	13,000	25,000
100,000	1,500	7,500	9,000	14,000	20,000	40,000

This table is based on first-class insulation; when insulation poor, double amount of refrigeration.

For *accurate estimates* the required refrigeration has to be calculated as follows:

Calculated Refrigeration.

By calculating the required refrigeration in a given case, we must consider the following points:

(a) *To cool the goods* from the temperature at which they enter the storage room down to the desired temperature. Example, to cool 30,000 lbs. of fresh meat a day from 95° to 35°, with an outside temperature of 85°.

$R_1 = P\,(t - t_1)\,s \qquad s = \text{spec. heat (on an average} = 0.8)$

$$= \frac{30{,}000\,(95 - 35)\,0.8}{24}$$

$= 60{,}000$ units per hour.

If the goods are cooled below 32° F., that is, frozen, the specific heat changes. (See table on Specific Heat.)

(b) *To offset radiation* through walls and floors.

The loss of cold is the total exposed area multiplied with the difference in temperature and the respective factors of heat transmission, which for average insulation can be taken as 3 units per degree of difference in temperature in 24 hrs. (See chapter on Insulation.)

Example: Chill room, 40 × 50 × 10 = 20,000 cb. ft.
Side walls of room = 1,800 sq. ft.
Ceiling and floor = 4,000 sq. ft.

Total surface = 5,800 sq. ft.

$$R_2 = A\,(t - t_1)\,3 = \frac{5{,}800\,(85 - 35)\,3}{24}$$

$$= 36{,}250 \text{ units per hour.}$$

(c) To offset loss of cold through opening of doors, etc.

Calculation is approximately 5 to 8% of total refrigeration (small boxes considerably more). Provide ante-rooms or gangways.

$$R_3 = \text{approx. } 7{,}850 \text{ units per hour.}$$

Loss through lights and the presence of persons may be calculated as follows:

Heat developed in one hour:
One workingman = 500 units.
One gas light = 3,600 units.
One incandescent light of 16 c. p. = 160 units.
One ordinary candle = 450 units.

Electric light preferable, as well as being convenient for turning on and off.

(d) An extra amout of refrigeration is required, where *forced air circulation* is used and the *total air is renewed* about 4 to 6 times daily. To maintain the conditions in the room as uniformly as possible, the renewal of the air should be continuous.

The loss of cold through air renewal depends upon the difference of in and outside temperature, frequency of air renewal and percentage of humidity of inner and outer air.

Example:

(1) *Refrigr. r_1 to precipitate the difference in moisture.*

The air leaves at 35° and 70% humidity and new air enters at 85° and 80% humidity.

One cb. ft. of air at 85° and 80% hum. contains 13 × 0.8 = 10.4 grains of moisture.

One cb. ft. of air at 35° and 70% hum. contains 2.44 × 0.7 = 1.7 grains of moisture.

As one pound of vapor contains 7,000 grains, the latent heat of one grain of moisture

$$= \frac{1090}{7000} = 0.15576 \text{ units.}$$

If the air is changed 6 times daily, it means

$$\frac{20{,}000 \times 6}{24} = 5000 \text{ cb. ft. of air in one hour.}$$

$$\text{Refrigeration } r_1 = 5000 \times 0.15576\,(10.4 - 1.7)$$
$$= 6780 \text{ units.}$$

(2) Refrig. r_2 to cool the air from 85° to 35°.

Weight of 1 cb. ft. dry air at 35° and atm. press. = 0.087.

Spec. heat of air at constant press. = 0.2375.

$$r_2 = 5000 \times 0.087 \times 0.2375\,(85 - 35)$$
$$= 5000 \text{ units.}$$
$$R_4 = r_1 + r_2$$
$$= 11{,}780 \text{ units per hour.}$$

This loss of cold is reduced to about 50% by providing a heat exchanger between the outgoing and incoming air, consisting of air ducts separated by thin sheet metal partitions.

$$R_4 = 5900 \text{ units per hour.}$$

Total amount of refrigeration $= R_1 + R_2 + R_3 + R_4 =$

$$R = 110{,}000 \text{ units per hour.}$$

If air at 35° and 70% humidity shall be reduced in the cooler to 21° and 70%, the reduction of temperature requires per cb. ft.

$$= 0.02\,(35 - 21) = 0.28 \text{ units.}$$

And to dry the air:

$$= 0.15576\,(2.44 \times 0.7 - 1.36 \times 0.7) = 0.117 \text{ units.}$$

A total of $0.28 + 0.117 = 0.4$ units per cb. ft.

Consequently $\dfrac{110{,}000}{0.4} = 275{,}000$ cb. ft. must pass every hour through the cooler, what would correspond to $\dfrac{275{,}000}{20{,}000} =$ nearly 14 air circulations of total cubic contents every hour.

The *area of main air ducts* will be, by assuming a velocity of 15 ft. per second

$$= \frac{275{,}000}{15 \times 3{,}600} = \text{about 5 sq. ft.}$$

The fan will require, assuming that 0.25 H. P. takes care of 35,000 cb. ft.

$$= \frac{275{,}000}{35{,}000} \times 0.25 = \text{about 2 H. P.}$$

As one H. P. is equivalent to 2,565 units, which are directly introduced into the circulated air, we have to correct the total amount of refrigeration by $2 \times 2{,}565 = 5{,}130$ units.

$$110{,}000 + 5{,}130 = 115{,}130 \text{ units per hour.}$$

$$= \frac{115{,}130 \times 24}{284{,}000}$$

= about 9.4 tons of refrigeration in 24 hrs.

Piping.

The pipes should be so arranged as to induce air circulation (see Fig. 19). Gutters and drip pans provided where necessary.

CUBIC FEET PER FOOT OF 2″ DIR. EXP. PIPE.

Size Bldg. in Cub. Ft.	Temperature. 0°	10°	20°	30°	40°	50°
100	0.5	2.3	3.6	4.5	6.5	9
1,000	1.8	7	10.6	14	20	33
10,000	3	10.5	17	22	30	48
30,000	3.5	14	23	30	42	68
100,000	4.5	17	28	37	56	100

These ratios are based on first-class insulation; when insulation Fig. 19). Gutters and drip pans provided where necessary.

No more than 1,200 feet 2″ pipe in one expansion.

For 1″ pipe use 1.8 times amount of 2″ pipe.

For 1¼″ pipe use 1.44 times amount of 2″ pipe.

When using disks, multiply amount of pipe with 4/7.

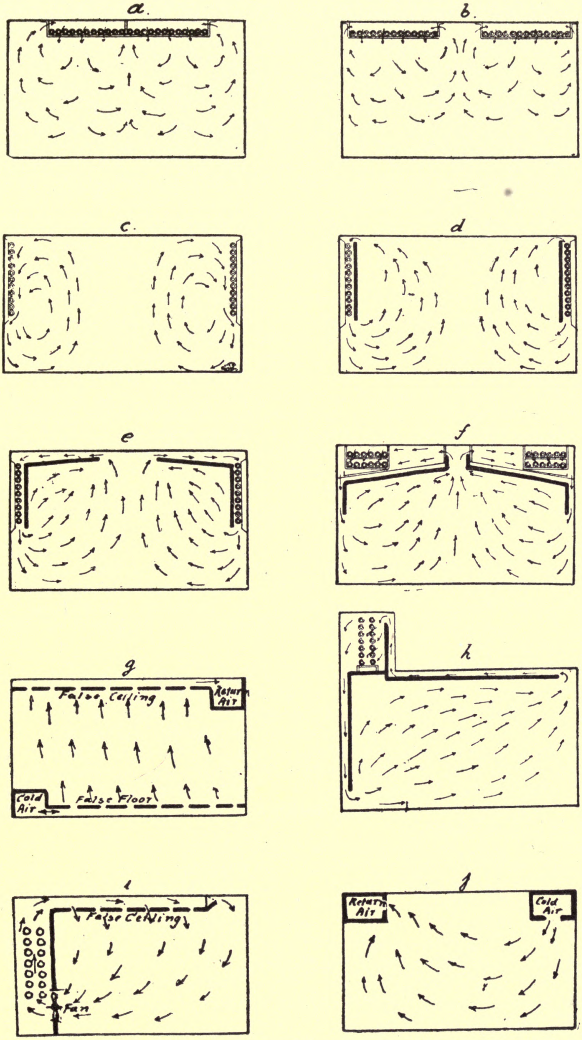


FIG. 19—ARRANGEMENT OF COOLING PIPES AND AIR DUCTS TO INDUCE AIR CIRCULATION.

a, b, pipes on ceiling; c, d, e, pipes on wall; f, h, pipes in overhead lofts; g, i, j, forced air circulation.

Brine Cooling System.

For *indirect expansion* (brine cooling) use 1½ times amount of pipe.

Brine Tank.—The size of the brine tank is calculated by allowing about 60 cb. ft. of brine per ton of refr.

The amount of expansion pipe in the tank is often taken equal to the amount of a submerged condenser. For safe calculation allow 120 *to* 150 *ft. of 2-inch pipe* (or its equivalent in other sizes) per ton of refrigeration in 24 hrs. In case of ice-making. double amount.)

The *coil and shell brine cooler* is based on 15 sq. ft. of pipe surface per ton of refr.

Brine Pump.—Velocity about 60 ft. per min. Builders usually figure the area of brine main by assuming one sq. inch per ton of refr. and a discharge of the pump = 4 gals. per min. per ton of refr.

For general cold storage purposes the direct expansion system may be well recommended, provided that the temperatures of the different rooms are almost the same and that the pipe runs are short. Long runs are liable to leak and, by discharging ammonia in the room, spoil the goods. Great care, therefore, must be taken by having only first-class pipe work and fittings used. The flanges must be soldered on the pipes, so as to make solid joints, and should be made male and female, so as to prevent the lead gasket from being blown out. If, however, the rooms are kept at widely different temperatures, it is difficult to regulate the ammonia so that it will flow evenly through all the rooms. The reason of this is found in the fact that ammonia tries to settle down in the coldest place it can find. If, for example, one room is kept at 20 degrees and the other at 40 degrees and both to be cooled in the same time by the same machine, the ammonia has the disposition to collect in the pipes of the coldest room. If the engineer in charge does not watch carefully, the pipes in the coldest room will fill with liquid ammonia, and hardly any ammonia is left in circulation.

Forced Air Circulation.

The cooling pipes (direct or indirect exp.) are calculated as above. They are arranged in a special chamber, which is connected with the rooms to be cooled by wooden air ducts. A fan or blower is provided which draws the air from the highest part of the room and forcing it through the cooler, brings it in contact with the cold coils, where it is cooled and dried. The cooled air leaves the cooler and is discharged back into the rooms from which it was taken.

The necessity of having two series of coils for successful, con-'inued operation, and the trouble of thawing off one of them and removing the drip-water, led to the construction of the "wet cooler." The refrigerating coils are arranged vertically with a gutter provided on the top of each to hold the brine. The brine is showered over the pipes and collects in a pan, from which it is drawn by a small centrifugal pump and returned to the gutter to be showered again over the pipes. The whole apparatus, which usually stands over the cold room, is enclosed in a well insulated chamber.

Instead of pumping brine over the expansion coils, Madison Cooper places Calcium Chloride in the gutters above the pipe coils. This Calcium, being highly hygroscopic, absorbs the moisture of the air and forms a strong brine, which trickles over the pipes.

The construction of air coolers must be so that a duct from the open air to the suction side of the fan is provided, through which fresh air can be drawn and led into the cool room when required. This duct can also be made use of if the cold room is needed in winter, when cold air from outside alone is blown into it.

In order to be able to warm the air in severe winter weather a series of steam coils is arranged on the delivery side of the fan. This method has not been found to answer well in very cold weather, because the air blown into the cold room through the lower air duct rises quickly upward and is led away by the upper duct without producing much effect, and the air remains almost unchanged in the lower part of the room. To obtain a sufficient supply of air for a very cold winter day there must be a third air duct laid on the floor of the cold room for carrying off the warm air at the same time that some passes out through the suction duct.

The air ducts are generally made of galvanized iron, which have to be, where the ducts run through the engine house or other warm places, properly insulated or they are made of tongued and grooved boards, saturated with chloride of zinc or protosulphate of iron. The American Linde Company gives the following rules:

The boards are planed smooth and laid close together and are supported by knee frames about 2″ × 1″ every 10 feet and fillets attached to the side wall and ceiling. The inside of the ducts is left perfectly smooth to avoid friction and eddy currents. The air is admitted and discharged through 10″ × 6″ openings, conveniently spaced along the ducts, the deliveries being in the bottom of the supply ducts and the suction duct holes on the side. The openings are fitted with hardwood doors, sliding in rebated runners, and afford an opportunity for regulating the amount of air and consequently the degree of cold in any room, irrespective of another, without the necessity of altering the speed of the fans or the temperature of the brine.

NOTES ON GENERAL COLD STORAGE:

Brewery Refrigeration

The process of making beer briefly consist of *malting* and *brewing*. *Malting* consists of:

1. Steeping the barley in water to supply moisture enough to cause it to germinate, when it is called *"malt."*

2. Drying the malt on a kiln by hot air.

Brewing consists of:

1. Mashing or mixing the malt, after it is ground, with water, the mixture being called *"wort."*

2. Boiling the wort in the brew kettle.

3. Cooling the hot wort in the *beer cooler.*

4. Fermenting the same in the *fermenting tubs.*

5. *Racking and storing.*

The boiling beer wort, coming from the *brew kettle,* Fig. 20,

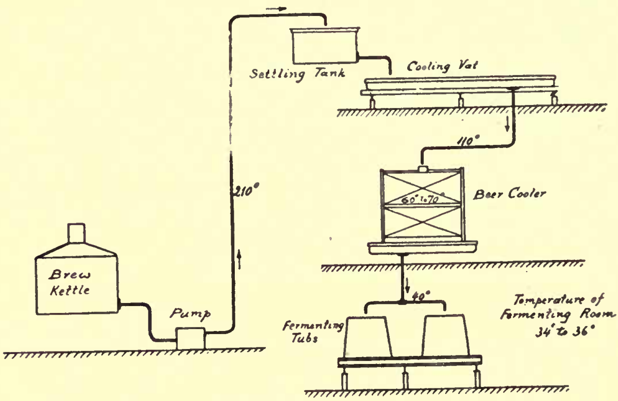


FIG. 20—DIAGRAM OF BREWING BEER.

is pumped into the *settling tank,* from where it flows into a *cooling vat,* exposed to the atmosphere (usually on the roof), where the wort is cooled down to about 110° F.

Being cooled to 40° F. (ale to 55°) in the *beer cooler,* it enters the *fermenting tubs,* where the heat developed by the fermentation of the wort is withdrawn by ATTEMPORATORS.

Refrigeration is applied to, (a) *beer cooler,* (b) *attemporators,* (c) *cellars and hop room.*

Beer Cooler.

The beer cooler (Baudelot cooler) consists of two sections, the upper section, through which well or hydrant water flows, which cools the wort down to 70° or 60° *Fahr.,* and the lower section, which cools the wort down to 40° *Fahr.* by means of cooled brine or direct expansion pipes (sometimes ice water).

Pipes are of 2-inch polished iron pipe. The cooling which is imparted to them by the wort prevents rusting. Pipes covered with copper are sometimes rendered non-conducting by lack of contact between pipe and copper covering.

DIMENSIONS OF LOWER SECTION OF BEER COOLER USING DIRECT EXPANSION.

Final Temperature of Wort 40° Fahr.

Twenty-four pipes—Initial temp. of wort 90°—

20 ft.	long	for	100	bbls.	per	hour	require	120	ton refr.
16 "	"	"	80	"	"	"	"	95	" "
12 "	"	"	60	"	"	"	"	70	" "

Twenty pipes—Initial temp. of wort 80°—

20 ft.	long	for	100	bbls.	per	hour	require	100	ton refr.
16 "	"	"	80	"	"	"	"	75	" "
12 "	"	"	60	"	"	"	"	58	" "

Sixteen pipes—Initial temp. of wort 70°

20 ft.	long	for	100	bbls.	per	hour	require	70	ton refr.
16 "	"	"	80	"	"	"	"	57	" "
12 "	"	"	60	"	"	"	"	43	" "

Twelve pipes—Initial temp. of wort 60°—

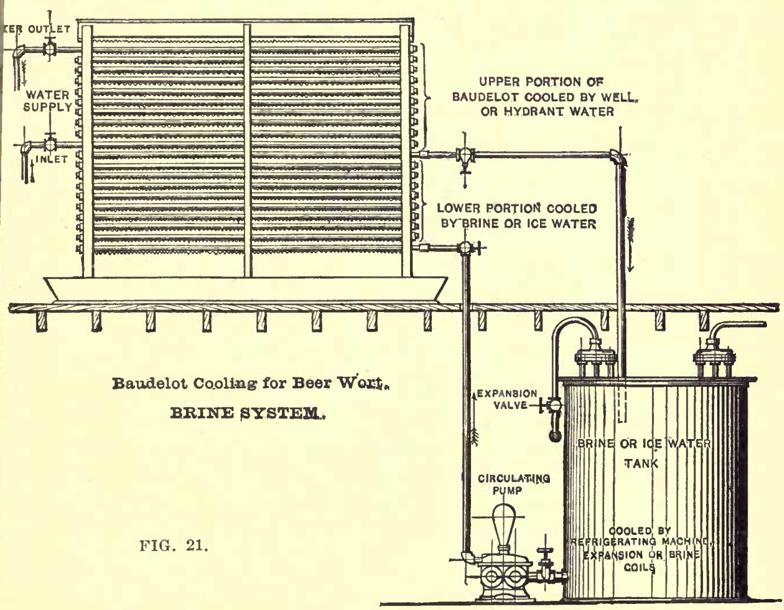
20 ft.	long	for	100	bbls.	per	hour	require	48	ton refr.
16 "	"	"	80	"	"	"	"	39	" "
12 "	"	"	60	"	"	"	"	30	" "

These figures are based *on five barrels of wort per hour per foot of pipe.*

If the cooling, as usually, is to be done in three hours, allow only one-third of the pipe.

One barrel equals 32 gallons, or 265 lbs.

In case of brine, add 20 per cent. pipe surface.



Baudelot Cooling for Beer Wort.

BRINE SYSTEM.

FIG. 21.

One hundred barrels of wort require 125 *lbs. of cooling water at* 56° *on upper section.*

One ton refrigeration required for twenty-five barrels of beer.

Attemporators.

The attemporator coils are suspended (mostly with swivel joints) in the fermenting tubs. They are made of iron, brass or copper, and of 1¼, 1½ or 2-inch size. Diameter of coil, about two thirds of tub.

Attemporators in cylinder form are usually made in two sizes:

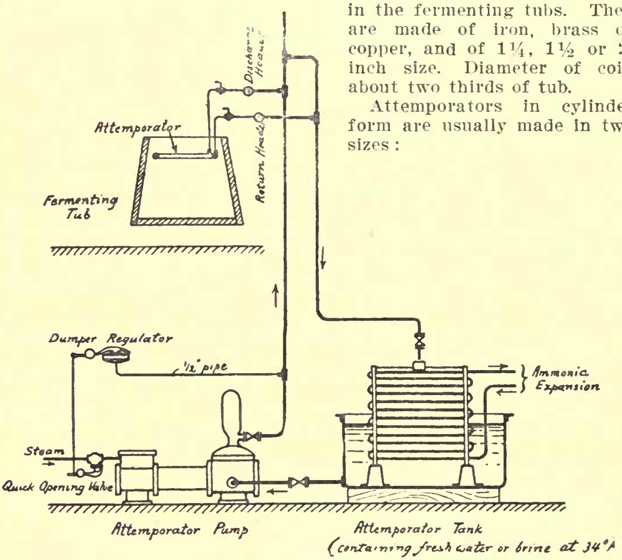


FIG. 22—ATTEMPERATOR SYSTEM.

18″ diam. × 18″ high, cooling surface, 14¼ sq. ft.
36″ diam. × 30″ high, cooling surface, 47 sq. ft.

100 barrels of wort require 12 *square feet of pipe surface* (19 *feet 2-inch pipe*).

The refrigeration is produced by means of *cooled fresh water* (safer in case of leaks) or brine (cheaper) circulated through the attemporators at about 34° Fahr.

Expansion pipe in attemperator tank about 12 *square feet of pipe surface per* 100 *barrels wort.*

Provide standpipe and pump regulator.

Piping of Cellars and Hop Room.

RATIOS FOR ALE BREWERIES.
2″ pipe direct expansion with 14″ disks per foot.

Room.	Temp. of Room.	10,000	15,000	20,000	30,000	40,000
		Size of Room in Cubic Feet.				
Fermenting	50°—60°	1:50	1:50	1:55	1:60	1:70
Vat or Ale Stor..	45°—50°	1:40	1:40	1:42	1:45	1:50
Ale Chip........	45°—50°	1:40	1:45	1:50	1:55	1:60
Ale Chip and Carbonating ...	33°—35°	1:30	1:32	1:35	1:40	1:45
Carbonating	32°—35°	1:25	1:28	1:30	1:35	1:38
Stock Ale........	50°—55°	1:50	1:50	1:55	1:58	1:60
Racking	32°—34°	1:20	1:23	1:25	1:28	1:30
		NO DISKS.				
		1,000	2,000	3,000	4,000	5,000
Starting	50°—55°	1:15	1:16	1:18	1:20	1:22
Yeast	32°	1:6	1:7	1:8	1:10	1:12

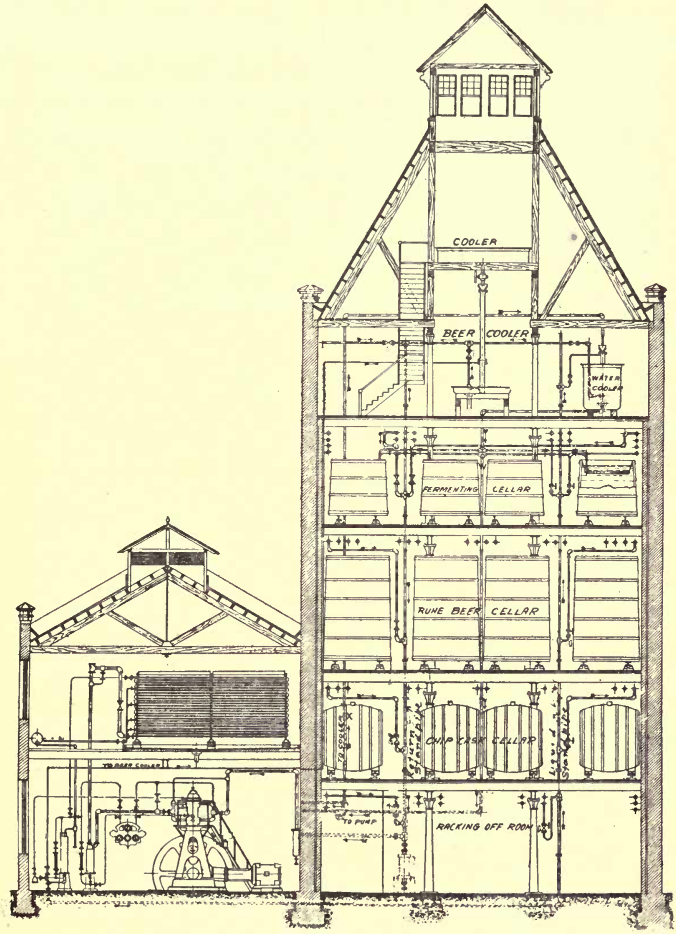


FIG. 23—MODERN BREWERY EQUIPPED WITH REFRIGERATING PLANT.

RATIOS FOR LAGER BEER BREWERIES.

2″ pipe direct expansion with 14″ disks per foot.

Room.	Temp. of Room.	Size of Room in Cubic Feet. 10,000	15,000	20,000	30,000	40,000
Starting tub.....	34°—36°	1:23	1:24	1:25	1:25	1:25
Fermenting	34°—36°	1:23	1:24	1:25	1:25	1:25
Storage (Rhue)..	32°—33°	1:38	1:40	1:43	1:45	1:47
Chip Cask (Spa).	32°—34°	1:40	1:43	1:45	1:47	1:50
Racking	34°—36°	1:23	1:24	1:25	1:28	1:30
		3,000	4,000	5,000	6,000	8,000
Hop Storage.....	32°	1 :20	1 :22	1.23	1 :24	1 :25

Example: Ratio, 1:23 means = 1 foot of pipe for 23 cubic feet of room.

Add 75% more pipe if without disks.

Weight of 1 foot 2 inch pipe, with disk and ice, about 75 pounds, length = 20 feet.

No more than 1,200 feet 2 inch pipe in one expansion (approx.).

One ton refrigeration for 120 feet 2 inch expansion pipe.

Wherever convenient, place piping on the ceiling.

Storage and Chip Cask.—Piping may be placed on the ceiling.

Fermenting Room.—Place piping over aisles or passageways, so as not to drip into the fermenting tubs.

Racking Room.—Piping may be placed on the ceiling and as much as possible about the door, to take up the outside heat as it enters.

Hop Storage.—Piping must be placed in a bank at the side of the room, so that all moisture can be easily drained away (forced air cooling preferred).

Brine vs. Direct Expansion.

It is customary to shut off all rooms from the pipe line during the short period of time, usually 3 hours, that the wort is cooled. Since this represents the *maximum amount of work* required from the refrigerating machine, its capacity is usually figured on the amount of work done in cooling a given quantity of hot beer wort within 3 hours.

Hettinger claims that, in case the wort is cooled by the brine system, only one-eighth of the refrigerating capacity is needed against that required in the case of direct expansion, because the cooling of the brine itself is extended over the entire 24 hours. No regulation of the expansion valves is required, since the temprature of the brine in the tank will only be raised 7.3 degrees F. during the entire period of cooling the wort, the capacity of the brine tank, being four times as great as the amount of the beer cooled.

A refrigerating machine using the brine system has to have double the capacity to a day's work in 12 hours that would be required to do the work in 24 hours.

Hettinger tries to disprove this by an example. He assumes a brewery plant, equipped with a 250-barrel beer kettle, the output being half lager and half stock and lively ale and the brewing of ale and lager beer being done alternately. Total space of the different rooms = 106,801 cubic feet. Allowing 7,000 cubic feet for 1 ton of refrigeration in 21 hours, the required number of tons of refr. = 15.26 tons. Heat of fermentation in 21 hours = 8 tons.

Cooling the beer through a racking cooler, allowing 6° in 8 hours = 4 tons. This means that the refrigerating machine will do 52 tons of refr. during 3 hours, and about 26 tons during the remaining 21 hours on the day lager beer is brewed. The next day when ale is brewed, the refrigeration required for cellars, fermenting room and racking room will be the same, that is, 26 tons in 21 hours. The ale storage does not require any refrigeration whatsoever.

The required capacity of the refrigerating machine, assuming that the ale will be cooled down 14 degrees in less than 2.5 hours and the wort having a strength of 15 per cent Balling: $(259 \times 250 \times 14 \times 10 \times 1.0614 \times 0.9) \div 284{,}000 = 30.49$ tons of refrigeration.

By doing the same amount of work with the brine system, in 24 hours, the calculation in tons will be as follows:

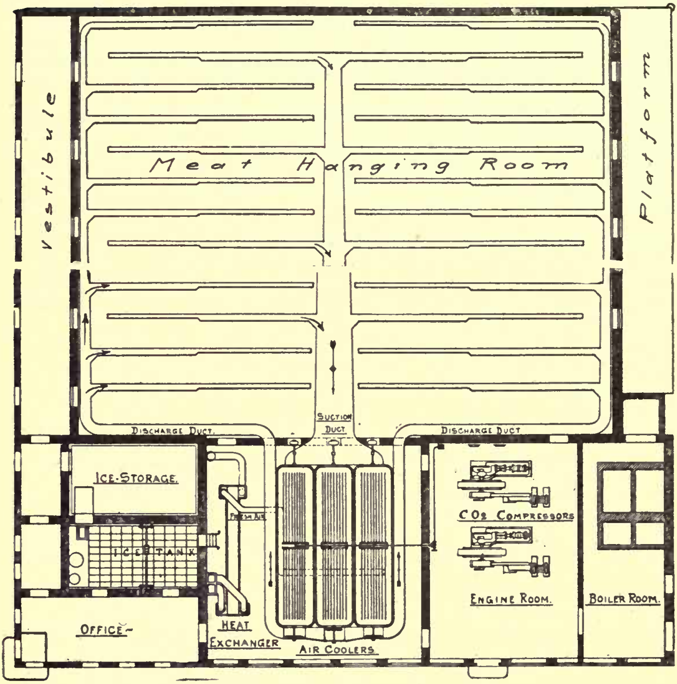
Cooling 125 barrels of wort for lager	3.25
Cooling cellars and rooms	13.35
Developing heat of 250 barrels of ale and lager	7.00

Chilling 125 barrels lager beer for racking	1.34
Cooling 125 barrels of wort for ale	1.50
Total	26.44

So that a machine of 26.44 tons is required to perform this amount of work in 24 hours, or a machine of 52.88 tons to do the work in 12 hours.

250 barrels is substituted for 125 barrels of ale and 125 barrels of lager because the work of the refrigerating machine, owing to the brine system, is extended over 48 hours, figuring one-half brew of ale and one-half brew of lager, the machine being calculated to run at the same speed and back pressure during the brewing of lager and ale.

NOTES ON BREWERY REFRIGERATION:



MODERN PACKING HOUSE EQUIPPED WITH FORCED AIR CIRCULATION.

Refrigeration should be produced by cold, dry air, which circulates freely around the meats, especially in the chill rooms, where the steam from the fresh killed animals and the foul gases have to be removed, so as not to affect the goods and the insulation.

Forced air circulation may cause a little more loss in weight in meat, but it is the soundest when viewed bacteriologically.

Recently a store room with direct expansion became invaded with phosphorescent bacteria. These bacteria produced a brilliant phosphorescence on a great many quarters of beef and carcases of mutton. The temperature of the room ranged about 35 to 40 degrees F. The germs can grow even at much lower temperatures, and they produce poisonous properties in meat.

To exterminate this bacillus from a room, the doors must be open, all ice and snow scraped away, and the pipes and the walls, floor and ceiling washed with solutions of lime, containing chloride of zinc. This zinc should exist in the wash in the proportion of 1 to 1,000. All meat that has become infected should be destroyed, as it is unfit for food.

Almost all European and Australian packing houses are refrigerated on the forced air cooling system with wet air coolers, which provides for the continuous ventilation of the chambers, and the purification of the air contained in them. Under these conditions there is no chance of the growth of bacteria which would be detrimental to health.

Refrigeration Required.

A. *Storage rooms,* which is estimated like *"General Cold Storage."*

B. *Chilling rooms,* either calculated like *General Cold Storage* or roughly estimated.

One ton refr. (24 *hrs.*) *for each of the following duties*:

15 to 24 hogs of 250 pounds each.

5 to 7 beeves of 700 pounds each.

45 to 55 calves of 90 pounds each.

50 to 70 sheep of 75 pounds each.

Hog chill rooms to be reduced to 32° F. in 24 hrs.

Beef chill rooms to be reduced to 32° F. in 36 hrs.

Chilling rooms to have ventilators on ceiling to allow steam and gases to escape, after which same have to be closed.

Space required.—Nine sq. f. per beef, 12 ft. high. Two sq. ft. per sheep, 8 ft. high. Meat rails about 27″ apart.

Piping to be estimated like *General Cold Storage,* with an *addition of* 13 *ft.* 2″ *direct expans. per ox,* and 6 *ft.* 2″ *pipe per hog.*

Piping to be arranged in overhead lofts.

C. *Freezing Rooms.* (Temperature 10° F. and below.)

Refrigeration is calculated like *General Cold Storage* with an *addition of one ton refr. per ton of meat.*

Piping estimated like *General Cold Storage,* with an *addition of* 30 *ft.* 2″ *direct expans. pipe per ox,* and 15 *ft.* 2″ *per hog.*

NOTES ON PACKING HOUSE REFRIGERATION:

Can Ice Plants

Capacity of Plant.—The ice making capacity is far below the refrigerating capacity, as we have to cool the water first from the ordinary temperature to 32°, and from there to the temperature of the brine. An allowance of 6 to 12 per cent. loss has to be made, due to radiation in freezing tank, pipes, etc. This would leave 60 *per cent. of the refrigerating capacity.*

Refr. tons	5	10	20	35	50	75	100	150	220	300	500
Ice, tons	2½	5	12	20	30	45	60	90	130	180	300

Time of Freezing.

The time of freezing depends on the temperature of the brine and the thickness of the ice. The following table is calculated by A. Siebert, on the assumption that the time of freezing is proportional to the square of the thickness.

FREEZING TIMES FOR DIFFERENT TEMPERATURES AND THICKNESSES OF CAN ICE.

Thickn'ss.	1 in.	2 in.	3 in.	4 in.	5 in.	6 in.	7 in.	8 in.	9 in.	10 in.	11 in.	12 in.
Temp. 10°	0.32	1.28	2.86	5.10	8.00	11.5	15.6	20.4	25.8	31.8	38.5	45.8
" 12°	0.35	1.40	3.15	5.60	8.75	12.6	17.3	22.4	28.4	35.0	42.3	50.4
" 14°	0.39	1.56	3.50	6.22	9.70	14.0	19.0	25.0	31.5	39.0	47.0	56.0
" 16°	0.44	1.75	3.94	7.00	11.0	15.8	21.5	28.0	35.5	43.7	53.0	63.0
" 18°	0.50	2.00	4.50	8.00	12.5	18.0	24.5	32.0	40.5	50.0	60.5	72.0
" 20°	0.58	2.32	5.25	9.30	14.6	21.0	28.5	37.3	47.2	58.3	70.5	84.0
" 22°	0.70	2.80	6.30	11.2	17.5	25.2	34.3	44.8	56.7	70.0	84.7	100.0
" 24°	0.88	3.50	7.86	14.0	21.0	31.5	42.8	56.0	71.0	87.5	106.0	126.0

The *sizes of the cans,* most in use, are given as follows:

Size of cans.	Weight of ice blocks.		Thickness of iron.		Time of freezing.	
	Nominal.	Actual.	Sides.	Bottom.	15° brine	18° brine.
6 x 12 x 26"	50	56	20"	20"	15 hrs	20 hrs.
8 x 16 x 32"	100	110	18"	16"	30 "	36 "
8 x 16 x 42"	150	165	18"	16"	30 "	36 "
11 x 22 x 32"	200	220	18"	14"	50 "	60 "
11 x 22 x 44"	300	315	16"	14"	50 "	60 "
11 x 22 x 57"	400	415	16"	14"	50 "	60 "

The temperature of the brine is about 10° higher than the ammonia in the expansion coils. By maintaining a good brine agitation, the temperature may be lowered a few degrees.

Back pressure, lbs. (gauge)...........	5	10	15	20	25	30
Brine temperature °F..............	—5	0	10	15	20	25

Freezing Tanks.

Expansion Pipe.—By good brine agitation and short expansions about 85 to 100 square feet of pipe per ton of ice will be sufficient. With a low back pressure the amount of pipe may be reduced.

The greatest efficiency is obtained with horizontal coils. In the case of vertical coils, top expansion is given the preference.

