

THE DESIGN, CONSTRUCTION, AND TEST
" OF A
FIVE TON STEAM JET REFRIGERATION UNIT

BY

A. W. HOWARD

and

C. W. VOGELIANG

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Approved:

In Charge of Major Work

Head of Department

Dean of Engineering

Chairman of Graduate Committee

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INTRODUCTION

In 1947 a paper was presented by Mr. Hugh S. Miles (17) on the design of a steam-jet refrigeration unit. This paper comprised the collection and correlation of available data from which the design of a sixty-five ton unit was evolved. Based on Miles' recommendation that a unit be built for testing the validity of the design, it was decided to construct and test a unit in the Mechanical Engineering Laboratory.

Although commercial companies have spent considerable time and money on the development of steam-jet refrigeration their results are not published. By the construction of this unit much experimental work could be done so that many conflicting points in available literature may be clarified.

It is the purpose of the authors to present the method of construction of a five ton steam-jet refrigeration unit and the test data that could be obtained in the time available.

REVIEW OF LITERATURE

Recent advances in Air Conditioning have opened a market for the practical application of steam-jet refrigeration. Although steam-jet refrigeration units have been on the market for a number of years the design of these units has not been available. From available literature Miles (17) evolved a procedure for the design of a steam-jet refrigeration unit. After a description of the apparatus Miles made an energy analysis to determine the conditions of the steam and vapor at various points as they passed through the unit. As the actual processes through the nozzle, mixing chamber, and diffuser did not follow the ideal or isentropic paths, it was necessary to introduce certain efficiency factors to account for these losses. For these efficiency factors, Miles used recommended values which he discussed along with other factors pertinent to the design of the unit. After the determination of the steam vapor ratio, which he found to be 1.53 pounds of steam per pound of vapor, the actual conditions at the throat and exit of the nozzle, the exit from the evaporator, and the inlet, throat and exit of the diffuser were found. With the conditions thus found calculations were made for determining the size of the evaporator, nozzle, and diffuser. Miles then recommended the construction of a small unit for testing the soundness of his design.

The literature examined on diffuser and entrainment efficiencies gave varying recommendations as to the efficiency to use. The following table is a summary of the values given:

Diffuser Efficiency	Entrainment Efficiency	Recommended By
60%	---	ASH&VE Guide, 1938 (2)
70%	65%	J. C. Bertch (4)
75%	65%	H. R. Sparks (23)
75%	---	H. G. Klrod, Jr. (8)
78%	---	F. R. B. Watson (25)
75-80%	---	G. Flugel (9)

Many types of spray arrangements and tanks were presented by the various authors. Bertch (4) used a perforated flat plate, a shower head arrangement and a perforated pipe in both vertical and horizontal cylindrical evaporator tanks. Ingersoll Rand Co. (13) used a weir in a rectangular evaporator with the length of the tank depending upon the number of ejectors on the unit. This method exposed a larger surface of the water but did not give a fine division of the water particles. Vertical tanks gave a longer fall for the water, thus permitting the water particles to separate from the vapor before it was carried off, but horizontal tanks gave a greater water surface. From this it would seem that the difference between horizontal and vertical tanks was small and the choice of the type to be used was of minor importance.

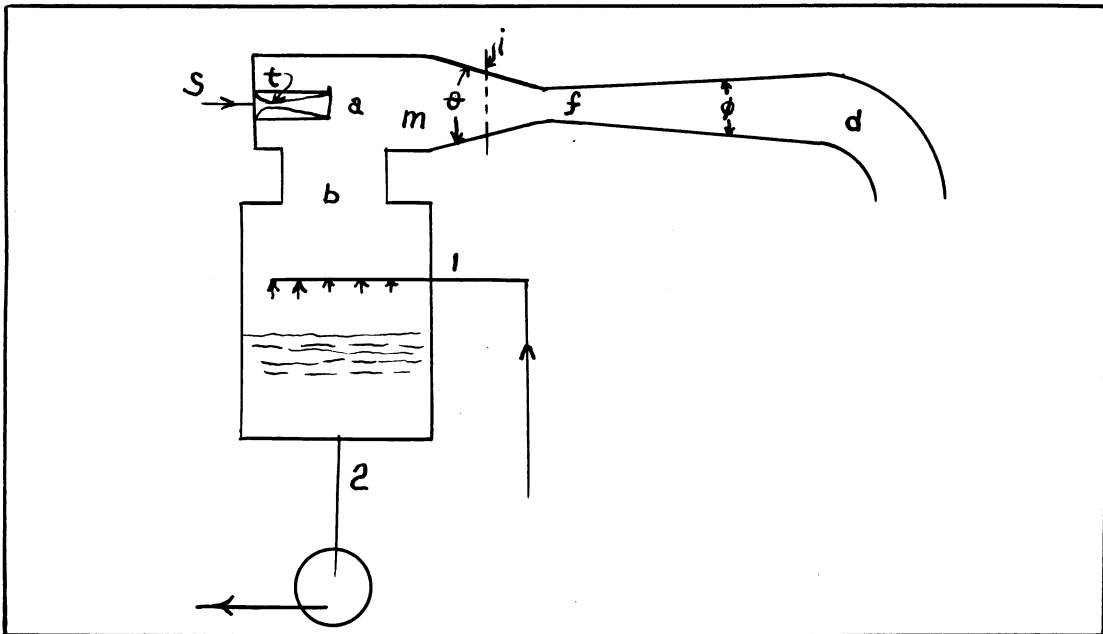
DESIGN AND SELECTION OF EQUIPMENT

A - Design Conditions

The apparatus will be assembled in the basement of the Mechanical engineering Laboratory using the Alberger condenser that is set up there as the condenser for the unit. The evaporating chamber, diffuser, nozzle and heat exchanger will be designed in accordance with the following conditions (17).

Steam Pressure Available -----	150 psig
Steam Pressure, Design -----	110 psig
Condenser Pressure -----	2 in. Hg. abs.
Evaporator Pressure -----	.361 in. Hg. abs.
Vapor Quality leaving evaporator -----	97%
Steam Quality -----	99.5%
Chilled Water Temperature -----	50° F
Return Water Temperature -----	60° F
Nozzle Efficiency (entrance to Throat) -----	95%
Nozzle efficiency, overall -----	85%
Diffuser Efficiency -----	75%
Entrainment Efficiency -----	65%
Refrigeration Effect -----	210-BTU/min/ton
Compression Ratio -----	5.5
Cone Angle of Nozzle -----	10°
Diffuser combining cone angle -----	32°
Diffuser divergent cone angle -----	6°
Steam Vapor Ratio -----	1.53 lb steam/lb vapor
Kinetic Energy (K_c) -----	127.2 BTU/lb

B - Table of Symbols



- a - nozzle exit
- A - an area, square inches
- b - connection between evaporator and diffuser
- d - diffuser discharge at entrance to condenser
- D - a diameter - inches
- f - diffuser throat
- g - acceleration due to gravity, 32.2 ft/sec/sec.
- h - an actual enthalpy, BTU/lb
- h' - an enthalpy after an isentropic process, BTU/lb
- Δh - an actual enthalpy change, BTU/lb
- $\Delta h'$ - an isentropic enthalpy change, BTU/lb
- i - diffuser inlet
- J - Joule's constant, 778 foot pounds per BTU
- K - kinetic energy, BTU/lb
- K_a - kinetic energy of the steam-jet per pound of steam supplied, BTU/lb
- K_c - kinetic energy required to compress one pound of the mixture, BTU/lb
- K_1 - kinetic energy of the mixture at the diffuser inlet per pound of steam supplied, BTU/lb
- $K_{1'}$ - kinetic energy of the mixture per pound of mixture, BTU/lb
- K_v - kinetic energy per pound of steam supplied available to compress the vapor, BTU/lb
- m - combining chamber
- M - weight of fluid, pounds
- psia - pounds per square inch absolute

psig pounds per square inch gauge
RE - refrigerating effect -- BTU/min/ton
s - entering steam
steam motive steam supplied through nozzle
t - nozzle throat
v - a volume, cu. ft/lb
V - a velocity, ft/sec
x - quality of steam
vapor steam evaporated from the water being chilled
1 - point just prior to water entering evaporator
2 - point just after water leaves evaporator
θ - combining cone angle
φ - diffuser cone angle

C - The Capacity of the Unit

$$\begin{aligned} RE &= M_b(h_b - h_1) \\ &= 1(1050 - 28) \\ &= 1022 \text{ BTU/lb of vapor} \end{aligned}$$

At a condition of 2 in. Hg. and $h_d = 1131$ BTU/lb of mixture,

$$v_d = 376 \text{ cu ft/lb.}$$

For a unit of five tons, the weight of vapor needed to be carried off

is:

$$\begin{aligned} w_v &= \frac{\text{Capacity (210)}}{60 \times RE} \\ &= \frac{5 \times 210}{60 \times 1022} \\ &= .0172 \text{ lb/sec} \end{aligned}$$

The size of the inlet into the condenser necessary to carry off this vapor at a velocity of 200 fps is:

$$\begin{aligned} A_d &= \frac{M_d \times v_d}{V_d} \\ &= \frac{.0436 \times 376}{200} \\ &= 0.082 \text{ sq. ft.} \end{aligned}$$

$$\begin{aligned} D_d &= (144 \times A_d / 0.7854)^{\frac{1}{2}} \\ &= (144 \times 0.082 / 0.7854)^{\frac{1}{2}} \\ &= 3.88 \text{ sq. in. say 4 in.} \end{aligned}$$

D - The Evaporator

The size of the evaporator outlet is determined by the volume of vapor that has to be carried off.

$$\begin{aligned}q_b &= w_v \lambda_b v_b \\&= 0.0172 \times 0.97 \times 1703.2 \\&= 28.4 \text{ cfs}\end{aligned}$$

The velocity of the vapor leaving the evaporator is assumed to be 100 fps (17).

$$\begin{aligned}A_b &= q_b / V_b \\&= 28.4 / 100 \\&= 0.284 \text{ sq. ft.} \\D_b &= \left(\frac{144 \times A_b}{0.7854} \right)^{\frac{1}{2}} \\&= \left(\frac{144 \times 0.284}{0.7854} \right)^{\frac{1}{2}} \\&= 7.18 \text{ in. say 8 in.}\end{aligned}$$

A discarded galvanized steel hot water tank was obtained that measured 14 inches in diameter, 40 inches long and had a wall thickness of 3/16 of an inch. According to the ASME Code for unfired pressure vessels with external pressure, a L/D ratio of 2.86 and a t/D ratio of 0.0125 gives an allowable pressure of 90 psi. This made the tank safe for use in this unit as the maximum pressure that the walls of the tank had to withstand was atmospheric pressure.

The flow required in the evaporator for a ten degree F differential between entering and exit circulating water is:

$$\begin{aligned}\text{Flow} &= \frac{\text{capacity} \times 200}{\text{water temp. rise} \times 62.4} \\ &= 5 \times 200/10 \times 62.4 \\ &= 1.62 \text{ cfm.}\end{aligned}$$

Since it is recommended that the fluid capacity of the evaporator be sufficient to handle the normal flow for one minute should the supply water be shut off (17), a volume of water greater than 1.62 cfm is needed in the tank. The cross sectional area of the tank is:

$$\begin{aligned}A &= 0.7854 \times D^2/144 \\ &= 0.7854 \times 196/144 \\ &= 1.07 \text{ sq. ft.}\end{aligned}$$

The necessary height of water in the tank is:

$$\begin{aligned}h &> \frac{V}{A} \\ &> 1.62/1.07 \\ &> 1.515 \text{ ft.} \\ \text{say } h &= 18\frac{1}{2} \text{ in.}\end{aligned}$$

This makes the fluid capacity slightly above that required for a normal flow of the circulating water for one minute.

