Equipment Components For Large Evaporator Plants: A Summary Report

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Equipment Components For Large Evaporator Plants: A Summary Report

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UNITED STATES DEPARTMENT OF THE INTERIOR • Stewart L. Udall, Secretary
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FOREWORD

This is one of a continuing series of reports designed to present accounts of progress in saline water conversion and the economics of its application. Such data are expected to contribute to the long-range development of economical processes applicable to low-cost demineralization of sea and other saline water.

Except for minor editing, the data herein are as contained in a report submitted by the contractor. The data and conclusions given in the report are essentially those of the contractor and are not necessarily endorsed by the Department of the Interior.

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OFFICE OF SALINE WATER

EQUIPMENT COMPONENTS FOR LARGE EVAPORATOR PLANTS: A SUMMARY REPORT

I. Spiewak
B. E. Mitchell

SEPTEMBER 1967

OAK RIDGE NATIONAL LABORATORY
Oak Ridge, Tennessee
operated by
UNION CARBIDE CORPORATION
for the
U. S. ATOMIC ENERGY COMMISSION

Contents

Foreword

The Office of Saline Water sponsored a program at ORNL to investigate the equipment components required for 50-Mgd and larger seawater distillation plants. This program was undertaken to determine if the large components and suitable materials were available from industry. Some of the equipment currently available was not designed for use with large multistage flash distillation plants, and the feasibility of using such equipment in desalting plants was evaluated along with factors relating to cost, performance, and estimated lifetime.

The following detailed reports were prepared on the equipment components:

Report Title	Number	Authors
Deaerators for 50-Mgd and Larger Desalination Plants	ORNL-CF-66-6-74[1]	M. Hanig S. C. Jacobs B. E. Mitchell R. K. Sood
Deaerator Ejectors and Blowers for 50-Mgd and Larger Desalination Plants	ORNL-CF-65-12-6[1]	S. C. Jacobs B. E. Mitchell R. K. Sood
Tubing Fabrication Methods for 50-Mgd and Larger Desalination Plants	ORNL-CF-65-12-7[1]	B. E. Mitchell F. J. Moran
Condenser Tube Bundle Configurations for 50-Mgd and Larger Desalination Plants	ORNL-CF-65-12-5[1]	H. M. Noritake R. W. Browell
Evaluation of Long Tubes for Evaporators	ORNL-CF-65-12-4[1]	B. E. Mitchell J. A. Smith
Mechanical Compressors for Vapor-Compression Distillation Plants	ORNL-CF-65-11-55[1]	B. E. Mitchell
Demisters for 50-Mgd and Larger Desalination Plants	ORNL-CF-65-11-54[1]	W. G. S. Fort B. E. Mitchell J. A. Smith
Valves for 50-Mgd and Larger Desalination Plants	ORNL-CF-65-11-53[1]	B. E. Mitchell J. A. Smith
Interstage Vapor Leakage in MSF Plants	ORNL-TM-1770	B. E. Mitchell J. A. Smith

[1]Unpublished report available only to Oak Ridge National Laboratory employees.

EQUIPMENT COMPONENTS FOR LARGE EVAPORATOR PLANTS: A SUMMARY REPORT

1. Introduction and Summary

The investigations summarized here were undertaken to determine whether the equipment components needed for 50 million gallons per day (Mgd) and larger seawater distillation plants were available from industry. These efforts reinforced conceptual design studies of 50- to 250-Mgd evaporators under contract with the Office of Saline Water (OSW).[1]

Included within the scope of the investigations were condenser tube bundles, condenser tubing, tubing fabrication methods, interstage tube seals, demisters, deaerators, deaerator jets and blowers, valves, and vapor compressors.

It was concluded that suitable equipment for desalting plant applications was generally available without extensive development by OSW with the exceptions noted below:

1. More information is needed on the service life of materials useful in evaporator components. The required information can be gained from test bed plants, pilot plants, and loops and does not require full-scale equipment. An applied materials program should reduce the cost and/or increase the reliability of all evaporator components.

2. Although adequate condenser tubing is available now, there are approaches for reducing the cost and/or increasing the reliability of tubing. Approaches included welded tubes of conventional alloys, the associated welded tubes of titanium, very long tubes, and methods of quality control.

3. The technology of deaerator design is not sufficiently advanced to guarantee the low levels of oxygen and carbon dioxide desired to minimize corrosion.

4. A more detailed knowledge of flash-chamber and condenser behavior could lead to the elimination of unnecessary design safety factors. Examples of areas where savings might be gained are in the elimination of demisters in some stages of multistage flash evaporators and in the reduction of "fouling" factors in condenser design.

The OSW has initiated active development programs in most of the areas indicated above.

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[1] *1965 Saline Water Conversion Report*, pp. 178, 233–35, and 277–95, U.S. Govt. Printing Office, Washington, D.C., 1966.

1.1 GROUND RULES FOR ECONOMIC EVALUATIONS

General. – The components investigated were assumed to be part of a 50-Mgd multistage flash distillation plant or of a larger plant containing 50-Mgd units. It is assumed that the energy source is a power plant which supplies electric power, exhaust steam to the brine heaters, and extraction steam as needed.

Evaporator Characteristics. – Although the components investigated would be useful in any evaporator, economic judgments were made on the basis of a multistage flash evaporator operated with a maximum brine temperature of 250°F, a performance ratio of 10 lb per 1000 Btu, a seawater temperature of 65°F, and a maximum brine concentration ratio of 2.0. Sulfuric acid feed treatment was assumed.

Capital Charges. – Capital charges include 5.185% per year for amortization, 0.25% for insurance, and interim replacement costs.

Utility Costs. – Several bases were used for utility costs, reflecting advancements in energy supply.

	Fossil Fuel	Commercial Nuclear	Advanced Nuclear
Exhaust steam, ¢/MBtu	21	10	5.5
100-psig steam, ¢/MBtu	31.8		10.9
Electricity, mills/kwhr	6.2	3.0	2.0

2. Flash Evaporator Components

2.1 CONDENSER TUBE BUNDLE

Large desalting plants will require condenser tube bundles similar to those used in central power stations. In order that conceptual designs of distillation plants of 50- to 250-Mgd capacity may be properly evaluated, and as an aid to initially creating a design, it is desirable that information on condenser design be extracted from the literature and made available to designers and evaluators as required by the desalination program.

2.1.1 Objectives and Scope of Work

The main objectives of the work were as follows:

1. to identify design parameters and extract from the literature the most reliable information on heat transfer, steam pressure losses, tube spacing, tube length, tube diameter, bundle shape, venting, condensate rain, and other technical aspects of condenser design,
2. to review and briefly describe current design practices used by the manufacturers of condensers,
3. to develop a computer program for general industry use and a graphical presentation of the data.

The scope of the work consisted of literature review, discussions with manufacturers, the development of a computer code for bundle design, and some sample calculations which illustrate the use of the code and permit generalizations about bundle design.

2.1.2 Literature Survey

A literature survey was made to obtain reliable information on heat transfer, steam pressure losses, tube spacing, tube length, tube diameter, tube bundle shape, methods of removing non-condensables, condensate rain, and other aspects of condenser design.

Useful data on particular aspects of condenser design were found in the following references:

1. forced-convection film coefficient, Sieder-Tate equation, ref. 2;
2. condensing coefficient, ref. 3;

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[2] W. H. McAdams, *Heat Transmission*, pp. 219ff, 3d ed., McGraw-Hill, New York, 1954.

[3] D. E. Briggs and E. H. Young, *The Condensing of Low Pressure Steam on Horizontal Titanium Tubes*, University of Michigan College of Engineering, Report No. 55, December 1965.

3. steam pressure loss, refs. 4–7;
4. effect of noncondensables, refs. 8 and 9;
5. method of removing noncondensables, discussions with industrial designers regarding current practice (see Sect. 2.1.3).

2.1.3 Current Condenser Design Practices

Manufacturers of steam surface condensers were consulted to obtain general data for use in design of evaporator condenser coils. The design details vary from one company to the next, but the following principles appear to be generally used:

1. The tube layout and steam lanes are used to converge the noncondensables toward collection points usually at the center or bottom center of the bundle.
2. An "air-cooler" section is used to concentrate noncondensables before they are evacuated. This section contains 2 to 5% of the tubes.
3. Steam lanes are not necessary in "small" condensers where the steam flow path through the bundle is less than 3 ft.
4. Steam lanes are provided in "large" power plant condensers for every two to four tube rows. The center of the bundle, near the vent, contains closely spaced tubes with no lanes.
5. High steam velocities, consistent with allowable pressure drop, are encouraged to sweep out noncondensables.
6. Long vertical rows of tubes are avoided to minimize condensate from rows above.
7. Flow-directing baffles and condensate-draining baffles are used in some cases.
8. Tubes are usually rolled into the tube sheets, although welded joints are also used.
9. Minimum web thickness on tube sheets is 0.25 in.

The conceptual designers of 50-Mgd and larger evaporators[1] are generally aware of the problems of condenser bundle design. Most of the designers selected bundle trains producing 3 to 6 Mgd per bundle; there were usually multiple bundles per flashing stream. Bundles of circular, rectangular, or polygonal cross section were used with cross-sectional areas approximating 20 ft^2. These would be "small" condensers based on criterion 3 above, but many condensers are used in a plant.

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[4] D. G. Boucher and C. E. Lapple, "Pressure Drop Across Tube Banks," *Chem. Eng. Progr.* 44(2), 117–34 (1948).

[5] J. E. Diehl, "Calculate Condenser Pressure Drop," *Petrol. Refiner* 36(10), 147–53 (1957).

[6] J. E. Diehl and C. H. Unruh, "Two-Phase Pressure Drop for Horizontal Crossflow Through Tube-banks," *Petrol. Refiner* 37(10), 124–28 (1958).

[7] E. C. Grimison, "Correlation and Utilization of New Data on Flow Resistance and Heat Transfer for Crossflow of Gases over Tube Banks," *Trans. ASME*, PRO-59-8, pp. 583–94, 1937.

[8] S. J. Meisenburg, R. M. Boarts, and W. L. Badger, "Influence of Small Concentrations of Air in Steam on the Steam Film Coefficient of Heat Transfer," *Trans. A.I.Ch.E.* 31, 622 (1936).

[9] R. H. Perry, C. H. Chilton, and C. H. Kirkpatrick, *Chemical Engineers' Handbook*, p. 11-39, 4th ed., McGraw-Hill, New York, 1963.

Several designers used bundle trains of 12.5 to 25 Mgd. They presented reasonable evidence that bundles this size would be workable.

The length of most bundle designs is 50 to 100 ft, with some designers using "long" tubes of 240 ft or more which go through many stages, and some selecting short single-stage bundles to facilitate replacement.

It does not seem possible at this time to select an "optimum" bundle cross section or length. Rather, the choices should be made on the basis of the overall plant layout and maintenance approach.

2.1.4 Computer Code for Bundle Design

Basis for Program. – A computer program was developed to calculate heat transfer values and steam-side pressure drop for various types of tube bundle configurations. Use of this program will enable detailed analysis of condenser tube bundles in each stage of a flash plant, thus allowing close predictions of performance. Incorporated in this analysis is the heat transfer performance of individual tubes as affected by their location within the bundle. Among the factors considered are the following:

1. the amount of condensate flowing down from the upper rows of tubes,
2. the saturation pressure and the saturation temperature at the internal tube rows as determined by pressure drops encountered by the vapor flowing through the bundle,
3. the effect of small amounts of noncondensables in the vapor stream upon the heat transfer coefficient, particularly as the noncondensable concentration increases with passage through the bundle,
4. the effect of pressure level upon the heat transfer and the pressure drops within the bundle,
5. the effect of change of bundle geometry, such as amount of face area presented to the flow, depth of the bundle, fraction of tubes that are shielded by baffles to form a gas-cooler section, etc.

The preferred method of using this program is to incorporate it into an evaporator plant program as a subroutine which makes the initial estimates of the bundle geometry, performs the necessary thermal and hydraulic analysis, and returns an overall heat transfer coefficient to the main program for use in finalizing the plant design. Also included in these calculations could be such additional information as the amount of noncondensables present and the fraction of tubes in a baffled gas-cooler section. Another use of this program could be as a separate program where information on bundle dimensions and vapor flow to the tube is inserted and bundle performance at various locations can be analyzed. These data can be used then to make modifications to the bundle-geometry results of the evaporator plant program, thus obtaining an optimum bundle configuration.

The heat transfer relationships used in the computer program are those cited in Sect. 2.1.2. The only exception to this is the fouling factor, for which there are no good references, and an arbitrary value is used.

Several geometries which can be calculated with the code are shown in Fig. 1.

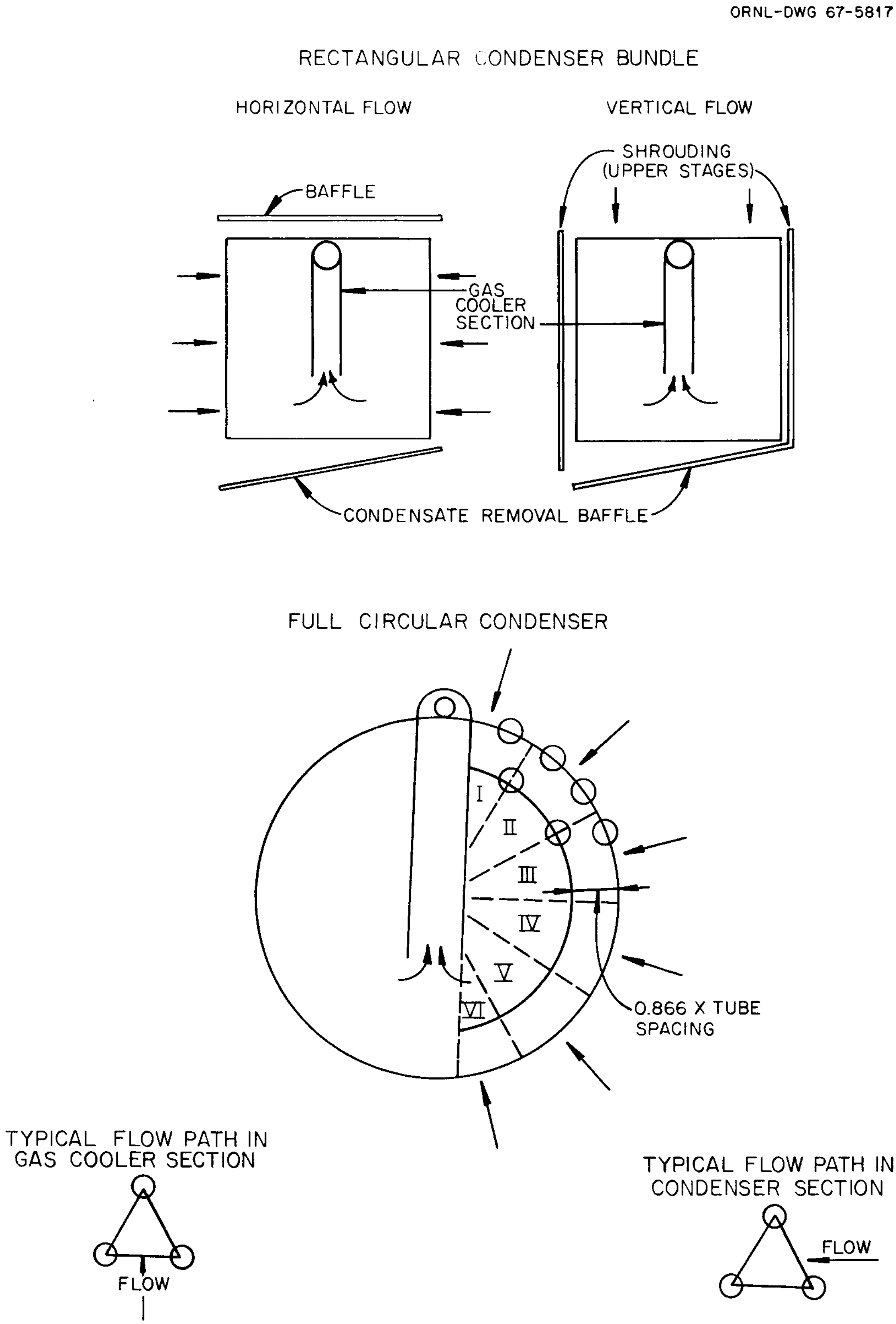

Fig. 1. Typical Condenser Geometries.

The results of the computation can be expressed in one of two ways. A fixed-geometry bundle can be analyzed, in which case the computer will determine how much steam is vented with the noncondensables. Alternatively, the steam input and vent streams are specified for a particular cross section, and the tube length is calculated by the computer.

Heat Transfer Equations. – Calculation of the heat transfer performance within the bundle is performed on a row-by-row basis using the following correlation:

Steam Side – Condensing Coefficient (Film Type)[3]

$$h_m = 0.725 \left[\frac{k_f^3 \rho_f^2 H_{FG} g_c}{\mu_f D_o (t_{\text{sat}} - t_w) n} \right]^{0.25} \times C \times F \, , \tag{1}$$

where

h_m = mean condensing coefficient, Btu hr^{-1} ft^{-2} °F^{-1};

k_f = thermal conductivity of the condensate at the film temperature, Btu hr^{-1} ft^{-2} (°F/ft)$^{-1}$;

ρ_f = density of the condensate at the film temperature, lb/ft^3;

H_{FG} = heat of vaporization at the film temperature, Btu/lb;

μ_f = viscosity of the condensate film, lb ft^{-1} hr^{-1};

D_o = outside diameter of the tube, ft;

t_{sat} = saturation temperature of the vapor, °F;

t_w = outside wall temperature, °F;

t_f = film temperature, °F, = $t_{\text{sat}} - 0.5(t_{\text{sat}} - t_w)$;

n = number of tubes in vertical row;

C = correction factor for n;

F = correction factor for the presence of noncondensables; and

g_c = gravitational constant, ft/hr^2.

The factor C for the effect of number of tubes is calculated as a function of n as follows:

$$C = 1.310 \qquad \text{for } n = 1.0 \, , \tag{2a}$$

$$= 1.2379 + 3.5361 \times 10^{-2} n - 1.5703 \times 10^{-3} n^2 \qquad \text{for } 2 \leqq n \leqq 16 \, , \tag{2b}$$

$$= 1.4017 \qquad \text{for } n > 16 \, . \tag{2c}$$

The effect of noncondensables when less than 4% by weight is calculated[8] as a function of the amount of noncondensables present as follows:

$$F = 1.0 - 3.4313 \times 10^1 X + 1.2268 \times 10^3 X^2 - 1.4923 \times 10^4 X^3 \, , \tag{3}$$

where

X = weight fraction of noncondensables.

Brine Side – Forced Convection Coefficient[2]

$$\frac{hD}{k} = 0.027 \mathrm{N}_{\mathrm{Re}}^{0.8} \mathrm{N}_{\mathrm{Pr}}^{0.333} \left(\frac{\mu_{\text{av}}}{\mu_w} \right)^{0.14} \, , \tag{4}$$

where

h = forced convection coefficient, Btu hr^{-1} ft^{-2} °F^{-1};

D = inside diameter of the tube, ft;

k = thermal conductivity of the brine, Btu hr^{-1} ft^{-2} (°F/ft)$^{-1}$;

N_{Re} = Reynolds number for brine flow, dimensionless;

N_{Pr} = Prandtl number for brine side, dimensionless;

μ_{av} = viscosity of the brine at average bulk temperature, lb ft^{-1} hr^{-1};

μ_w = viscosity of the brine at wall temperature, lb ft^{-1} hr^{-1}.

Iterations are performed to determine t_{wi}, the wall temperature on the brine side, and t_{wo}, the wall temperature on the steam side, in order to define the heat transfer coefficients for both inside and outside surfaces.

When the amount of noncondensables becomes greater than 4% by weight, a weighted gas-vapor coefficient is calculated[9] as follows:

$$\frac{1}{h_{vg}} = \frac{Q_G}{Q_T h_g} + \frac{1}{h_m}, \tag{5}$$

where

h_{vg} = condensing coefficient with noncondensables present,

h_g = gas coefficient based on properties of the noncondensable,

Q_G = sensible heat removed from gas,

Q_T = total heat removed from gas and vapor,

h_m = condensing coefficient with no noncondensables.

The gas coefficient is calculated by the correlation[2]

$$\frac{h_g D}{k} = 0.33 N_{Pr}^{0.33} N_{Re}^{0.6}, \tag{6}$$

where

h_g = forced convection coefficient, Btu hr^{-1} ft^{-2} °F^{-1};

D = outside diameter, ft;

k = thermal conductivity of gas, Btu hr^{-1} ft^{-2} (°F/ft)$^{-1}$;

N_{Re} = Reynolds number for the gas, dimensionless;

N_{Pr} = Prandtl number for the gas, dimensionless.

Pressure Drop Equations. – Pressure drop calculations within the tube bundle use two-phase flow correlations as a multiplier for the friction factor data on tube banks. Since the friction factors for tube banks as a function of Reynolds number, longitudinal pitch, and transverse pitch, as correlated by Grimison,[7] have resulted in curves of such complicated shapes, they have been incorporated into the program as a table rather than as equations. A subroutine is used to perform an interpolation between points of this table.

The two-phase flow correlation of Diehl and Unruh[6] for horizontal flow through tube banks is used as a multiplier to modify the single-phase pressure drop data and has been correlated as a series of equations as follows:

$$\frac{\Delta P_{TP}}{\Delta P_G} = 1.0 \qquad \text{for } X \leqq 0.15, \tag{6a}$$

$$= \exp\left[-0.93213 - 0.74257 \ln X - 0.13174(\ln X)^2\right] \qquad \text{for } 0.15 < X \leqq 2.0\ , \tag{6b}$$

$$= \exp\left[-0.85714 - 0.94972 \ln X - 0.015453(\ln X)^2\right] \qquad \text{for } X > 2.0\ , \tag{6c}$$

where

$$X = \frac{\text{LVF}}{\rho_g/\rho_L}\ ,$$

ΔP_{TP} = pressure drop calculated for two-phase flow,

ΔP_G = pressure drop calculated for total flow occurring in gas,

LVF = liquid volume fraction.

For pressure drop in vertical downflow through tube banks, the two-phase correlation developed by Diehl[5] has been fitted over several ranges as follows:

$$\frac{\Delta P_{TP}}{\Delta P_G} = 1.0 \qquad \text{for } X_D \leqq 0.001\ , \tag{7a}$$

$$= \exp\left[-4.0317 + 0.54538 \ln X_D + 1.0329\ (\ln X_D)^2 + 1.03929\ (\ln X_D)^3\right]$$
$$\text{for } 0.001 < X \leqq 0.02\ , \tag{7b}$$

$$= 0.0225 X_D^{-1.02} \qquad \text{for } X > 0.02\ , \tag{7c}$$

where

$$X_D = \frac{\text{LVF}}{N_{Re}^{0.5}(\rho_g/\rho_L)}\ .$$

Results. – Calculations made for rectangular condensers of a representative multistage flash plant indicate that the pressure drop for vapor flowing through the condenser bank is very small over a major portion of the plant (see Table 1). Since the majority of the stages operate with very small vapor pressure drops, it appears that the bundles for the condenser should be laid out with a minimum allowable web thickness. This can lead to relatively large temperature drops in the last two or three stages, but the overall penalty incurred due to these few stages will be small. On the other hand, the closely spaced tube banks will provide positive sweeping of noncondensables from the tube banks, and this should enhance the performance of those stages where noncondensable buildup is a problem.

Since the pressure drop is very low, steam lanes should not be required in the evaporator section condenser bundles. The heat reject section of a flash evaporator plant might use a more open tube spacing and/or steam lanes, since the vapor volumes to be handled are larger at this end of the plant.

Table 1. Typical Condenser Bundle Dimensions and Pressure Drop

Table applies to a bundle train producing 3.125 Mgd of product in a 56-stage multistage flash evaporator

Stage Number	Condenser Temperature (°F)	Tube Diameter (in.)	Ratio of Tube Spacing to Diameter	Number of Rows		Bundle Dimension (ft)		Vapor Pressure Drop		$\bar{U}$ (Btu hr^{-1} ft^{-2} $°F^{-1}$)	ΔT (°F)
				Horizontal	Vertical	Height	Width	psi	°F		
1	245.0	0.664	1.30	73	44	4.60	3.22	0.003	0.006	684.	11.5
5	234.5	0.664	1.30	73	44	4.60	3.22	0.003	0.007	671.	11.5
10	221.4	0.664	1.30	73	44	4.60	3.22	0.004	0.010	662.	11.6
15	208.2	0.664	1.30	73	44	4.60	3.22	0.004	0.015	645.	11.7
20	195.1	0.664	1.30	73	44	4.60	3.22	0.005	0.022	626.	11.8
25	182.0	0.664	1.30	73	44	4.60	3.22	0.006	0.033	613.	11.9
30	168.8	0.664	1.30	73	44	4.60	3.22	0.007	0.051	572.	12.0
35	155.6	0.664	1.30	73	44	4.60	3.22	0.008	0.080	555.	12.1
40	142.4	0.664	1.30	73	44	4.60	3.22	0.010	0.128	534.	12.1
45	129.1	0.664	1.30	73	44	4.60	3.22	0.012	0.208	496.	12.2
50	115.7	0.664	1.30	73	44	4.60	3.22	0.014	0.340	475.	12.1
54	104.9	0.664	1.30	73	44	4.60	3.22	0.017	0.511	440.	12.0
55	97.7	0.625	1.30	82	27	4.80	1.81	0.004	0.153	384.	11.1
56	89.8	0.625	1.30	82	27	4.80	1.81	0.007	0.313	340.	15.0

Calculations performed with this code have indicated that there are no major advantages or disadvantages of the rectangular-shaped bundle over the circular bundle. Consequently, the shape of the condenser bundle used, whether rectangular or circular, will be dictated primarily by the design of the shell containing the bundles and the shape of the vapor flow path to the condenser. Aside from these considerations, it appears that for high-pressure stages where vapor velocities are low and pressure losses small, a shrouded bundle design such as the rectangular condenser with vertical flow may be desirable from the standpoint of increasing vapor velocity for sweeping out noncondensables. On the other hand, for a very-low-pressure operation where velocities become high and pressure losses become significant, a bundle design presenting a larger face area, such as the circular layout, would appear to be preferable. In this case the horizontal flow alternative seems to give lower pressure drops than the vertical flow system for rectangular bundles and is therefore preferred if a rectangular bundle is used.

Row-by-Row Performance. – A computation was made to illustrate the row-by-row performance of a rectangular bundle at the low-temperature end of an MSF plant, shown in Fig. 2. Ten percent of the total tubes were assumed to be located in the gas-cooler portion of the bundle. As shown in this figure, a pressure drop of 0.06 psi was incurred in passing through the condenser portion of the bundle, and a further loss of 0.03 psi was incurred in the gas-cooler section. At a pressure level corresponding to 85°F, these pressure losses correspond to temperature losses of 3.5°F in the condenser section and an additional 2°F loss for the gas cooler, making a total drop of 5.5°F. The sharper decreases in temperature and pressure in the gas-cooler section are due to two factors, namely, the higher gas velocity achieved in this section and the conservative pressure loss correlation assumed for two-phase flow moving vertically upward.

The overall heat transfer coefficient in the condenser section on an area-weighted basis was 435.6 Btu hr^{-1} ft^{-2} $°F^{-1}$ and 382.8 in the gas-cooler section portion with a combined overall coefficient of 430.9 Btu hr^{-1} ft^{-2} $°F^{-1}$.

The upper line in Fig. 2 shows the decreasing percentage of steam remaining uncondensed as the vapor passes through the bundle. In this calculation, 13% of the entering steam was vented from the top of the air-cooler section. If desired, the code could be used to resize the bundle to give some other desired exit steam fraction.

Effect of Tube Spacing on Condenser Performance. – A study was made to illustrate the effect of changing the tube spacing on the bundle of Fig. 2. The results of the study are illustrated in Fig. 3.

As the tube spacing is increased, thus decreasing the pressure drop within the bundle, the gain in thermal driving force or the log mean Δt improves the performance of the condenser and is shown in Fig. 3 to reduce the uncondensed fraction of steam from 13% to 3% as the tube-spacing-to-tube-diameter ratio is increased from 1.3 to 2.0. A substantial part of this gain is achieved by increasing the spacing from 1.3 to 1.5.

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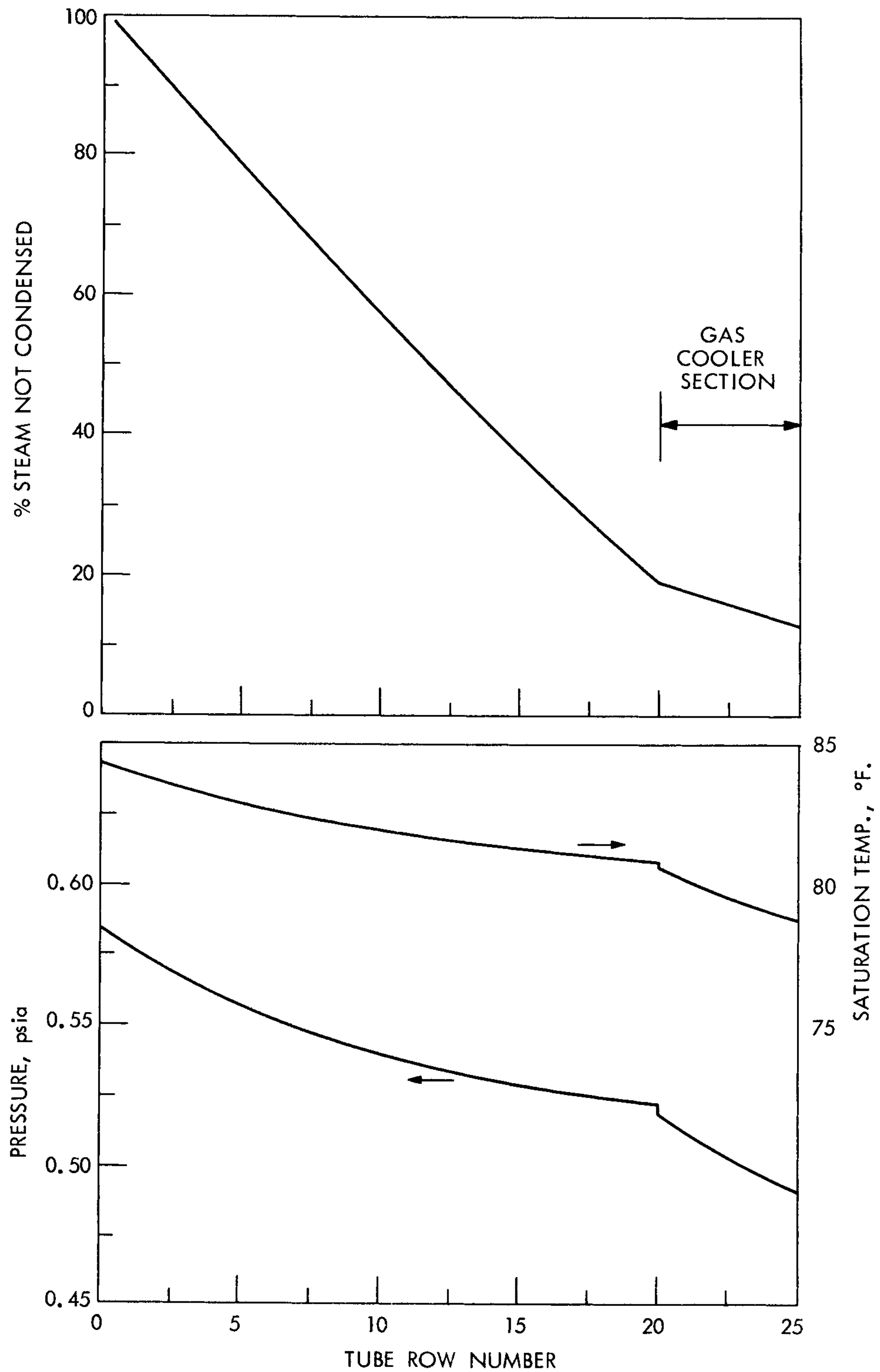

Fig. 2. Temperature, Pressure, and Flow Profile in a Rectangular Condenser Bundle.