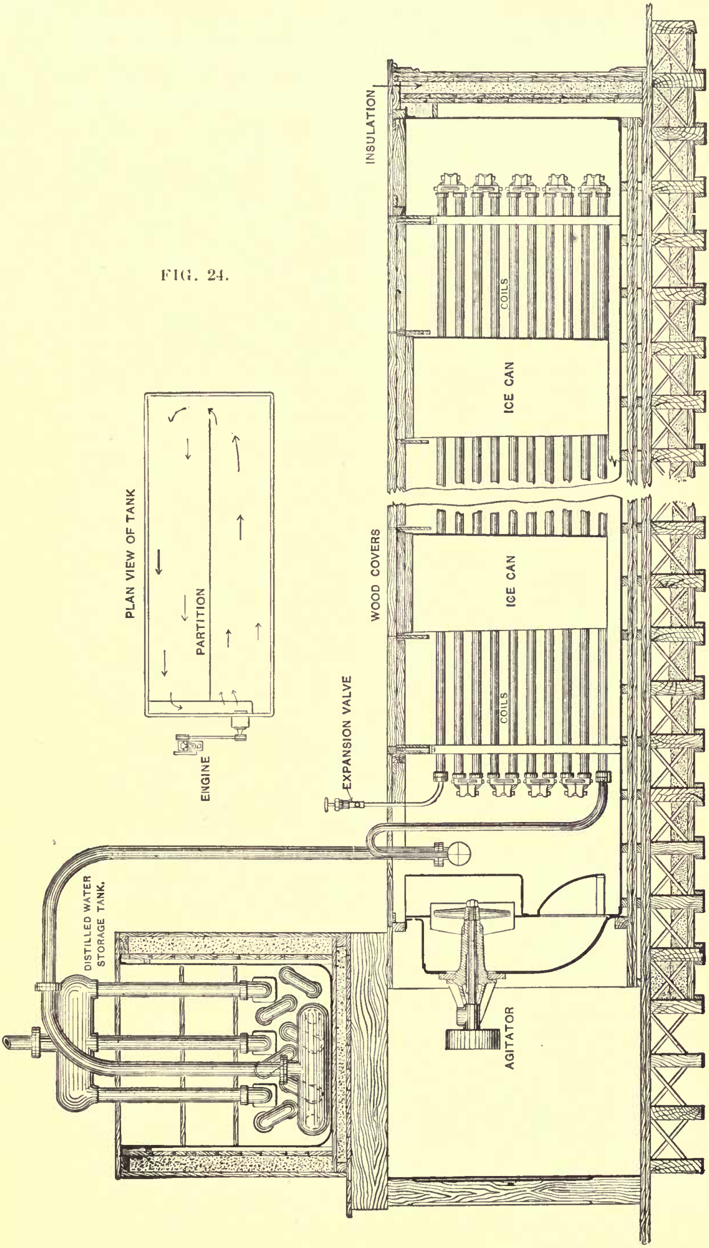
Amount of pipe per ton of ice.

15° brine.	18° brine.
400 ft. of 1" pipe	450 ft. of 1" pipe
320 ft. of 1¼" pipe	360 ft. of 1¼" pipe
270 ft. of 1½" pipe	310 ft. of 1½" pipe
210 ft. of 2" pipe	240 ft. of 2" pipe

Greatest length of one expansion is 1,200 ft.

Brine Circulation.—The brine is generally kept in motion by a propeller, driven by belt or direct connected to electric motor.

FIG. 24.



DETAIL ARRANGEMENT OF FREEZING TANKS FOR ICE FACTORIES

In tanks up to 10 tons use a 12″ propeller at 225 rev. per minute; in tanks from 10 to 25 tons use an 18″ propeller. In larger tanks use two propellers, or, still better, a centrifugal pump.

Allow 7¼ lbs. of salt per cb. ft. of tank. (See chapter on brine.)

Size.—The size of the tank depends on the *size of the cans, time of freezing* and *size of expansion pipe.*

The following table is based on 18° brine and 2″ expansion pipe:

Weight of blocks.	Number of cans.	Size of tank.
	5-TON TANK.	
100 lbs.	19 × 8 = 152	37′- 0 × 10′-4 × 36″
150 "	13 × 8 = 104	26′- 0 × 10′-4 × 46″
300 "	14 × 6 = 84	34′- 2 × 9′-8 × 4′-0
	10-TON TANK.	
150 lbs.	20 × 10 = 200	40′- 4 × 12′-6 × 46″
300 "	22 × 8 = 176	49′-10 × 12′-8 × 4′-0
	15-TON TANK.	
150 lbs.	30 × 10 = 300	58′- 0 × 12′-6 × 46″
200 "	38 × 10 = 380	87′- 7 × 15′-0 × 36″
300 "	25 × 10 = 250	58′- 7 × 15′-0 × 4′-0
	20-TON TANK.	
200 lbs.	42 × 12 = 504	96′- 4 × 17′-8 × 36″
300 "	34 × 10 = 340	78′- 7 × 15′-0 × 4′-0

Ice Storage.

By calculating the size of the ice storage room we assume that 50 *cubic feet* of ice, as usually stored, equal one ton.

Storage and ante-room have to be piped. The refrigeration and amount of piping can be calculated after the rules applying for *General Cold Storage.* Frequently a ratio of 1 :14 to 1 :20 is taken for 2″ direct expansion and about one-third to one-half more for brine piping. Pipes to be placed on ceiling.

The room should be well insulated and be provided with proper ventilation from the highest point and have thorough drainage.

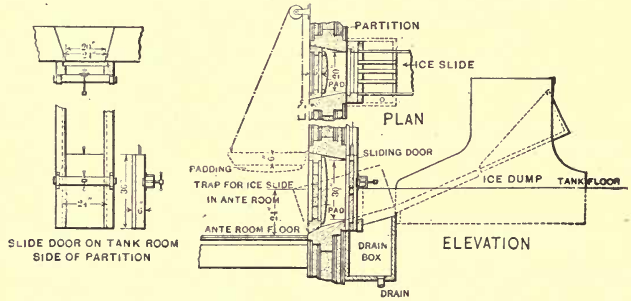


FIG. 25—DETAILS OF SLIDE DOOR ON TANK ROOM.

Cost of Ice.

The cost of ice varies considerably with the size of the plant, the price of coal and other items.

The following table gives an approximate estimate. But necessary alterations for price of coal and addition for cost of delivery, interest and other things must be made in each case, which may increase the total cost of ice from 20 per cent. in small plants to 50 per cent. in large plants.

The table shows *cost of ice put in the ice house ready to sell.*

APPROXIMATE COST OF OPERATING ICE FACTORIES

Tons Ice per day.	Engineers $1.50 to $5.00 per day.		Night Engineer or Oilers $1.50 per day.		Firemen $1.50 per day.		Tankmen and Laborers $1.00 per day.		Pipe Fitter or Machinist $2.50 per day.		Coal 15 cts. per cwt. or $3.00 per ton.		Oil, Waste, Lights and Sundries.	Daily Operating Expenses.	Ice per ton.
1	1	$1.50		....		....	1	$1.00		....	900	$1.35	$0.50	$4.35	$4.35
2	1	1.50		....		....	1	1.00		....	1,500	2.25	0.50	5.25	2.63
3	1	2.00		....		....	1	1 00		...	1,800	2.70	0.50	6.20	2.10
5	1	2.00	1	$1.50		...	2	2.00		....	2,500	3.75	1.00	10.25	2.05
7½	1	2.00	1	1.50	1	$1.50	2	2.00		....	3,200	4.80	1.25	13.05	1.74
10	1	2.50	1	1.50	2	3.00	2	2.00		....	3,600	5.40	1.25	15.65	1.57
15	1	2.50	1	1.50	2	3.00	3	3.00		....	5,000	7.50	1.50	19.00	1.27
18	1	2.75	1	1.50	2	3.00	3	3.00		....	5,500	8.25	1.80	20.30	1.15
20	2	4.50	1	1.50	2	3.00	3	3.00		....	6,000	9.00	2.00	23.00	1.15
25	2	5.00	1	1.50	2	3.00	4	4.00		....	7,500	11.25	2.50	27.25	1.09
30	2	5.00	2	3.00	2	3.00	4	4.00		....	9,000	13.50	3.00	31.50	1.05
35	2	6.00	2	3.00	2	3.00	5	5.00		....	10,500	15.75	3.50	36.25	1.03
40	2	6.00	2	3.00	2	3.00	5	5.00		....	12,000	18.00	4.00	39.00	1.00
50	2	6.50	2	3.00	3	4.50	6	6.00	1	$2.50	15,000	22.50	5.00	50.00	1.00
60	2	7.00	2	3.00	4	6.00	7	7.00	1	2.50	18,000	27.00	6.00	58.50	1.00
70	2	7.50	2	3.00	5	7.50	8	8.00	1	2.50	21,000	31.50	7.00	67.00	.99
80	2	9.00	2	3.00	5	7.50	10	10.00	2	5.00	24,000	36.00	8.00	78.50	.98
90	3	10.00	2	3.00	6	9.00	11	11.00	2	5.00	27,000	40.50	9.00	87.50	.96
100	3	10.00	2	3.00	7	10.50	12	12.00	2	5.00	30,000	45.00	10.00	95.50	.95

Coal Consumption.

The coal consumption depends on the size of plant, kind of engine, temperature of feedwater and quality of coal. The following table is based on an evaporative capacity of steam boilers of 10 lbs. of water per lb. of coal. For other ratios the coal consumption changes in direct proportion.

One ton of coal for	4 tons of ice in a	1 ton ice plant.
	5 " " "	10 " "
	6 " " "	25 " "
	7 " " "	50 " "
	8 " " "	100 " "
	11 " " "	large plants using evaporators.

Water Consumption.

Water is greatly economized in a can ice plant, as the same water is used first over the ammonia condensers, then in the steam condenser and, if it is of good quality, as feed water for the steam boilers. It leaves the boilers in the form of live steam to drive the engines, the exhaust steam of which is condensed, purified and used as the water from which the ice is made.

It is evident that the colder the water the less will be needed. An ice plant should always have a reliable supply of *four to six gallons of water per minute for every ton of ice.*

WATER CONSUMPTION PER TON OF ICE.

Temperature of Water.				
Over ammonia condensers..............	55°	60°	70°	80°
Entering steam condensers.............	80°	85°	90°	95°
Leaving steam condensers..............	125°	125°	125°	125°
Gallons per minute......................	4	4.5	5.15	6

Note.—For every 5 degrees increase in temperature of the cooling water the coal consumption increases 8 per cent., if the quantity of the water remains the same.

Distilling Apparatus.

The exhaust steam from the engine and pumps is generally used to supply the distilled water. The deficiency in supply, which increases with the size of the plants, is taken direct from the boiler.

The steam has to be deprived of the oil and, after being condensed, is subjected to a purifying process before it is allowed to go into the cans. It is impossible to give any rules for size and number of filters required on different plants, as it may be necessary to treat the water specially according to the quality of the water in the locality.

The usual course of distilling and filtering is as follows: *Engine, grease separator, steam condenser, skimmer and reboiler, charcoal filter, storage tank.*

Grease Separator.

These work on the principle that the steam strikes with great force against surfaces and deposits the oil.

Linde's grease separator consists of a vertical cylindrical tank with an upright spiral partition in the interior. The steam enters near the bottom and strikes against this baffle plate where its speed is reduced to one-fifteenth of the initial speed. The oil collects at the bottom and is drawn off.

Baldwin's grease separator is a cylindrical tank, either horizontal or vertical, filled to about one-fourth with water. The steam strikes against the water surface and deposits the oil. Baffle plates assist this process. These separators have proved very efficient.

"York's" grease separator is placed in the exhaust steam pipe in line with the pipe. The inlet nozzle is surrounded by corrugated baffle plates through which the steam must pass and which effectually separate the oil.

In the *coke filter* the steam has to pass through a large mass of coke, which is well adapted for extracting the oil from the steam.

Dimensions of Grease Separators.

	"De La Vergne" Coke Filter				"York" Grease S.	
Tons Ice	dia.	height	Steam pipe	Bushels of Coke required	dia.	height
5	2 ft.	4½ ft	4"	6½	22"	27½"
15	2½ "	5 "	5"	12½	26"	34"
20	3 "	6 "	6"	21½	"	"
30	3½ "	7 "	7"	34	30"	39¾"
60	4 "	7 "	8"	48	"	"
90	5 "	8 "	10"	91	36"	52½"
130	6 "	10 "	12"	170	"	"

Steam Condenser.

1. *Amount of Cooling Water* per ton of distilled water in 24 hrs.

$$P = \frac{2000 \times 960}{t - t_1}$$

t_1 = initial temp. of water, t = final temp. of water.
960 = latent heat of steam.

Example: $t_1 = 85°$ F., $t = 125°$ F.

$$P = \frac{2000 \times 960}{125 - 85} = 48{,}000 \text{ lbs. in 24 hrs.}$$

$$= \frac{48{,}000}{24 \times 60 \times 8.3} = 4 \text{ gals. per min.}$$

2. *Cooling Surface* in sq. ft. per ton of water in 24 hrs.

$$S = \frac{2000 \times 960}{(t_1 - t)\, n \times 24}$$

t = mean temp. of cooling water.
t_1 = average temp. in condenser (about 210° F.).
n = heat transmission per sq. ft. per hour per degree of difference in temp. (about 200 to 500).

Example (continued):

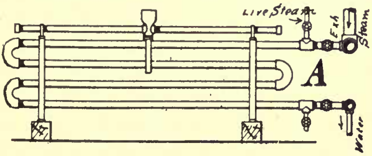
$$S = \frac{2000 \times 960}{(210 - 105)\, 200 \times 24} = \text{about 4 sq. ft.}$$

For practical calculations allow:
10 sq. ft. of pipe for one ton in *Open Air* condensers.
6 sq. ft. of pipe for one ton in *Surface* condensers.
14 sq. ft. of pipe for one ton in *Submerged* condensers.

Constructional Details.

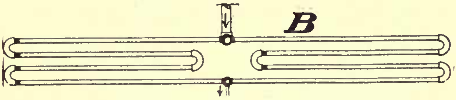
Every condenser must be provided with a back pressure or relief valve, which acts as a safety valve in case not all of the steam can be condensed on account of lack of condensing water, or for any other reason.

Fig. A illustrates an *atmospheric condenser,* a number of independent coils connected to two headers. Each coil is provided



with a stop valve on inlet and outlet, and a live steam and purging connection, so that any coil can be cleaned while the balance is in operation.

For *large plants* this type is also made as shown in Fig. B,

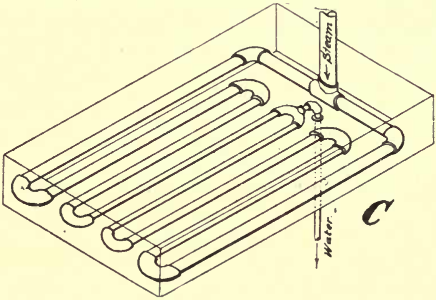


The object of this arrangement is the division of the area of the large main exhaust pipe into the many small areas of the coils as close as possible to the main inlet, without spacing the coils too close, which would prevent the cleaning of the outside surfaces of the pipes.

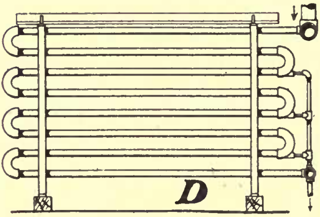
Where a very hard condensing water must be used and much cleaning of the outside surfaces of the pipes is necessary, *submerged coils,* as shown in Fig. C, have been used successfully. The area of the main exhaust pipe is divided into two branches, and the size of the pipes can be gradually reduced toward the outlet in proportion of the amount of steam condensed in each pipe while passing through the coil.

Submerged condensers can be well drained by giving all the pipes some slope toward the outlet.

The *condenser in Fig. D* is similar to to the De La Vergne ammonia condenser, having a number of outlets through which the water of condensation is drained off.

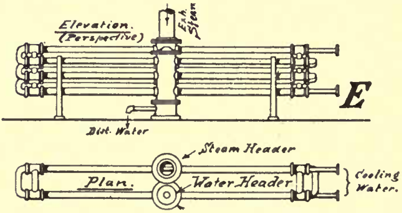


The *York double pipe condenser* is illustrated in Fig. E. Each section consists of two coils which are connected by return bends at both ends. At the center of each coil is a vertical header, one of which is for the steam inlet and the other for the water outlet. The exhaust steam enters the header on top. On its way



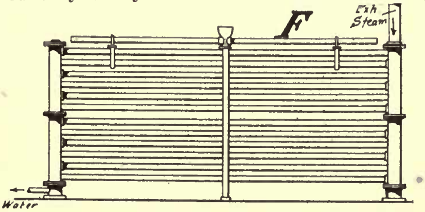
to the water header it has to pass but one return bend, all of which bends have a slope toward the water header for a perfect drainage.

The standard atmospheric condenser of the York Mfg. Co. is illustrated in Fig. F. These coils are made up with headers which

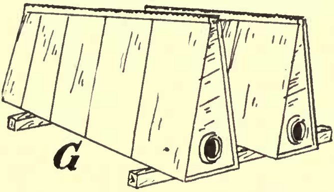


are connected with straight pipes. The steam is admitted to all pipes at the same time and has not to pass through cramped passages or to change its direction. If placed horizontally, the coils could be used in a submerged condenser.

The *Triumph condenser,* Fig. G, uses as the condensing surface sheet metal instead of pipes, in the form of V-shaped boxes. The condensing water can be used economically and the flat surfaces can be cleaned very easily while the condenser is in operation.



For special purposes and local condition the *shell condenser*, Fig. H, is used, both horizontally and vertically. It consists of



a shell, within which are a great number of small sized seamless brass or copper tubes, through which the condensing water passes, the shell being filled with the exhaust steam.

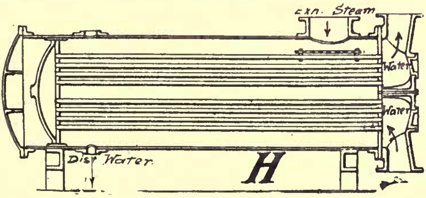


FIG. 26—A to H.

This type is very efficient, takes little space and can be placed anywhere inside the building.

DIMENSIONS OF SURFACE CONDENSER "H."

Cooling Surface sq ft	Capacity lbs of Steam p h.	H P at 20 lbs Rate	Dimensions over all		Tubes			Pipe Connections				Weight lbs
			length	width, height	No	length	size	Main Exh	Air	Circ Inlet	Circ Outlet	
100	1000	50	70	21	125	62"	5/8"	4"	2½"	2½"	2½"	1335
200	2000	100	82½	21	200	75½"	"	6	3	3	3	1890
300	3000	150	87½	27	300	75½"	"	8	"	5	5	2500
400	4000	200	87	28	404	75	"	8	"	"	"	2850
500	5000	250	104	29	410	92	"	10	"	"	"	3425
600	6000	300	111	30	456	99	"	10	"	"	"	3820
700	7000	350	117	32	518	102½	"	10	4	7	7	4260
800	8000	400	115	36	608	99	"	12	"	"	"	4720
1000	10000	500	112	39	788	95½	"	12	"	"	"	5590
1200	12000	600	114	40	950	95	"	14	"	8	8	6390
1400	14000	700	120	41	1030	102	"	14	5	"	"	7060
1500	15000	750	120	42	1102	102	"	14	"	"	"	7480
1600	16000	800	126	42	1155	104	"	16	"	10	10	7920
1800	18000	900	138	44	1160	116	"	16	6	"	"	8680
2000	20000	1000	138	44	1290	116	"	18	"	"	"	9360

Skimmer and Reboiler.

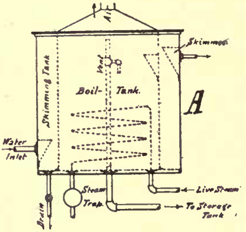
During the condensation of the exhaust steam more or less air or gas is absorbed by the water. The reboiling drives the impurities to the surface where they are skimmed off, while the air and gases escape through a vent into the outer atmosphere. The water should enter at the highest possible temperature (about 110° to 112°) so as to economize in steam for reboiling.

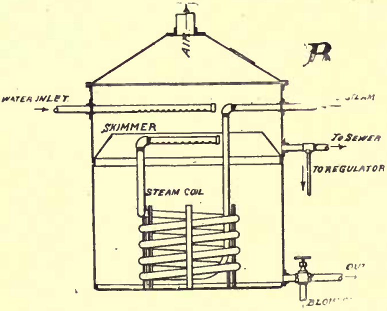
The steam coil is either closed or open. In the open coil the steam pressure is reduced to a few pounds and the condensation passes through the perforations and mixes with the water. The closed coil needs no regulation and is supplied with high pressure steam. The condensation is either carried back to the boiler by gravity, or is discharged into the steam condenser by a steam trap.

Mostly used are the *cylindrical tanks* and the *rectangular shallow pans*. The advantages claimed for the former, greater body of water, large skimming line, small floor space and simple construction; for the latter, large surface and small depth of boiling water, which are said to better assist the escape of the air and foul gases, constant current of water toward skimmer, possible division of surface into parts of decreasing ebullition.

Leading builders use from two to four square feet of W. I. pipe surface per ton of ice making capacity (less for brass or copper tubing).

Constructional Details.



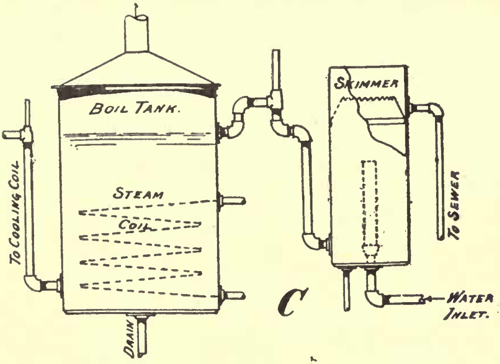


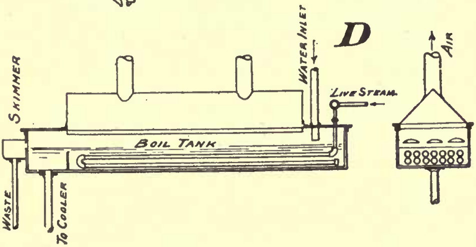
The De La Vergne Reboiler, A, has the boiling tank placed centrally within a larger tank, the annular space between both forming the skimming tank. Being placed at the same level with a hot water storage tank, the water level is always kept full, and the ebullition is confined to the boil tank, leaving skimmer in a state of rest. The steam coil is closed and provided with a steam trap.

The Triumph Reboiler, B, is also cylindrical, the skimmer being a V-shaped annular trough within the reboiler. The steam coil is open, discharging the condensation near the surface of the water.

Another cylindrical type, Fig. C, is used by *Fred W. Wolf* and a number of other builders. It has no automatic regulator. The water level in the skimmer and the boil tank is kept constant by goose neck outlets.

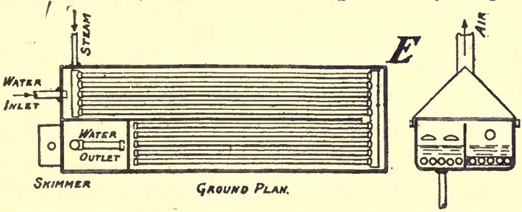
The York reboiler, D, is of the rectangular shape with open steam coil. The oil and impurities are carried by the water current into the skimming chamber, where they are skimmed by means



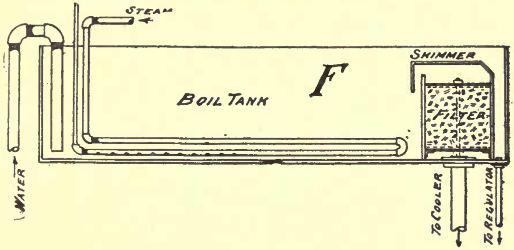


of V-shaped openings in the end of tank into a trough at the end of the reboiler. The pure water is discharged from the bottom of this chamber.

The Frick reboiler, E, is divided lengthwise by a partition,



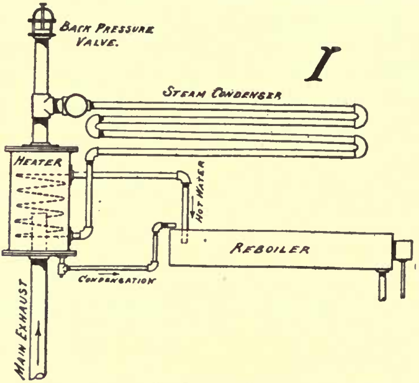
which not only lengthens the travel of the water, but brings same in a counter-current to the flow of steam which is doubled



by this division. The pipes of the open steam coil are not perforated, but are closed with caps, each of which has a small hole for the discharge of condensation. The skimming and discharge of the pure water are similar to those of the York reboiler.

The Wingrove reboiler, F, is a combination with a filter for the outgoing pure water. The steam coil is open and perforated at the end of the pipes. The oil and floating impurities are carried into the skimming chamber over a special shaped plate above the filter.

The Bertsch reboiler, I, is a combination with a heater in-

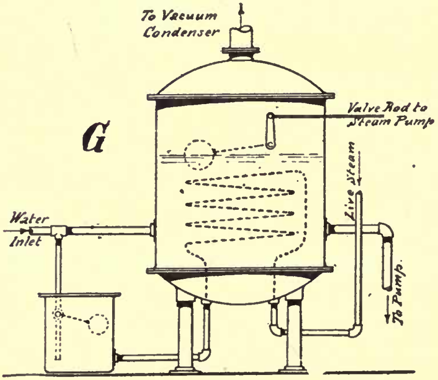


serted in the exhaust line in front of the condenser, the purpose of which is to deliver the condensed water to the reboiler at the temperature of the exhaust steam.

The condensed water from the condenser passes on its way to the reboiler through the coil of the heater. The condensation from the heater can be drained into the reboiler or float tank.

In connection with a condensing engine and a vacuum steam condenser, a *vacuum reboiler* saves steam, because the boiling point is much lower, and it saves cooling water, because the boiling temperature corresponding to the vacuum is not above 140° F.

In the *De La Vergne vacuum reboiler,* G, the water from the



vacuum steam condenser enters the reboiler by gravity near the bottom, and is removed and delivered to the hot water storage tank by a pump which is regulated by a float within the reboiler, raised or lowered by the variation of the water level. The air and gases are drawn into the steam condenser and removed by the air pump creating the vacuum. The closed steam coil discharges the condensation into a pot, from which it is siphoned into the reboiler through the water inlet line whenever the float within the pot opens the valve.

The York vacuum reboiler, H, contains within an air tight shell a series of shallow pans, each of which has an overflow and a dam to maintain a certain depth of water. The water

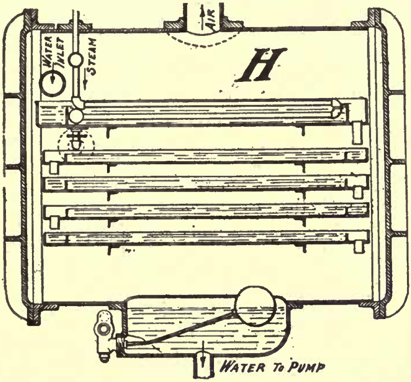


FIG. 27—A to H.

drops from one pan to the other and circulates through each pan. The top-most pan is provided with a closed steam coil for boiling. At the bottom of the shell is a float tank for the accumulation of the pure water which is removed by a pump. The float in the float tank regulates the steam for the water pump, which forces the pure water through the cooler and filters. At the top of the shell is the air outlet, which is either direct connected to an air pump, or to a vacuum steam condenser.

Frick Reboiler, 18 in. high, 30 in. wide.

Length: 1 to 6	ton	plants	=3 ft. 6 in.
8 to 12	"	"	=7 ft.
15	"	"	=10 ft. 6 in.
25	"	"	=13 ft. 9 in.
50	"	"	=20 ft. 6 in.
100	"	"	=23 ft. 9 in.

De La Vergne Reboiler.

2 to 15	ton	plants	=3 ft. dia., 4 ft. high.
20 to 30	"	"	=3 ft. 6 in. dia., 4 ft. 8½ in. high.
40 to 60	"	"	=4 ft. dia., 5 ft. high.

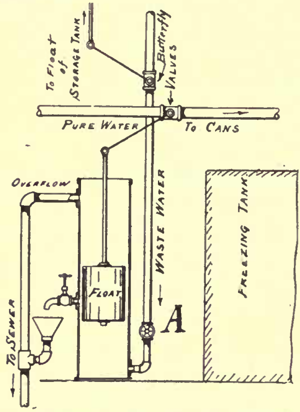
Frick Steam Condenser, 8 pipes high.

5	ton	plant	=1 coil, 15 ft. long.
10	"	"	=2 coils, 15 ft. long.
20	"	"	=4 coils, 15 ft. long.
50	"	"	=9 coils, 15 ft. long.
100	"	"	=17 coils, 15 ft. long.

Water Regulator.

The flow of the water leaving the reboiler must be automatically regulated before entering the cooler. The principle of such regulators is the automatic opening and closing of a valve (butterfly or quick opening) in the distilled water line.

A good regulator must allow a great variation in the quantity of water passing at each operation, as well as in the number of operations.

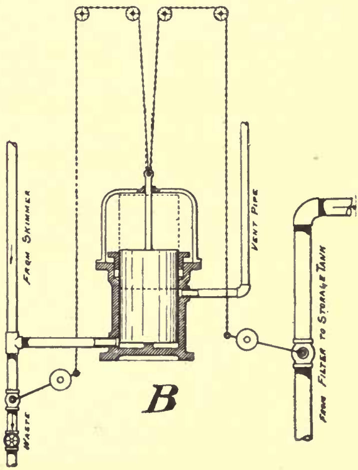


The De La Vergne regulator, A, consists of an open cylinder with a float and is operated by the waste water of the steam condenser. It can be placed anywhere near the distilled water supply pipe.

The operation is as follows: As long as the water level in the hot water storage tank is at normal height the butterfly valve in the waste water line is open and admits water to the regulator, thereby raising the float which opens the butterfly valve in the pure water line and allows the water to pass to the freezing tank. When the water in the hot water storage tank is low, both butterfly valves close and stay closed until the pure water in the storage tank reaches again the normal height, when the same operation is repeated.

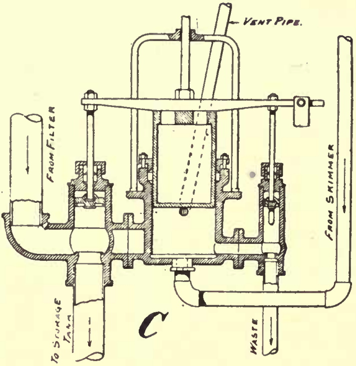
The York regulator, B, consists of a cylinder with a plunger to which two valves are attached, one for the pure water and one for the waste water. The water from the skimmer is used for operating the regulators, and the operation is as follows:

Whenever the reboiler is skimming, the mixture of oil and water fills the pipe connecting the skimmer with the regulator. As soon as the water column in this pipe is of sufficient height, the pressure so created elevates the plunger, whereby both valves are opened. The pure water then passes from the filter to the storage tank, and the skimming water drains through the waste pipe. The skimming in the reboiler stops and the water in the regulator and its supply pipe drains out, causing the plunger to lower and both valves to close, until the reboiler skims again. For the relief of the air which might get into the cylinder, a



vent is provided, which opens when the plunger is in its highest position. By the use of the skimming water the plunger is always well lubricated.

The Wingrove regulator, C, differs from the York regulator only in the mechanical means, and the principle is exactly the same in both and covered by the same description.

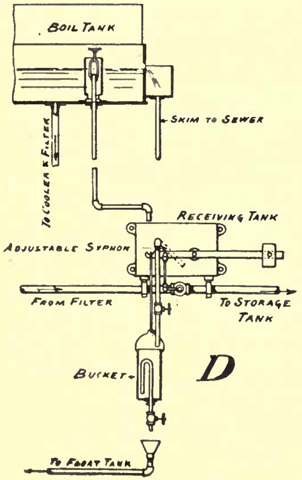


The Frick regulator, D, consists of two principal parts, the receiving tank and the counterbalanced bucket which operates the pure water valve. When the water in the reboiler reaches the overflow tube by which the skimming is regulated, the receiving tank begins to fill to the top of the siphon, after which the water passes through the siphon to the bucket.

As soon as the weight of the water overcomes the balance weight, the bucket lowers and the pure water valve opens, allowing the pure water to pass to the storage tank. After the bucket is filled to the top of its own siphon, it begins to empty its contents into the float tank from which the water is pumped back to the reboiler.

When the water in the reboiler is lowered below the top of the overflow tube, the supply to the receiving tank and the bucket stops, and the bucket is siphoned empty and becomes lighter than the balance weight, which raises the bucket and closes the pure water valve.

Bertsch's regulator, E, is a combination of the float and siphon types. The water pressure against the valve seat is counterbalanced by an adjustable weight. As soon as the reboiler is skimming, the float tank fills, the float rises and relieves the valve, allowing the water to pass to the storage or freezing tank. When the float reaches a certain height, the lever opens the drain pipe and starts the siphon which empties the float tank in the desired time, and this is regulated by the drain valve.



Condensed Water Cooler.

Its purpose is to cool the boiling hot water, as it comes from the reboiler, as nearly as possible to the temperature of the cooling water, after which any further cooling must be done by mechanical refrigeration.

Each cooling coil should be provided with a drain or washout connection at the bottom, and a steam connection at the top, as during the

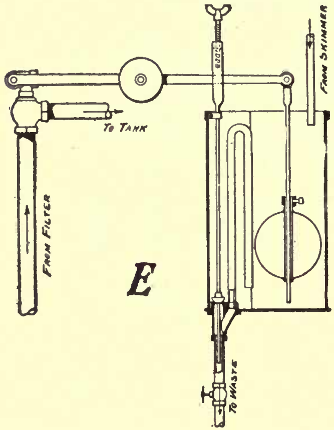


FIG. 28. A TO E.

cooling of the water some of the oil contained therein is separated and forms a coating on the inside surface of the pipes, which can only be removed by a blow of live steam.

The cooler is of the *double pipe* and more commonly of the *atmospheric type.* Its construction is sufficiently illustrated in the various arrangements of the different builders below.

Filter.

The cooled water receives a final filtration, in order to free it from any odors and foreign matters still contained therein. The most common place for the filters is after the cooling coils, and, again, right before the can filler. As the filtering media are mostly used sand, crushed quartz, maple charcoal, bone black (animal charcoal), pulp and felt or cotton cloth.

All of these materials have a purely mechanical action upon the water, with the exception of the wood and animal charcoal, which combine with the mechanical action also a chemical one, inasmuch as they have power to absorb any kind of odor. The charcoal filters are therefore also called *"deodorizers."*

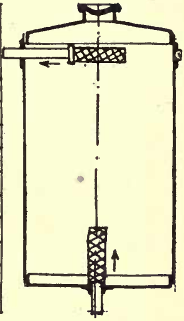
The method of filtering differs. Some filter from bottom to top, for which method it is claimed that the heavy particles in the water tend to fall to the bottom instead of clogging the filtering material. Others filter from top to bottom and the claim is that the oily substance contained in the water remains floating on top instead of being forced down through the filtering material. To cleanse these filters, the flow of the water is reversed in order to loosen the packet material and to wash the same. Where steam is used for cleansing, the content of the filter is first blown with live steam, and afterwards washed in the way as before stated.

Dimensions of Condensed Water Coolers

	"Frick" (Atm.)			"De La Vergne" (Atm.)			"York" (Double Pipe)	
Tons Ice	No.	length	2" pipes high	No.	length	2" pipes high	length overall	pipes high, 1¼" and 2"
1	1	7½ ft	2	1	5 ft	2		
2	"	"	4	"	"	4	18 ft.	1
3	"	"	8	"	"	6	"	2
5	"	"	10	"	"	12	"	2
8	"	"	14	"	"	"	"	3
10	"	15 ft	10	"	10 ft	"	"	4
15	"	"	14	"	"	16	"	6
20	2	"	10	"	"	20	"	8
25	"	"	14	"	"	"	"	10
30	"	"	14	"	15 ft	"	"	12
35	3	"	10	"	"	"		
40	"	"	14	"	20 ft	"		
60	4	"	14	2	15 ft	"		

Dimensions of Charcoal Filters.

