UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION

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HYDRAULIC LABORATORY REPORT NO. 68

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LABORATORY INVESTIGATION OF ORIFICE DESIGNS FOR AIR-LIFT ICE PREVENTION SYSTEM OF THE GRAND COULEE DAM - COLUMBIA BASIN PROJECT, WASHINGTON

By

T. G. OWEN

- - -

Denver, Colorado
December 12, 1939

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UNITED STATES
DEPARTMENT OF THE INTERIOR
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Branch of Design and Construction
Engineering and Geological Control
and Research Division
Denver, Colorado
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Laboratory Report No. 68
Hydraulic Laboratory
Compiled by: T. G. Owen

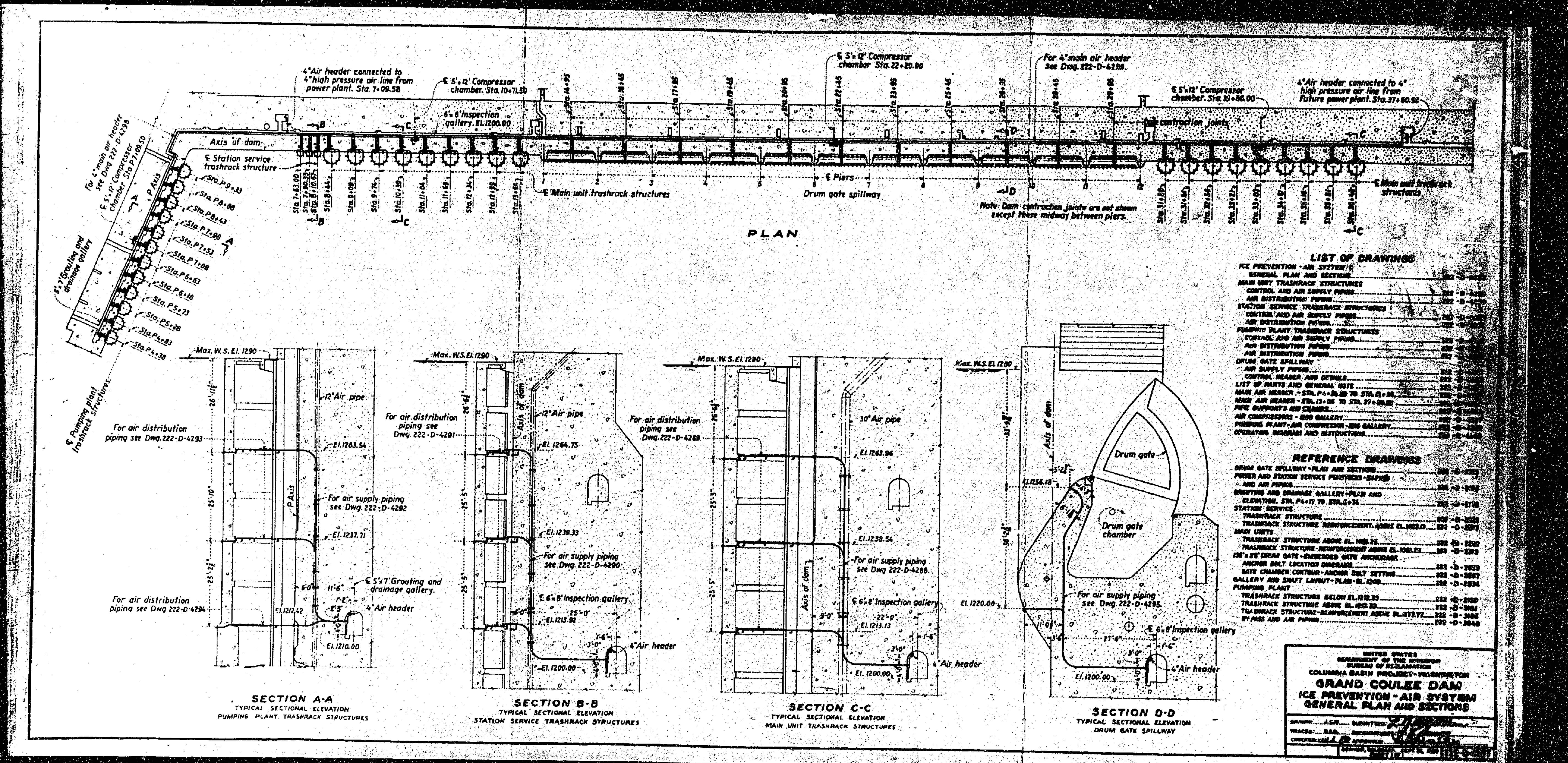
Reviewed by: J. E. Warnock

Subject: Laboratory investigation of orifice designs in connection with ice prevention by the air system for Grand Coulee Dam, Columbia Basin Project, Washington.

PRELIMINARY TESTS

1. Introduction. The formation of ice in front of the trashracks and drum gates of Grand Coulee Dam is both detrimental to good operation and dangerous to the structure. At times, the air temperature at the dam site has been known to drop to -28° F. and remain at that level for a period of several days. The problem confronting the designers was to maintain free of ice the reservoir surface immediately adjacent to the upstream face of the dam. Many methods of accomplishing this result were studied but discarded in favor of the air-lift system because of its simplicity. In the air-lift system compressed air is forced into the reservoir at a depth at which the water temperature is relatively high, and the stirring and mixing action of the rising air induces the upward flow of warm water which either melts the ice or prevents its formation.

The proposed location of the air-lift ice prevention or de-icing units for the Grand Coulee Dam is shown in figure 1. As shown also in that figure, provision is made to introduce air at three different elevations in front of the trashracks and at two separate levels in front of the drum gates. This feature allows air to be introduced at a depth equal to or greater than 10 feet


PLAN
4" Air header connected to 4" high pressure air line from power plant. Sta. 7+09.58
℄ 5'x 12' Compressor chamber. Sta. 10+71.50
6'x 8' Inspection gallery. El. 1200.00
℄ 5'x 12' Compressor chamber Sta. 22+20.80
For 4" main air header See Dwg. 222-D-4289.
℄ 5'x12' Compressor chamber. Sta. 33+86.00
4" Air header connected to 4" high pressure air line from future power plant. Sta. 37+80.50
Dam contraction joints
Axis of dam
℄ Station service trashrack structure
℄ Main unit trashrack structures
℄ Piers
Drum gate spillway
Note: Dam contraction joints are not shown except those midway between piers.
℄ Main unit trashrack structures
For 4" main air header see Dwg 222-D-4298
℄ 5'x 12' Compressor chamber Sta P7+09.50
5'x 7' Grouting and drainage gallery
℄ Pumping plant trashrack structures
P Axis
SECTION A-A
TYPICAL SECTIONAL ELEVATION
PUMPING PLANT TRASHRACK STRUCTURES
Max. W.S. El. 1290
12" Air pipe
For air distribution piping see Dwg. 222-D-4293
For air supply piping see Dwg. 222-D-4292
For air distribution piping see Dwg. 222-D-4294
℄ 5'x 7' Grouting and drainage gallery.
4" Air header
SECTION B-B
TYPICAL SECTIONAL ELEVATION
STATION SERVICE TRASHRACK STRUCTURES
For air distribution piping see Dwg 222-D-4291
For air supply piping see Dwg. 222-D-4290
℄ 6'x 8' Inspection gallery
SECTION C-C
TYPICAL SECTIONAL ELEVATION
MAIN UNIT TRASHRACK STRUCTURES
30" Air pipe
For air distribution piping see Dwg. 222-D-4289
For air supply piping see Dwg 222-D-4288.
SECTION D-D
TYPICAL SECTIONAL ELEVATION
DRUM GATE SPILLWAY
Drum gate
Drum gate chamber
For air supply piping see Dwg. 222-D-4295.
LIST OF DRAWINGS
REFERENCE DRAWINGS
UNITED STATES
DEPARTMENT OF THE INTERIOR
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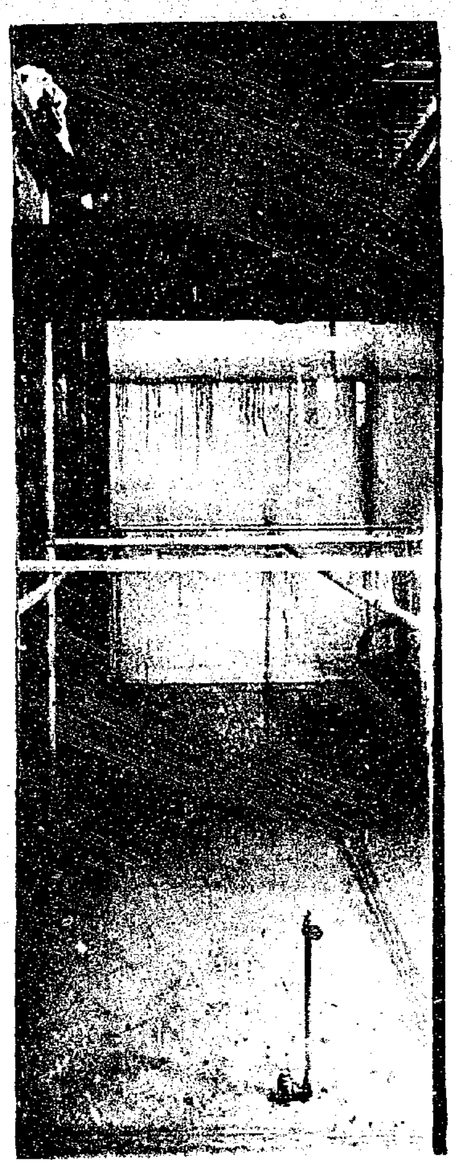
COLUMBIA BASIN PROJECT-WASHINGTON
GRAND COULEE DAM
ICE PREVENTION - AIR SYSTEM
GENERAL PLAN AND SECTIONS

(which has been found to be the minimum depth for best results), regardless of the reservoir elevation.