E - The Nozzle

One booster ejector is used with only one nozzle in the ejector. The steam discharged by the nozzle will be:

$$\begin{aligned} W_n &= \text{vapor ratio} \times W_v \\ &= 1.53 \times 0.0172 \\ &= 0.0263 \text{ lb/ sec} \end{aligned}$$

With equilibrium conditions existing in the nozzle and initially wet steam the pressure at the throat is determined:

$$\begin{aligned} P_t &= 0.58 \times P_s \\ &= 0.58 \times 123.7 \\ &= 71.7 \text{ psia} \\ \\ h_t &= h_s - 0.95 (h_s - h_t^*) \\ &= 1186 - 0.95 (1186 - 1144) \\ &= 1146.1 \text{ BTU/lb of steam} \\ \\ V_t &= 223.7 (h_s - h_t)^{\frac{1}{2}} \\ &= 223.7 (1186 - 1146.1)^{\frac{1}{2}} \\ &= 1411 \text{ fps} \end{aligned}$$

With the pressure and enthalpy at the throat known the quality of the steam is found to be 96.1%.

$$\begin{aligned} v_t &= x_t \times v \text{ (saturated vapor)} \\ &= 0.961 \times 6.068 \\ &= 5.83 \text{ cu ft/lb of steam} \end{aligned}$$

From the above conditions the diameter of the nozzle throat is:

$$\begin{aligned}A_t &= w_n v_t / V_t \\&= 0.02635 \times 5.83 \times 144 / 1411 \\&= 0.0156 \text{ sq. in.}\end{aligned}$$

$$\begin{aligned}D_t &= (A_t / 0.7854)^{\frac{1}{2}} \\&= (0.0156 / 0.7854)^{\frac{1}{2}} \\&= 0.141 \text{ in.}\end{aligned}$$

At the exit from the nozzle the quality of the steam is 79.2% and the enthalpy is 862 BTU from which is found the specific volume $v_g = 1349$ cu ft/lb.

$$\begin{aligned}V_g &= 223.7 \times K_g^{\frac{1}{2}} \\&= 223.7 \times 324^{\frac{1}{2}} \\&= 4025 \text{ fps}\end{aligned}$$

$$\begin{aligned}A_g &= w_n v_g / V_g \\&= 0.02635 \times 1349 \times 144 / 4025 \\&= 1.27 \text{ sq. in.}\end{aligned}$$

$$\begin{aligned}D_g &= (A / 0.7854)^{\frac{1}{2}} \\&= (1.27 / 0.7854)^{\frac{1}{2}} \\&= 1.26 \text{ say } 1\frac{1}{4} \text{ inches}\end{aligned}$$

Having the throat diameter, exit diameter, and the cone angle of 10° .

the length of the nozzle is calculated.

$$L = \frac{D_a - D_t}{2 \times \tan 5^\circ}$$

$$L = \frac{1.25 - 0.141}{2 \times 0.08748}$$

$$L = 5.78 \quad \text{say } 5\text{-}3/4 \text{ inches.}$$

The inlet to the nozzle is the arc of a circle whose radius is three times the throat diameter, and the entrance end of the nozzle will be threaded with $1\frac{1}{2}$ inch American Standard Pipe threads. The nozzle is made of brass to prevent rusting.

The length of thread on the entrance end of the nozzle is computed:

$$\begin{aligned} L &= (0.8D \div 6.8)P \\ &= (0.8 \times 1.66 \div 6.8) 0.087 \\ &= 0.7076 \quad \text{say } 3/4 \text{ inch} \end{aligned}$$

where: L = length of effective thread in inches
D = pipe diameter in inches
P = pitch of thread

The threaded portion of the base is $1\frac{1}{2}$ inches long and the hexagonal head on the nozzle is centered on the throat. The total length of the nozzle is $1\frac{1}{2} \div 1\frac{1}{2} \div 2 \div 5\text{-}3/4 = 8$ inches. The rim at the exit of the nozzle is made $1/8$ of an inch thick, thus making the outside diameter of the diverging portion of the nozzle $1\frac{1}{2}$ inches. The hexagonal head is made $2\frac{1}{2}$

inches across flats and $1\frac{1}{8}$ inches wide.

The nozzle is placed in position by fastening it to a piece of $1\frac{1}{4}$ inch pipe by means of a coupling and screwing the pipe into the flange. This method is used for positioning the nozzle since by varying the length of the pipe, the nozzle can be placed in any desired position. The flange was fastened to the tee by means of eight, $3/4$ x 3 inch bolts.

F - The Diffuser

From page (7) the kinetic energy (K_c) of the mixture at the inlet to the diffuser is 127.2 BTU/lb. This kinetic energy corresponds to a velocity of:

$$\begin{aligned}V_1 &= 223.7(K_1)^{\frac{1}{2}} \\V_1 &= 223.7(127.2)^{\frac{1}{2}} \\&= 2520 \text{ fps}\end{aligned}$$

The weight of the mixture, w_m , is the weight of the steam and the vapor.

$$\begin{aligned}w_m &= w_s + w_v \\&= 0.0264 + 0.0172 \\&= 0.0436 \text{ lb/sec}\end{aligned}$$

The enthalpy of the mixture at the inlet to the diffuser, (h_1), is 1004 BTU/lb and the pressure corresponds to that in the evaporator which is 0.362 psia. The specific volume of the vapor at these conditions is,

$$v_1 = 1577 \text{ cu ft/lb.}$$

From the above conditions the area of the inlet to the diffuser can be calculated.

$$\begin{aligned} A_1 &= w_m \times v_1 / V_1 \\ &= 0.0436 \times 1577 \times 144 / 2520 \\ &= 3.93 \text{ sq in.} \end{aligned}$$

$$\begin{aligned} D_1 &= (A / 0.7854)^{\frac{1}{2}} \\ &= (3.93 / 0.7854)^{\frac{1}{2}} \\ &= 2.24 \text{ inches} \end{aligned}$$

To determine the throat diameter of the diffuser a series of approximations were made at various pressures to determine the pressure at which the area is a minimum. As the pressure at the throat is approximately $P_d/2$ or 1 inch Hg. abs., calculations are made at 1.0, 1.1, and 1.2 inches of Hg. abs. pressure. The kinetic energy at d is considered negligible. Assuming isentropic compression from the inlet to 1.1 inches Hg.,

$$\begin{aligned} h_{1.1} - h_1 &= 1063 - 1004 \\ &= 59 \text{ BTU/lb} \end{aligned}$$

As the actual compression path to 1.1 inches Hg. abs. requires more work it is necessary to use a diffuser efficiency which will account for the loss of energy.

$$\begin{aligned}h_{1.1} - h_1 &= \frac{h_{1.1}^i - h_1}{\phi_d} \\ &= 59/.75 \\ &= 78.7 \text{ BTU/lb}\end{aligned}$$

$$\begin{aligned}h_{1.1} &= 1004 + 78.7 \\ &= 1082.7 \text{ BTU/lb}\end{aligned}$$

From the general energy equation between the throat of the diffuser and the discharge into the condenser the mixture velocity at the throat is determined:

$$\begin{aligned}h_{1.1} + v_{1.1}^2/2g &= h_d + v_d^2/2g \\ 1082.7 + v_{1.1}^2/2g &= 1131 + 0 \\ v_{1.1} &= 1555 \text{ fps.}\end{aligned}$$

The specific volume of the vapor at a pressure of 1.1 inches Hg. abs. and an enthalpy of 1082.7 is: $v_{1.1} = 588 \text{ cu. ft/lb.}$

The area of the throat at these conditions is:

$$\begin{aligned}A_{1.1} &= wv_{1.1}/v_{1.1} \\ &= 0.0436 \times 588 \times 144/1555 \\ &= 2.375 \text{ sq. inches.}\end{aligned}$$

By the same method of calculation the area at 1.0 inches Hg. abs. is found to be 2.415 sq. inches, the area at 1.2 inches Hg. abs., 2.420 sq.

inches. As the smallest area occurred at 1.1 inches of Hg. abs., this area is used to determine the diameter of the throat.

$$\begin{aligned} D &= (A_{1.1}/0.7854)^{\frac{1}{2}} \\ &= (2.375/0.7854)^{\frac{1}{2}} \\ &= 1.746 \text{ inches.} \end{aligned}$$

The length of the straight section of the throat is taken as two diameters, or $L = 2 \times 1.746 = 3\frac{1}{2}$ inches long. A combining cone angle of 32° is used and the combining cone is joined to the throat by an arc with a radius of eight throat diameters, or $R = 8 \times 1.746 = 14$ inches.

The size of the evaporator outlet is 8 inches (pagell) so an eight inch cast iron tee is used for the combining chamber. This made it possible to have the outside diameter of the combining cone of the diffuser $7\frac{1}{2}$ inches with an inside diameter of $6\frac{1}{2}$ inches.

The length of the converging portion of the diffuser is determined by:

$$\begin{aligned} L &= \frac{D_o - D_i}{2 \tan 16^\circ} \\ &= \frac{6.46 - 2.25}{2 \tan 16^\circ} \\ &= 7.33 \quad \text{say } 7\frac{1}{4} \text{ inches.} \end{aligned}$$

The length of the chord from the inlet to the throat is determined as:

$$L = \left[R^2 - \left\{ R - \frac{(D_o - D_i)}{2} \right\}^2 \right]^{\frac{1}{2}}$$

$$\begin{aligned} &= \left[14^2 - \left\{ 14 - \frac{(2.25-1.75)}{2} \right\}^2 \right]^{\frac{1}{2}} \\ &= 2.56 \quad \text{say } 2-9/16 \text{ inches.} \end{aligned}$$

The diffuser is designed and cast in two parts which facilitates the machining of the inner surfaces. The diffuser is made of cast iron and the walls are cast one-half an inch thick. A portion of the throat section $2\frac{1}{2}$ inches long is included in the low pressure end, making the total length of the low pressure end $7\frac{1}{4} + 2-9/16 + 2\frac{1}{2} = 12-5/16$ inches. A standard flanged fitting is positioned on the low pressure section of the diffuser so that the rim on the entrance of the diffuser is one-half inch from the edge of the evaporator vapor outlet.

Assuming a velocity of 200 fps at the discharge from the diffuser and with the enthalpy and pressure known, the specific volume of the mixture is found to be $v_d = 376$ cu. ft/lb. The diameter of the diffuser exit is then determined:

$$\begin{aligned} A_d &= w_m v_d / V_d \\ &= 0.0436 \times 376 / 200 \\ &= 11.8 \text{ sq. inches.} \\ D_d &= (A / 0.7854)^{\frac{1}{2}} \\ &= (11.8 / 0.7854)^{\frac{1}{2}} \\ &= 3.88 \quad \text{say } 4 \text{ inches.} \end{aligned}$$

The diffuser is cast with a standard four inch flange on the exit in

order that a standard four inch 90° elbow can be used to connect it to the inlet on the condenser. A standard 1-3/4 inch flange is cast on the throat end of each portion of the diffuser.

The length of the high pressure diverging portion of the diffuser is determined by:

$$\begin{aligned} L &= \frac{D_d - D_t}{2 \tan 3^\circ} \\ &= \frac{4 - 1.75}{2 \times 0.0524} \\ &= 21\frac{1}{2} \text{ inches.} \end{aligned}$$

A one inch straight section of the throat is added to the high pressure section thus making the total length $21\frac{1}{2} + 1 = 22\frac{1}{2}$ inches.

The position of the nozzle exit in relation to the inlet of the diffuser is determined by projecting the cone of the nozzle to the inlet of the diffuser. This distance is determined as:

$$\begin{aligned} \text{distance a-1} &= \frac{D_1 - D_2}{2 \tan 5^\circ} \\ &= \frac{2.25 - 1.25}{2 \tan 5^\circ} \\ &= 5.725 \text{ inches.} \end{aligned}$$

The nozzle is placed 5.725 minus 7.250 or $1\frac{1}{2}$ inches inside the diffuser entrance. The diffuser entrance is 18 inches (distance between flange faces on the tee) minus $4\frac{1}{2}$ inches or $13\frac{1}{2}$ inches from the back face of the tee.

The exit of the nozzle is therefore $13\frac{1}{2}$ plus $1\frac{1}{2}$ or 15 inches from the back face of the tee.

G - The Heat Exchanger

As the primary object of the investigation was the refrigeration of the unit a method had to be devised to put a load on the unit. The simplest method of doing this was to circulate the chilled water through a heat exchanger.

The heat exchanger was designed according to the principles in Gaffert's (16), to furnish the heat equivalent of eight tons of refrigeration, using a steam supply of 140 psi gage pressure. The heat exchanger was made with an excess of three tons capacity to handle any extra refrigeration that the unit might have and to make it possible to vary the temperature of the circulating chilled water.

H - The Chilled Water Circulating Pump

A suitable centrifugal pump available in the laboratory was used as a circulating pump. The pump was an Ingersoll Rand Model B, Series 0940543, Type 2LRH-NSI rated at 10 gallons per minute against a 70 foot head. A flow of ten gallons per minute is the equivalent of 1.40 cfm which is slightly below the desired flow. The reduced flow would not materially change the results other than having a circulating water temperature difference greater than 10 degrees.

I - The Spray Nozzle

The spray nozzle is made from a $1\frac{1}{2}$ inch section of $\frac{3}{4}$ inch wrought iron pipe. The necessary area for the discharge of the return water at 1.40 cfs was determined by making the total area of the discharge holes two-thirds of the flow area of the pipe. With discharge holes $\frac{7}{64}$ of an inch in diameter, 34 holes were needed. The discharge velocity from these holes is:

$$\begin{aligned} V &= \frac{\text{Flow}}{A} \\ &= \frac{(1.40)(144)}{(0.1091)^2 \times 0.7854 \times 34 \times 60} \\ &= 10.52 \text{ fps} \end{aligned}$$

At this velocity the water will strike the sides of the tank and break up into a fine spray.

J - The Measuring Instruments

1 - The Water Meters. In order to determine the refrigerating effect it was necessary to measure the rate of flow of the chilled water. A Tropic rotary disk water meter, number 1561922, was procured that would measure the flow of water in hundredths of a cubic foot. This meter was manufactured by the Pittsburg Equitable Meter Company.