	"Frick"				"De La Vergne"			
Tons Ice	No.	dia.	height	Bushels Charcoal each	No.	dia.	height	Bushels Charcoal each
1	#	24"	48"	10	1	30"	36"	7½
2 & 4	2	"	"	"	1	"	"	"
5	"	"	60"	13	1	"	"	"
8	"	30"	48"	14	1	"	"	"
10	"	"	60"	17	1	"	48"	10
15	"	"	"	"	1	36"	60"	18
20	"	"	"	"	1	"	72"	21½
25	"	36"	48"	20	1	"	"	"
30	"	"	60"	26	1	48"	60"	38
40	3	30"	"	17	2	36"	72"	21½



Storage Tank.

The storage tank serves for the purpose of storing up a large amount of distilled water. A wooden float generally covers the whole area of the water to prevent any reabsorption of air.

Many builders use the storage also as a fore cooler, having ammonia coils in the inside. The tanks are made either cylindrical or rectangular, of wood or of iron, and the cooling pipes are either an independent coil or simply an expansion of the ammonia suction pipe. The latter method is used in all plants where the machine can not work with backfrost, and the storage tank is used as much for preventing back-frost as for cooling the distilled water. The temperature of the water can be regulated at will where an independent coil is used for cooling. Where the return from the freezing tank is used for cooling, the temperature of the water depends entirely on the amount of heat the returning vapor can take up, which in many cases is very little.

Each can is filled separately by means of hose and can filler, which delivers the water to the bottom of the can, so that the water does not absorb more air as it rushes in.

DIMENSIONS OF CYLINDRICAL TANKS (NO COILS).

Tons ice.	Dia.	Height.
5	2½ ft.	3½ ft.
10	3 ft.	4 ft.
20	3½ ft.	5 ft.
40	4 ft.	6 ft.

DIMENSIONS OF SQUARE TANKS (EXP. COILS).

Tons ice.	Length.	Width.	Height.	2 in. Pipe.	Size of water pipe.
10	10 ft.	2½ ft.	3½ ft.	58 ft.	1 in.
20	11 ft.	3½ ft.	4 ft.	145 ft.	1 in.
30	12 ft.	4½ ft.	4½ ft.	218 ft.	1¼ in.
40	12 ft.	4½ ft.	5½ ft.	290 ft.	1½ in.
50	14½ ft.	4½ ft.	5½ ft.	363 ft.	1½ ft.
75	25 ft.	4½ ft.	5½ ft.	544 ft.	2 in.
100	17 ft.	7½ ft.	5½ ft.	725 ft.	2½ in.
200	24 ft.	9½ ft.	5½ ft.	1,450 ft.	3 ft.

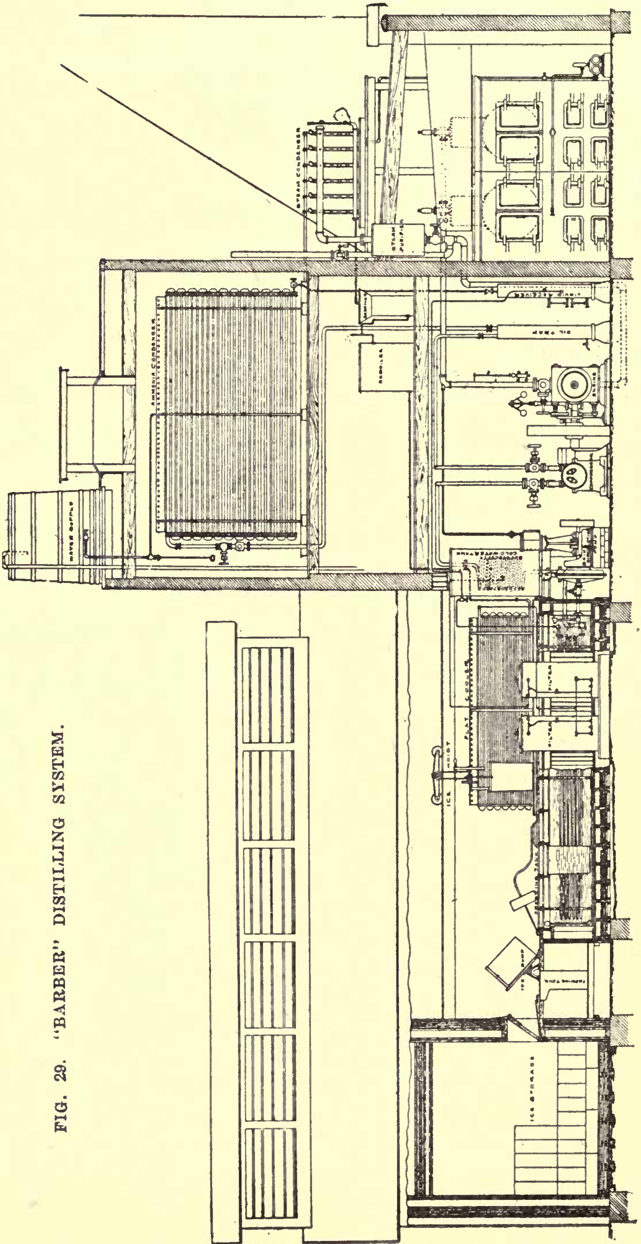


FIG. 29. "BARBER" DISTILLING SYSTEM.

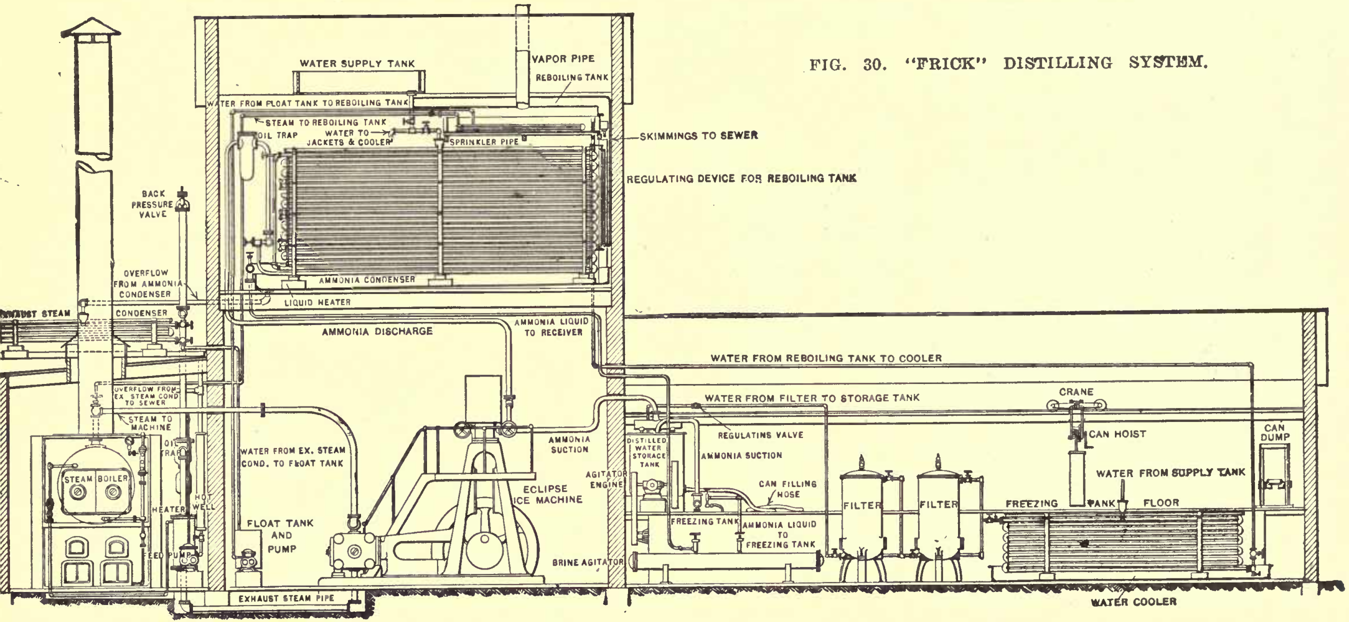


FIG. 30. "FRICK" DISTILLING SYSTEM.

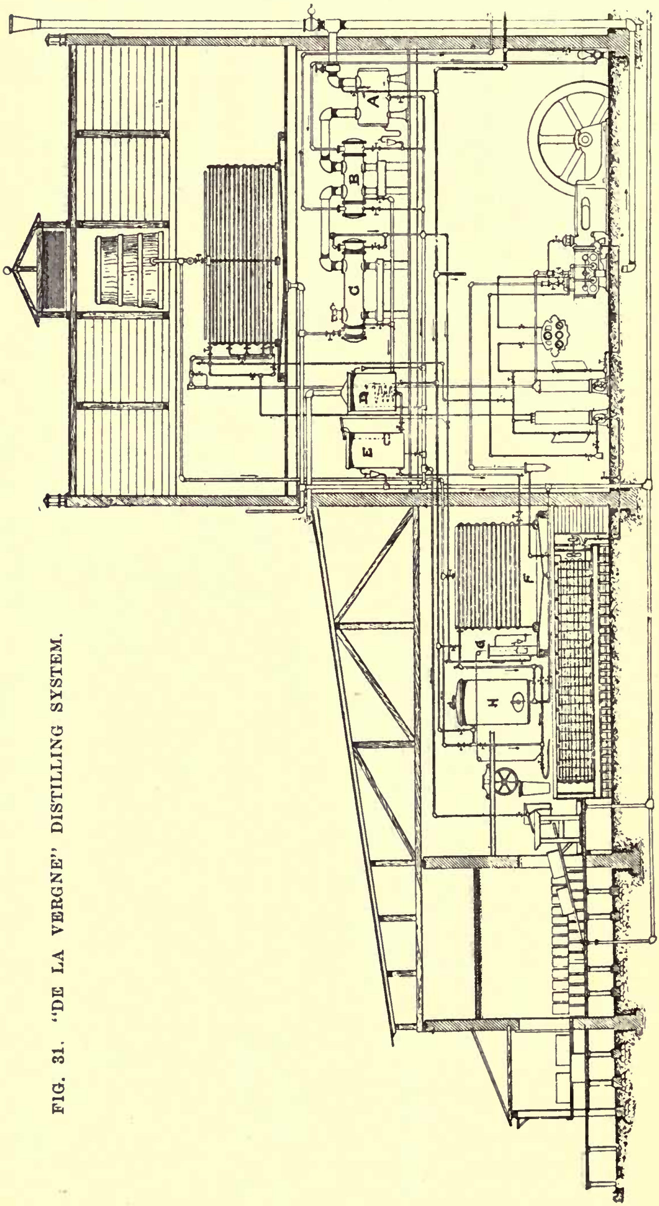


FIG. 31. "DE LA VERGNE" DISTILLING SYSTEM.

The Evaporator System.

The economy of ice production depends upon the efficiency of the boiler. If the boiler evaporates 8 lbs. of water per pound of coal and we lose 25 per cent. by steam cylinder condensation, condensation in exhaust pipe and loss by reboiling and skimming, we may produce 6 tons of ice per ton of coal.

Efforts were made to improve the economy and the use of compound condensing engines in connection with an evaporator in which the exhaust steam is used to produce additional distilled water was resorted to.

In all ice making plants with evaporators now in operation, the Lillie evaporator has been used. It consists of a cast-iron shell and is provided with copper tubes. Near one end is the tube head which divides the evaporator into two parts, the steam space and the vapor space. One end of the copper tubes is expanded in the tube head, the other end is closed, but the closed

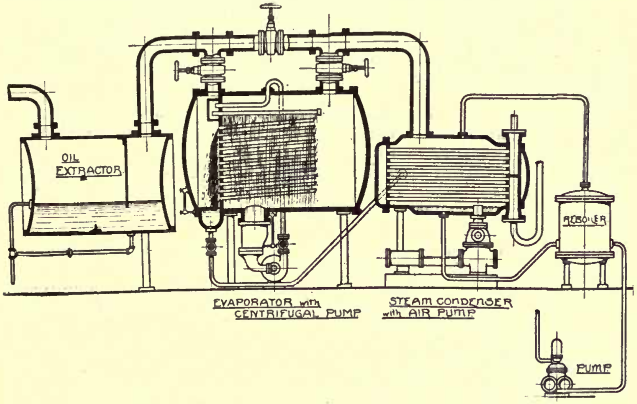


FIG. 32. DIAGRAM OF EVAPORATOR SYSTEM.

ends are each provided with a very small air vent hole. Under the evaporator a centrifugal pump is placed which serves to circulate the water over the tubes, a float in the float box keeps the water at a pre-determined level.

The exhaust steam from the low pressure cylinder, usually under a vacuum of 18″ and a temperature of 169° Fahr., enters the steam space of the evaporator and thence the copper tubes, the water which is showered over the tubes evaporates owing to the lower vacuum, 25″ or 26″, which, by means of the condenser and air pump is maintained in this space. The temperature of vapor under a vacuum of 26″ is 126°, and the difference between 126° and 169° is quite sufficient to produce boiling and consequently evaporation. The steam which enters the copper tubes is condensed, drops to the bottom of the steam space and from there is periodically discharged into the steam condenser.

The vapor is, of course, pure, clean and free from any odor owing to the fact that it is distilled at a low temperature; the steam, however, which has done its work in both the high and low pressure cylinders of the engine, contains all the impurities which such steam is subject to in any ice plant, viz., oil, oxide of iron and free ammonia. In order to free it from the oil and oxide of iron it must be washed or passed through a coke scrubber in the usual way except that in this case the oil extractor or coke scrubber must be operated under the same vacuum which is maintained in the steam space of the evaporator.

The vapor after it leaves the evaporator enters the top of the steam condenser, the air pump by taking away the air and most of the ammoniacal gases which have not yet been re-absorbed by the distilled water maintains a vacuum of from 25 to 26″.

The condensed steam leaves at the bottom of the condenser and flows over to the reboiler, whose vacuum is maintained through a by-pass with the vacuum part of the steam condenser. It enters the reboiler under a vacuum of 26″ and a temperature of 120° and needs only to be heated to 126° in order to boil.

When the water level within has risen to a certain height, a float inside will act upon the steam valve of the pump, which will commence to pump the water away up to the storage tank on the next floor, from which it passes through the usual course of cooling and filtering before entering the cans.

With the Lillie evaporator seven-eighths of a pound of vapor can be produced for every pound of steam. To produce 100 tons of distilled water would required fifty-five tons of exhaust steam, but in order to have that quantity enter the evaporator seventy-three or seventy-four tons must have entered the high pressure steam cylinder and this determines the economy of the plant.

In practice, 10 to 11 tons of distilled water can ice can be made per ton of coal if the latter evaporates eight tons of water under the working pressure in the boiler per ton of coal.

The exhaust steam from auxiliary machinery and pumps is used for heating the boiler feed water, and the water for the evaporator, if it is suitable, is heated by using it for cooling the distilled water.

The operation of such a plant is extremely simple, and it is not difficult for the operating engineer to understand it, in fact it requires no more attention than an ice plant with compressors driven by compound condensing steam engine. (L. Block, Trans. A. S. R. E. 1906, Abridged.)

Multiple Effect Evaporators.

Very large plants are enabled to use highly economical engines by having a double or triple effect evaporator. In this way the exhaust steam may be able to produce almost 3 times as much distilled water as exhaust steam is condensed, as we will see from the following calculation:

Assumed steam consumption = 2,000 lbs. per hour.
Distilled water required = 4,500 lbs. per hour.

The exhaust steam enters the first evaporator under a back pressure of 5 lbs. above the atmosphere. The last evaporator is in connection with a surface condenser with air pump, and a high vacuum is maintained in its vapor end. A moderate vacuum is maintained in No. II and a low vacuum in No. I.

Let us assume that the supply of water (which may be used first in the steam condenser) enters No. I at a temperature of 120°.

1. The first operation will be to raise the 4,500 lbs. of water from 120° to 203° F. (temp. of vaporization in No. I).

4,500 (203 — 120) = 373,500 units, which requires an equivalent of $\frac{373{,}500}{952}$ = 390 lbs. steam, condensed. (952 = lat. heat at 5 lbs. G. Press.) Deducting this from 2,000 lbs. initial steam, leaves 1,610 lbs. of steam, the condensation of which will cause a certain amount of water being evaporated; 952 being the latent heat of the steam in No. I, and 972 that of the water at 203°, the amount of vapor formed by the condensation of 1,610 lbs. of steam will be

$$\frac{1610 \times 952}{972} = 1{,}580 \text{ lbs. of vapor passing to No. II.}$$

Deducting this weight from the total of 4,500 lbs. = 4,500 — 1,580 = 2,920 lbs. of water passing to No. II.

2. This water enters at 203°. But as the temperature in No. II, due to the better vacuum is only 181°, it will, in falling 203 — 181 = 22°, give off vapor as follows:

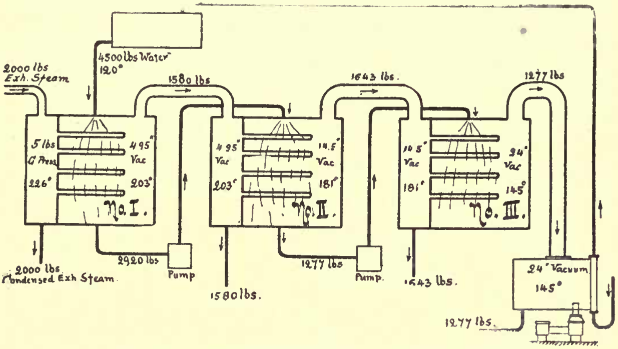


FIG. 33. TRIPLE EFFECT EVAPORATOR.

$$\frac{2920 \times 22}{992 \text{ (lat. heat)}} = 63 \text{ lbs. of vapor.}$$

As the 1,580 lbs. of vapor from No. I are condensed in No. II, it will under the better vacuum and lower temperature evaporate nearly the same weight of water. Adding 1,580 to 63 gives a total = 1,643 lbs. of vapor passing to No. III. Deducting this weight from 2,920 = 1,643 — 2,920 =1,277 lbs. of water passing to No. III.

3. Evaporator No. III has a vacuum of 24″ and a corresponding temperature of 145°.

The water in falling 181 — 145 = 36°, will give off vapor as follows:

$$\frac{1277 \times 36}{1012 \text{ (lat. heat)}} = 45 \text{ lbs. of vapor.}$$

As in No. II, taking the evaporation in No. III equal in weight to the condensation, or 1,643 lbs., the total will be 1,643 + 45 = 1,688 lbs.

This is far in excess of what is actually left to evaporate, namely, 1,277 lbs. It shows that the capacity of the triple effect is too great, or in other words, that less steam was needed to evaporate the initial amount of water.

The sum of the different weights of vapor passing out of the three vessels to be condensed for the supply of the ice cans is:

$$1{,}580 + 1{,}643 + 1{,}277 = 4{,}500 \text{ lbs.}$$

By calculations we find out that only about 1,860 lbs. of exhaust steam are required to distill that amount of water from an initial temperature of 120°.

This gives a ratio of $\frac{4{,}500}{1{,}860}$ = 2.42 lbs. of distilled water for each lb. of exhaust steam.

If the water is heated up to 200° before entering No. I, the ratio will be about 3 lbs. of water per lb. of steam.

The condensed steam, not being required for ice making, will be returned to the boiler as boiler feed water.

The vapor pipes are increased in size so as to make the fall of the temperature between the vessels as slight as possible.

Space Required for Can Ice Plants.

The illustrations below give an approximate idea of the space required for a given size plant. Of course, these dimensions can be varied greatly to suit local conditions.

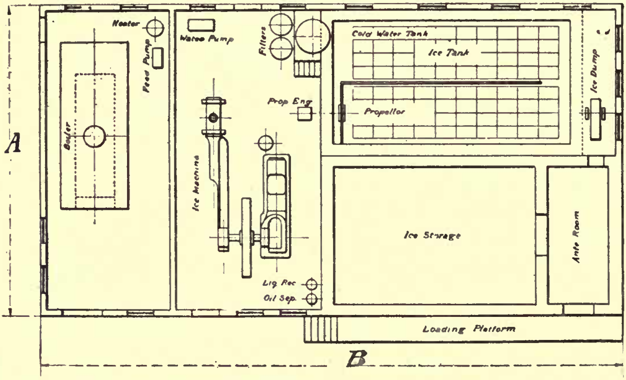


FIG. 34. HORIZONTAL D. A. MACHINE (WOLF).

Capacity tons ..	5	10	15	20	25	30	40	50	60	80	100
A in ft.........	30	35	37	40	42	42	49	49	54	59	73
B in ft.........	56	73	78	85	95	107	120	135	150	154	160

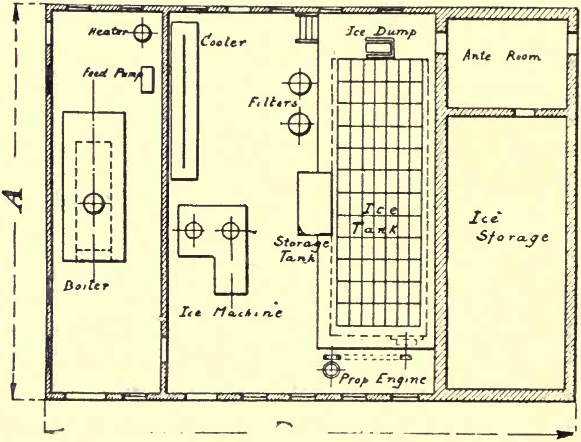
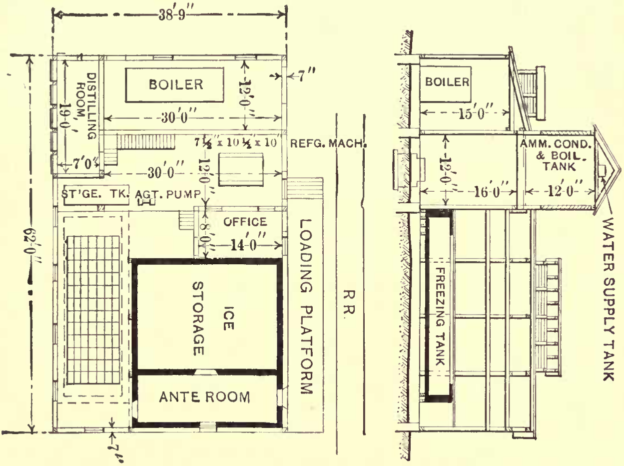
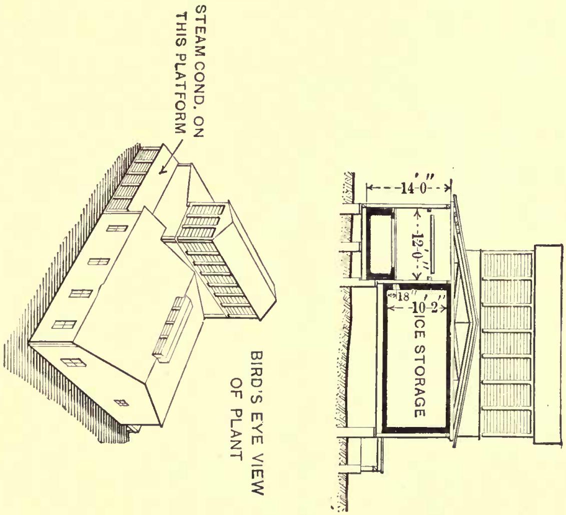


FIG. 35. VERTICAL S. A. MACHINE (YORK).

Capacity tons ..	6	10	15	20	25	30	40	50	60	75	100
A in ft.........	40	44	47	50	53	56	60	64	69	72	76
B in ft.........	53	64	75	87	97	108	121	135	150	163	174

Through the courtesy of the Frick Co. we are enabled to show in the following pages complete lay-outs of ice plants ranging from a daily capacity of 6 tons to 60 tons.





6-TON ICE FACTORY

FIG. 36.

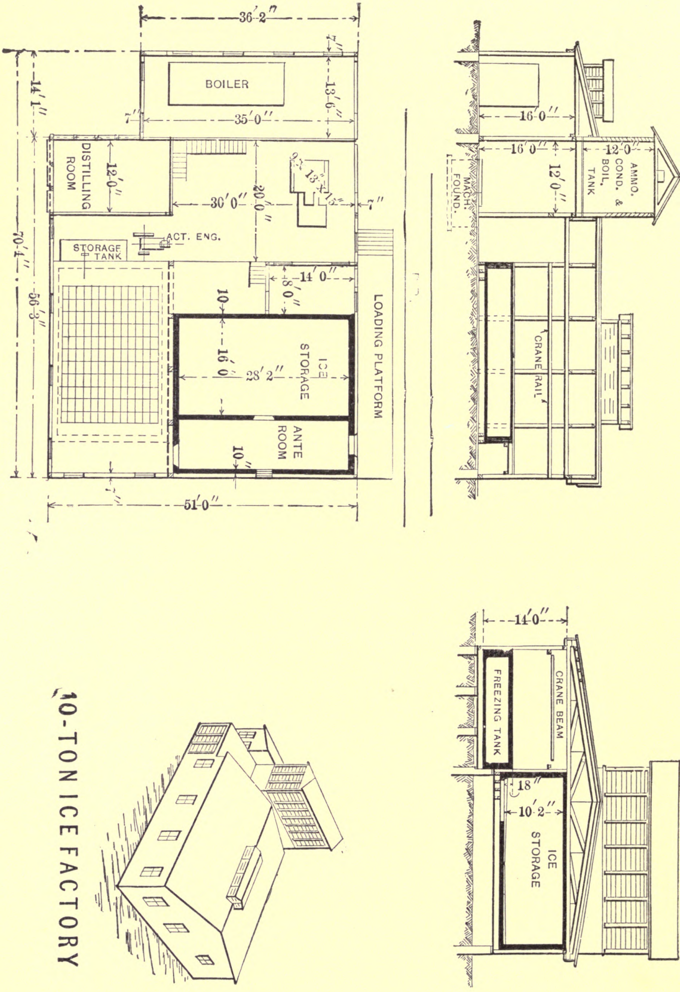
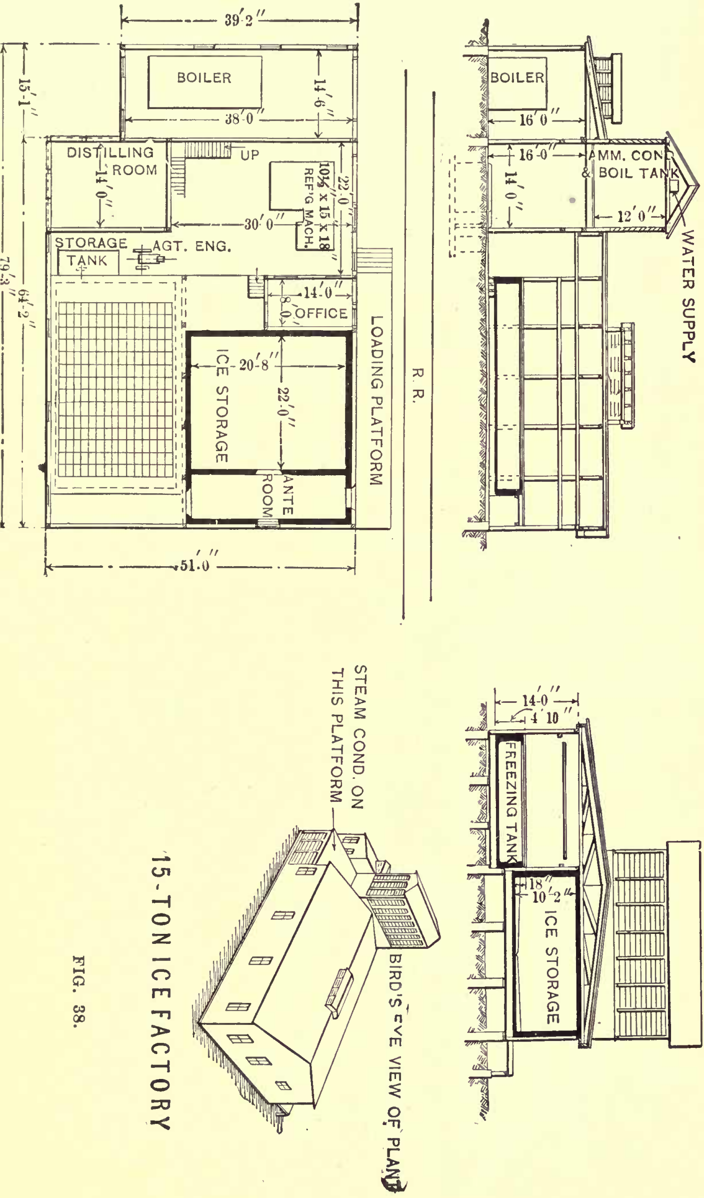

10-TON ICE FACTORY
BOILER
DISTILLING ROOM
STORAGE TANK
ACT. ENG.
ICE STORAGE
ANTE ROOM
LOADING PLATFORM
MACH. FOUND.
AMMO. COND. & BOIL. TANK
CRANE RAIL
FREEZING TANK
CRANE BEAM
ICE STORAGE

FIG. 37.



15-TON ICE FACTORY

FIG. 38.

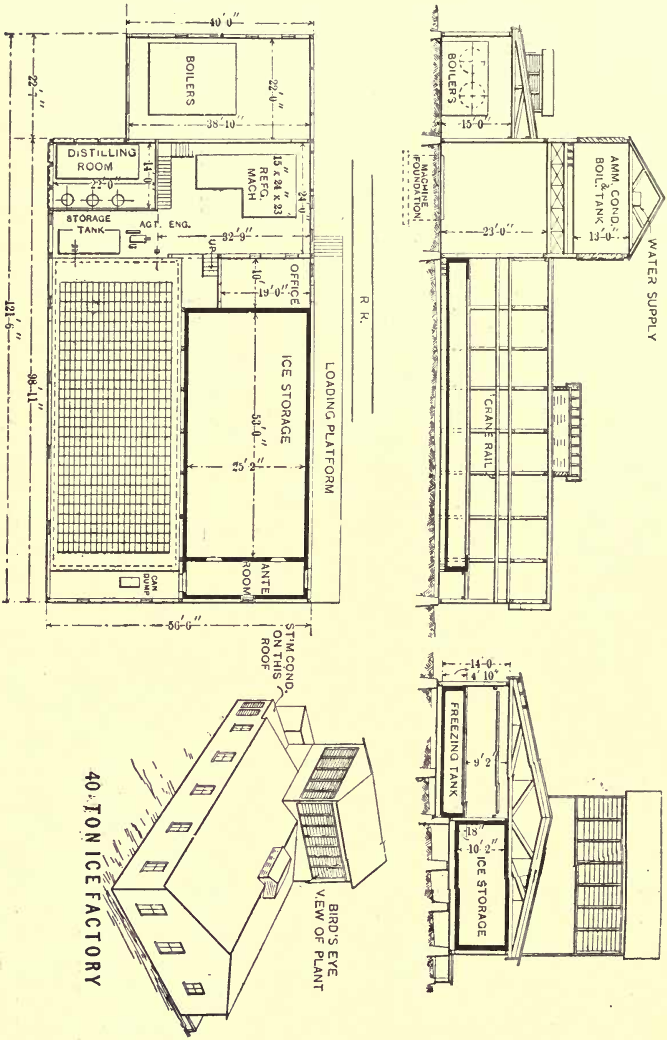
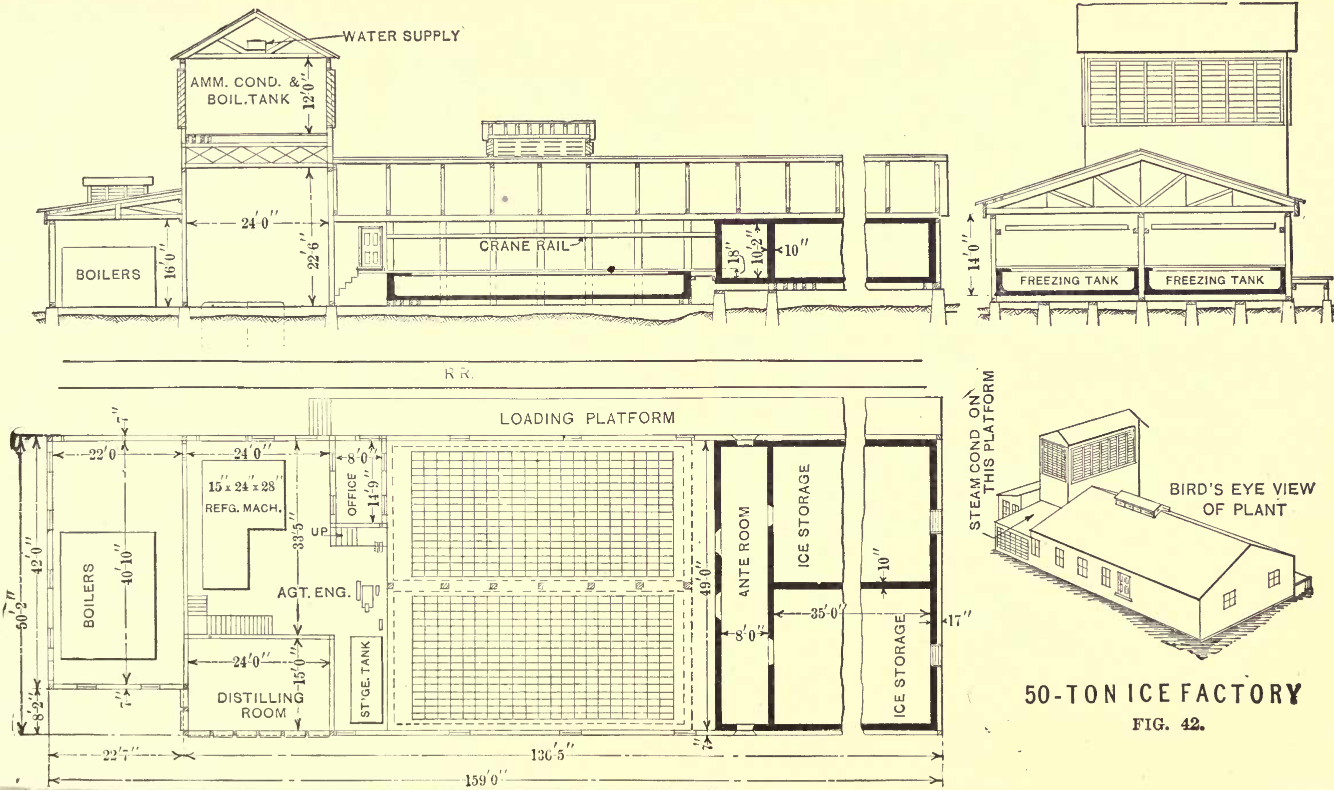
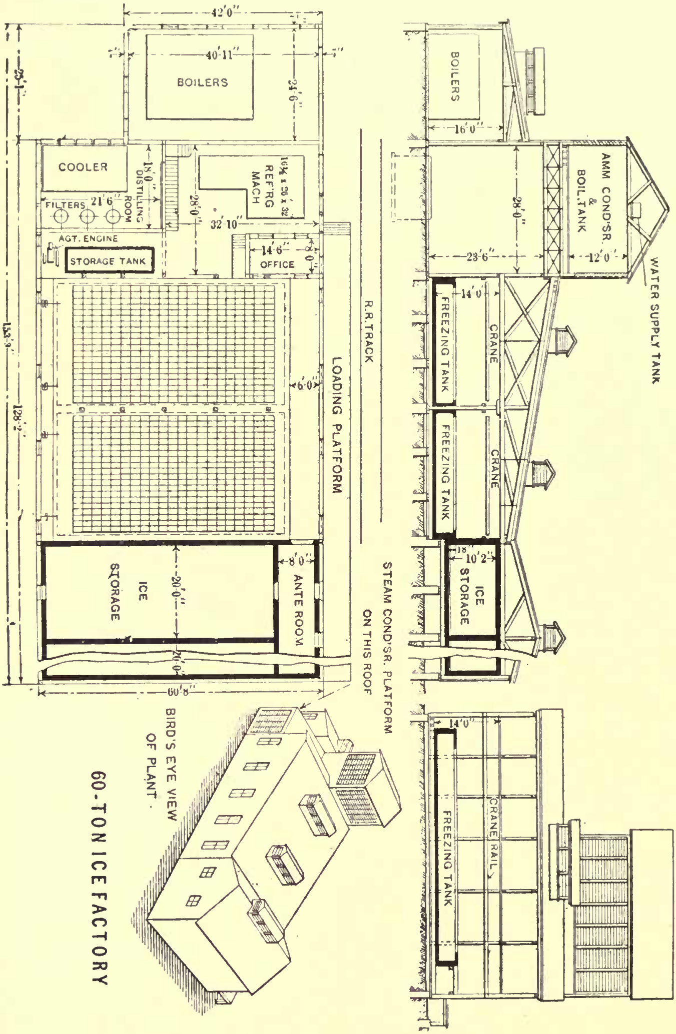

BOILERS
DISTILLING ROOM
STORAGE TANK
AGT. ENG.
15 x 24 x 23 REFG. MACH
OFFICE
ICE STORAGE
ANTE ROOM
CAN DUMP
LOADING PLATFORM
R. R.
ST'M COND. ON THIS ROOF
AMM. COND. & BOIL. TANK
WATER SUPPLY
MACHINE FOUNDATION
CRANE RAIL
FREEZING TANK
ICE STORAGE
BIRD'S EYE VIEW OF PLANT
40 TON ICE FACTORY

FIG. 41.