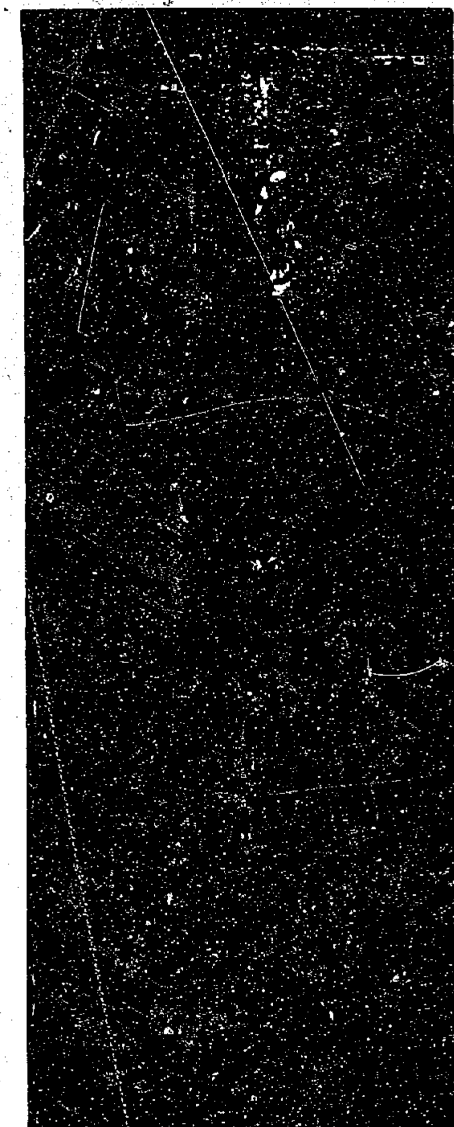
In the design of the de-icing units, many problems arose which could not be accurately answered without the aid of laboratory investigation. The problem of determining the best size, shape, and direction of discharge of the orifice was one of these. Also, the question arose concerning whether or not the cooling due to expansion of the air through the orifice would cause ice to form in the orifice. The laboratory investigation was divided into two parts: First, a preliminary cursory examination of several orifice designs; and second, a detailed research concerning the cooling effect due to expansion.

2. <u>Apparatus.</u> In the preliminary tests the equipment consisted of a glass-sided tank, near the bottom of which was located the orifice to be tested. The orifice was connected with the high-pressure air line and a throttling valve was placed in the line to control the pressure. By means of two mercury U-tubes the air-line pressure at the orifice and the static water pressure at the orifice elevation were measured. The difference between these two observed pressures gave the differential pressure across the orifice. Figure 2-A shows a typical set-up with the orifice placed to discharge horizontally and toward the observer.

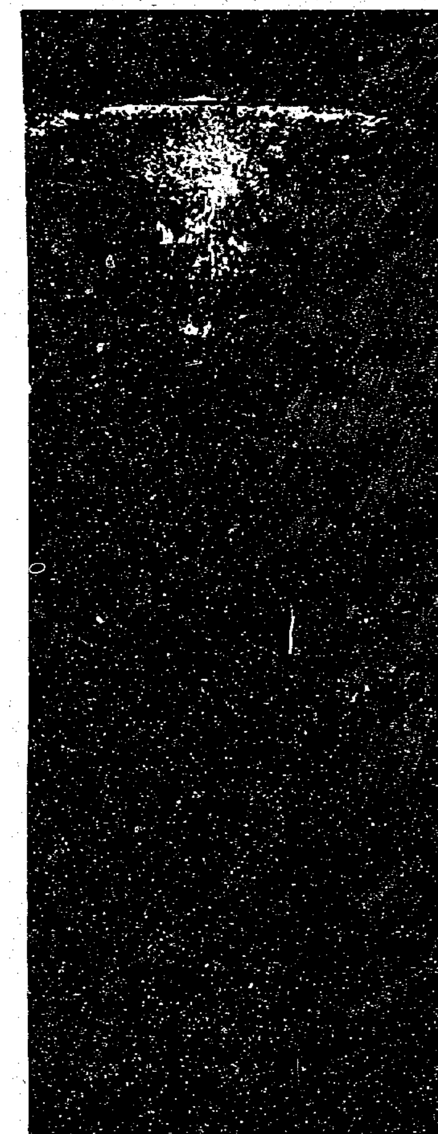
3. <u>Procedure.</u> In the experiments on orifices A, B, and C, the tests were purely visual. The flow patterns at three pounds per square inch differential pressure for horizontal discharge of