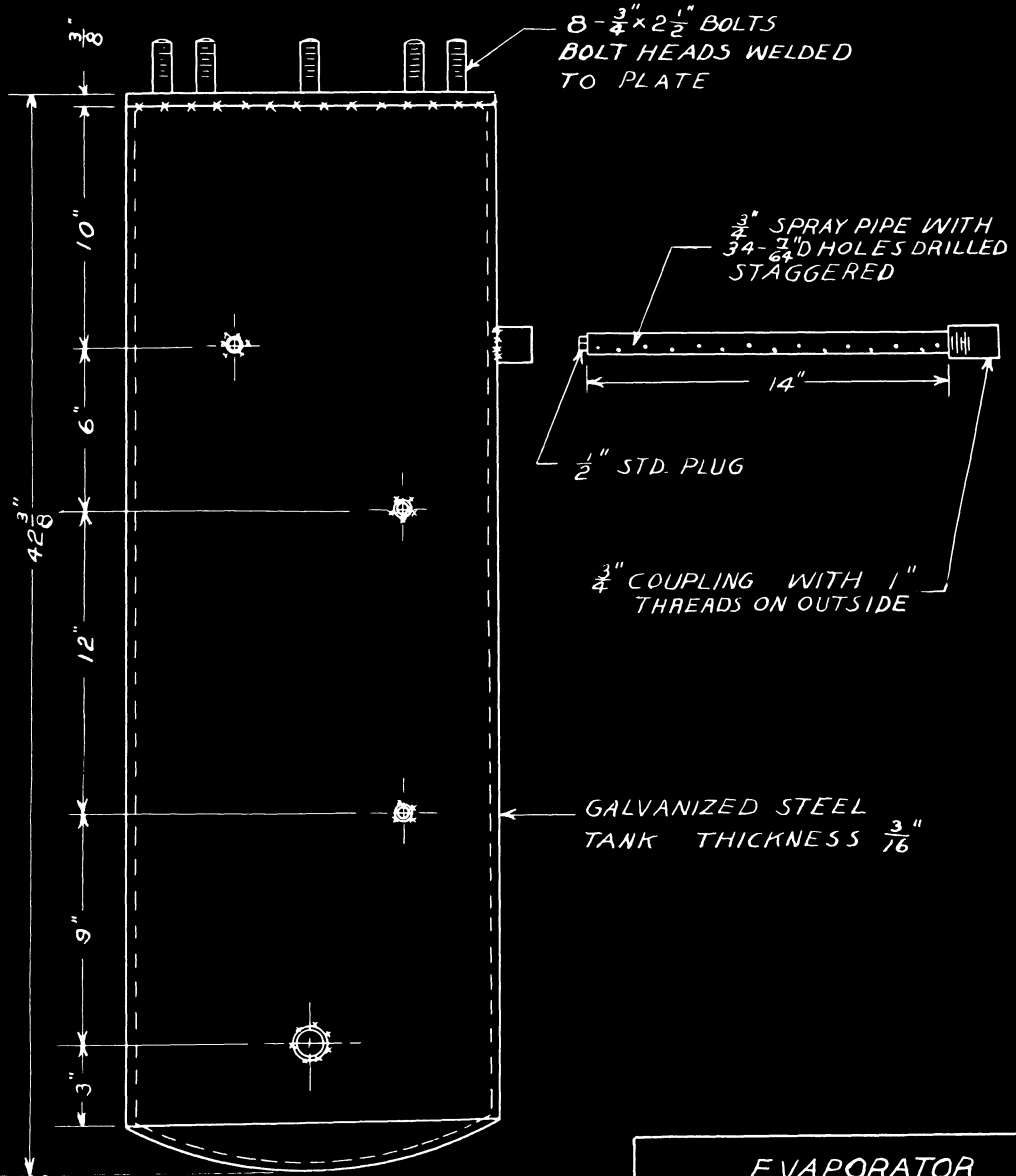
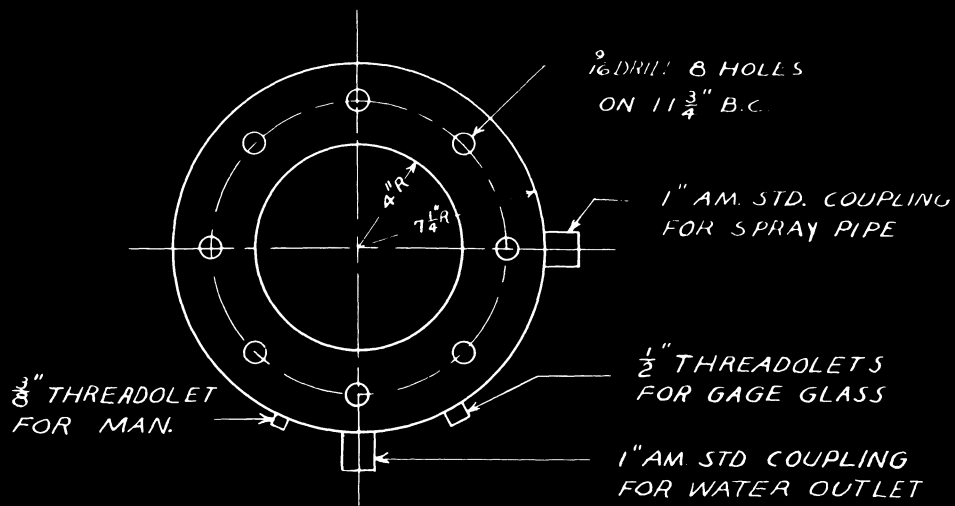
A water meter was also needed in the make up water line to determine the amount of make up water supplied to the evaporator. A Worthington rotary disk water meter, number 1334006, which measured the flow of water in

gallons was obtained.

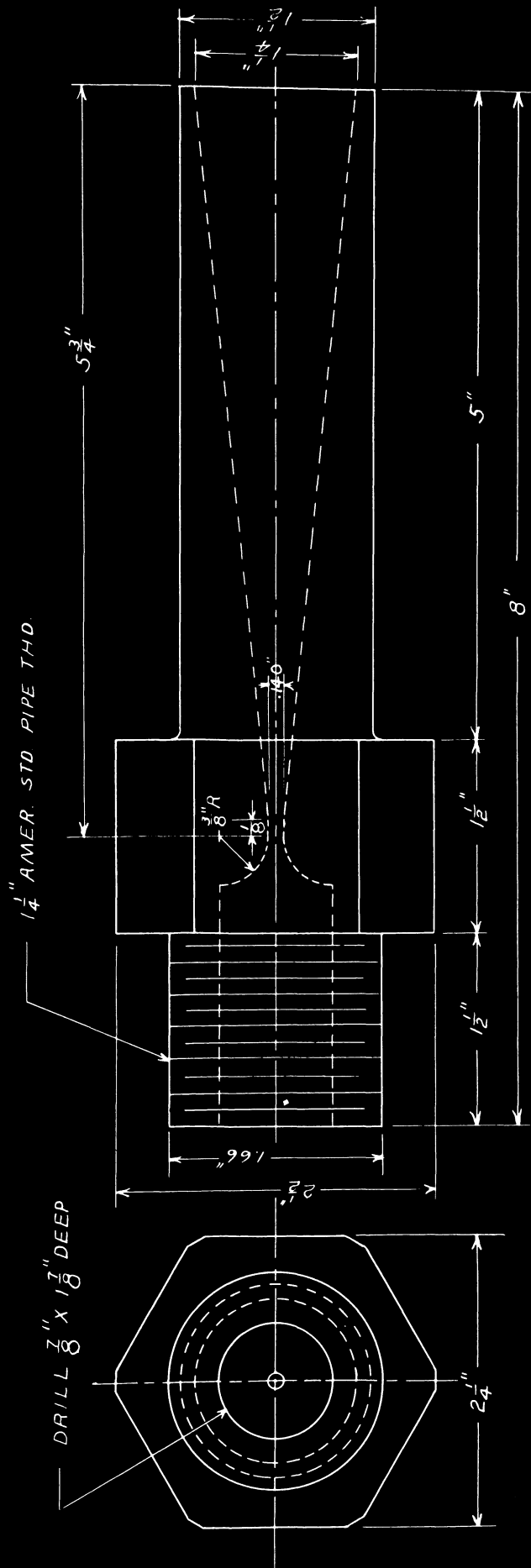
2 - The Pressure Gauge. A Lenersan pressure gauge, Mechanical Engineering Laboratory number 107, was secured to measure the steam pressure supplied to the nozzle of the steam-jet ejector. This pressure gauge had a range from 30 inches Hg. vacuum to 150 pounds per square inch pressure.

3 - The Pressure Regulating Valve. A K & T pressure regulating valve was used to maintain a constant pressure on the steam nozzle.

4 - The Thermometers. Two nitrogen filled Mercury thermometers, ranging from minus 30 degrees to plus 120 degrees Fahrenheit, and made by the Fisher Scientific Company were used to measure the temperature of the chilled water and the return warm water.

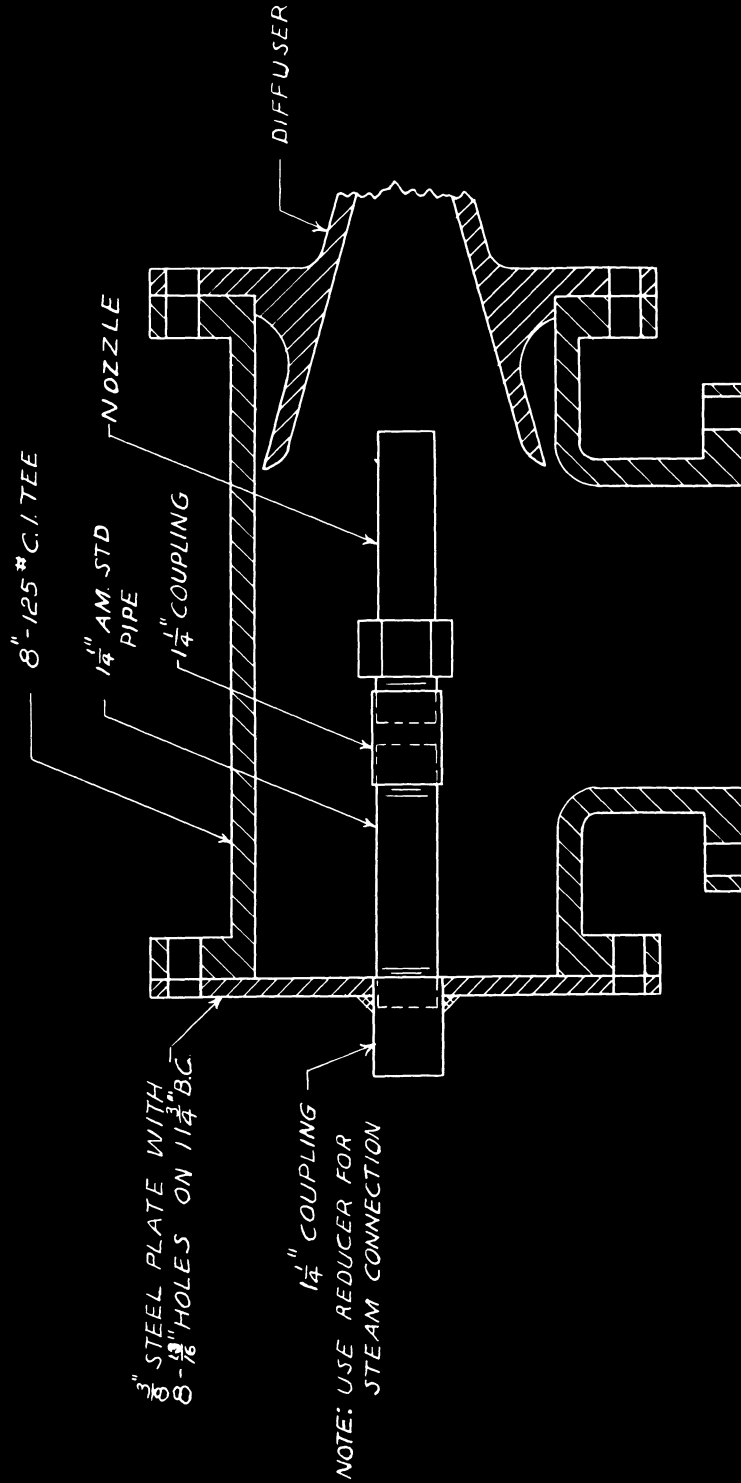


EVAPORATOR
FOR
STEAM JET UNIT
DRAWN BY: A.W.H. APRIL 23 48
CHECKED BY: C.W.V. SCALE 1"=6"



MATERIAL: BRONZE
 ONE REQUIRED
 FINISH ALL OVER

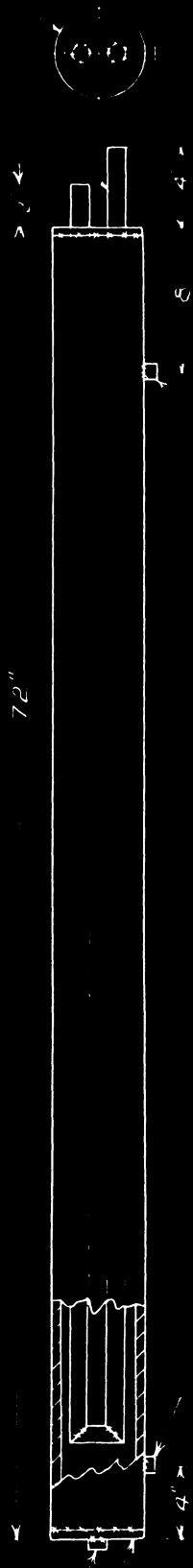
NOZZLE
 FOR
 STEAM JET UNIT
 DRAWN BY: A.W.H. NOV 20, 1947
 CHECKED BY: SCALE: 1"=1"



NOZZLE ASSEMBLY
 FOR
 STEAM JET UNIT
 DRAWN BY: A.W.H. APRIL 26 '48
 CHECKED BY: C.W.V. SCALE 1"=4"

3.12.48
0.1 4 3/4" 4.0

1.44 5.10 PIP



1/2" THREDOLET FOR
ALT. STEAM INLET

72"

1" THREDOLET FOR
STEAM INLET

1/2" THREDOLET FOR
STEAM DRAIN

1/4" STEEL PLATE

**HEAT EXCHANGER
FOR
STEAM JET UNIT**
 DRAWN BY AWH APRIL 28 1948
 CHECKED BY CWV SCALE 1"=8"

IV

CONSTRUCTION

The initial step in the construction of the steam-jet unit was the fabrication of the evaporator tank from a section of an old water tank. After brazing the top, the threadollets, and couplings into position, the evaporator was tested for air leaks by subjecting it to 50 psig air pressure and by applying a soap solution to the brazed points.