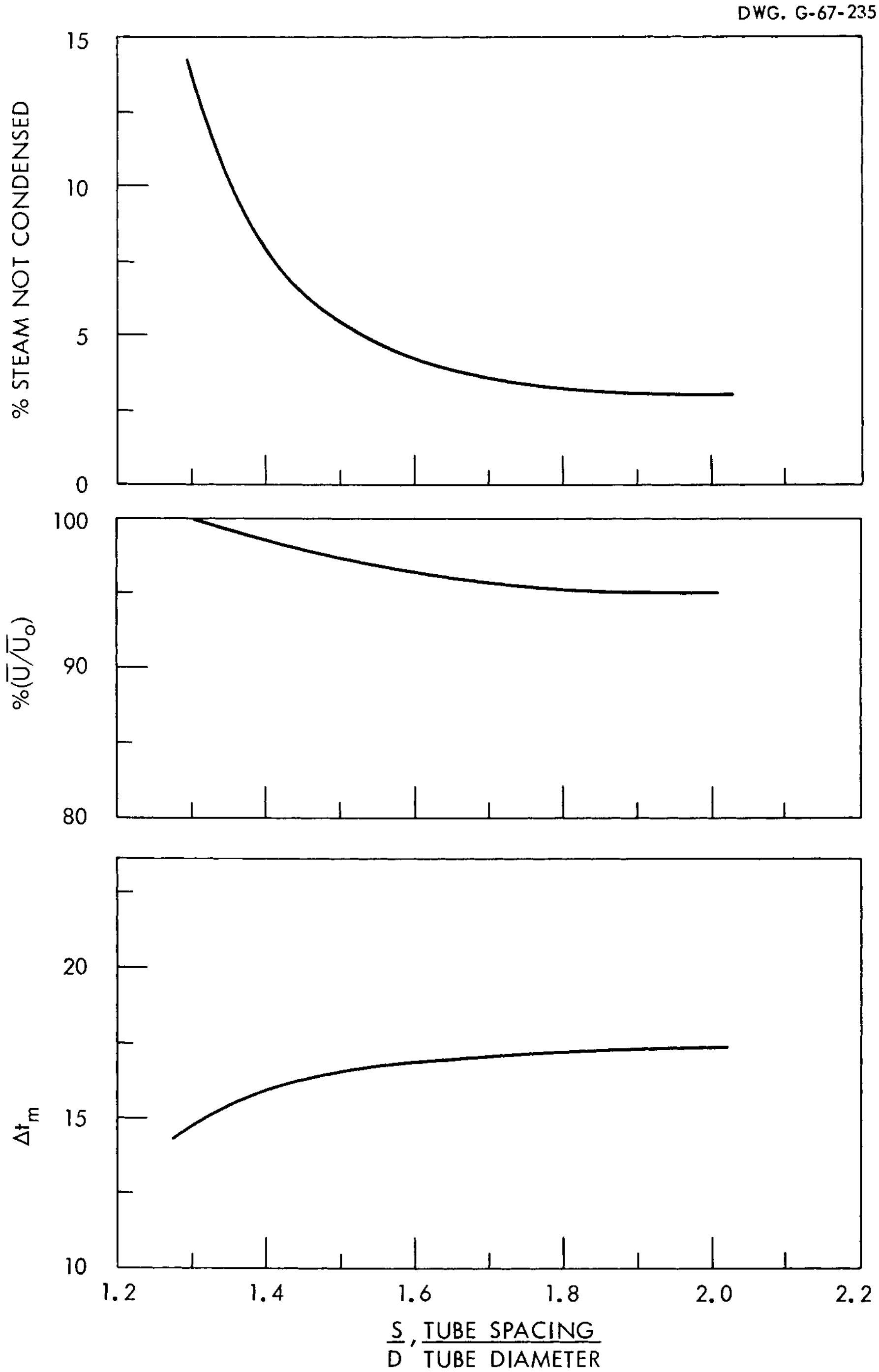

Fig. 3. Effect of Tube Spacing Ratio upon Condenser Performance in the Condenser of Fig. 2.

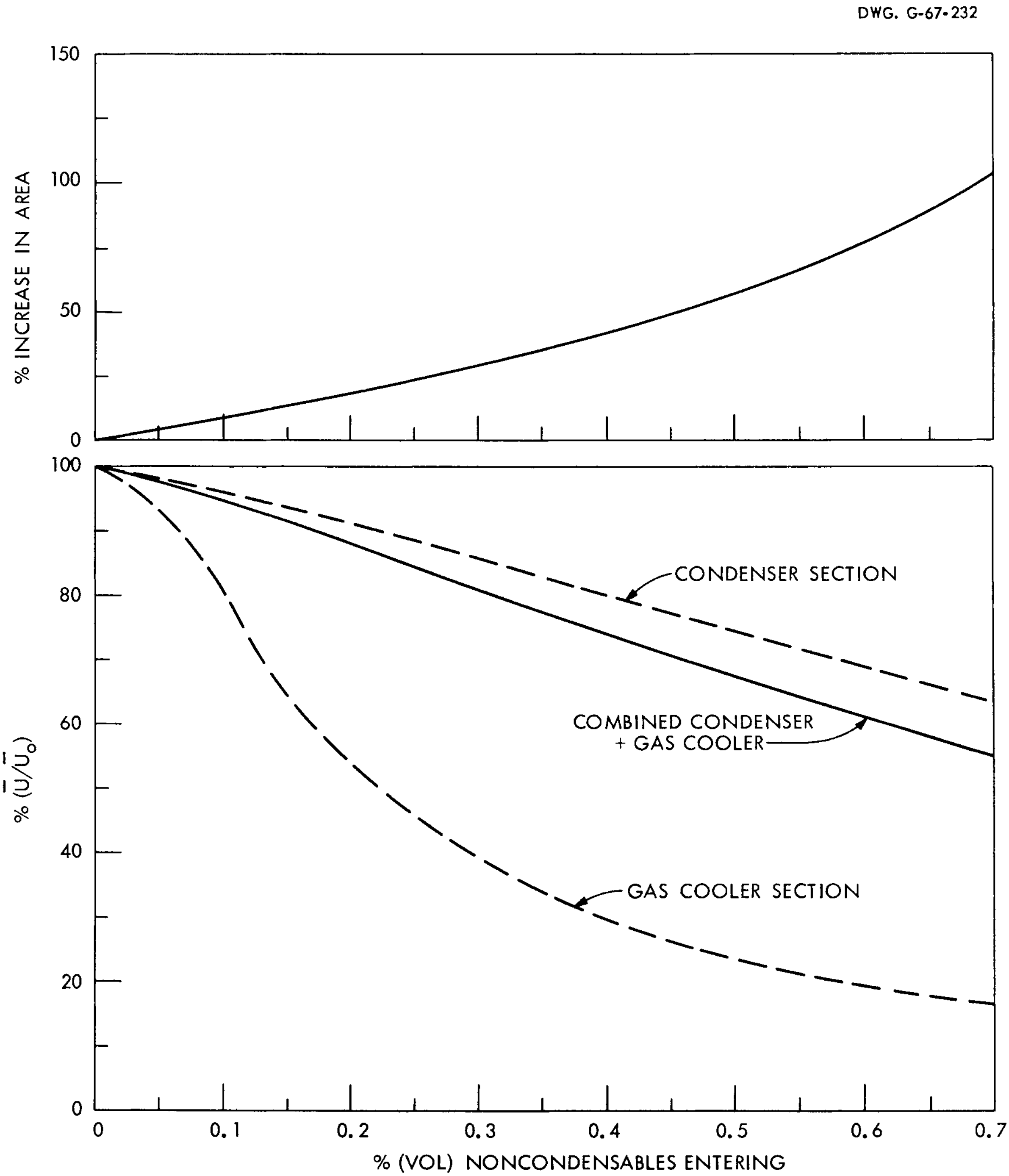

Fig. 4. Effect of Noncondensables upon Condenser Area Requirements.

As may be surmised, the value of wide tube spacing is significant only for lower-pressure operations where vapor velocities are high and pressure drop losses are an appreciable factor. Under these conditions, steam lanes are frequently used by condenser manufacturers.

Effect of Noncondensables on Condenser Area. – A study was made to illustrate the effect of noncondensables on heat exchange area requirements, as shown in Fig. 4. As the entering amount of noncondensables increases from 0 to 0.7 vol %, the required heat transfer area increases by almost 100%. The overall heat transfer coefficient in the condenser section decreases to 63% of its initial value, while the *U* value in the gas-cooler section drops even faster to the vicinity of 20% of the initial value because of the increasing fraction of inerts.

2.1.5 Conclusions

It was concluded that large condenser bundles can be designed on the basis of published correlations. To refine the design equations further would require measurements of local heat transfer and pressure drop in large bundles.

No major differences were found between rectangular and circular bundle configurations from the standpoint of condenser performance.

The use of steam lanes or of increased tube spacing should be considered for bundles operating at 2 psia or below. Relatively small amounts of noncondensables, on the order of 1 vol % or less, can detract seriously from heat transfer performance of a condenser.

2.2 EVALUATION OF LONG TUBES

For several years manufacturers of seawater distillation equipment have been studying designs incorporating tubes of considerable length. Plants with tubes 75 ft long have been built, and plants with tubes up to 300 ft long have been carried at least to the conceptual design stage. To date, such plants have been of the multistage flash type, and the tubes have passed in tandem through several stages. The object of this study was to investigate the feasibility of long-tube conceptual designs.

2.2.1 Scope of Work

The scope of work consisted of the following:

1. analysis of structural considerations – length between supports, provisions for differential expansion rates between tube and vessel, and methods of tubing and retubing in the field,
2. study of fabrication and handling – shop vs on-site fabrication, practical mill lengths, wall thicknesses required for stability in handling, shipping problems, etc.,
3. heat transfer and thermodynamics – effect on pressure drop in systems,
4. evaluation of cost factors – mill extras for various lengths, possible effect on base price and mill extras of large orders optimally spaced, tube sheet drilling costs, tube support plate drilling costs, tube rolling or end welding costs, etc.,
5. long-tube insertion tests – to be conducted to determine the feasibility of inserting and handling long tubes in a test fixture,
6. overall economic and operational comparison between evaporators using long tubes and conventional shorter bundles.

2.2.2 Review of Conceptual Designs

Among the 18 conceptual designs reviewed (ref. 1), the following used tubes over 120 ft long:

1. Aqua-Chem, Inc., used 1-in. by 0.016-in.-wall titanium tubes 900 ft long.
2. Badger Company used 0.625-in. by 0.016-in.-wall titanium tubes 284 ft long.
3. C. F. Braun Company used 0.75-in. by 0.016-in.-wall titanium tubes up to 240 ft long in their MSF feed heater.
4. Burns and Roe used 1-in. by 0.016-in.-wall titanium tubes up to 240 ft long in one of their designs.
5. Oak Ridge National Laboratory used 0.664-in. by 0.042-in.-wall 70-30 CuNi tubes 273 ft long in their 250-Mgd MSF design.
6. Ralph M. Parsons Company used 1-in. by 0.049-in.-wall 70-30 CuNi tubes up to 348 ft long.

In every case the designer anticipated the tubing life to be equal to the plant life of 30 years because of the corrosion resistance of the material selected. None of the nine designers who used aluminum brass or 90-10 CuNi chose long tubes.

2.2.3 Thermal Stress Problems

Provisions are required for tube expansion in the long, continuous tube design concept because of difference in overall thermal expansion of the tubes and of the evaporator shell in which the tubes and water boxes are mounted. The differences in thermal expansion are caused by the following:

1. Differences in heating and cooling rates for the relatively massive evaporator shell and the thin tubes at the time of evaporator startup or shutdown. This can be a large effect unless startup or shutdown is very slow.
2. Differences in steady-state operating temperature between the evaporator shell and tubes. The tubes will be at a few degrees higher temperature than the shell at each point in the plant.
3. Differences in thermal expansion coefficient of the evaporator shell material and tubes. For example, the expansion coefficient for 70-30 CuNi tubes is $9.0 \times 10^{-6}/°F$; for steel, $6.5 \times 10^{-6}/°F$; and for concrete, $7.9 \times 10^{-6}/°F$.

In addition to the above three factors, changes in overall length between the shell and the tubes can be caused by the provision of expansion joints in the evaporator shell. If shell structural and foundation problems are provided for by shell expansion joints at right angles to the tubes, the ends of the shell will remain in a fixed position and the effect will be as though the thermal expansion of the shell were zero. This will give the greatest relative motion between the freely expanding tubes and the shell.

Fixed Tube Sheets. – Since the most straightforward method of providing for tube expansion is for all tube supports to be fixed with the tubes fastened to the end sheets and sliding through the intermediate supports, an investigation was made to determine the feasibility of this method.

The calculational model is illustrated in Fig. 5. A FORTRAN computer program was prepared for calculating deflection, end moments, length of deflection curve, and maximum stress for any given tube, span distance, and end load. Another program was prepared for calculating the stage and cumulative thermal expansion of any tube. Results of the calculations for various alternative tubes are tabulated in Table 2. It can be seen that all the conditions selected stress the tubes well beyond their allowable limit.

Methods of Relieving Expansion Stresses. – Analyses have been made of several methods of reducing tube stresses induced by thermal expansion. Investigation of the alternative methods indicated that the floating tube sheet is the most practical method (Fig. 6). Other methods investigated were:

1. slotted tube supports – where the interstage supports permitted changes in slope of the tube at the walls to relieve bending stresses,
2. expansion rods – where the tube supports were tied together with expansion rods and expanded with the tubes,
3. floating water box – where the entire water box moved with the tubes, instead of just in the tube sheet as in Fig. 6,
4. independently floating tubes – where the tubes were joined only to one tube sheet and were sealed to the other with sliding seals.

2.2.4 Methods of Tube Fabrication and Handling

Tubes made of the alloys used in desalting plants are not supplied in lengths over 100 ft at present. Other tubing, such as copper water tubing, is produced in coils many hundreds of feet long. The following methods of producing and shipping long tubes were suggested by manufacturers:

1. shop fabrication of long coils, shipping, and straightening at the plant site,
2. shipping of strip material, with on-site forming and seam welding of tubes,
3. shop fabrication of long tubes, with shipping on special coupled rail cars,
4. shop fabrication of shells and on-site drawing of tubes.

Tubing will be inspected by either ultrasonic or eddy-current methods to discover hidden flaws. This is particularly important for welded tubes and will preferably be done simultaneously with the welding, so that defective joints can be immediately detected.

Consideration has been given to several methods for splicing tubes. In general, the methods discussed with the tube manufacturers were upset welded joints, brazed sleeve joints, butt-welded joints with filler metals, and butt-welded joints by fusion welding with a split-type weld head.

Tubes will be stacked and stored after testing and inspection. The tubes are rather fragile and difficult to handle; for example, a $\frac{3}{4}$-in. by 0.035-in.-wall 90-10 CuNi tube 275 ft long weighs about 85 lb, must be supported every 25 ft to avoid damage from its own weight, and can be bent on a 14-ft radius without taking a permanent set. They must be stored carefully with adequate support to avoid bending, crushing, or denting.

Table 2. Thermal Stresses in Long Tubes

Tube Number[a]	Tube Temperature, Inlet (°F)	Tube Temperature, Outlet (°F)	Stage Length (in.)	Number of Stages	Total Thermal Expansion (in.)	Average Thermal Expansion per Stage	Y_2, Deflection (in.)	Stress ($lb/in.^2$) Bending + End Load	End Load (lb)	Tensile Strength (psi)	Yield Strength (psi)	ASME Code Allowable (psi)
1	88.0	226.0	101.0	40	3.60	0.090	1.91	37,806	1037	50,000	18,000	12,000
2	88.0	226.0	101.0	40	2.12	0.053	1.49	31,849	355	58,000	40,000	14,500
3	88.0	226.0	71.0	40	2.56	0.064	1.35	58,356	2115	50,000	18,000	12,000
4	88.0	226.0	71.0	40	1.48	0.037	1.03	48,491	757	58,000	40,000	14,500
5	88.0	226.0	71.0	40	2.36	0.059	1.30	42,965	695	38,000	15,000	9,000
6	71.4	225.2	71.3	40	1.52	0.038	1.03	32,549	296	58,000	40,000	14,500
7	71.4	225.2	71.3	40	2.40	0.060	1.30	42,965	695	38,000	15,000	9,000
8	88.0	226.0	140.0	20	2.56	0.128	2.69	26,489	485	50,000	18,000	12,000
9	88.0	226.0	140.0	20	1.48	0.074	2.11	22,309	160	58,000	40,000	14,500

[a]Description of tubes:

Tube Number	Description
1	1.0 in. OD × 0.049 in. wall × 338.9 ft aluminum brass
2	1.0 in. OD × 0.016 in. wall × 338.9 ft titanium
3	1.0 in. OD × 0.049 in. wall × 236.7 ft aluminum brass
4	1.0 in. OD × 0.016 in. wall × 236.7 ft titanium
5	0.75 in. OD × 0.035 in. wall × 236.7 ft 10% 90-10 CuNi
6	0.75 in. OD × 0.016 in. wall × 271 ft titanium
7	0.75 in. OD × 0.035 in. wall × 271 ft 10% 90-10 CuNi
8	1.0 in. OD × 0.049 in. wall × 236.7 ft aluminum brass
9	1.0 in. OD × 0.016 in. wall × 236.7 ft titanium

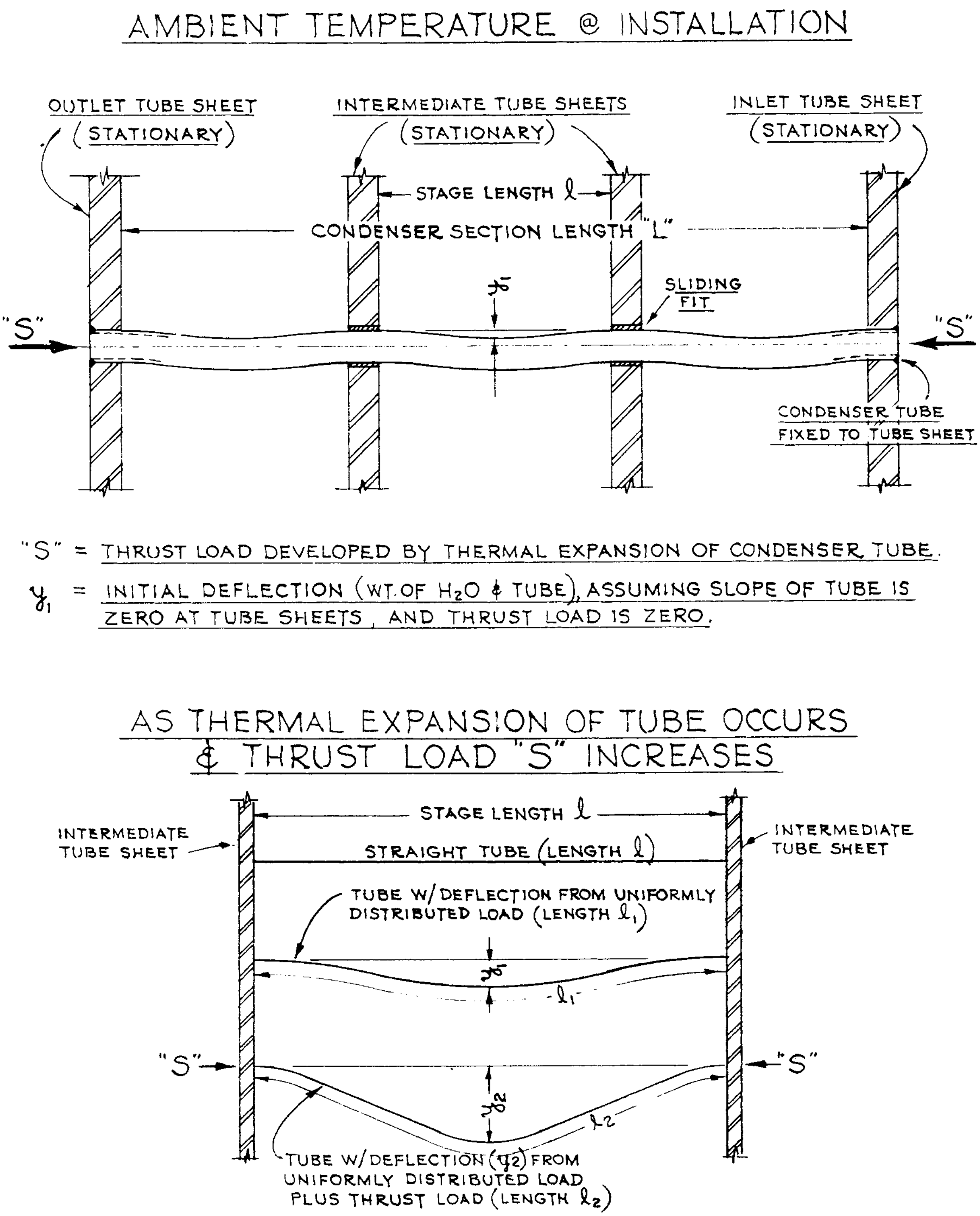

Fig. 5. Tubing Thermal Expansion Model.

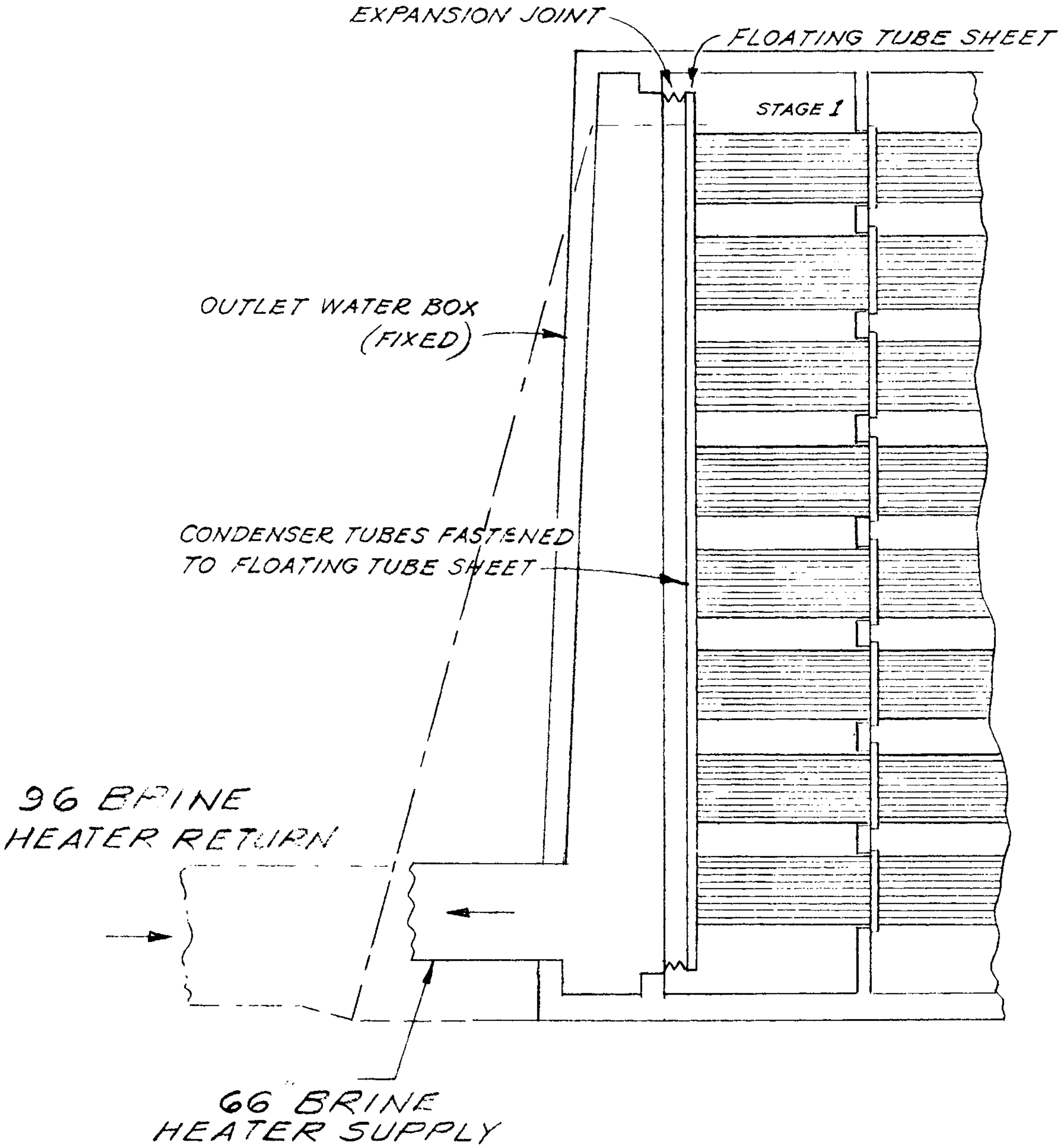

Fig. 6. Floating Tube Sheet.

2.2.5 Methods of Installing Long Tubes

A number of methods were analyzed for installing 275-ft-long tubes into a 250-Mgd 40-stage evaporator. The methods involved different ways of driving and guiding individual tubes through many preplaced tube supports.

The preferred method is shown in Figs. 7–9. Specially designed grooved trays would act as guiding devices. The tubes are inserted one horizontal row at a time. They are power-driven through the tube support holes with the hydraulically operated tube pusher shown in Fig. 8.

When the tubes are to be inserted in the evaporator, they will be loaded on a train of carts on tracks running from the storage point to the evaporator. The carts preferably will be loaded with the correct number of tubes for one complete tube bundle and arranged in the proper number of horizontal and vertical rows on spacer supports on the carts. The tube supports on the carts will be designed as frictionless rollers so that the tubes can be easily pulled longitudinally from the carts.

When the train is loaded, any special tube preparation, such as swaging bullet noses on the tubes to facilitate insertion, will be done using portable tools. The train will then be pulled along the tracks to scaffolding at the evaporator in line with the opening where the tubes will be inserted. It is not necessary to leave clearance in line with the evaporator for straight-in insertion of the tubes, since they can be brought around a fairly tight curve without damage to the tubes. Where the train is brought around a curve, it may be necessary to use radius rods on the carts to prevent derailment because of side forces required to bend the tubes.

2.2.6 Long-Tube Insertion Tests

The test system (Fig. 10) consisted of a stand with simulated condenser tube sheets at 7-ft intervals across a span of 364 ft, a tube pusher mechanism for installing the long tubes, a dynamometer for measuring the pushing force, and several 365-ft tubes, which were fabricated from 50-ft, 0.049-in.-wall, $\frac{3}{4}$-in.-OD 90-10 CuNi tubes spliced together. The semicircular tube sheets contained ten 0.760-in.-diam (TEMS standard fit) holes for tubes without interstage seals and ten 0.875-in.-diam holes for tubes with interstage seals. These holes were beveled on each side of the tube sheet. All the tube sheets were aligned to within 0.036 in. on the horizontal and vertical axes with respect to the center line of the tube sheet housing and were parallel to each other. A surveyor's transit and a tube sheet fixture were used to attain this alignment. Two-hole brackets were later fabricated and attached to the original tube sheets in order to conduct tube insertion tests on tube sheet hole sizes between 0.752 and 0.760 in. These holes were not beveled, as the larger holes were; more tube binding and tube scratching resulted when these holes were used.

All the following tube insertion tests were conducted with tubes which measured 0.751 to 0.752 in. in outside diameter.

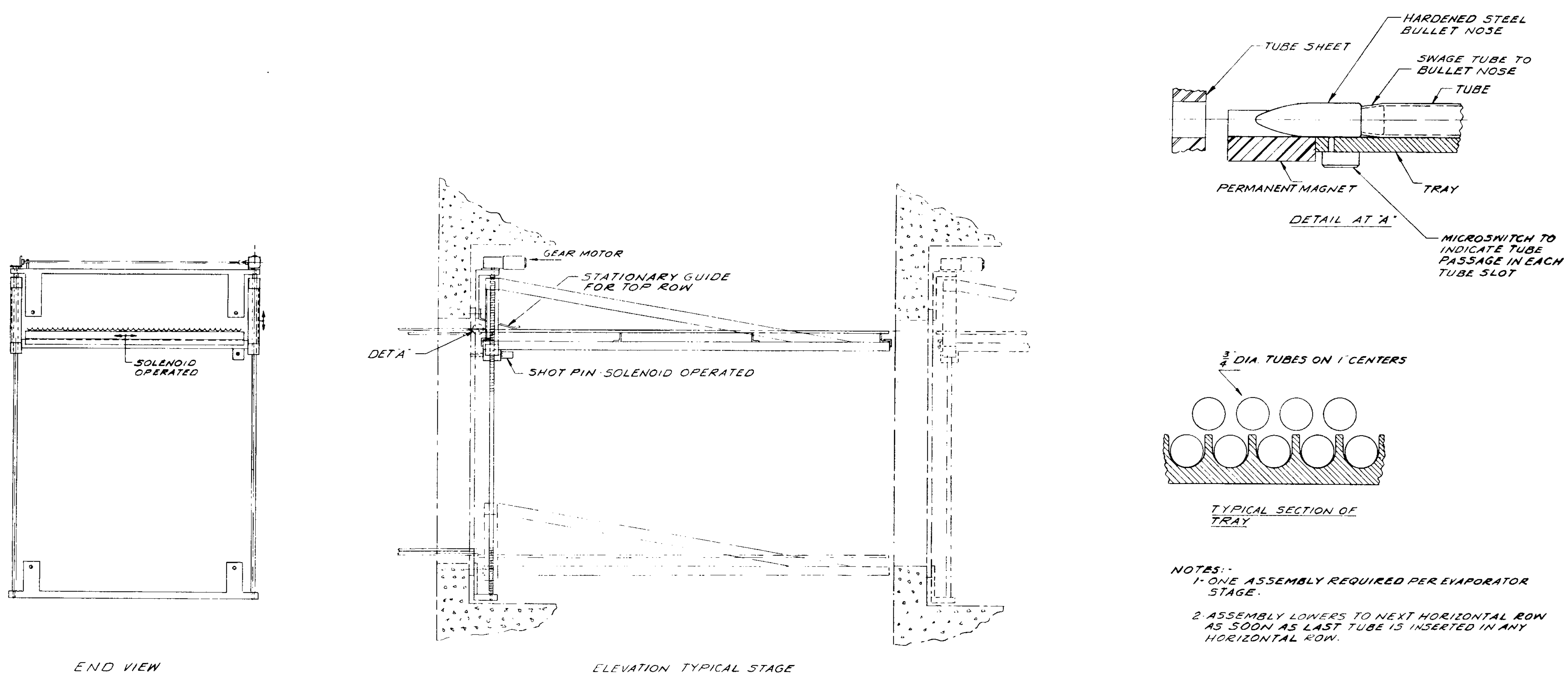

Fig. 7. Automatic Tube Guide.

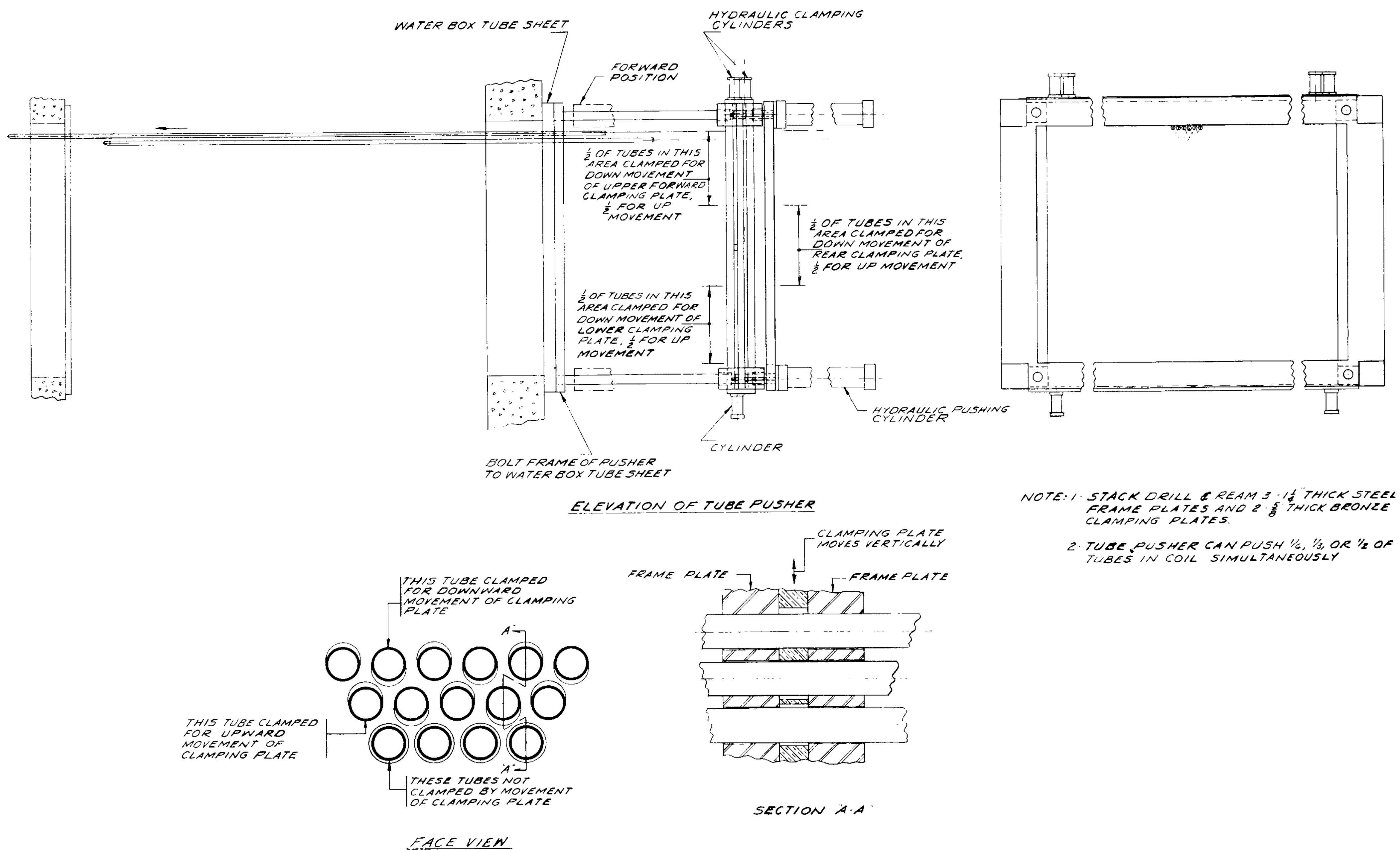

Fig. 8. Tube Pusher.