A. Typical Test Installation

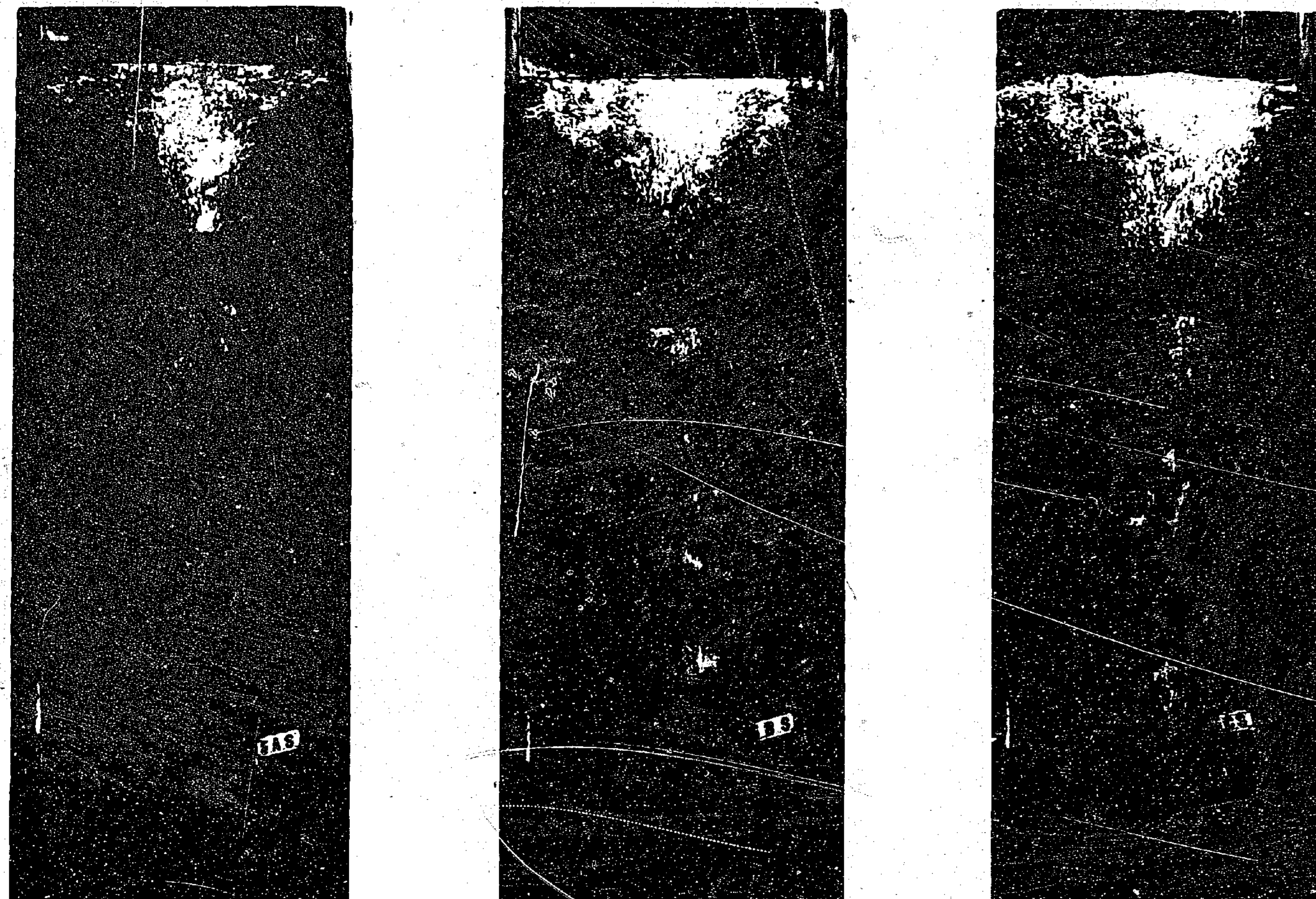


B. Orifice A (Area 0.0052 sq. in.)



C. Orifice B (Area 0.0106 sq. in.)

TYPICAL SET-UP AND HORIZONTAL-DISCHARGE FLOW PATTERNS AT 3 LB. PER SQ. IN. DIFFERENTIAL PRESSURE

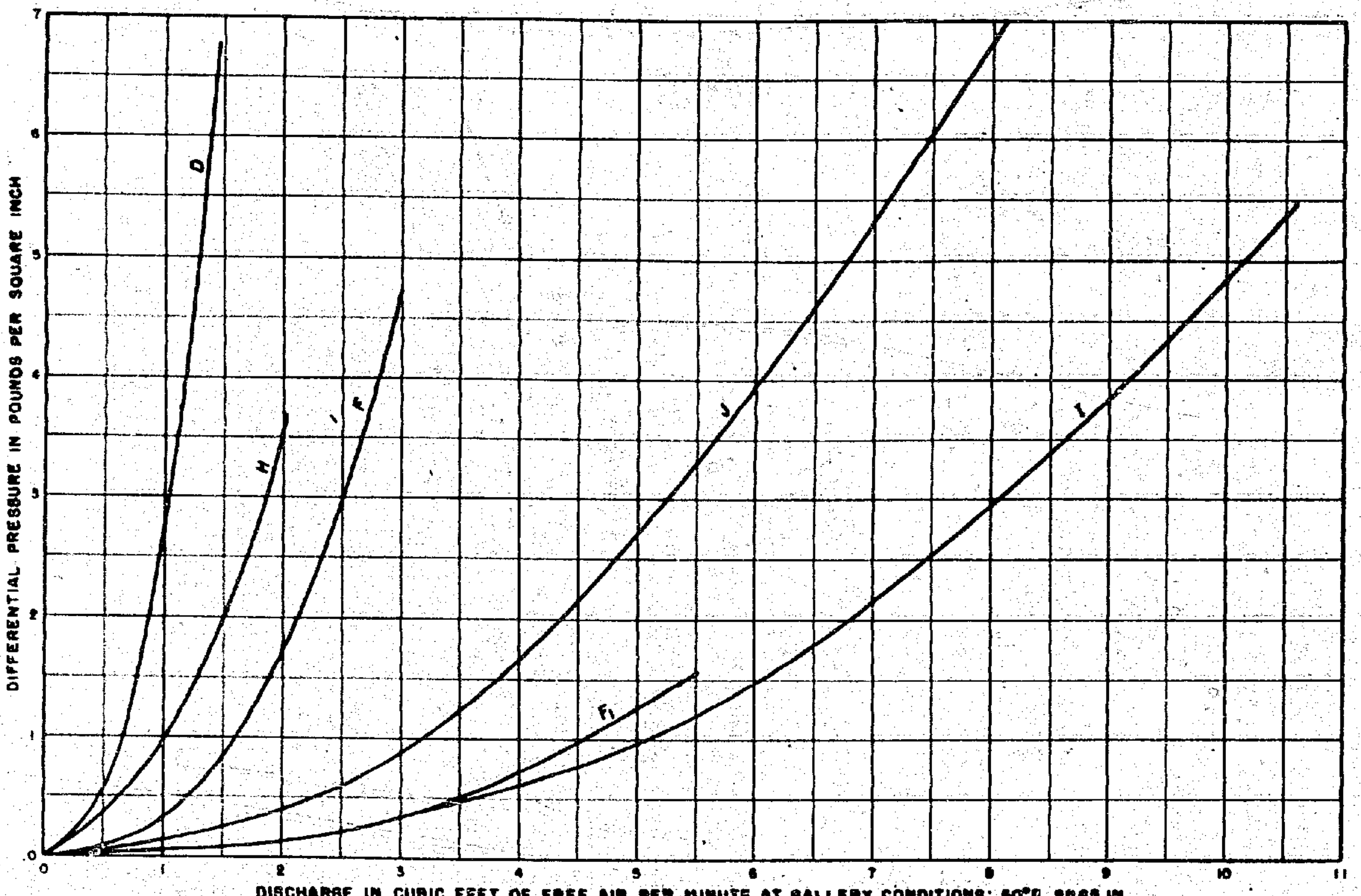


A. Orifice A (Area 0.0052 sq. in.)

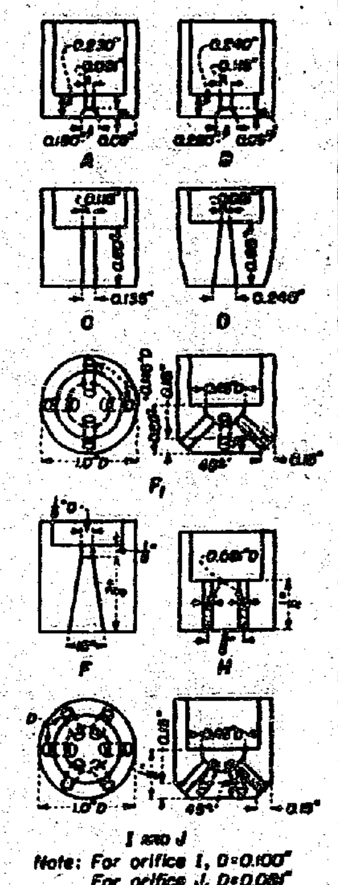
B. Orifice B (Area 0.0106 sq. in.)

C. Orifice C (Area 0.0106 sq. in.)

VERTICAL-DISCHARGE FLOW PATTERNS AT 3 LB. PER SQ. IN. DIFFERENTIAL PRESSURE



A. PRESSURE-DISCHARGE CURVES BASED ON AVERAGE STATIC WATER HEAD ON ORIFICE OF 1.987 POUNDS PER SQUARE INCH AND AVERAGE LABORATORY BAROMETER OF 12.142 POUNDS PER SQUARE INCH



B. ORIFICE DETAILS

GRAND COULEE DE-ICING EXPERIMENTS

PRESSURE-DISCHARGE CURVES AND ORIFICE DETAILS

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orifices A and B are shown in figures 2-A and -B. The flow patterns with discharge vertically downward and at three pounds per square inch differential pressure for orifices A, B, and C are shown in figure 3, and the general details of these orifices are shown in figure 4-B. From these tests it was determined that discharge directed vertically downward gave the best flow pattern, the criterion being the largest cross-sectional area of the rising air current and the fineness of division of air bubbles. It was observed further that an orifice of type C (figure 4-B) gave a better pattern than type B, which had the same cross-sectional area.

After these initial tests, a gas meter of the displacement type was installed in order to obtain the discharge characteristics of the various orifices. Orifice D (figure 4-B) was installed and its flow pattern was found to be good but its discharge was too low (figure 4-A) to establish a strong upward water current. Orifice F was then tested and found to give a good flow pattern and also a good circulation of water. The orifices thus far tested had only one hole. The multiple-hole nozzles were then examined. This group embraces orifices F_1, H, I, and J (figure 4-B). Nozzle H did not give a better flow pattern than F, and the water current established was inferior to that of F. Nozzles F_1, I, and J all gave excellent flow patterns and induced strong water currents but the discharge of each was so high that an excessive size of air compressor would have been required with any one of these three nozzles. These three nozzles had the further disadvantage of having holes of smaller diameter than orifice F, which fact greatly increased the possibility of the holes becoming plugged or fouled by foreign matter.

4. Preliminary conclusions. From these preliminary tests the following conclusions were drawn:

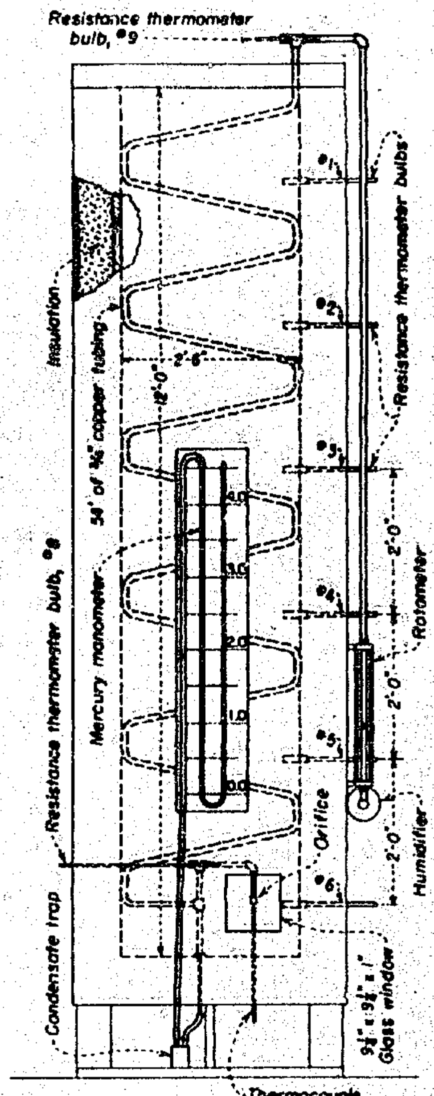
(a) The direction of discharge vertically downward gave a superior flow pattern.

(b) An orifice with an 18-degree tapered exit similar to type F (figure 4-B) gave the best flow pattern of all single-hole types tested.

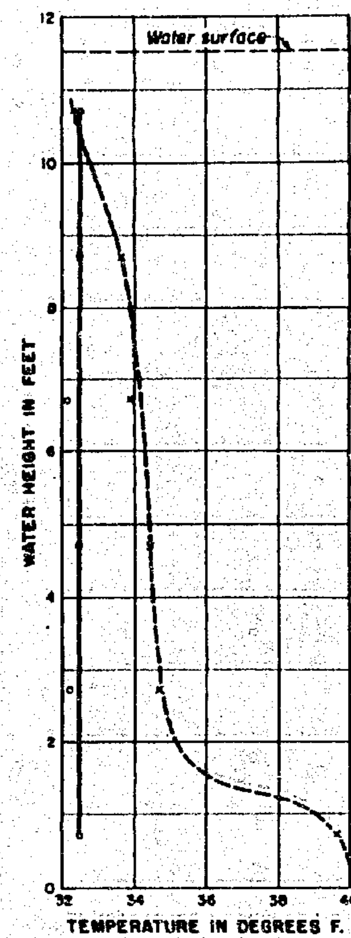
(c) A discharge of two cubic feet of free air per minute at a differential pressure of two pounds per square inch is sufficient to establish a strong upward water current. This value for discharge also is compatible with practical limits of the compressor size required for the Grand Coulee air de-icing system.