The evaporator stand (which consisted of a steel plate brazed to one end of a four inch pipe) was placed in position at the end of the Alberger condenser. This position and the height of the stand were determined by the location of the steam inlet at the top of the condenser. Then the evaporator was placed on the stand and braced.

The heat exchanger was made from a six-foot section of five-inch pipe by brazing the ends and threadollets into position. It, too, was tested for leaks, and then made secure on the wall. The near-by high-pressure steam line was tapped, and a line was installed to the steam connection on the heat exchanger. An impulse steam trap was put into the drain line which carried the condensate from the heat exchanger to the main drain system in the building.

When the chilled-water pump was obtained, it was installed temporarily along with the piping for the water circulating system. A priming line and a line to the gland seal were added to prevent air from leaking into the system. A water meter having been placed in both the circulating line and the make-up line, the main water line close by was tapped, and the make-up line connected. Thermometer wells were placed in the make-up line, and in

the discharge and entrance lines to the evaporator, in order that temperatures could be obtained for calculating the external load or refrigerating effect of the unit.

The throat diameter of the nozzle was larger than specified in the design, so a tapered insert was made for the entrance end in order to attain the desired throat diameter. The inside surface of the diffuser was unfinished when it arrived, but, by use of the lathe in the laboratory, and of a special boring tool which was made, the inside of the diffuser was machined to a reasonably good surface.

After the tee was bolted down on top of the evaporator, the diffuser was bolted to the tee. The four-inch elbow was fastened to the condenser entrance, and the four-by-eight-inch flanged nipple inserted between the elbow and the diffuser exit, thus completing the installation of the diffuser section.

A rubber gasket was used between the tee and the evaporator, but for all other joints, asbestos gaskets were employed.

From the steam line going to the heat exchanger, a steam connection was made to the nozzle assembly. When a weight-and-lever type steam reducing valve had been put in this line to reduce the steam pressure, a steam pressure gage was installed for reading the pressure. A throttling calorimeter was also attached to this steam line in order that the quality of the steam entering the nozzle could be determined.

It was felt that the Bourdon type vacuum gage on the condenser would not give sufficiently accurate readings of the back pressure, so a vertical

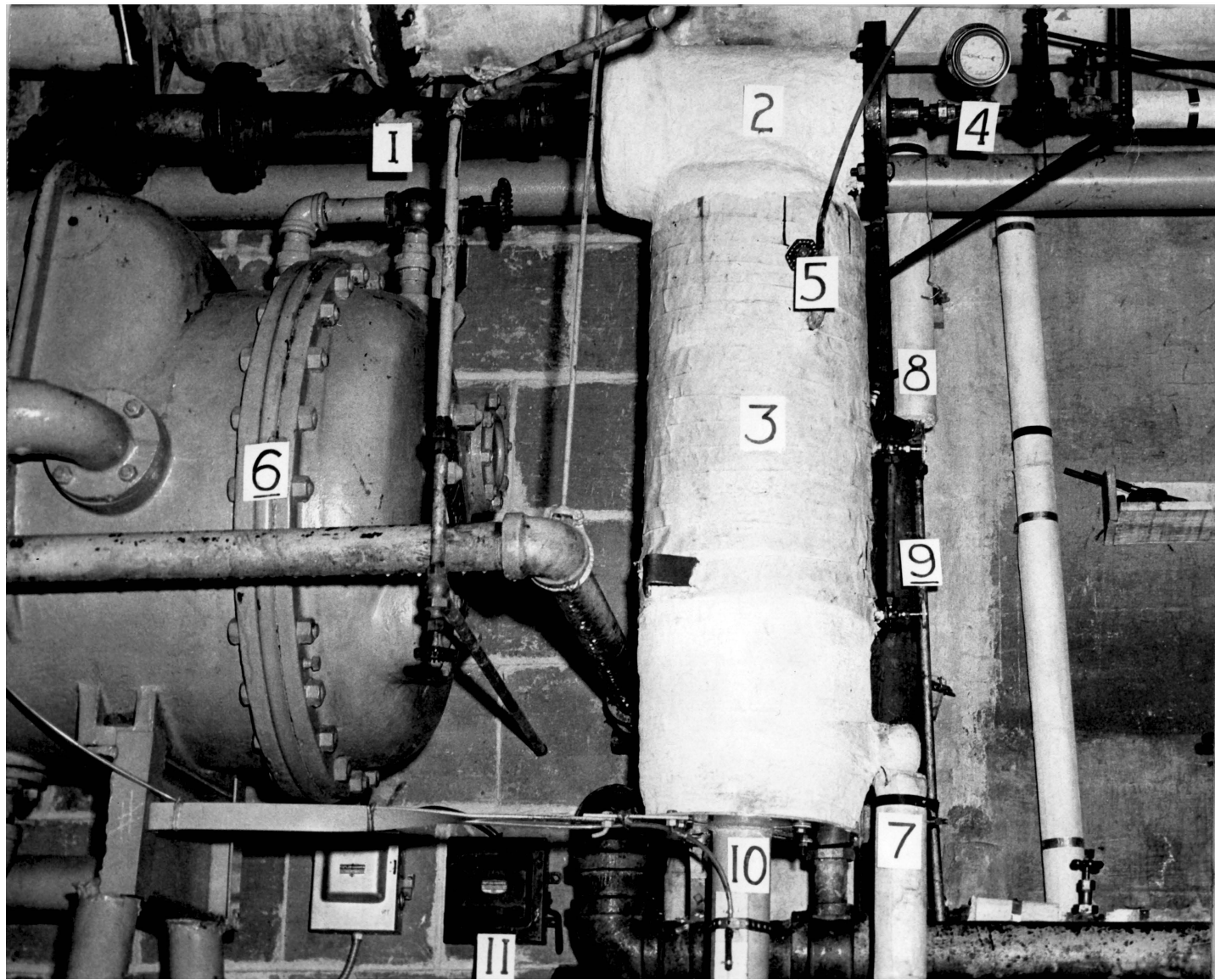
U-tube mercury monometer was constructed and connected to the condenser. The monometer used for measuring the pressure in the evaporator was a thirty-degree inclined U-tube mercury monometer, since a very small pressure change here would have considerable effect on the results.

A blind flange was bolted to the exit end of the diffuser, and the entire unit was put under a 40 psig air pressure and tested for leaks by applying a soap solution. It should be noted that, during the construction, every precaution was taken to eliminate the possibility of leaks by applying pipe compound to all gasket and threaded pipe joints.

Approximately one inch of asbestos insulation was used to cover the entire evaporator and tee section of the unit to decrease the amount of heat transferred to the unit from the surrounding air. Magnesia pipe insulation was put on all the steam lines to decrease condensation of the steam, and corrugated asbestos insulation on the chilled-water lines to insure the best regulation of load at the heat exchanger.

Calibration of the instruments was a relatively simple matter, as the only instruments to be calibrated were the water meters, the steam-pressure gage, and the thermometers. The water meters were checked by weighing the water flowing through them on a set of platform scales for various rates of flow. The water meters were within one per cent accuracy, so no calibration curve was necessary. The pressure gage was tested on a Shaeffer and Budenburg Dead-weight Pressure Gage Tester. The gage was found to be accurate to within two pounds for the range from 35 psig to 150 psig. This accuracy was deemed sufficient for all practical purposes.

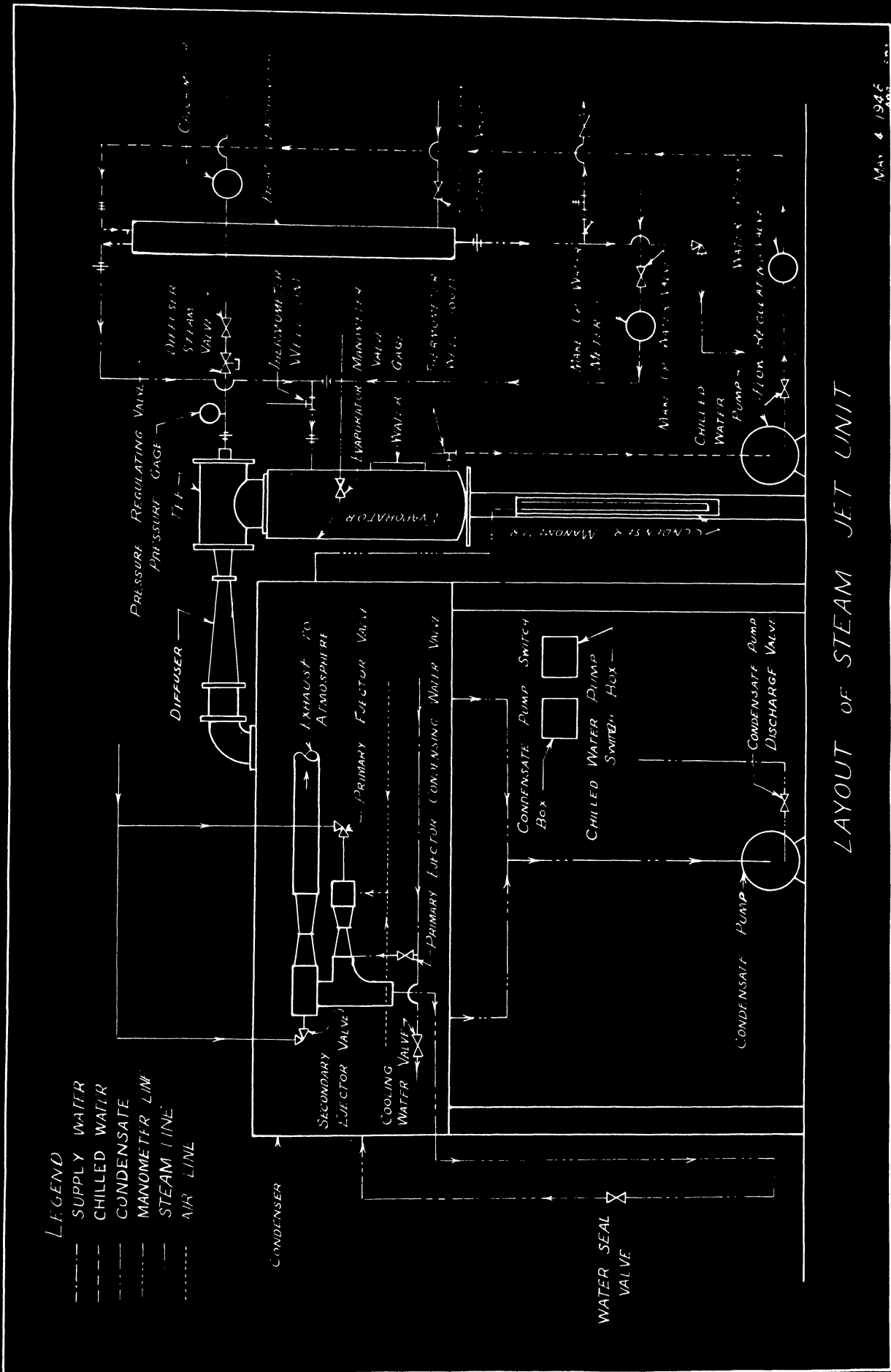
Several thermometers were obtained and comparison of their readings made at various temperatures. Two thermometers which read the same at all temperatures within the range of measurements desired were used for determining the chilled water and return water temperatures.