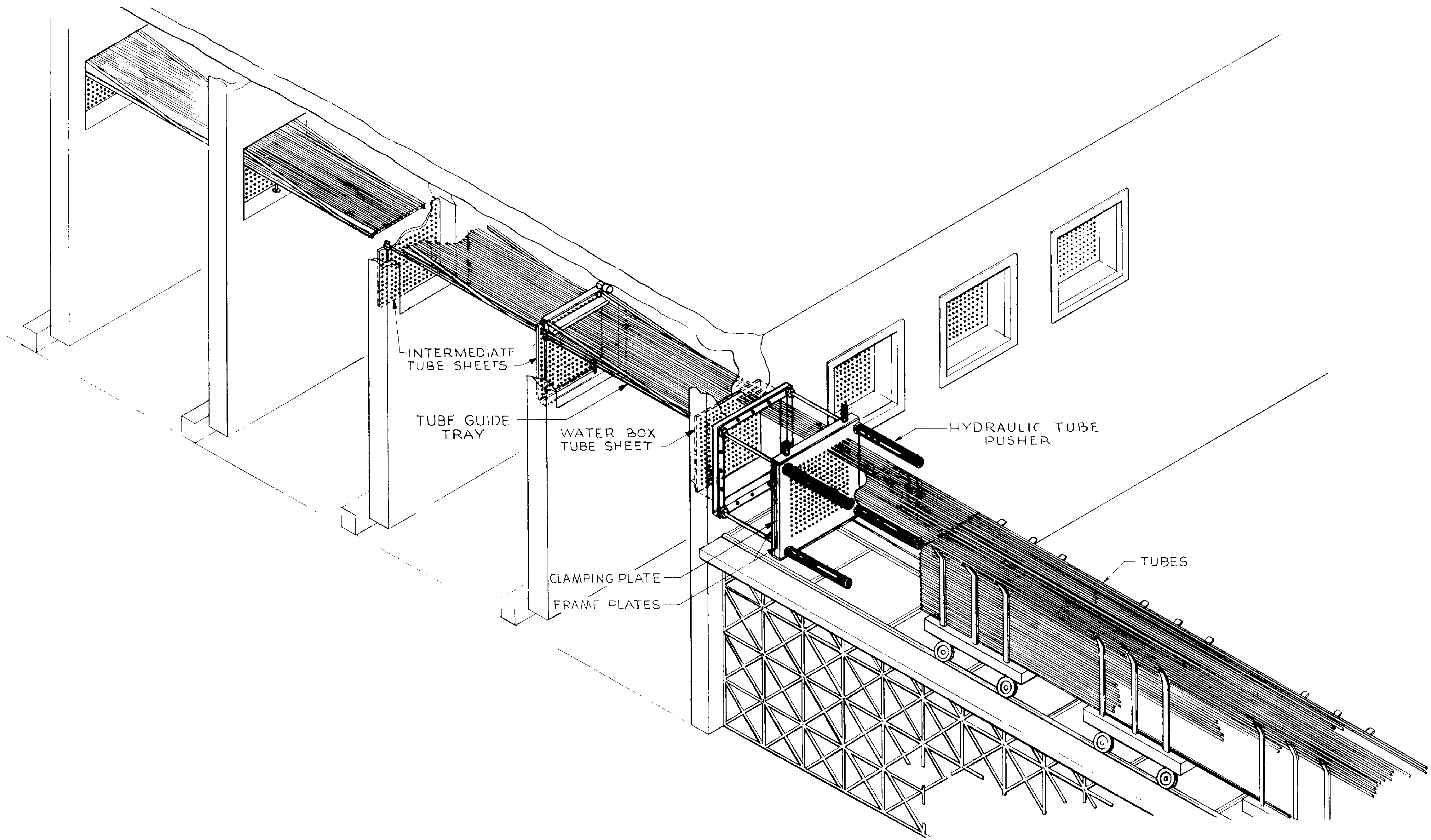

Fig. 9. Insertion of Full-Length Tubes in Evaporator.

PHOTO G-66-959

Fig. 10. Long-Tube Test Facility.

The initial test consisted of manually installing a 365-ft-long tube through tube sheets with 0.760-in. holes. With one man guiding the tube through the tube sheets, one man supporting the end of the long tube, and one man providing a pushing force, the tube was installed with ease.

Two-hole brackets were then attached to the intermediate tube supports to obtain a range of tube hole sizes between 0.752 and 0.760 in. As was anticipated, it was found that the 0.752-in.-OD tube could not be inserted through the tube sheets with 0.752-in. holes. The tube could be inserted through tube sheets with 0.754-in. holes, but with a degree of difficulty which was impractical. The tube could be inserted through tube sheets with 0.755-in. (TEMS special close fit) holes with difficulty but within a practical range. Thus, we determined that the minimum differential diameter size between the tube and the tube sheet hole was 0.003 in. or an annular clearance of 0.0015 in. and that this differential was dependent upon close alignment of all tube sheets. Tubes of 0.752 in. OD can be installed through tube sheets with hole sizes from 0.756 to 0.760 in. with relative ease.

Tube vibration occurs as the tubes are inserted through close-fitting tube sheet holes. When this occurs, the required driving forces increase sharply to a point where the drive rolls slip, stopping the forward motion of the tube. In such instances, it was necessary to interrupt tube insertion to allow the tube vibration to cease. This problem was less pronounced as the tube sheet holes were increased in size. The tube vibration increased the time of tube insertion from approximately 330 sec for 0.760-in. tube sheet holes to approximately 860 sec for 0.755-in. holes. Vibration also occurred when the tube was removed but was not as pronounced. The tube pusher provided the driving force in both directions – pushing for inserting and pulling for removing. The results of these tests are summarized in Table 3.

Table 3. Tests with Properly Aligned Tube Sheets

The data below represent the results of tube insertion tests with all tube sheets properly aligned on the horizontal and vertical axes with respect to the center line of the tube sheet housing and with all tube sheets parallel to each other.

Test Number	Tube Size, OD (in.)	Tube Sheet Hole Size (in.)	Annular Clearance (in.)	Dynamometer Reading (lb)	Insertion Speed (fpm)
1	0.752	0.760	0.004	30–40	66.4
2	0.752	0.754	0.001	High	12.2
3	0.752	0.755	0.0015	80–250	25.5
4	0.752	0.756	0.002	50–250	
5[a]	0.752	0.753	0.0005	80–100	57.3
6[b]	0.752	0.760	0.004	80–100	

[a]With polypropylene seals in tube sheet holes.
[b]Automatic alignment of tube.

Successful tests of tube insertion were conducted with tube-to-tube-sheet seals of polypropylene. The seals were installed in the 0.875-in. tube sheet holes and had an opening measuring 0.753 in. in diameter. Thus, the annular clearance between the tube and the seal was 0.0005 in. The results of this test are also summarized in Table 3.

A demonstration was made to show that the concept of power feeding in conjunction with automatic alignment of the tube with the tube sheet holes is feasible and practical. V-shaped tube guide brackets, which were fabricated from 1- by $\frac{1}{8}$-in. angle iron, were positioned atop and between two adjacent tubes which were already in place. The V-shaped bracket was thus in position to accept a bullet-nose-equipped tube and to guide the tube into the proper tube sheet hole. Fifteen such brackets were used, and a tube was power fed and automatically aligned over a total distance of 105 ft.

Tests were conducted to simulate the effect of tube sheet misalignment caused by settling of the evaporator building. This misalignment could cause concern if it becomes necessary to remove old tubes and install new ones. This condition was simulated in the 364-ft-long tube insertion test rig by raising the $\frac{1}{4}$ and $\frac{3}{4}$ supporting points of the testing rig $\frac{1}{4}$ in. and by lowering the $\frac{1}{2}$ support point $\frac{1}{4}$ in. Tube insertion tests with three misaligned plates demonstrated that a tube could be installed through tube sheets with 0.760-in. holes but could not be installed through tube sheets with 0.755- and 0.756-in. holes. In further tests with seven plates similarly misaligned, tube insertions through 0.760-in. holes were successful.

Tests were conducted also to determine the effect of tube sheets which were not parallel to each other. This was accomplished by rotating two tube sheets 2.4° in opposite directions. With this condition, tubes could not be pushed through even the largest size (0.760-in.) holes. Parallel misalignment has a greater effect on tube insertion efforts than either vertical or horizontal misalignment.

2.2.7 Operation and Maintenance Problems

Methods of Removing or Replacing Tubes. – The long tubes can be removed, if necessary, by using a rotary cutter to cut the inside of the tube behind the tube sheet at both ends, removing the rolled ends, inserting expansion rod devices into both tube ends, and withdrawing the tube by applying a pulling force with a winch. A pulling rod or cable should be pulled in from the opposite end as the tube is pulled out. A short section of the tube or the entire tube could be replaced and reinstalled by this method. It is possible that impingement erosion near the water box might require replacing a short section of each tube. This can be done by pulling out a section of each tube, as described above, replacing the faulty section, pulling the tube back into place, and rolling the tube ends into the end tube sheets. In a similar manner, the entire tube can be replaced, if desired. Dummy support plates approximately 6 in. from each end tube sheet would be required on the original installation to hold the tube in position after the ends were cut, and good access to the end tube sheets would be required to successfully remove and replace the tubes.

Plugging of Tubes. – Tubes will be plugged as leaks develop. If power plant practice were followed, an evaporator might be shut down for replacing all the tubing after 10 to 15% had been plugged. Long-tube plants using 70-30 CuNi or titanium would presumably not require retubing during their 30-year life.

With 360-ft-long tubes, plugging would be simple since the tube ends are easily accessible. With shorter tube lengths and intermediate water boxes, tube plugging would be more difficult since access would be required to the interior of all water boxes for testing and plugging.

Tube Support. – Vibration of sufficient amplitude can rapidly destroy condenser tube bundles by fatiguing the tubes at points of maximum stress.

In the long-tube design, tubes will be supported at least at every stage wall where they pass through an intermediate tube support. If more frequent support is required, one or more additional supports can be provided within each stage. For a large plant, support may be provided by an additional intermediate tube support in each stage since the tooling will be available for drilling it along with the stage separator supports. Several tube-lacing or bar-separator schemes are in use which can be installed after the tubes have already been inserted.

The standard practice for support spacing in steam surface condensers is at intervals of 50 to 60 tube diameters. For the long-tube evaporator design, which warrants a high premium for maximum protection of the tubes, supports at the 50- to 60-diam spacing might be assumed in the absence of contrary data.

Tube Leakage. – The product water is contaminated by the leakage from condenser tubes. A completely ruptured tube in the 250-Mgd plant design would increase the salt content of the product by about 5 ppm. Possibly five leaks of this magnitude could be tolerated out of 188,160 long tubes before part of the evaporator need be shut down for repair.

For purposes of planning interim replacements, it is assumed that 70-30 CuNi tubes will have a 30-year life. This is interpreted to mean that after 30 years it would be more economical to replace the tubes than to continue to operate the original set. Of the 188,160 tubes in a 250-Mgd plant, possibly 10% would be plugged. Tube failures after the initial shakedown period are likely to be infrequent for the first 20 years and increasingly frequent thereafter. Shutdowns of a portion of the plant for tube plugging are probably not required for many years; such repairs could be carried out during plant outages.

A plant containing 60-ft tubes is likely to experience two to three times as many failures as the 360-ft-tube plant because many failures are associated with inlet turbulence. The fraction of surface lost after 30 years is likely to be less than half as much, however, because $\frac{1}{6}$ as much tubing is lost per tube failure.

The long-tube design offers an advantage in making rapid repairs when tube leaks occur. The water boxes are at the ends of the evaporator heat recovery and reject sections and are fairly accessible. To find a leak, tubes would be plugged temporarily (for example, with rubber stoppers) at one end. Then a slight positive air pressure in the evaporator would cause air flow in any leaking tube; this flow could be detected by a simple test, such as a soap bubble check at the unplugged tube end.

There are some service disadvantages to long tubes. Each long tube must withstand the full range of service conditions, while the short bundles can have tubes of different wall thicknesses and materials adapted to the exact conditions. In the event some condition of temperature, pressure, or noncondensables causes accelerated attack on tubes in only a few stages, repair of the short-tube design would be easier than for the long-tube design.

To recapitulate, a long-tube plant would have perhaps two to three times as much surface out of service at a given time relative to a short-tube plant. More downtime and much more maintenance labor are required for leak hunting and tube plugging in the short-tube plant. Most of these casualties would occur after 20 years in a 70-30 CuNi system.

In the absence of real service data, it is difficult to predict a significant difference between long- and short-tube plants in leakage costs.

2.2.8 Economic Analysis

Capital cost breakdowns were prepared for 250-Mgd plants containing 60- and 360-ft tubes. There was an estimated difference of about $2 million in favor of the long tubes over the short tubes, of which 80% was a saving in the cost of water boxes. This analysis assumed the tube costs would be equal.

Table 4. Comparison of Annual Costs of Long and Short Tubes (250-Mgd Plant)

	Prefabricated Bundle, 60 ft	Field-Fabricated Bundle	
		60 ft	360 ft
Annual fixed costs			
Amortization at 5.185%	$2,679,050	$2,653,336	$2,560,352
Insurance at 0.25%	129,173	127,933	123,450
Interim replacement at 0.35%	180,842	179,106	172,830
Total fixed costs	$2,989,065	$2,960,375	$2,856,632
Annual operating costs			
Power losses in water boxes at $40.20 per box per year	16,100	16,100	
Total operating costs	$ 16,100	$ 16,100	
Total annual costs	$3,005,165	$2,976,475	$2,856,632
Differential cost of 60-ft tubes over 360-ft tubes	$ 148,533	$ 119,843	
Differential cost, cents per 1000 gal	0.181	0.146	

The short-tube design has a number of pressure losses at the water boxes which do not exist in the long-tube design. In a 250-Mgd plant with 5.65 fps tube velocity and five intermediate water boxes per bundle train, there would be a power loss of $16,100 per year (2 mills/kwhr power cost).

A potentially important difference between the cost of long and short tubes is in the maintenance cost. In the absence of service data, reliable projection of a difference in cost (see Sect. 2.2.7) cannot be made.

Table 4 summarizes the cost differences and shows a difference in water cost of 0.146 to 0.181¢ per 1000 gal in favor of the long tubes for the assumed conditions.

2.2.9 Conclusions

The studies and tests reported here indicate that the use of tubes up to 350 ft long is feasible, although good alignment of tube supports is required for a many-stage plant.

It would be expected that the long-tube design would be most attractive when combined with tubing expected to have a life as long as the plant.

The cost advantages of the long-tube evaporator are estimated to be about 3% reduction in capital cost and 1% reduction in water cost over short-tube designs.

2.3 INTERSTAGE VAPOR LEAKAGE

2.3.1 Objectives and Scope of Work

In multistage flash distillation plants incorporating long tubes passing in tandem through two or more stages, a problem exists at the place where the tubes pass through the interstage dividing plates. If the holes through which the tubes pass are left large enough for ease in assembly and, more particularly, for later retubing, the leakage of water vapor from one stage to the next could appreciably reduce the plant economy. If these holes are reduced or eliminated to reduce leakage, initial fabrication and subsequent retubing become more difficult, if not impossible.

The objectives of this investigation were as follows:

1. to calculate for selected typical cases what the leakage would be and how much effect it would have on the economy of the plant,
2. to find out what methods of sealing have been used in the past and what the experience with them has been,
3. to find out what other methods have been proposed and to examine them with respect to probable antileakage effectiveness, fabrication and retubing problems, and cost.

The scope of work consisted of a survey of industry practice, an analysis of leakage factors in multistage flash evaporators, the fabrication of seals, and the measurement of leakage with and without seals.

2.3.2 Review of Industry Practice

Commercial Evaporators. – Evaporator manufacturers normally use seals. Investigation reveals that several different methods have been used commercially for sealing the annulus between the tube and the tube sheet. These include the following:

1. *Labyrinth Seals.* – A labyrinth-type seal takes the pressure drop between the two stages. This type of seal should provide adequate sealing but would be quite expensive in comparison with some of the others.

2. *O-Ring Seals.* – A rubber O-ring is installed in a recess in the tube sheet hole. This type of seal has provided good sealing in an existing evaporator for about five years, but grooving the support plates at low cost requires special equipment.

3. *Metal Ferrule Seals.* – A metal ferrule is inserted between the tube and the tube sheet. This method of sealing will work only where the tubing diameter is controlled closely. Since there is friction between the tube and ferrule, the method does not lend itself to long tubes.

4. *Seal Inserts.* – A plastic or rubber insert seals the annulus. Some designs have the disadvantage that the condenser tubes cannot be put on a minimum spacing.

5. *Packed Tube Sheet.* – The seal consists of a secondary support plate secured to the interstage support plate, and the space between the two support plates is filled with a plastic packing or a packing of metal or glass shot. This packing would insulate some of the condenser area from useful condensation; this would amount to about 2% of the condenser area.

Table 5 gives ORNL estimates of the costs of these various sealing methods.

Table 5. Cost of Sealing Methods Used by Manufacturers

Method	Cost per Seal
Labyrinth	94¢
O-ring	8¢
Metal ferrule	7¢
Ring insert	25¢
Packed tube sheet	4¢

Conceptual Designs (see ref. 1, Sect. 2.1.6). – Most of the study contractors analyzed the problem of leakage of steam from one stage to the next through the gaps between the tubes and the holes in the stage dividers. Foster Wheeler reported their calculations in detail. They concluded that nylon ferrules should be used in the first 36 stages of a 72-stage plant.

Parsons reported the results of their analysis from which they concluded that the leakage rate would be tolerable when the normal TEMS machinery tolerance for baffles of $\frac{1}{64}$ in. is used. Parsons' design with 98 stages is characterized by very low interstage pressure difference, and therefore a low driving force for interstage leakage.

Aqua-Chem and AREL also conclude that ferrules are unnecessary. Worthington provides swelling fiber inserts for the first 11 stages. Fluor provides inserts in the higher temperature stages.

Braun proposes a plastic seal around the tubes. The stage dividers are made from 2-in.-thick asbestos-reinforced furan plastic. The Badger Company proposes either to provide split collars of fibrous or elastomeric material or to spray a liquid elastomer around each tube on the higher-pressure side.

Burns and Roe proposes to cast the stage divider around the tubes in place, using 1-in.-thick plastic. Lockheed avoids the problem by use of a drilled insert plate.

2.3.3 Calculated Leakage

Methods were developed to compute vapor leakage through the annulus formed between tube and tube sheet hole. The mathematical model employed[10] is an annulus with complete eccentricity, which appears to best fit the test results. Flow through the annulus is laminar.

Figure 11 gives a procedure for computing the interstage annulus leak rate for an MSF plant.

If heat delivered through the brine heater were allowed to leak through all stage partitions of a flash plant to the final heat reject stage, there would be a production from this component of 1 lb of product per 1000 Btu of heat. The heat that does not leak produces R lb of product per 1000 Btu, where R is normally called performance ratio. There is a loss in the amount of regenerative use of heat proportional to the quantity of leakage multiplied by the stage ΔT through which the leakage occurs.

To a first approximation, the fraction of the total steam availability (or cost) lost by leakage in one stage is

$$f = \frac{q_m \, \Delta T_m \, (R - 1)}{Q(\mathrm{FR})R} \,, \qquad (1)$$

where

f = fraction of steam availability (or cost) lost in one stage (m) relative to the useful steam availability (or cost) at the brine heater;

q_m = interstage leakage at stage m, lb/hr;

Q = steam condensed at brine heater, lb/hr;

ΔT_m = stage temperature decrement, °F;

FR = flashing range of brine, °F;

R = evaporator performance ratio, lb per 1000 Btu.

[10] W. R. Snyder and G. A. Goldstein, "An Analysis of Fully Developed Laminar Flow in an Eccentric Annulus," *A.I.Ch.E. J.* 11, 462 (1965).

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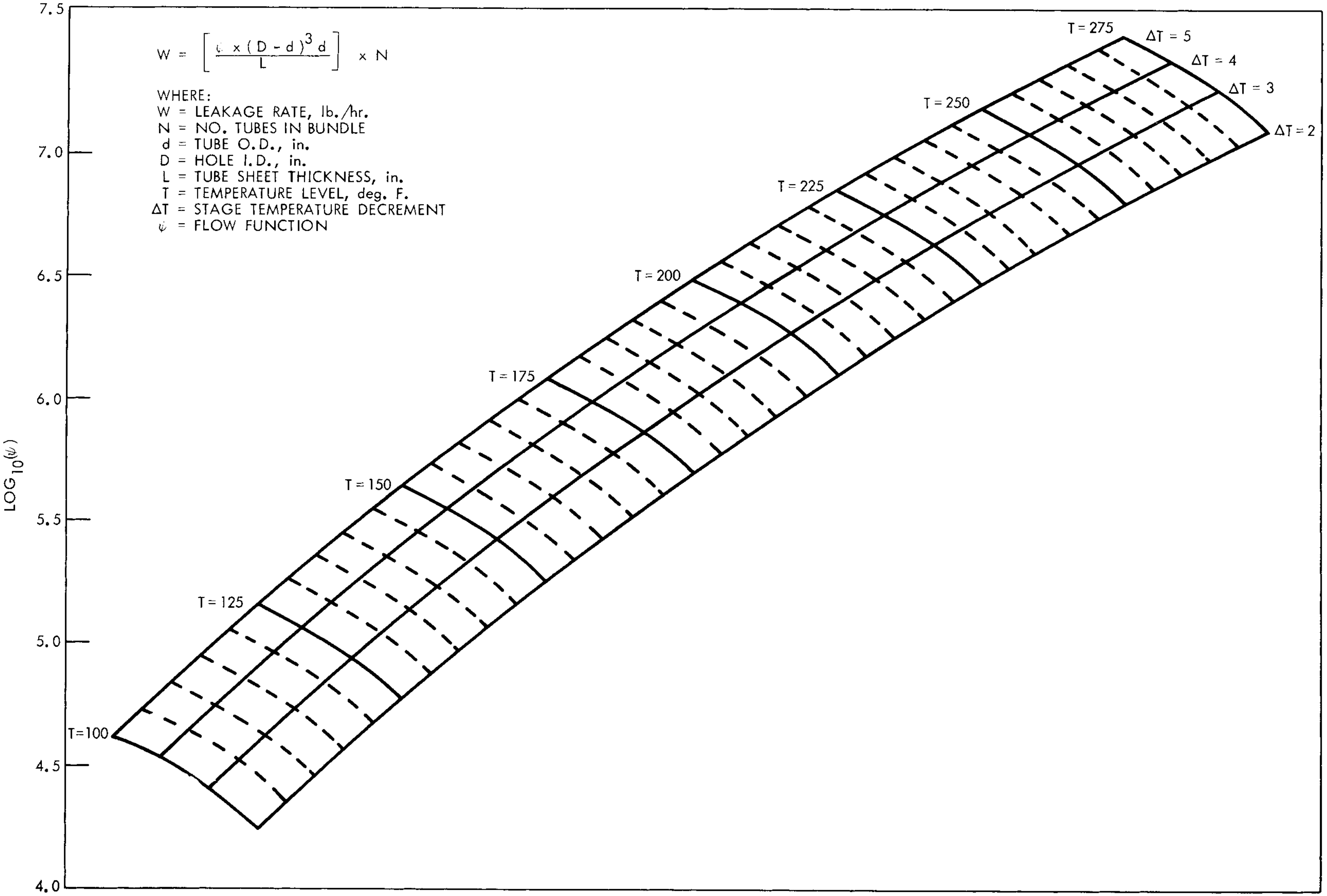

Fig. 11. Interstage Leakage vs Temperature, Temperature Decrement, and Annulus Geometry.

The break-even seal investment per tube is obtained by equating the annual leakage loss in stage m to the annual cost of seals as follows:

$$(\mathrm{BEI})NA = fC ,$$

or

$$(\mathrm{BEI}) = \frac{fC}{NA}, \tag{2}$$

where

(BEI) = break-even investment, \$/seal;

C = cost of steam at brine heater, \$/year;

A = annual charge rate, 1/year;

N = total number of tubes.

We have calculated f and break-even investment for a typical evaporator, which has design parameters shown in Table 6. Figure 12 presents a plot of f as a function of stage brine temperature and stage position. Figure 13 presents a plot of the break-even investment for seals as a function of brine temperature and stage position. Imposed on this plot is the most likely installed seal cost. It is seen here that for our "typical plants" the first 27 stages would require seals.

Table 7 gives the change in leakage fraction (f) and break-even investment (BEI) for plants with design characteristics which vary from the reference conditions of Table 6. Table 8 gives some typical results for various design characteristics.

Table 6. Assumptions Regarding the Typical Plant

Product rate	50 Mgd
Performance ratio	10 lb per 1000 Btu
Maximum brine temperature	250°F
Blowdown temperature	90°F
Flashing range	160°F
Total number of stages	53
Stage temperature decrement	3°F
Tube outside diameter	0.750 in.
Tube sheet hole diameter	0.758 in.
Tube sheet thickness	0.750 in.
Exhaust steam cost	10¢/MBtu
Annual charge rate	0.05185
Plant load factor	0.9
Total number of tubes	40,026
Cost of a seal installed	\$0.084
Annular clearance	0.004 in.

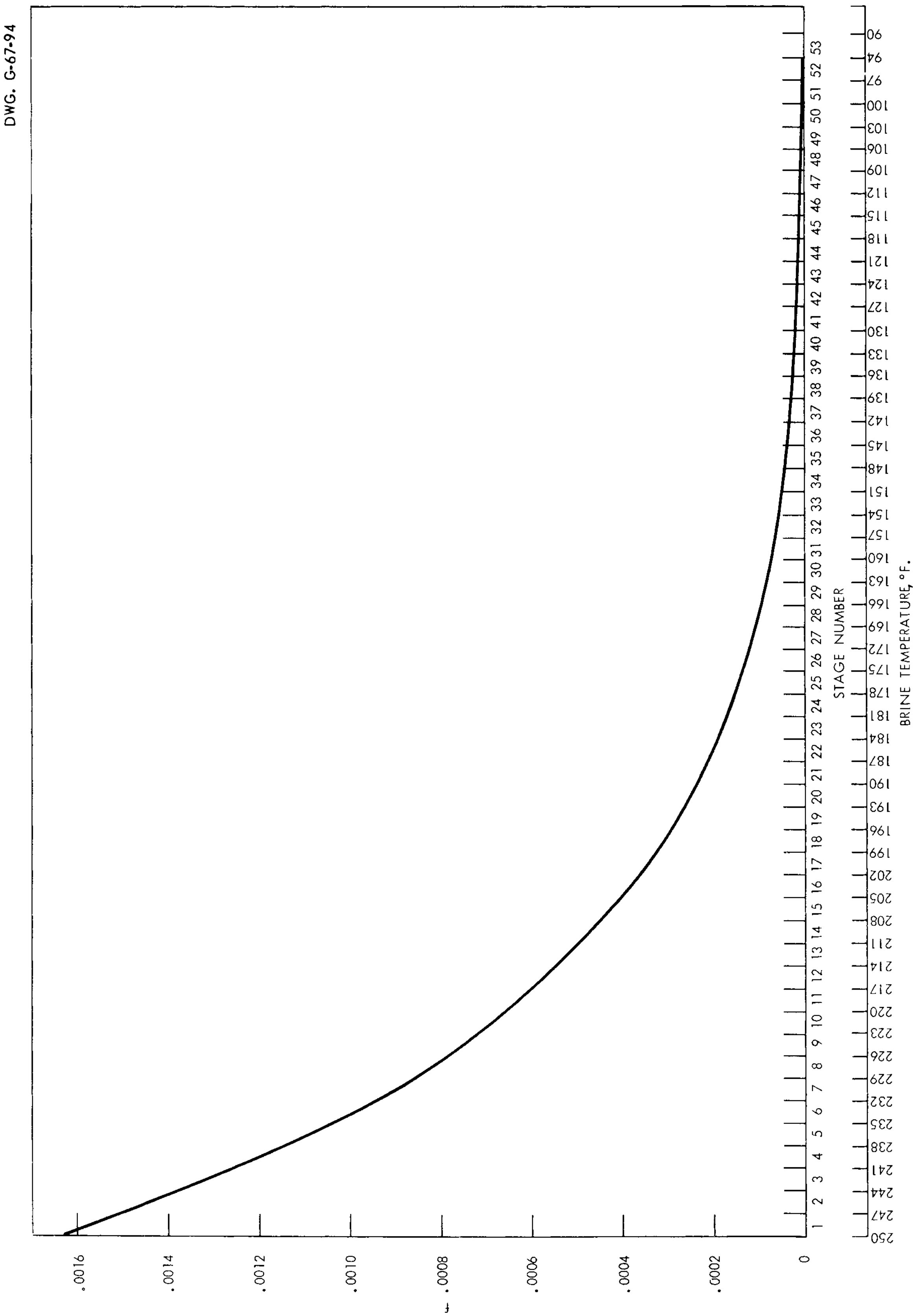

Fig. 12. Steam Availability Loss in Each Stage.

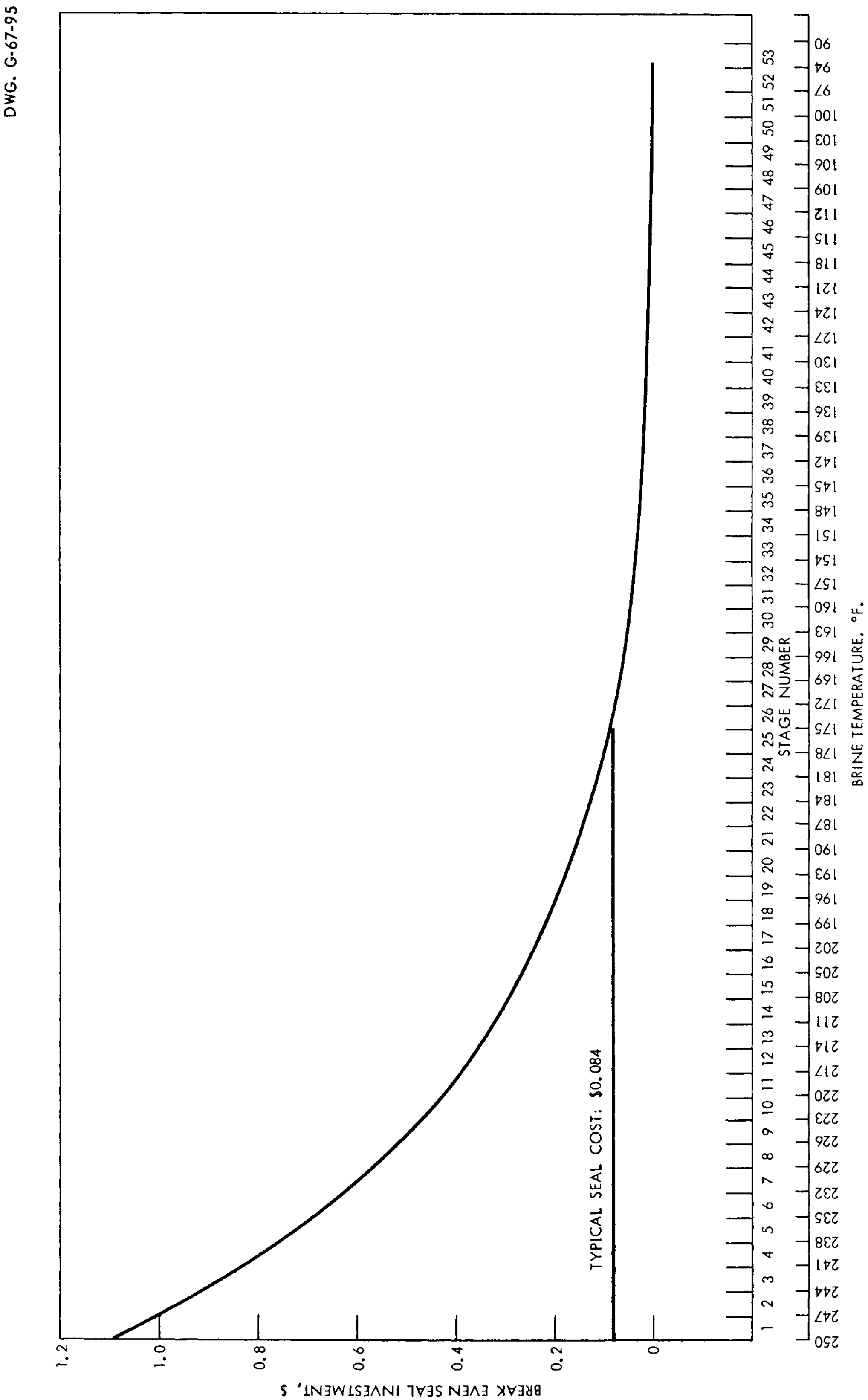

Fig. 13. Break-Even Sealing Investment.