DETAILED INVESTIGATION

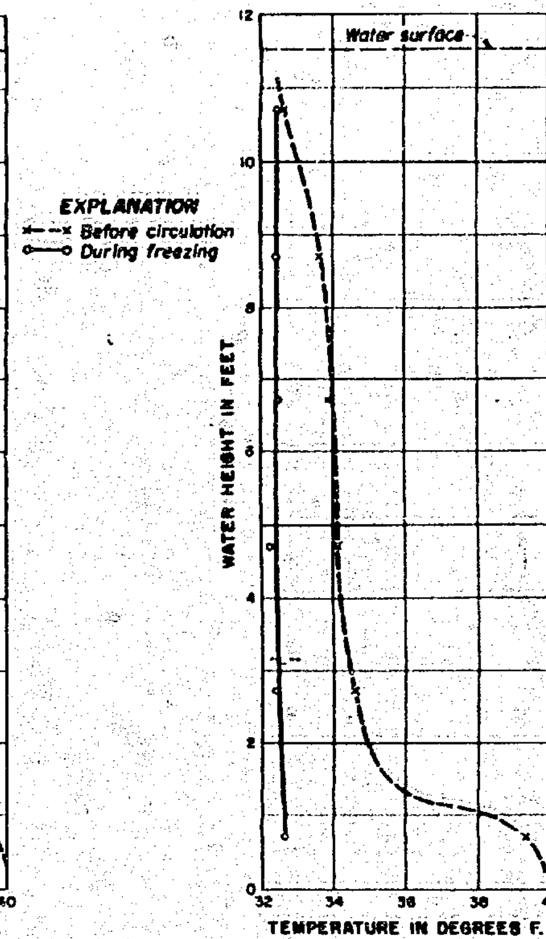
5. Equipment and procedure. For these tests a special tank was constructed and equipped with instruments especially adaptable for the investigation. The tank was made 12 feet deep so that at least 10 feet of static water head could be imposed on the orifice. Inside the tank about 54 feet of 3/4-inch copper pipe were placed in coils and at the end of which was located the orifice to be tested. In one side of the tank and at the elevation of the orifice a piezometer opening and a resistance thermometer bulb were located. The function of the piezometer opening was to measure the static water pressure at the orifice elevation; the purpose of the



A. DETAILS OF DE-ICING EQUIPMENT



B. TEMPERATURE GRADIENT CURVES, TEST: GIC-11



C. TEMPERATURE GRADIENT CURVES, TEST: GIC-12

GRAND COULEE DE-ICING EXPERIMENTS

DETAILS OF APPARATUS AND TEMPERATURE GRADIENT CURVES

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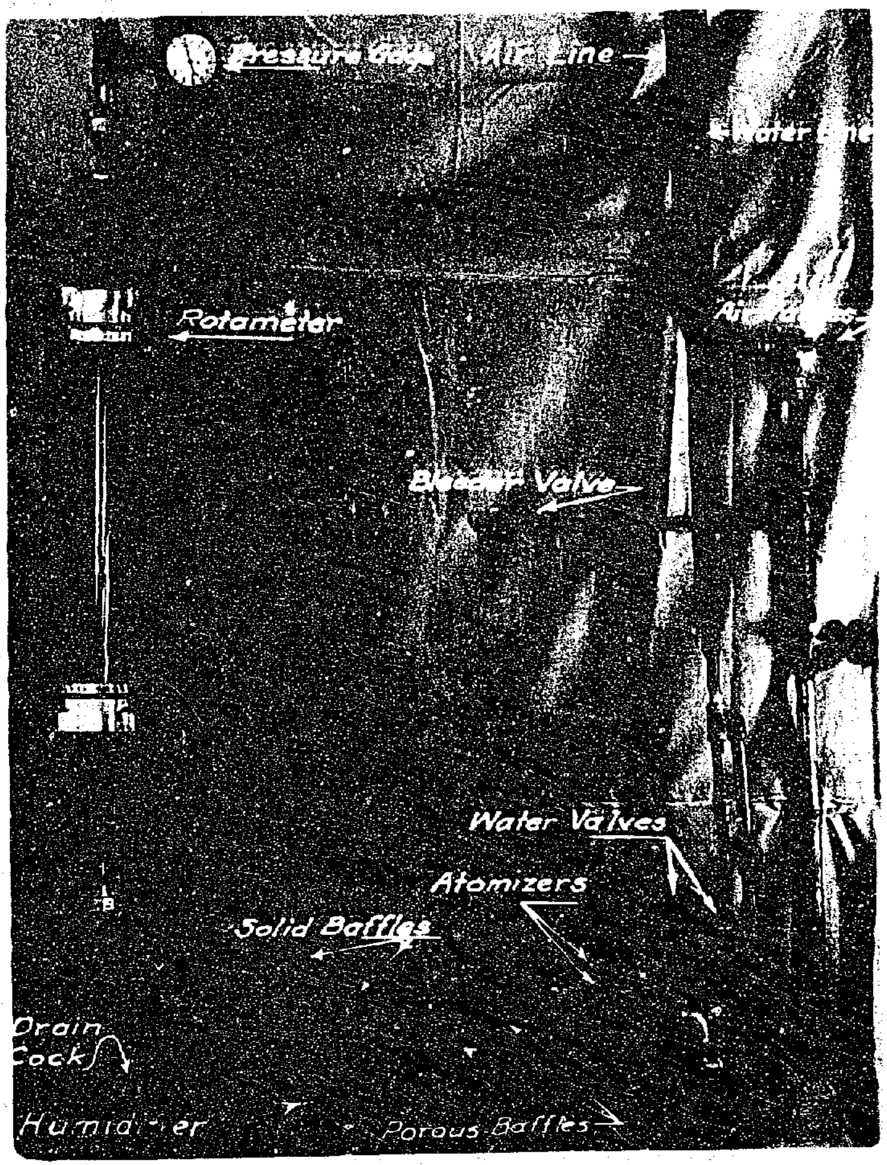
resistance thermometer bulb was to determine the temperature of the water at that level. Located in the same side of the tank at two-foot intervals vertically above the thermometer bulb at the orifice level were five other thermometer bulbs, the purpose of which was to determine the temperature gradient during testing. These details are shown in figure 5-A. The resistance thermometers were connected through a selector switch to a wheatstone bridge and the resistance balance was indicated by a very sensitive light-beam galvanometer. The temperature of the air before entering the copper coils was measured by resistance thermometer No. 9, and the air temperature before expansion through the orifice, by thermometer bulb No. 8. In the first experiments, another resistance thermometer bulb about 1/4 inch in diameter and 3 inches long was so placed that it would be directly in the discharging air stream. It was found, however, that due to the oscillation of the air jet, this thermometer bulb was not continuously surrounded by air alone, so the bulb was re-placed by a thermocouple which, it was hoped, could be placed near enough to the orifice that it would be constantly in the issuing air stream. The thermocouple was connected through a selector switch to the galvanometer and enough resistance introduced in the circuit such that one division on the galvanometer scale was approximately equivalent to one degree difference in temperature between the hot and cold junctions of the thermocouple. The temperature of the "hot" junction of the thermocouple was known since it was placed in a vacuum bottle filled with a mixture of cracked ice and water. The

FIGURE 6



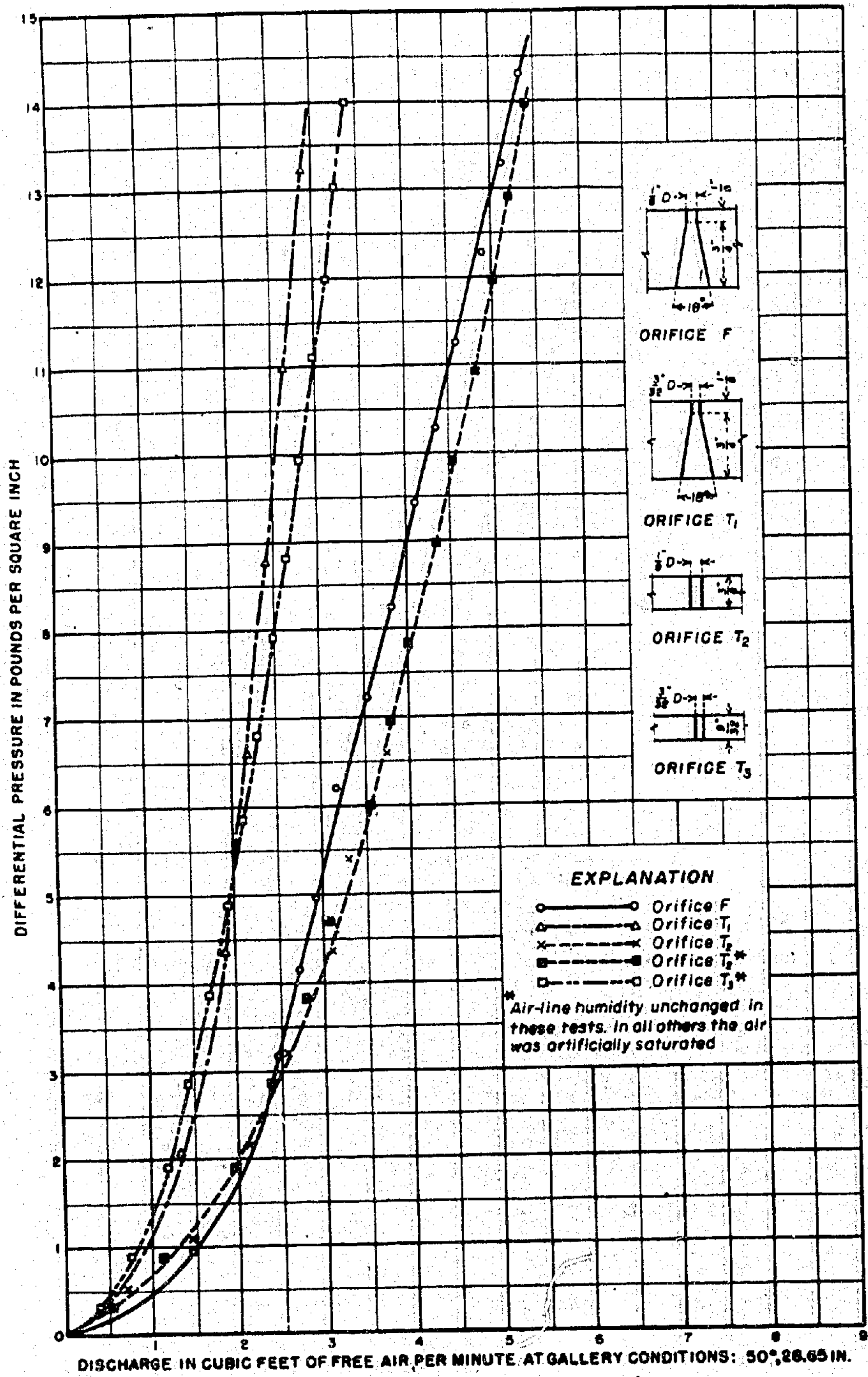
A. Humidifier and Air Rotameter Assembly

B. Air Rotameter

HUMIDIFYING AND AIR-METERING APPARATUS

cold junction was located immediately below the orifice. Because of the oscillation of the air jet and the vibration of the thermocouple, it was felt that the indicated temperatures were unreliable and for that reason are not included in this report. Insulating material was placed between the inner and outer tank walls, separated by 2- by 6-inch studding. The bottom of the tank was insulated and the top was equipped with a close-fitting, removable cover, also insulated. The air pressure was controlled by a system of three valves (figure 6-A) so arranged that variations in the supply-line pressure would produce the least change in pressure at the orifice. The humidity of the air from the supply line was increased by passing the air through the humidifier, which consisted of two atomizers which sprayed water on two porous baffles. Mechanically entrained water was removed by a system of three solid baffles (figure 6-A). From the humidifier the air passed through the air rotameter, a device for measuring rate of flow. The air rotameter (figure 6-B) consists of an accurately machined glass tube tapered to increase in bore from bottom to top. A spinning metal rotor floats in the air stream and its position, read on a scale on the glass tube, indicates the amount of air flowing. (The correction for pressure and temperature will be subsequently discussed.) Just above the rotameter, a Bourdon type pressure gage was located. For more accurate air pressure measurements a mercury manometer was used. For visual observation and photographic recording, two 9-1/4- by 9-1/4-inch glass windows were located on opposite sides of the tank at the orifice level.