50-TON ICE FACTORY

FIG. 42.



BOILERS
COOLER
DISTILLING ROOM
FILTERS
REF'RG MACH
AGT. ENGINE
STORAGE TANK
OFFICE
ICE STORAGE
ANTE ROOM
LOADING PLATFORM
R.R.TRACK
STEAM COND'SR. PLATFORM ON THIS ROOF
AMM. COND'SR. & BOIL.TANK
WATER SUPPLY TANK
FREEZING TANK
CRANE
CRANE RAIL
BIRD'S EYE VIEW OF PLANT
60-TON ICE FACTORY

FIG. 43.

Plate Ice Plants

Plate Ice having its growth in thickness from one side only, the formation of ice proceeds from the freezing plate outward, and certain undesirable properties of the water held in solution or mechanically suspended or other than chemically fixed, are separated and rejected by the slowly freezing water. The residual or unfrozen water, at the termination of the freezing period, is drained off, the tanks then being refilled with fresh water.

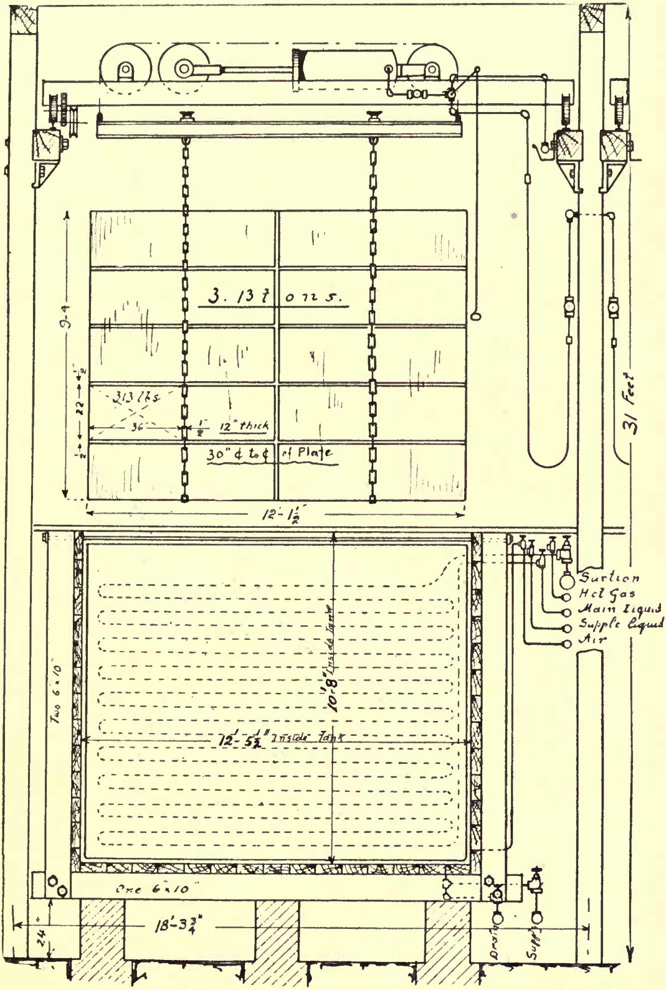


FIG. 44. DIRECT EXPANSION PLATE PLANT.

Plate ice is made by the following methods: The *direct expansion plate;* the *direct expansion plate, with still brine,* known as the "Smith" plate; the *brine cell plate;* the *brine coil plate,* and the *block system* with either direct expansion or brine coils.

The *direct expansion plate* is the simplest in construction and consists of direct expansion zigzag coils with ⅛-inch plates of iron bolted or riveted in place. The thawing off of the face of ice is accomplished by turning the hot ammonia gas from the machine direct into the tank coils.

The *direct expansion plate with still brine,* known as the *"Smith"* plate, is similar in construction, excepting that the coil is immersed in a brine solution contained in a water and brine tight cell. Thawing off is accomplished by turning hot gas into the coils.

The *brine cell plate* consists of a tightly caulked and riveted cell or tank about four inches thick, provided with proper bulkheads or distributing pipes, to give an even distribution of brine throughout the plate. The thawing off of the face of the ice is accomplished by circulating warm brine through the plate.

The *brine coil plate* is similar to the direct expansion plate, excepting that brine is circulated through the coil instead of ammonia. Thawing off is accomplished by means of warm brine circulated through the coils.

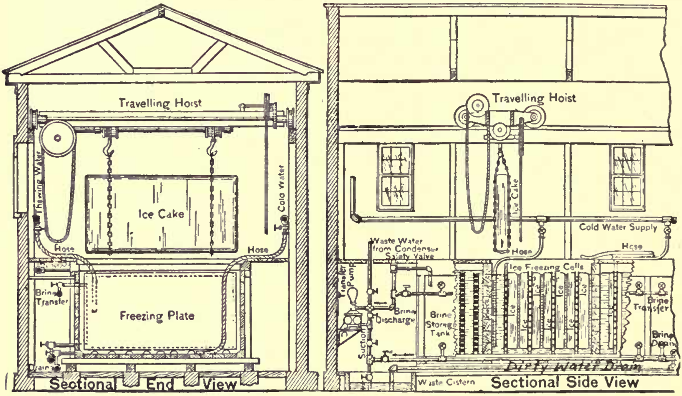


FIG. 45. BRINE COIL PLATE PLANT.

In the *block system* the ice is formed directly on the coils, through which either ammonia or brine is circulated. After tempering, the ice is cut off in blocks the full depth of the plate by means of steam cutters, which are guided through the ice close to the coils.

The method of harvesting is similar in all of the foregoing systems, excepting that in use for harvesting block ice. Some use hollow lifting rods and thaw them out with steam; others use solid rods and cut them out when cutting up the ice; and others again use chains which are slipped around the cake when it floats up in the tank.

Cutting up the plate is accomplished by means of steam cutters, power saws and hand plows. In the block system, however, where the ice is cut off the plate in the tank, it only remains to remove the cakes by means of a light crane and hoist and divide them into the required sizes with an axe or bar.

Agitation is accomplished by means of air jets located midway between the plates, sometimes in the center, sometimes three or four feet from one end and sometimes at both ends of the plates.

In well designed plants the production of a square top has been fairly well solved and it only remains for the owner to see to it that a constant water level is maintained in the tank while the ice is in process of formation.

From an economic standpoint, it is immaterial whether the ice as harvested from the tank has round or square ends, unless the tank be so designed that no ice is formed between thaw pipes or in back of thaw planks. This is especially true if the scrap ice can be utilized.

A thawing system has been designed requiring for its proper operation iron freezing tanks. The ice is formed up to the bottom and sides of the tank and on the outside of the tank around each cell consisting of two plates of ice, a hollow space is formed by means of studding and sheathing. In this space are steam coils which heat the outside of the iron tank and thus loosen the ice from the bottom and ends.

American Linde Plate System.—The freezing plates are constructed of square pipes, which, lying closely together, make a perfect sheet. They consist of two zig-zag coils, which interlock in each other. Through one of these coils (having the larger area) cold ammonia vapors are passed and through the smaller one brine is passed.

The working of these freezing plates is as follows:

When the cold ammonia vapors are passed through the ammonia coil, the cold is evenly transmitted through the whole surface of the pipes, and the brine coil, which is surrounded on two sides by the cold ammonia coil, will have nearly the same temperature as the ammonia coil, so that the freezing along the whole plate will take place just as fast as if the plate consisted entirely of one ammonia coil. When we want to loosen the plate of ice from

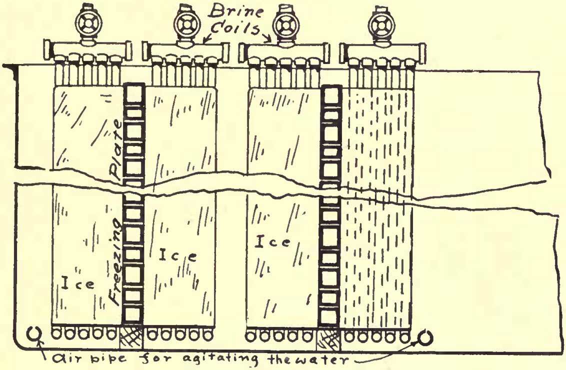


FIG. 46. AMERICAN LINDE PLATE SYSTEM.

the freezing plate, shut off the supply of liquid ammonia and open the valve which allows warm brine to pass through the brine coil.

After the plate is loosened, close the brine valve and open the valve which lets the liquid ammonia pass through the ammonia coil. To get the ice plates square the brine pipes are covered with sheet iron. The plate of ice forms inside this sheet and when it has formd thick enough and needs to be loosened, the same valve which lets warm brine pass through the brine coil interlocked with the freezing coil also lets brine pass through these coils, so that the ice is loosened from the plate.

An absorption machine under the right conditions should produce up to 12 tons of ice per ton of coal burned.

This figure includes all of the coal burned to provide steam for the water pump, ammonia pump, condensed steam pump, agitating apparatus, crane operation and so forth. Actual results on a season's business show 10 tons of ice sold per ton of coal bought.

Another advantage of such a plant is that cheap coal can be burned, providing a proper boiler plant has been installed.

The following costs per ton for operating a 50-ton plant may be interesting:

Coal at $2.20 per ton	$0.22
Labor	.34
Ammonia	.06
Incidentals and repairs	.24
Interest on investment	.25
Taxes and insurance	.11
Total to produce 1 ton of ice	$1.26

The factory cost of the ice is 86 cents per ton, including repairs.

A compression machine with compound condensing engine and with all pumps, etc., driven by the compressor engine would require at least 130 H. P. for a 50-ton ice-making plant and with an evaporation of 7-1 in the boiler plant, it would require the burning of 4½ tons of coal per day which would be equivalent to the making of 11 tons of ice per ton of coal burned. It is safe to say that not over ten tons of ice per ton of coal burned would be sold. *So that from the standpoint of coal economy the two plants would be practically equal.*

The cost per ton for operating a 50-ton compression plant would be about as follows:

Coal at $3.20 per ton	$0.32
Labor	.34
Ammonia	.03
Incidentals and repairs	.18
Interest on the investment	.25
Taxes and insurance	.11
Total to produce 1 ton of ice	$1.23

In this case the factory cost of the ice is 87 cents, including repairs.

The difference in factory cost per ton is so small that the whole matter resolves itself into the question as to which type of machine is best adapted to the particular conditions existing in the immediate vicinity in which the plant is to be erected.

A stil greater economy in the production of plate ice may be attained by a *combination of the absorption and compression machines.* The steam consumption of both types of machines is a well known quantity. If, then, the combination plant be so proportioned that all of the steam required to operate a simple Corliss engine be utilized in an absorption machine at, say, ten pounds pressure, either the absorption machine or the compression machine will be operated at no cost for coal.

Assume that a 100-ton plate plant be so designed. Then a 30-ton compression ice-making machine will drive a 70-ton absorption ice-making machine with its exhaust steam after the steam has done its work in the compressor engine. A plant designed on these lines would turn out 14 *tons of ice per ton of coal* burned and the cost per ton for operation would be about as follows:

Coal at $2.20 per ton	$0.16
Labor	.30
Ammonia	.05
Incidentals and repairs	.21
Interest on investment	.25
Taxes and insurance	.11
Total to produce 1 ton of ice	$1.08

The factory cost per ton of ice is in this instance reduced to 72 cents and the difference in the cost of production in favor of the combined plant is 15 cents per ton, which on a yearly output of 20,000 tons, gives the substantial sum of $3,000 per annum saved. (K. Wegeman, Trans. West. Ice Ass'n. 1907. Abridged.)

About 250 square feet of freezing surface will be required per ton per 24 hours on a brine plant and in a direct expansion plant about 275. The brine plants are more easy to operate than the direct expansion plants, for the reason that the plant can be operated more continuously under the same conditions. That is, the condition does not fluctuate so easily, and the ice can be made of a more uniform thickness for the reason that the temperature of the freezing surface is more uniform.

In a direct expansion plant the freezing surface that is not backed with the liquid ammonia will have one temperature, and the freezing surface that has gas inside of it will have an entirely different temperature, and the range is considerable.

The difficulty with the brine plants is the impossibility of making plates that won't leak. The displacement per ton for the compressors of a brine plant is less than the direct expansion plants.

If the expansion coils can be kept very nearly flooded with liquid we obtain a higher efficiency and a more uniform temperature.

The difficulty with the direct expansion plant is the ammonia leaks; the expansion coils being subject to such a range of temperatures. The loss of ammonia on a direct expansion plant is considerably more than on a brine plant.

If we use brine, we will have to use a slightly lower back pressure than if we use direct expansion. Few brine plants are running at much better than 10 or 12 pounds back pressure, whereas the direct expansion plant will run higher. The accumulator system will run as high as 14 or 16 pounds.

Plate ice can be made as pure as any can ice ever produced. There are two means at hand to accomplish this end:

Sterlization and Ozonization.—Where plenty of exhaust steam is at hand, sterilization is the best means, but in most plants ozonization will be found the more convenient method.

Treatment by ozone will reduce the number of bacteria from 3,000 to 7 per cubic centimeter, and the 7 remaining bacteria are of the harmless kind. The investment runs from $12 to $20 per ton of ice-making capacity, including filters; the power required is about one H. P. per hr. The German standard for pure potable water is 100 bacteria per cubic centimeter. The treatment would therefore more than meet the requirements of the health board.

A sterilizing equipment is both higher in first cost and cost of operation, and has the added disadvantage of sending the water to the forecooler at a considerably higher temperature.

Plate System vs. Can System.

The principal elements in the selection of "plate" system and "can" system contrasted:

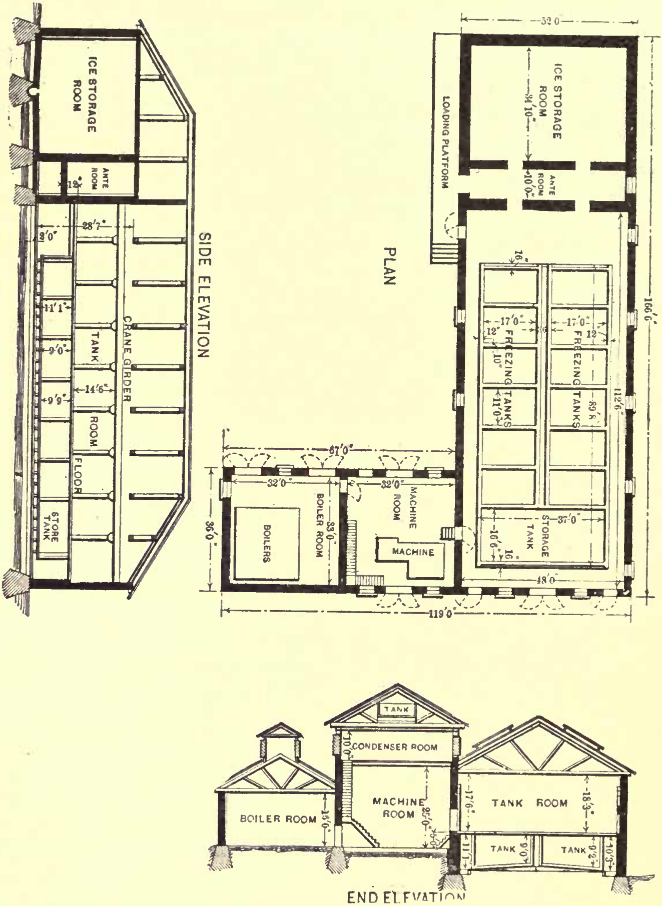


FIG. 47. 20-TON PLATE ICE PLANT.

Quality of Ice.—Both systems under intelligent management will produce ice of good quality, but the "can" system depends upon a complicated arrangement of distilling and filtering apparatus which permits rapid deterioration in quality if not carefully watched and kept in effective working condition.

Power.—Water, gas, electricity or any cheap motive power can be used for producing plate ice, but when distilled water is required, the "can" system *must* use steam.

Water.—Where water is highly impregnated with lime, etc., or gaseous products capable of vaporization and condensation, the "plate" system can be used if operated at a slow rate of freezing, as, for instance, sea water can be frozen on the "plate" system while very opaque and difficult to handle on the "can" system.

Investment or First Cost.—For producing ice 12 to 14 inches thick, the investment is greater in the "plate" than in the "can" system, where steam is used, by 33 to 75 per cent. This is due largely to the increased area of buildings required, high pressure compound condensing steam engines, power traveling cranes, expensive construction of freezing tanks and cells, etc.

Cash Available.—Given a limited cash capital you are enabled for one-half the money to buy and equip a "can" system of same tonnage capacity, occupying but one-half the space.

Ice for Cooling Cars.—When crushed ice is required solely for cooling purposes, the "can" system is by all means the cheapest

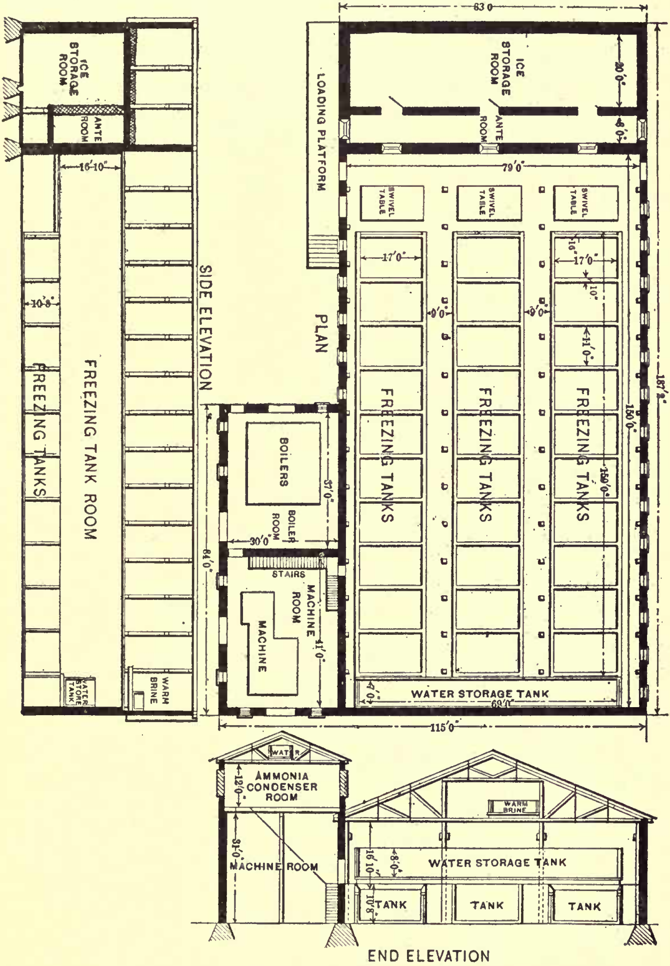


FIG. 48. 50-TON PLATE ICE PLANT.

to operate, as the ice may be made in thin, quick-freezing moulds, the distilling system and steam boiler dispensed with, and any motive power used for driving the compressor.

To secure best economy in large "plate" system installation, the equipment should include power hoisting crane for lifting ice from tanks; automatic machinery for sawing large cakes into blocks; power ice handlers and conveyors; ample, well insulated ice storage rooms; the main tank freezing cells, plates or coils thoroughly well made with a view to long life and avoiding leakage; abundant fore-cooling water storage. Where steam must be used, adopt high pressure water tube boiler and best make of compound condensing engine, preferably of the Corliss type.

(Penny. Trans. A. S. R. E. 1906. Abridged.)

NOTES ON ICE PLANTS:

Pipe Line Refrigeration

(J. E. Starr, A. S. R. E. Trans. 1906. Abridged.)

Pipe lines are laid, by virtue of public franchise, under the streets and public places of cities for supplying refrigeration to individual consumers. Two methods have been employed for distribution: (a) Brine Circulation; (b) Direct Expansion.

The relation of income and length of main is on an average $12,000 gross income per mile. The various installations range from one mile of mains to seventeen miles.

Brine lines have the usual two pipe flow and return system with refrigerator coils connected in multiple. The brine is cooled in brine coolers of the shell and coil type. Brine pumps are of the triplex type driven by direct connected engines.

The power required for distributing the brine varies directly with the head and the range of the brine. Assuming a range of 5 deg. between the outgoing and incoming brine and a head of 120 feet we have $\frac{200 \times 120}{5 \times 33,000} = 0.14$ H. P. per ton of refrigeration delivered to the brine as measured by the brine. This will call for from 0.23 to 0.28 H. P. at the motor per ton of refrigeration.

The insulation of the mains is effected by laying the pipe in a wooden box and covering with an insulating material soaked in some moisture resisting compound. (Hair felt soaked with a mixture of rosin and paraffin oil or granulated cork soaked in pitch.) Above ground all service lines must, of course, be insulated to and from the wall of the refrigerator.

The loss of refrigerating power by reception of heat coming through the insulation of mains is constant on a given length of main for each division of temperature of the atmosphere, but varies directly: as to percentage of total load, with the load—that is, the greater the load the less the percentage of loss—accurate thermometer readings of brine temperature in the mains at the station and at various points on the line are needed to establish this point.

Ammonia lines have been laid under the three-pipe system, consisting of a liquid line carrying the liquid ammonia under pressure by main and branch to the expansion valves at the refrigerators; a return or vapor line carrying back the gas; and a third line called the vacuum line.

The expansion coils in the refrigerators are connected in multiple between the liquid and the vapor line. The vacuum line is connected at each expansion coil on the coil side of the stop valves on the liquid and vapor lines. Repairs at any refrigerator can thus be made without disturbing the balance of the system. The vacuum line is also connected at manholes for repair purposes on the main lines. It can be used as a bridge line to carry liquid over a block where there may be a leak on the liquid line. Its use is also imperative in extensions of existing lines to carry air or ammonia to test out new lines without disturbing the operation of old ones. The ammonia lines are laid in a conduit of vitrified or salt glazed split sewer pipe. The lower half of the conduit being first laid in concrete, then the ammonia lines are run and tested, then the top half of the conduit is laid on and cemented. Manholes are provided at street intersections in the usual manner of all street service work.

The expansion piping is rather liberally installed, the idea being to have enough piping to superheat the gas to nearly the temperature of the box and prevent frosting out into the return main.

In small refrigerators it is very difficult to prevent frosting out, and wherever possible such boxes are connected in series with other boxes. Where a number of small boxes are grouped as in a hotel or restaurant, a brine cooler is installed, fed from the street lines, and brine circulation is used locally.

Laying out the central station as to tonnage of machinery and provision for increase involves a study of average weather conditions. The annual output must be divided into periods showing average demands by periods and of course the plant must be machined for the highest daily load and for the absolute peak. By taking the average mean monthly temperatures and subtracting from each monthly mean the figure 30 (a little below freezing) the remainders will represent the distribution of the load by months. Working these figures into percentages of the total we have our monthly load curve.

For laying out piping for the distribution of liquid, a drop in pressure between the condenser pressure and the pressure due to the highest temperature likely to exist at any point on the liquid line is to be taken as basis for friction head. As most installations so far are on comparatively level ground, static pressure has not figured extensively, but it carries a limitation if liquid lines running to the upper stories of high buildings are involved. Such lines can not be carried to a height where the loss of head would involve a pressure below the boiling point of the liquid at the temperature surrounding the pipe.

The temperature of the mains in the conduit seldom rises above 75° in the summer. This corresponds to an ammonia pressure of 126.5 lbs. With a condensing temperature of 150 lbs. the distribution of a given tonnage or its corresponding amount of liquid could be calculated on a drop of 23.5 lbs. In practice a drop of 15 lbs. has been considered about the outside allowance for friction head. There always remains in case of change of conditions the alternative of raising the condenser pressure to keep the ammonia in liquid form up to the expansion valves.

It is desirable to hold the back pressure at the station as low as possible in order to obtain the greatest available friction head, thus keeping down the cost of line and also retaining the ability to give low temperatures at refrigerators far from the station and to keep down the cost of expansion piping. For this reason the absorption type of machine has been used largely for pipe line systems as it possesses the advantage of working with economy at low back pressures.

Avoiding freezing business all other classes of refrigeration, say from 28° up, can be carried on the basis of 25 pounds for the highest pressure on the return line and 5 to 10 lbs. at the station, giving a friction head of from 15 to 20 lbs.

In July and August one ton of refrigeration takes care of 2,800 cubic feet of space. One cubic foot of space requires .07 ton per annum. One square foot of insulation requires .204 ton per annum.

The most important question in direct expansion pipe line work is that of leakage of ammonia. In fact, experience has shown that the financial success of the system must stand or fall on this item. Various methods have been tried; finally a system was adopted of anchoring the pipes at definite intervals with expansion joints at definite distance from each anchor, confining the expansion and contraction to definite distances and to calculable limits. The later developments include welding the pipes in a continuous length from manhole to manhole and putting expansion joints at the manhole or U bends on the run.

Of late, apparently successful attempts have been made to weld the pipes in situ by the thermit welding process. This process consists in thoroughly cleaning the ends of the pipes and butting them together. Strong clamps hold the ends firmly one to the other. An iron mould is then clamped around the pipe having an annular opening all around the joint. The thermit is then poured into the mould from a hand crucible. The lighter slag first pours out of the mould, followed by thermit steel, which sinks to the bottom, filling the mould about half way up with steel, and the displaced slag fills the balance of the mould. The great heat of the thermit brings the metal of the pipe to a welding heat. The clamps are drawn towards each other, compressing the butted ends of the pipe, and the weld is complete.

While undoubtedly the major cause of loss from leakage has resulted from worn out or badly put together joints in the line, as a result of expansion and contraction, there will always remain a certain amount of what might be termed insensible leakage. While this will doubtless always exist, its aggregate will not be sufficient to cut a large figure in line expense.

Automatic Refrigerating Machines

In the last few years a machine has been put on the market which is said to be automatic and which may be adapted to any small compressor. The accompanying diagram shows the arrangements of these parts.

The switchboard is equipped with the motor-controlling rheostat, switches, voltmeter, ammeter and scale light, with terminal con-

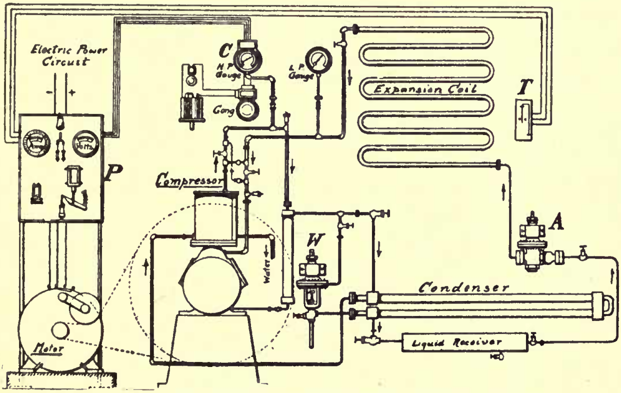


FIG. 49. AUTOMATIC REFRIGERATING MACHINE.

tacts for all wire connections on the back of this panel. The thermostat in the refrigerator is adjusted to operate at any two temperatures: one, above which the temperature in the box must be allowed to rise; and the other, below which it must not fall.

After the plant has been started it will operate until the lower or cold limit of temperature has been reached in the refrigerator. Electric contact is then made in the thermostat, automatically opening the switch so as to stop the motor. The stopping of the process of refrigeration results in the gradual rise of temperature in the refrigerator to the higher limit, when electric contact is made in the thermostat automatically closing the switch and starting the motor again.

As a rule the thermostat is adjusted so that the plant will operate and produce refrigeration within a range of 3° to 4° of variation in the refrigerator; in other words, if the minimum of 36° is desired the plant will operate until this temperature is obtained, when it will stop and not operate again until the temperature rises to 39° or 40°, according to the adjustment of the thermostat.

While in operation the motor and compressor are both working at full load and highest efficiency, and when stopped all expense of operation ceases.

The Automatic Expansion Valve is regulated in the following manner:

Within the valve chamber is fitted an accurately constructed valve mechanism which will only allow a feed of the liquid from the compression side of the system into the expansion side, when the vapor pressure of the expansion side is less than an adjustable and opposed pressure. The proper proportion of feed to meet the requirements of refrigeration in each specific plant can always be determined and regulated by the adjustment provision.

Perfect regulation by this automatic valve is insured by the thermostat control of the motive power, stopping the plant when the temperature has fallen to the desired limit.

The *Automatic Water Regulator* allows the pressure in the condenser pipes to act against a flexible diaphragm, which in turn actuates the valve stem or plunger in the chamber of this regulator; the reverse action being that of a tension spring adjusted to prevent a flow of water when the pressure in the condenser is reduced below the normal, that is, when the plant has been stopped.

The water circuit is provided with a by-pass connection, hand controlled, to permit a flow of water at other times, for example, to flush the water circuit when the plant has been out of service for a long period as it might be during cold weather.

The *Automatic High-pressure Cut-off* is attached to the high-pressure gauge and is so arranged that if the pressure of the condenser, as indicated by the gauge, should for any cause rise far above its normal, then the thermostat circuit is automatically interrupted so as to open the motor switch and stop the operation of compression.

When the pressure falls to the normal or predetermined level, the mechanism restores the control of the plant to the thermostat, which in turn will start or stop the motor-driven compressor in accordance with the temperature conditions in the refrigerator. When the pressure cut-off operates to shut down the plant a special signal gong is automatically sounded to indicate the cause as being abnormal, and an auxiliary bell, on a primary battery circuit, can be placed at a distance so as to indicate each stopping of the plant from this cause, if the water supply should be irregular.

The compressor shown in the present illustrations is built by the Automatic Refrigerating Company of Hartford, Conn.

PART IV—OPERATION OF COMPRESSION PLANT

Erection and Management

The installation of the plant comprises the *proper erection* of machine and apparatus, *testing* the different parts under air pressure and *charging* the system, after which an *efficiency test* is made.

Foundation.

The foundation for engine and compressor must be finished at least two weeks before setting the machines. The following rules should be strictly observed:

Digging.—Dig down to a good, solid bottom, which is never to be less than called for on drawing. Break and remove adjacent rocks, to avoid vibration. Depth of foundation varies from 5 to 8 ft. for small and medium sized machines. As a general rule, the foundation shall weigh approximately 5 times as much as machine.

Concrete.—For the concrete, only Portland cement, sharp and clean stones and sand are to be used. It is to consist of 1 part cement, 3 parts sand, 5 parts stone.

The concrete is to be well rammed down and is to have a level surface. The template should set square and approximately level. The bolts must firmly fit the washers and are then blocked up and adjusted with the nuts, until the bolt ends are level with each other and at the right height above engine house floor. Around the bolts, beginning within 12 inches from the anchor plates, a space 4 × 4 is to be left clear of mortar and other material, or the bolts are encased in a pipe about 4 inches diam., which is removed before machine is put in place.

Surrounding Buildings or Posts.—The concrete should not touch any surrounding parts of the building or post foundations, should not bind on any pipes or other structure, and the contractor has to make sure that no damage can be done by vibration of machine.

Grouting.—After machine is in place, grout with either *cement, sulphur* or *iron rust.* For *cement,* mix equal parts of Portland cement and sharp sand. Add water to make a thin, freely running grout. Build up one layer of bricks around bed plate and foot of machine, then pour in cement, until it sets solid underneath and about half or one inch up on the casting. It will be dry and set properly in two or three days. When using sulphur, make a stiff clay around bed plate, melt sulphur in a large pot over a slow fire and pour quickly with the hand ladle (boils at 239°).

Pipe Connections.—They are usually laid out carefully in the drawing and made up in the shop, measuring not over 4 ft. one way and 20 ft. the other way. All joints on ammonia pipes are screwed and soldered except on some final connections, which must be fitted on job. Suitable hangers must be provided according to character of walls and ceiling.

Testing Plant.

It is important, before introducing the charge of gas into the machine system, to carefully test every part of the apparatus, and make it thoroughly tight under at least 300 pounds air pressure, which pressure may be obtained by working the ammonia compressor and allowing free air to flow into suction side of pump by opening special valves provided for this purpose, the entire system being thus filled with compressed air at the desired pressure. While this pressure is being maintained, a search is instituted for leaks, every pipe, joint, and square inch of surface being tediously scrutinized. One method is to cover all surfaces with a thick lather of soap, leaks showing themselves by formation of soap bubbles. In the case of condenser and brine tank coils, the tanks are allowed to fill with water, the bubbles of air escaping through the water locating the leak. It is important that the apparatus be thoroughly tight, and while each separate piece is carefully tested in the works, transportation and handling may damage, besides a few joints are made on the premises, and it is necessary to go over the entire surface to be sure. While the machine is engaged in pumping air into the system, advantage should always be taken of this opportunity to *purge the system of all dirt and moisture.* To do this properly, valves are provided so the apparatus may be blown out by sections, removing valve bonnets, loosening joints for this purpose, so that it is *positively known* that each pipe, valve and space is strictly clean and purged of all dirt and traces of moisure.

A final test may then be had by *pumping a pressure of* 300 *pounds upon the entire system,* and allowing the apparatus to stand for some hours, estimating the leakage, if any, by noting the degrees of pressure as shown by the pressure gauge connected to system. The air pressure will shrink somewhat at first, by reason of losing heat gained during compression by the pumps. As soon as the air parts with its heat and returns to its normal temperature, the gauge will come to a standstill and remain at a fixed point (depending upon the barometer and changing temperature of the room), if the system is tight.

Do not charge the system until it is well cleansed, purged and tight.