Table 7. Effect of Design Parameters on Leakage and Seal Investment[a]

Parameter	Correction Factor for Change in Parameter (Multiply by Indicated Ratio)		Limits of Correlation
	To Correct f in Fig. 12	To Correct (BEI) in Fig. 13	
Tube diameter, 0.75 in.	$d/0.75$	$d/0.75$	0.5 to 1.0 in.
Diametral clearance, 0.008 in.	$\left(\frac{D-d}{0.008}\right)^3$	$\left(\frac{D-d}{0.008}\right)^3$	0 to 0.010 in.
Tube sheet thickness, 0.75 in.	0.75/tube sheet thickness	0.75/tube sheet thickness	0.25 to 2 in.
Stage temperature decrement, 3°F	$(\Delta T/3)^2$	$(\Delta T/3)^2$	0 to 4°F
Annual charge rate, 0.05185 per year		$0.05185/A$	
Steam cost at brine heater, 10¢/MBtu.		c/10	
Plant factor, 0.9		factor/0.9	0 to 1
R, 10 lb per 1000 Btu	See Eq. (1)	Eqs. (1) and (2)	5 to 15 lb per 1000 Btu
FR, 160°F	160/(FR)	160/(FR)	100 to 180°F

[a]For the reference conditions of the above table (see also Table 6) the total annual steam leakage in a 50-Mgd module without seals would be $24,270 (1.78¢ of the total steam cost). The cost of seals for 27 stages is $90,778, corresponding to an annual cost of $4707. The steam leakage with seals is $3491 per year, assuming seals reduce 90% of the leakage in the stages where installed.

Table 8. Use of Seals in Several Typical MSF Designs

Steam Cost at Brine Heater (¢/MBtu)	Stage Temperature Decrement (°F per stage)	Annular Clearance (in.)	Number of Stages Where Seals Are Justified[a] for Seal Costs of –	
			2.6¢ Each	8.4¢ Each
5.5	3.0	0.002	12	0
5.5	3.0	0.004	31	22
5.5	4.0	0.002	18	6
5.5	4.0	0.004	36	27
10.0	3.0	0.002	18	6
10.0	3.0	0.004	36	27
10.0	4.0	0.002	24	12
10.0	4.0	0.004	41	31

[a]Based on a 53-stage plant.

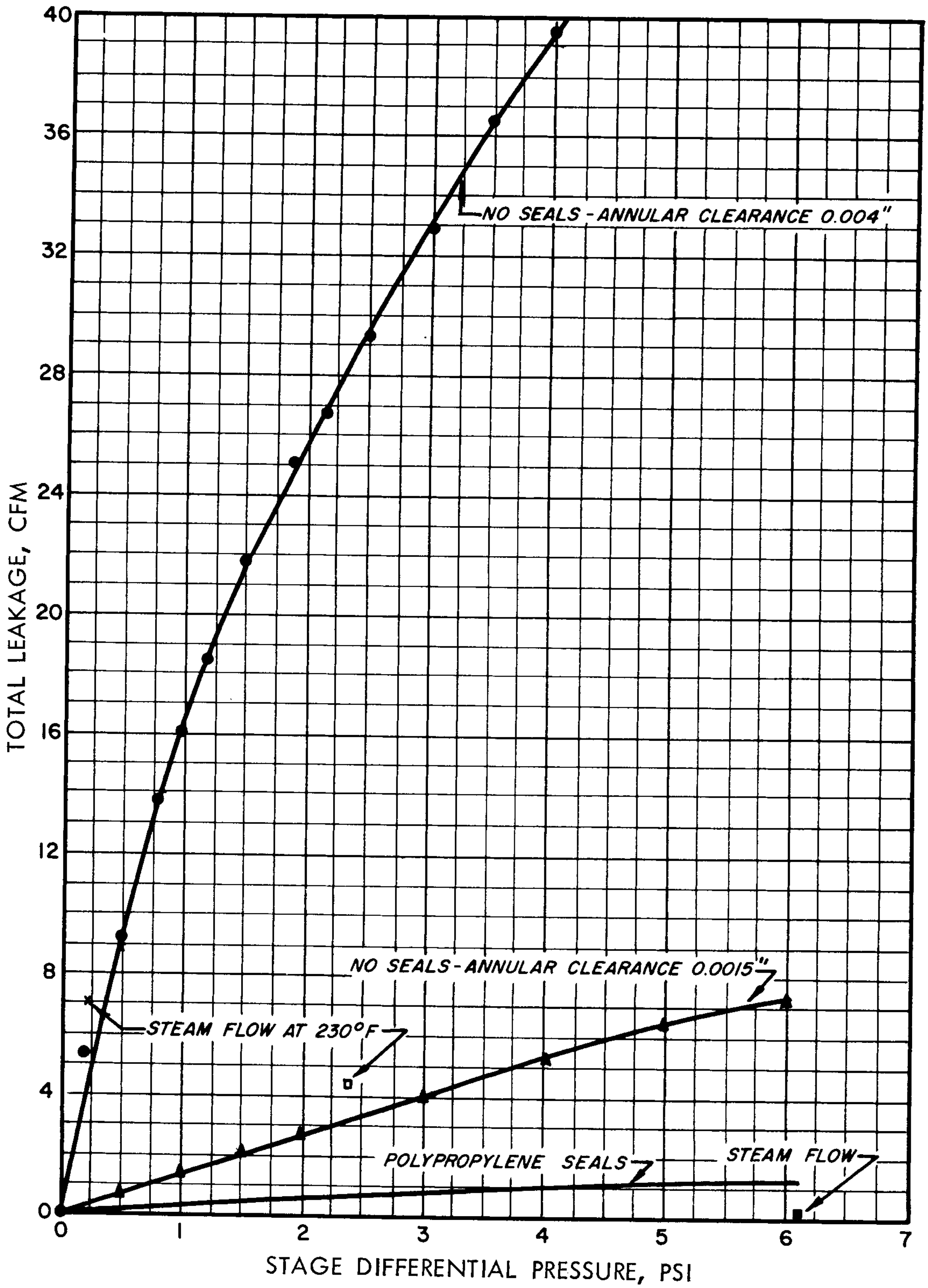

Fig. 14. Leak Tests for a 40-Tube Array of 0.75-in. Tubes.

2.3.4 Tube Leak Tests

A series of tests was performed to verify the calculated leakages and to test the effectiveness of various types of sealing devices.

The test facility consisted of a steam condensing chamber equipped with removable intermediate and end support sheets to accommodate forty $\frac{3}{4}$-in. condensing tubes and a separate condensing coil on the inside of the shell. When the tubes were used, it was anticipated that "wet" leakage would occur because some of the leakage steam would condense in the annuli and that some sealing would occur because of this action. When the coil was used, it was anticipated that "dry" leakage would occur with no steam condensed in the annuli. Thus, no sealing due to condensate in the annuli could be expected. Instrumentation was provided to measure flows, temperatures, and pressures.

Tests were run with steam leakage at about 212°F and with air at 70°F. Annular clearances varied from 0.0015 to 0.0040 in., and differential pressures were 0 to 7 psi. Some typical results are plotted in Fig. 14.

The test results correlated within ±20% of the predictions of Fig. 11. The "wet" leakage was about the same as "dry" leakage within experimental scatter.

Steam leakage through polypropylene ferrules with an interference fit relative to the tube (seal diameter 0.750, tube diameter 0.752) was about $\frac{1}{15}$ of the leakage through a 0.0015-in. annulus. Air leakage through the same seals was about $\frac{1}{5}$ of the leakage through a 0.0015-in. annulus (see Fig. 14).

2.3.5 Seal Designs

Several methods of sealing that appeared to have potential for low cost were evaluated. They are described below.

Plastic Ferrule Inserts (Fig. 15). – This method should provide good sealing at low cost. The major disadvantages of the method are that (1) the inside diameter of the insert must be matched with the fabricated outer diameter of the tube and (2) in some cases the tube spacing might have to be increased to accommodate the seals.

Seals of this type were machined from polypropylene and tested in the leakage facility (Sect. 2.3.4). While dismantling the test vessel following the steam leakage tests on the design 1 polypropylene seal, considerable sticking between the seal and the tubing was encountered. This could be troublesome in an MSF plant if tubes stuck to seals instead of sliding freely.

Additional testing of seal materials is required to support the choice of any plastic for long life in this application.

Based on information from plastics suppliers, the most promising materials are probably ethylene-propylene synthetic rubbers (Du Pont Nordel or similar) or moisture-resisting nylon.

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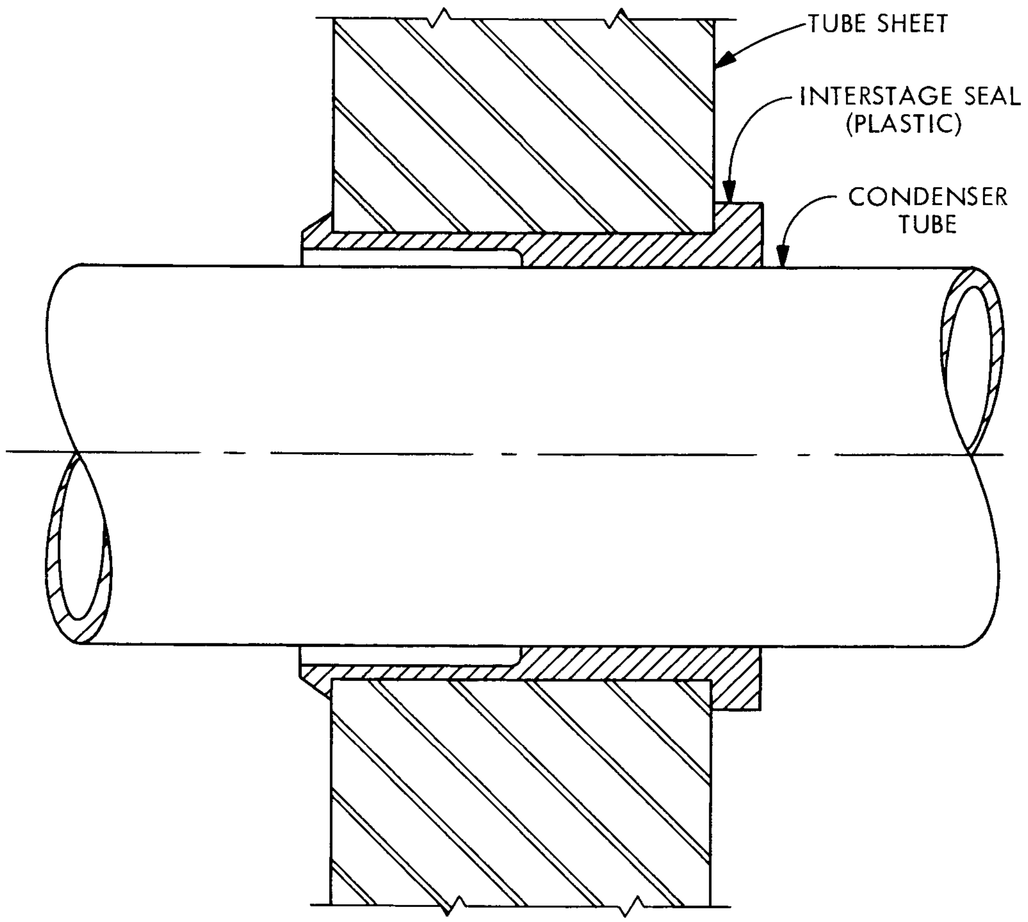

Fig. 15. Seal Design 1. Tube diameter, 0.750 in.; hole diameter, 0.875 in.

Clip Inserts (Figs. 16 and 17). – Clip insert seals were developed to eliminate the disadvantages of the plastic ferrule inserts.

With the design of Fig. 16, sealing at the support plate is dependent upon the four metal retaining clips holding the face of the seal tight against the support plate. The tapered end is designed to provide sealing for tubing that varies in size over a 0.010-in. range (0.750 ± 0.005 in.).

Several seals were molded from polypropylene, Teflon, and nylon, but tests showed that any material rigid at room temperature was not suitable. The seal material must be flexible at room temperature for ease of tube insertion.

Seals were therefore molded from silicone rubber (Silastic 503RTV), polyvinyl chloride (PVC), Hypalon, and Nordel. The silicone seal was too elastic and pulled loose from the metal retainer

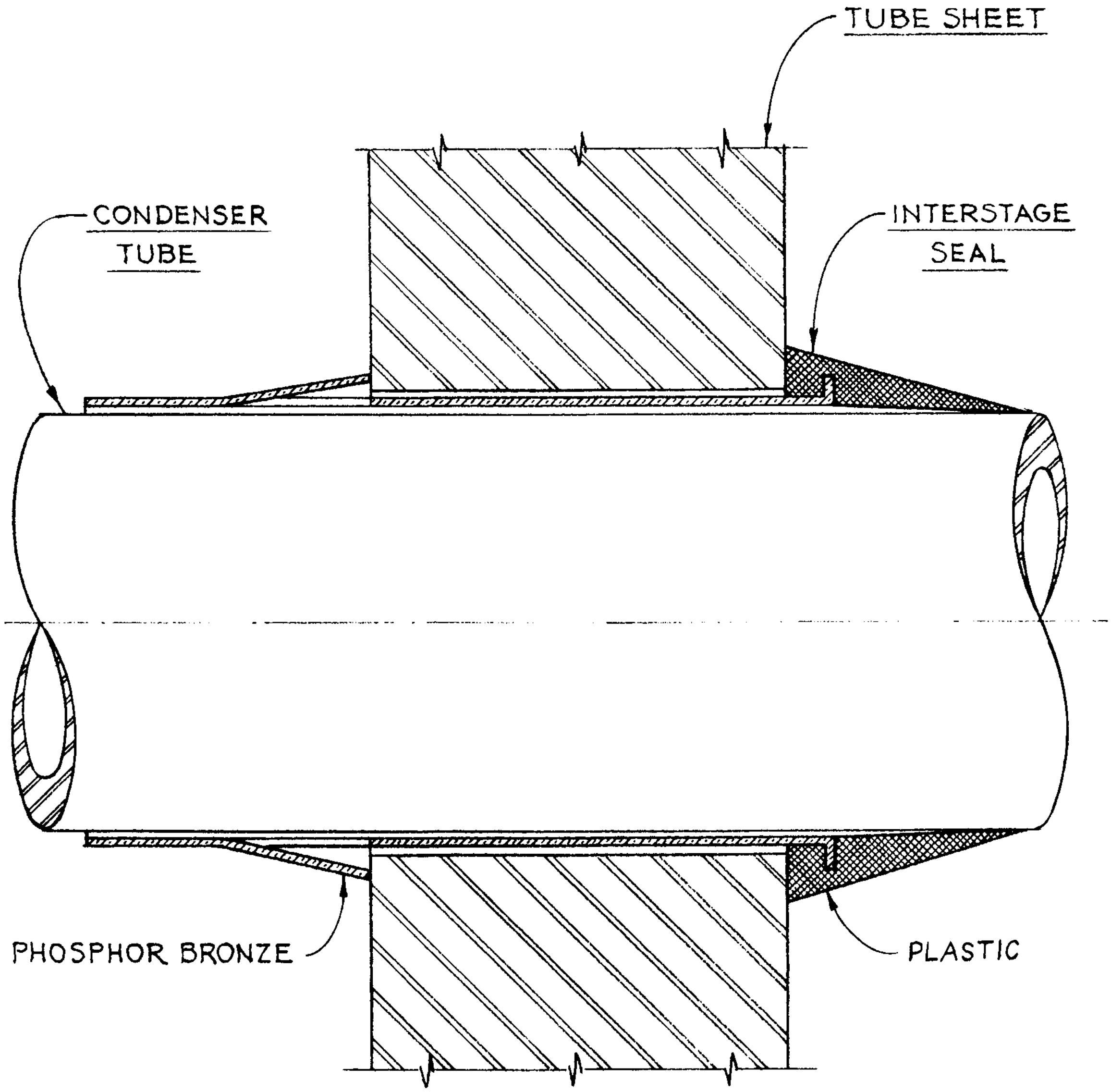

Fig. 16. Seal Design 2. Tube diameter, 0.750 in.; hole diameter, 0.875 in.

when the tube was inserted. Other silicones might do a more effective job, but this particular silicone was not satisfactory. The PVC seal was satisfactory for tube insertion, but considerable leakage occurred during the air test – primarily because of excessive leakage at the support plate. As the seal was pressed against the plate to allow the retaining clips to spring out, the PVC tended to move away from the metal retainer. This demonstrated the need to bond any elastomer seal to the metal clip. The Hypalon seal was bonded to the metal clip with Pliobond adhesive for a series of air tests. These tests showed that the Hypalon seal did an effective job of sealing.

When the test chamber was dismantled it was found that the Hypalon seal had been pulled apart at the thinnest portion of the seal and that the Nordel seal had been pulled loose where it

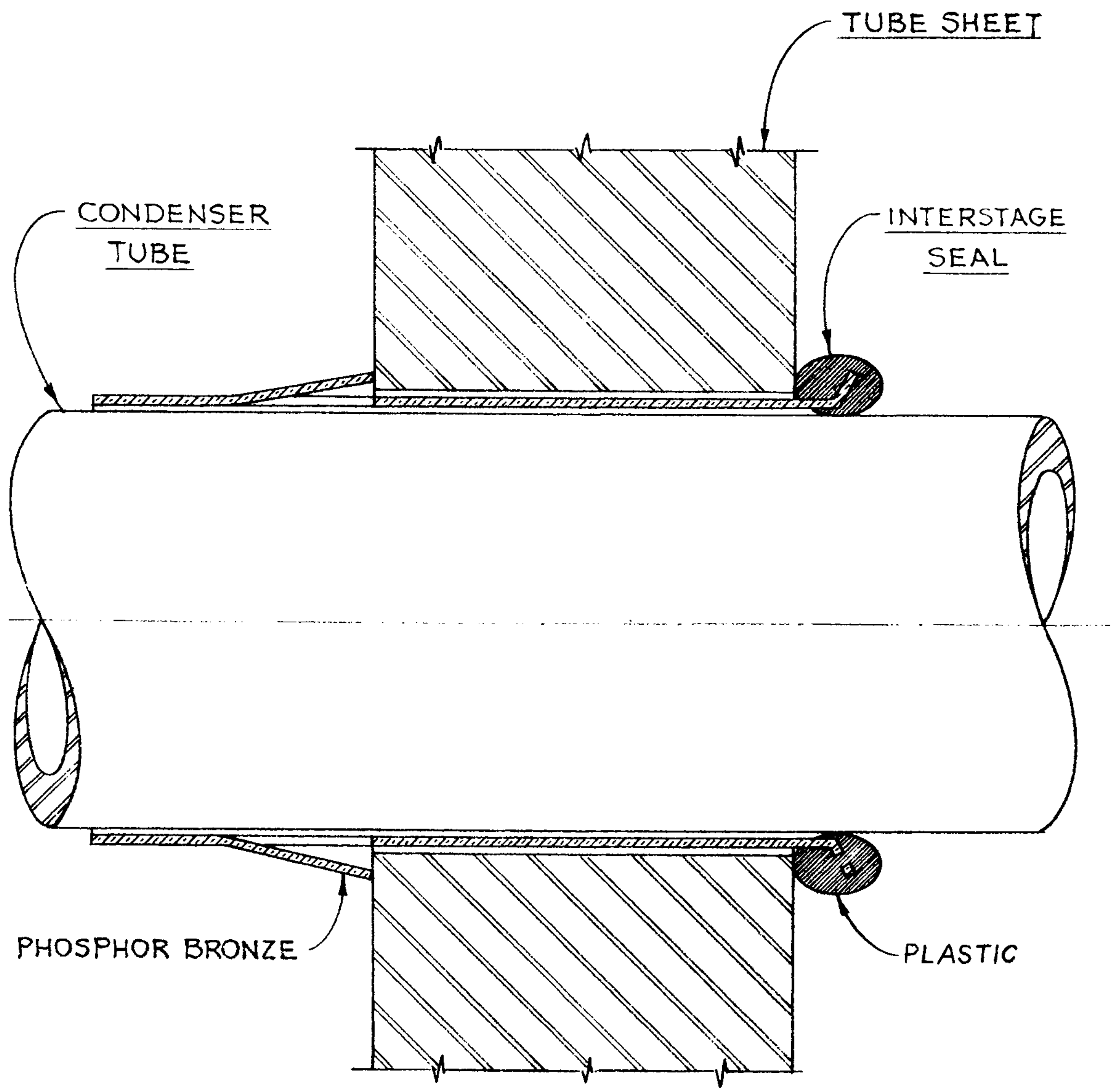

Fig. 17. Seal Design 2A. Tube diameter, 0.750 in.; hole diameter, 0.875 in.

was bonded to the metal retaining clip. These occurrences were attributed to the differential steam pressure pressing the seal against the tube, thus resulting in the noted occurrences when the tube was moved. This suggests the possibility that any elastomer could cause similar problems in this or similar designs of seals.

The seal configuration shown in Fig. 17 might be sturdier than that of Fig. 16. However, seals of this type were not fabricated because of lack of time and funds.

Because of the sticking noted earlier between the tube and the design 1 polypropylene seal, tests were conducted to compare the effort required to slide tubes through design 2 Hypalon, Nordel, fluorothene, nylon, polypropylene, and Teflon seals under simulated operating conditions.

The tests were conducted using the vapor-leakage test equipment at a vessel steam pressure of 6 psig and a temperature of 227°F. The effort required to move the tube, in the order of increasing difficulty, was as follows:

Material	Estimated Factor of Moving Difficulty
Teflon	1.0
Nylon	1.25
Nordel	2.75–3.0
Fluorothene	3.0–3.5
Hypalon	3.25–3.70
Polypropylene	6.5–7.5 (unsatisfactory)

Expansion of the Tube Against the Tube Sheet. – As an alternative to the use of plastic seals, a hydraulically actuated expander could expand each tube locally at the tube support. Similar tube and support plate materials would be used, thus avoiding electrochemical corrosion. The support plate could be quite thin ($\frac{1}{8}$ in.) since the tubes would provide reinforcement. Thus, the material and drilling costs for intermediate support plates would be reduced appreciably. The entire intermediate tube support would have to slide, and the tube sheet edges would have to be sealed by a bellows or some close-fitting arrangement.

A disadvantage of the above method is that neither sections nor entire lengths of individual tubes can be replaced. As leaks develop, the tubes would be plugged and the entire plant retubed if necessary. In some of the other methods of sealing, individual tubes could be replaced if desired.

Minimum Annulus. – Drilling of the tube sheet holes 0.002 in. larger than the outside diameter of the tube to reduce the annular opening to 0.001 in. should reduce the leakage to a point where seals would not be economical. However, the outside diameter of the tubes would have to be controlled closely in order to make the method work. Normal ASTM tolerance on tubing diameter is ±0.003 in. from nominal, while the proposed OSW specification gives ±0.005 in. In some cases the holes could be drilled to match the tube diameter.

Based on the tests reported in Sect. 2.2, only short (up to 60 ft) tube designs could use such an approach.

2.3.6 Evaluation of Sealing Materials

Inquiries were made to manufacturers where the literature suggested a possible sealing material that would suit the application. These included manufacturers of both plastic and elastomer materials. The following is a summary of the findings:

Plastics. – *Polypropylene.* – There are apparently no available polypropylene formulations that are flexible enough at room temperature and that will withstand a 250° steam environment for long periods of time. Certain polypropylenes (high heat stabilized) are used and give good service in 180 to 200°F water in laundry equipment, but their service life in an MSF plant is questionable. In addition, the rigidity at room temperature and the sticking experienced in the test program are negative factors. The Dow Chemical Company and Allegheny Plastics stated that none of the polypropylene materials would give a 30-year service life in an MSF plant.

Nylon (Polyamide Resins). – Most high-temperature nylon materials are stabilized to resist heat but not moisture. No commercial nylon resins were found on the market that would withstand both heat and moisture for long periods of time. Du Pont has suggested an experimental resin, Zytel FE 2403, as being the most promising for the tube-seal application. The material should have good characteristics with respect to tube sliding but is quite rigid at room temperature.

Teflon (Polytetrafluoroethylene). – Teflon will withstand heat and moisture very well and has good tube sliding qualities. The material is less rigid at room temperature than some of the other plastics but is still too rigid for a design 2 seal. Material costs are high compared with some of the other plastics. Because of the incompatibility of Teflon and titanium (crevice corrosion), Teflon probably could not be used if the plant were tubed with titanium tubing.

Fluorothene (Polychlortrifluoroethylene). – Fluorothene will withstand heat and moisture very well. However, the material is quite rigid at room temperature, and material costs are very high.

Rubber Elastomers. – *Hypalon (Chlorosulfonated Polyethylene).* – Hypalon will withstand the environment for a limited time, but Du Pont felt that Nordel would give a longer service life.

Viton Fluoroelastomer (Copolymer of Vinylidene Fluoride and Hexafluoropropylene). – This material will withstand the temperature, but the leaching action of the steam makes it unsuitable for an MSF plant environment.

Nordel (Ethylene-Propylene-Diene Polymer). – This material was recommended by Du Pont as their most promising elastomer for withstanding the high-temperature, wet environment. This material has been used successfully in high-pressure steam (150 lb and 358°F) applications. Du Pont would not guarantee a 30-year life in an MSF plant.

Silicone. – Silicone rubber will withstand the temperature in an MSF plant, but the seal parts exposed to the steam will have certain ingredients leached out, and the useful service life of the silicone rubber will thus be shortened. The Union Carbide Silicones Division suggested that a silicone rubber with a Teflon coating be considered. The silicone should withstand the temperature and provide the required flexibility. The Teflon coating will withstand the environment, provide the required protection for the silicone, and allow the tube to slide freely through the seal. The seal would be relatively expensive, but less expensive than a full Teflon seal. This combination of materials should be satisfactory for a copper-nickel-tube plant, but not for a titanium-tube plant.

2.3.7 Cost Comparison of Seals

The estimated costs of design 1 (Fig. 15) and design 2 (Fig. 16) type seals made from several sealing materials are presented in Table 9.

A cost analysis of Nordel seal inserts for a 250-Mgd plant is presented in Table 10.

Table 9. Cost Comparison of Seals

Material	Specific Gravity	Cost per Seal (cents)						
		Design 1[a]			Design 2[b]			
		Material	Installation	Total	Metal Clip	Sealing Material	Installation	Total
Polypropylene	0.905	1.55[a]	1.0	2.55	6.0	1.37[b]	1.0	8.37
Nylon	1.15	2.67	1.0	3.67	6.0	1.93	1.0	8.93
Teflon	2.1	14.0	1.0	15.0	6.0	7.6	1.0	14.6
Fluorothene	2.3	22.2	1.0	23.2	6.0	11.7	1.0	18.7
Hypalon	1.2	1.93	1.0	2.93	6.0	1.56	1.0	8.56
Viton	1.85	23.7	1.0	24.7	6.0	12.4	1.0	19.4
Nordel	0.86	1.51	1.0	2.51	6.0	1.36	1.0	8.36
Silicone	1.2	5.59	1.0	6.59	6.0	3.4	1.0	10.4
Silicone with 10 mils of Teflon	1.3	6.32	1.0	7.32	6.0	3.8	1.0	10.8

[a]Based on 5 g of polypropylene per seal.
[b]Based on 2.5 g of polypropylene per seal.

Table 10. Cost of Sealing 250-Mgd Plant

	Cost
Seal inserts, design 2 type seal for 27 stages at \$0.084 per seal installed	\$426,747
Cost of leakage steam,[a] capitalized at 5.5% per year, assuming 90% sealing in stages 1 through 27 (\$11,550 per year); no sealing in stages 28 through 53 (\$5910 per year)	270,090
Total cost (seals + leakage)	\$696,837
Water cost	0.05¢ per 1000 gal

[a]For the plant containing five modules as detailed in Table 6, which uses steam for 10¢/MBtu.

2.3.8 Conclusions

The loss associated with interstage vapor leakage through the annular openings between the tubes and the intermediate support plates in MSF plants is dependent upon several variables, which include the annular clearance, the number of tubes, the stage temperature, the stage temperature decrement, the cost of steam to the brine heater, the value of the leakage steam, the plant load factor, and the annual charge rate. By assigning numerical values to each of these variables, the break-even investment for sealing devices can be determined from equations that have been developed. The break-even investment can then be compared with the cost of sealing devices to determine whether the cost of sealing is warranted.

In general, it will be found that sealing devices are not needed where the annular clearances between the tube and the intermediate support plate holes are 0.001 in. or less and that sealing devices can be afforded for a few high-temperature stages where the annular clearances are 0.002 in. or larger. With rigid control of the tubing outside diameter, the intermediate support plates can be drilled 0.002 in. larger than the tube to provide an annular clearance of 0.001 in., and the cost of sealing for plants with short (30- to 60-ft) tubes can be eliminated. This method would not work for plants with long (300-ft) tubes. Plastic ferrule inserts will work with long tubes.

If the tubing size is allowed to vary over a size range of ±0.005 in., the intermediate support plates would have to be drilled to accommodate the largest size tube, and sealing devices would be required to provide sealing over the entire range of annular clearances (0.002 to 0.007 in.). Under these conditions, sealing can be provided by expanding the tube against the tube support or by providing a seal insert with an elastomer seal ring (similar to design 2).

The most common seals used commercially at present, metal ferrules and rubber O-rings, are not necessarily the most promising for large plants. Additional testing is needed of various plastics and of seal designs which incorporate plastics in the MSF plant environment. If a long-tube design using corrosion-resistant materials were developed, additional work on the tube expansion method would be justified.

2.4 DEMISTERS

One of the problems of seawater distillation equipment is separation of vapor from the brine in which it is generated. Traditionally, this has been accomplished in two steps: (1) separation of the vapor from the bulk of the brine at a brine-vapor interface, followed by (2) separation of entrained droplets of brine from the stream of vapor in a device interposed in the vapor stream for this purpose. These devices are known in the industry by various names; the term "demister" is used here. Hook-vane and plate-type demisters have been used widely on shipboard distillation plants. More recently, woven wire mesh demisters have been used successfully on both shipboard and land-based plants. Distillate purities of the order of 1 ppm total dissolved solids are regularly achieved with properly designed combinations of disengaging surface, demister installation, and evaporator configuration. For large land-based plants of high economy ratio, existing demisters may not be optimum. The pressure loss suffered by water vapor in passing through the demister is critical for high-economy flash distillation cycles because of the effect of adiabatic expansion on the mean temperature difference across the condenser heat transfer surfaces. On the other hand, distillate purities obtainable with existing demisters are unnecessarily high; product water in the range from 50 to 100 ppm total dissolved solids (tds) would be acceptable for normal municipal use. Finally, present demisters are expensive, and they impose certain restrictions on the design of evaporator vessels, which, if removed, could lead to economies in construction.

The objective of this study is to evaluate demisters specifically adapted to distillation plants of 50- to 250-Mgd capacity. The effort is aimed at reduction of cost and of pressure drop, within acceptable limits of distillate purity and with established methods of creating the primary disengagement of vapor from brine.

The study includes the following aspects:

1. The literature on demisters which are now being used or have been used or are seriously proposed for use in seawater distillation equipment was surveyed.
2. Examinations of the most successful or promising designs and configurations for reducing cost and pressure losses are to be made. Design factors studied will include materials, fabrication methods, methods of installation in evaporators, and effect on evaporator design.

2.4.1 Review of Literature and Industry Practice

Literature Survey. – A literature survey was made to obtain information on demisters that might be used for removal of entrainment in seawater evaporators. Several articles were found that covered the processes by which entrainment occurs; these include splashing, foaming, heat flux density, depth of bubble generation, bubble diameter, surface tension, viscosity, and velocity of the rising vapor. Information was obtained that covered various types of liquid entrainment removal systems. Very little information was located that could be applied directly to the design of desalination plants. Of all the literature studied, the article by O. H. York and E. W. Poppele[11] on wire mesh mist eliminators was the most informative and useful.

A wire mesh entrainment separator can be installed without difficulty in process equipment such as evaporators, scrubbers, and distillation columns. The only equipment modification required for vessels up to 6 ft in diameter is a support ring fixed to the inside of the vessel to hold the mist eliminator assembly, which consists of the wire mesh sandwiched between a bottom support and top hold-down grid. In larger vessels intermediate supports are required in addition to the annular ring.

Although knitted wire mesh has been used by industry for a broad range of entrainment elimination applications, the volume of fundamental work published regarding their performance characteristics is small.

[11] E. W. Poppele and O. H. York, "Wire Mesh Mist Eliminators," *Chem. Eng. Progr.* 59(6), (1963); reprinted as Bulletin 631 by the Otto H. York Co., Inc.

As generally used, the knitted wire mesh eliminator consists of a bed, usually 4 to 6 in. deep, of fine-diameter wires interlocked by a knitting operation to form a wire mesh pad with a high free volume, usually between 97 and 99%. The wire mesh construction is altered to change performance characteristics such as entrainment removal efficiency, liquid draining rate, gas-handling capacity, and pressure drop. Separation efficiencies greater than 99.9% are obtained over a wide velocity range.

The pressure drop across the wire mesh is sufficiently low, usually less than an inch of water, to be considered negligible for most applications. The pressure drop through wire mesh is influenced by gas and liquid rates. As is the case with packed towers, the pressure drop rises with increased liquid load, since at higher entrainment rates, the lower surface of the mesh more completely fills with liquid, thereby reducing the cross-sectional area available for vapor flow.

Frankel[12] reports tests in a flash evaporator system. Various types of separators were tried, including cyclones, wire mesh separators, impingement-type baffles, etc. On the whole, a suitable adaptation of a hook-vane separator, combining impingement with centrifugal effect, was found to be the most satisfactory. This type of separator has been used for a long time in conventional marine evaporators. In order to avoid joints and machined surfaces inside the evaporator or vessel, the separators were mounted in very simply arranged water seals.

References 13–15 are concerned with entrainment generation and transport. Unfortunately, no references report the quantity and quality of entrainment generated in flash evaporators in a manner useful for design. References 12, 16, 17, and 18 make it clear that the quantity of entrainment is related to flash chamber geometry, throughput, and local properties of the seawater (including the optional use of antifoam).

Industry Practice. – Available information indicates that all U. S. flash evaporator vendors normally provide wire mesh demisters in all stages. The wire is usually Monel or 316 stainless steel.