FIGURE 7



GRAND COULEE DE-ICING EXPERIMENTS
PRESSURE-DISCHARGE CURVES

Before starting a test, ice was put in the water of the tank. The usual charge of ice was about 1,200 pounds, introduced in small pieces of approximately 25 pounds each. After allowing the water to be cooled, the test was started. The initial temperature gradient as indicated by thermometer bulbs Nos. 1 to 6, inclusive (figure 5-A), was recorded, and figures 5-B and -C show typical temperature gradient curves. The static water head on the orifice was recorded and then the air turned on. Water and air temperatures, air pressure, and the quantity of air discharged were observed. The differential pressure across the orifice was progressively increased by increments of about one pound per square inch until the orifice froze or until the pressure limit of the apparatus was attained.

6. Orifice with 18° taper, 1/8-inch diameter (F). The preliminary tests indicated that orifice F had the most desirable properties of all others tested. In the new set-up a check calibration was made on this orifice, and the pressure-discharge curve plotted (figure 7). Tests on the cooling effect due to expansion of the air indicated that the orifice would freeze solidly, forming a cone of ice in the tapered exit when the air temperature before expansion was 32.38° F. and the differential pressure was 5.405 pounds per square inch. The results of freezing tests on all orifices are shown in table I.

7. Orifice with 18° taper, 3/32-inch diameter (T_1). As an alternative design, orifice T_1 was built. It is the same in every detail as orifice F except that its initial diameter is 3/32 of an

TABLE I

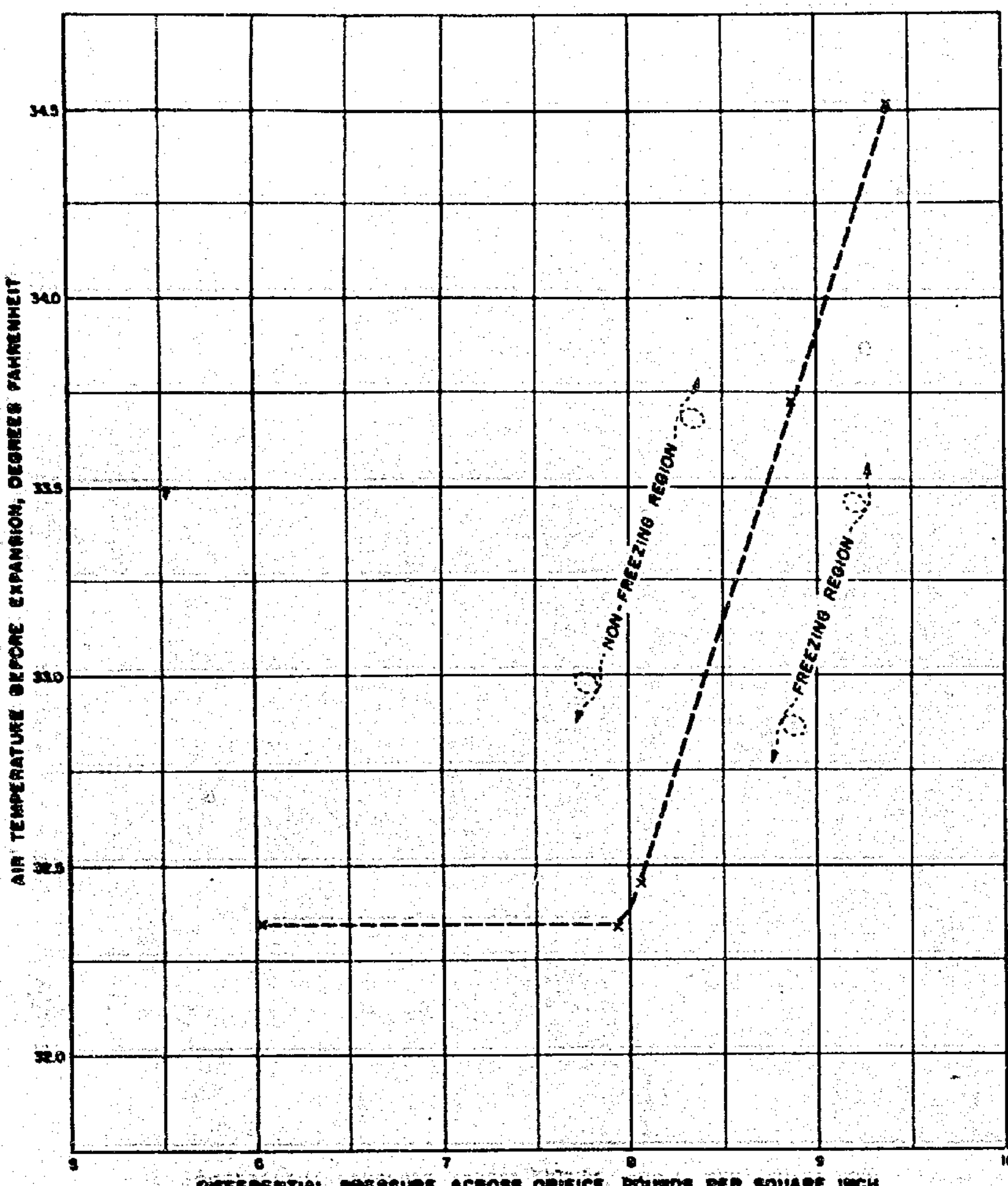
OBSERVED CONDITIONS UNDER WHICH ORIFICE FROZE

1	2	3	4	5	6
		Differential pressure across orifice	Air-line temperature, deg. F.		Temperature of water surrounding orifice
Orifice	Test No.	lb. per sq. in.	Before entering cooling coils (bulb No. 9)	Before expansion (bulb No. 8)	(bulb No. 6)
F	GIC-4	5.405	72.36	32.38	32.29
T_3	GIC-8	5.877	72.04	32.59	32.44
T_2	GIC-10	6.012	74.40	32.34	32.48
T_2	GIC-11	7.928	74.36	32.34	32.49
T_2	GIC-12	8.051	74.33	32.45	32.65
T_2	GIC-12	9.018	74.94	32.80	32.76
T_2	GIC-13	8.865	71.63	33.72	33.64
T_2	GIC-13	9.867	71.91	33.71	33.62
T_2	GIC-13	9.389	72.52	34.51	33.96
T_2	GIC-13	9.507	72.94	34.25	34.13
T_2	GIC-13	9.507	165.11	34.29	34.15
T_2	GIC-13	11.022	73.16	34.49	34.38

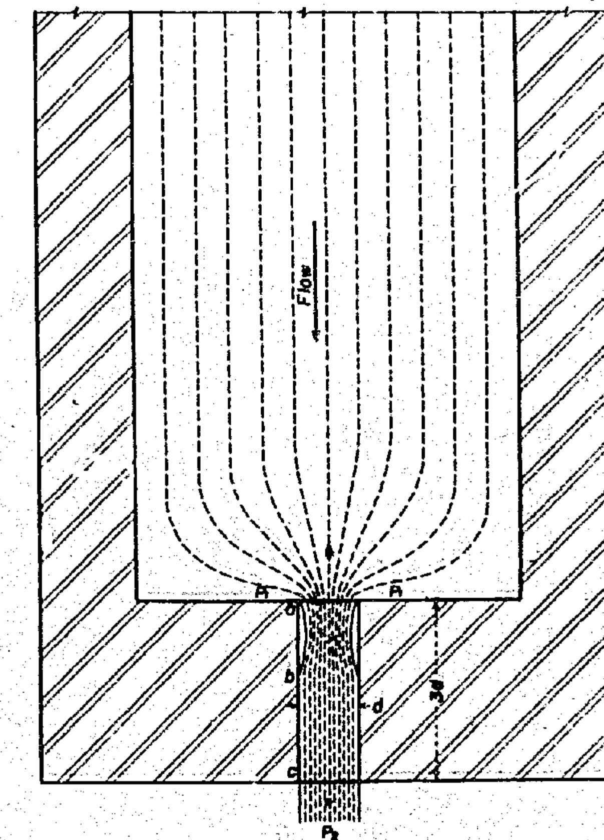
inch instead of 1/8 inch, which makes its area roughly half that of orifice F. The calibration curve for this orifice is shown in figure 7. Temperature tests were made but the range of air temperature before expansion was too high to cause freezing. The highest differential pressure during the test was 17.54 pounds per square inch, and the air temperature before expansion was 35.39° F. No further temperature tests were made on this orifice because it was decided to try an orifice of the short-tube type.