After machinery has been made perfectly tight, *air must be exhausted from the entire system* by working the pumps and discharging the air through the valves provided for this purpose. When the escape of air ceases and the pressure gauges show a full vacuum, it is well to close all outlets and allow the machinery to stand for some time, to test the capacity of the apparatus to withstand external pressure without leakage; in some cases it has been discovered that parts while tight from internal pressure, owing to loose particles lodging over leaks and acting as plugs to prevent escape, these same points, when subjected to an external pressure, give way and disclose the leakage.

Charging Plant.

Connect the flask of ammonia to the charging valve, the gauge still showing a vacuum, close the expansion valve in main liquid pipe connecting receiver to brine tanks. Then open valve on

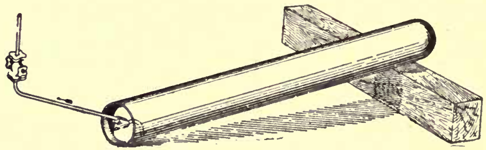


FIG 50.

Position of the tank should be as in Fig 50, the outlet valve pointing upwards and the other end of the tank raised 12″ to 15″. The connection between the outlet valve of the tank and the inlet valve of the system should be a ⅜″ pipe.

ammonia flask and allow the liquid to be exhausted into the system. We recommend placing the flask on small platform scales, in order to weigh the contents and *know positively when cask is exhausted.* Each standard tank contains from 100 to 110 lbs. of ammonia.

The machine may be run all this time at a slow speed, with discharge and suction valves wide open. As one flask is exhausted, place another on scales, and continue until the liquid receiver is shown to be partly full, by the glass gauge thereon. Then shut the charging valve and open and regulate the main expansion valve; the machine is then sufficiently charged to do work, as shown by the pressure gauges and gradual cooling of the brine and frosting of expansion pipe leading to brine tank coils.

While the system is being charged, *water is allowed to flow over the condenser,* and time diligently employed in searching further for leaks, which can readily be detected by sense of smell, each point being again gone over.

Ammonia is a great solvent, and in some cases leaks may be opened up by reason of the gas dissolving substances that may have stopped defective places and withstood the air test.

Amount of liquid in system:

Tons of refr. in 24 hrs..	5	10	15	20	25	50	100	150	200
Lbs. of liquid...........	150	200	250	350	375	425	500	550	750

Add to the above one-third lb. for 1 ft. of 2-inch expansion pipe. Sulphur dioxide machines use about 3 to 4 times, and carbonic acid machines 5 to 6 times as much liquid.

Air in the System.—Negligence in regulating the expansion valve and needlessly pumping a vacuum on the brine tank, carelessly allowing leaky stuffing boxes, may allow air to get into the system, as will also taking the apparatus apart without expelling the air, before the re-introduction of the ammonia gas.

The presence of air in considerable quantity is readily noticed by an expert, by the intermittent action of the expansion valve and singing noise, rise of condensing pressure, loss of efficiency in the condenser, etc. Purging valves are provided on the condenser and other points to allow the imprisoned air to escape, and restore the apparatus to its normal condition of pressure and efficiency.

Pumping Out Connections.

Every compressor should be provided with a by-pass, which enables the engineer to exhaust the ammonia from any part of the system, and temporarily store it in any other part until the repairs or examinations are made.

The by-pass is also used for exhausting the compressors themselves before the heads are removed for examination. By these means we are able to reverse the action of the pumps and exhaust the ammonia from the condenser, storing it in the expansion coils.

In each case, after the examination of any part, the air may be exhausted therefrom and the charge of ammonia re-introduced without the admixture of air.

While the same rules apply to all compressors, we append here some directions governing specific makes, as given by their builders.

Directions for Safety Head Compressors.

To pump out compressor B.—All valves closed. Open main discharge stop valve A1 and by-pass valves 2 and 3. Run machine slowly until compressor cylinder is exhausted, then close by-pass valve 3 and cylinder head may be removed. After replacing cylinder head the air may be expelled by closing main stop valve A1 and discharging through purging valve on head of cylinder A.

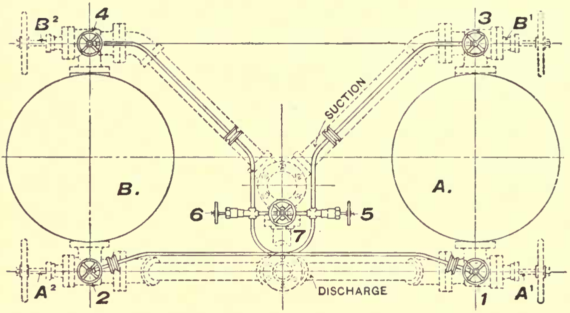


FIG 51—BY-PASS OF SAFETY HEAD COMPRESSOR.

To pump out compressor A, proceed in same manner, using opposite set of valves.

To pump out ammonia condenser and store in evaporating coils or low-pressure side: Open main discharge stop valve A1, by-pass valves 1 and 4, thus connecting to suction of cylinder B, and expelling gas by opening by-pass valves, 2, 5 and 7 into main suction pipe. Run machine slowly.

By using oposite set of valves the other cylinder may be used, as one is used to exhaust the gas from the discharge through by-pass, while the others expels it through the other portion of by-pass into the suction pipe and low-pressure side.

Directions for "Oil" Compressors.

To Pump Out a Condenser.—Close cocks 4, 5, 6 and 8 of those condensers which you *don't* want to pump out. Close cocks 40 and 44 of those condensers you *want* to pump out, the other condensers working during all this time. Open cock 1 and then close main liquid cock 36 and main return cock 42 and run at

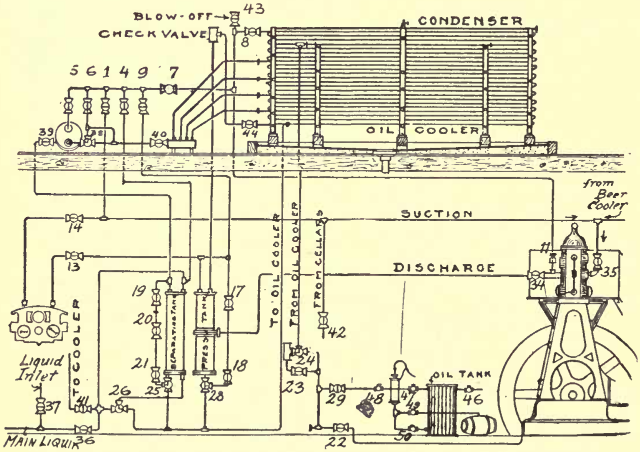


FIG 52—PUMPING-OUT CONNECTIONS OF OIL COMPRESSOR.

reduced speed. Now lower your back pressure to 0 lbs. and keep it there until there is no more frost on the condensers you want to pump out. Don't cut off the water from the condenser you want to pump out. Now close cock 8 of the condenser in question; furthermore, cock 1, and open cocks 4, 5, 6 and cock 8 of the other condensers. You can now break any joint of the condenser in question.

When the joints of the condenser have been made again, open cock 44 of the condenser in question a little, allowing the air to escape at joint of cocks near condenser. When you smell ammonia strongly close this joint and open cock 44 fully; further, cocks 8 and 40 and your condenser is in proper working order.

To pump out main liquid line.—Close cock 36 and also all the expansion cocks but one. Also close all the return cocks except the one corresponding with the expansion cock that was left open, and reduce the back pressure to 0 lbs., and keep it there as long as the pipe shows frost. Then close the last expansion cock and stop the machine. You can now break any joint of this pipe, but you must not touch any cock connecting with it. When all the joints have been made tight again, open cock 36 a little and allow the air confined in the pipe to escape at the farthest joint broken until you smell ammonia strongly. Then close the joint, and you are ready to start the machine.

To Pump Out Brine Cooler, Beer Cooler, Etc.—Close expansion cock leading to the cooler or cellar you want to pump out and see that the corresponding return cock is open. Close main liquid cock 36 and all other 2-inch return cocks, and then reduce your back pressure to 0 pounds, until it will not go up again when you stop the machine or when you run the machine at its slowest speed. Then close the return cock mentioned before, and you can now break any joint of the cooler or cellar expansion, not touching the cocks. The machine may be working during this time and doing work in the other cellars or coolers. If you want it to do this, open all 2-inch return cocks except the one belonging to cooler or cellar you wish to repair, and open cock 36 again, allowing the back pressure to go up to its usual height. When all your joints have been made again, open the expansion cock, before closed, a little, so as to allow some ammonia to enter the cooler or cellar, and then close it again, allowing the air to escape at the joint of the respective cooler or cellar near return cock until you smell ammonia strongly. Then close the joint and open the respective return cock. You can now expand again in this cooler or cellar.

Pumping out storage tank, separating tank, etc., is done in similar manner and no further instructions are required.

Efficiency Test of Refrigerating Plant

The purpose of the test is to determine how the *refrigeration produced* compares with the *amount of work expended* and the *amount of coal consumed.*

Getting ready for test:

1. Engine and compressor have to be provided with *indicators.*
2. Condensing water and circulated brine have to be connected with a *meter.*
3. Temperature of in and outgoing brine and condensing water is to be measured by *thermometers.*
4. Also temperature of ammonia gas, by placing *mercury wells* in the suction and discharge pipe near the compressor.

Indicator Diagram.

The diagram shows:

(a) The actual work done by (engine) or applied to (compressor) a piston during each stroke. H. P. of compressor is product of mean pressure, piston area and piston speed divided by 33,000.

The mean pressure in the compressor may, in the absence of an indicator diagram, be found approximately in the following table.

MEAN PRESSURE IN COMPRESSOR.

Condenser Pressure.		103	115	127	139	153	168	184	200	218
Condenser Temperature.		65°	70°	75°	80°	85°	90°	95°	100°	105°
Refrigerator Pressure.	Refrigerator Temperature									
4	—20°	41.46	43 91	46 34	48 77	51.23	53.68	56.11	58 54	60 99
6	—15°	42.72	45 38	47 90	50 74	53.40	56 08	58 86	61 40	64 08
9	—10°	44.40	47 38	50.33	53 29	56.25	59.20	62.16	65.14	68 09
13	— 5°	45.86	49 15	52 42	55 70	58.97	62.25	65.53	68 81	72.08
16	0°	46 94	50 56	54 16	57 78	61 40	65 00	68.62	72.22	75 84
20	5°	47 74	51 73	55 70	59 68	63 67	67 66	71 62	75 61	79 61
24	10°	48 04	52 40	56 77	61 13	65 51	69 86	74.24	78 59	82 97
28	15°	47 88	52 67	57 44	62.23	67 02	71 81	76 60	81 39	86.18
33	20°	47 08	52 30	57 53	62.75	67.98	73.23	78.46	83 68	88 91
39	25°	45 06	51.34	57 05	62 75	68.46	74 17	79.88	85 58	91 29
45	30°	43.16	49,71	55.92	62 14	68 35	74 56	80.77	86 98	93 19
51	35°	40 52	47 26	54.02	60.76	67.52	74.28	81 02	87 78	94.52

(b) The conditions of pressure at the different positions of the piston, the working of the valves and the changes of temperature.

Figs. 1 to 6 show defective cards.

Figs. 7 and 8 show good cards.

Fig. 9 shows how to plot the isothermal and adiabatic lines by means of the two tables below.

To plot the adiabatic line by means of Table I; Find in the horizontal line with p the number corresponding to the absolute back pressure on your card. Then in the same vertical column that contains your absolute back pressure, and opposite p_9, find the value of p_9. Lay this off on line 9 (Fig. 53, No. 9), from b to b_1, to the same scale as that of your indicator spring. Do the same for p_8, p_7 to p_1. You then have a series of points through which you draw the smooth curve a, b, c. This curve is the adiabatic.

To plot the isothermal line by means of Table II proceed the same as explained in regard to the adiabatic line.

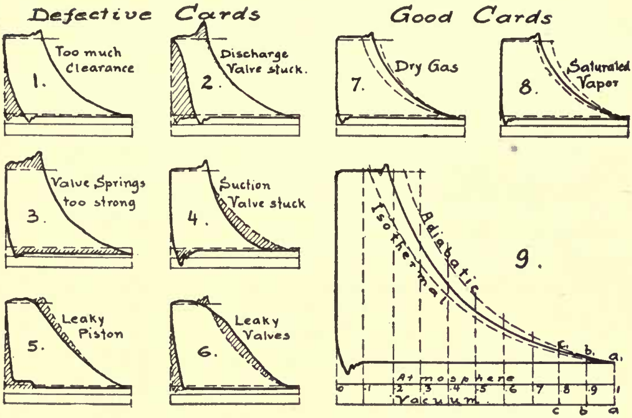


FIG 53.

TABLE I.

ADIABATIC CONSTANTS.

p.	15	16	17	18	19	20	21	22	23	24
p.9	17 2	18 4	19.5	20 6	21.8	22.9	24.1	25.2	26 4	27.6
p.8	20.0	21.4	22.7	24.3	25.4	28.7	28.1	29.4	30.7	33.1
p.7	24.9	25.4	27.0	28.6	30.2	31.8	33.4	35.0	38.5	38.1
p.6	29.2	31.1	33.1	35.0	36.9	88.6	40.8	42.8	44.7	46.7
p.5	37.0	39.5	41.8	44.4	46.8	49.3	51.8	54.3	56.7	59.2
p.4	49.3	52.7	56.8	59.2	62.5	65.8	69.1	72.4	75.7	79.0
p.3	71.7	76.5	81.3	86.1	90.8	95.6	100.3	105.2	110.0	114.8
p.2	122.0	130.0	138.2	146.3	154.5	162.7	170.6	178.9	187.0	195.1
p.1	300.0	320.0	340.0	360.0	380,0	400.0	420.0	440.0	460.0	480.0

p.	25	26	27	28	29	30	31	32	33	34
p.9	28.7	29.8	31,0	32.1	33.2	34.4	35.6	36.7	37.9	39.0
p.8	33.4	34.7	36.1	37.4	38.8	40.1	41.4	42.8	44.1	45.4
p.7	39.7	41.3	42.8	44.5	46.2	47.7	49.3	50.8	52.4	54.0
p.6	48.7	50.6	52.5	54.4	56.4	58.3	60.3	62.2	64.2	66.2
p.5	61.7	64.2	66.6	69.1	71.5	74.0	76.4	78.8	81.4	83.8
p.4	82.3	85.5	88.8	92.2	95.4	98.5	102.0	105.2	108.6	111.8
p.3	119.6	124.2	129.0	133.9	138.7	143.4	148.2	153.0	157.8	162.5
p.2	203.2	211.4	219.4	227.8	235.8	244.0	252.0	260.0	268.2	276.3
p.1	500.0	520.0	540.0	560.0	580.0	600.0	620.0	640.0	660.0	680.0

p.	35	36	37	38	39	40	41	42	43	44
p.9	40.2	41.3	42.5	43.6	44.7	45.9	47.1	48.2	49.3	50.5
p.8	46.8	48.1	49.4	50.8	52.1	53.4	54 8	56.1	57.4	58.8
p.7	55.6	57.2	58.8	60.4	62 0	63.6	65.2	66.7	68.3	69 8
p.6	68.0	70.0	71.9	73.8	75.8	77 8	79.7	81.6	83.6	85.5
p.5	86.3	88.7	91.2	93.7	96.2	98.7	101.0	103.5	106.0	108.3
p.4	115.1	118.4	121.7	125.0	128 2	131.6	135.0	138.1	141.5	144.7
p.3	167.3	172.1	177.0	181.6	186.5	191.2	196.0	200.8	205.8	213.0
p.2	284.7	292 7	300.7	309.0	317.0	325.0	333.4	341.2	349.2	357.8
p.1	700 0	720 0	740 0	760.0	780 0	800.0	820.0	840 0	860 0	880.0

p.	45	46	47	48	49	50	51	52	53	54
p.9	51.7	52.7	53 9	55.1	56.2	57.4	58.5	59.7	60.8	62.0
p.8	60.2	61.5	62.8	64.1	65.4	66.8	68.1	69.4	70.8	72.1
p.7	71.5	73.1	74.7	76.3	77.8	79.5	81.0	82.6	84 2	85.8
p.6	87.5	89 4	91 4	93.8	95.2	97.2	99.1	101.0	103.0	105.0
p.5	110.8	113.2	115.8	118.2	120.5	123.0	125.6	128.0	130.6	133.0
p.4	148.0	151 3	154.6	158.0	161.2	164.5	167.7	171.0	174.3	177.6
p.3	215.0	220.0	224.8	229.5	234.2	239.0	243.8	248.6	253.5	258.1
p.2	366.0	374 0	382.0	390.0	398.0	407.0	414.0	422.0	432.0	438.0
p.1	900.0	920.0	940.0	960.0	980.0	1000.0	1020.0	1040.0	1060.0	1080.0

p.	55	56	57	58	59	60
p.9	63.2	64.3	65.4	66 6	67.7	68.8
p.8	73.5	74.8	76.1	77.5	78.8	80.1
p.7	82.4	89.0	90.5	92.2	93.7	95.3
p.6	107.0	108.9	110.8	112.8	114.8	116.8
p.5	135.5	138.0	140.4	142.8	145.2	147.7
p.4	181.0	184.2	187.5	190.7	194.0	197.8
p.3	263.0	267.8	272.4	277.2	282.0	287.0
p.2	447.0	466.0	463.0	472.0	480.0	487.0
p.1	1100.0	1120.0	1140.0	1160.0	1180.0	1200.0

TABLE II.

ISOTHERMAL CONSTANTS.

p.	15	16	17	18	19	20	21	22	23	21
p.9	16 7	17 8	18 9	20 0	21 1	22 2	23.3	24.5	25 6	26 7
p.8	18.7	20.0	21 2	22 5	23.7	25 0	26.2	27 5	28 7	30 0
p.7	21 4	22 8	24 3	25 7	27 1	28 6	30.0	31 4	32.8	34.3
p.6	25.0	26 7	27 3	30 0	31 7	33 4	35 0	36 7	38.4	40 0
p.5	30 0	32.0	34 0	36 0	38 0	40 0	42 0	44.0	46 0	48.0
p.4	37 5	40 0	42 5	45 0	47 5	50 0	52.5	55.0	57 5	60 0
p.3	50.1	53 4	56 7	60 1	63 4	66.7	70 1	73 4	76 7	80 1
p.2	75 0	80 0	85 0	90 0	95.0	100 0	105 0	110.0	115 0	120.0
p.1	150.0	160 0	170.0	180 0	190 0	200.0	210 0	220 0	230 0	240.0

p.	25	26	27	28	29	30	31	32	33	34
p.9	27 8	28 9	30 0	31 1	32 2	33 3	34 4	35 6	36 7	37 8
p.8	31 2	32.5	33 7	35 0	36 2	37 5	38 7	40 0	41 2	42.5
p.7	35.7	37 1	38 6	40 0	41 4	42 8	44 3	45 7	47 2	48 6
p.6	41.7	43 4	45 0	46 7	48 3	50 0	51 7	53 4	55 0	56.7
p.5	50 0	52 0	54.0	56 0	58 0	60 0	62 0	64 0	66.0	68.0
p.4	62.5	65 0	67 5	70 0	72 5	75 0	77 5	80 0	82.5	85 0
p.3	83.4	86 7	90 1	93 4	96 7	100 1	103 4	106 7	110.1	113 4
p.2	125 0	130 0	135 0	140 0	145 0	150.0	155 0	160.0	165 0	170.0
p.1	250 0	260.0	270.0	280 0	290.0	300 0	310 0	320 0	330 0	340 0

p	35	36	37	38	39	40	41	42	43	44
p.9	38 9	40 0	41 2	42 3	43 4	44 5	45 6	46 7	47 8	48 9
p.8	43 7	45 0	46 2	47 5	48 7	50 0	51.2	52 5	53 7	55 0
p.7	50 0	51 4	52.8	54 3	55 7	57 2	58 6	60 0	61 4	62.8
p.6	58.4	60.0	61 7	63 4	65 0	66 7	68 4	70 0	71 7	73 4
p.5	70 0	72 0	74 0	76 0	78 0	80 0	82.0	84 0	86 0	88 0
p.4	87 5	90 0	92 5	95 0	97 5	100 0	102 5	105 0	107 5	110 0
p.3	116 7	120.1	123 4	126 7	130.1	133 4	136 7	140.1	143 4	146 7
p.2	175 0	180.0	185 0	190 0	195.0	200 0	205 0	210.0	215 0	220 0
p.1	350.0	360 0	370 0	380.0	390 0	400.0	410.0	420.0	430.0	440.0

p.	45	46	47	48	49	50	51	32	53	54
p.9	50 0	51 2	52 3	53 4	54 5	55 6	56 7	57 8	58 9	60 0
p.8	56.2	57 5	58 7	60 0	61 2	62 5	63 7	65 0	66 2	67 5
p.7	64 3	65 7	67 2	68 5	70 0	71 4	72 8	74 3	75 7	77 2
p.6	75 0	76 7	78.4	80 0	81 7	83 4	85 0	86 7	88 4	90.0
p.5	90 0	92 0	94 0	96.0	98.0	100 0	102.0	104 0	106 0	108.0
p.4	112 5	115.0	117.5	120.0	122.5	125 0	127.5	130 0	132.5	135.0
p.3	150 0	153 4	156 7	160.0	163.4	166 7	170 0	173 4	176 7	180 0
p.2	225 0	230 0	235.0	240 0	245 0	250.0	255 0	260 0	265.0	270 0
p.1	450.0	460.0	470.0	480.0	490 0	500.0	510.0	520.0	530.0	540.0

p.	55	56	57	58	59	60
p.9	61 2	62.3	63.4	64.5	65 6	66 7
p.8	68 7	70 0	71 2	72 5	73 7	75 0
p.7	78.5	80 0	81.4	82 8	84.3	85 7
p.6	91 7	93.4	95.0	96 7	98 4	100 0
p.5	110.0	112.0	114 0	116 0	118 0	120 0
p.4	137.5	140.0	142.5	145 0	147 5	150.0
p.3	183 4	186 7	190.1	193.4	196 7	200 1
p.2	275 0	280.0	285.0	290 0	295 0	300.0
p.1	550.0	560.0	570.0	580.0	590 0	600 0

RECORD OF A TEST MADE WITH A "DE LA VERGNE" 32-TON MACHINE AT THE PACKINGHOUSE OF RICHARD WEBBER.

Readings were made every hour for 12 hrs. in succession and the average taken.

Brine meter, 660 cb. ft. p. hr.; water meter, 235 cb. ft. p. hr.; steam gauge, 90 lbs.; back pressure, 22 lbs.; cond. pressure, 140 lbs.; number of rev., 2,880 p. hr. = 48 p. min.; temp. of feed water, 165°; brine temp., initial 17°, final 27°.

Spec. gravity of brine at 60° = 1.119; spec. heat = 0.8326; weight of 1 cb. ft. = 69.83 lbs.; coal used = 3,988 lbs.

Actual refr. capacity $R = P \times s \times (t - t_1) \div 284{,}000$.

P = lbs. of brine circulated in 24 hrs. = 660 × 69.83 × 24 = 1,106,000 lbs.; t = final temp. of brine = 27; t_1 = initial temp. = 17; s = spec. heat of brine = 0.8326.

$R = 1{,}106{,}000 \times 0.8326\ (27 - 17) \div 284{,}000 = 32.3$ *tons in* 24 *hrs.*

Condensing water used per minute = 235 × 7.5 ÷ 60 = 29.3 *gallons.* (1 cb. ft. = 7.5 gallons.)

Rules for Testing Refrigerating Machines.

(Abridged from Preliminary Report to A. S. M. E.)

The unit to measure the cooling effect or the refrigeration is the heat required to melt 1 pound of ice, which is 144 British thermal units, and by dividing the refrigeration measured in British thermal units by 144, the ice melting capacity in pounds is obtained. The unit for a ton of 2,000 pounds of ice melting capacity is therefore 288,000 British thermal units. The tonnage capacity is the refrigerating capacity expressed in tons of ice-melting capacity in 24 hours, and is equivalent to the abstraction of 288,000 British thermal units in 24 hours, or to 12,000 British thermal units per hour, or 200 British thermal units per minute.

The unit for measuring the commercial tonnage capacity is based upon the actual weight of refrigerating fluid circulated between the condenser and the refrigerator, and actually evaporated in the refrigerator.

The actual refrigerating capacity of a machine may be determined from the quantity and range of temperature of the brine, water, or other secondary refrigerating liquid circulated as a refrigerant, and the actual refrigerating capacity under the standard set of conditions should correspond closely to the commercial tonnage capacity.

The standard set of conditions are those which often exist in ice making, namely that the temperature of the saturated vapor at the point of liquefaction in the condenser is 90 degrees F. and the temperature of the evaporation of the liquid in the refrigerator 0 degrees F. This corresponds for ammonia to a condenser pressure of about 168 pounds gauge pressure, and to a gauge pressure of about 15 pounds in the refrigerator.

In the case of air machines, the actual tonnage capacity for a specified set of conditions is obtained by basing the refrigeration on the amount of air cooled and the amount which it is lowered in temperature.

In the Code of Rules the primary refrigerating fluid is considered to be ammonia, but the rules will apply no matter what the refrigerating fluid may be.

In a brine circulating system where brine coils are made use of to produce the refrigerating the capacity of these coils is not therefore taken into account. A test made with a brine heater gives correctly the capacities herein specified.

Calibration of Thermometers.

All the thermometers used should be carefully calibrated before employing them in a test. The 32 degree point may be determined by noting their readings when surrounded by melting ice, and other points by comparing with a standard thermometer which should also be calibrated at its ice point in order to make sure that it is correct.

Thermometers having the graduations marked directly on the glass stems should be used, and these should be placed in wells containing brine or mercury, the wells to extend for at least 2 inches into the space where the fluid circulates. The mercury in the stem of the thermometer should stand a little higher than the top of the well, in order that the readings may be obtained without moving the thermometer. Where the range of temperature through which the refrigerating fluid is cooled is measured in order to determine the capacity of the machine, it is often necessary to measure this range with the highest degree of refinement. For example, if a refrigerating machine cools brine through a range of 5 degrees, one-tenth of a degree will be equivalent to 2 per cent. of the range of temperature, and it is therefore essential that the range should be determined with as great accuracy as possible. In general, it is well to interchange the thermometers which are used for measuring the temperatures of the inlet and outlet brine several times during a test, making note of such changes on the record of the test.

Calibration of Water and Brine Meters.

Where meters are used for determining the amount of refrigerating fluid which is circulated they should be carefully calibrated, both before and after a test, and in some cases, where long tests are made, they should also be calibrated during the test.

In calibrating a meter the measurements should be made with the meter in the position in which it is installed in the test. This is especially necessary where the liquid which is measured is circulated by means of a pump which produces pulsations in the pressure, because the pulsations, as well as the total pressure, must be the same in calibrating the meter as exist in the actual test. In calibrating a meter with either water or brine the temperature of the fluid should be about the same as exists in the test.

Duration of Test.

The duration of a test depends upon its character. If a test is made of an ice making plant, and it is desired to obtain the actual amount of ice made per pound of steam consumed, it may be necessary to make tests of a week or more in duration in order to eliminate as far as possible any error in estimating the amount of ice and cold stored in the freezing tank, which should be made as nearly as possible the same at the end as at the beginning of the test.

Where the refrigerating capacity is measured, the conditions should be made as nearly the same as possible at the beginning and the ending of a test. By making the test of a long enough duration, any error involved through irregularities will be practically eliminated and in most cases all tests should be of at least 8 hours duration.

It is essential that the average temperature of that part of the brine between the points where its temperature is measured and where it is cooled by the evaporation of the ammonia, as well as the quantity of this part of the brine, be the same at the end as at the start of the test. If there is much difference in temperature or quantity, a correction should be applied.

Conditions Existing in Tests.

Where a machine is guaranteed to develop a certain capacity with a certain quantity of condensing water at a certain temperature, it is often necessary to heat the condensing water to the temperature specified in the contract (circulating the water through a heater in which steam is admitted).

All conditions specified in a contract should be followed as closely as possible in making a test.

Amount of Ammonia Circulated and Evaporated.

The anhydrous ammonia must necessarily be measured under pressure. The best method is actually to weigh it, employing two tanks having flexible metallic pipe connections for the purpose.

The arrangement of the two ammonia cylinders for measuring

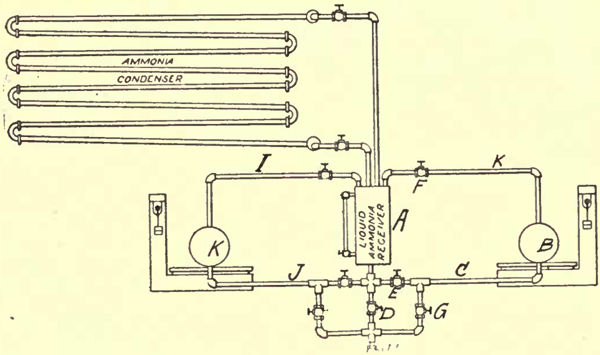


FIG 54—MEASURING ANHYDROUS AMMONIA.

the anhydrous ammonia is shown in diagram. The ammonia receiver installed with the machine is marked *A*, and one of the two tanks for weighing the anhydrous ammonia *B* and the other *K*. In using the tanks for weighing anhydrous ammonia the valve *D* is closed. In filling the tank *B*, the valves *E* and *F* are opened and the valve *G* is closed. After the tank *B* is filled, the valve *E* is closed and the weight determined, after which the valve *G* is opened, and the anhydrous ammonia is allowed to flow from the tank through the throttle valve or cock *H* into the refrigerator. During the time that the anhydrous ammonia is allowed to flow from the tank *B* through the throttle valve or cock *H*, the second tank, *K*, similar in construction to *B*, which is connected to the pipes *I* and *J*, is being filled.

In setting up the apparatus, care must be taken that the horizontal pipes *C*, *K*, *I* and *J* leading to the two tanks, are long enough to allow sufficient flexibility to insure the proper working of the scales. Care must be taken also that the pipes *I* and *K* are so connected that no liquid ammonia can enter them, while the tanks for weighing the ammonia are being emptied. The liquid ammonia receiver must be large enough to allow the level of the liquid to be carried at all times well below the inlets of the pipes *I* and *K*. The tanks *B* and *K* may be covered with a nonconductive covering to diminish the heating or cooling effect of the atmosphere on them. There should be little or no tendency to evaporate the liquid ammonia or to condense the ammonia vapor in the tanks *B* and *K*, and that such is the case may be determined by allowing them to stand for some time with the vent pipes open to the ammonia receiver *A*, and noting whether they gain or lose in weight.

Actual Refrigerating Capacity.—In determining the actual refrigerating capacity of the machine the conditions must be those specified in the contract. For example, if a machine is guaranteed to produce a certain tonnage of refrigeration in cooling a storehouse in summer weather, the test should be made in the summer, if possible, or the capacity of the coils, which are used for refrigerating the various rooms, may be tested by employing relatively warmer brine. If the heat given to the brine is then not sufficient, a heater may be readily constructed of a coil through which the brine passes, which is immersed in steam, so that the required amount of heat is added to the brine.

Specific Heat of Brine Used.—In all cases where the actual refrigerating effect is measured by the cooling produced in the brine circulated, the specific heat of the brine should be determined.

Temperature and Pressure of Ammonia Gas Leaving Refrigerating Coils.—It is necessary in computing the commercial refrigerating capacity from the weight of anhydrous ammonia circulated that the pressure and the temperature of the gas leaving the refrigerator be known. As the pressure of the gas leaving the refrigerator is nearly that existing in the refrigerator, it may be taken as such without sensible error. Unless the gas leaving the refrigerator is superheated, there may be some liquid anhydrous ammonia leaving the refrigerator coils along with the gas. A thermometer at this point is necessary in all tests, because if any liquid ammonia leaves the refrigerator the calculated results will be too great and the machine will be doing less refrigeration than indicated by the measured amount of ammonia circulated.

Temperature of Ammonia at the Expansion Valve.—It is necessary in computing the commercial tonnage capacity that the temperature of the anhydrous ammonia be known on the high pressure side of the expansion valve. A thermometer well should be inserted in the pipe for this purpose.

Commercial Tonnage Capacity.—The commercial tonnage capacity should be computed from the formula:

$$R = \frac{W}{12{,}000}\,[L_2 - q + cp\ (t_1 - t)] \quad (1)$$

Where R = the commercial tonnage capacity or the tons of ice melting capacity per 24 hours.

W = the weight of anhydrous ammonia evaporated in the refrigerating coils in pounds per hour.

L_2 = the total heat above 32 degrees F. of 1 pound of the saturated ammonia gas at the pressure of the refrigerator.

q = the sensible heat above 32 degrees F. contained in 1 pound of the liquid ammonia at the temperature observed before it passes through the expansion valve.

cp = the specific heat of ammonia gas at constant pressure of 0.51.

t_1 = the temperature of the superheated ammonia gas leaving the refrigerator in degrees F.

t = the temperature corresponding to the pressure at which the ammonia gas leaves the refrigerator in degrees F.

The specific heat of liquid anhydrous ammonia is very nearly unity, and if taken at this figure, we obtain (2):

$$R = \frac{W}{12{,}000}\,[H_2 - (T_1 - T_2) + cp\ (t_1 - t)\ ] \quad (2)$$

Where H_2 = the latent heat of evaporation of 1 pound of anhydrous ammonia at the pressure of the refrigerator.

T_1 = the temperature of anhydrous ammonia observed just before it passes through the expansion valve in degrees F.

T_2 = the temperature corresponding to the pressure of the ammonia gas in the refrigerator in degrees F., and the remainder of the notation is the same as in equation (1).

In determining the commercial tonnage capacity it is necessary to make sure that the anhydrous ammonia is pure. In the case of absorption machines, there is usually some water present in the ammonia. The quantity of water should be determined.

Actual Refrigerating Capacity.—The actual refrigerating capacity should be computed from the formula:

$$R_1 = \frac{W_1 c}{12{,}000} (t_2 - t_3) \qquad (3)$$

Where R_1 = the actual tonnage capacity, or the tons of ice melting capacity per 24 hours.

W_1 = the weight of refrigerating fluid circulated per hour.

c = the specific heat of the refrigerating fluid for the range of temperature existing in the tests.

t_2 = the temperature of refrigerating fluid returned to the machine, and

t_3 = the temperature of refrigerating fluid leaving the machine.

Indicator Cards, etc.—Indicator cards should be taken from the steam and ammonia cylinders of a compression machine. Thermometer wells should be placed in the inlet and exit ammonia pipes of a compressor, and the temperatures observed.

Strength of Liquors in Absorption Machine.—The density of the strong and weak liquors should be determined in testing an absorption machine. It is essential in doing this that no gas be allowed to escape from the liquids on drawing from the machine. The liquors should be drawn off through a pipe which is surrounded with cold brine or some other refrigerant, and the density should be determined at a temperature at which there is practically no evaporation.

Heat Balance.—A balance should be made of the various quantities of heat received and rejected by a machine. This is important as proving the accuracy of a test. The following table gives the essential data and results for a test to determine the commercial tonnage capacity:

1. Duration of test..hours
2. Anhydrous ammonia evaporated per hour in the refrigerating coils (W)..lbs.
3. Average condenser pressure above atmosphere, or gauge, pressure (made as near 168 lbs. a square inch above the atmosphere as possible) ....lbs. per sq. in.
4. Average refrigerator pressure above atmosphere or gauge pressure (made as near 15 lbs. a square inch above the atmosphere as possible).........................lbs. per sq. in.
5. Average temperature of liquid ammonia on high pressure side of the throttling valve or cock (T_1)..............deg. F
6. Average temperature of ammonia gas leaving the refrigerator (t_1)..deg. F.
7. Temperature of saturated ammonia gas corresponding to the average refrigerator pressure (T_2)................deg. F.
8. Total heat above 32 degrees F. of 1 pound of saturated ammonia gas at the average refrigerator pressure (L_2)...B. t. u.
9. Sensible heat above 32 degrees F. contained in 1 pound of liquid ammonia at the temperature observed before it passes through the throttle valve or cock (q)..............B. t. u.
10. Commercial tonnage capacity = R as figured by equations (1) and (2).