The conceptual designers[19] used wire mesh demisters with two exceptions. No other types of separators were used.

El-Saie[17] reports that product purities of 80 ppm tds are obtained in the Kuwait evaporators without demisters. On the other hand, demisters are used in all the Curacao evaporators[18] where

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[12] A. Frankel, "Flash Evaporators for the Distillation of Seawater," *Proc. Inst. Mech. Engrs.* **174**(7), 312–24 (1960).

[13] H. E. O'Connell and E. S. Pettyjohn, "Liquid Carry-Over in a Horizontal Tube Evaporator," *Trans. Am. Inst. Chem. Engrs.* **42**, 795–814 (1946).

[14] D. M. Newitt, N. Dombrowski, and F. H. Knelman, "Liquid Entrainment, 1. The Mechanism of Drop Formation from Gas or Vapor Bubbles," *Trans. Inst. Chem. Engrs.* **32**, 244–61 (1954).

[15] F. H. Garner, S. R. M. Ellis, and J. A. Lacey, "The Size Distribution and Entrainment of Droplets," *Trans. Inst. Chem. Engrs.* **32**, 222–35 (1954).

[16] R. S. Silver, "A Review of Distillation Processes for Fresh Water Production from the Sea," a paper presented at the First European Symposium on Fresh Water from the Sea, Athens, June 1962.

[17] M. H. Ali El-Saie, "Water Production Experience of the City of Kuwait," paper No. SWD/30, presented at First International Symposium on Water Desalination, Washington, D.C., Oct. 3–9, 1965.

[18] F. C. A. A. Van Berkel, J. W. Van Hasselet, and J. H. Van Der Torren, "Experiences with Large Sea-Water Flash Evaporators," paper No. SWD/94, presented at First International Symposium on Water Desalination, Washington, D.C., Oct. 3–9, 1965.

[19] *1965 Saline Water Conversion Report*, pp. 178, 233–35, and 277–95, U.S. Government Printing Office Washington, D.C., 1966.

the desired impurity level is below 4 ppm. Demisters must be reinforced in some cases with antifoam addition to the seawater.

The procedures used by the conceptual designers for design of demisters generally are similar to those of M. W. Kellogg.[20] There is a wide variation in the accepted parameters, especially in the amount of entrainment, the preferred velocity in the demister, and the efficiency of the demister.

2.4.2 Design Equations

Sizing demisters requires three separate groups of design information:

1. the entrainment expected from the flash evaporators expressed as a function of the operating and geometrical parameters of the evaporators,
2. the efficiency of liquid removal by the demisters expressed as a function of demister characteristics and operating conditions,
3. the pressure drop across the demisters as a function of demister parameters and operating conditions.

Entrainment Generation. – References 13, 15, and 21 provide data on the amount of entrainment in submerged tube evaporators. These data indicate that at low vapor production rates the amount of entrainment decreases as vapor production increases, while at high production rates entrainment increases. The latter is what one would intuitively expect. The following equations were derived for design use:

$$E = 34{,}100(G^3v^2 \times 10^{-8})^{-0.157} \qquad \text{for } (G^3v^2 \times 10^{-8}) < 347, \tag{1}$$

and

$$= 84(G^3v^2 \times 10^{-8})^{0.870} \qquad \text{for } (G^3v^2 \times 10^{-8}) > 347, \tag{2}$$

where

E = entrainment, lb of liquid per 10^6 lb of vapor,

G = lb of vapor per hour per square foot of evaporator surface,

v = specific volume of vapor, ft^3/lb.

For a typical evaporator, the Point Loma plant of OSW, G varied from 130 to 110 and $G^3v^2 \times 10^{-8}$ from 8 to 1700 in going from the hot end of the plant (230 to 250°F) to the cold end (90 to 100°F). More recent commercial evaporators tend to have higher vapor rates than Point Loma. The above relationship is believed to be quite conservative when applied to flash evaporators. Experimental data should be obtained for various flash chamber configurations to revise predictions of entrainment.

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[20]M. W. Kellogg Co., "Saline Water Conversion Technical Data Book," Issue No. 2, OSW 700 to 720 (1965).

[21]C. L. Carpenter, doctoral dissertation, Brooklyn Polytechnic Institute, 1951.

Demister Efficiency. – The equation for liquid removal from the vapor is:

$$E_o = E_i e^{-\alpha t} , \tag{3}$$

where

E_o = entrainment on downstream side of demister,

E_i = entrainment on upstream side of demister,

e = base of natural logarithms = 2.718+,

t = demister thickness (in.),

and alpha is given by

$$\alpha = 0.666 \ln U - 0.1873 \tag{4}$$

for the type 421 demister,[21] where U = superficial vapor velocity at demister, fps.

Recommended Vapor Velocity. – York recommends a velocity

$$U = 0.35[(\rho_g - \rho_l)/\rho_g]^{0.5}$$

for general design.

A modified version of this equation was derived from Fig. 8 of Poppele and York[11] for use in the present design calculations:

$$\ln \left[U^2 \frac{a\rho_g(u_l)^{0.2}}{g\rho_g\epsilon^3\rho_l} \right] = -4.995 - 0.7252 \ln \left(\frac{L}{G} \sqrt{\frac{\rho_g}{\rho_l}} \right)$$

$$-0.03016 \left[\ln \left(\frac{L}{G} \sqrt{\frac{\rho_g}{\rho_l}} \right) \right]^2 , \tag{5}$$

where

a = specific surface area of demisters, ft^2/ft^3;

ρ_g = vapor density, lb/ft^3;

ρ_l = liquid density, lb/ft^3;

u_l = liquid viscosity, centipoises;

ϵ = void fraction of demister, dimensionless;

L/G = ratio of liquid to vapor incident on demister, that is, $E \times 10^{-6}$;

g = acceleration of gravity = 32.2 ft/sec^2.

The recommended velocity is greater than that which M. W. Kellogg[20] indicates has the highest efficiency and is also about 130% of that used at Point Loma.

Pressure Drop. – The pressure drop through the demisters is calculated as the sum of a pressure drop caused by friction against the wires of the demister plus a pressure drop due to the liquid load in the demister. These two drops are calculated from Figs. 5 and 7 of ref. 11 by the equations

$$\frac{\Delta P_d \, g \epsilon^3}{l_a \rho_g U^2} = 1.5685 \left(\frac{\rho_g U}{a u} \right)^{-0.2747} , \tag{6}$$

where

ΔP_d = dry pressure drop, lb/ft^2,

l = demister thickness, ft,

u = viscosity of vapor, lb ft^{-1} sec^{-1};

and

$$L = [(L/G) U \rho_g \times 3600] , \tag{6a}$$

$$U_r = U \sqrt{\frac{\rho_g}{\rho_l - \rho_g}} , \tag{6b}$$

$$\Delta P_l = 0.012 e^{\phi} , \tag{7}$$

where

$$\phi = [10.51 - 1.287 \log L + 0.3784 (\log L)^2] \cdot (U_r - 0.075) \tag{7a}$$

for the type 421 demister where

ΔP_l = pressure drop due to liquid load, in. H_2O.

The total pressure drop is then calculated as

$$\text{DPT} = (0.193 \, \Delta P_d) + \Delta P_l \text{ in. } H_2O, \text{ or} \tag{8}$$

$$\Delta P_T = \Delta P_d + (\Delta P_l / 0.193) \text{ lb/ft}^2 . \tag{9}$$

2.4.3 Calculated Entrainment Separator Designs

Table 11 gives some calculated results using the above equations. A typical MSF evaporator would use about 650 ft^2 of 4-in. wire mesh per million gallons of product per day.

Table 11. Demister Design[a]

TEMPERATURE 90.0800 DEG F.
VAPOR PRESSURE 0.7000 PSIA

G	E	SPA	DP	PPM/CR	DT
20.0	21833.0	2.219	0.633	0.565	1.021
40.0	15751.7	2.158	0.667	0.411	1.076
60.0	17579.5	2.177	0.656	0.457	1.057
80.0	37247.4	2.341	0.577	0.954	0.933
100.0	66685.5	2.505	0.517	1.698	0.836
120.0	107323.3	2.668	0.467	2.726	0.757
140.0	160481.6	2.829	0.426	4.071	0.691
160.0	227396.7	2.988	0.390	5.765	0.634
180.0	309237.6	3.145	0.359	7.836	0.584
200.0	407117.1	3.301	0.332	10.314	0.541
220.0	522100.6	3.455	0.309	13.225	0.502
240.0	655212.9	3.608	0.287	16.596	0.468
260.0	807442.3	3.760	0.268	20.450	0.437
280.0	979746.0	3.911	0.251	24.813	0.409
300.0	1173052.8	4.061	0.236	29.708	0.384

TEMPERATURE 98.2400 DEG F.
VAPOR PRESSURE 0.9000 PSIA

G	E	SPA	DP	PPM/CR	DT
20.0	23519.3	1.992	0.610	0.607	0.795
40.0	16968.4	1.933	0.644	0.442	0.838
60.0	14318.5	1.902	0.664	0.368	0.864
80.0	24662.8	2.001	0.605	0.636	0.788
100.0	44154.8	2.131	0.545	1.128	0.711
120.0	71062.5	2.260	0.495	1.808	0.647
140.0	106260.5	2.388	0.454	2.699	0.593
160.0	150567.4	2.514	0.418	3.820	0.547
180.0	204757.1	2.639	0.387	5.191	0.507
200.0	269566.5	2.763	0.360	6.832	0.471
220.0	345701.2	2.886	0.335	8.759	0.440
240.0	433839.4	3.008	0.314	10.991	0.411
260.0	534635.9	3.129	0.294	13.543	0.386
280.0	648724.2	3.249	0.276	16.431	0.363
300.0	776719.4	3.368	0.260	19.672	0.341

[a]Demisters are 4-in.-thick York 421 type.
G is pounds per hour of vapor per square foot of brine surface.
E is entrainment, pounds of brine per million pounds of vapor.
SPA is square feet of demister required per thousand pounds of vapor per hour.
DP is pressure drop across demister in inches of water.
PPM/CR is ppm carried over divided by concentration ratio of brine.
DT is equivalent temperature rise in degrees F.

Table 11 (continued)

TEMPERATURE 107.9200 DEG F.
VAPOR PRESSURE 1.2000 PSIA

G	E	SPA	DP	PPM/CR	DT
20.0	25609.7	1.763	0.584	0.660	0.595
40.0	18476.5	1.707	0.617	0.480	0.629
60.0	15264.4	1.677	0.637	0.399	0.649
80.0	15386.0	1.678	0.636	0.402	0.648
100.0	27546.2	1.776	0.576	0.709	0.588
120.0	44332.7	1.875	0.527	1.132	0.538
140.0	66291.2	1.973	0.486	1.688	0.496
160.0	93932.2	2.070	0.450	2.387	0.460
180.0	127738.8	2.167	0.419	3.242	0.429
200.0	168170.5	2.262	0.392	4.265	0.401
220.0	215667.5	2.357	0.367	5.467	0.375
240.0	270653.0	2.450	0.345	6.859	0.353
260.0	333535.4	2.543	0.325	8.451	0.332
280.0	404710.1	2.636	0.306	10.253	0.314
300.0	484560.6	2.728	0.289	12.275	0.296

TEMPERATURE 117.9900 DEG F.
VAPOR PRESSURE 1.6000 PSIA

G	E	SPA	DP	PPM/CR	DT
20.0	27881.0	1.563	0.558	0.717	0.445
40.0	20115.1	1.510	0.591	0.521	0.472
60.0	16618.2	1.482	0.611	0.433	0.487
80.0	14512.4	1.464	0.625	0.380	0.498
100.0	17201.3	1.487	0.608	0.448	0.485
120.0	27683.6	1.562	0.559	0.712	0.446
140.0	41395.6	1.637	0.518	1.058	0.413
160.0	58656.1	1.712	0.482	1.494	0.385
180.0	79766.7	1.786	0.451	2.028	0.361
200.0	105014.3	1.859	0.424	2.667	0.339
220.0	134673.9	1.932	0.399	3.417	0.319
240.0	169009.6	2.004	0.376	4.286	0.301
260.0	208276.6	2.076	0.356	5.280	0.284
280.0	252721.7	2.147	0.337	6.405	0.270
300.0	302584.5	2.218	0.320	7.667	0.256

TEMPERATURE 129.6200 DEG F.
VAPOR PRESSURE 2.5000 PSIA

G	E	SPA	DP	PPM/CR	DT
20.0	30629.5	1.371	0.530	0.786	0.285
40.0	22098.1	1.322	0.563	0.571	0.303
60.0	18256.4	1.295	0.583	0.474	0.314
80.0	15943.0	1.278	0.597	0.416	0.321
100.0	14352.4	1.265	0.607	0.376	0.327
120.0	16442.3	1.282	0.593	0.428	0.319
140.0	24586.3	1.337	0.553	0.634	0.297
160.0	34837.9	1.393	0.517	0.892	0.279
180.0	47376.2	1.448	0.486	1.209	0.262
200.0	62371.7	1.503	0.459	1.588	0.247
220.0	79987.6	1.558	0.434	2.034	0.234
240.0	100380.8	1.612	0.411	2.549	0.222
260.0	123702.9	1.666	0.390	3.139	0.210
280.0	150100.4	1.719	0.371	3.807	0.200
300.0	179715.6	1.772	0.354	4.557	0.191

Table 11 (continued)

TEMPERATURE 141.4800 DEG F.
VAPOR PRESSURE 3.0000 PSIA

G	E	SPA	DP	PPM/CR	DT
20.0	33563.0	1.210	0.504	0.860	0.236
40.0	24214.5	1.163	0.537	0.624	0.251
60.0	20004.9	1.139	0.556	0.518	0.260
80.0	17469.9	1.122	0.569	0.454	0.266
100.0	15727.0	1.110	0.580	0.410	0.271
120.0	14432.8	1.100	0.589	0.378	0.275
140.0	14811.0	1.103	0.586	0.387	0.274
160.0	20986.6	1.145	0.551	0.543	0.258
180.0	28539.8	1.186	0.520	0.733	0.243
200.0	37573.2	1.227	0.493	0.961	0.230
220.0	48185.1	1.269	0.468	1.229	0.219
240.0	60470.1	1.310	0.445	1.540	0.208
260.0	74519.5	1.350	0.424	1.895	0.199
280.0	90421.5	1.391	0.405	2.297	0.190
300.0	108262.0	1.431	0.388	2.749	0.181

TEMPERATURE 162.2400 DEG F.
VAPOR PRESSURE 5.0000 PSIA

G	E	SPA	DP	PPM/CR	DT
20.0	39012.0	0.989	0.461	0.997	0.140
40.0	28145.7	0.948	0.493	0.723	0.150
60.0	23252.7	0.925	0.512	0.599	0.156
80.0	20306.2	0.910	0.526	0.525	0.160
100.0	18280.3	0.899	0.536	0.474	0.163
120.0	16776.0	0.891	0.545	0.436	0.165
140.0	15601.2	0.884	0.552	0.407	0.168
160.0	14650.2	0.878	0.558	0.383	0.169
180.0	13859.6	0.872	0.564	0.363	0.171
200.0	16323.6	0.888	0.547	0.425	0.166
220.0	20933.9	0.914	0.523	0.541	0.159
240.0	26271.1	0.939	0.500	0.676	0.152
260.0	32374.8	0.965	0.480	0.830	0.146
280.0	39283.4	0.990	0.461	1.004	0.140
300.0	47034.2	1.016	0.443	1.200	0.135

TEMPERATURE 193.2100 DEG F.
VAPOR PRESSURE 10.0000 PSIA

G	E	SPA	DP	PPM/CR	DT
20.0	47829.8	0.760	0.406	3.120	0.069
40.0	34507.5	0.724	0.437	1.982	0.074
60.0	28508.5	0.705	0.456	1.527	0.077
80.0	24895.9	0.692	0.469	1.272	0.080
100.0	22412.1	0.683	0.479	1.105	0.081
120.0	20567.9	0.675	0.487	0.986	0.083
140.0	19127.5	0.669	0.494	0.896	0.084
160.0	17961.5	0.664	0.500	0.825	0.085
180.0	16992.2	0.659	0.506	0.767	0.086
200.0	16169.6	0.655	0.510	0.719	0.087
220.0	15459.7	0.652	0.515	0.679	0.087
240.0	14839.0	0.649	0.519	0.644	0.088
260.0	14290.0	0.646	0.522	0.613	0.089
280.0	13799.8	0.643	0.526	0.586	0.089
300.0	15205.4	0.651	0.516	0.664	0.088

Table 11 (continued)

TEMPERATURE 213.0300 DEG F.
VAPOR PRESSURE 15.0000 PSIA

G	E	SPA	DP	PPM/CR	DT
20.0	53880.9	0.656	0.374	6.494	0.045
40.0	38873.2	0.623	0.405	4.088	0.049
60.0	32115.2	0.605	0.423	3.132	0.051
80.0	28045.6	0.593	0.436	2.597	0.053
100.0	25247.6	0.585	0.446	2.249	0.054
120.0	23170.0	0.578	0.454	2.000	0.055
140.0	21547.3	0.572	0.461	1.813	0.056
160.0	20233.9	0.567	0.467	1.666	0.057
180.0	19142.0	0.563	0.472	1.546	0.057
200.0	18215.2	0.559	0.477	1.447	0.058
220.0	17415.6	0.556	0.481	1.363	0.058
240.0	16716.3	0.553	0.485	1.291	0.059
260.0	16097.8	0.551	0.489	1.228	0.059
280.0	15545.6	0.548	0.492	1.172	0.060
300.0	15048.6	0.546	0.495	1.123	0.060

TEMPERATURE 227.9600 DEG F.
VAPOR PRESSURE 20.0000 PSIA

G	E	SPA	DP	PPM/CR	DT
20.0	58630.1	0.592	0.352	11.000	0.034
40.0	42299.5	0.561	0.383	6.882	0.037
60.0	34945.9	0.544	0.400	5.252	0.038
80.0	30517.6	0.533	0.413	4.344	0.040
100.0	27473.0	0.525	0.423	3.753	0.040
120.0	25212.2	0.518	0.431	3.333	0.041
140.0	23446.6	0.513	0.438	3.016	0.042
160.0	22017.3	0.508	0.444	2.767	0.042
180.0	20829.2	0.505	0.449	2.565	0.043
200.0	19820.8	0.501	0.454	2.398	0.043
220.0	18950.7	0.498	0.458	2.256	0.044
240.0	18189.7	0.495	0.462	2.135	0.044
260.0	17516.7	0.493	0.466	2.029	0.045
280.0	16915.9	0.491	0.469	1.936	0.045
300.0	16375.0	0.488	0.472	1.853	0.045

TEMPERATURE 250.3300 DEG F.
VAPOR PRESSURE 30.0000 PSIA

G	E	SPA	DP	PPM/CR	DT
20.0	66048.3	0.514	0.321	23.375	0.022
40.0	47651.5	0.485	0.351	14.499	0.024
60.0	39367.5	0.470	0.369	11.010	0.025
80.0	34378.9	0.460	0.381	9.073	0.026
100.0	30949.0	0.452	0.391	7.816	0.027
120.0	28402.2	0.447	0.399	6.925	0.027
140.0	26413.2	0.442	0.406	6.254	0.028
160.0	24803.1	0.437	0.412	5.727	0.028
180.0	23464.6	0.434	0.417	5.301	0.029
200.0	22328.6	0.431	0.422	4.948	0.029
220.0	21348.4	0.428	0.426	4.650	0.029
240.0	20491.2	0.425	0.430	4.394	0.029
260.0	19733.1	0.423	0.433	4.172	0.030
280.0	19056.2	0.421	0.436	3.977	0.030
300.0	18446.9	0.419	0.439	3.803	0.030

Table 12. Cost of 4-in. York 421 Demisters

Material	Cost per Square Foot		
	Mesh	Top and Bottom Grids	Total
316 SS	$10.90	$6.30	$17.20
Monel	12.30	8.05	20.35

2.4.4 Cost of Demisters

Meaningful cost information was obtained only for Otto H. York Company wire mesh demisters (See Table 12).

Possibilities for reducing the cost of mesh demisters were explored. However, it was found that the cost of wire is about 80% of the price of the woven pads, and the only possibilities appear to be lower-density pads or cheaper materials.

The York Company advises that plastic or ceramic-fiber demisters have been built and work well. The costs of these demisters are not promising at present.

The usefulness of lower-density pads remains to be explored in MSF application.

Hook-vane demisters are known to be more compact than mesh demisters but are heavier and therefore more expensive. There is, however, the possibility of extruded plastic hook-vane demisters, which might be more economical than metal mesh. Such hook-vane separators would be especially useful if space were restricted.

2.4.5 Conclusions

Flash evaporator designs require demisters in at least some of the lower-temperature stages. Wire mesh demisters are the type normally used and would be adequate for 50-Mgd plants.

Several experimental approaches are likely to provide lower-cost demister designs. First, the liquid entrainment from flash evaporators should be measured under various operating conditions with particular emphasis on low-temperature stages where the vapor velocity is high. The amount of entrainment, in general, will be a function of the weir geometry, the mixing efficiency in the chamber, the size of the chamber, the freeboard, etc. Thus it appears that model tests of proposed weir and flash-chamber geometries should be used to predict the entrainment. It is possible that one could design chamber geometries that generate little entrainment.

Second, several demister designs should be tested under actual operating conditions to accurately measure the separation efficiency and pressure drop. The development of simple, sturdy, inexpensive demisters with high efficiency should be emphasized. Moreover, the study should consider those facets of demister design that will maximize the entrainment removal and minimize the pressure drop.

Third, life tests of plastic separators should be started to evaluate the applicability of lower-cost materials.

2.5 TUBING FABRICATION

Tubing used in evaporators, usually of copper alloy, represents up to half the equipment cost. There appears to be an opportunity to reduce this cost significantly by applying new specifications, production methods, and inspection procedures to large production runs.

There is currently increasing interest in the possibilities of premium materials such as titanium. Their potential for high corrosion resistance to seawater makes possible the use of tubing with much-thinner-than-conventional walls. Means must be found for fabricating thin-walled premium tubing in long tubes (100 to 350 ft), as it is anticipated that the most economical evaporator designs may call for tube bundles of great length.

The objectives of the study are as follows:

1. to ascertain the present state of the art of fabrication of copper-nickel and titanium alloy tubing, including methods, limitations (as on length, diameter, wall thickness, grade of material, etc.), and costs; from this to identify possible means of reducing costs and/or improving the product; and to predict whether tubing development can ultimately result in substantial savings in evaporator costs in comparison with the costs obtained with present materials;
2. to stimulate the industry to develop improved methods along the lines stated above.

The scope of the work reported here covers the following:

1. an evaluation of the state of the art,
2. an investigation of quality control standards and minimum specifications for evaporator tubing.

Work is being continued, under contract to industry, to develop low-cost tubing.

2.5.1 State-of-the-Art Review

Tubing used in condensers has been largely of the seamless, drawn type in lengths of 30 to 100 ft. Lengths up to 200 ft are encountered occasionally. Tube materials presently in general use include steel, stainless steel, aluminum, copper, copper alloy, and graphite. Depending on operating and site conditions, tube performance life will usually vary from 10 to 30 years. Recent improvements in raw material costs and manufacturing practice have also brought into consideration "premium" tube materials such as titanium and Hastelloy C, which might provide a tube life in excess of 30 years. However, because of the present low-volume market, these materials still cost significantly more than copper-base alloys.

Seamless 90-10 CuNi and aluminum-brass tubing are the most common materials used in seawater distillation condensers. The 70-30 CuNi is recommended for polluted seawater. A 1.5-Mgd MSF evaporator using titanium tubes is being operated at St. Croix, Virgin Islands.

The 50-Mgd and larger conceptual designs used tubing of $\frac{5}{8}$ to 1 in. outside diameter with the following distribution of materials selection:

90-10 CuNi	7
70-30 CuNi	4
Aluminum brass	4
Titanium	5

The current fabrication methods being used for producing seamless copper-nickel tubing are (1) tube reducers, (2) draw benches, and (3) bull blocks. The major portion of the seamless tubing is made on single draw benches. There are a few multitube draw benches in existence which can draw five tubes simultaneously in lengths up to 200 ft. Lately, several of the tube producers have been investigating more aggressively the bull block method for producing tubes. Studies indicate that this technique has potential for producing low-cost seamless tubes. Several producers make welded copper-alloy tubing by the high-frequency process. Only a small amount of tubing is made by this process currently; however, this method is being considered seriously as an advanced and more economic fabricating concept by many of the major producers. One method which would require considerable development but which might have the lowest ultimate cost is the continuous casting and cold forging method.

The potentially higher corrosion resistance of materials such as titanium in seawater should make possible the use of tubes with very thin walls (since little or no corrosion allowance would be necessary). New tube fabrication techniques would be required to put these materials in a competitive position. One method that is being considered is the welding of thin strip material into tubes by the high-frequency resistance welding process. This process has been used for the past few years in making ornamental stainless steel and aluminum tubular material at speeds up to 1000 fpm. In addition, this process has been used for making thin-wall stainless steel tubing for steam surface condensers (stainless steel tubing now comprises about 30% of the steam surface condenser tubing market). The application of this welding process to Hastelloy C and titanium tubing should prove to be economical if the required development work is carried out.

2.5.2 Tubing Specification

A set of specifications has been drafted to cover the manufacture of tubes for MSF evaporators. The objective of these specifications is to permit economies of manufacture by taking into account the actual requirements of the desalination plants without unduly lowering tube quality. The specifications, therefore, are somewhat less stringent than the ASTM specifications for condenser tubes. The specifications cover copper-nickel and titanium as both seamless and welded tubing.

Some of the areas where these specifications represent a relaxation of ASTM B-111 are the following:

1. The outside diameter of $^5/_8$- to 1-in. tubing is permitted to vary by ±0.004 from the specified diameter; ASTM B-111 allows ±0.003 for $^1/_2$- to $^3/_4$-in. tubing sizes and ±0.0035 for $^3/_4$- to 1-in. sizes.
2. Welded tubing is permitted.
3. Tubing is supplied in the as-drawn or as-welded temper; minimum tensile strength of 70-30 CuNi is 65,000 psi and of 90-10 CuNi, 50,000 psi.
4. A 100% inspection with a suitable nondestructive device is required.
5. Hydrostatic test is not required.
6. Splice welds are permitted in limited numbers.
7. Tube lengths over 100 ft are included in the specification.

Comments from user groups have been incorporated into the specifications to the extent possible. The copper-nickel specification has been submitted to the ASTM B5 committee for review. The titanium specification will be submitted to ASTM after fabricating experience has been gained on thin-wall seamless and welded tubing.

2.5.3 Quality Control Investigations

Nondestructive Tube Tests. – Nondestructive tube-testing equipment suitable for 100% on-line testing of long tubes has been investigated for both seamless and welded types at fabricating speeds of 200 to 500 fpm.

Ultrasonic testing does not appear to lend itself very well to such applications since the customary rotating of the tube in conjunction with a stationary ultrasonic probe is not feasible with long tubes. One company markets a rotating-probe-type instrument, but the maximum testing speed is only 150 fpm. While makers of ultrasonic testers feel their testers are quite adequate for detecting defects in seamless tubes, they are hesitant to recommend them for welded tubes unless the outside and inside weld flash is removed flush with the tube surfaces; otherwise, serious interfering background reflections occur.

Eddy-current tube testers appear to have the capability of testing at the required high speeds. While such devices in their conventional differential output form appear adequate for detecting the types of flaws that are occasioned in seamless tubes, they normally are unable to detect lengthy discontinuities such as long cracks in seam welds or weld faults resulting from either overheat or underheat conditions in welded-type tubes. Surveys disclosed two eddy-current instruments which seem to possess the full capability for reliable high-speed testing of long welded tubes. These instruments provide not only a differential output signal to reflect pinholes, inclusions, and short hairline cracks but also an absolute output signal to reflect long cracks or open seam welds. The manufacturers of these types of tube testers are Microwave Instrument Company of Corona Del Mar, California, and Magentic Analysis Corporation of Mt. Vernon, New York.

Samples of defective welded-type copper-nickel tubes are being obtained for testing in an effort to make a conclusive determination of the suitability of these instruments for testing welded tubes.

2.5.4 Long Tube Splicing

Development work has been done with two companies (Astro-Arc in Sun Valley, California, and Merrick Engineering in Nashville, Tennessee) that market split-type weld heads for fusion butt welding of tubes. Excellent weldments have been obtained with titanium tube samples, but some porosity was encountered with the copper-nickel joints. The major portion of their past work has been directed at titanium with the aircraft industry, and these companies have had no previous experience with copper-nickel materials. It is felt that successful copper-nickel joints can be obtained by optimizing the weld parameters, such as arc length, shielding, current, and speed. In addition, a slight modification in the weld head design is needed to improve alignment of tube ends.

2.5.5 Tube Development Contracts

We have been successful in stimulating tube producers to make development studies to investigate new, low-cost fabrication methods. In cases where industry was not interested in doing development work on their own, we have issued a selected group of cost-assistance contracts to encourage industry participation to develop the required information.

One company is developing a low-cost fabrication concept for welded thin-wall (0.016- to 0.020-in.) titanium tubing with their own funds. Another company is developing a fabrication concept for thin-wall (0.012- to 0.016-in.) Hastelloy C tubing with their own funds.

Cost-assistance contracts have been let (1) to develop fabrication concepts for thin-wall (0.012- to 0.020-in.) seamless titanium tubing, (2) to develop bull-block fabrication methods for forming 90-10 and 70-30 CuNi tubing, (3) to develop a fabrication concept for welded 90-10 (0.035-in.-wall) CuNi tubing, and (4) to demonstrate the welding of thin-wall titanium tubes to titanium-clad tube sheets. The output of these contracts is as follows: (1) the economics of shipping target-size coils from the factory to the site and of shipping long straight lengths to the plant site, (2) the demonstration of the feasibility of the different fabrication methods, (3) projected tube prices to 1970, (4) the development of on-line, reliable inspection methods, and (5) production of tubing for OSW corrosion tests.

2.5.6 Tubing Prices

Copper-Based Alloys. – Current prices (1965) were obtained (Table 13) from five major tube producers on 50- and 100-ft tubes based on quantities of 60,000 ft and over. Only two of these producers would make any commitment on projected 1970 prices for longer tubes up to 360 ft in length. Their opinions were that in 1970 the unit price of a 360-ft tube would be the same as it is today for a 100-ft tube, if splicing is permitted to minimize scrap losses.

The difference in tubing prices for optimally spaced orders or for orders placed with one or several companies has been investigated. The tube manufacturers have stated that the tubing prices for a large order will be the same regardless of whether the order is placed with one or three companies. They have further indicated that a large order of, say, 95,000,000 ft of $^5/_8$-in.-OD copper-nickel tubing can be produced within two years with present industry capacity if the order is split among three major tube producers. Industry has ample capacity for producing large quantities of the primary thick-wall shells, but they would be short on draw-bench facilities for drawing shells into finished-size tubes.

An analysis was made of the components of 90-10 CuNi tubing prices in order to evaluate the possibilities of lower tube prices in the future (Table 14). Whereas the present market indicates about 60¢/lb for the fabrication steps in producing copper-nickel tubes, some future fabrication systems may be 20¢/lb. The projected reduction in price would be about 38% if raw materials costs were constant at 41¢/lb.

Table 13. Current Condenser Tube Prices for Large MSF Plants[a]

Material	O.D. (in.)	Wall (in.)	Tube Price (dollars/ft)						
			Company A, 100-ft Lengths	Company B, 100-ft Lengths	Company C, 100-ft Lengths	Company D		Company E	
						50-ft Lengths	100-ft Lengths[b]	50-ft Lengths	100-ft Lengths[b]
Aluminum brass	$\frac{5}{8}$	0.049	0.288	0.288	0.288	0.241	0.279	0.261	0.289
	$\frac{3}{4}$	0.049	0.351	0.351	0.351	0.296	0.347	0.316	0.352
	1	0.049	0.472	0.472	0.472	0.396	0.456	0.361	0.472
	$1\frac{1}{4}$	0.049	0.698	0.579	0.622	0.502	0.569	0.503	0.579
90-10 cupronickel	$\frac{5}{8}$	0.042	0.315	0.314	0.314	0.267	0.305	0.287	0.315
	$\frac{3}{4}$	0.042	0.378	0.377	0.377	0.322	0.366	0.342	0.378
	1	0.042	0.501	0.501	0.501	0.426	0.485	0.379	0.502
	$1\frac{1}{4}$	0.042	0.720	0.613	0.613	0.536	0.602		0.614
	$\frac{5}{8}$	0.035	0.310	0.274		0.227	0.275	0.226	
	$\frac{3}{4}$	0.035	0.369	0.329		0.273	0.329	0.274	
	1	0.035	0.477	0.435		0.359	0.435	0.359	
	$1\frac{1}{4}$	0.035				0.451	0.536	0.451	

[a]Quotations are current prices for 50- and 100-ft seamless tubes in quantities of 60,000 ft and over.

[b]Lengths of up to 360 ft could be supplied at the same price by 1970.