PHOTOGRAPH OF STEAM-JET UNIT
(see next page)

KEY TO PHOTOGRAPH

- 1 - Diffuser
- 2 - Tee or combining chamber
- 3 - Evaporator Tank
- 4 - Nozzle Assembly
Pressure Gage
Reducing Valve
- 5 - Outlet for Manometer
- 6 - Alberger Condenser
- 7 - Chilled-Water Outlet
- 8 - Water Line to Spray Nozzle
- 9 - Make-Up Water Line
- 10 - Condenser Manometer on Evaporator Stand
- 11 - Switch Box for Chilled-Water Circulating Pump



- LEGEND**
- SUPPLY WATER
 - CHILLED WATER
 - CONDENSATE
 - MANOMETER LINE
 - STEAM LINE
 - AIR LINE

LAYOUT OF STEAM JET UNIT

FIG. 1

May 4 1946

PROCEDURE

The unit was started by the filling of the system and evaporator to the proper water level; the turning on of the circulating water for the condenser to allow cool water to pass through the condenser tubes; the opening of the steam valve to the second stage of the condenser ejector; and the starting of the condensate pump. The cooling water to the inter-condenser on the first stage ejector was opened to about three-fourths of a turn. When the vacuum in the condenser was about twenty-two inches of mercury, the first stage ejector was cut on, and after the condenser pressure was reduced to two inches of mercury absolute, the booster ejector on the evaporator was turned on, and the chilled-water pump started to allow the cooling water to be circulated through the load or heat exchanger. When the desired value of the chilled-water temperature had been reached, heat was added to the water by means of steam admitted to the heat exchanger, in order that the water might be kept at the correct temperature.

Upon completion of the preliminary tests, which were run to familiarize the authors with the operation of the unit, the test runs were begun by placing a one and one-quarter-inch nipple in the nozzle assembly in order to put the exit end of the nozzle fourteen and three-quarter inches from the back face of the tee. This was approximately at the calculated nozzle position of fifteen inches. The condenser pressure was set at two inches of mercury absolute by the regulation of the cooling water, and the steam pressure reducing valve was regulated so that the steam pressure gauge read 110 psig. The temperature of the water leaving the evaporator was maintained

at fifty degrees Fahrenheit by the addition of steam to the heat exchanger. When equilibrium conditions were attained, the following readings were recorded: condenser pressure, steam pressure, barometric pressure, temperatures of the water entering and leaving the evaporator, room temperature, temperature of the flange on the four inch elbow, chilled water flow, and the time. Also, the relative temperatures of the diffuser were determined by feeling the various sections, especially at the throat. Similar runs were made with the condenser pressure at 1.7, 1.4, and 1.0 inches of mercury absolute. The unit was operated at least ten minutes after equilibrium conditions were established before readings were taken. These tests were repeated with the nozzle in eleven other positions, ranging from thirteen and one-quarter inches to sixteen and one-sixteenth inches.

Tests were made at one and two inches condenser pressure with steam pressures ranging from thirty-five psig to one-hundred and forty psig while the nozzle exit was fifteen and one-quarter inches from the back flange. A set of tests was also made with the chilled water temperature varying from 40° F to 70° F, while the steam pressure was one-hundred and ten psig, and the condenser pressure one inch mercury.

When the above tests were completed, the steam pressure gage was again calibrated, and was found to have a constant negative error of nineteen psi. The gage was corrected and tests repeated for seven different nozzle positions at one inch and two inches condenser pressure, with the steam pressure at 110 psig. A set of runs was made with the chilled-water temperature varying from 40° F to 75° F while the condenser pressure was two

inches mercury, and the nozzle exit was fifteen and one quarter inches. Additional tests were run for the unit operating under different steam pressures. The maximum steam pressure available was 140 psig.

VI

DATA AND RESULTS

Table 1

Varying Back Pressures at Different Nozzle Positions

Nozzle Position 13 $\frac{1}{2}$ inches

Cond. Vacuum In. Hg.	Barm. Rdg. In. Hg.	Temp. H ₂ O out °F	Temp. H ₂ O in °F	Temp. Room °F	Temp. Flange °F	Stm. Press. psig	Corr. Stm. Press. psig	Quant. Flow cfs	H. E. @ 50° ton	Comments*
26.20	28.22	65.0	66.5	90	107	110	91	1.42		Warm
26.20	28.22	<u>66.0</u> 65.5	<u>67.5</u> 67	90	110	110	91	<u>1.42</u> 1.42	.37	Warm
26.50	28.22	57.0	58.8	90	110	110	91	1.42		Warm
26.50	28.22	<u>56.7</u> 56.9	<u>58.4</u> 58.6	90	110	110	91	<u>1.42</u> 1.42	.63	Warm
26.80	28.22	50.0	53.2	86	106	111	92	1.42		Cold-3
26.80	28.22	<u>50.0</u> 50.0	<u>53.2</u> 53.2	88	104	111	92	<u>1.42</u> 1.42	1.37	Cold-2
27.20	28.22	50.0	53.5	89	105	111	92	1.42		Cold-3
27.20	28.22	<u>50.1</u> 50.1	<u>53.8</u> 53.7	89	106	111	92	<u>1.42</u> 1.42	1.60	Cold-2

*The flange at the throat of the diffuser is used as a reference point for the cold spot.
 A (-) indicates the distance from the throat on the low pressure end of the diffuser and
 a (+) refers to a point on the high pressure end.

Table 1 continued

Varying Back Pressures at Different Nozzle Positions

Nozzle Position 13-3/4 inches

Cond. Vacuum In. Hg.	Baro. Rdg. In. Hg.	Temp. H ₂ O out °F	Temp. H ₂ O in °F	Temp. Room °F	Temp. Flange °F	Stm. Press. psig	Corr. Stm. Press. psig	Quant. Flow cfs	R. E. @ 50° ton	Comments
26.34	28.33	61.9	63.8	85	112	110	91	1.42		Warm 0
26.32	28.33	<u>62.1</u> 62.0	<u>64.0</u> 63.9	85	112	110	91	<u>1.42</u> 1.42	.55	Warm 0
26.63	28.33	57.0	58.8	85	114	110	91	1.42		Warm 0
26.63	28.33	<u>57.0</u> 57.0	<u>58.7</u> 58.7	85	112	110	91	<u>1.42</u> 1.42	.60	Warm 0
26.94	28.33	50.0	53.2	85	100	110	91	1.42		Cool-2
26.95	28.33	<u>50.0</u> 50.0	<u>53.2</u> 53.2	85	98	110	91	<u>1.42</u> 1.42	1.42	
27.34	28.33	50.0	53.9	85	100	110	91	1.42		Cool/2
27.34	28.33	<u>50.0</u> 50.0	<u>53.9</u> 53.9	85	98	110	91	<u>1.42</u> 1.42	1.73	Cool/1

Table 1 continued

Varying Back Pressures at Different Nozzle Positions

Nozzle Position 14 inches

Cond. Vacuum In. Hg.	Baro. Edg. In. Hg.	Temp. H ₂ O out °F	Temp. H ₂ O in °F	Temp. Room °F	Temp. Flange °F	Stm. Press. psig	Corr. Stm. Press. psig	Quant. Flow cfs	R.E. @ 50° ton	Comments
26.00	27.97	63.7	64.8	99	110	111	92	1.42		Warm 0
26.00	27.97	<u>63.7</u> 63.7	<u>64.8</u> 64.8	99	112	110	91	<u>1.42</u> 1.42	.28	
26.30	27.97	56.2	57.8	99	109	110	91	1.42		Cool-
26.30	27.97	<u>56.1</u> 56.2	<u>57.6</u> 57.7	99	110	110	91	<u>1.42</u> 1.42	.58	
26.60	27.97	50.0	53.1	99	112	112	93	1.42		Cool 0
26.60	27.97	<u>50.1</u> 50.1	<u>53.1</u> 53.1	100	108	112	93	<u>1.42</u> 1.42	1.33	
27.00	27.97	50.0	53.1	101	108	110	91	1.42		Cold 0
27.00	27.97	<u>50.0</u> 50.0	<u>53.0</u> 53.0	101	-	110	91	<u>1.42</u> 1.42	1.33	

Table 1 continued

Varying Back Pressures at Different Nozzle Positions

Nozzle Position 14-3/8 inches

Cond. Vacuum In. Hg.	Baro. Rdg. In. Hg.	Temp. H ₂ O out °F	Temp. H ₂ O in °F	Temp. Room °F	Temp. Flange °F	Stm. Press. psig	Corr. Stm. Press. psig	Quant. Flow cfs	R.E. @ 50° ton	Comments
26.20	28.18	62.0	63.3	91	110	110	91	1.42		Cool-7
26.20	28.18	<u>62.0</u> 62.0	<u>63.3</u> 63.3	91	110	110	91	<u>1.42</u> 1.42	.38	
26.50	28.18	54.8	56.4	93	108	110	91	1.42		Cool-7
26.50	28.18	<u>55.2</u> 55.0	<u>56.8</u> 56.6	93	108	110	91	<u>1.42</u> 1.42	.60	
26.80	28.18	50.0	52.7	93	103	110	91	1.42		
26.80	28.18	<u>50.1</u> 50.1	<u>52.5</u> 52.6	91	100	110	91	<u>1.42</u> 1.42	1.11	Cold-6
27.20	28.18	50.1	54.6	91	94	110	91	1.42		
27.20	28.18	<u>50.0</u> 50.1	<u>54.5</u> 54.6	91	92	110	91	<u>1.42</u> 1.42	1.55	Cold 0

Table 1 continued

Varying Back Pressures at Different Nozzle Positions

Nozzle Position 14-3/4 inches

Cond. Vacuum In. Hg.	Barm. Rdg. In. Hg.	Temp. H ₂ O out °F	Temp. H ₂ O in °F	Temp. Room °F	Temp. Flange °F	Stm. Press. psig	Corr. Stm. Press. psig	Quant. Flow cfs	R. K. @ 50° ton	Comments
26.15	28.13	66.5	68.0	91	112	110	91	1.42		Cool 0
26.13	28.13	<u>67.1</u> 66.8	<u>68.6</u> 68.3	91	112	110	91	<u>1.42</u> 1.42	.35	Cool 0
26.43	28.13	61.0	63.3	91	112	110	91	1.42		Cold-7
26.43	28.13	<u>61.0</u> 61.0	<u>63.3</u> 63.3	91	112	110	91	<u>1.42</u> 1.42	.68	Cold-7
26.78	28.14	50.8	53.2	91	105	110	91	1.42		Cold-7
26.78	28.14	<u>50.8</u> 50.8	<u>53.2</u> 53.2	91	105	110	91	<u>1.42</u> 1.42	1.06	Cold-7
27.19	28.14	50.1	53.4	87	103	110	91	1.42		Cold-1
27.13	28.14	<u>50.5</u> 50.3	<u>53.8</u> 53.6	87	103	110	91	<u>1.42</u> 1.42	1.46	Cold-1

Table 1 continued

Varying Back Pressures at Different Nozzle Positions

Nozzle Position 15-1/8 inches

Cond. Vacuum In. Hg.	Baro. Rdg. In. Hg.	Temp. H ₂ O out °F	Temp. H ₂ O in °F	Temp. Room °F	Temp. Flange °F	Stm. Press. psig	Corr. Stm. Press. psig	Quant. Flow cfs	R. X. @ 50° ton	Comments
25.80	27.82	63.0	64.8	104	120	110	91	1.42		Cool-7
25.80	27.82	<u>63.0</u> 63.0	<u>64.7</u> 64.7	104	122	110	91	<u>1.42</u> 1.42	.48	
26.10	27.82	58.5	58.5	102	-	110	91	1.42		Cool-7
26.10	27.82	<u>58.5</u> 58.5	<u>58.5</u> 58.5	102	-	110	91	<u>1.42</u> 1.42	.74	
26.40	27.82	52.0	54.5	102	105	110	91	1.42		Cool 0
26.40	27.82	<u>51.0</u> 51.5	<u>53.5</u> 54.0	102	108	110	91	<u>1.42</u> 1.42	1.11	
26.80	27.82	50.0	54.4	103	93	110	91	1.42		
26.80	27.82	<u>50.0</u> 50.0	<u>54.4</u> 54.4	103	93	110	91	<u>1.42</u> 1.42	1.95	

Table 1 continued

Varying Back Pressures at Different Nozzle Positions

Nozzle Position 15 1/2 inches

Cond. Vacuum In. Hg.	Baro. Rdg. In. Hg.	Temp. H ₂ O out °F	Temp. H ₂ O in °F	Temp. Room °F	Temp. Flange °F	Stm. Press. psig	Corr. Stm. Press. psig	Quant. Flow cfs	R. E. @ 50° ton	Comments
26.34	28.22	51.4	53.4	91	106	110	91	1.42		Cold-5
26.22	28.22	<u>52.0</u>	<u>54.0</u>	91	107	110	91	<u>1.42</u>		
		51.7	53.7					1.42	.67	
26.50	28.22	50.1	53.1	90	107	110	91	1.42		Cool 0 Cool 0
26.52	28.22	<u>50.0</u>	<u>53.2</u>	90	107	110	91	<u>1.42</u>		
		50.1	53.1					1.42	1.33	
26.82	28.22	50.0	54.5	90	106	110	91	1.42		Cold-1 Cold-1
26.82	28.22	<u>50.0</u>	<u>54.5</u>	90	104	110	91	<u>1.42</u>		
		50.0	54.5					1.42	1.99	
27.22	28.22	50.0	54.7	93	93	110	91	1.42		Cold-0 Cold-0
27.22	28.22	<u>50.0</u>	<u>54.7</u>	93	92	110	91	<u>1.42</u>		
		50.0	54.7					1.42	2.21	