8. Orifice of short-tube type, 1/8-inch diameter (T_2). This orifice was built and tested because it was believed it would have a lesser tendency to cause plugging due to ice formation than an orifice having a tapered exit, such as for T_1. The general dimensions and calibration curve are shown in figure 7. The results of the temperature investigation are shown in table I, and from these data a "critical freezing curve" was determined for this orifice by plotting air temperature before expansion against differential pressure. This curve is shown in figure 8-A. The curve passes through the points which lie furthermost to the left and upwards. Thus, it divides the temperature-pressure plane into two regions: one region lying upward and to the left of the curve, in which no combination of initial air temperature and differential pressure will cause freezing; the other region lying downward and to the right of the curve, in which all combinations of initial temperature and pressure result in ice formation in the orifice. Figure 9 shows orifice T_2 during freezing when the initial air temperature was 32.37° F. and the dif-

Figure 6



A. CRITICAL FREEZING CURVE FOR ORIFICE "T2"



B. SECTION ON ℄ OF ORIFICE "T2"

GRAND COULEE DE-ICING EXPERIMENTS
CRITICAL FREEZING CURVE AND FLOW DIAGRAM FOR ORIFICE "T2"

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288-D-276

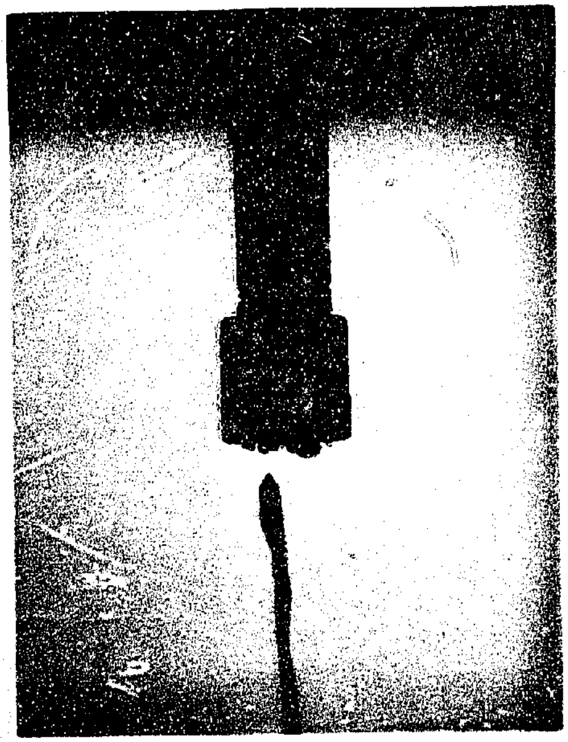
A. Unobstructed Flow Before Ice Started Forming

B. Flow Impeded by Ice Formation

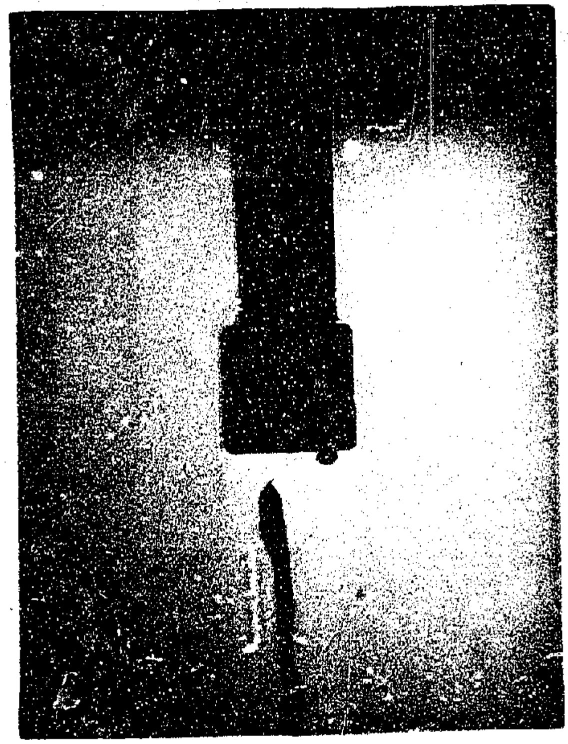
C. Jet Deflected Due to Ice Formation

D. Deflected and Restricted Flow Due to Ice Formation

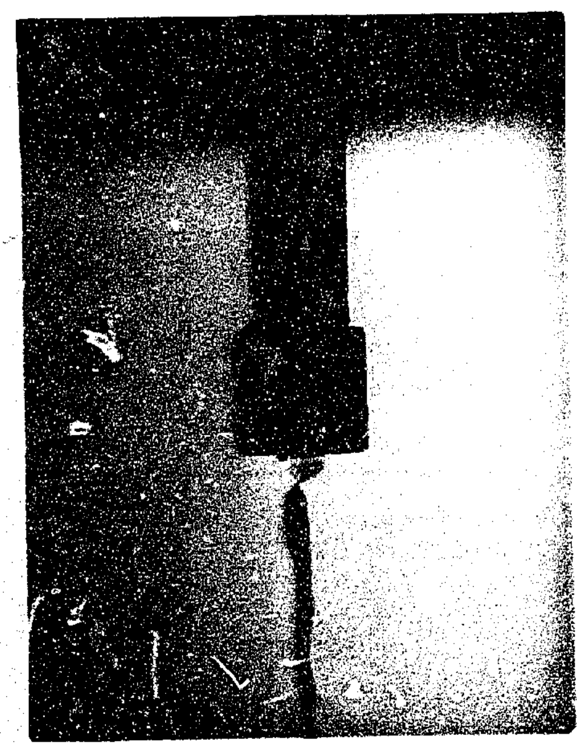
FLOW CONDITIONS DURING FREEZING OF ORIFICE "T_2". DIFFERENTIAL PRESSURE 19.25 LB. PER SQ. IN. INITIAL TEMPERATURE 32.37° F.



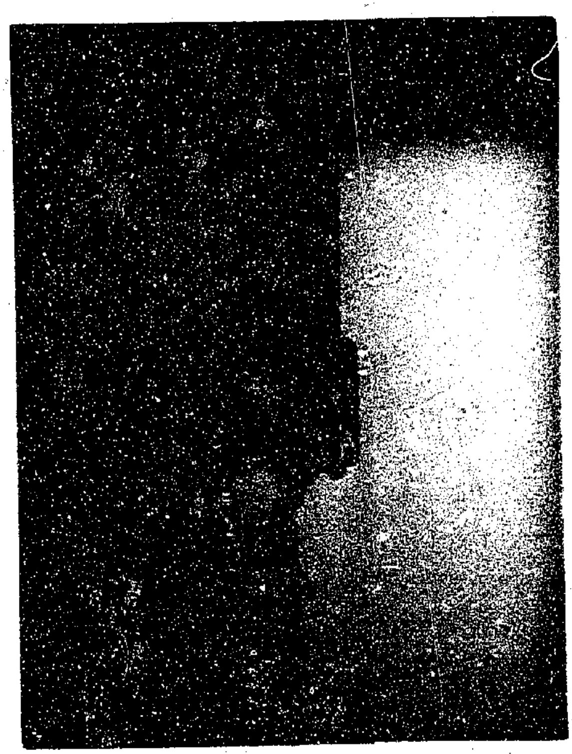
A. Orifice Frozen, Ice Cone Barely Visible



B. Orifice Frozen, Ice Cone More Pronounced



C. Orifice Frozen, Ice Cone is Larger



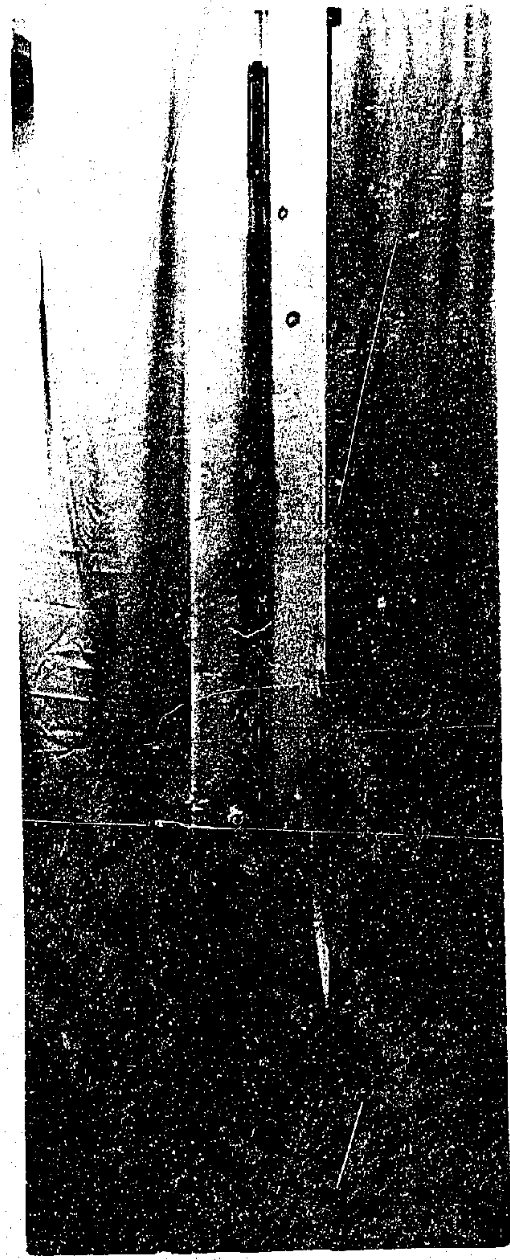
D. Orifice Frozen, Large Dense Cone Resulting from Longer Run

ICE CONES ON ORIFICE "T_2". DIFFERENTIAL PRESSURE 19.25 LB. PER SQ. IN. INITIAL AIR TEMPERATURE 32.37° F.