Table 14. Breakdown of 90-10 Cupronickel Tube Prices

0.75-in. OD × 0.035-in. wall, 100 ft or over

	Current Production Concepts		Advanced Fabrication Concepts	
	Seamless Tube	Welded Tube	Seamless Tube, Cast and Reduced[a]	Welded Tube[b]
Price of raw material,[c] ¢/lb	0.409	0.409	0.409	0.409
Price of fabricating strip, ¢/lb		0.371[d]		0.143
Price of fabricating tube, ¢/lb	0.608	0.109	0.206	0.099
Total price of tube, ¢/lb	0.995[e]	0.889	0.615	0.651
Price, ¢/ft	0.318	0.284	0.197	0.208

[a]Reduced to finished size in a single-step cold-forging operation from a continuous-cast shell.
[b]Strip fabricated from continuous-cast slab.
[c]Copper at 36¢/lb and nickel at 85.25¢/lb.
[d]17% discount from list price.
[e]20% discount from list price.

The strong impact of competitive factors on tube prices is illustrated by an analysis of tubing bids for the West Coast MSF Module (Table 15). The contract conversion price of 43¢/lb for government-supplied metal leads to an effective discount from tubing "list" price of 33% on a 1-million-ft order. If improved fabrication methods were developed, competitive factors would still enter into the "discounts" bid.

Titanium and Hastelloy C. – Several major titanium producers were contacted for estimates of 1970 tube prices. The leading producer of Hastelloy C was similarly contacted. Estimated tube prices are given in Table 16. Prices are quoted both for commercially pure titanium and for the palladium alloy which possesses improved resistance to crevice corrosion. It is likely that commercially pure titanium or an alloy close to it in price will be adequate.

One of the titanium producers has estimated prices considerably below the others, in fact, so low that a 25% contingency over their quotation has been added in making the comparative evaluation. Their low price seems to be justified in light of the large and improved facilities this company has under construction – including new sponge production facilities and a vacuum-anneal, roll-forming facility for continuous production of strip from slabs. This company anticipates that the growth of the current market plus the requirements of the desalination program will reduce titanium mill product costs to a low value by 1970. A degree of price scaling was noted in the projections: a 3 : 1 expansion over the quantity of tubes required for a 50-Mgd module effected a cost reduction of approximately 1.5%; for a 5 : 1 expansion, the reduction was 3.5%.

Table 17 presents a breakdown of titanium and Hastelloy C tube prices derived from vendor information.

Table 15. Analysis of Bids for West Coast Module Tubing[a]

Quotes of September 15, 1966

	Low Bid, Anaconda, F.O.B. Houston[b]		Second Low Bid, Scovill, F.O.B. Houston[c]	
	per lb	per ft	per lb	per ft
Average price of tubes	0.8281	0.2650	0.8889	0.2844
Cost of government metal	0.3888	0.1244	0.3888	0.1244
Shipping cost	0.0144	0.0046	*d*	*d*
Conversion price	0.4249	0.1360	0.5001	0.1600

[a]These bids were for 90-10 cupronickel tubes, 0.75-in. O.D. with 0.035-in. wall thickness. The quotes were for 8600 tubes of 68-ft length, 6500 tubes of 61-ft length, and 6900 tubes of 17-ft length.

[b]Total price of order, $291,089.24.

[c]Total price of order, $312,445.96.

[d]Included in conversion price.

Table 16. Titanium and Hastelloy C Tubing Price Projections (1970)[a]

OD (in.)	Wall Thickness (in.)	Tubing Price (dollars/ft)					
		Company A[b]		Company B		Company C,	Company D,
		Titanium	Ti + 0.15% Pd	Titanium	Ti + 0.15% Pd	Titanium	Hastelloy C
3/4	0.012	0.241	0.295	0.53	0.64		
	0.016	0.303	0.375	0.53	0.66	0.76	0.57
	0.020	0.356	0.445				0.64
	0.025	0.454	0.564				
1	0.012	0.298	0.370	0.70	0.84		
	0.016	0.368	0.465	0.69	0.87	0.81	0.68
	0.020	0.465	0.586				0.77
	0.025	0.568	0.717				
1 1/4	0.012	0.354	0.446	0.86	1.05		
	0.016	0.460	0.581	0.86	1.07	0.91	0.81
	0.020	0.559	0.710				0.93
	0.025	0.695	0.882				

[a]All titanium estimates are for 360-ft tubes produced on site in 10,000,000- to 84,000,000-ft quantities; estimates for Hastelloy C are based similarly but on 12,000,000-ft quantities.

[b]Company A costs include a 25% contingency added by ORNL to the Company A estimates.

Table 17. Breakdown of Titanium and Hastelloy C Tubing Prices

a. Titanium Tubing Prices

	Welded,[a] Current		Seamless, Current		Welded, 1970		Seamless, 1970	
	Ti	+0.2% Pd	Ti	+0.2% Pd	Ti	+0.2% Pd	Ti	+0.2% Pd
Price of raw material, $/lb								
Titanium sponge	1.32	1.32	1.32	1.32	1.00	1.00	1.00	1.00
0.2% Pd		1.20		1.20		1.00		1.00
Price of fabricating strip, $/lb	3.58	4.28			2.78	2.77		
Price of fabricating tube, $/lb	8.70	10.88	15.44	19.27	0.42	0.42	6.34	7.14
Total price of tube, $/lb	13.60	17.68	16.76	21.79	4.20	5.19	7.34	9.14
Price, $/ft	1.22	1.59	1.21	1.57	0.303	0.375	0.53	0.66
Tube dimensions								
OD, in.	0.75	0.75	0.75	0.75	0.75	0.75	0.75	0.75
Wall, in.	0.020[a]	0.020[a]	0.016	0.016	0.016	0.016	0.016	0.016

[a]Welded tubing is not currently listed with less than 0.020-in.-thick walls.

b. Hastelloy C, Welded Tubing Prices

$^3/_4$-in. OD, 0.016-in. wall

	Advanced	Current
Price of raw material, $/lb	1.35	2.35
Price of fabricating sheet, $/lb	0.55	1.26
Price of fabricating tube, $/lb	0.18	4.02
Total price of tube, $/lb	2.08	7.63
Price, $/ft	0.42	1.55

2.5.7 Conclusions

The cost of condenser tubing is usually the largest single cost in large seawater evaporators. Although the industry appears to have the capability of supplying adequate tubing, it appears to be desirable to develop and apply lower-cost fabrication and quality control methods which can be reflected in reduced plant cost and water cost. The areas for development include:

1. thin-wall welded and seamless titanium tubes,
2. thin-wall welded Hastelloy C tubes,
3. welded copper-nickel tubing,
4. advanced methods of producing seamless copper-nickel tubes,
5. methods of on-site (field) fabrication of tubes,
6. nondestructive test methods,
7. specifications which are tailored to the quality of tubing required for seawater evaporators.

All these developments, in combination, should permit reduced tubing prices for large evaporators. This would imply a continuing demand from customers for large plants.

3. Process Components

3.1 DEAERATOR JETS AND BLOWERS

All high-economy seawater distillation plants operate with a major portion of the evaporation and condensation taking place at subatmospheric pressures. Noncondensable gases (primarily oxygen, nitrogen, and carbon dioxide) can enter plants dissolved in the seawater makeup and also by leakage into vessels operating at subatmospheric pressures. If the evaporation and condensation processes are to proceed at maximum rates, accumulation of noncondensable gases must be prevented. It is therefore necessary to provide equipment which will remove such gases from the evaporators and discharge them at atmospheric pressure. In the past, steam-jet air ejectors have been used most commonly; however, water jets and various types of mechanical compressors have been used or suggested. None of the equipment currently available has been designed for use with large distillation plants of 50- to 250-Mgd capacity.

The objectives of the study were as follows:

1. to ascertain the present state of the art of blower and ejector equipment suitable for removal of noncondensable gases from large distillation plants,
2. to evaluate the technical and economic problems in available systems.

3.1.1 Review of Industrial Practice

Commercial Evaporators. – In the majority of existing desalination plants the evacuation systems are equipped with steam-driven ejectors. Condenser evacuation systems for large power stations commonly utilize either steam-driven ejectors or rotary-vane (liquid-sealed) type vacuum pumps. Many other types of removal equipment are being used successfully by the petrochemical industry for various applications. Although industrial experience in this field is extensive, the size of equipment used is small in relation to the operating requirements of a typical 50-Mgd plant.

Evacuation systems normally include precondensers and intercondensers to reduce pumping loads. Both tubular and barometric-type condensers are used; in the former case there is some product recovery.

Foster and Herlihy[22] report considerable difficulty with materials in the Point Loma air ejector system following the use of acid treatment of the brine feed (which evolves CO_2). There

[22] A. C. Foster and J. P. Herlihy, "Operating Experience at San Diego Flash Distillation Plant," paper No. SWD/46, presented at First International Symposium on Water Desalination, Washington, D.C., Oct. 3–9, 1965.

was rapid corrosion and erosion of the steel venturi throats and failure of aluminum brass and 90-10 condenser tubes. Type 316 stainless steel was recommended for tubing, and stainless steels were recommended for other components.

Type 316 stainless steel is currently being used for air ejector systems.

Conceptual Designs of 50-Mgd and Larger Systems. – Conceptual designers[23] varied very widely in their estimates of evacuation loads, in their equipment selection, and in their choice of materials.

Most of the designers selected the steam-driven ejector as the basic evacuation unit, although blowers were also used. The number of stages used varied from two to four; and while some used single, full-capacity ejectors, others used as many as four in parallel.

Both surface-type and barometric-type condensers were used; in some cases product water was used as the coolant.

Most conceptual designers used atmospheric CO_2 degassing ponds or towers to reduce the load on the air ejectors.

The great variety of equipment and design data used by the conceptual designers is indicative of the absence of standardization in the design of large air ejector systems.

3.1.2 Evacuation Loads

Two cases were selected as typical of the capacities that will be required for 50-Mgd distillation plants. On the low-capacity side, Case I is for a plant equipped with an atmospheric decarbonating pond or tower. This plant is also expected to be designed and built with extreme care and good-quality workmanship so that the rate of air inleakage will be at a minimum. On the high-capacity side, Case II is for a plant where all dissolved gases are released in the deaerator. In addition, the air inleakage into the plant is assumed to be rather high. In both cases it is assumed that the air inleakage will be evacuated along with the deaerator gases by a single system of evacuating equipment taking its suction from the lowest pressure in the evaporator system (0.9 psia).

Table 18 gives the gas loads that are anticipated for each of the two limiting cases. In Case I, 70% of the carbon dioxide produced by the addition of sulfuric acid for scale control is released in the decarbonation basin. Other gases are assumed to remain unaffected through the decarbonation process and are evolved in the deaerator along with the balance of the CO_2. The quantities of dissolved gases given in Table 18 correspond to a seawater temperature of 65°F.[24]

The median evacuation loads of the 50-Mgd conceptual designs were 36,000 to 100,000 lb of noncondensables per day, although some estimates were both above and below this range. These estimates are of the same order as those of Table 18.

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[23] *1965 Saline Water Conversion Report*, pp. 178, 233–35, and 277–95. U.S. Government Printing Office, Washington, D.C. (1966).

[24] *Saline Water Conversion Technical Data Book*, Issue No. 2, M. W. Kellogg Co., 1965.

Table 18. Estimated Evacuation Loads for a Typical 50-Mgd Plant

	Case I[a]	Case II[b]
Deaerator pressure, psia	0.9056	0.9056
Deaerator temperature, °F	88	88
Water vapor, lb/day	62,083	156,869
Inleakage air, lb/day	200	2,400
Argon, lb/day	400	400
Nitrogen, lb/day	10,700	10,700
Oxygen, lb/day	6,720	6,720
Carbon dioxide, lb/day	28,500	85,500
Total noncondensables, lb/day	46,520	105,720

[a]Plant with outside degassing pond and low air inleakage.
[b]Plant without outside degassing pond and high air inleakage.

3.1.3 Alternative Designs

There are many equipment choices to be made in the deaerator ejector system. The best choice of equipment for one part of the system may be coupled to choices made elsewhere in the system. Some of these interlocking choices are as follows:

1. blowers vs ejectors,
2. number of stages of blowers and ejectors,
3. barometric precondensers vs surface precondensers vs no precondensers,
4. barometric vs surface intercondensers,
5. barometric vs surface vs no aftercondensers,
6. atmospheric decarbonation vs none.

A study was made of commercial ejector equipment. Analysis of the results indicated that for 50-Mgd plants, steam ejectors or centrifugal compressors were more desirable than rotary-vane vacuum pumps, rotary lobe compressors, water jets, or air jets. The latter group was eliminated from detailed economic evaluation.

In keeping with a design philosophy of simplicity in design and of low maintenance cost, it was decided to use single, full-capacity units of equipment rather than several smaller units in parallel. Blowers can be operated for long periods with minimum maintenance. In addition, the systems (evaporator and evacuation) can tolerate minor malfunctions for moderate periods until the next plant shutdown.

3.1.4 Material Selection

One of the most difficult aspects of this study has been the selection of materials that are suitable in all respects. These factors include corrosion resistance, strength, and durability to give a 30-year life; material cost; ease of fabrication; and experience of prospective manufacturers in working with these materials.

The mixture of gases from the deaerator consists of oxygen, carbon dioxide, water vapor, and traces of entrained seawater containing sodium chloride. The two most versatile and inexpensive materials of construction, carbon steel and cast iron, are subject to severe corrosion when in direct contact with wet carbon dioxide and oxygen.

The next best choice is stainless steel. Type 316 stainless steel is now used in many desalting plant ejector systems, and no failures have been reported to date. Considerable caution is required in the use of stainless steels in chloride-containing environments subject to wetting and drying; a considerable amount of stress corrosion cracking has been experienced in the power and nuclear industries.

Among nonferrous alloys the materials considered were Monel, Inconel, Hastelloy C, titanium, and Karbate (impervious graphite).

Although Monel and Inconel give excellent service in seawater under turbulent conditions, marine organisms may accumulate under stagnant seawater and induce local oxygen concentration cell action to cause pitting-type corrosion. Monel is also subject to corrosion under the environment of wet carbon dioxide. Fabrication limitations eliminated Karbate as the material for some parts of ejectors and heat exchangers. Hastelloy C and titanium seem to have the best potential for use in this service. Since their use is expected to increase in the coming years, the cost of these premium alloys is expected to drop in relation to other materials.

Several commercially available protective coatings were considered. Many of these coatings have given excellent service, but the life expectancy cannot be accurately predicted at high temperatures since no long-term operating data are available. However, the cost advantage of using protective coatings over using premium materials is great, even if the coating has to be reapplied a few times during the life of the plant. For applications where the temperature is low and the conditions are not very erosive, the protective coatings probably can be used advantageously for some of the equipment under consideration. Since the cost of the coatings is a very small fraction of the total equipment cost, it will not influence the ultimate results of this study. The individual plant designer should select the best coatings available at the time of actual plant construction. However, for purposes of this study Heresite was selected as the protective coating. It is a phenol formaldehyde synthetic resin compound.

The following materials of construction are recommended for the subject equipment.

Centrifugal Blowers. – This is expensive equipment which requires precise and complex design work. Investigation reveals that the commercially available blowers of the required size are made mostly of ferrous alloys. Making blowers of special alloys would involve considerable development work. Therefore, a protective coating seems to be the best solution; it is recommended that blowers be constructed of cast iron containing 3% nickel and coated with a Heresite coating suitable for the anticipated temperature level.

Ejectors. – Because of the high velocity, temperature, and erosive conditions involved, a protective coating is not expected to give satisfactory service. Furthermore, the cost of the ejector is small relative to the cost of the steam used to drive the ejector; inefficiencies induced by the wear of nozzles and diffusers are expected to reduce the efficiency of the steam jets. The most wear-resistant materials should be used for this service, namely Hastelloy C or titanium. Type 316 stainless steel is an alternate, but there is more risk of corrosive attack.

Surface Condensers. – The use of protective coatings on the tubes and in the shell is impractical. The heads can be easily coated and thus can be made of carbon steel or cast iron. Since 65°F cooling water is available and the outside film heat transfer coefficient is low due to noncondensables, the tube wall temperature will not be very high. This would suggest that stainless steel tubes could be used, although it is necessary that local hot spots be avoided.

In the cost studies reported later, Hastelloy C was assumed to be the tube material of intercondensers; type 316 stainless steel was used for precondensers.

Barometric Condensers. – Because of the highly erosive nature of falling water, coatings are not expected to last very long on those surfaces which are under the direct impact of water. However, such surfaces will remain cool because of direct contact with cooling water.

Type 316 stainless steel is recommended, but hot spots originating from the steam entry should be eliminated or a Heresite coating could be applied locally.

3.1.5 Unit Costs

The costs used in the study are as follows:

1. Steam ejectors, Fig. 18.
2. Blowers, Fig. 19.
3. Barometric condensers, Fig. 20.
4. Hastelloy C surface condensers, Fig. 21.
5. Stainless steel surface condensers, Fig. 22.
6. Steam, electric power: costs were assumed corresponding to "fossil fuel" and to "advanced nuclear" as listed in Sect. 1.1.
7. Cooling water: the cost of cooling-water capacity (fossil fuel) was fixed at \$1.75 per gpm per year derived as follows:

Capital investment (piping and pumps)	\$7.27 per gpm
Annual cost	
Fixed charges, 5.78%	\$0.42 per gpm
Maintenance, 1% of capital investment	\$0.07 per gpm
Power at 6.2 mills/kwhr, 120 ft head	\$1.26 per gpm
Total annual cost	\$1.75 per gpm

The inlet seawater temperature was fixed at 65°F. The cost of cooling water (advanced nuclear) was calculated similarly to be \$0.90 per gpm.

8. Value of product water: the condensate collected in the surface condensers was credited at $0.36 per 1000 gal for "fossil fuel" and half that for "nuclear."
9. Fixed charges: the fixed charge assumed throughout the study was 5.78%.

3.1.6 Economic Evaluations and Design Calculations

Intercooling and Pressure Ratios. – The most efficient way to divide an adiabatic gas compression process for intercooling is to split it into steps of equal compression ratios. Except for the pressure drop through the intercondensers, the compression range of the blower systems in this study was divided into two or three steps to provide equal compression ratios for each

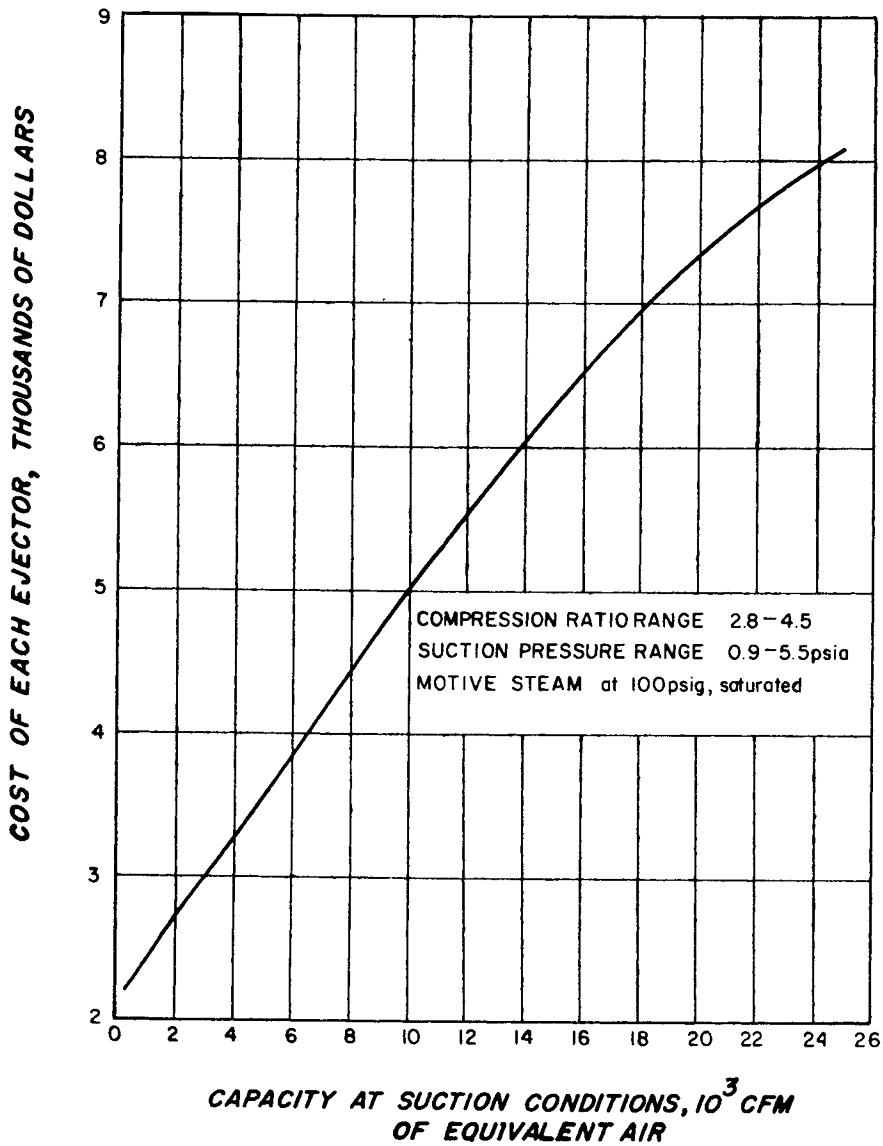

Fig. 18. Capacity vs Ejector Cost. Hastelloy construction.

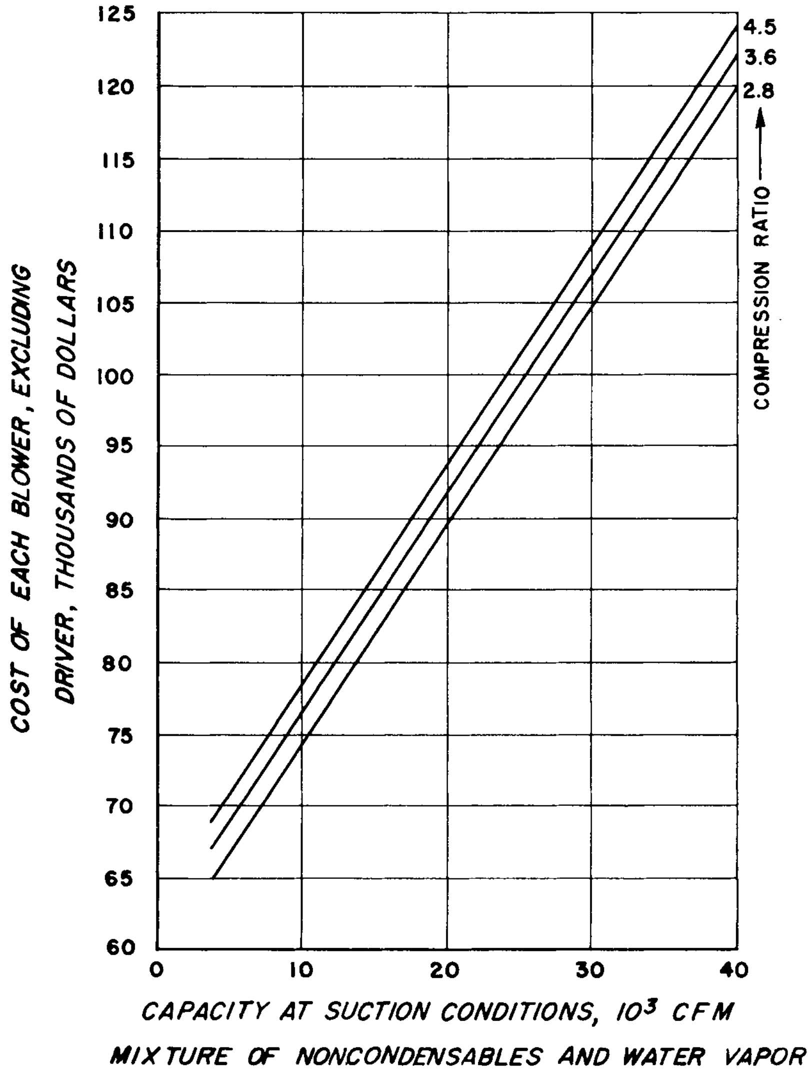

Fig. 19. Capacity vs Blower Cost. Cast iron with 3% nickel.

blower. The gas was cooled to provide the same temperature at the inlet to each blower. In order to confine the study and to keep the comparison on an equal basis, the pressure ratios and the inlet temperatures for the ejectors were held at the same values as those for the corresponding blowers. Table 19 gives the suction and discharge pressures calculated for two- and three-stage systems.

Efficiency. – The power requirements for the blowers were based on 70% adiabatic efficiency. Estimates of the steam requirements for the ejectors were confirmed by information obtained from manufacturers and by that available in literature.[25–27]

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[25]Frank H. Dotterweich and C. V. Mooney, "How to Design and Operate Gas Jet Compressors," *Petrol. Refiner* 34(10), 104–7 (1955).

[26]V. V. Fondrk, "Figure What an Ejector Will Cost," *Petrol. Refiner* 37(12), 101–5 (1958).

[27]R. B. Power, "Steam-Jet Air Ejectors," *Hydrocarbon Process. Petrol. Refiner* 43(2), 121–26 (1964).

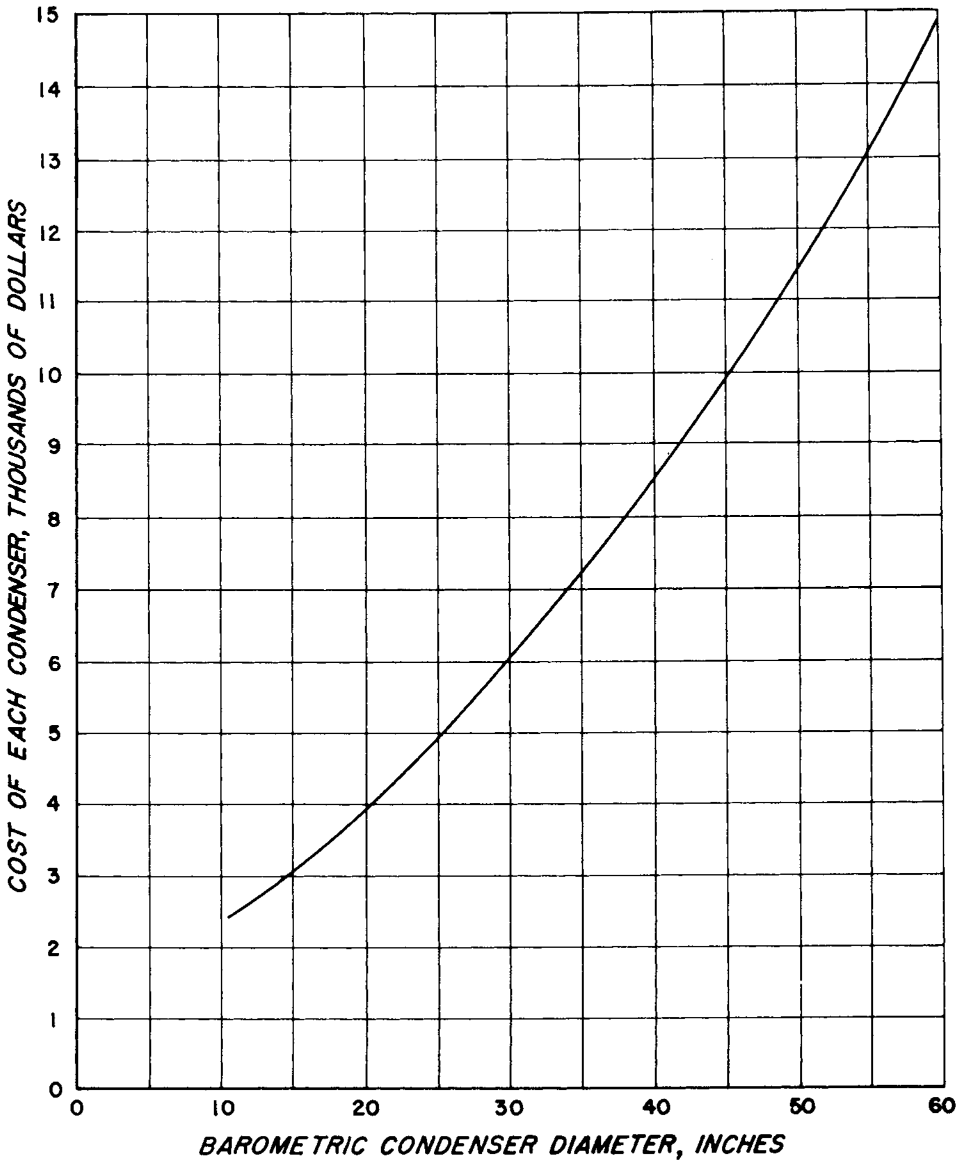

Fig. 20. Barometric Condenser Costs. Type 316 SS.

Gas Temperature Out of Condenser. – Consideration was also given to the temperature to which the gases should be cooled in each of the condensers. Condenser gas outlet temperatures of 75 and 80°F were selected and the overall cost difference evaluated for both the barometric and the surface condensers. At the lower temperature the required surface or the quantity of cooling water is larger, but at the higher temperature the energy requirements of the subsequent ejector or blower are larger. The difference proved to be small, but it favored the 75°F value selected for this study. (Seawater at 65°F was assumed to be available for cooling.)

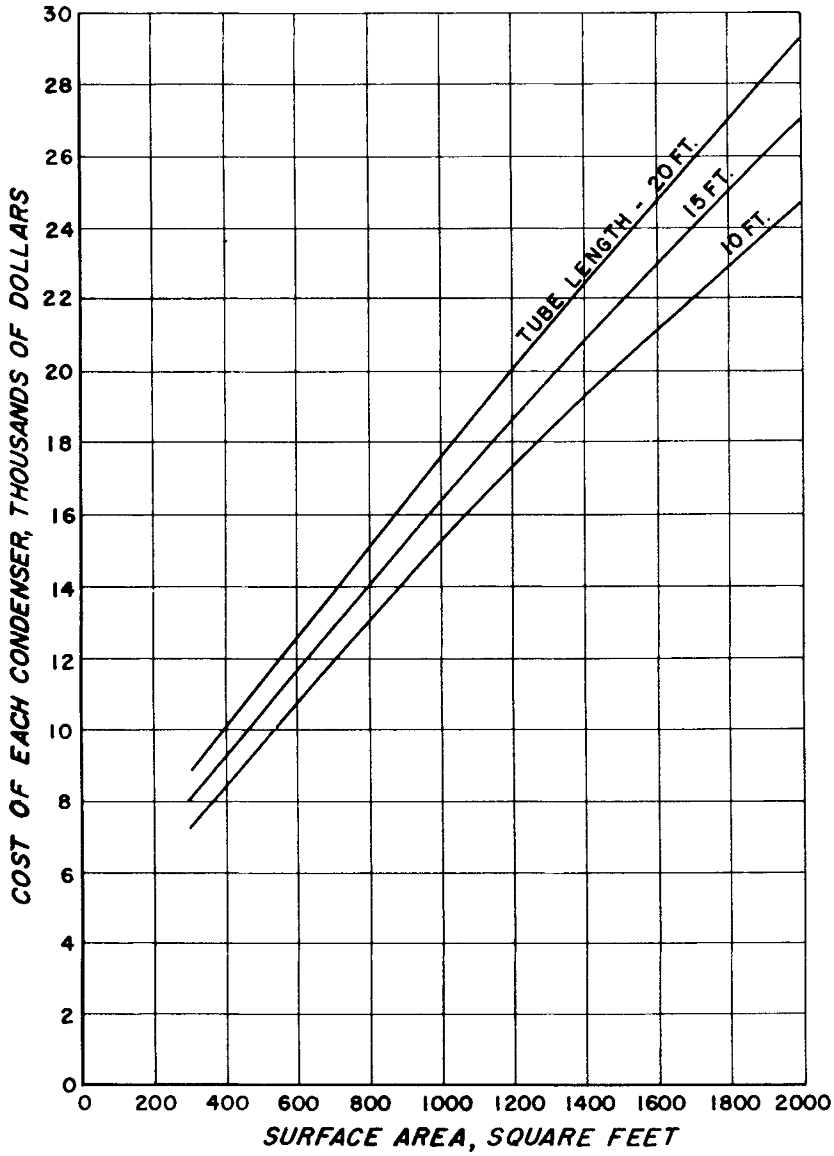

Fig. 21. Surface Condenser Costs. Multipass condensers with fixed heads and flexible tube sheets. Hastelloy C construction except heads, which are steel lined with Heresite.

Condenser Design. – In the design of barometric condensers, the minimum terminal temperature difference was established as 8°F, and the maximum temperature of cooling water leaving the condenser was fixed at 110°F. The gas pressure drop through the barometric condensers was considered to be negligible. After calculating the cooling-water requirements from the total heat load, the standard size of the barometric condenser was selected from the data presented in "Standards of Heat Exchanger Institute."[28]

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[28] *Standards for Direct Contact Barometric and Low Level Condensers*, 4th ed., Heat Exchanger Institute, New York, 1951.