Table 1 continued

Varying Back Pressures at Different Nozzle Positions

Nozzle Position 15-5/16 inches

Cond. Vacuum In. Hg.	Barn. Rdg. In. Hg.	Temp. H ₂ O out °F	Temp. H ₂ O in °F	Temp. Room °F	Temp. Flange °F	Stm. Press. psig	Corr. Stm. Press. psig	Quant. Flow cfs	R.E. @ 50° ton	Comments
25.76	27.74	56.0	57.0	96	106	112	93	1.42		Cold-6
25.76	27.74	<u>54.7</u> 54.8	<u>56.5</u> 56.7	95	107	111	92	<u>1.42</u> 1.42 1.43	.76	
26.10	27.74	50.1	53.5	94	104	111	92	1.42		
26.10	27.74	<u>50.0</u> 50.1	<u>54.0</u> 54.7	95	103	111	92	<u>1.42</u> 1.42 1.42	1.64	Cold-5
26.34	27.74	50.0	54.4	94	103	110	91	1.42		Cold-4
26.34	27.74	<u>50.0</u> 50.0	<u>54.5</u> 54.4	95	98	111	92	<u>1.42</u> 1.42 1.42	1.99	
26.75	27.74	50.3	55.2	94	88	110	91	1.42		
26.75	27.74	<u>50.2</u> 50.3	<u>55.2</u> 55.2	94	81	111	92	<u>1.42</u> 1.42 1.42	2.17	Cold/1 Cold/2

-2-

Table 1 continued

Varying Back Pressures at Different Nozzle Positions

Nozzle Position 15-7/16 inches

Cond. Vacuum In. Hg.	Barn. Rdg. In. Hg.	Temp. H ₂ O out °F	Temp. H ₂ O in °F	Temp. Room °F	Temp. Flange °F	Sta. Press. psig	Corr. Sta. Press. psig	Quant. Flow cfs	R. E. @ 50° ton	Comments
25.75	27.78	56.6	58.3	102	104	112	93	1.42		Cool-4
25.78	27.78	<u>56.0</u> 56.3	<u>57.7</u> 58.0	102	109	110	91	<u>1.42</u> 1.42	.64	Cool-4
26.10	27.78	50.3	53.0	102	106	108	89	1.42		Cool-6
26.10	27.78	<u>50.2</u> 50.3	<u>52.7</u> 52.8	102	115	110	91	<u>1.42</u> 1.42	1.10	Cool-6
26.40	27.78	50.0	53.5	103	114	110	91	1.42		Cold-4
26.40	27.78	<u>50.0</u> 50.0	<u>53.3</u> 53.4	103	112	110	91	<u>1.42</u> 1.42	1.51	Cold-4
26.80	27.78	50.0	54.3	103	96	110	91	1.42		Cold-1
26.80	27.78	<u>50.0</u> 50.0	<u>54.4</u> 54.4	102	88	110	91	<u>1.42</u> 1.42	1.95	Cold 0

1
6

Table 1 continued

Varying Back Pressures at Different Nozzle Positions

Nozzle Position 15-5/8 inches

Cond. Vacuum In. Hg.	Barn. Hdg. In. Hg.	Temp. H ₂ O out °F	Temp. H ₂ O in °F	Temp. Room °F	Temp. Flange °F	Stm. Press. psig	Corr. Stm. Press. psig	Quant. Flow cfs	R. M. @ 50° ton	Comments
26.30	28.26	67.6	69.3	82	108	110	93	1.42		Warm
26.30	28.26	<u>67.7</u> 67.7	<u>69.8</u> 69.0	82	109	110	93	<u>1.42</u> 1.42	.30	Warm
26.56	28.26	60.0	61.6	82	107	110	93	1.42		Cool-7
26.56	28.26	<u>60.0</u> 60.0	<u>61.6</u> 61.6	82	107	110	93	<u>1.42</u> 1.42	.50	
26.86	28.26	52.8	54.4	82	102	110	93	1.42		Cold-5
26.86	28.26	<u>52.2</u> 52.5	<u>54.3</u> 54.3	82	98	110	93	<u>1.42</u> 1.42	.73	Cold-5
27.26	28.26	50.1	54.0	82	92	110	93	1.42		Cold/2
27.26	28.26	<u>50.1</u> 50.1	<u>54.0</u> 54.0	82	88	110	93	<u>1.42</u> 1.42	1.73	Cold(-1)

Table 1 continued

Varying Back Pressures at Different Nozzle Positions

Nozzle Position 16-1/16 inches

Cond. Vacuum In. Hg.	Baro. Rdg. In. Hg.	Temp. H ₂ O _{out} °F	Temp. H ₂ O _{in} °F	Temp. Room °F	Temp. Flange °F	Sta. Press. psig	Corr. Sta. Press. psig	Quant. Flow cfs	R.E. @ 50° ton	Comments
26.22	28.23	69.0	70.0	85	108	110	91	1.42		Cool-7
26.22	28.23	<u>69.0</u>	<u>70.0</u>	85	109	110	91	<u>1.42</u>		Cool-7
		69.0	70.0					1.42	.32	
26.55	28.23	63.5	65.0	84	107	110	91	1.42		Cool-2
26.55	28.23	<u>63.5</u>	<u>65.1</u>	84	107	110	91	<u>1.42</u>		Cool-2
		63.5	65.0					1.42	.41	
26.82	28.23	50.4	52.4	83	96	110	91	1.42		Cold-2
26.82	28.23	<u>51.0</u>	<u>53.0</u>	83	95	110	91	<u>1.42</u>		Cold-2
		50.7	52.7					1.42	.89	
26.22	28.23	50.0	53.2	82	89	110	91	1.42		Cold-1
26.23	28.23	<u>50.1</u>	<u>53.6</u>	82	88	110	91	<u>1.42</u>		Cold-1
		50.1	53.4					1.42	1.42	

Table 1 continued

Varying Back Pressures at Different Nozzle Positions

Nozzle Position 16-9/16 inches

Cond. Vacuum In. Hg.	Barn. Rdg. In. Hg.	Temp. H ₂ O out °F	Temp. H ₂ O in °F	Temp. Room °F	Temp. Flange °F	Stm. Press. psig	Corr. Stm. Press. psig	Quant. Flow cfs	R.H. @ 50° ton	Comments
25.92	27.95	68.0	69.0	105	105	110	91	1.42		Cool 0
25.92	27.95	<u>69.0</u> 68.5	<u>70.0</u> 69.5	105	107	110	91	<u>1.42</u> 1.42	.23	
26.28	27.95	63.6	64.7	105	115	110	91	1.42		Cool 0
26.27	27.95	<u>63.4</u> 63.5	<u>64.5</u> 64.6	105	116	110	91	<u>1.42</u> 1.42	.30	
26.58	27.95	50.0	51.8	106	117	110	91	1.42		Cold 0
26.58	27.95	<u>50.0</u> 50.0	<u>51.9</u> 51.8	106	118	110	91	<u>1.42</u> 1.42	.80	Cold 0
27.0	27.95	50.0	53.0	106	117	110	91	1.42		Cold 1/2
27.0	27.95	<u>50.0</u> 50.0	<u>53.0</u> 53.0	108	117	110	91	<u>1.42</u> 1.42	1.33	Cold 1/2

Table 2

Varying Back Pressures at Different Nozzle Positions

Nozzle Position 13 $\frac{1}{4}$ inches

Cond. Vacuum In. Hg.	Baro. Rdg. In. Hg.	Temp. H ₂ O out °F	Temp. H ₂ O in °F	Temp. Room °F	Temp. Flange °F	Stm. Press. psig	Corr. Stm. Press. psig	Quant. Flow cfs	H.E. @ 50° ton	Comments
25.68	27.68	52.9	54.0	93	108	110	110	1.42	.46	Cool
26.68	27.68	50.0	54.4	92	100	110	110	1.42	1.95	Cold 0

Nozzle Position 14 inches

26.0	27.94	62.5	64.1	98	105	110	110	1.42	.45	Cool
27.0	27.94	50.4	53.0	98	95	110	110	1.42	1.15	Cold 3

Nozzle Position 14 $\frac{1}{2}$ inches

25.68	27.68	56.5	57.6	90	102	110	110	1.42	.40	Warm
26.68	27.68	52.0	53.6	90	105	110	110	1.42	.71	Cold 4

Nozzle Position 15-1/16 inches

25.82	27.82	57.0	58.5	91	105	110	110	1.42	.545	Cool
26.82	27.82	51.0	54.6	91	109	110	110	1.42	1.69	Cold 5

Nozzle Position 15 $\frac{1}{2}$ inches

25.82	27.82	50	54.4	95	112	110	110	1.42	1.95	Cold-7
26.82	27.82	50.4	55.3	94	76	110	110	1.42	2.17	Cold 8

Table 2 continued

Varying Back Pressures at Different Nozzle Positions

Cond. Vacuum In. Hg.	Baro. Rdg. In. Hg.	Temp. H ₂ O out °F	Temp. H ₂ O in °F	Temp. Room °F	Temp. Flange °F	Sta. Press. psig	Corr. Sta. Press. psig	Quant. Flow cfs	R.E. @ 50° ton	Comments
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Nozzle Position 15 1/2 inches

25.82	27.82	52.8	54.4	90	108	110	110	1.42	.67	Cool
26.82	27.82	50.1	54.5	90	101	110	110	1.42	1.95	Cold/4

Nozzle Position 16-1/8 inches

25.94	27.94	63.0	54.2	103	95	110	110	1.42	.33	Cool-7
26.94	27.94	50.0	53.0	93	93	110	110	1.42	1.33	Cold/3

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Table 3

Varying Steam Pressures

Cond. Vacuum In. Hg.	Baro. Rdg. In. Hg.	Temp. H ₂ O out ° F	Temp. H ₂ O in ° F	Temp. Nozzle ° F	Temp. Flange ° F	Stm. Press. psig	Corr. Stm. Press. psig	Quant. Flow cfs	H.E. @ 50 ton	Comments
Nozzle Position 15 $\frac{1}{2}$ inches, Back Pressure 1 inch Hg.										
26.8	27.78	50.0	55.0	101	90	110	91	1.42	2.21	Cold 0
26.8	27.78	50.1	55.1	101	95	125	106	1.42	2.21	Cold 0
26.8	27.78	50.0	55.0	101	105	130	111	1.42	2.22	Cold-2
26.8	27.78	50.0	55.0	96	104	95	76	1.42	2.22	Cold-2
26.8	27.78	50.0	55.0	97	93	80	61	1.42	2.22	Cold-2
26.8	27.78	50.1	55.0	97	90	65	46	1.42	2.17	Cold-3
26.8	27.78	50.0	53.6	93	88	50	31	1.42	1.59	Cold-4
26.8	27.78	50.5	60.0	96	88	35	16	1.42	.66	Cool-7
27.0	27.93	50.0	54.8	93	78	150	131	1.42	2.12	Cold-6
27.0	27.93	50.0	55.0	93	79	75	56	1.42	2.22	Cold-2
27.0	27.93	50.2	50.0	93	80	60	41	1.42	2.12	Cold-7
27.0	27.93	50.0	52.2	93	81	40	21	1.42	.93	Cold-7
26.8	27.82	50.1	55.0	94	84	140	140	1.42	2.17	Cold-12
26.8	27.82	50.4	55.3	94	76	110	110	1.42	2.17	Cold-4