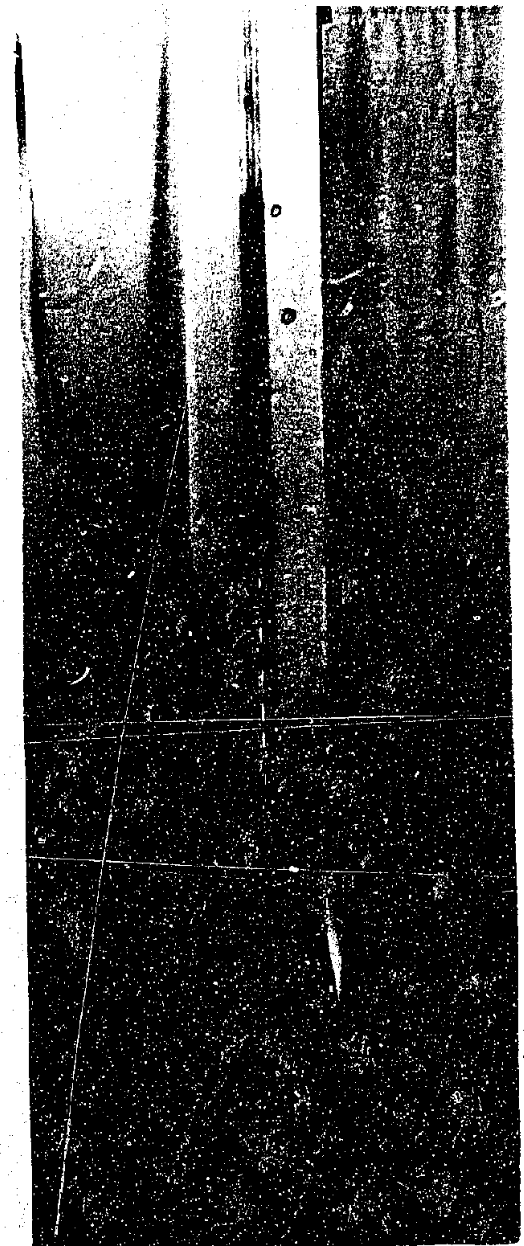
ferential pressure was 19.25 pounds per square inch. In figure 10 are shown four ice cones which were formed below orifice T_2 under the above conditions of pressure and temperature. They vary in size due, in part, to the length of time the formation process was allowed to take place, and, in part, due to the way in which ice formation first started.

9. Orifice of short-tube type, 3/32-inch diameter (T_3). Orifice T_3 is a short tube similar to T_2, having its length equal to three diameters (figure 7). Its area, however, is approximately half that of T_2. The calibration curve for this orifice is shown in figure 7 and its freezing characteristics shown in table I. At this point in the test program it was decided to adopt orifice T_2 so no further temperature investigations were conducted on T_3.

10. Air-line tests. In the air-lift de-icing system for the Grand Coulee Dam the proposed design incorporates risers of about 60 feet in height from the compressor to the top units (figure 1). During the periods when the compressor is not being operated, water will fill these risers. The question arose concerning whether or not the water could be blown out of the vertical risers by the air from the compressors. This problem was also investigated in the laboratory. Three 5-foot lengths of 1-inch inside diameter pyrex glass tubing were joined together with short rubber unions. By suitable piping the upper end of the glass tubing was connected to an orifice whose area was equivalent to the combined areas of four orifices of a de-icing unit. The lower end of the glass riser



A. Start of Process, Air Pressure Very Low



B. View Showing Water Slipping Past Air Air Pressure 10 lb. per sq. in.

CONDITIONS IN VERTICAL AIR LINE WHILE WATER IS BEING FORCED OUT BY AIR was connected to both air-and water-supply lines. By this arrangement conditions during the time of blowing out the vertical line could be studied visually. The glass tube was first filled with colored water, then the air was turned on. Figure 11-A shows large air bubbles rising in the glass tube. As these bubbles rose they moved some of the water upward; the rest slipped between the rising bubble and the outside of the tube. At the start of the process, water was forced out of the nozzle until the first rising bubble reached it. Then, for a time, the flow of water through the orifice was intermittent, a quantity of water, then air, the amount of water discharged decreasing as the process continued, until finally all water discharged came out in the form of spray. As the air pressure was increased, the air velocity in the tube increased. At about 10 pounds gage the velocity through the tube is about 37 feet per second based on the full area of the glass tube. Under this condition, water remaining in the riser collects in thin sheets next to the tube wall and "hangs" in air stream or slips past it, as shown in figure 11-B.

From this investigation it was concluded that the water could be successfully blown out of the risers by air. It was found that the water did not all go out in one continuous stream but that the flow was pulsating. Most of the water was forced out by the air and it was felt that the remaining water did not impair operation and would ultimately be evaporated by the air.

11. Discussion of experimental results. The method of

measuring the rate of flow of air by using the rotameter deserves mention in regard to accuracy of measurements and the necessary corrections to convert the discharge to terms of free air at gallery conditions. The rotameter, as previously described, consists of a glass tube tapered from a large diameter at the top to a smaller diameter at the bottom. The flow of the fluid to be metered is from the bottom to the top of the tube. Etched on the glass tube of the rotameter used in this investigation are two scales: One, a reference scale graduated in millimeters; the other, a calibration scale for air at standard atmospheric pressure and 70° F. Inside the glass tube there is a metal rotor which spins and hangs in the air stream at a definite height above the bottom of the tube for any particular discharge. Obviously the forces on the rotor in a vertical direction are in equilibrium when the rotor floats at a constant position with respect to the tube; that is, the weight of the rotor just balances the drag on it due to the air stream. The drag on the rotor is a function of its shape, its projected area, the square of the velocity of the fluid, and the density of the fluid (in this case air). Thus:

$$D = C_D \rho A \frac{V^2}{2}$$

where

D = the drag

C_D = a coefficient dependent on the shape of the rotor

ρ = the density of the fluid

A = the projected area of the rotor

V = the velocity of the fluid passing the rotor.

Now, consider metering under two sets of conditions: The first at standard atmospheric pressure (29.92 inches) and 70° F; the second, at some new set of values of pressure and temperature such that the density has changed. Denote the former by subscript "1"; the latter by subscript "2". Let us assume that the drag in the two cases is the same; in other words, that the rotor floats at the same position in the glass tube. Then it follows that:

$D_1 = D_2$

$C_{D_1} = C_{D_2}$ since C_D is a function only of the shape of the rotor

$A_1 = A_2$ since obviously the projected area of the rotor is the same in both cases

then,

$$C_{D_1} \rho_1 A_1 \frac{V_1^2}{2} = C_{D_2} \rho_2 A_2 \frac{V_2^2}{2}$$

or

$$\rho_1 V_1^2 = \rho_2 V_2^2$$

but

$$V = Q/a$$

where V = as before, the air velocity past the rotor

Q = the discharge, cubic feet per second

a = the area between the rotor and the tube wall

Then,

$$\rho_1 \frac{Q_1^2}{a_1^2} = \rho_2 \frac{Q_2^2}{a_2^2}$$

but

$a_1 = a_2$ since the cross-sectional area between the rotor and tube wall does not change if the rotor remains in the same position.

Therefore,

$$Q_2 = Q_1 \sqrt{\frac{\rho_1}{\rho_2}}$$

but

$$\rho = \text{mass per unit volume}$$

$$= \frac{m}{v}$$

and

$$\frac{P_1 v_1}{T_1} = \frac{P_2 v_2}{T_2}$$

or

$$v_2 = \frac{T_2}{T_1} \times \frac{P_1}{P_2} \times v_1$$

Assuming the same mass in each case:

$$\frac{\rho_1}{\rho_2} = \frac{m_1/v_1}{m_2/v_2} = \frac{v_2}{v_1} = \frac{\frac{T_2 \, P_1}{T_1 \, P_2} \times v_1}{v_1}$$

and

$$Q_2 = Q_1 \sqrt{\frac{T_2}{T_1} \times \frac{P_2}{P_1}}$$

where m = mass

v = volume

T_1 = calibration temperature (absolute)

$= 459.4 + 70 = 529.4$

T_2 = air-line temperature (absolute)

P_1 = calibration pressure = 29.92 inches = 14.696 pounds per square inch (absolute)

P_2 = air-line pressure in pounds per square inch (absolute)

By substituting the proper values of T_1, T_2, P_1, P_2, and Q_1 in the above equation, the value was found of Q_2 which is the discharge in cubic feet per second of air at the absolute air-line temperature and pressure. Q_2 was then converted to free air at gallery conditions (28.65 inches and 50° F.) by the equation:

$$Q_3 = \frac{P_2}{P_3} \times \frac{T_3}{T_2} \times Q_2$$

where

Q_3 = cubic feet of free air per minute at gallery conditions

P_2 = absolute air-line pressure

P_3 = absolute gallery pressure (14.073 pounds per square inch)

T_2 = absolute air-line temperature

T_3 = absolute gallery temperature

= 459.4 + 50 = 509.4° F.

To determine the discharge by the rotameter, a calibration curve was constructed from the reference scale (in millimeters) and the discharge calibration scale etched on the glass tube. In calibrating the orifices, the reference scale only was read since a more precise reading could be obtained on this scale. From this reading Q_1 was obtained from the rotameter calibration curve. Then, having the air-line temperature and gage pressure, and assuming the barometric pressure to be the average laboratory barometric pressure, Q_2 was computed by the formula developed above. Q_3 was then found as outlined above.

The accuracy of the calibrations of orifices F, T_1, T_2, and T_3 depends upon several factors: The accuracy of the rotameter calibration curve, the preciseness of reading the reference scale, the accuracy in reading the air-line pressure gage, and the correctness in assuming that the actual laboratory barometer was equivalent to the average barometric pressure (24.72 inches). The latter assumption is probably the greatest source of error and could be as much as eight percent. If all errors were cumulative, the accuracy of the orifice calibrations is still probably within a 12 percent maximum deviation from the true value.