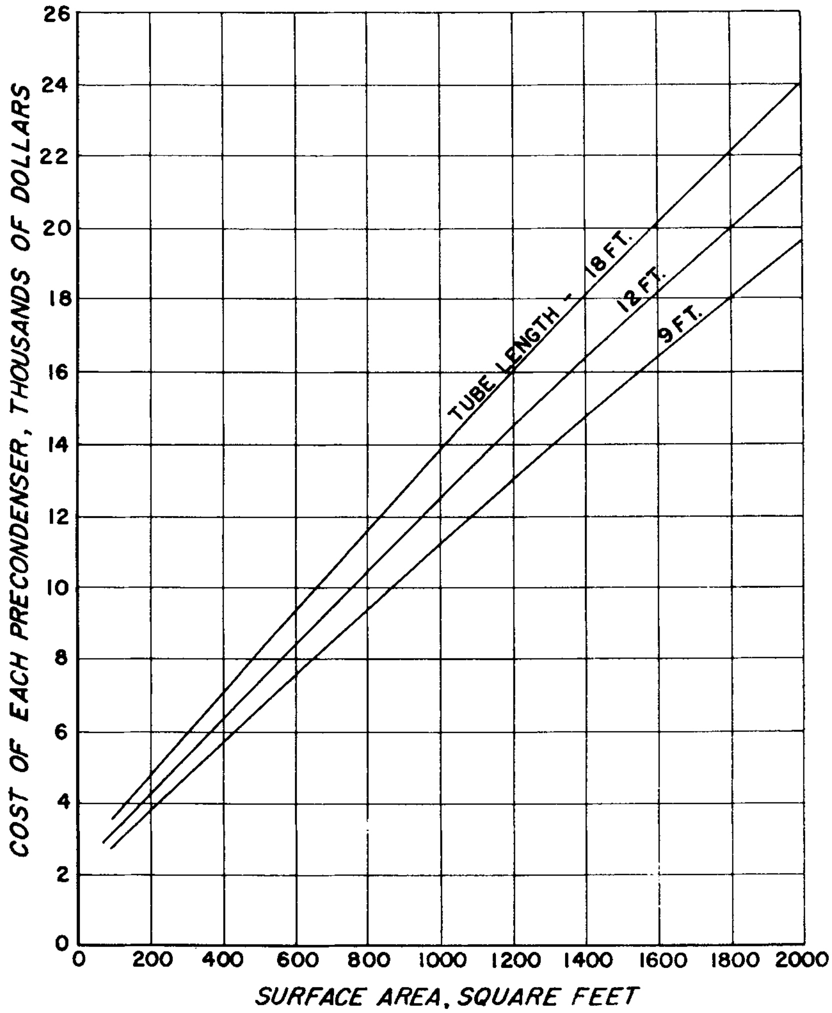


Fig. 22. Surface Precondenser Costs. Tubes and shell of 316 SS. Fixed heads of coated carbon steel.

Table 19. Pressures in a Multistage Blower or Ejector System

	Stage	Pressure (psia)			
		Suction		Discharge	
		Barometric Condensers	Surface Condensers	Barometric Condensers	Surface Condensers
Two-stage system	1	0.86	0.82	3.65	3.65
	2	3.47	3.28	14.7	14.7
Three-stage system	1	0.86	0.82	2.29	2.29
	2	2.18	2.06	5.81	5.81
	3	5.52	5.23	14.7	14.7

The design of surface condensers proved to be complex. When a mixture of noncondensables and steam is exposed to a surface colder than its dew point, condensation occurs and forms a thin layer of liquid on the surface. Next to this layer a film of the gas mixture is formed which is depleted in vapor and rich in noncondensables. Because of the difference in partial pressures of the vapor in this film and of the bulk stream, vapor diffuses from the main body through the gas film to condense on the cold surface. The rate of condensation is thus limited and governed by the laws of diffusion of vapor through a gas.

The method of Colburn, Chilton, and Hougen as presented by Kern[29] was utilized as the basis of calculation.

Optimum Number of Stages for Blowers and Ejectors. – The optimum number of stages for blowers is 2 because the capital costs for more stages rise faster than savings in energy. One stage is not practical because of high temperature rise. The optimum number of stages for an ejector system is 3, which is also the number of stages for minimum steam consumption based on available information.

Barometric vs Surface Condensers. – For the most suitable design in each case, the barometric condensers are cheaper than surface condensers when all costs and savings are considered. The advantage favoring the surface condensers is recovery of condensate, while barometric condensers have the advantage of price, low pressure drop, less use of cooling water, and low maintenance cost. The difference in cost, in any event, is small compared with the overall system cost.

The use of product as a coolant for barometric intercondensers was suggested by many of the conceptual designers. This possibility was not evaluated in the present study, but it should be considered in future work for intercondensers coupled to steam jets (where there is significant potential for recovery of product). There is a disadvantage in that the gases would be warmer, leading to less efficient pumping. The latter objection could be overcome by a two-stage barometric condenser – the first stage cooled by product and the second by cooling water. Such complexities might be justified in very large plants.

Decarbonating Pond. – Although designing the atmospheric decarbonator is outside the scope of this report, a representative design was evaluated to determine its usefulness in reducing the cost of the vacuum equipment. A five-step cascade- or waterfall-type decarbonation basin made of concrete was selected for this service. The cost of such a basin to handle the feed flow for a 50-Mgd plant is estimated to be \$150,000, including the incremental cost of feed pumps for the additional head requirement.

The cascade-type degassing pond ($\Delta P = 12.5$ psi approximately) chosen turned out to be uneconomical. It was cheaper to evacuate the carbon dioxide in the vapor phase than to pump the liquid. However, a degassing pond with a lower pressure drop ($\Delta P = 2$ psi approximately) may prove to be economical if its initial cost is not very high.

[29]D. Q. Kern, *Process Heat Transfer*, pp. 339–46, McGraw-Hill, New York, 1950.

Table 20. Optimum System – Two-Stage Blower System Using Barometric Condensers (Case II)[a]

	Blowers	
	No. 1	No. 2
Inlet flow		
Water vapor, lb/hr	1968	279
Noncondensables,[a] lb/hr	4405	4405
Pressure, psia	0.86	3.466
Temperature, °F	75	75
Discharge		
Pressure, psia	3.649	14.7
Temperature, °F	390	399
Brake horsepower	240	151
	Barometric Condensers	
	Precondenser	Intercondenser
Size (diameter), in.	36	15
Cooling-water flow, gpm	640	98
Cooling-water inlet temperature, °F	65	65
Cooling-water outlet temperature, °F	80	110

[a]See Table 18 for gas composition and definition of case II.

The decarbonator has secondary benefits in that the pond provides time for acid to react with the feed, it may be useful for clarifying the feed, and the deaerator size is reduced slightly.

Complete System Analysis. – Sixteen systems (containing combinations of blowers or ejectors, two or three stages, high or low evacuation loads, surface or barometric condensers) were designed and their costs estimated. Typical design parameters are shown in Tables 20, 21, and 22. Cost estimates are given in Tables 23 and 24.

3.1.7 Conclusions

This study has demonstrated that equipment of the required size and of materials with an expected life of 30 years in a desalination plant is available only as special equipment. Experience with the fabrication of premium materials required for this service is not readily available.

The economic evaluations, based on equipment which can be purchased now, have led to the following conclusions.

1. A two-stage blower system with barometric condensers is the most economical centralized station arrangement for fossil-fueled 50-Mgd plants. For plants getting low-cost nuclear steam,

Table 21. System with Surface Condensers Using Two-Stage Blowers (Case II)[a]

Blowers		
	No. 1	No. 2
Inlet flow		
Water vapor, lb/hr	2200	300
Noncondensables,[a] lb/hr	4405	4405
Pressure, psia	0.86	3.28
Temperature, °F	75	75
Discharge		
Pressure, psia	3.649	14.7
Temperature, °F	403	413
Brake horsepower	262	157
Surface Condensers		
	Precondenser	Intercondenser
Surface, ft^2	2070	800
Tube length, ft	17	10.6
Tube size	$\frac{3}{4}$ in. OD × 20 BWG	$\frac{3}{4}$ in. OD × 20 BWG
Number of passes	4	4
Shell diameter, in.	30	24
Cooling-water flow, gpm	724	456
Cooling-water velocity, fps	5	5
Cooling-water inlet temperature, °F	65	65
Cooling-water outlet temperature, °F	77	76
Overall U, Btu hr^{-1} ft^{-2} $(^{\circ}F)^{-1}$	192	38

[a]See Table 18 for gas composition and definition of case II.

two-stage blowers and three-stage ejectors are competitive. For approximately equal projected cost, ejectors would be preferred because of their inherent simplicity.

2. The energy costs are over half the total ejector system costs.

3. A precondenser should be used to reduce the amount of water vapor to be evacuated.

4. Barometric pre- and intercondensers are preferred to surface condensers because of lower capital cost, less severe service for materials, less use of cooling water, and significant reductions in pumping power because of lower pressure drop. These factors more than offset the condensate which might be saved. The possibility of cooling barometric intercondensers with product to recover condensate needs further study.

5. A cascade-type (high head loss) decarbonating pond used in this study proved to be uneconomical. However, a degassing pond of the low head loss (weir) type may prove to be economical and should be considered in future design studies.

6. Premium materials are required. Type 316 stainless steel should be useful up to 150 to 200°F. Hastelloy C or titanium is recommended for parts exposed to higher temperatures. The use of coatings, such as Heresite, over less costly materials is also attractive.

Table 22. Lowest Cost System for Ejectors – Three-Stage Ejector System Using Barometric Condensers (Case I)[a]

Ejectors			
	No. 1	No. 2	No. 3
Inlet flow			
Water vapor, lb/hr	941	227	77
Noncondensables,[a] lb/hr	1938	1938	1938
Pressure, psia	0.86	2.18	5.52
Temperature, °F	75	75	75
Discharge			
Pressure, psia	2.29	5.81	14.7
Temperature, °F	181	208	221
Motive steam			
Flow, lb/hr	6900	6900	6900
Pressure, psia	114.7	114.7	114.7
Temperature, °F	338	338	338

Barometric Condensers			
	Precondenser	Intercondenser	
		No. 1	No. 2
Size (diameter), in.	21	18	18
Cooling-water flow, gpm	231	160	124
Cooling-water inlet temperature, °F	65	65	65
Cooling-water outlet temperature, °F	80	110	110

[a]See Table 18 for gas composition and definition of case I.

Table 23. Blower Systems Cost Summary

Ground Rules	Case	Number of Stages	Condenser Type	Total Cooling Water (gpm)	Total Blower Power Consumption (hp)	Installed Cost (dollars)					Annual Operating Cost (dollars)					Total Annual Cost
						Blowers	Condensers	Instrumentation	Pond	Total	Fixed Charges	Maintenance	Cooling Water	Electricity	Condensate Credit	
Fossil	I	2	Surface	434	233	174,000	25,300	5000	150,000	354,300	20,500	3500	760	33,500	(800)	57,500
	II	2	Surface	1180	419	210,300	44,900	5000		260,200	15,000	2600	2065	17,500	(2100)	35,100
	I	3	Surface	475	245	235,700	34,000	7500	150,000	427,200	24,700	4300	831	34,000	(900)	62,900
	II	3	Surface	1208	410	272,800	55,300	7500		335,600	19,400	3400	2114	17,100	(2200)	39,800
	I	2	Barometric	300	219	168,700	9,700	5000	150,000	333,400	19,300	3300	500	32,900		56,000
	II	2	Barometric	750	391	202,000	13,100	5000		220,100	12,700	2200	1300	16,300		32,500
	I	3	Barometric	300	230	228,400	13,800	7500	150,000	399,700	23,100	4000	500	33,400		61,000
	II	3	Barometric	800	381	262,500	17,300	7500		287,300	16,600	2900	1400	15,900		36,800
Advanced nuclear	I	2	Surface	434	233	174,000	25,300	5000	150,000	354,300	20,500	3500	390	10,800	(400)	34,400
	II	2	Surface	1180	419	210,300	44,900	5000		260,200	15,000	2600	1060	5,600	(1050)	22,200
	I	3	Surface	475	245	235,700	34,000	7500	150,000	427,200	24,700	4300	428	11,000	(450)	39,500
	II	3	Surface	1208	410	272,800	55,300	7500		335,600	19,400	3400	1087	5,500	(1100)	27,200
	I	2	Barometric	300	219	168,700	9,700	5000	150,000	333,400	19,300	3300	200	10,700		33,500
	II	2	Barometric	750	391	202,000	13,100	5000		220,100	12,700	2200	700	5,300		20,900
	I	3	Barometric	300	230	228,400	13,800	7500	150,000	399,700	23,100	4000	300	10,800		38,200
	II	3	Barometric	800	381	262,500	17,300	7500		287,300	16,600	2900	700	5,100		25,300

Table 24. Ejector Systems Cost Summary[a]

Ground Rules	Case	Number of Stages	Condenser Type	Cooling Water (gpm)	Total Steam Consumption (lb/hr)	Installed Cost (dollars): Ejectors	Installed Cost (dollars): Condensers	Installed Cost (dollars): Instrumentation and Controls	Installed Cost (dollars): CO_2 Release Pond	Installed Cost (dollars): Total	Annual Costs (dollars): Fixed Charges	Annual Costs (dollars): Maintenance	Annual Costs (dollars): Steam	Annual Costs (dollars): Cooling Water	Annual Costs (dollars): Pond Pumping Power	Annual Costs (dollars): Condensate Credit	Annual Costs (dollars): Total
Fossil	I	2	Surface	937	10,000	9,200	31,000	5000	150,000	195,200	11,300	2000	30,080	1640	23,800	(2630)	66,200
	II	2	Surface	2247	21,800	13,200	56,800	5000		75,000	4,300	800	65,680	3930		(6020)	68,700
	I	3	Surface	1059	8,100	12,600	40,900	7500	150,000	211,000	12,200	2100	24,470	1850	23,800	(2840)	61,600
	II	3	Surface	2422	17,700	17,300	71,300	7500		96,100	5,600	1000	53,220	4240		(6480)	57,600
	I	2	Barometric	498	9,000	9,200	11,000	5000	150,000	175,200	10,100	1800	27,000	893	23,800		63,600
	II	2	Barometric	1210	19,300	13,200	16,200	5000		34,400	2,000	300	58,060	2120			62,500
	I	3	Barometric	515	6,900	12,600	13,800	7500	150,000	183,900	10,600	1800	20,740	900	23,800		57,800
	II	3	Barometric	1260	15,400	17,300	20,000	7500		44,800	2,600	500	46,450	2213			51,800
Advanced nuclear[b]	I	2	Surface	937	10,000	9,200	31,000	5000	150,000	195,200	11,300	2000	10,290	843	7,700	(1315)	30,900
	II	2	Surface	2247	21,800	13,200	56,800	5000		75,100	4,300	800	22,450	2023		(3010)	26,600
	I	3	Surface	1059	8,100	12,600	40,900	7500	150,000	211,000	12,200	2100	8,355	953	7,700	(1420)	29,900
	II	3	Surface	2422	17,700	17,330	71,300	7500		96,100	5,600	1000	18,200	2180		(3250)	23,800
	I	2	Barometric	498	9,000	9,200	11,000	5000	150,000	175,200	10,100	1800	9,225	450	7,700		29,300
	II	2	Barometric	1210	19,300	13,200	16,200	5000		34,400	2,000	300	19,880	1080			23,260
	I	3	Barometric	515	6,900	12,600	13,800	7500	150,000	183,900	10,600	1800	7,070	460	7,700		27,600
	II	3	Barometric	1260	15,400	17,300	20,000	7500		44,800	2,600	500	15,825	1130			20,100

[a]Cases I and II defined in Table 18.
[b]Costs based on energy from large nuclear plants.

3.2 DEAERATORS

Virtually complete removal of dissolved gases is an important step in the treatment of seawater in most of the planned distillation systems. The liberation of noncondensables from seawater as it passes through the evaporator can cause inefficiencies in the condensers and major corrosion problems for conventional materials of construction.

Acidified seawater contains about 8 ppm of O_2 and 100 ppm of CO_2. Oxygen removal to 10 ppb and CO_2 removal to 1 ppm is the tentative objective. The ultimate requirement for gas removal cannot be given without quantitative knowledge of the effect of traces of gas on the corrosion of the various materials used for evaporator design.

The stated objectives of the work were as follows:

1. To ascertain the present state of the art of deaerators suitable for deaerating the feedwater to the required low levels for large distillation plants.
2. To develop design parameters for deaerators that will reduce the oxygen to 10 ppb and the CO_2 to 1 ppm.
3. To provide an improved basis for the overall economic appraisal of the deaerator problem by the calculation for selected plant designs of the costs for the quantity of stripping steam required for operating tray- and packed-column-type deaerators and the pressure drop across these types of deaerators. This information, coupled with the equipment costs from 2, above, would permit much more meaningful economic evaluations than are available today.

The scope of work included the following:

1. expeditious study of the information available in the existing literature on deaerators and CO_2 release ponds to determine the applicability to large distillation plants,
2. review of the experience of OSW contractors,
3. selection of the most promising methods from a review of the deaerators and the CO_2 ponds proposed by all the conceptual designers of large desalination plants to obtain information which may be pertinent to the design of new or proposed deaerators,
4. recommendation of a scope of work, test procedures, and a location for carrying out additional development based on the information generated in the above items 1 through 3.

The program was subsequently broadened to include the development, under contract, of suitable oxygen analyzers and the testing of OSW deaerators. This more recent work, in progress, will not be reported here.

3.2.1 Types of Deaerators

Types Requiring Stripping Steam. – *Tray Columns (Fig. 23).* – There are many tray columns which may be used as deaerators with steam in parallel or counter flow used as a stripping medium. The most conventional type is that which has bubble caps as for vapor distillation; this type is expensive to fabricate and has associated with it undesirably large pressure drops. There are many tray designs which have lower pressure drops and are of simpler construction. Among these are turbogrid trays, baffle trays, sieve trays, and ripple trays. On the basis of some recent work on the evaluation of a number of these types in ammonia absorption, it appears that the ripple and turbogrid trays are the most attractive from the point of view of low operating pressure drop and high tray efficiency.

Packed Columns (Fig. 23). – The equipment normally consists of a vertical cylindrical column containing a bed of packing which breaks up the water to create a maximum exposed surface. Many different types, shapes, and sizes of packing are available.

In the actual process the water is spread or sprayed over the top of the packing bed, from where it trickles down through the packing. Stripping steam is supplied at the bottom, and it travels upward through the packing. By virtue of Henry's law for concentration of a dissolved gas, the gases are separated from the water and are carried away by the stripping steam. If the supply steam is pure, the dissolved gases in water can be made to approach zero. An average liquid loading rate of 20,000 lb hr^{-1} ft^{-2} is considered practical. The flow rate for the stripping steam is a function of the deaeration levels required, the amount of steam required to minimize channeling, and the characteristics of the particular type of packing.

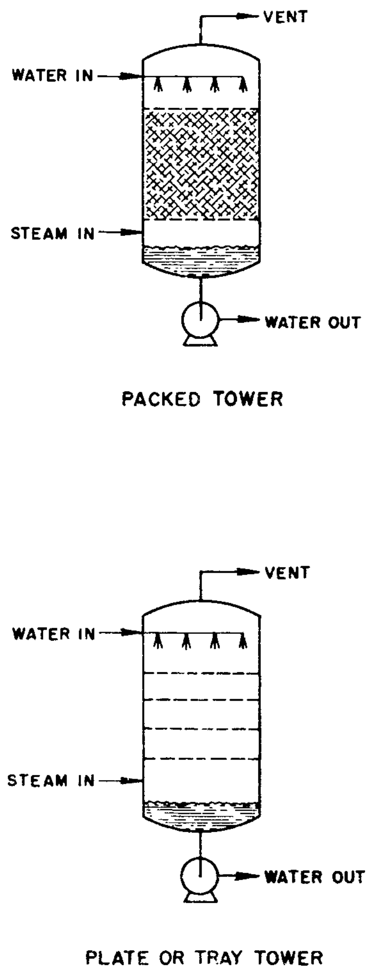


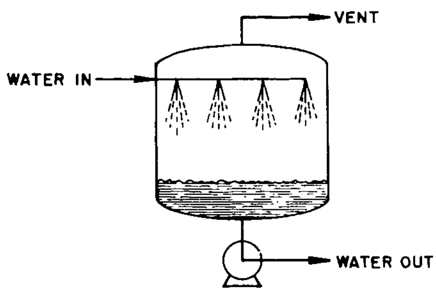
Fig. 23. Packed and Tray Tower Deaerators.

It is especially difficult to prevent undesirable channeling in large-diameter packed columns.

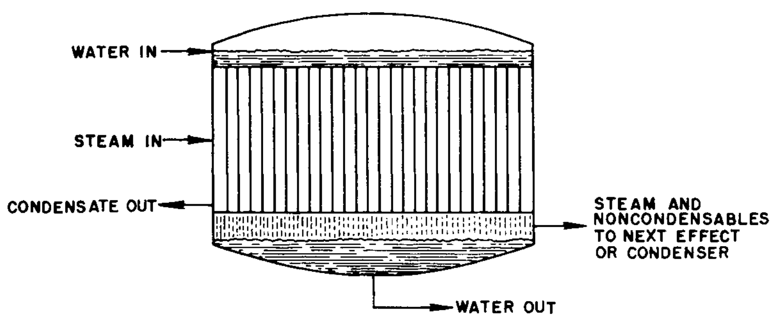
Steam-Jet. – Steam-jet deaerators have been used for many years in relatively small power plants, and they do an excellent job of removing oxygen to very low levels in the range of 0 to 7 ppb. Usually the process operates at elevated temperatures and above atmospheric pressures. The basic principle of the process involves violently mixing steam and water together to bring the water to saturation temperature. Part of the steam is condensed, and the excess steam provides the necessary turbulence and stripping action to degasify the water. Although the deaerating capability of such an apparatus is high, the high operating costs involved are prohibitive where large plants are concerned.

Wetted-Wall Towers. – The wetted-wall tower design is the ultimate extension of the counterflow tray or packed-tower concept. Although this concept has the virtue of an extremely low pressure drop, the cost of the mass transfer surface required may render the method uneconomic.

Types Requiring No Stripping Steam. – *Flash Unit (Fig. 24).* – Based on mass transfer equilibrium principles, it is possible to subject water to a flash (sudden reduction in pressure to below the saturation value) and cause the ebullition of the dissolved oxygen and carbon dioxide.



FLASH TYPE



VTE TYPE

Fig. 24. Flash- and VTE-Type Deaerators.

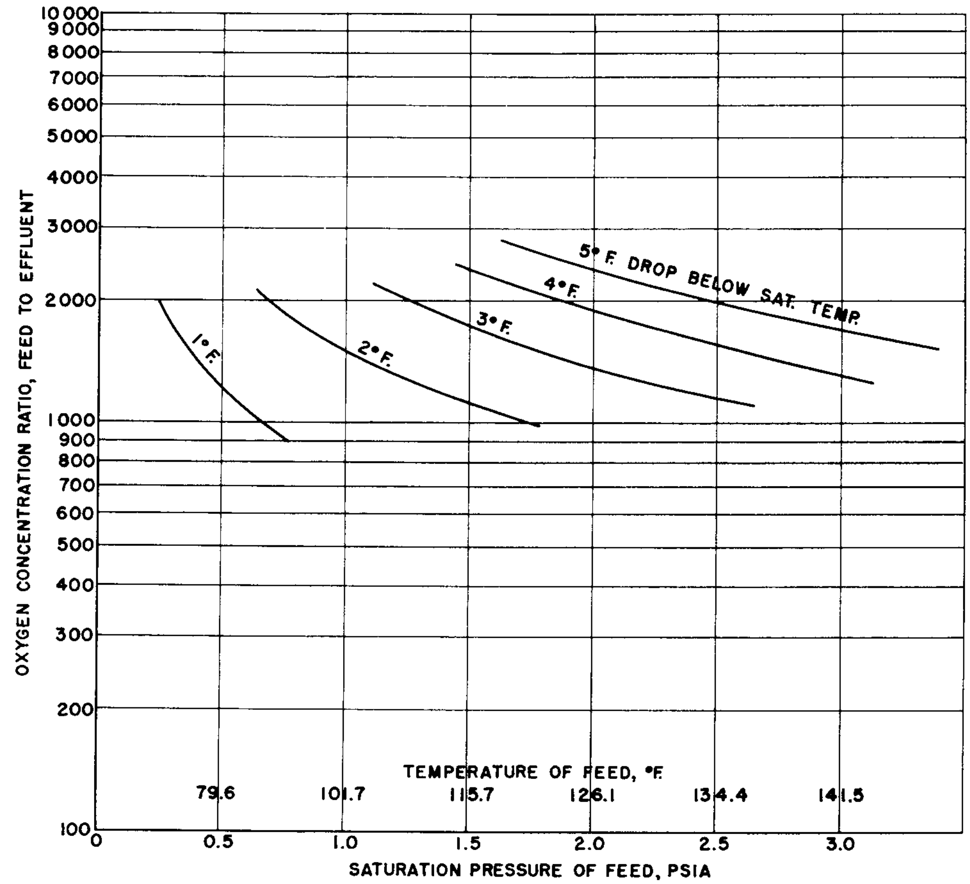


Fig. 25. Ideal Flash Process. Equilibrium conditions assumed.

Figure 25 shows the factor by which the oxygen content of water is reduced in an equilibrium flashdown. The desired factor of about 800 is achieved in a 1° flashdown at 90°F or about 2° at 130°F.

A real flash system will be much less efficient than the ideal system, but the process appears to have considerable potential.

VTE Effect. – In a vertical-tube evaporator plant it may be possible to accomplish the deaeration in one of the evaporator effects. The feedwater to be deaerated is introduced at the top of the tubes and falls as a thin film inside the tubes, which are heated from the outside with steam. A nozzle or weir controls the flow pattern of the falling brine. The heat transferred through the wall generates steam in the boundary layer, and the steam provides stripping action to degasify the water.

Combinations. – If it is not possible to achieve the required efficiency in a single flashing stage, a likely combination is flashing through spray nozzles followed by supplementary trays or packing.

3.2.2 Types of Decarbonators

When sulfuric acid is added to plant makeup water, the bicarbonates are decomposed and approximately 100 ppm (by weight) of free carbon dioxide is evolved. Many designers of sea-water distillation plants have proposed that better economy can be achieved for the deaeration process if most of this free carbon dioxide is allowed to escape from the water at atmospheric pressure. The economic evaluations of Sect. 3.1 of this report confirmed the usefulness of de-carbonators provided they were of a low-pressure-drop type.

The following types have been considered:

Pond Types (Fig. 26). – *Cascade Steps.* – This design consists of a set of stairlike steps leading down to a retention pond. The water is introduced at the top step and cascades down

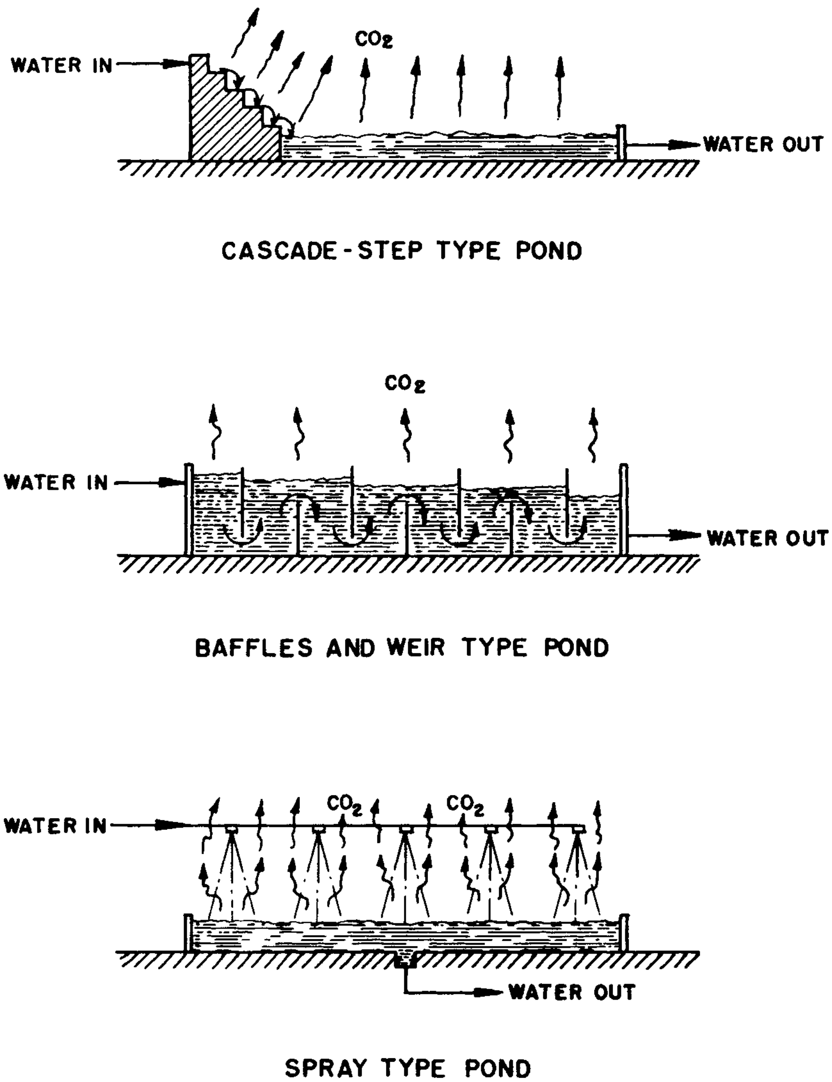


Fig. 26. Pond-Type Atmospheric Decarbonators.

the steps into the retention reservoir. It is withdrawn from the opposite end of the reservoir. The fall-splash process at each step continuously renews the liquid surface to effect good gas release. Betz Laboratories, Inc., has done some experimental work on the cascade pond and concluded that 60 to 70% of the carbon dioxide can be easily removed. The proposed design for a 50-Mgd plant consists of a weir, five steps (each 3 ft tread, 3 ft rise, and 40 ft wide), and a retention tank (200 ft long, 40 ft wide, and 4 ft deep). Total retention time is approximately 7 min. Head loss is expected to be approximately 8.5 psi.

Baffles and Weirs. – Numerous variations of this type of design have been proposed. Basically, the design calls for a large open pond with numerous flow deflectors to bring as much water as close to the surface as possible. A typical design uses alternate submerged weirs and surface weirs to make the water flow in a vertical zig-zag path. No experimental data on such designs have been published. This type of system offers a minimum pressure drop, but the degree of effectiveness is unknown.

Spray Ponds. – Several of the nozzle manufacturers were contacted for design information on sprays. The only data that we could obtain from them were the size of drops and the cone angle that can be expected under a particular set of conditions. At its saline water test station at Millstone Point, Connecticut, the AMF Company is successfully using a spray pond to decarbonate acidified seawater down to 15 to 20 ppm. Although the pressure drop through such a degasifier is expected to be as high as that of a cascade-step-type pond, the capital investment would be lower because of the small size of the pond.

Tower Types. – Packed towers and modified cooling towers using atmospheric air as the contacting medium have been proposed as decarbonators by several conceptual designers of 50-Mgd plants.[30] Although these devices would undoubtedly be more efficient than the pond-type systems, it would be hard to justify their higher capital and operating costs by compensating savings in the deaerator and ejector systems.

3.2.3 Review of Deaerator Experience

Survey of the Literature and of OSW Contracts. – Although a considerable amount of literature is available on the deaeration of fresh water (power plants), very little information in the general literature can be directly applied to problems of designing deaerators for seawater. While most of the deaeration work has been done on hot distilled water, desalination plant designs require that seawater deaeration be done at low pressures and temperatures. Most freshwater deaerators have to strip only oxygen, but the seawater deaeration process is more complex because of added requirements for removal of carbon dioxide. In addition, the equilibrium behavior of carbon dioxide in water is very complex, and transformations between bicarbonates,

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[30] *1965 Saline Water Conversion Report*, pp. 178, 233–35, and 277–95, U.S. Government Printing Office, Washington, D.C. (1966).

carbonates, and free carbon dioxide further complicate the classical analysis of mass transfer in a conventional stripping tower.

M. W. Kellogg Contract. – Experimental work done by M. W. Kellogg Company[31] is the only known attempt to relate conventional two-phase exchange technology to seawater in a countercurrent packed tower. Although it fell short of providing enough information to design a large deaerator, it made an important step toward understanding seawater deaeration.

The major portion of their work was an attempt to establish equilibrium data for oxygen-seawater and carbon dioxide–seawater systems and to determine mass transfer coefficients for packed beds operating at temperatures and pressures at which large deaerators are expected to operate. Their tests were run in a tower 24 ft tall and 8 in. in inside diameter. Two packing beds, each 5 ft high, were stacked one above the other with a redistributor plate between them. The packing was 1-in. ceramic Berl saddles. Liquid flow rates were varied from 10,000 to 20,000 lb hr^{-1} ft^{-2}, inlet liquid temperature from 100 to 170°F, and stripping steam rates from 0 to 100 lb hr^{-1} ft^{-2}. Effluent O_2 levels down to 10 ppb are reported. On the basis of their test data, Kellogg believes that the tower mass transfer coefficients for seawater are comparable with those for fresh water.

Dow Tests of the Freeport Deaerator.[32] – The Freeport deaerator is a packed tower 46 ft in height and 6 ft in diameter. It was originally designed to contain a 16-ft-high bed of 3-in. ceramic Raschig rings. It was the objective of the Dow Chemical Company to evaluate the design and performance of the deaerator, and, if required, suggest necessary modifications. Dow conducted a test and established the following values as typical performance of the unit:

Inlet brine flow	488,000 lb/hr
Inlet brine temperature and pH	122°F; 4.0
Effluent gas content in seawater	10 ppb oxygen; 2.8 ppm carbon dioxide
Mode of operation	Countercurrent
Vapor leaving	1050 lb/hr
Vapor entering	Not given
Operating pressure	1.6 psia

Dow suggested that an 8-ft-high bed of Maspac (3.75-in. type FN-90) should provide an economical substitute for the 16-ft bed of Raschig rings. (One objection to the ceramic rings was that they were scoring the steel tower.) Such a bed was apparently substituted and was indicated to reduce oxygen to 5 ppb and carbon dioxide to 3.6 ppm.