Table 3 continued

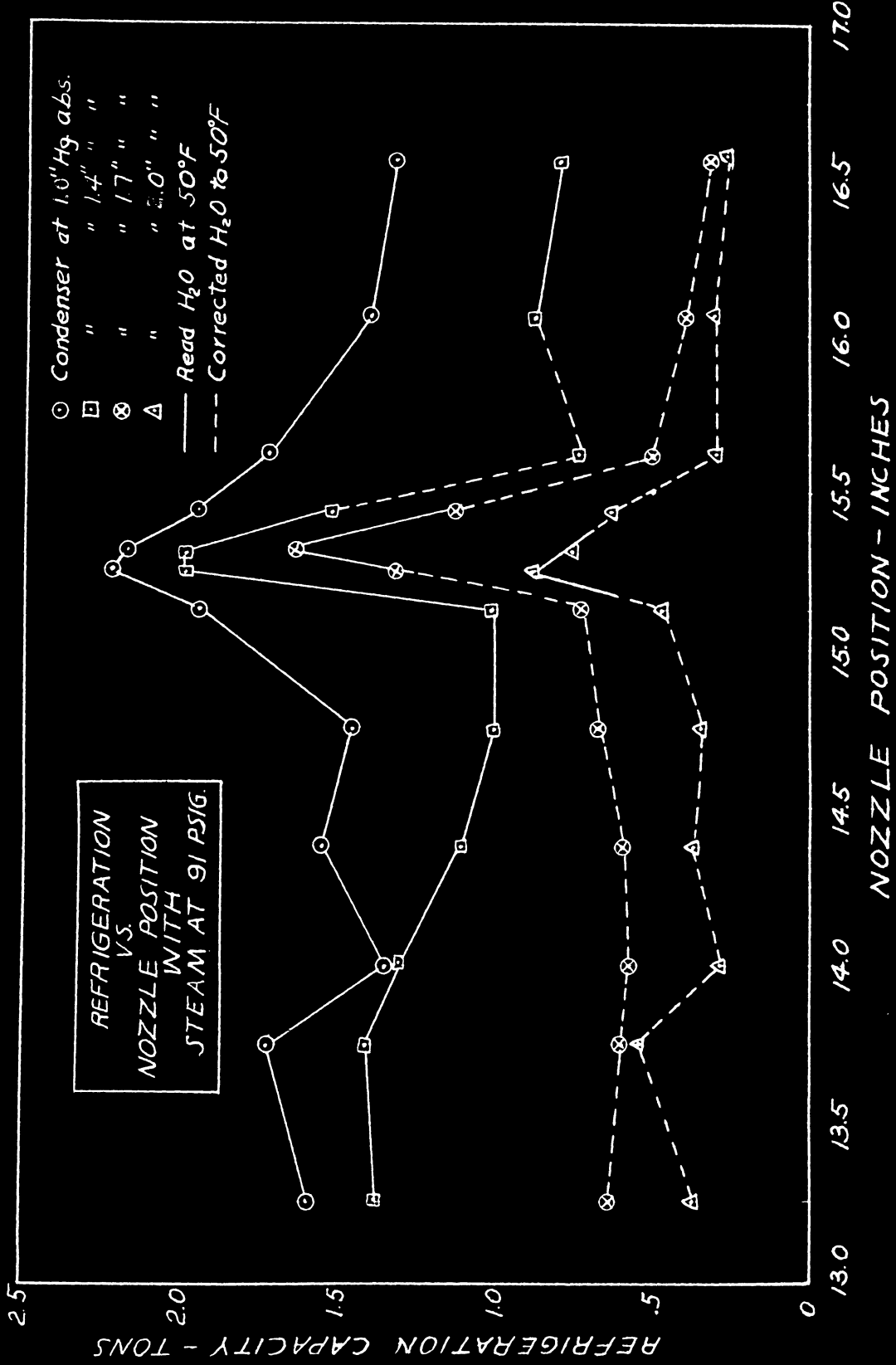
Varying Steam Pressures

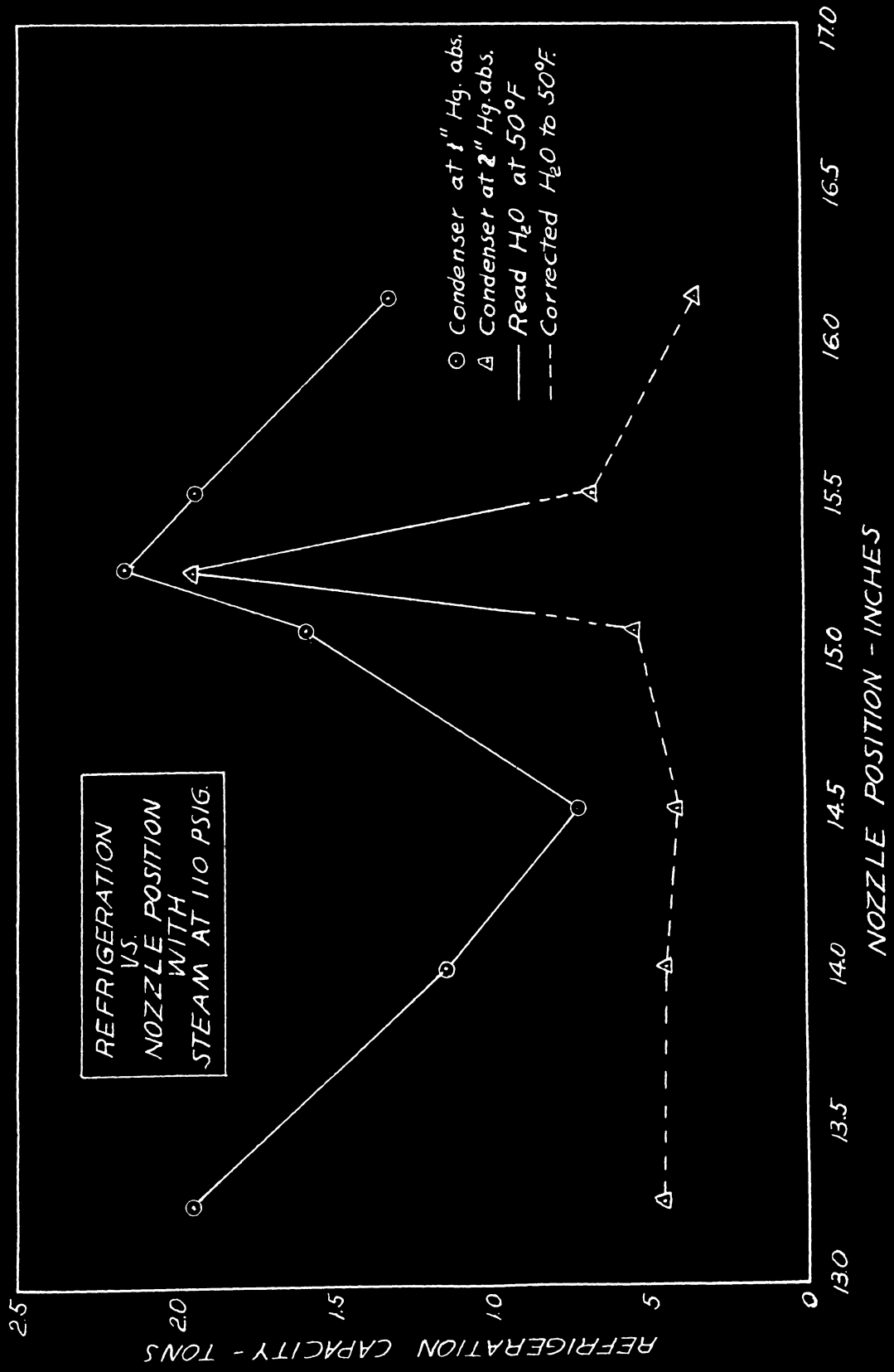
Cond. Vacuum In. Hg.	Baro. Rdg. In. Hg.	Temp. H ₂ O out °F	Temp. H ₂ O in °F	Temp. Room °F	Temp. Flange °F	Stm. Press. psig	Corr. Stm. Press. psig	Quant. Flow cfs	E.E. @ 50° ton	Comments
Nozzle Position 15½ inches, Back Pressure 2 inches Hg.										
26.0	27.95	50.0	54.5	93	110	150	131	1.42	2.0	Cold-7
26.0	27.95	50.0	54.1	93	110	132	113	1.42	1.82	Cold-7
26.0	27.95	50.0	53.5	92	112	130	101	1.42	1.55	Cold-7
26.0	27.95	50.0	52.2	92	112	110	91	1.42	0.97	Cold-7
26.0	27.95	57.0	59.0	92	112	100	81	1.42	.78	Cold-7
26.0	27.95	50.0	52.8	92	112	115	96	1.42	1.23	Cold-7
26.0	27.95	50.0	54.3	92	112	129	110	1.42	1.91	Cold-7
26.0	27.95	50.0	54.4	92	112	140	121	1.42	1.95	Cold-7
25.8	27.82	50.0	55.0	98	112	137	137	1.42	2.22	Cold-7
25.8	27.82	50.0	54.4	95	112	110	110	1.42	1.95	Cold-7