PART V—THE STEAM PLANT

Steam Engines

Horse-Power.

The *indicated horse-power* is found by the following formula: I. H. P. = a s p ÷ 33,000.

a = piston area in inches (deduct area of rod).
s = piston speed in ft. per min. = 2 × stroke × rev. p. min.
p = mean effective pressure in lbs. p. sq. inch of piston.

The Actual or Brake Horse-Power equals the indicated horse-power less the power required to run the engine itself, which is ordinarily 25% of the total power. The ratio between the indicated and brake horse-power is called *Mechanical Efficiency.*

The *Mean Effective Pressure* is computed from an indicator diagram, or may be obtained approximately from table below.

MEAN EFFECTIVE STEAM PRESSURE.

Cut-off at		$\frac{1}{10}$	$\frac{1}{9}$	$\frac{1}{8}$	$\frac{1}{7}$	$\frac{1}{6}$	$\frac{1}{5}$	$\frac{1}{4}$	$\frac{3}{10}$	$\frac{1}{3}$	$\frac{2}{5}$	$\frac{1}{2}$
Apparent Ratio of Expansion.		10	9	8	7	6	5	4	3.33	3	2 5	2
M. E. P. per Lb. Initial Pressure.		.330	.355	.385	.421	465	.523	.596	.661	.699	.776	.846
Initial Pressure.		Mean Effective Pressure from Full Area of Ideal Diagram.										
Gauge	Absolute.											
40	54.7	18.07	19.42	21.06	23.03	25.44	28.55	32.63	36 15	38.26	41.93	46 31
45	59.7	19.72	21.19	22.98	25.12	27.76	31.16	35.62	39.46	41 76	45.76	50.54
50	64.7	21.37	22.97	24.91	27.23	30.09	33.77	38.60	42.76	45.26	49.59	54.77
55	69.7	23.02	24.74	26.83	29.34	32 31	36.38	41.58	46.07	48.76	53.43	59.00
60	74.7	24.67	26.52	28.75	31.45	34.74	38.99	44 56	49.37	52.26	57.26	63.24
65	79 7	26.32	28.29	30.68	33.55	37 06	41.59	47.55	52 67	55.75	61.09	67.47
70	84.7	27.97	30.07	32.60	35.66	39.39	44.20	50.53	55.98	59.25	64.92	71.70
75	89.7	29.62	31.84	34.53	37 76	41.71	46.81	53.51	59.28	62 75	68.76	75.94
80	94.7	31.28	33.62	36.45	39 87	44.04	49.42	56.50	62.59	66.25	72.59	80.17
85	99.7	32.93	35.39	38.38	41 97	46.36	52.03	59.48	65.89	69.74	76.42	84.40
90	104.7	34.58	37.17	40.30	44.08	48.69	54.64	62.46	69.20	73.24	80.26	88.63
95	109.7	36.23	38.94	42.23	46.18	51.01	57.25	65.44	72.50	76.74	84.09	92.87
100	114.7	37.88	40.72	44.15	48.20	53.34	59.86	68.43	75.81	80.24	87.92	97.10
110	124.7	41.18	44.27	48.00	52.50	57.98	65.08	74.39	82.41	87.23	95.59	105.26
120	134.7	44.49	47.82	51 85	56.71	62.64	70.30	80.36	89.02	94.23	103.25	114.04
130	144.7	47.79	51.37	55.70	60.92	67.29	75.52	86.32	95.63	101.22	110.91	122.50
140	154.7	51.09	54.92	59.55	65.13	71.94	80.74	92.29	102.24	108.22	118.58	130.96
150	164.7	54.39	58.47	63.40	69.34	76.59	85.96	98.26	108.85	115.21	126.26	139.43
160	174.7	57 70	61.02	67.25	73.55	81.24	91.17	104.22	115.46	122.21	133.91	147.89
170	184 7	61.00	65.57	71.10	77.76	85.89	96 39	110.19	122.07	129.20	141 90	156.36
180	194.7	64.30	69.12	74.95	81.97	90.54	101.61	116.15	128.68	136.20	149.24	164.82
190	204.7	67 60	72.67	78.79	86.18	95.19	106.83	122.12	135.29	143.19	156.91	173.29
200	214.7	70.91	76.22	82.64	90.39	99.84	112 05	128.08	141.90	150.19	164.57	181.85
210	224.7	74.21	79.78	86.49	94.60	104.49	117.27	134 05	148.51	157.19	172.24	190.22

To find the highest M. E. P. realized in practice, subtract from the ideal values given in table, 7 lbs. for condensing engines, and 20 lbs. in the case of non-condensing engines.

The ideal M. E. P. for any initial gauge pressure not given in table is found by multiplying your absolute pressure by the M. E. P. per pound of initial, as given in third line of table.

PISTON SPEED IN FEET PER MINUTE.

Ordinary direct-acting pumping engines (non-rotative)	90 to 130
Ordinary horizontal engines	200 to 400
Horiz. comp. and triple-expans. mill engines	400 to 800
Ordinary marine engines	400 to 650
Engines for large high-speed steamships	700 to 900
Locomotive engines (express)	800 to 1,000
Engines for torpedo-boats	1,000 to 1,200

STEAM PER HORSE-POWER PER HOUR.

Plain slide valve engine	60 to 70 lbs.
High speed automatic engine	30 to 50 lbs.
Corliss simple non-cond	25 to 28 lbs.
Corliss comp. non-cond	23 to 26 lbs.
Corliss simple condensing	19 to 21 lbs.
Corliss comp. condensing	13 to 15 lbs.

Valve Setting of Corliss Engine.

The following instructions are given by the Frick Co. and apply to all Corliss engines:

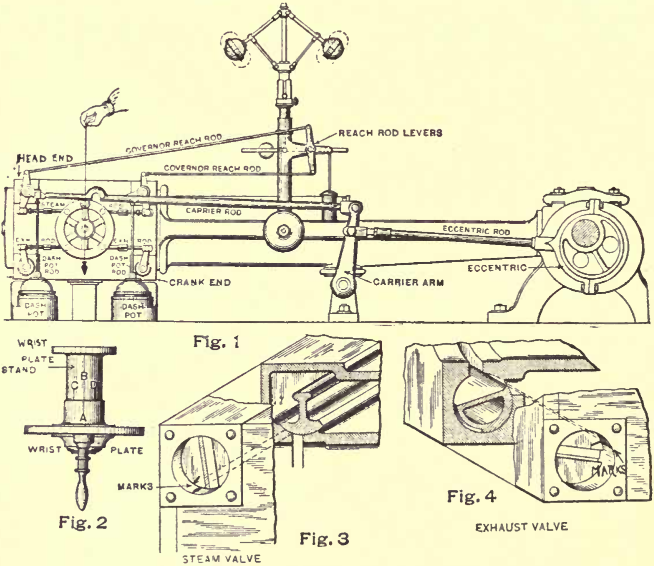


FIG 56—VALVE SETTING OF CORLISS ENGINE.

The Steam and Exhaust Valves.—Take off the back valve chest cover and upon the bore of the seats you will find a mark which coincides with the closing edge of the port. (See Figs. 3 and 4.) Look upon the end of the valve and find a mark running towards the center of valve; this line coincides with the closing edge of valve. Note that in case of the exhaust valve the valve controls the part leading into the exhaust passage and not the opening from the cylinder downward. The upper edge of the exhaust port is the closing edge, and the outer edges of the steam ports are the closing edges.

The Wrist Plate.—You will find a mark upon the hub and corresponding marks upon the hub of the wrist plate, when it is moved back and forth by the eccentric. The wrist plate should be located exactly central between the four valves.

To test the marks on wrist plate hub connect the eccentric rods and engage or drop the carrier rod upon the wrist plate stud; then rotate the eccentric upon the shaft the full extent of its throw or movement each way, and observe if the marks upon the hub of wrist plate at full throw agree with the marks upon the bracket; if not, disconnect the box trap of eccentric rod at carrier arm and adjust the screw on stub end by lengthening or shortening (as required), until the marks do agree on both extremes of movement.

To Set the Valves.—Place the wrist plate in a vertical position (at the central mark); turn the valves around in their seats until the steam valves show by the closing edge marks upon their ends by comparison with the port line marks in the seats that the steam valve edges lap over or cover the ports ¼ *of an inch for* 18-*inch bore of engine cylinder,* ⅜ *for* 24-*inch cylinder,* and 7/16 *for* 30-*inch cylinder.* The *exhaust valves* should show from 1/16 *to* ⅛ *opening,* according to size of cylinder.

In connecting the wrist plate see first that the cut-off latch is hooked on the stud or is engaged. Connect the wrist plate and steam and exhaust valve arms so the wrist plate stands at the central mark or vertical, and the steam and exhaust valve have the proper lap and opening as instructed, the proper amount of steam lap and exhaust opening being determined as above by the size of engine.

To Make Final Adjustments.—Now with the carrier rod hooked upon the wrist plate stud, place the engine upon the center, knowing which way the engine shaft is to run, turn the eccentric upon the shaft (it being loose) in the same direction in which shaft is run, a little more than at right angles ahead of the crank or until the steam valve on the same end as the piston is just beginning to open, *say* 1/32 *of an inch;* in this position secure the eccentric on the shaft by means of the set screws in the hub (*see in all cases that the steam valves are hooked up or engaged by the cut-off mechanism*), then turn the engine on the opposite center and see if the steam valve on that end has the same amount of opening; if not, you can make the adjustment by lengthening or shortening the wrist plate rod attached to this valve.

To Adjust the Cut-off.—See that the governor and connections are put together properly, and block the governor about halfway in the slot; then fasten the reach or cam rod lever so it stands about at right angles to a line drawn midway between the reach rods; then lengthen or shorten the reach rods until the cam or trip levers stand vertical or plumb. The governor and connections now occupy the proper relative positions, and it remains to make the exact adjustment and to equalize the cut-off, so as the same amount of steam is admitted at each end of the stroke. Also, lower the governor and observe when the governor is down that the cut-off mechanism does not unhook, but allows steam to be taken full stroke, after which place the engine at 1-5 of the stroke, which can be done by measuring upon the engine bed guides from each end and turning the engine (*with all parts connected up*) until crosshead is fair with the mark, then slowly raise the governor until the cut-off on the end taking steam trips or unhooks, and to ensure this point being accurately determined it is well to stand by with the hand pressing down upon the dash pot rod; now block the governor in this position and try the cut-off on the other stroke same distance from the end. After a few trials back and forth, and adjusting the length of the cam rods, the cut-off can be made to drop at precisely the same point of stroke. Take care to secure everything permanently when done.

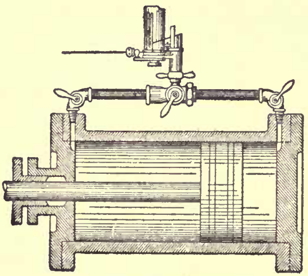
Note: on Automatic Safety Attachment.—As most engines are fitted with safety automatic cams, designed to act only when governor has fallen to bottom of slot in the governor column, before finishing your adjustment see that when the governor is at its proper height it will trip the cut-off. When resting on the high part of the slotted safety collar, the valve gear will follow full stroke, and when safety collar has been turned to bring the notch opposite slot, the governor will drop low enough to allow the safety cams or knock-off lever to be brought into play so as not to permit the valves to be opened.

The dash pot rod should be adjusted in length so the steam valve arm, resting thereon, when the dash-pot plunger is home, or at the bottom of the pot, is in such a position that the latch is sure to hook over the latch stud and the stud lies midway between the latch die and the closing shoulder. This will insure on the one hand the positive engagement of the latch, and on the other hand prevent the shoulder from jamming down upon the latch stud. If the dash-pot rod is too short, the latch will not hook on.

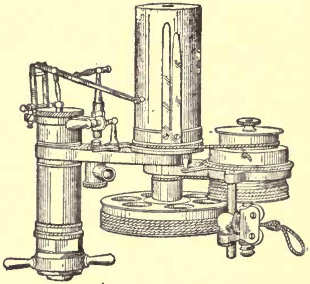
The regular gag pot is used on Corliss Engines to prevent over-sensitiveness of the governor and its response to trivial changes. Use only coal or kerosene oil in this pot, and regulate the screw in the piston if required to give greater freedom of motion. *See that all parts of the governor move freely.*

If the latch dies have a tendency to slip, the latch spring may be at fault. It can be made stronger by twisting the spring stud, bringing more tension against the latch. If the stoppage comes from wear, take out the latch or stud die and turn it, thus presenting a new wearing surface, or sharpen edge by applying to a grindstone. Do not bring any more pressure on the spring than necessary, as when steel dies are in good condition the tension of spring can be very light. Keep the cushion leathers in good order and your valve gear working noiseless and smooth.

Using a Steam Engine Indicator
to test the correctness of valve setting is the most approved method known, and should be applied in cases where an indicator can be obtained. Recollect that *to adjust the point of cut-off*



Cross-pipe connection.
FIG 57.



Indicator and reducing-wheel.
FIG 58.

to take same amount of steam at each end, adjust the cam or reach rods. To *give* more or less steam lead adjust the wrist plate rods. *Lengthening* them increases the *lap* and *shortening* them gives more *lead.* The same with the exhaust valves, the cushion or release being effected thereby. If the eccentric is properly set, it is not necessary to disturb it in ordinary cases.

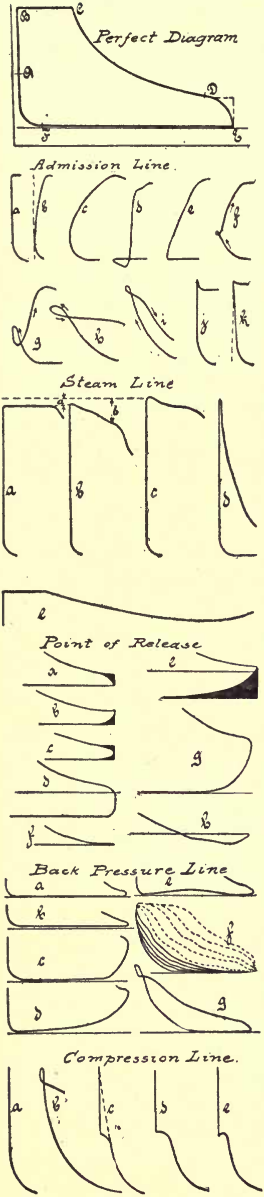


FIG 59.

The lines of a *perfect diagram* are as follows:

A to B is the *"admission line,"* showing that ports and clearance space are filled with steam.

B to C is the *"steam line,"* showing that sufficient steam is admitted to the cylinder up to the point of cut-off at C.

C to D is the *"expansion line,"* showing the work done by the expansion of the steam while piston travels from point of cut-off to point of release at D.

D to E is the *"release line,"* where the exhaust valve opening lets the steam escape from the cylinder.

E to F is the *back pressure line,* showing the amount of pressure on the back of the piston.

At F occurs the *exhaust closure,* and F to A is the *coompression line,* showing how the pressure is raised.

ADMISSION LINE.

a. Normal. b. Not sufficient lead.
c. Not sufficient lead (slide valve).
d. Steam admitted too late.
e. Exhaust valve closing too late.
f and g. Too much compression for late steam opening.
h and i. Too much compression (slide valve).
j and k. Too much lead.

STEAM LINE.

a. Normal. b. Steam ports or steam pipe too small.
c. Too large steam chest area.
d. No load on engine.
e. Piston speed too great (slide valve).

POINT OF RELEASE.

a. Normal. b. Release too late.
c. Counterpressure at moment of normal release.
d. Release too early.
e. Release too late (condensing).
f and h. Light load or early cut-off.
g. Late cut-off.

BACK PRESSURE LINE.

a. Normal, b, c and d. Insufficient exhaust area.
e. Small exhaust ports.
f. Continuous diagram with varying load.
g. Early closure of valve.

COMPRESSION LINE.

a. Normal. b. Excess. compression.
c and d. Leakage in valves or piston.
e. Leakage in piston.

Taking Care of Corliss Engine.

Before starting your engine, see that all the water is blown out of the steam pipe by means of the drip valve provided on steam valve elbow; then open the steam valve a little and allow the steam to blow through the cylinder, first one end, then the other, by moving the wrist plate by hand sufficient to let the steam pass through the valves. The cylinder soon becomes warm, and all water is expelled into the exhaust pipe, the exhaust drain cock having been left open to allow it to run off. When ready to start, *let the engine move slowly* until you are satisfied everything is all right, then open stop-valve wide, and leave same open at all times.

Don't work the wrist plate motion by hand and run engine backward and forward; the carrier rod is provided with a detachable hook so wrist plate may be worked for the purpose of warming up steam cylinder and blowing through.

When machine is stopped, wipe it down clean, and examine all bearings and parts. Before starting again, see that all oil cups are properly filled and in working order, and all oil holes clear. Use none but the best oil, and use no more of it than is required to keep bearings in good working condition.

Air Pumps.

For a *jet-condensing engine* the capacity of the vertical single-acting pump varies from 1/5 to 1/10 of the capacity of the low-pressure cylinder, and from 1/8 to 1/16 in case of a horizontal double-acting pump.

For a *surface-condensing engine* the capacity of the s. a. pump would be from 1/10 to 1/8, and of a d. a. pump 1/15 to 1/25 of that of the low-pressure cylinder.

The above proportions are for pumps having the same number of strokes as the piston of the low-pressure cylinder.

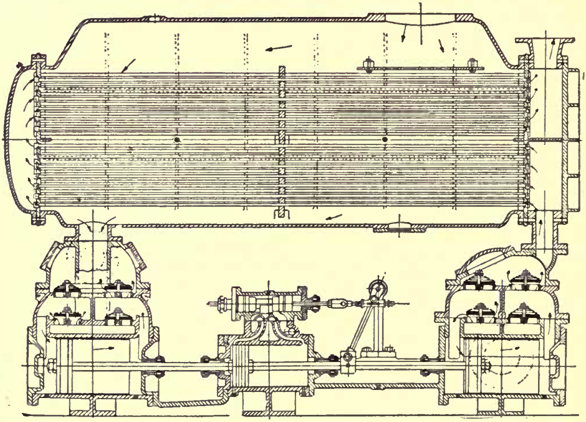


FIG 60—SURFACE CONDENSER WITH AIR AND CIRCULATING PUMP.

SIZES AND DIMENSIONS OF STANDARD CORLISS ENGINES.

SIZE OF CYLIND'R.		No. of Revolutions.	INDICATED HORSE POWER.									FLY WHEEL.			CRANK SHAFT					STEAM PIPES.		Approx.
			80 lbs. Pressure.			90 lbs. Pressure.			100 lbs. Pressure.													Shipping
Diameter.	Stroke.		1-5 Cut-off.	1-4 Cut-off.	1-3 Cut-off.	1-5 Cut-off.	1-4 Cut-off.	1-3 Cut-off.	1-5 Cut-off.	1-4 Cut-off	1-3 Cut-off	Diameter.	Face.	Weight in Pounds.	Diameter.	Length.	Center Shaft above Found	Center Shaft to back of Cyl'd'r Head		Diam. Steam	Diam. Exh'st.	Weight Pounds.
Inch.	Inch.											Ft.	In.		In.	Ft.	In.	Ft.	In.	In.	In.	
10	24	90	30	36	43	34	41	48	39	49	60	8	13	3500	5	6	25	10	11	3	4	10700
10	30	90	37	45	53	42	51	60	48	58	68	8	13	4300	5	6	25	13		3	4	11750
12	30	90	54	65	76	62	74	87	69	83	97	9	15	5700	6	6½	25	13	4	3½	4½	15550
12	36	85	61	73	86	70	84	98	78	94	110	10	17	6300	6	7	25	15	5	3½	4½	16900
14	36	85	83	100	118	95	114	134	107	128	150	10	19	8900	7	7½	25½	15	6	4	5	22300
14	42	82	93	112	132	107	128	140	120	144	168	12	19	8900	7	7½	25½	17	7	4	6	22700
16	36	82	105	126	140	120	144	167	135	162	187	12	21	10500	8	8	27	15	9	4½	6	26610
16	42	78	116	139	163	133	159	188	150	179	210	12	23	11800	8	8	27	17	10	4½	6	28930
18	36	80	129	155	182	148	177	208	166	199	233	12	25	12600	9	9	30	15	11	5	7	30000
18	42	78	147	176	208	168	202	237	189	227	266	14	25	13300	9	9	30	18		5	7	33800
18	48	75	162	194	227	185	222	259	208	249	291	15	25	15300	9	9	30	20	1	5	7	39570
20	42	75	175	210	246	200	240	280	225	270	314	15	29	16600	10	9	30	18	4	6	8	43450
20	48	72	192	230	268	219	263	307	246	296	346	16	29	18700	10	9	30	20	5	6	8	47612
22	42	75	211	254	298	242	290	340	271	326	382	16	31	21000	11	10	32	18	5	6	8	50600
22	48	72	232	278	308	265	318	368	298	358	410	16	31	23100	11	10	32	20	6	6	8	56220
24	48	70	268	322	380	307	368	432	345	414	486	18	33	24400	12	11	32	20	11	7	9	65140
24	60	65	311	374	438	356	427	500	401	421	561	18	37	30500	12	11	32	25	2	7	9	75740
26	48	70	315	378	445	360	432	507	405	486	570	18	37	29000	13	11½	32	21		7	9	77210
26	60	65	366	439	512	418	502	586	470	564	658	18	37	31000	13	11½	32	25	3	7	10	80000
28	48	68	355	426	503	406	487	568	457	548	680	18	37	31000	14	12	34	21	3	8	12	77920
28	54	68	400	477	554	461	546	631	523	616	709	18	37	31000	14	12	34	23	4	8	12	83000
28	60	65	424	509	594	485	582	655	545	654	763	18	37	31000	14	12	34	25	6	8	12	89000
30	48	68	407	489	571	466	559	652	524	629	734	22	46	33000	15	12½	36	21	8	8	12	93000
30	60	62	464	557	650	531	637	743	597	717	837	24	52	38500	15	12½	36	25	11	8	12	104000
30	72	55	494	593	693	565	678	791	635	762	890	24	60	52000	15	13	36	30	2			127000
32	48	65	443	532	622	507	608	710	570	684	800	24	48	34500	16	13	36	21	11			
32	60	62	529	634	739	604	725	846	680	816	952	24	60	43900	16	13	36	26	2			
32	72	55	563	675	787	643	772	902	723	868	1013	24	60	58200	16	14	36	30	5			

Horse Power.

The standard rating is as follows: *One horse-power equals* 30 *lbs. of water evaporated p. hr., from feed water, at* 100° *F. into dry steam of* 70 *lbs. gauge pressure.*

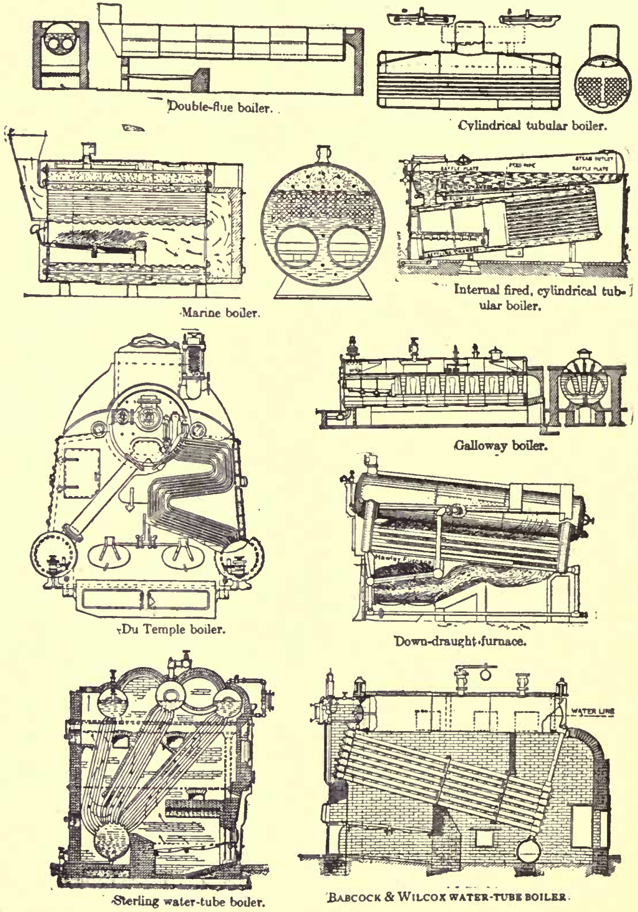


FIG 61—VARIOUS TYPES OF STEAM BOILERS.

This is equivalent to the evaporation of 34.5 lbs. of water from a feed water temp. of 212° F. into dry steam at the same temp. and under atm. press.

APPROXIMATE PROPORTION OF HEATING-SURFACE AND GRATE-SURFACE PER HORSE-POWER, ETC., OF VARIOUS TYPES OF BOILERS.

TYPE OF BOILER.	Square feet of heating-surface per horse-power.	Coal per square foot of heating-surface.	Relative economy.	Relative rapidity of steaming.	Heating-surface per square foot of grate.	Pounds of coal per square foot of grate.	Pounds of water per pound of coal.
Water-tube.....	10 to 12	.3	1.00	1.00	35 to 50	12 to 20	9 to 12
Cylind'l tubular.	14 " 16	.25	.91	.60	25 " 35	10 " 15	8 " 11
Vertical tube....	15 " 20	.25	.80	.60	25 " 30	10 " 15	8 " 10
Locomotive.....	12 " 16	.275	.85	.55	50 " 100	20 " 40	8 " 11
Flue	8 " 12	.4	.79	.25	20 " 25	10 " 20	8 " 10
Plain cylindrical.	6 " 10	.5	.69	.20	15 " 20	15 " 25	7 " 9

Horizontal Tubular Boilers.

Capacity tons in 24 hrs.		Boilers required		Dimensions of each boiler			Appr. Weight of Boilers		Brick Setting above floor line		Iron Smoke Stacks		Duplex Boiler Feed Pumps	Space req. for One Boiler inch.
Refr.	Ice	No	H.P.	Dia. inch.	Length ft.	No. of 3" tubes	Fixtures lbs.	Total lbs.	No. of Red Bricks	No. of Fire Bricks	Dia. in.	Weight ft.		L W H
5	2.3	1	20	36	10	24	2650	5300	5500	400	15	30	4½ × 2¾ × 4	170 × 72 × 89
10	4.5	1	35	44	12	48	5000	8500	6500	500	20	35	"	200 × 80 × 97
15	—	7	45	48	14	52	6600	9700	10000	600	20	40	"	224 × 92 × 103
20-25	10	1	60	54	14	66	7870	12800	12500	700	26	40	"	206 × 112 × 118
—	15	1	70	54	16	66	8770	13600	13500	750	26	50	"	230 × 112 × 118
35	20	1	80	60	16	80	10700	17000	17500	800	28	60	"	254 × 112 × 118
50	25	1	100	66	16	102	13300	20000	20000	1000	32	60	5¼ × 3½ × 5	256 × 118 × 125
75	30	1	125	72	16	132	16200	25000	22500	1100	35	60	"	260 × 124 × 131
90	40	1	150	72	18	132	17700	26800	24000	1300	35	60	"	284 × 124 × 131
100-20	50	2	100	66	16	102	26600	42000	34000	2000	45	60	"	
150	60	2	125	72	16	132	32400	53000	38000	2200	48	70	"	
200-20	75	2	150	72	18	132	35400	57000	42000	2600	48	70	6 × 4 × 6	
	90	3	115	66	18	102	43800	70000	49500	3000	55	60	"	
	100	3	125	72	16	132	48600	80000	53000	3300	60	70	"	
	120	3	150	72	18	132	53100	95000	56000	3900	60	70	7½" × 4½" × 6"	

When possible each boiler to be furnished with independent stack.
We recommend in all cases One more boiler than specified.

Fuel.

The value of *fuel* is measured by the number of heat units which its combustion will generate. The fuel is composed of carbon and hydrogen, and ash, with sometimes small quantities of other substances not materially affecting its value.

"Combustible" is that portion which will burn; the ash or residue varying from 2 to 36 per cent. in different fuels.

"Slack" or the screenings from coal, when properly mixed—anthracite and bituminous—and burned by means of a blower is nearly equal in value of combustible to coal, but its percentage of refuse is greater.

Petroleum has a heating capacity, when fully burned, equal to from 21,000 to 22,000 B. T. U. per pound, or say 50 per cent. more than coal. But owing to the ability to burn it with less losses, it has been found that it is equal to 1.8 pounds of coal. A gallon of petroleum is equivalent to twelve pounds of coal, and 190 gallons are equal to a gross ton of coal. It is very easy with these data to determine the relative cost.

It has been estimated that on an average one pound of coal is equal, for steam-making purposes, to 2 lbs. dry peat, 2¼ to 2½ lbs. dry wood, 2:½ to 3 lbs. dried tanbark, 2½ to 3 lbs. sun-dried bagasse, 2¾ to 3 lbs. cotton stalks, 3¼ to 3¾ lbs. wheat or barley straw, 5 to 6 lbs. wet bagasse, and 6 to 8 pounds wet tan-bark.

Natural gas varies in quality, but is usually worth 2 to 2½ times the same weight of coal, or about 30,000 cubic feet are equal to a ton of coal.

TABLE OF COMBUSTIBLES.

Kind of Combustible.	Air Required.	Temperature of Combustion.				Theoretical Value.		Highest Attainable Value under Boiler.	
	In Pounds per pound of Combustible.	With Theoretical Supply of Air.	With 1½ Times the Theoretical Supply of Air.	With Twice the Theoretical Supply of Air.	With Three Times the Theoretical Supply of Air.	In Pounds of Water raised 1° per pound of Combustible.	In Pounds of Water evaporated from & at 212°, with 1 lb. Combustible.	With Chimney Draft.	With Blast, Theoretical Supply of Air at 60°, Gas 320°.
Hydrogen	36.00	5750	3860	2860	1940	62032	64.20		
Petroleum	15.43	5050	3515	2710	1850	21000	21.74	18.55	19.90
Carbon { Charcoal, Coke, Anthracite Coal, }	12.13	4580	3215	2440	1650	14500	15.00	13.30	14.14
Coal—Cumberland	12.06	4900	3360	2550	1730	15370	15.90	14.28	15.06
" Coking Bituminous	11.73	5140	3520	2680	1810	15837	16.00	14.45	15.19
" Cannel	11.80	4850	3330	2540	1720	15080	15.60	14.01	74.76
" Lignite	9.30	4600	3210	2490	1670	11745	12.15	10.78	11.46
Peat—Kiln dried	7.68	4470	3140	2420	1660	9660	10.00	8.92	9.42
" Air dried 25 per cent. water	5.76	4000	2820	2240	1550	7000	7.25	6.41	6.78
Wood—Kiln dried	6.00	4080	2910	2260	1530	7245	7.50	6.64	7.02
" Air dried 20 per cent. water	4.80	3700	2607	2100	1490	5600	5.80	4.08	4.39

AMERICAN COALS.

COAL. State. Kind of Coal.	Per cent. of Ash.	Theoretical Value.		COAL. State. Kind of Coal.	Per cent of Ash.	Theoretical Value	
		in Heat Units.	Pounds of water evap.			in Heat Units.	Pounds of water evap.
Penn. Anthracite	3.49	14.199	14.70	Ill. Bureau Co.	5.20	13,025	13.48
" "	6.13	13,535	14.01	" Mercer Co.	5.60	13,123	13.58
" "	2.90	14,221	14.72	" Montauk	5.50	12,659	13.10
" Cannel	15.02	13,143	13.60	Ind. Block	2.50	13,588	14.38
" Connellsville	6.50	13,368	13.84	" Caking	5.66	14 146	14.04
" Semi-bit'nous	10.70	13,155	13.62	" Cannel	6.00	13,097	13.56
" Stone's Gas	5.00	14,021	14.51	Md. Cumberland	13.88	12,226	12.65
" Youghiogheny	5.60	14,265	14.76	Ark. Lignite	5.00	9,215	9.54
" Brown	9.50	12,324	12.75	Col. "	9.25	13,562	14.04
Kentucky Caking	2.75	14,391	14.89	" "	4.50	13,866	14.35
" Cannel	2.00	15,198	16.76	Texas "	4.50	12,962	13.41
" "	14.80	13,360	13.84	Wash. Ter. Lignite	3.40	11,551	11.96
" Lignite	7.00	9,326	9.65	Penn. Petroleum		20,746	21.47

SIZES OF CHIMNEYS WITH APPROPRIATE HORSE-POWER BOILERS.

Dia. in inches.	Height of Chimneys. / Commercial Horse-Power.											Effective Area, square ft.	Actual Area, square ft.	Side of square of ap'roximate area. Inches.
	50 ft	60 ft	70 ft	80 ft.	90 ft	100 ft.	110 ft.	125 ft.	150 ft.	175 ft.	200 ft.			
18	23	25	27									0.97	1.77	16
21	35	38	41									1.47	2.41	19
24	49	54	58	62								2.08	3.14	22
27	65	72	78	83								2.78	3.98	24
30	84	92	100	107	113							3.58	4.91	27
33		115	125	133	141							4.47	5.94	30
36		141	152	163	173	182						5.47	7.07	32
39			183	196	208	219						6.57	8.30	35
42			216	231	245	258	271					7.76	9.62	38
48				311	330	348	365	389				10.44	12.57	43
54				363	427	449	472	503	551			13.51	15.90	48
60				505	539	565	593	632	692	748		16.98	19.64	54
66					658	694	728	776	849	918	981	20.83	23.76	59
72					792	835	876	934	1023	1105	1181	25.08	28.27	64
78						995	1038	1107	1212	1310	1400	29.73	33.18	70
84						1163	1214	1294	1418	1531	1637	34.76	38.48	75
90						1344	1415	1496	1639	1770	1893	40.19	44.18	80
96						1537	1616	1720	1876	2027	2167	46.01	50.27	86

Water for Feeding Boilers.

should be soft, and deposit no sediment in the boiler. When it contains a large amount of scale-forming material it is usually advisable to purify it before allowing it to enter the boiler, instead of attempting the prevention of scale by the introduction of chemicals into the boiler.

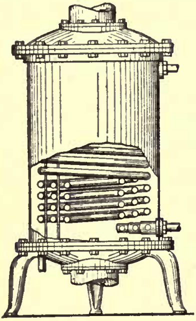
Carbonates of lime and magnesia may be removed to a considerable extent by simply heating the water in an exhaust-steam feed water heater or still better by a live-steam heater.

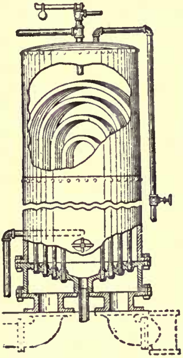
When the water is very bad, it is best treated with chemicals—lime, soda-ash, caustic soda, etc.

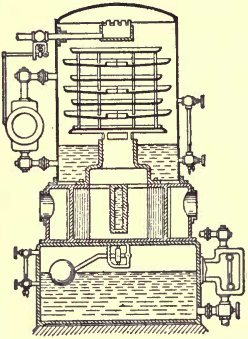
TREATMENT OF BOILER FEED WATER.