It should be pointed out that in these calibration tests, a change in the back pressure (water head) on the orifice will change the differential pressure at which critical pressure occurs, which, in turn, will change slightly the shape of the discharge curves. To illustrate how the water head on the orifice (back pressure) affects the differential pressure at which critical pressure occurs, consider the case of orifices F, T_1, and T_2. The back pressure under which these orifices were tested was 10.54 feet of water or 4.57 pounds per square inch, causing critical pressure to occur at 14.82 pounds per square inch differential pressure. However, had the back pressure been, say, 35 feet of water or 15.17 pounds per square inch, critical pressure would have occurred at 24.22 pounds differential pressure. The differential pressure at which critical pressure occurs affects the orifice calibration curve (orifice discharge plotted against differential pressure) in the following manner. Until critical pressure is reached, increasing the air-line pressure by increments of, say, one pound per square inch results in increasing by one pound the differential pressure causing flow. Above critical pressure, however, an increase of one pound in the air-line pressure results in an increase of only 0.47 of a pound in the pressure difference causing flow. It is apparent then, that the back pressure, and hence the differential pressure at which critical pressure occurs will affect the shape of the discharge curve.

In constructing the critical freezing curve (figure 8-A), differential pressure was plotted against air temperature before expansion. Ice formation in the orifice is a function of at least the differential pressure across the orifice, the air temperature before expansion, and temperature of the surrounding water; but since the air temperature before expansion is, in turn, a function of the temperature of the surrounding water, the initial air temperature was chosen as one of the coordinates for the freezing curve. It will be seen from table I, in the next to the last line, that the air temperature before entering the cooling coils (column 4) was much higher than in other tests, while the temperature before expansion (column 5) was not changed appreciably. The length of the pipe in the cooling coils was 54 feet. This indicates that the air temperature before expansion will be sensibly the same as that of the surrounding medium, regardless of its initial temperature.

The critical freezing curve (figure 8-A) does not give all the data in regard to the freezing of orifice T_2. While it gives the combinations of initial air temperature and differential pressure at which freezing starts, it does not indicate the degree or seriousness of freezing. In the range of initial air temperatures up to about 33° F. it was observed that it was possible to have the orifice freeze solidly at points (representing pressure and temperature conditions) lying on the critical freezing curve. However, these instances were exceptions rather than the usual result; generally ice would start to form in the orifice, partially fill it, and then be blown out. In order to obtain quick freezing and ice cones (such as shown in figure 10) the differential pressure was increased to 19.25 pounds per square inch. In the range of initial air temperature lying above 33° F., freezing of the orifice was not such a serious factor within the differential pressure range tested. The start of freezing in the range above 33° F. became progressively more and more difficult to detect by observation of the jet as the differential pressure necessary to produce freezing increased, until in the vicinity of 10 or 11 pounds per square inch it could only be detected by the instruments. When the mercury manometer, indicating the air-line gage pressure, showed an increase in pressure, and when the rotameter showed concurrently a decrease in discharge, it was definitely established that freezing had started. In the region of temperature lying above 33° F. it was observed that the duration of the decreased discharge due to ice formation in the orifice became shorter as the values of temperature and differential pressure necessary to cause freezing became greater. Clearing of the orifice was manifested by a sudden increase in discharge and decrease in differential pressure.

In attempting to explain the phenomenon of ice formation in the orifices, the writer proposes the following interpretation. A theoretical analysis was made for expansion of saturated air through orifice T_2, taking the actual values of initial and final pressure and initial air temperature found in test GIC-10 (table I). The expansion was considered adiabatic and account was taken of the variation of the specific heat of the saturated air with temperature. To simplify computations it was first assumed that the amount of water vapor present was constant throughout expansion and equal to that present in saturated air at the temperature before expansion. Thus, the temperature of the gas mixture after expansion was obtained, neglecting moisture condensed out. Account was then taken of the amount of water vapor which condensed and froze, giving to the mixture its heat of vaporization and its heat of fusion. Thus, by a cut and try process, the final temperature of the gas mixture was obtained, arriving at a balance such that: the water vapor necessary to saturate the air at the final temperature plus the amount that condensed and froze just equaled the original amount present before expansion; and the heat given up by the moisture which condensed and froze was just sufficient to raise the temperature of the gas mixture from that value found by neglecting condensation and freezing to the final value. For the particular case investigated, the initial temperature was 491.74 degrees absolute, the initial pressure, 22.92 pounds per square inch, and the final pressure, 16.91 pounds per square inch. From these data, and by the method just outlined, the final temperature after expansion was found to be approximately 2.34° F. This analysis has taken no consideration of heat transfer from the orifice to the gas mixture although there must have been some. It will be remembered that this set of values of pressure and initial temperature constituted one point on the critical freezing curve, which means that freezing of the orifice just started under these conditions. The theoretical final temperature of 2.34° F. is far below the freezing point and it is difficult to conceive that the orifice whose temperature cannot be greater than that of the surrounding water (32.48° F.) could raise the air temperature to such a point that freezing just starts under these conditions. Another question rising from a study of the freezing phenomenon is how frozen moisture particles can stick to the walls of the orifice when moving at such a high velocity, approximately 525 feet per second or 358 miles per hour for orifice T_2 at a differential pressure of 6.012 pounds per square inch. A solution which would satisfactorily explain both of these points is proposed and reference will be made to figure 8-B. The mixture of air and water vapor at pressure P_1 expands through the orifice to pressure P_2 following the approximate streamlines as shown in the figure. From point a to point b the jet contracts, passes through the vena contracta, and then expands again following the walls of the orifice between points b and c. The cold air cools the walls of the orifice between points b and c to some temperature which is higher than that of the air and lower than that of the water surrounding the orifice. As the saturated air expands the temperature drops, resulting in moisture being condensed out of the air. The resulting fluid is then a mixture of moisture particles and saturated air, the air temperature being below freezing. It is conceivable that the particles of moisture do not freeze instantly because there must be a heat transfer before this can take place, and therefore, because of the high velocity through the orifice, it could be possible that unfrozen moisture particles could come in contact with the walls of the orifice between points b and c. It will be remembered from studies of refrigeration that a moist finger will stick instantly to a metal surface when that surface is at a temperature of 15° F. or lower. (This is known as the "stick test.") If the walls of the orifice were at 15° F., then the moisture particles would stick instantly upon contact and freezing of the orifice would start. This hypothesis would satisfy both the question of low air temperature and that of the high air velocity. It is of interest to note that for the one point on the critical freezing curve for which the theoretical final air temperature was computed and found to be 2.34° F., the average of this final air temperature and that of the surrounding water is exactly 15° F., which is the highest temperature at which the "stick test" will occur.

By extending this reasoning further it can be explained why freezing is more serious (that is, the orifice may freeze solidly) when it occurs with the temperature of the surrounding water in the region of 32° to 33° F. The orifices tested in the laboratory were made of wrought iron and lead. The thermal conductivity at 64° F. is 20.1 for lead, and 34.9 for wrought iron.[1] For ice, the thermal conductivity[1] is given as 1.26. This means that a

[1]Marks' Handbook, Third Edition, Seventh Printing, tables 1 and 3, pp. 396-397, McGraw-Hill Co., N. Y.

temperature gradient plotted from the surrounding water, through the metal forming the orifice and to the inside wall can be of a relatively flat slope, but from the inside wall through the ice to the cold air stream the temperature gradient has a relatively very steep slope. Therefore, when freezing of the orifice starts with the temperature of the surrounding water in the region of 32° to 33° F., the temperature of the inside wall of the orifice can conceivably remain below the melting point, the bond between metal and ice remain unbroken, and the freezing process continue on to ultimate restriction of air flow. Whereas, with a higher temperature of the surrounding water, the freezing of the orifice may start, but as the ice deposit increases in thickness, the temperature of the inside wall of the orifice rises until the melting point is reached, the bond between the ice and metal is broken, the flow-restricting ice deposit is blown out, and complete freezing of the orifice is prevented.

12. Summary. From this investigation it was concluded that:

(a) The direction of discharge vertically downward from the orifice gave a superior flow pattern.

(b) An orifice with an 18-degree tapered exit similar to type F gave the best flow pattern of all single-hole types tested. The multiple-hole types were at once abandoned because of the high discharge or because of danger of becoming plugged if the diameter of the holes was decreased so that the discharge would be equivalent to the single-hole type.

(c) A discharge of two cubic feet of free air per minute at a differential pressure of two pounds per square inch is sufficient to induce a strong upward water current. This value for discharge is also compatible with practical limits of compressor size required for the Grand Coulee air de-icing system.

(d) An orifice of the short-tube type (T_2) was found to be somewhat superior to the type represented by orifice F as far as freezing is concerned; its flow pattern was not materially worse, and its discharge characteristics about the same. Therefore, this orifice was adopted for use in the Grand Coulee de-icing system.

(e) When freezing in the orifice occurs with the initial air temperature and surrounding water temperature in the range of 32° to 33° F., the orifice may freeze completely and stop flow. No complete freezing was observed in the tests when the temperatures of the air and surrounding water were above 33° F. and the differential pressure within the range available with the laboratory apparatus. However, complete freezing is probably possible at higher differential pressures.

(f) Freezing of the orifice is a function of the initial air temperature, the differential pressure, the type of orifice, the temperature of the surrounding water, the thickness of the orifice walls, and the thermal conductivity of the material of which the orifice is made. (It is presupposed that the air used has a high enough initial humidity that moisture will be condensed out during expansion.)

Where the danger of freezing is imminent, a sharp-edged orifice should be used. If, however, an orifice of the short-tube type is used because of its superior flow pattern, the freezing hazard may be decreased by making the operating differential pressure small, the walls of the tube as thin as possible and of a material of high thermal conductivity such as copper.

(g) Laboratory tests on 1-inch inside diameter tubing indicated that the water in the vertical supply pipes to the orifices can be blown out with air. The water does not go out in one continuous stream but, at first, as alternate quantities of water and air, and finally, as a spray of water and air mixed. At 10 pounds per square inch differential pressure the air velocity through the tube is 37 feet per second. This velocity was insufficient to carry all the water out of the riser. That which remained hung in thin sheets between the walls and the air stream. It was thought that this remaining water did not impair operation and would finally be either evaporated by the air or taken out as finely divided particles in a mechanical mixture with the air.

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