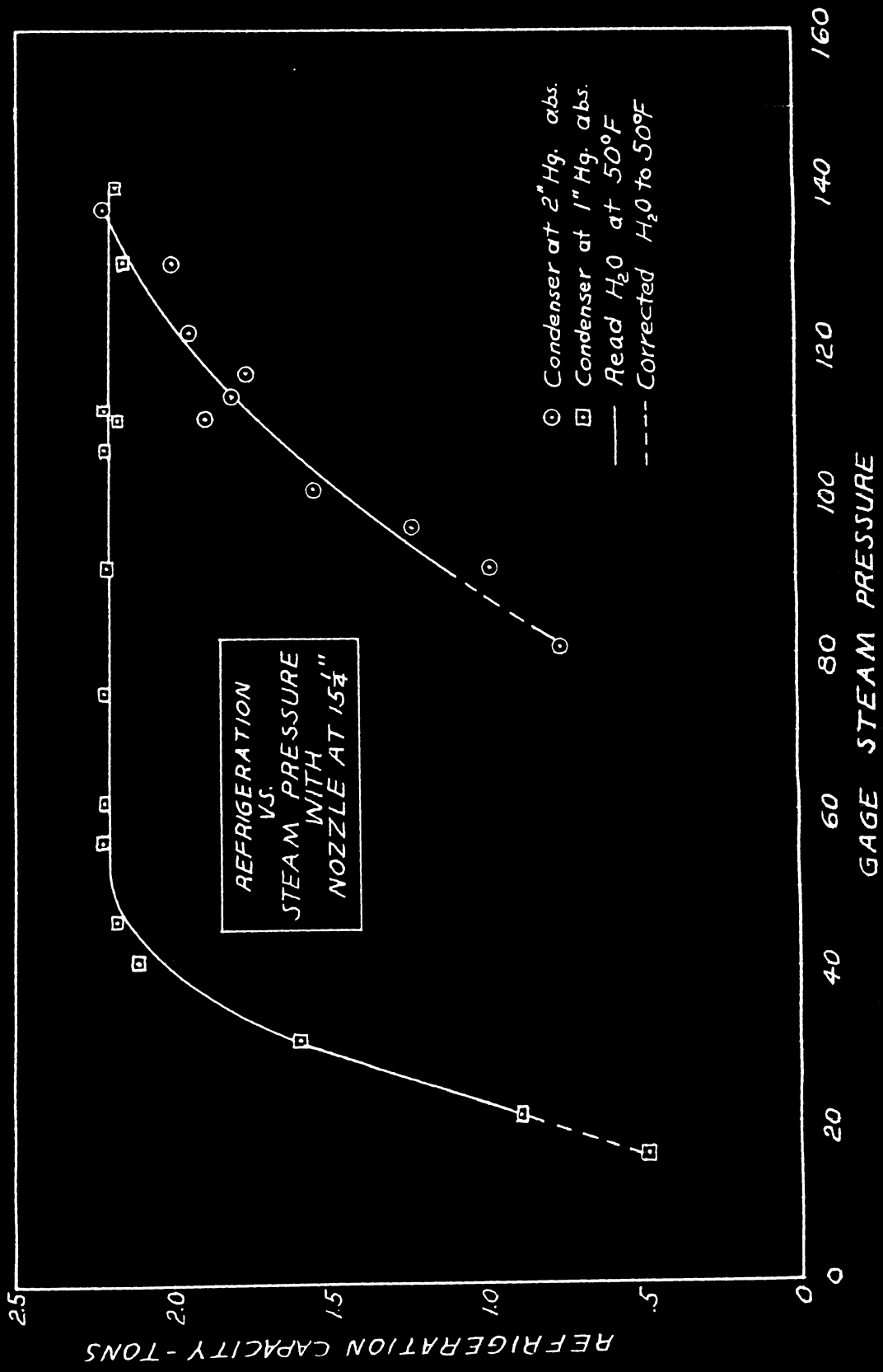
Table 4

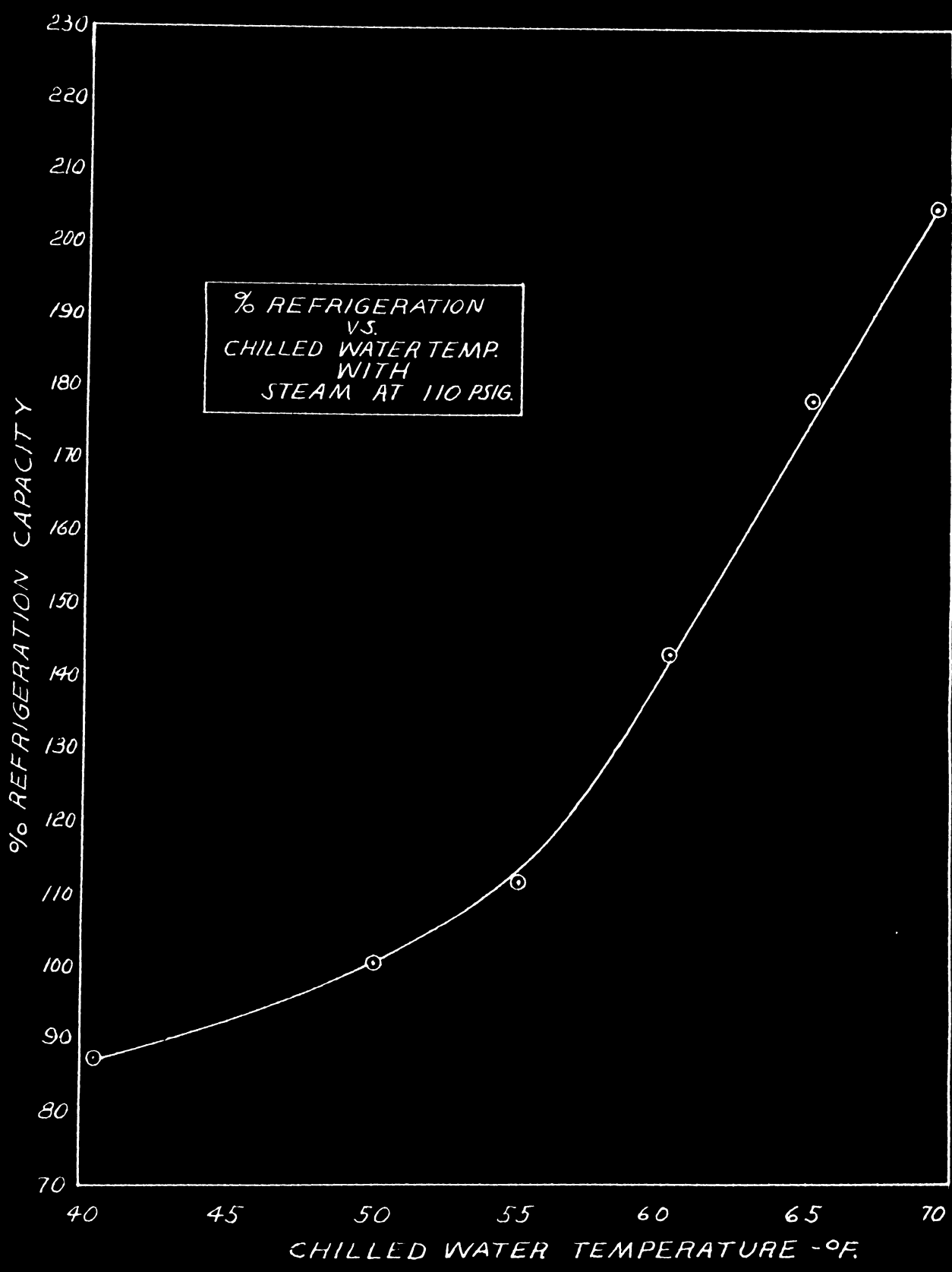
Varying Chilled-Water Temperatures

Cond. Vacuum In. Hg.	Barn. Rdg. In. Hg.	Temp. H ₂ O out °F	Temp. H ₂ O in °F	Temp. Room °F	Temp. Flange °F	Stm. Press. psig	Corr. Stm. Press. psig	Quant. Flow cfs	R.E. @ 50° ton	Comments
Nozzle Position 15½ inches, Back Pressure 2 inches Hg.										
25.82	27.82	40	43.8	95	108	110	110	1.42	1.68	Cold-7
25.82	27.82	50	54.4	95	112	110	110	1.42	1.98	Cold-7
25.82	27.82	55	60.0	94	113	110	110	1.42	2.22	Cold-7
25.82	27.82	60.7	67.0	94	114	110	110	1.42	2.79	Cold-7
25.82	27.82	65.4	73.2	94	114	110	110	1.42	3.46	Cool-3
25.82	27.82	70.9	80.0	94	112	110	110	1.42	4.05	Cool-3
25.82	27.82	75.0	86.0	94	104	110	110	1.42	4.87	Cool-3









AWH. C.W.V.

FIG. 5

VII

DISCUSSION

This unit was designed to produce five tons of refrigeration, but the maximum obtained during the tests at the designed conditions was 1.95 tons. This impairment of the performance could be attributed to many factors, and much time was spent in attempting to determine the factors responsible for the decreased performance of the unit. It is believed, however, that the assumed diffuser efficiency of seventy-five per cent was too high for a unit of this size and construction.

A boring bar of twenty-four inches was necessary in machining the inside surface of the diffuser and the preferred smoothness of the surface was unattainable due to the vibration of the cutting tool. The manufactured diffusers for which these recommended efficiencies apply, undoubtedly have a much smoother surface than could be obtained by inexperienced machinists in the laboratory shop. As the velocity at the throat section was 1555 feet per second, the roughness of the surface tended to increase the internal friction, which should have been accounted for by the assumption of a lower diffuser efficiency.

It seems logical that the efficiency of a small unit should be less than that of larger units, since the ratio of the circumference to the cross sectional area is increased as the size decreases. The recommendations of the various companies probably are made with reference to units larger than the one built, because it was found from manufacturer's catalogues that units of less than 20-ton capacity are seldom constructed.

Proceeding on this basis, calculations were made following the same

design procedure, but with different efficiencies assumed, to determine to what extent the capacity would be affected by the decrease in diffuser efficiency. The quality of the steam was discovered by means of the throttling calorimeter to be ninety-five per cent. Since the quality was assumed to be ninety-nine and five-tenths per cent for the design, new calculations were made for ninety-five per cent quality steam. The following table shows the results of these calculations.

Diffuser Efficiency	Entrainment Efficiency	Steam Quality	<u>lb Vapor</u> <u>lb Steam</u>	Vapor Quality	Tons Capacity
75%	65%	95%	1.63	97%	4.78
75%	65%	95%	1.58	95%	4.62
70%	65%	95%	1.91	95%	3.95
65%	65%	95%	2.375	95%	3.182
65%	60%	95%	3.32	95%	2.28
75%	60%	95%	2.00	95%	3.70

These assumed diffuser efficiencies are not unusually low since values as low as sixty per cent were recommended in the Heating, Ventilating and Air Conditioning Guide (2).

The temperatures which were measured at the flange of the nipple connecting the diffuser to the condenser indicated that the loss in this section was appreciable, since the pressures corresponding to the temperatures read were in all cases, higher than the ones in the condenser. The temperature

of the two flanges for the nipple were at a higher temperature than the pipe on either side of the flanges. This is believed to have been due to the increased turbulence at the flange joints.

It is possible that more heat was gained from the atmosphere than was accounted for by the ten BTU per ton per minute introduced in the design to account for these losses, but calculations of a heat balance on the unit showed that this assumption was reasonable. The amount of vapor evaporated was determined by calculating the volume of water removed from the tank for a given period of time. The internal load thus determined checked very closely with the external load.

The water was not broken up into as fine particles by the spray pipe as were expected, so a set of six Buffalo spray nozzles was used so that the effect of a better spray could be determined. The unit was also operated with the water discharged into the tank without a spray nozzle. The difference in the results for these three spray conditions was negligible.

Although the data and curves for the varying nozzle positions at ninety-one psig are included to show the general shape of the curves, the data of most significance was that obtained at the normal operating conditions of one hundred and ten psig steam pressure, and two inches back pressure. In the curves (Figures 2 and 3) it is shown that by slight changes in the nozzle position, the performance of the unit was noticeably affected. The most effective position seemed to be fifteen and one quarter inches, which was very close to the calculated position of fifteen inches. By moving the nozzle a fraction of an inch in either direction, an appreciable

difference in the results was observed. When the nozzle position was fifteen and one quarter inches, and the steam pressure one hundred and one psig, capacity was 1.95 tons, but by moving the nozzle a quarter of an inch either way, the capacity was reduced by approximately two-thirds. According to Watson's experiments maximum results were obtained at only one position of the nozzle. Since the calculated nozzle position was fifteen inches, these results conform with the statements made by manufacturers that the nozzle position is determined experimentally after the units are built.

Changing the steam pressure with a two-inch condenser pressure had a definite affect upon the performance (Figure 4). The capacity was increased from about 1.8 tons at one hundred and ten psig to 2.25 tons at one hundred and forty psig the highest pressure obtainable. When the pressure was below one hundred and ten psig, the performance was greatly affected, and at eighty psig, the capacity was reduced to three quarters of a ton. Thus, the performance of the unit was definitely impaired by lowering the steam pressure below designed conditions, and a small increase in capacity was effected by raising the steam pressure.

The lower the chilled water temperature, the less was the refrigeration (Figure 5), because there is a perceptible change in the specific volume of steam vapor at this low pressure and low temperature range. For instance, the specific volume for saturated steam at fifty degrees Fahrenheit is 1703 cubic feet per pound, and is only 867 cubic feet per pound at seventy degrees Fahrenheit. As the temperature increased, therefore, the weight of the vapor removed was correspondingly increased, although the volume removed

might have been changed very little.

If the temperature was decreased to thirty-two degrees, the water in the tank was slowly frozen as the latent heat of fusion was removed. This freezing was not instantaneous due to the constant movement of the water, and the installation was not endangered. Freezing seemed to begin at the surface of the water, and was continued downward as the heat of fusion was removed. Circulation was not stopped because holes were melted in the ice by the spray, and water continued to be circulated.

For some of the tests, a chilled-water temperature of fifty degrees could not be attained because there was always some heat picked up in passing through the heat exchanger and the water pipes. In these cases, the equivalent refrigeration effect at fifty degrees was obtained by use of the curve for refrigeration capacity versus chilled-water temperatures (Figure 5).

Although it was not the primary object of the investigation to determine the effects of varying back pressures, tests were run at one inch back pressure for steam pressures and for nozzle positions. The capacity of the unit at one inch was not visibly altered (Figure 4) by changing the steam pressure from sixty psig to one hundred and forty psig, but was definitely impaired when the pressure fell below forty psig. Therefore, the most efficient range of steam pressures with one inch back pressure seemed to be from forty psig to sixty psig. For this pressure range, the coldest spot on the diffuser was at the entrance end, which indicated that compression was occurring near the diffuser inlet. The coldest spot moved toward the condenser as the pressure was increased, and at one hundred and ten psig, it

was about eight inches beyond the throat on the diverging end. When the back pressure was decreased, the amount of energy required for compression was also reduced. Therefore, the vapors tended to be forced by the kinetic energy of the steam beyond the throat section before they could be compressed. At one hundred and forty psig the capacity with a two inch back pressure was nearly equal to the capacity at one inch back pressure. At this higher pressure range, the performance was not increased by decreasing the back pressure to one inch, because the compression was not occurring at the inlet to the diffuser. It is possible, however, that when the back pressure is changed, the most effective nozzle position is not fifteen and one quarter inches. This fact was indicated by the data taken when the nozzle was thirteen and one quarter inches, in which position there was a considerable increase in the capacity, and compression was occurring at the throat as indicated by the temperature of the diffuser.

VIII

SUMMARY AND CONCLUSIONS

It was decided at the beginning of the investigation to build a five ton unit instead of a larger one because in the construction of the five ton unit, the Alberger condenser installed in the laboratory could be utilized. This decision was also based on the fact that the time and expense of constructing a large unit were prohibitive. The construction was carried out with the idea of using, as much as possible, equipment and material available in the laboratory. The time spent in the actual erection of the unit was negligible compared to the time consumed in obtaining and fabricating the various pieces of the unit.

The object of the investigation prior to the preliminary tests was to obtain data for various steam pressures at different nozzle positions. This plan was abandoned, however, when it was ascertained that the unit did not operate at the design conditions. Consequently, tests were made to secure information for the correction of the design.

The tests performed on the constructed steam-jet refrigeration unit did, by no means, give a complete set of performance data for the unit operating under all varying conditions, but from the data which was obtained, the authors believe the following statements were substantiated:

- a) The design as set forth in H.S. Miles' thesis is valid, and it can readily be applied to the design of a steam-jet unit, provided that the efficiencies chosen are appropriate to the particular size and type of the unit.
- b) For a designed steam pressure and back pressure, there is an operating position for the steam nozzle at which the

best results are obtained, and this position must be determined experimentally.

- c) With the nozzle in the most effective position, and with the designed back pressure, the refrigeration capacity is increased with an increase of steam pressure. The capacity is reduced perceptibly with a reduction of the steam pressure.
- d) When the unit is operating at the designed nozzle position, and if the back pressure is reduced, there should also be a reduction of the steam pressure for the most efficient operation of the unit.
- e) When a steam-jet is built, it should be operated at the conditions for which it was designed.

IX

RECOMMENDATIONS

As an extensive study of the operation of the unit was prevented by the time consumed in its design and construction, there are still many factors remaining to be investigated. A comprehensive study of various types of spray nozzles with different rates of flow should be made to find the combination that will produce the maximum refrigeration. An investigation should be conducted by changing the steam pressure, with the nozzle at various positions to determine whether or not there is an interaction.

It is estimated that, due to the low diffuser efficiency, a project would be benefactory whereby the interior surfaces would be very smoothly finished, and tests run that the performance might be checked.

In conjunction with this, it is suggested that a diffuser be cast of clear plastic, in order that the internal processes might be studied. Since the temperatures are low, there would be no danger to the plastic diffuser through overheating. The plastic diffuser could be of a construction differing from that used for the metal diffuser, in order that results might be obtained with diffusers of different dimensions.

In test runs with varying nozzle positions, a great amount of time was lost in changing the position of the nozzle, and test procedures would be expedited by the incorporation of a packing gland arrangement on the nozzle assembly. By means of this improvement, the nozzle position could be altered without the removal of the nozzle assembly.

It has been recommended by several manufacturers, that better entrainment is achieved with a nest of nozzles than with a single nozzle.

It is stated by Watson (25) that the success of the present forms of steam-jet refrigeration units are the outcome of extensive experimental investigation; it is believed that those who are not directly connected with a company developing these units, will be able to obtain this information by a continuation of this project.

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XI

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