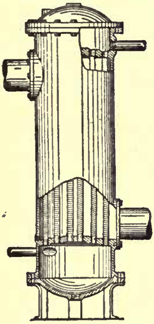
Cause of trouble.	Incrustation.	Treatment of water.
Carbonate of lime	Soft scale	Slaked lime, sal-soda.
" of magnesia	"	" " "
Sulphate of lime	Hard scale	Sal-soda, caustic soda.
" of magnesia	"	Slaked lime and sal-soda.
Chloride of magnesia	Corrosion	Sal-soda, or caustic soda.
Sediment of sand, clay, and mud	Precipitation, or soft scale.	Alum, and filter.
Organic matter	Foaming and corrosion	Slaked lime, sal-soda, or caustic soda.
Alkaline water	Foaming	Frequent blowing off from boiler, or neutralize with hydrochloric acid.
Acid waters	Corrosion	Slaked lime, sal-soda.

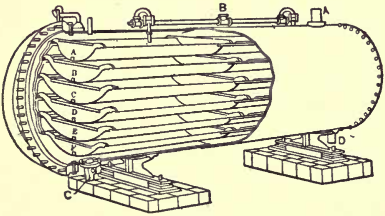
Feed Water Heaters.


Multicoil-heater.


Berryman heater.


Cookson heater, purifier, and oil-separator.


Wainwright heater.



Hoppes feed-water heater.

FIG 62—VARIOUS TYPES OF FEED WATER HEATERS.

Percentage of Saving in Fuel by Heating Feed-Water. Steam at 70 Pounds Gauge-Pressure.

Initial temperature (feed.)	Temperature to Which Feed is Heated.													
	100°	110°	120°	130°	140°	150°	160°	170°	180°	190°	200°	210°	220°	250°	300°
35°	5.53	6.38	7.24	8.09	8.95	9.89	10.66	11.52	12.38	13.24	14.09	14.95	15.81	19.40	29.34
40°	5.12	5.97	6.84	7.69	8.66	9.42	10.28	11.14	12.00	12.87	13.73	14.59	15.45	18.89	28.78
45°	4.71	5.57	6.44	7.30	8.16	9.03	9.90	10.76	11.62	12.49	13.36	14.22	15.09	18.37	28.22
50°	4.30	5.16	6.03	6.89	7.76	8.64	9.51	10.38	11.24	12.11	12.98	13.85	14.72	17 87	27.67
55°	3.89	4.75	5.63	6.49	7.37	8.24	9.11	9.99	10.85	11.73	12.60	13.48	14.35	17.38	27 12
60°	3.47	4.34	5.21	6.08	6.96	7.84	8.72	9.60	10.47	11.34	12.22	13.10	13.98	16.86	26.56
65°	3.05	3.92	4.80	5.67	6.56	7.44	8.32	9.20	10.08	10.96	11.84	12.72	13.60	16.35	26.02
70°	2.62	3.50	4.38	5.26	6.15	7.03	7.92	8.80	9.68	10.57	11.45	12.34	13.22	15.84	25.47
75°	2.19	3.07	3.96	4.84	5.73	6.62	7.51	8.40	9.28	10.17	11 06	11 95	12.84	15.33	24.92
80°	1.76	2.65	3.54	4.42	5.32	6.21	7.11	8.00	8.8	9.78	10.67	11.57	12.46	14.81	24.37
85°	1.30	2.22	3.11	4.00	4.90	5.80	6.70	7.59	8.48	9.38	10.28	11.18	12.07	14.32	23.82
90°	0.89	1.78	2.68	3.58	4.48	5.38	6.28	7.18	8.07	8.98	9.88	10.78	11.68	13.82	23.27
95°	0.45	1.34	2.25	3.15	4.05	4.96	5.86	6.77	7.66	8.57	9.47	10.38	11.29	13.31	22.73
100°	0.00	0.90	1.81	2.71	3.62	4.53	5.44	6.35	7.25	8.16	9.07	9.98	10.88	12.80	22.18

Steam.

"Saturated Steam" is steam of the temperature due to its pressure—not superheated. "Superheated Steam" is steam heated to a temperature above that due to its pressure.

"Dry Steam" is steam which contains no moisture. It may be either saturated or superheated.

"Wet Steam" is steam containing intermingled moisture, mist or spray. It has the same temperature as dry saturated steam of the same pressure.

Flow of Steam in Pipes.

The flow of steam through pipes is calculated after the following formula:

$$W = 87\sqrt{\frac{D\ (P_1 - P_2)\ d^5}{L\left(1 + \frac{3.6}{d}\right)}}$$

W = weight of steam in lbs., which will flow per minute through a pipe of the length L in feet and the diameter d in inches; P_1 = initial pressure; P_2 = pressure at end of pipe; D = weight per cubic foot of the steam.

Steam at atmospheric pressure flows into a vacuum at the rate of about 1,550 feet per second, and flows into the atmosphere at the rate of 650 feet per second.

Heating by Steam

One square foot radiating surface, with steam at 212°, will heat 100 cubic feet of air per hour from zero to 150°, or 300 cubic feet from zero to 100° in the same time.

Where the condensed water is returned to the boiler, or where low pressure of steam is used, the diameter of mains leading from the boiler to the radiating surface should be equal, in inches, to *one-tenth the square root of the radiating surface, mains included,* in square feet. Thus a 1-inch pipe will supply 100 square feet of surface, itself included. Return pipes should be at least ¾ inch in diameter, and never less than one-half the diameter of the main—longer returns requiring larger pipe.

One square foot of boiler surface will supply from 7 to 10 square feet of radiating surface. Small boilers for house use should be much larger proportionately than large plants. Each horse-power of boiler will supply from 240 to 360 feet of 1-inch steam pipe, or 80 to 120 square feet of radiating surface.

Under ordinary conditions one horse-power will heat, approximately, in

Brick dwellings, in blocks, as in cities.... 15,000 to 20,000 cub. ft.
Brick stores, in block .... 10,000 to 15,000 cub. ft.
Brick dwellings, exposed all round........ 10,000 to 15,000 cub. ft.
Brick mills, shops, factories, etc......... 7,000 to 10,000 cub. ft.
Wooden dwelling, exposed............... 7,000 to 10,000 cub. ft.
Foundries and wooden shops............. 6,000 to 10,000 cub. ft.
Exhibition buildings, largely glass, etc... 4,000 to 15,000 cub. ft.

In heating buildings care should be taken to supply the neces-

PROPERTIES OF SATURATED STEAM.

Gauge Pressure per Square in Pounds.	Pressure above Vacuum per Sq. Inch in Pounds.	Temperature in Degrees Fahrenheit.	Total Heat in Heat Units at 32° F.	Heat in Liquid from 32° in Heat Units.	Heat of vaporization or latent Heat in Heat Units	Density or Weight of 1 Cubic Foot in Pounds.	Volume of one Pound in Cubic Feet.	Factor of equivalent Evaporation at 212° Fahrenheit.
....	1	101.99	1113.1	70.0	1043 0	0.00299	334 50	0.9661
....	2	126 27	1120.5	94.4	1026 1	0.00576	173.60	.9738
....	3	141 62	1125.1	109.8	1015.3	0 00844	118 50	.9786
....	4	153 09	1128 6	121 4	1007 2	0 01107	90.33	.9822
....	5	162 34	1131.5	130 7	1000.8	0 01366	73.21	.9852
....	6	170 14	1133 8	138.6	995 2	0.01622	61 65	9876
....	7	176.80	1135.9	145.4	990 5	0.01874	53 39	9897
....	8	182.02	1137.7	151.5	986.2	0 02125	47.06	9916
....	9	188.33	1139.4	156.9	982.5	0 02374	42.12	.9934
....	10	193.25	1140.9	161.9	979 0	0.02621	38.15	.9949
0	14.7	212 00	1146.6	180 7	966 0	0.03793	26.78	1 0000
03	15	213 03	1146.9	181.8	965 1	0.03826	26.14	1 0003
5.3	20	227 95	1151.5	196 9	954.6	0.05023	19.91	1 0051
10.3	25	240 04	1155.1	209.1	946 0	0.06199	16.13	1.0099
15.3	30	250.27	1158.3	219.4	938.9	0.07360	13.59	1.0129
20.3	35	259.19	1161 0	228.4	932.6	0.08508	11.75	1.0157
25.3	40	207.13	1163 4	236.4	927.0	0.09644	10.37	1 0182
30.3	45	274.29	1165.6	243.6	922 0	0.1077	9.285	1.0205
35.3	50	280 85	1167 6	250.2	917 4	0.1188	8.418	1.0225
40.3	55	286.89	1179.4	256 3	913.1	0.1299	7 698	1.0245
45.3	60	292.51	1171.2	261.9	909.3	0.1409	7.097	1.0263
50.3	65	297 77	1172.7	267.2	905 9	0.1519	6.583	1.0280
55.3	70	302 71	1174.3	272.2	902.1	0.1628	6 143	1.0295
60.3	75	307 38	1175 7	276.9	898 8	0.1736	5.760	1.0309
65.3	80	311 80	1177 0	281.4	895 6	0.1843	5.426	1.0323
70.3	85	316.02	1178 3	285.8	892 5	0 1951	5.126	1.0337
75.3	90	320.04	1179 6	290.0	889 6	0 2058	4.859	1.0350
80.3	95	323 89	1180.7	294.0	886 7	0.2165	4.619	1.0362
85.3	100	327 58	1181.9	297.9	884.0	0.2271	4.403	1.0374
90.3	105	331.13	1182.9	301.6	881 3	0 2378	4 205	1.0385
95.3	110	334.56	1184.0	305.2	878.8	0.2484	4 026	1.0396
100.3	115	337.86	1185.0	308.7	876 3	0 2589	3.862	1.0406
105.3	120	341.05	1186.0	312 0	874 0	0.2695	3 711	1.0416
110.3	125	344.13	1186 9	315 2	871 7	0.2800	3.571	1.0426
115 3	130	347.12	1187.8	318.4	869.4	0.2904	3.444	1.0435
125.3	140	352.85	1189.5	324.4	865.1	0.3113	3.212	1.0453
135.3	150	358 26	1191.2	330.0	861.2	0.3321	3.011	1.0470
145.3	160	363 40	1192.8	335.4	857 4	0 3530	2 833	1.0486
155.3	170	368.29	1194 3	340.5	853.8	0.3737	2 676	1.0502
165.3	180	372.07	1195.7	345 4	850 3	0.3945	2.535	1.0517
175.3	190	377.44	1197.1	350.1	847.0	0.4153	2.408	1.0531
185.3	200	381.73	1198.4	354 6	843.8	0.4359	2.294	1.0545
210.3	225	391.79	1201.4	365.1	836 3	0.4876	2.051	1.0576
235.3	250	400.99	1204 2	374 7	829.5	0.5393	1.854	1.0605
260.3	275	409 50	1206 8	383 6	823.2	0.5913	1.691	1.0632
285.3	300	417 42	1209 3	391.9	817.4	0.644	1.553	1.0657
310.3	325	424 82	1211.5	399 6	811.9	0.696	1.437	1.0680
335.3	350	431 90	1213 7	406 9	806 8	0.728	1.3:37	1.0703
360.3	375	438 40	1215.7	414.2	801.5	0 800	1 250	1.0724
385.3	400	445 15	1217.7	421.4	796.3	0.853	1 172	1.0745
485.3	500	466 57	1224 2	444 3	779 9	1 065	0 939	1 0812

sary moisture to keep the air from becoming "dry" and uncomfortable. For comfort, air should be kept at about "50 per cent. saturated." This would require one pound of vapor to be added each 2,500 cubic feet heated from 32° to 70°.

Care of Boilers.

1. *Safety Valves.*—Great care should be exercised to see that these valves are ample in size and in working order. *Overloading* or *neglect* frequently lead to the most disastrous results. Safety valves should be tried at least once every day to see that they will act freely.

2. *Pressure Gauge.*—The steam gauge should stand at zero when the pressure is off, and it should show same pressure as the safety valve when that is blowing off. If not, then one is wrong, and the gauge should be tested by one known to be correct.

3. *Water Level.*—The first duty of an engineer before starting or at the beginning of his watch, is to see that the water is at the proper height. Do not rely on glass gauges, floats or water alarms, but try the gauge cocks. If they do not agree with water gauge, learn the cause and correct it.

4. *Gauge Cocks and Water Gauges* must be kept clean. Water gauge should be blown out frequently, and the glasses and passages to gauge kept clean.

5. *Feed Pump or Injector.*—These should be kept in perfect order, and be of ample size. It is always safe to have two means of feeding a boiler. Check valves, and self-acting feed valves should be frequently examined and cleaned. Satisfy yourself frequently that the valve is acting when the feed pump is at work.

6. *Low Water.*—In case of low water, immediately cover the fire with ashes (wet if possible) or any earth that may be at hand . If nothing else is handy use fresh coal. Draw fire as soon as it can be done without increasing the heat. Neither turn on the feed, start or stop engine, or lift safety valve until fires are out, and the boiler cooled down.

7. *Blisters and Cracks.*—These are liable to occur in the best plate iron. When the first indication appears there must be no delay in having it carefully examined and properly cared for.

8. Fusible Plugs, when used, must be examined when the boiler is cleaned, and carefully scraped clean on both the water and fire sides, or they are liable not to act.

9. *Firing.*—Fire evenly and regularly, a little at a time. Moderately thick fires are most economical, but thin firing must be used where the draught is poor. Take care to keep grates evenly covered, and allow no air-holes in the fire. Do not "clean" fires oftener than necessary. With bituminous coal, a "coking fire," i. e., firing in front, shoving back when coked, gives best results if properly managed.

10. *Cleaning.*—All heating surfaces must be kept clean outside and in, or there will be a serious waste of fuel. Never allow over 1/16 inch scale or soot to collect on surfaces between cleanings. Handholes should be frequently removed and surfaces examined, particularly in case of a new boiler, until proper intervals have been established by experience.

The *exterior* of tubes can be kept clean by the use of blowing pipe and hose. In using smoky fuel, it is best to occasionally brush the surfaces when steam is off.

11. *Hot Feed Water.*—Cold water should never be fed into any boiler when it can be avoided, but when necessary it should be caused to mix with the heated water before coming in contact with any portion of the boiler.

12. *Foaming.*—When foaming occurs in a boiler, checking the outflow of steam will usually stop it. If caused by dirty

water, blowing down and pumping up will generally cure it. In cases of violent foaming, check the draft and fires.

13. *Air Leaks.*—Be sure that all openings for admission of air to boiler or flues, except through the fire, are carefully stopped. This is frequently an unsuspected cause of serious waste.

14. *Blowing Off.*—If feed-water is muddy or salt, blow off a portion frequently, according to condition of water. Empty the boiler every week or two, and fill up afresh. When surface blow-cocks are used, they should be often opened for a few minutes at a time. Make sure no water is escaping from the blow-off cock when it is supposed to be closed. Blow-off cocks and check-valves should be examined every time the boiler is cleaned.

15. *Leaks.*—When leaks are discovered, they should be repaired as soon as possible.

16. *Blowing Off.*—Never empty the boiler while the brick-work is hot.

17. *Dampness.*—Take care that no water comes in contact with the exterior of the boiler from any cause, as it tends to corrode and weaken the boiler. Beware of all dampness in seatings or coverings.

18. *Galvanic Action.*—Examine frequently parts in contact with copper or brass, where water is present, for signs of corrosion. If water is salt or acid, some metallic zinc placed in the boiler will usually prevent corrosion, but it will need attention and renewal from time to time.

19. *Rapid Firing.*—In boilers with thick plates or seams exposed to the fire, steam should be raised slowly, and rapid or intense firing avoided.

20. *Standing Unused.*—If a boiler is not required for some time, empty and dry it thoroughly. If this is impracticable, fill it quite full of water, and put in a quantity of common washing soda. External parts exposed to dampness should receive a coating of linseed oil.

21. *General Cleanliness.*—All things about the boiler room should be kept clean and in good order. Negligence tends to waste and decay.

Rules for Conducting Boiler Test.

The Committee of the A. S. M. E. on Boiler Tests, consisting of Wm. Kent (chairman), J. C. Hoadley, R. H. Thurston, Chas. E. Emery, and Chas. T. Porter, recommended the following code of rules for boiler tests (Trans., vol. vi., p. 256):

Preliminaries to a Test.

I. In preparing for and conducting trials of steam boilers the specific object of the proposed trial should be clearly defined and steadily kept in view.

II. Measure and record the dimensions, position, etc., of grate and heating surfaces, flues and chimneys, proportion of air space in the grate surface, kind of draught, natural or forced.

III. Put the boiler in good condition. Have heating surface clean inside and out, grate bars and sides of furnace free from clinkers, dust and ashes removed from back connections, leaks in masonry stopped, and all obstructions to draught removed. See that the damper will open to full extent, and that it may be closed when desired. Test for leaks in masonry by firing a little smoky fuel and immediately closing damper. The smoke will then escape through the leaks.

IV. Have an understanding with the parties in whose interest the test is to be made as to the character of the coal to be used. The coal must be dry, or, if wet, a sample must be dried carefully and a determination of the amount of moisture in the coal made, and the calculation of the results of the test corrected accordingly. Wherever possible, the test should be made with standard coal of a known quality. For that portion of the country east of the Allegheny Mountains good anthracite egg coal or Cumberland semi-bituminous coal may be taken as the standard for making tests. West of the Allegheny Mountains and east of the Missouri River, Pittsburg lump coal may be used. *

V. In all important tests a sample of coal should be selected for chemical analysis.

VI. Establish the correctness of all apparatus used in the test for weighing and measuring. These are: 1. Scales for weighing coal, ashes, and water. 2. Tanks, or water meters for measuring water. Water-meters, as a rule, should only be used as a check on other measurements. For accurate work the water should be weighed or measured in a tank. 3. Thermometers and pyrometers for taking temperatures of air, steam, feed water, waste gases, etc. 4. Pressure gauges, draught gauges, etc.

VII. Before beginning a test, the boiler and chimney should be thoroughly heated to their usual working temperature. If the boiler is new, it should be in continuous use at least a week before testing, so as to dry the mortar thoroughly and heat the walls.

VIII. Before beginning a test, the boiler and connections should be free from leaks, and all water connections, including blow and extra feed pipes, should be disconnected or stopped with blank flanges, except the particular pipe through which water is to be fed to the boiler during the trial. In locations where the reliability of the power is so important that an extra feed pipe must be kept in position, and in general when for any other reason water pipes other than the feed pipes cannot be disconnected, such pipes may be drilled so as to leave openings in their lower sides, which should be kept open throughout the test as a means of detecting leaks, or accidental or unauthorized opening of valves. During the test the blow-off pipe should remain exposed.

If an injector is used it must receive steam directly from the boiler being tested, and not from a steam pipe or from any other boiler.

See that the steam pipe is so arranged that water of condensation cannot run back into the boiler. If the steam pipe has such an inclination that the water of condensation from any portion of the steam pipe system may run back into the boiler, it must be trapped so as to prevent this water getting into the boiler without being measured.

Starting and Stopping a Test.

A test should last at least ten hours of continuous running, and twenty-four hours whenever practicable. The conditions of the boiler and furnace in all respects should be, as nearly as possible, the same at the end as at the beginning of the test. The steam pressure should be the same, the water level the same, the fire upon the grates should be the same in quantity and condition, and the walls, flues, etc., should be of the same temperature. To secure as near an approximation to exact uniformity as possible in conditions of the fire and in temperatures of the walls and flues, the following method of starting and stopping a test should be adopted:

X. *Standard Method.*—Steam being raised to the working pressure, remove rapidly all the fire from the grate, close the damper, clean the ash pit, and as quickly as possible start a new fire with weighed wood and coal, noting the time of starting the test and the

* These coals are selected because they are about the only coals which contain the essentials of excellence of quality, adaptability to various kinds of furnaces, grates, boilers, and methods of firing, and wide distribution and general accessibility in the markets.

height of the water level while the water is in a quiescent state, just before lighting the fire.

At the end of the test remove the whole fire, clean the grates and ash pit, and note the water level when the water is in a quiescent state; record the time of hauling the fire as the end of the test. The water level should be as nearly as possible the same as at the beginning of the test. If it is not the same, a correction should be made by computation, and not by operating pump after test is completed. It will generally be necessary to regulate the discharge of steam from the boiler tested by means of the stop-valve for a time while fires are being hauled at the beginning and at the end of the test, in order to keep the steam pressure in the boiler at those times up to the average during the test.

XI. *Alternate Method.*—Instead of the Standard Method above described, the following may be employed where local conditions render it necessary:

At the regular time for slicing and cleaning fires have them burned rather low, as is usual before cleaning, and then thoroughly cleaned; note the amount of coal left on the grate as nearly as it can be estimated; note the pressure of steam and the height of the water level—which should be at the medium height to be carried throughout the test—at the same time; and note this time as the time of starting the test. Fresh coal, which has been weighed, should now be fired. The ash pits should be thoroughly cleaned at once after starting. Before the end of the test the fires should be burned low, just as before the start, and the fires cleaned in such a manner as to leave the same amount of fire, and in the same condition, on the grates as at the start. The water level and steam pressure should be brought to the same point as at the start, and the time of the ending of the test should be noted just before fresh coal is fired.

During the Test.

XII. *Keep the Conditions Uniform.*—The boiler should be run continuously, without stopping for meal times or for rise or fall of pressure of steam due to change of demand for steam. The draught being adjusted to the rate of evaporation or combustion desired before the test is begun, it should be retained constant during the test by means of the damper.

If the boiler is not connected to the same steam pipe with other boilers, an extra outlet for steam with valve in same should be provided, so that in case the pressure should rise to that at which the safety valve is set it may be reduced to the desired point by opening the extra outlet, without checking the fires.

If the boiler is connected to a main steam pipe with other boilers, the safety valve on the boiler being tested should be set a few pounds higher than those of the other boilers, so that in case of a rise in pressure the other boilers may blow off, and the pressure be reduced by closing their dampers, allowing the damper of the boiler being tested to remain open, and firing as usual.

All the conditions should be kept as nearly uniform as possible, such as force of draught, pressure of steam, and height of water. The time of cleaning the fires will depend upon the character of the fuel, the rapidity of combustion, and the kind of grates. When very good coal is used, and the combustion not too rapid, a ten-hour test may be run without any cleaning of the grates, other than just before the beginning and just before the end of the test. But in case the grates have to be cleaned during the test, the intervals between one cleaning and another should be uniform.

XIII. *Keeping the Records.*—The coal should be weighed and delivered to the firemen in equal portions, each sufficient for about one hour's run, and a fresh portion should not be delivered until the previous one has all been fired. The time required to consume each portion should be noted, the time being recorded at the instant of firing the first of each new portion. It is desirable that at the same time the amount of water fed into the boiler should be accurately noted and recorded, including the height of the water in the boiler and the average pressure of steam and temperature of feed during the time. By thus recording the amount of water evaporated by successive portions of coal, the record of the test may be divided into several divisions, if desired, at the end of the test, to discover the degree of uniformity of combustion, evaporation, and economy at different stages of the test.

XIV. *Priming Tests.*—In all tests in which accuracy of results is important, calorimeter tests should be made of the percentage of moisture in the steam, or of the degree of superheating. At least ten such tests should be made during the trial of the boiler, or so many as to reduce the probable average error to less than one per cent., and the final records of the boiler test corrected according to the average results of the calorimeter tests.

On account of the difficulty of securing accuracy in these tests, the greatest care should be taken in the measurements of weights and temperatures. The thermometers should be accurate within a tenth of a degree, and the scales on which the water is weighed to within one hundredth of a pound.

Analyses of Gases.—Measurement of Air-Supply, Etc.

XV. In tests for purposes of scientific research, in which the determination of all the variables entering into the test is desired, certain observations should be made which are in general not necessary in tests for commercial purposes. These are the measurement of the air supply, the determination of its contained moisture, the measurement and analysis of the flue gases, the determination of the amount of heat lost by radiation, of the amount of infiltration of air through the setting, the direct determination by calorimeter experiments of the absolute heating value of the fuel, and (by condensation of all the steam made by the boiler) of the total heat imparted to the water.

The analysis of the flue gases is an especially valuable method of determining the relative value of different methods of firing, or of different kinds of furnaces. In making these analysis great care should be taken to procure average samples—since the composition is apt to vary at different points of the flue.

Record of the Test.

XVI. A "log" of the test should be kept on properly prepared blanks, containing headings as follows:

Time.	Pressures.			Temperatures.					Fuel.		Feed water.	
	Barometer.	Steam-gauge.	Draught-gauge.	External Air.	Boiler-room.	Flue.	Feed-water.	Steam.	Time.	Lbs.	Time.	Lbs. or cu. ft.

1. Date of trial..................................			
2. Duration of trial	hours.		
DIMENSIONS AND PROPORTIONS.			
eave space for complete description.			
3. Grate-surface....wide....long....area.......	sq. ft.		
4. Water-heating surface.........................	sq. ft.		
5. Superheating surface........................	sq. ft.		
6. Ratio of water-heating surface to grate-surface..................................			
AVERAGE PRESSURES.			
7. Steam-pressure in boiler, by gauge....	lbs.		
8. Absolute steam-pressure........................	lbs.		
9. Atmospheric pressure, per barometer........	in.		
10. Force of draught in inches of water.........	in.		
AVERAGE TEMPERATURES.			
11. Of external air............................	deg.		
12. Of fire-room..................................	deg.		
13. Of steam....................................	deg.		
14. Of escaping gases..........................	deg.		
15. Of feed-water..............................	deg.		
FUEL.			
16. Total amount of coal consumed	lbs.		
17. Moisture in coal..............................	per cent.		
18. Dry coal consumed........................	lbs.		
19. Total refuse, dry.......pounds =..........	per cent.		
20. Total combustible (dry weight of coal, Item 18; less refuse. Item 19).......	lbs.		
21. Dry coal consumed per hour.	lbs.		
22. Combustible consumed per hour............	lbs.		
RESULTS OF CALORIMETRIC TESTS.			
23. Quality of steam, dry steam being taken as unity..			
24. Percentage of moisture in steam.............	per cent.		
25. Number of degrees superheated..............	deg.		
WATER.			
26. Total weight of water pumped into boiler and apparently evaporated	lbs.		
27. Water actually evaporated, corrected for quality of steam	lbs.		
28. Equivalent water evaporated into dry steam from and at 212° F.	lbs.		
29. Equivalent total heat derived from fuel in British thermal units	B.T.U		
30. Equivalent water evaporated into dry steam from and at 212° F. per hour.............	lbs.		
ECONOMIC EVAPORATION.			
31. Water actually evaporated per pound of dry coal, from actual pressure and temperature	lbs.		
32. Equivalent water evaporated per pound of dry coal from and at 212° F.	lbs.		
33. Equivalent water evaporated per pound of combustible from and at 212° F.	lbs.		
COMMERCIAL EVAPORATION.			
34. Equivalent water evaporated per pound of dry coal with one sixth refuse, at 70 pounds gauge-pressure, from temperature of 100° F. = Item 33 × 0.7249..........................	lbs		

	RATE OF COMBUSTION.			
35.	Dry coal actually burned per square foot of grate-surface per hour........................	lbs.		
36.	Consumption of dry coal per hour. Coal assumed with one sixth refuse. — Per sq. ft. of grate-surface......	lbs.		
37.	Consumption of dry coal per hour. Coal assumed with one sixth refuse. — Per sq. ft. of water-heating surface..	lbs.		
38.	Consumption of dry coal per hour. Coal assumed with one sixth refuse. — Per sq. ft. of least area for draught.	lbs.		
	RATE OF EVAPORATION.			
39.	Water evaporated from and at 212° F. per sq. ft. of heating-surface per hour...........	lbs.		
40.	Water evaporated per hour from temperature of 100° F. into steam of 70 lbs. gauge-pressure. — Per sq. ft. of grate-surface..........	lbs.		
41.	Water evaporated per hour from temperature of 100° F. into steam of 70 lbs. gauge-pressure. — Per sq. ft. of water-heating surface.	lbs.		
42.	Water evaporated per hour from temperature of 100° F. into steam of 70 lbs. gauge-pressure. — Per sq. ft. of least area for draught.	lbs.		
	COMMERCIAL HORSE-POWER.			
43.	On basis of thirty pounds of water per hour evaporated from temperature of 100° F. into steam of 70 pounds gauge-pressure (= 34½ lbs. from and at 212°)	H.P.		
44.	Horse power, builders' rating, at... ...square feet per horse power............... .	H.P.		
45.	Per cent developed above, or below, rating§.	per cent.		

Reporting the Trial.

XVII. The final results should be recorded upon a properly prepared blank, and should include as many of the following items as are adopted for the specific object for which the trial is made.

Results of the trial of a ....

Boiler at ....

To determine ....

NOTES ON STEAM BOILERS:

Pumps

Pressure and Head.

To find the *pressure* in lbs. per square inch of a column of water, multiply the height of the column in feet by .433.

To find the *height* of a column of water in feet, the pressure being known, multiply the pressure shown on gauge by 2.309.

The mean pressure of the atmosphere is usually estimated at 14.7 lbs. per square inch, so that with a perfect vacuum it will sustain a column of mercury 29.9 inches, or a column of water 33.9 feet high, at sea level.

PRESSURE AND HEAD.

Feet Head.	Pressure per Square Inch.	Feet Head.	Pressure per Square Inch.	Feet Head.	Pressure per Square Inch.	Feet Head.	Pressure per Square Inch.	Feet Head.	Pressure per Square Inch.
1	0.43	64	27.72	127	55.01	190	82.30	253	109.59
2	0.86	65	28.15	128	55.44	191	82.73	254	110.03
3	1.30	66	28.58	129	55.88	192	83.17	255	110.46
4	1.73	67	29.02	130	56.31	193	83.60	256	110.89
5	2.16	68	29.45	131	56.74	194	84.03	257	111.32
6	2.59	69	29.88	132	57.18	195	84.47	258	111.76
7	3.03	70	30.32	133	57.61	196	84.90	259	112.19
8	3.46	71	30.75	134	58.04	197	85.33	260	112.62
9	3.89	72	31.18	135	58.48	198	85.76	261	113.06
10	4.33	73	31.62	136	58.91	199	86.20	262	113.49
11	4.76	74	32.05	137	59.34	200	86.63	263	113.92
12	5.20	75	32.48	138	59.77	201	87.07	264	114.36
13	5.63	76	32.92	139	60.21	202	87.50	265	114.79
14	6.06	77	33.35	140	60.64	203	87.93	266	115.22
15	6.49	78	33.78	141	61.07	204	88.36	267	115.66
16	6.93	79	34.21	142	61.51	205	88.80	268	116.09
17	7.36	80	34.65	143	61.94	206	89.23	269	116.52
18	7.79	81	35.08	144	62.37	207	89.66	270	116.96
19	8.22	82	35.52	145	62.81	208	90.10	271	117.39
20	8.66	83	35.95	146	63.24	209	90.53	272	117.82
21	9.09	84	36.39	147	63.67	210	90.96	273	118.26
22	9.53	85	36.82	148	64.10	211	91.39	274	118.69
23	9.96	86	37.25	149	64.54	212	91.83	275	119.12
24	10.39	87	37.68	150	64.97	213	92.26	276	119.56
25	10.82	88	38.12	151	65.40	214	92.69	277	119.99
26	11.26	89	38.55	152	65.84	215	93.13	278	120.42
27	11.69	90	38.98	153	66.27	216	93.56	279	120.85
28	12.12	91	39.42	154	66.70	217	93.99	280	121.29
29	12.55	92	39.85	155	67.14	218	94.43	281	121.72
30	12.99	93	40.28	156	67.57	219	94.86	282	122.15
31	13.42	94	40.72	157	68.00	220	95.30	283	122.59
32	13.86	95	41.15	158	68.43	221	95.73	284	123.02
33	14.29	96	41.58	159	68.87	222	96.16	285	123.45
34	14.72	97	42.01	160	69.31	223	96.60	286	123.89
35	15.16	98	42.45	161	69.74	224	97.03	287	124.32
36	15.59	99	42.88	162	70.17	225	97.46	288	124.75
37	16.02	100	43.31	163	70.61	226	97.90	289	125.18
38	16.45	101	43.75	164	71.04	227	98.33	290	125.62
39	16.89	102	44.18	165	71.47	228	98.76	291	126.05
40	17.32	103	44.61	166	71.91	229	99.20	292	126.48
41	17.75	104	45.05	167	72.34	230	99.63	293	126.92
42	18.19	105	45.48	168	72.77	231	100.06	294	127.35
43	18.62	106	45.91	169	73.20	232	100.49	295	127.78
44	19.05	107	46.34	170	73.64	233	100.93	296	128.22
45	19.49	108	46.78	171	74.07	234	101.36	297	128.65
46	19.92	109	47.21	172	74.50	235	101.79	298	129.08
47	20.35	110	47.64	173	74.94	236	102.23	299	129.51
48	20.79	111	48.08	174	75.37	237	102.66	300	129.95
49	21.22	112	48.51	175	75.80	238	103.09	310	134.28
50	21.65	113	48.94	176	76.23	239	103.53	320	138.62
51	22.09	114	49.38	177	76.67	240	103.96	330	142.95
52	22.52	115	49.81	178	77.10	241	104.39	340	147.28
53	22.95	116	50.24	179	77.53	242	104.83	350	151.61
54	23.39	117	50.68	180	77.97	243	105.26	360	155.94
55	23.82	118	51.11	181	78.40	244	105.69	370	160.27
56	24.26	119	51.54	182	78.84	245	106.13	380	164.61
57	24.69	120	51.98	183	79.27	246	106.56	390	168.94
58	25.12	121	52.41	184	79.70	247	106.99	400	173.27
59	25.55	122	52.84	185	80.14	248	107.43	500	216.58
60	25.99	123	53.28	186	80.57	249	107.86	600	259.90
61	26.42	124	53.71	187	81.00	250	108.29	700	303.22
62	26.85	125	54.15	188	81.43	251	108.73	800	346.54
63	27.29	126	54.58	189	81.87	252	109.16	900	389.86
								1000	433.18

Horse-Power.

The *theoretical horse-power* required to elevate water to a given height is found by multiplying the total weight of water in lbs. by the height in ft. and dividing by 33,000; or, by multiplying the gallons per minute by the height in ft. and dividing by 4,000. (Allowance of 25 per cent. should be added for friction.)

PUMP HORSE POWER REQUIRED TO RAISE WATER.

Gals. per min.	5′	10′	15′	20′	25′	30′	35′	40′	45′	50′	60′	75′	90′	100′	125′	150′
5	.006	.012	.019	.025	031	.037	.044	.05	.06	.06	.07	.09	.11	.12	16	.19
10	012	.025	.037	.050	062	.075	.087	.10	.11	.12	.15	.19	.22	.25	.31	.37
15	019	.037	.056	.075	.094	.112	.131	.15	.17	.19	.22	.28	.34	.37	.47	.56
20	.025	.050	.075	.100	.125	.150	.175	.20	.22	.25	.30	.37	.45	.50	62	.75
25	.031	.062	.093	.125	.156	.187	.219	.25	.28	.31	.37	.47	.56	.62	78	.94
30	.037	.075	.112	.150	.187	.225	.262	.30	.34	.37	.45	.56	.67	.75	.94	1.12
35	043	.087	.131	.175	.219	.262	306	.35	.39	.44	.52	.66	.79	.87	1.08	1.31
40	050	.100	.150	.200	.250	.300	.350	.40	.45	.50	.60	.75	.90	1.00	1.25	1.50
45	.056	.112	.168	.225	.281	.337	.394	.45	.51	.56	.67	.84	1.01	1.12	1.41	1.69
50	.062	125	.187	.250	.312	.375	.437	.50	.56	.62	.75	.94	1.12	1.25	1.56	1.87
60	.075	.150	.225	.300	.375	.450	.525	.60	.67	.75	.90	1.12	1.35	1.50	1.87	2.25
75	093	.187	.281	.375	.469	.562	.656	.75	.84	.94	1.12	1.40	1.69	1.87	2.34	2.81
90	.112	.225	.337	.450	.562	.675	.787	.90	1.01	1.12	1.35	1.68	2.02	2.25	2.81	3.37
100	.125	.250	.375	.500	.625	.750	.875	1.00	1.12	1.25	1.50	1.87	2.25	2.50	3.12	3.75
125	.156	312	.469	.625	.781	.937	1.094	1.25	1.41	1.56	1.87	2.34	2.81	3.12	3.91	4.69
150	.187	.375	.562	.750	.937	1.125	1.312	1.50	1.69	1.87	2.25	2.81	3.37	3.75	4.69	5.62
175	.219	.437	.656	.875	1.093	1.312	1.531	1.75	1.97	2.19	2.62	3.28	3.94	4.37	5.47	6.56
200	.250	.500	.750	1.000	1.250	1.500	1.750	2.00	2.25	2.50	3.00	3.75	4.50	5.00	6.25	7.50
250	.312	.625	.937	1.250	1.562	1.875	2.187	2.50	2.81	3.12	3.75	4.69	5.62	6.25	7 81	9.37
300	.375	.750	1.125	1.500	1.875	2.250	2.625	3.00	3.37	3.75	4.50	5.62	6.75	7.50	9.37	11.25
350	.437	.875	1.312	1.750	2.187	2.625	3.062	3 50	3.94	4.37	5.25	6.56	7.87	8.75	10.94	13.12
400	500	1 000	1.500	2.000	2.500	3.000	3.500	4 00	4.50	5.00	6.00	7 50	9.00	10.00	12.50	15.00
500	625	1.250	1.875	2.500	3.125	3.750	4.375	5.00	5.62	6.25	7.50	9.37	11.25	12.50	15.62	18.75

The *actual horse-power* for 100 ft. lift is 1.7 times the theoretical horse-power, for a 200 ft. lift 1.45 times, and for a 300 ft. lift 1.25 times.

It is estimated that it requires approximately *one horse-power, including friction, to raise sixty gallons of water per minute thirty-three feet high.*

Capacity of Pump.

To find the capacity of a cylinder in gallons, multiply the area in inches by the length of stroke in inches; divide this amount by 231 (which is the cubical contents of a gallon of water), and the quotient is the capacity in gallons.

A U. S. gallon of water weighs 8½ lbs. and contains 231 cubic inches. A cub. ft. of water weighs 62.4 lbs. and contains 1,728 cb. inches, or 7.48 gallons.

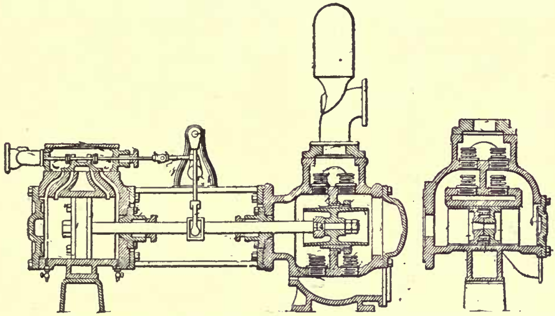
To find quantity of water elevated in one minute running at 100 feet of piston speed per minute, square the diameter of water cylinder in inches and multiply by 4. *Example*: Capacity of a five-inch cylinder is desired. The square of the diameter (5 inches) is 25, and multiplied by 4 gives 100, which is gallons per minute (approximately).

To find the diameter of a pump cylinder to move a given quantity of water per minute (100 feet of piston travel being the speed), divide the number of gallons by 4, then extract the square root, which will be the required diameter in inches.

TABLE OF EFFICIENCY OF PUMPING MACHINES.

Description.	Duty in Million Foot Pounds per 110 lbs. Coal.	Per Centage of Thermal Value of Steam Used.	Equivalent in Coal per Hourly Horse-power.
Pumping Engines.......	30 to 110	3.89 to 13.25	6.68 to 1.95
Steam pumps, large size,	15 to 30	1.94 " 3.89	13.4 " 6.68
Steam pumps, small size,	8 to 15	1.04 " 1.94	25.00 " 13.40
Vacuum pumps..........	3 to 10	0.39 " 1.30	66.6 " 25.00
Injectors, lifting water only.	2 to 5	0.26 " 0.65	100 " 66.60

TANK OR LIGHT-SERVICE DUPLEX PUMP (WORKING PRESSURE OF 75 LBS.)



PLUNGER AND RING PATTERN PISTON PATTERN WATER EN

FIG. 63. DUPLEX PUMP.

Sizes.			Gallons per Stroke.	Capacity per min. at Given Speed.		Length in inches.	Width in inches.	Sizes of Pipes.			
Steam-cylinder.	Water-cylinder.	Length of Stroke.		Strokes.	Gallons.			Steam.	Exhaust.	Suction.	Discharge.
4	4	5	.27	130	35	33	9½	½	¾	2	1½
5	4	7	.38	125	48	45½	15	¾	1	3	2½
5½	5½	7	.72	125	90	45½	15	¾	1	3	2½
7½	7½	10	1.91	110	210	58	17	1	1½	5	4
8	6	12	1.46	100	146	67	20½	1	1½	4	4
6	7	12	2.00	100	200	66	17	¾	1	4	4
8	7	12	2.00	100	200	67	20½	1	1½	5	4
8	8	12	2.61	100	261	68	30	1	1½	5	5
10	8	12	2.61	100	261	68½	30	1½	2	5	5
8	10	12	4.08	100	408	68	20½	1	1½	8	8
10	10	12	4.08	100	408	68½	30	1½	2	8	8
12	10	12	4.08	100	408	64	24	2	2½	8	8

SINGLE ACTING TRIPLEX PUMP.

Diameter of Pump Plungers.	Stroke of Pump Plungers.	Capacity, gallons, one revolution of Crank Shaft.	Gallons, per min., 40 rev. of Crank Shaft.	Size Suction Pipe, inches.	Size Delivery Pipe, inches.	Ratio of Gearing.
4	6	1.00	40	2½	2	7½ to 1
5	6	1.50	60	3	2½	7 to 1
5	8	2.00	80	3	2½	7 to 1
6	8	2.93	117	4	3	7 to 1
7	8	4.00	160	5	4	7 to 1
8	8	5.20	208	6	5	7 to 1
8	10	6.50	260	6	5	7 to 1

CENTRIFUGAL PUMPS (FOR LIFTS FROM 15 to 35 FT.)

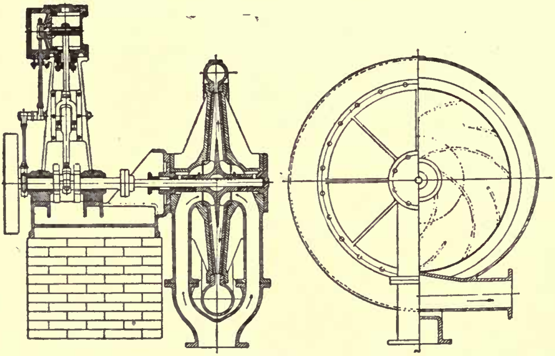


FIG 64—CENTRIFUGAL PUMP.

No. of Pump.	Suction-pipe, in.	Discharge-pipe, in.	Economical Capacity, gals. per min.	H.P. for each foot of lift.	Weight, lbs.	No. of Pump.	Suction-pipe, in.	Discharge-pipe, in.	Economical Capacity, gals. per min.	H.P. for each foot of lift.	Weight, lbs.
1	1½	1	25	.028	65	10	10	10	3000	1.60	3000
1½	2	1½	70	.05	230	12	12	12	4200	2.15	6800
2	2½	2	100	.08	265	15	15	15	7000	3.50	8840
3	3½	3	250	.15	500	18	18	18	10000	5.00	10000
4	4½	4	450	.27	680	24	24	24	18000	7.60	9000*
5	6	5	700	.36	1032	30	30	30	25000	10.50	20000*
6	6	6	1200	.65	1260	36	36	36	35000	14.75	22000*
8	8	8	2000	1.10	2460						

Directions for Connecting and Running Pumps.

The suction pipe of a pump should be perfectly air-tight. A leak in the suction pipe will destroy the vacuum, and prevent the water rising in the pipe.

The suction and discharging pipes should be run with as few bends and elbows as possible, to avoid water-hammer and undue friction. The diameters should never be less than called for by the openings on the pumps.

When drawing or forcing water long distances or at high speeds, the diameters of the pipes should be greater than called for by the openings on pumps, and should be large enough to convey the fluids with the minimum of friction. This is particularly essential for the suction pipe, which has only the atmospheric pressure to force the water from the source of supply to the pumps.

A strainer should be attached to suction pipe to prevent the entrance of foreign substances, and the *total area of the strainer holes* should be *from two to five times the area of the pipe.*

A *large vacuum chamber* on suction pipe near the pump is advantageous, and when high speeds are desired without noise, becomes a necessity.

Hot water cannot be lifted by suction any desirable height, and the difficulty increases with the temperature. To handle hot water efficiently it should gravitate to the pump.

During cold weather, if in an exposed situation, the pump and pipes should be thoroughly drained after stopping, to insure safety against frost.

The steam and exhaust pipes should be connected so that they may be drained of their water of condensation. When a steam pump is not to be used for some time, the steam cylinder and valve gear should be well oiled before stopping.

The stuffing-boxes should be kept clean and carefully packed, to avoid excessive friction by being screwed down too tight.

Short Rule for Piping a Pump.—To find the size of steam pipe, divide the cross-sectional area of steam piston by 64. To find the size of exhaust pipe divide the cross-sectional area of steam piston by 32. To find the size of the discharge pipe divide the cross-sectional area of plunger by 3. To find the size of suction pipe, divide the cross-sectional area of plunger by 2. Give the water valves the same area of opening as the suction pipe.

Duty Trials of Pumping Engines.

(Abridged from Trans. A. S. M. E., XII, 530.)

The new unit chosen, foot pounds of work per million heat units furnished by the boiler is the equivalent of 100 lbs. of coal in cases where each pound of coal imparts 10,000 heat units to the water in the boiler, or where the evaporation is 10,000 : 965.7 = 10,355 lbs. of water from and at 212° per pound of fuel.

The work done is determined by plunger displacement, after making a test for leakage, instead of by measurement of flow by weirs, which, however, may help to obtain additional data.

The necessary data having been obtained, the duty of an engine may be computed by the use of the following formulae:

$$1.\ \text{Duty} = \frac{\text{Foot-pounds of work done}}{\text{Total number of heat-units consumed}} \times 1{,}000{,}000,$$

$$= \frac{A(P \pm p + s) \times L \times N}{H} \times 1{,}000{,}000 \text{ (foot-pounds).}$$

$$2.\ \text{Percentage of leakage} = \frac{C \times 144}{A \times L \times N} \times 100 \text{ (per cent).}$$

3. Capacity = number of gallons of water discharged in 24 hours

$$= \frac{A \times L \times N \times 7.4805 \times 24}{D \times 144} = \frac{A \times L \times N \times 1.24675}{D} \text{ (gallons).}$$

4. Percentage of total frictions,

$$= \left[\frac{\text{I.H.P.} - \dfrac{A(P \pm p + s) \times L \times N}{D \times 60 \times 33{,}000}}{\text{I.H.P.}}\right] \times 100$$

$$= \left[1 - \frac{A(P \pm p + s) \times L \times N}{A_s \times \text{M.E.P.} \times L_s \times N_s}\right] \times 100 \text{ (per cent);}$$

or, in the usual case, where the length of the stroke and number of strokes of the plunger are the same as that of the steam-piston, this last formula becomes:

$$\text{Percentage of total frictions} = \left[1 - \frac{A(P \pm p + s)}{A_s \times \text{M.E.P.}}\right] \times 100 \text{ (per cent).}$$

In these formulae the letters refer to the following quantities:

A = Area, in square inches, of pump plunger or piston, corrected for area of piston rod or rods.

P = Pressure, in pounds per square inch, indicated by the gauge on the force main.

p = Pressure, in pounds per square inch, corresponding to indication of the vacuum gauge on suction main (or pressurer gauge, if the suction pipe is under a head). The indication of the vacuum gauge, in inches of mercury, may be converted into pounds by dividing it by 2.035.

s = Pressure, in pounds per square inch, corresponding to distance between the centres of the two gauges. The computation for this pressure is made by multiplying the distance, expressed in feet, by the weight of one cubic foot of water at the temperature of the pump well, and dividing the product by 144.

L = Average length of stroke of pump plunger, in feet.

N = Total number of single strokes of pump plunger made during the trial.

As = Area of steam cylinder, in square inches, corrected for area of piston rod. The quantity $As \times M.E.P.$, in an engine having more than one cylinder, is the sum of the various quantities relating to the respective cylinders.

Ls = Average length of stroke of steam piston, in feet.

Ns = Total number of single strokes of steam piston during trial.

$M.E.P.$ = Average mean effective pressure, in pounds per square inch, measured from the indicator diagrams taken from the steam cylinder.

$I.H.P.$ = Indicated horse power developed by the steam cylinder.

C = Total number of cubic feet of water which leaked by the pump plunger during the trial, estimated from the results of the leakage test.

D = Duration of trial in hours.

H = Total number of heat units (B. T. U.) consumed by engine = weight of water supplied to boiler by main feed-pump × total heat of steam of boiler pressure reckoned from temperature of main feed water + weight of water supplied by jacket pump × total heat of steam of boiler pressure reckoned from temperature of jacket water + weight of any other water supplied × total heat of steam reckoned from its temperature of supply. The total heat of the steam is corrected for the moisture or superheat which the steam may contain. No allowance is made for water added to the feed water, which is derived from any source, except the engine or some accessory of the engine. Heat added to the water by the use of a flue heater at the boiler is not to be deducted. Should heat be abstracted from the flue by means of a steam reheater connected with the intermediate receiver of the engine, this heat must be included in the total quantity supplied by the boiler.

Leakage Test of Pump.

The leakage of an inside plunger (the only type which requires testing) is most satisfactorily determined by making the test with the cylinder head removed. A wide board or plank may be temporarily bolted to the lower part of the end of the cylinder, so as to hold back the water in the manner of a dam, and an opening made in the temporary head thus provided for the reception of an overflow pipe. The plunger is blocked at some intermediate point in the stroke (or, if this position is not practicable, at the end of the stroke), and the water from the force main is admitted at full pressure behind it. The leakage escapes through the overflow pipe, and it is collected in barrels and measured. The test should be made, if possible, with the plunger in various positions.

In the case of a pump so planned that it is difficult to remove the cylinder head, it may be desirable to take the leakage from one of the openings which are provided for the inspection of the suction valves, the head being allowed to remain in place.

It is assumed that there is a practical absence of valve leakage. Examination for such leakage should be made, and if it occurs, and it is found to be due to disordered valves, it should be remedied before making the plunger test. Leakage of the discharge valves will be shown by water passing down into the empty cylinder at either end when they are under pressure. Leakage of the suction valves will be shown by the disappearance of water which covers them.

If valve leakage is found which cannot be remedied the quantity of water thus lost should also be tested. One method is to measure the amount of water required to maintain a certain pressure in the pump cylinder when this is introduced through a pipe temporarily erected, no water being allowed to enter through the discharge valves of the pump.

Table of Data and Results.

In order that uniformity may be secured, it is suggested that the data and results, worked out in accordance with the standard method, be tabulated in the manner indicated in the following scheme:

DUTY TRIAL OF ENGINE.

DIMENSIONS.

1. Number of steam cylinders........................
2. Diameter of steam cylinders........................ ins.
3. Diameter of piston rods of steam cylinders.......... ins.
4. Nominal stroke of steam pistons.................. ft.
5. Number of water plungers........................
6. Diameter of plungers............................ ins.
7. Diameter of piston rods of water cylinders.......... ins.
8. Nominal stroke of plungers........................ ft.
9. Net area of steam pistons........................ sq. ins.
10. Net area of plungers............................ sq. ins.
11. Average length of stroke of steam pistons during trial ft.
12. Average length of stroke of plungers during trial.... ft.

(Give also complete description of plant.)

TEMPERATURES.

13. Temperature of water in pump well................ degs.
14. Temperature of water supplied to boiler by main feed pump .... degs.
15. Temperature of water supplied to boiler from various other sources................................ degs.

FEED WATER.

16. Weight of water supplied to boiler by main feed pump lbs.
17. Weight of water supplied to boiler from various other sources .... lbs.
18. Total weight of feed water supplied from all sources.. lbs.

PRESSURES.

19. Boiler pressure indicated by gauge................. lbs.
20. Pressure indicated by gauge on force main.......... lbs.
21. Vacuum indicated by gauge on suction main........ ins.
22. Pressure corresponding to vacuum given in preceding line .... lbs.
23. Vertical distance between the centres of the two gauges .... ins.
24. Pressure equivalent to distance between the two gauges. lbs.

MISCELLANEOUS DATA.

25. Duration of trial................................ hrs.
26. Total number of single strokes during trial......
27. Percentage of moisture in steam supplied to engine, or number of degrees of superheating.............. % or deg.
28. Total leakage of pump during trial, determined from results of leakage test.......................... lbs.
29. Mean effective pressure, measured from diagrams taken from steam cylinders.................... M.E.P.

PRINCIPAL RESULTS.

30. Duty .... ft. lbs.
31. Percentage of leakage .... %
32. Capacity .... gals.
33. Percentage of total friction .... %

ADDITIONAL RESULTS.

34. Number of double strokes of steam piston per minute.
35. Indicated horse power developed by the various steam cylinders .... I.H.P.
36. Feed water consumed by the plant per hour........ lbs.
37. Feed water consumed by the plant per indicated horse-power per hour, corrected for moisture in steam.... lbs.
38. Number of heat units consumed per indicated horse-power per hour .... B.T.U.
39. Number of heat units consumed per indicated horse power per minute .... B.T.U.
40. Steam accounted for by indicator at cut-off and release in the various steam cylinders.................. lbs.
41. Proportion which steam accounted for by indicator bears to the feed water consumption............
42. Number of double strokes of pump per minute....
43. Mean effective pressure, measured from pump diagrams .... M.E.P.
44. Indicated horse power exerted in pump cylinders.... I.H.P.
45. Work done (or duty) per 100 lbs. of coal............ ft. lbs.

SAMPLE DIAGRAM TAKEN FROM STEAM CYLINDERS.

(Also, if possible, full measurement of the diagrams, embracing pressures at the initial point, cut-off, release, and compression; also back pressure, and the proportions of the stroke completed at the various points noted.)

SAMPLE DIAGRAM TAKEN FROM PUMP CYLINDERS.

These are not necessary to the main object, but it is desirable to give them.

NOTES ON PUMPS:

Miscellaneous

Belt Transmission.

HORSE POWER OF SHAFTING.

Diameter of Shaft in Inches.	Revolutions per Minute.				
	100	125	150	175	200
	h. p.	h. p.	h. p.	h. p.	h. p.
15-16	1.2	1.4	1.7	2.1	2.4
1 3-16	2.4	3.1	3.7	4.3	4.9
1 7-16	4.3	5.3	6.4	7.4	8.5
1 11-16	6.7	8.4	10.1	11.7	13.4
1 15-16	10.0	12.5	15.0	17.5	20.0
2 3-16	14.3	17.8	21.4	24.9	28.5
2 7-16	19.5	24.4	29.3	34.1	39.0
2 11-16	26.0	32.5	39.0	43.5	52.0
2 15-16	33.8	42.2	50.6	59.1	67.5
3 3-16	43.0	53.6	64.4	75.1	85.8
3 7-16	53.6	67.0	79.4	93.8	107.2
3 11-16	65.9	82.4	97.9	115.4	121.8
3 15-16	80.0	100.0	120.0	140.0	160.0
4 7-16	113.9	142.4	170.8	199.8	227.8
4 15-16	156.3	195.3	234.4	273.4	312.5

HORSE POWER OF BELTING.

TABLE FOR SINGLE LEATHER, 4-PLY RUBBER AND 4-PLY COTTON BELTING, BELTS NOT OVERLOADED. (ONE INCH WIDE, 800 FEET PER MINUTE = 1-HORSE POWER.)

Speed in Ft. Per Minute.	Width of Belts in Inches.											
	2	3	4	5	6	8	10	12	14	16	18	20
	h.p	h.p	h.p	h.p	h.p	h.p	h.p	h.p	h.p	h.p	h. p.	h. p.
400	1	1½	2	2½	3	4	5	6	7	8	9	10
600	1½	2¼	3	3¾	4½	6	7½	9	10½	12	13½	15
800	2½	3	4	5	6	8	10	12	14	16	18	20
1,000	2	3¾	5	6¼	7½	10	12½	15	17½	20	22½	25
1,200	3	4½	6	7½	9	12	15	18	21	24	27	30
1,500	3¾	5¾	7½	9½	11½	15	18¾	22½	26½	30	33¾	37½
1,800	4½	6¾	9	11¼	13½	18	22½	27	31½	36	40½	45
2,000	5	7½	10	12½	15	20	25	30	35	40	45	50
2,400	6	9	12	15	18	24	30	36	42	48	54	60
2,800	7	10½	14	17½	21	28	35	42	49	56	63	70
3,000	7½	11¼	15	18¾	22½	30	37½	45	52½	60	67½	75
3,500	8¾	13	17½	22	26	35	44	52½	61	70	79	88
4,000	10	15	20	25	30	40	50	60	70	80	90	100
4,500	11¼	17	22½	28	34	45	57	69	78	90	102	114
5,000	12½	19	25	31	37½	50	62½	75	87½	100	112	125

Double leather, 6-ply rubber or 6-ply cotton belting will transmit 50 to 75 per cent. more power than is shown in this table.

A simple rule for ascertaining transmitting power of belting, without first computing speed per minute that it travels, is as follows: Multiply diameter of pulley in inches by its number of revolutions per minute, and this product by width of the belt in inches; divide this product by 3,300 for single belting, or by 2,100 for double belting, and the quotient will be the amount of horse power that can be safely transmitted.

Equivalent Values.

Equivalent Values of Electrical and Mechanical Units.

Unit.	Equivalent Value in Other Units.
1 K. W. Hour =	1,000 watt hours. 1.34 horse-power hours. 2,654,200 ft.-lbs. 3,600,000 joules. 3,412 heat-units. 367,000 kilogram metres. .235 lb. carbon oxidized with perfect efficiency. 3.53 lbs. water evap. from and at 212° F. 22.75 lbs. of water raised from 62° to 212° F.
1 H.P. Hour =	.746 K. W. hours. 1,980,000 ft.-lbs. 2,545 heat-units. 273,740 k. g. m. .175 lb. carbon oxidized with perfect efficiency. 2.64 lbs. water evaporated from and at 212° F. 17.0 lbs. water raised from 62° F. to 212° F.
1 Kilo-watt =	1,000 watts. 1.34 horse-power. 2,654,200 ft.-lbs. per hour. 44,240 ft.-lbs. per minute. 737.3 ft.-lbs. per second. 3,412 heat-units per hour. 56.9 heat-units per minute. .948 heat-unit per second. .2275 lb. carbon oxidized per hour. 3.53 lbs. water evap. per hour from and at 212° F.

Unit.	Equivalent Value in Other Units.
1 H.P. =	746 watts. .746 K. W. 33,000 ft.-lbs. per minute. 550 ft.-lbs. per second. 2,545 heat-units per hour. 42.4 heat-units per minute. .707 heat-units per second. .175 lbs. carbon oxidized per hour. 2.64 lbs. water evap. per hour from and at 212° F.
1 Joule =	1 watt second. .000000278 K. W. hour. .102 k. g. m. .0009477 heat-units. .7373 ft.-lb.
1 Ft.-lb =	1.356 joules. .1383 k. g. m. .000000377 K. W. hours. .001285 heat-units. .0000005 H.P. hour.
1 Watt =	1 joule per second. .00134 H.P. 3.412 heat-units per hour. .7373 ft.-lbs. per second. .0035 lbs. water evap. per hr. 44.24 ft.-lbs. per minute.
1 Watt per sq. in. =	8.19 heat-units per sq. ft. per minute. 6371 ft.-lbs. per sq. ft. per minute. .193 H.P. per sq. ft.

Unit.	Equivalent Value in Other Units.
1 Heat-unit =	1,055 watt seconds. 778 ft.-lbs. 107.6 kilogram metres. .000293 K. W. hour. .000393 H.P. hour. .0000688 lbs. carbon oxidized. .001036 lbs. water evap. from and at 212° F.
1 Heat-unit per Sq. Ft. per min. =	.122 watts per square in. .0176 K. W. per sq. ft. .0236 H.P. per sq. ft.
1 Kilogram Metre =	7.233 ft.-lbs. .00000365 H.P. hour. .00000272 K. W. hour. .0093 heat-units.
1 lb. Carbon Oxidized with perfect Efficiency =	14,544 heat-units. 1.11 lb. Anth'cite coal ox. 2.5 lbs. dry wood oxidized. 21 cu. ft. illuminating-gas. 4.26 K. W. hours. 5.71 H.P. hours. 11,315,000 ft.-lbs. 15 lbs. of water evap. from and at 212° F.
1 lb. Water Evapor'ed from and at 212° F. =	.283 K. W. hour. .379 H.P. hour. 965.7 heat-units. 103,900 k. g. m. 1,019,000 joules. 751,300 ft.-lbs. .0664 lb. of carbon oxidized.

Cooling Towers.

Cooling towers possess operative advantages of considerable importance. There is, of course, a certain loss of water by evaporation, but this rarely exceeds 10 per cent. of the water coooled, while under favorable conditions of the air it does not exceed 5 *per cent.*

It is advisable to have *separate towers for steam condenser and ammonia condenser,* as the results are better in each case. The efficiency of the cooling tower is lowered very fast, when the water for the ammonia condenser is much above 80°, whereas for steam condenser, if the water be reduced to 100° the tower will be fairly efficient.

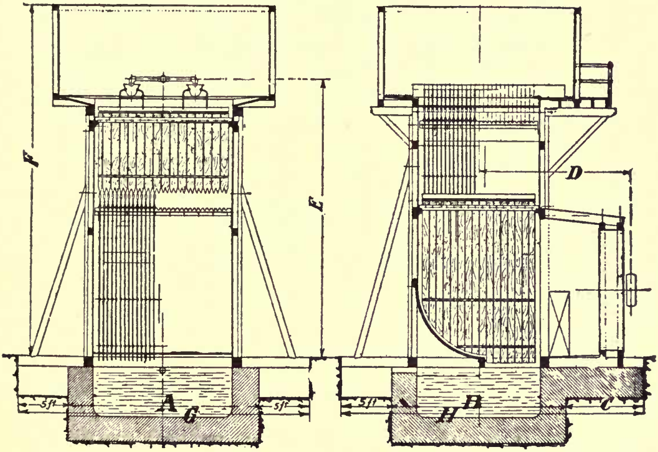


FIG 55--FORCED DRAFT COOLING TOWER.

The following data show the results in cooling obtained by the use of cooling towers:

For *ammonia condensers,* with the air at 95° F. and 37 per cent. humidity:

Initial temperature of water entering cooling tower....... 100° F.
Final temperature of water leaving cooling tower......... 71° F.

Result in cooling .... 29° F.

For *steam condensers,* with the air at 95° F. and 44 per cent. humidity:

Initial temperature of water entering cooling tower....... 160° F.
Final temperature of water leaving cooling tower......... 81° F.

Result in cooling .... 79° F.

As the forced draft tower seems to have met with general favor, we append a few tables, stating general dimensions and capacity.

Size and Weight of Cooling Towers.

No. of Tower.	MAIN DIMENSIONS. A	B	C	D	E	F	G	H	Weight in lbs.
I	8'11½"	8' 6½"	6 ft.	9' 1"	24' 9"	32'	18'11½"	19' 6½"	25,000
II	9' 9½"	8'11½"	6 ft.	9' 3"	24' 9"	32'	19' 9½"	19'11½"	28,500
III	10' 2¾"	9' 9½"	6 ft.	9'10"	24' 9"	32'	20' 2¾"	20' 9½"	32,000
IV	11' 5½"	10' 7½"	7 ft.	10' 4"	24' 9"	32'	21' 5½"	22' 7½"	39,000
V	13' 3½"	12' 5½"	7 ft.	11' 4"	24, 9"	32'	23' 3½"	24' 5½"	46,000
VI	14' 6¾"	13' 3½"	7 ft.	12' 6"	25' 8"	32' 9"	24' 6¾"	25' 3½"	53,000
VII	16' 4¾"	15' 1½"	7 ft.	13' 4"	25' 8"	32' 9"	26' 4¾"	27' 1½"	59,000
VIII	17' 7½"	16' 4½"	8 ft.	14' 9"	27' 4"	34' 7"	27' 7½"	29' 4½"	65,700
IX	18'10¼"	17' 2½"	8 ft.	15' 3"	27' 4"	34' 7"	28'10¼"	30' 2½"	71,700

Cooling Capacity of Cooling Towers and Size of Fans.

No. of Tower	Cooling Capacity in Gallons in 24 hours for: Ammonia CONDENSERS	Steam CONDENSERS	H.P. of comp. cond. engine supplied with condens. water.	Size & Number of Fans	Size of Pulley.	Revol. of Pulley per minute.	H.P. for Fans.
I	50,000	100,000	50	1— 6ft.	15"x 8"	100—125	1 — 1½
II	75,000	150,000	75	1— 6ft.	15"x 8"	150—170	1½— 2
III	100,000	200,000	100	1— 7ft.	18"x 9"	140—150	2 — 2½
IV	150,000	300,000	150	1— 8ft.	24"x 9	140—150	3½— 4
V	200,000	400,000	200	1— 9ft.	28"x10"	130—140	5 — 6
VI	250,000	500,000	250	1—10ft.	30"x11"	130—140	7 — 9
VII	300,000	600,000	300	1—10ft.	30"x11"	145—150	10 —12
VIII	400,000	800,000	400	1—12ft.	36"x12"	110—120	13 —15
IX	500,000	1,000,000	500	1—12ft.	36"x12"	140—150	16 —20

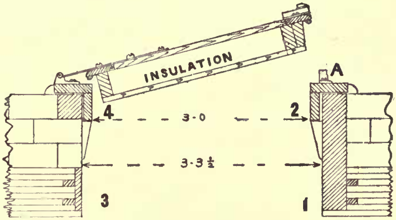
MISCELLANEOUS NOTES:

Doors

Doors are a weak point in all storage rooms. Their insulation is important, but their tightness and quick operation is vastly more so. A leak is an endless expense. Slow moving doors are hardly less so. Doors that bind and work badly are shut only when the workman can find no excuse for leaving them open, which is seldom, if ever.

The following sketches show a construction which is patented, and which is especially contrived to avoid these troubles.

The door makes an overlapping contact, with a soft hemp gasket in the joint, and is held to its seat against the front of the door frame by powerful elastic hardware. The thick portion of the door fits loosely, so that considerable change of size, form and position, due to wear, swelling, etc., does not make it leak or bind.



Where all old style doors, when they work badly or leak, must be eased, thus forever destroying their fit, a slight readjustment of the door frame of these doors restores them to their original perfection of fit and freedom in a minute at no expense.

As these doors do not stand in the doorway when open, it can be six inches less in width than old style doorways—an important economy in refrigeration.

As constructed in this year, 1908, the opening in wall to receive these door frames should be 3½ inches wider, and 4 inches higher, than the size of the doorway in the clear. Follow construction numbered 1 and 2. For overhead track doors this rough opening should extend 13 inches above the lower edge of track. Door frames are secured with lag screws, ⅜x4 inches, through front casing, inserted at A.

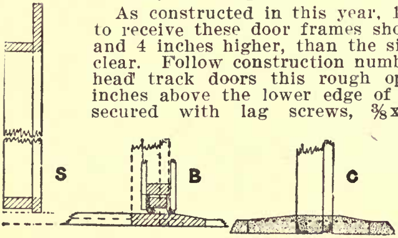


Figure B shows wooden beveled threshold, 1¾ inches thick, which connects lower ends of door frame and forms a part of it, let down into floor. No feather edge, no jolt, no splinters. For warehouses. Accommodates trucks.

Figure C, cement floor, shows lower ends of door frame extending down into the door a distance of three inches, and connected by angle irons extending across doorway from one side to the other below the surface.

Figure S shows door frame with full standard sill and head used on all sizes of door frames. Suited only to walking through.

Special doors on a modified plan for intermittent or continuous freezers, as well as for general purposes, perfectly tight and perfectly free, regardless of temperature, moisture or accumulation of ice in any degree.

Metal covered fireproof doors.

Combined self-closing ice door and chute of three styles.

Ice counters.

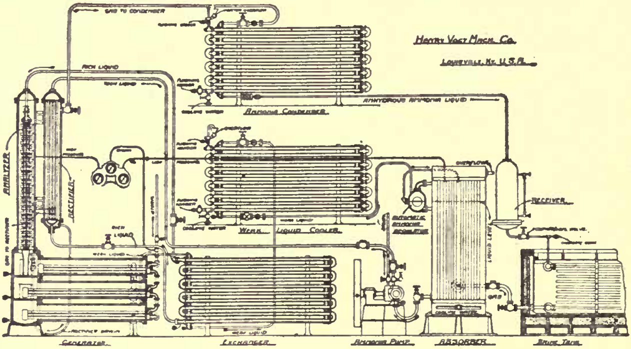
Patents on every valuable feature of this work are granted to or applied for by the STEVENSON CO., CHESTER, PA.

Absorption Machines.

Since going to press the author's attention has been called to the latest design of the Vogt Absorption Machine, which differs in some respects from the one given on page 23. In order to bring the book up to date the following brief description is here appended:

The strong liquor is drawn from the absorber and pumped into the upper end of the rectifier and passes down through the small pipes and out from the bottom of rectifier to the bottom pipes of the exchanger, where it passes upward through the inner pipes and out from the top of exchanger to top of analyzer, where the liquid falls in a spray from one pan to another until it reaches the top compartment of the generator.

The gas generated passes upward in the analyzer and is cooled and deprived of a portion of its moisture by coming in contact with the liquid trickling down from pan to pan in the analyzer. The gas passes on and enters the rectifier at bottom and completely surrounds the tubes through which the rich aqua is flowing, and as



VOGT ABSORPTION MACHINE.

the rich aqua is comparatively cool as against the gas, the moisture in the gas will condense and deposit itself on the tubes as the gas is forced upward, allowing the gas to pass over dry to the condenser. The moisture withdrawn and adhering to the tubes will drain out at the bottom of the rectifier and back into the top compartment of the generator.

The gas from the rectifier is admitted to the top of the condensing coils, where it quickly liquefies and is conducted from the bottom of the condenser to the liquid ammonia receiver.

The weak liquid having in the meantime passed from bottom of generator to top of exchanger, and down through the outer pipes of same, is conducted to the weak liquid cooler to be further reduced in temperature, and is finally conducted to the absorber, where the gas from the refrigerating coils is rapidly absorbed, and the double cycle of circulation is thus completed.

NOTES.

NOTES.

NOTES.

NOTES.

NOTES.

NOTES.

NOTES.

NOTES.

NOTES.

NOTES.

NOTES.

NOTES.

NOTES.

General Offices
CARBONDALE, PA.

New York Boston Baltimore Chicago Pittsburgh