During the switch from Raschig rings to Maspac, it was decided to run the deaerator without any packing in it. Dow reports: "The effluent showed between 10 and 100 ppb of oxygen and

[31] *Desorption of Carbon Dioxide and Oxygen from Seawater*, OSW R and D Progr. Rept. No. 158 (October 1965).

[32] Dow Freeport Deaerator Tests, *An Engineering Evaluation of the Long Tube Vertical Falling Film Distillation Process*, Dow Chemical Company, OSW R and D Progress Report No. 139, February 1965.

a gross carbon dioxide content of about 8 ppm as determined by gravimetric analyses." Through conversations with their personnel, it was discovered that some stripping steam was used during the above-mentioned test, but the quantity is unknown.

American Machine and Foundry Tests on Decarbonator and Flasharator.[33] – The deaeration process used by AMF at Millstone Point, Connecticut, is a two-step process consisting of an atmospheric decarbonator and a vacuum deaerator.

The decarbonator is a simple free-falling-spray device designed to handle 100 gpm. The spray nozzle is specially designed to provide a high degree of surface breakup and operates with a pressure drop of 7 psi. The spray trickles through four layers of window screen before it falls into the deaerator intake.

The AMF deaerator is a horizontal cylindrical tank with flat heads and measures approximately 4 ft in diameter and 4 ft in length. The incoming water is sprayed through two perforated pipes into the vacuum chamber maintained at 1.5 in. Hg absolute pressure. The ratio of throughput to recirculation is approximately 1:5, and the recirculation water is undergoing a flashdown of 3 to 5°F. The incoming water does not undergo any flashdown.

AMF reports 15 to 18 ppm of carbon dioxide in decarbonator effluent and 4 to 6 ppm of carbon dioxide in deaerator effluent. The deaerator is believed to bring oxygen down to about 50 ppb.

Baldwin-Lima-Hamilton Corporation (BLH) Decarbonator and Deaerator Operation. – BLH[34] operates a 4-ft-high decarbonation tower (countercurrent with air) and a 13-ft-high deaerator column (countercurrent with steam) at the OSW Saline Water Research Station at Wrightsville Beach. Both units operate at about 135°F and contain 2-in. Maspac packing. The oxygen content of the deaerated feed is about 100 ppb; the CO_2 in the effluent has not been measured to date.

Point Loma Deaerator.[35] – This deaerator contains 25 Monel trays 2 ft wide × 15 ft long in a 12.4-ft-high tower. Stripping steam is admitted in parallel flow. Up to 860 gpm of makeup flow, pH about 4 to 5, was deaerated to a reported 5 ppb of O_2 at 1 in. Hg absolute pressure. The CO_2 content of the deaerated seawater varied widely for reasons that were unknown.

Scaleup of Tower Diameter. – The principal factors affecting the efficiencies of packings are resistances to mass transfer in the liquid and vapor phases and channeling. The mass transfer resistances are properties of the fluids, but channeling is a function of physical size and arrangement of the packings and the tower. Various attempts[36,37] have been made to determine the effect of tower diameter on channeling. It is well established that mass transfer

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[33] R. K. Sood and M. Hanig, *Trip Report: AMF Company and M. W. Kellogg Company*, Appendix E in ORNL-CF-66-6-84, April 21, 1966.

[34] R. K. Sood and S. C. Jacobs, *Trip Report: Wrightsville Beach, N.C., March 7–8, 1966*, Appendix G in ORNL-CF-66-6-84.

[35] *Second Annual Report, Saline Water Conversion Demonstration Plant, Point Loma I*, OSW R and D Progress Report No. 114 (1964).

[36] R. J. Hengstebeck, *Distillation Principles and Design Procedures*, p. 240, Reinhold, New York, 1961.

[37] D. P. Murch, "Height of Equivalent Theoretical Plate in Packed Fractionating Columns," *Ind. Eng. Chem.* 45(12), 2616 (1953).

efficiency decreases with increased diameters, but most of the experimental work done has been on column diameters much smaller than the desalting plant requirements of 40- to 50-ft-diam columns. An attempt to extrapolate the available data over such a wide range is likely to give inaccurate results.

Methods of Measuring Dissolved Oxygen and Carbon Dioxide in Seawater. – Analytical techniques for determining dissolved carbon dioxide and oxygen in water are described in ASTM designations D513-57[38] and D888-65T[39,40] respectively. These techniques have long been used successfully in analyzing distilled water, but attempts to apply these methods per se to seawater have yielded erroneous results in some cases. A literature survey was made, and several instrument manufacturers were contacted to determine if methods were available for measuring low levels of oxygen and carbon dioxide dissolved in seawater. The modified ASTM (Winkler) analysis proposed by Potter and White[41] appears to be the most suitable (wet chemistry) for low levels of oxygen in seawater.

Contacts with instrument manufacturers have uncovered an instrument which seems to hold promise in eliminating most of the problems encountered in analyzing seawater for dissolved oxygen. The unique feature of this instrument is the "semidry" electrode, which has no contact with the sample stream and therefore does not become contaminated and is not affected by other chemicals present in the seawater. A known quantity of hydrogen is purged through the sample water, where it picks up the oxygen. These gases are then introduced into a gas analyzing cell, which determines the oxygen content. This instrument is made by Cambridge Instrument Company, who claim the following performance features:

Range	0 to 20 ppb of dissolved oxygen
Response	90% of reading in 1 min, 100% in 3 min
Accuracy	±5% of full scale

The instrument is being subjected to seawater tests using the modified Winkler analysis as a referee test.

The ASTM method for measuring CO_2, the evolution-titration method, appears to be satisfactory.

Commercial Deaerators. – A survey was made of deaerator manufacturers for units designed to process 100 Mgd of feed and to provide an effluent containing 5 ppb of oxygen.

Three positive replies were received. One supplier offered a tray-type deaerator which could be "guaranteed" when operated with countercurrent stripping steam. The cost of the trays was estimated to be $892,000.

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[38] *ASTM Standards on Industrial Water; Atmospheric Analysis*, Society for Testing Materials, Part 23, D 513–57, pp. 29–34 (1965).

[39] *Ibid.*, D 888-65T, pp. 95–98.

[40] *Ibid.*, D 888-65T, pp. 98–102.

[41] E. C. Potter and J. F. White, *J. Appl. Chem.* 1 (Parts I, II, III), 285–327 (1957).

Two suppliers offered packed-bed deaerators which would operate with a small amount of stripping steam. The designs appeared to be speculative ones in that no evidence was given that the units would provide the desired oxygen concentration of 5 ppb. The prices estimated were $372,000 and $615,000 respectively.

Conceptual Designs. [42] – A very wide variety of designs (summarized in Table 25) were proposed by the conceptual designers.

All except one proposed atmospheric decarbonation ponds or towers as a first treatment. Three contractors (Ferguson, Foster-Wheeler, and General Electric) proposed that decarbonation be done at the feed temperature of 65°F. Most contractors proposed to preheat the brine moderately before acidification and decarbonation. BLH performed this step at 145°F.

Most deaerators were designed to operate at 74 to 90°F. BLH deaerated at 145°F, General Electric at 222°F, and Badger at 288°F. Some systems used stripping steam.

3.2.4 Economic Evaluation

A deaerator for use with a desalination plant is an integral part of the process, with brine and vapor streams being received from and returned to the primary plant. Temperature and phase changes are experienced by the streams, altering their usefulness in the evaporator. Vapor is usually produced or consumed, affecting the evaporator capacity.

Accurate assessment of the total annual cost of various types of deaerators is quite complex and is not possible without more accurate design data. Some generalizations are, nevertheless, possible. The costs might be broken down as follows:

1. atmospheric decarbonator cost,
2. deaerator capital cost,
3. cost of pumping feed through the deaerator,
4. cost of the evacuation system,
5. cost of steam availability used in the deaerator and lost to the evaporator,
6. incremental cost of condenser surface used with steam-gas mixtures over the cost required for condensing pure steam,
7. credit for distillate produced in deaerator,
8. cost of changes in the evaporator required to accommodate the deaerator.

These costs vary widely in the designs which have been proposed. Potentially the largest single cost is the cost of steam availability lost to the evaporator. For this reason flash deaerators, which do not require stripping steam, are of promise in relation to tray types, which require large amounts of steam.

If tray-type or packed deaerators will ultimately be needed to achieve the desired efficiency, steam from the last stage of the evaporator may be used for stripping. The availability expended

[42] *1965 Saline Water Conversion Report*, pp. 178, 233–35, and 277–95, U.S. Government Printing Office, Washington, D.C. (1966).

Table 25. Summary of Decarbonator and Deaerator Designs by Conceptual Designers of 50-Mgd Distillation Plants

Designer	Acid Treatment	Atmospheric Decarbonator	Deaerator				Target Assays (ppm)	
			Construction	Pressure	Temperature (°F)	Stripping Medium	CO_2	O_2
Allis-Chalmers	H_2SO_4	Submerged weir, Amer-plate lined. Exit CO_2 24.5 ppm, 78.5°F	Horizontal cylinders of baffle spill type, Monel-lined carbon steel	Vacuum	*a*	None	2	0.02
Applied Research	H_2SO_4	Forced-draft tower made of bitumen-coated concrete packed with Surfpac. Water heated, CO_2 reduced to 50 ppm	Three horizontal chambers series-connected. Water sprayed in first chamber and flashed down 5° through successive chambers. Flashed steam condensed on feed-water-cooled tube bundles. Monel	*a*	90	None	*a*	0.07
Aqua-Chem	H_2SO_4	Steel with cypress spray trays. Water scrubbed with air at 72.5°. CO_2 = 10 ppm, O_2 = 18 ppb, H_2S = 2 ppm	Spray, packed-bed type, using Pall rings in last heat reject stage. Made of steel-lined concrete	Vacuum, but not low	*a*	Steam from flashing brine	6	0.007
Badger	H_2SO_4	None	Part of stages 1 and 2. Several Ti trays in stage 1	2° flash	288	None	*a*	*a*
Baldwin-Lima-Hamilton	H_2SO_4	Redwood commercial-type cooling tower at 145°. Stripped with air. "All" CO_2 and 40% of dissolved air removed at atmospheric pressure	Spray and packed type. Steel shell lined with Heresite and packed with polypropylene packing	3.2 psia	145	Steam	*a*	0.1
C. F. Braun	H_2SO_4	Concrete basin coated with neoprene asphalt. Two-thirds of CO_2 removed at 91°	Concrete with several decks of trays	*a*	*a*	Steam from last stage	*a*	0.001
Burns and Roe	H_2SO_4	Open concrete basin, plastic lined (Koppers Bitumastic 300 M or equivalent). Feed heated to 80°; two-thirds of CO_2 removed and air reduced to 23.5 ppm	Spray, perforated-tray type. Hypalon-lined concrete shell. Cypress wood decks	1 in. Hg abs	*a*	None	*a*	0.7
Dow Chemical	H_2SO_4	Steel-shell, rubber-lined tower packed with Maspac. 111° at 1 atm, stripped with air. CO_2 reduced to 0.8 ppm, 30% of other gases removed	Steel-shell, rubber-lined packed tower, four vertical sections packed with Maspac. Flash steam at each section pumped to next higher pressure section to serve as stripping steam	*a*	111	Steam and noncondensables generated in the deaerator	<0.8	0.009
H. K. Ferguson	H_2SO_4	Rubber- and bituminous-lined concrete basin. Low temperature (65°), $2\frac{1}{4}$-hr residence time	Raschig-ring-packed horizontal cylinder. Water sprayed in	*a*	85	None	*b*	*b*
Fluor	H_2SO_4	Concrete basin made of special cement (Portland type V). 85°, CO_2 reduced to 25 ppm	Spray type; flashing action	*a*	85	Steam	*a*	*a*
Foster Wheeler	H_2SO_4	Concrete	Heresite-lined carbon steel packed with ceramic Raschig rings. Flashing action	0.4 psia	75	None	2	0.1
General Electric	H_2SO_4	Butyl-rubber-lined concrete basin. Water sprayed at 65°, 90% of CO_2 released	Water spread over two layers of Maspac from a tray with 5452 notched downcomers	>1 atm	222	None	*a*	*a*
Lockheed	H_2SO_4	Concrete basin, 80°; CO_2 reduced to 25 ppm	Water sprayed into a 316 SS vacuum tank		80	Steam	*c*	*c*
Ralph M. Parsons	H_2SO_4	Concrete pond coated with Carbomastic No. 2. CO_2 reduced to 34 ppm	Stainless steel countercurrent tower packed with 1-in.-high bed of 2-in. stainless steel Raschig rings	0.49 psia	78.8	Steam	0.00068	0.12
Worthington	H_2SO_4	Bitumastic-lined concrete with open canals; 74°; CO_2 reduced to 20 ppm	Vertical cylinder, packed with plastic; water sprayed in	*a*	74	None	1.5	0.028

[a] Value not given.
[b] 10 ppm total gas content.
[c] 1440 lb/hr of CO_2 and 1300 lb/hr of air removed.

in this case costs the evaporator very little because it is degraded in the final condenser anyway. The major cost (other than the deaerator itself) is in the increased surface required in the heat reject condenser.

A number of flowsheets containing various types of equipment and operating at various temperatures were investigated. The advantages (+) and disadvantages (–) of these systems are summarized below:

Flash Deaerator	Countercurrent Packed Column	Countercurrent Tray Column
Low operating cost (+)	High operating cost (–)	High operating cost (–)
Low equipment cost (+)	Moderate equipment cost	High equipment cost (–)
Probability that one stage will do incomplete job (–)	Channeling makes it uncertain complete job can be achieved (–)	Probability complete job can be designed now (+)

90° Deaerator	140° Deaerator	250° Deaerator
Fits well into SEMS plant (+)	Awkward for SEMS, fits into MEMS (–)	Fits well into SEMS plant (+)
Large steam specific volume (–)	Moderate steam specific volume (+)	Small steam specific volume (+)
Minimum use of premium materials (+)	Moderate use of premium materials	High use of premium materials (–)
Stripping steam available at low cost (+)	Stripping steam has high availability cost (–)	Stripping steam has high availability cost (–)
High-cost jets or blowers (–)	Moderate-cost jets or blowers	No jets or blowers (+)
Lowest steam requirements (+)	Moderate steam requirements	Highest steam requirements (–)
Flash deaeration requires low flashdown (+)	Flash deaeration requires moderate flashdown	Flash deaeration requires high flashdown. Impossible in one stage (–)

Carbon Dioxide Pond	No Pond
Low-cost jets and blowers (+)	High-cost jets and blowers (–)
Low-cost pond not demonstrated to date (–)	No equipment cost for pond
More reaction time for acid (+)	Special care must be taken to give acid time to react before deaerator (–)
High operating cost	No operating cost

Weighing the advantages and disadvantages of the various systems, the following approach seems to make the most sense (see Fig. 27):

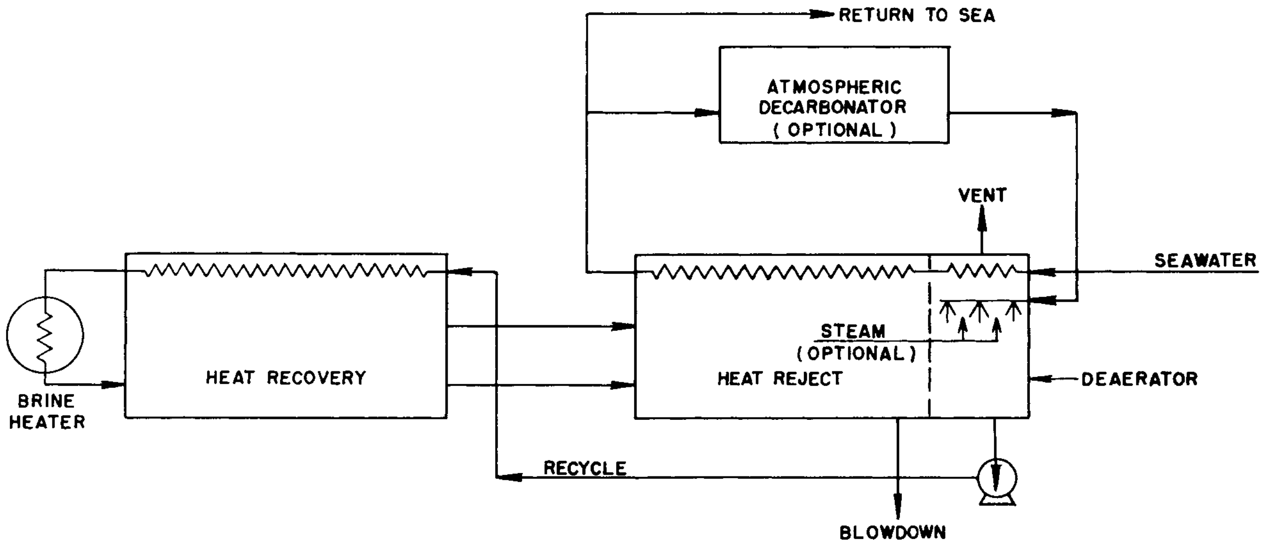

Fig. 27. Deaerator System Coupled to the Heat-Reject Section of a Flash Evaporator. Deaeration is at low pressure end of the plant (90° F). Deaerator is part of the heat-reject section and can be either flash, sieve tray, or packed column. Steam may be supplied to the deaerator from the last stage. A flash deaerator can also be a free-standing unit.

1. a low-head decarbonation pond if a satisfactory design can be demonstrated, operating at heat reject temperature (90°),
2. a 90°F flashing device followed by (if required) supplementary trays or packing.

One particular advantage of this approach is that it could be adapted to all MSF or VTE flowsheets, therefore limiting the amount of development.

3.2.5 Conclusions and Recommended Additional Work

Seawater deaerator design procedures have not yet reached the stage where large, economical units can be guaranteed for limiting O_2 to 10 ppb and CO_2 to 1 ppm. Some deaerators have been reported to approach or achieve these goals; however, the methods of chemical analysis used (and therefore the reported results) are open to some question.

Flash (spray) deaerators have very favorable equilibrium characteristics and do not require expensive stripping steam. They can be readily fitted into evaporator flowsheets. Unfortunately, virtually no test data are available on the actual mass transfer behavior. Tests are recommended to explore the following variables:

1. types of nozzle,
2. nozzle pressure drop,
3. amount of flashdown,
4. deaerator pressure,

5. loading (lb/ft^2) of column,
6. height of column,
7. performance of a two-stage column;
8. after the above variables are optimized, effect of column diameter should be investigated.

Packed-column and tray-type deaerators are of potential value for polishing the flashed feed. Additional mass transfer measurements should be made on seawater deaerators operating at and near rated conditions. Work is needed on the problems of distributing liquid to large-diameter columns.

Reported CO_2 analyses from deaerator systems are frequently inconsistent with the O_2 analyses. We are inclined to attribute these inconsistencies to the kinetics of reactions between acid and carbonates (particularly with suspended solids). From this point of view a decarbonation pond would be desirable for providing more time for the acid reaction. The type of pond which should be developed (from the cost standpoint) would be a low-head type containing baffles to bring bubbles to the surface.

There is a need for accurate analytical methods for low levels of CO_2 and O_2 in seawater. It is likely that existing methods with minor refinements will prove adequate. The uncertainties of the existing procedures have inhibited the proper development of test information from seawater deaerators.

3.3 VALVES

Water flow rates which will be encountered in distillation plants of 50- to 250-Mgd capacity will require valves of sizes heretofore encountered primarily in hydroelectric power systems, large aqueducts, and sanitary water plants. Valves designed for such applications are not necessarily optimum for distillation plants, where absolute pressures, differential pressures, and exposure conditions are different.

The objective of the study was to examine existing designs for cost and suitability for application on distillation plants of 50- to 250-Mgd capacity.

The scope of the study was as follows:

1. establish functional requirements for the major valve applications in principal evaporator cycles of 50- to 250-Mgd capacity,
2. solicit manufacturers of large valves to provide outline drawings, weights, recommended materials, pressure losses when fully open, prices, and any other pertinent information on the basis of the functional requirements determined in item 1,
3. tabulate and correlate the information obtained in item 2, and evaluate the designs offered.

The study should include examination of valve types, materials of construction, and control and mechanical operator problems. Corrosion, erosion, abrasion, cavitation, valve balance, and cost should be considered in evaluating designs and materials.

3.3.1 Valve Selection

Large valves are needed in desalting plants to throttle and control the flow of seawater and recirculating brine, usually in parallel streams. A single 50-Mgd flash evaporator train will typically have a seawater flow of 140,000 gpm and a recycle flow of 250,000 gpm. The size valves used would typically be 72 in. and 96 in. respectively.

Valves are required also to shut off and isolate evaporator trains, pumps, etc., so that these pieces of equipment can be maintained or repaired.

Several large power plants and sanitary water plants were inspected, and valve manufacturers were consulted to determine the applicability of existing experience to valves for service in 50- to 250-Mgd distillation plants. As a result of these investigations, it was concluded that butterfly valves are best suited for general evaporator plant service because their first cost is lower; they weigh less, require less room and support, have fewer working parts, and require less maintenance; although they suffer slightly more energy loss than gate valves, this penalty is more than offset by the lower initial cost.

Modern powerhouses and sanitary water plants generally use large butterfly valves in lieu of gate valves or check valves, where applicable. For example, the Chicago Central District Filtration Plant, which is the world's largest, with a rated capacity of 960 Mgd and a capability of 1700 Mgd, is equipped with 777 butterfly valves ranging in size from 14 to 108 in.

3.3.2 Materials of Construction

Experience at the saline water demonstration plants at Freeport, Texas, and San Diego, California, was reviewed to assess the problems which were experienced with valves in these plants. Both plants were equipped with lined butterfly and gate valves of relatively small size (largest 12 in.) compared with those needed for distillation plants of 50- to 250-Mgd capacity.

The Cyrus W. Rice Company has evaluated Freeport information as follows:

"The basic valve installed in this plant is a butterfly type with carbon steel body, Monel shaft, bronze disk, and Hycar or butyl seat. These valves have provided a low-maintenance installation which is satisfactory for a service not requiring positive flow shutoff. Typical installations are: brine pump discharge (with pneumatic positioner and gear operator) to control the brine transfer flow from effect to effect; condensate pump discharge (with gear operator) to allow manual control of condensate pumping flow rate; brine, condensate, and seawater feed pump suction (with handle operator); and vapor inflow and outflow at deaerator (with pneumatic positioner and diaphragm operator) to control deaerator vapor flow. Butterfly valve failures in this plant have been minimal and have been limited to deterioration of the resilient valve seats, which are easily removed and replaced.

"In general, valves have not been high-maintenance items in the plant, and the types of valves installed and materials of construction used have been suitable for the intended service, except for the inability to provide shutoff consistently."

Table 26. Materials of Construction for Large Butterfly Valves

Application	Material			
	Body	Disk	Shaft	Trim
Seawater intake	Ni-Resist	Ni-Resist	316 SS	316 SS
Deaerator feed	Ni-Resist	Ni-Resist	316 SS	316 SS
Makeup water	Ni-Resist	Ni-Resist	316 SS	316 SS
Seawater reject	Ni-Resist	Ni-Resist	316 SS	316 SS
Brine recycle	Mild carbon steel	Mild carbon steel	CN-7M-SS	CN-7M-SS
Product water	Mild carbon steel	Mild carbon steel	CN-7M-SS	CN-7M-SS
Blowdown	Mild carbon steel	Mild carbon steel	CN-7M-SS	CN-7M-SS
Drain	Mild carbon steel	Mild carbon steel	CN-7M-SS	CN-7M-SS

At San Diego, trouble was experienced with the Hycar lining because of bonding failure. The lining was removed, and these valves performed satisfactorily until the plant was dismantled. In view of the trouble experienced with lined valves, valve materials should be selected which will give satisfactory service without protective coatings or linings.

Based on the above experience and on recommendations from valve suppliers and users, the materials recommended for large butterfly valves are given in Table 26.

Valve damage is attributed to cavitation, corrosion, and erosion.

Cavitation resistance of the selected valve materials can be listed in the following order: 316 SS, carbon steel, Ni-Resist, and cast iron. Cavitation can occur in a butterfly valve when the valve is operated in a slightly open position or when the process requires large pressure drops across the valve. Thus, butterfly valves should not be operated continuously in a slightly open position, and good process design is needed to avoid flashing where pressure drops across the valve are high.

Corrosion resistance is improved by good deaeration and by the use of especially corrosion-resistant trim such as 316 stainless steel. There are potential stress corrosion problems with stainless steels at above 150 to 200°F, and other alloys might have to be used. Fortunately most large valves in evaporator plants operate below these temperatures.

Abrasion of the valve materials occurs because of wearing or grinding properties of the fluid. Abrasion of the valve bodies and/or seats should not be a problem if properly designed seawater intake systems are utilized. For instance, in Southern California it has been determined that if the seawater intake system utilizes submarine lines, most of the sand will be prevented from entering the evaporator system.

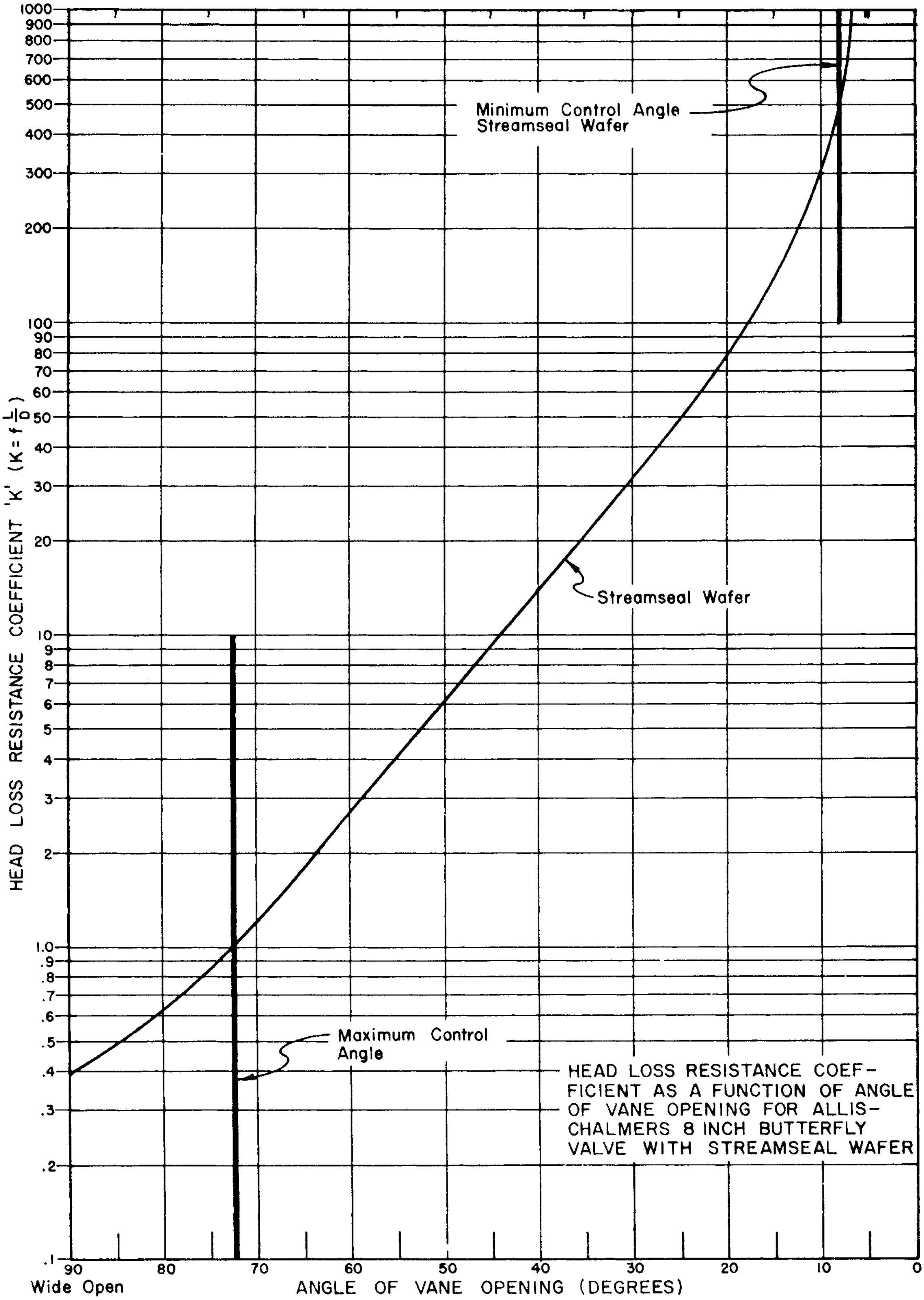

Fig. 28. Head Loss Coefficient as a Function of Angle of Vane Opening for 8-in. Butterfly Valve with Streamseal Wafer.

3.3.3 Design and Service Data on Butterfly Valves

Butterfly valves can be used for either on-off or throttling service. When completely closed, the edge of the disk compresses a rubber-type seat to produce a tight closure; continued operating force is not required to maintain the disk in the closed position. The valve disk is contoured for low head loss and is internally reinforced to withstand both static and dynamic forces. These valves are also adaptable to flow and level control, inasmuch as the coefficient of flow changes uniformly with the degree of opening. Figure 28 gives one manufacturer's data on flow coefficients. For cold seawater service, a Buna-N rubber material is used for valve seats. This material possesses a good combination of wear properties, resilience, and resistance to brine and seawater. The valve seat can be replaced without removing the valve from the line. For high-temperature service, a chlorobutyl rubber is employed. One valve supplier stated that this elastomer is good for extended service at temperatures up to 350°F. However, another indicated that experience in high-temperature service is insufficient for selecting a material which will give trouble-free service over long periods of time and many operating cycles.

The valves are powered by trunnion-mounted hydraulic cylinder operators which permit remote, local, automatic, or manual control systems and open-close, pressure, and flow control service.

A positioner is added to valves used for control purposes to maintain the proper vane position by means of fluid pressure from a control device. In the event of loss of operating fluid pressure, the valve is prevented from slamming closed suddenly, with resultant damage due to water hammer, by incorporating a gradual bleedoff feature in the cylinder design.

For subatmospheric pressure service, air inleakage is avoided by flooding the valve shaft packing glands with positive-pressure product water.

The primary disadvantage of a butterfly valve is that it causes a higher pressure drop in the fully open position than a gate valve for the same flow conditions. However, in an evaporator plant equipped with motor-driven pumps, the majority of valves are used for throttling service, and pressure drop is not the determining factor in the overall selection. A second disadvantage is that the unbalanced pressure across the valve seat requires an operating force to open the valve and hold it in the operating position. Valves with low seat differential pressures are expensive and impractical for large distillation plants.

3.3.4 Valve Prices

Functional specifications were forwarded to six large valve manufacturers to obtain design and cost information for valves ranging from 30 to 156 in., with head pressures from 70 to 300 ft, and in various materials of construction. At least two of these manufacturers stated they could supply the valves, but only one was in a position to provide estimated prices. These prices are given in Tables 27 and 28 and in cost curves, Figs. 29 through 35. There are listed the basic valve price, the operator price, and the charge for engineering and tooling.

Table 27. Estimating Prices for Butterfly Valves

Valve Size (in.)	Cost of Materials[a] (dollars)				Cost of Operator[b] (dollars)	
	1	2	3	4	M	C
			Maximum ΔH = 70 ft Head			
24	746	808	1,470	4,270	340	340
30	1,081	1,200	1,606	4,806	450	360
36	1,511	1,746	1,872	5,572	625	415
42	1,967	2,285	2,387	6,787	625	415
48	2,646	3,106	2,944	8,244	625	415
54	3,342	3,985	3,437	9,737	790	455
60	4,224	5,100	4,130	11,530	790	455
66	5,250	6,444	5,325	14,125	790	530
84	9,500	11,800	14,550	29,550	1,100	530
96	14,700	18,600	21,600	44,500	2,100	675
108			26,000	55,000	2,100	721
120			38,400	79,900	3,700	980
144			59,600	123,300	8,500	1300
150			65,800	136,000	10,200	1850
162			79,300	163,000	10,200	1850
			Maximum ΔH = 100 ft Head			
48	2,920	3,380	3,210	8,510	790	460
60	4,840	5,710	4,740	12,150	1,100	530
66	5,940	7,128	6,010	14,800	1,100	675
96	16,720	20,820	23,632	47,700	3,700	725
120			42,250	86,800	10,200	1170
144			67,350	134,250	10,200	1300
150			74,050	147,630	10,200	1850
162			89,650	176,800	28,000	1850
			Maximum ΔH = 200 ft Head			
48	3,510	4,550	4,570	12,800	1,100	675
60	5,650	7,950	6,410	17,900	2,100	790
66	7,100	10,100	8,335	22,000	2,100	980
96			33,480	69,000	8,500	1263
108			45,050	94,100	10,200	1820
120			59,520	123,900	28,000	3400
144			92,380	191,200	28,000	3625
150			101,990	210,700	33,000	3625
162			123,000	251,600	33,000	5000
184			168,300	343,400	33,000	6400
			Maximum ΔH = 275 ft Head			
48	3,680	4,700	5,250	14,700	2,100	725
108			51,800	108,200	28,000	2100
184			193,600	395,000	40,000	8000

Table 27 (continued)

Valve Size (in.)	Cost of Materials[a] (dollars)				Cost of Operator[b] (dollars)	
	1	2	3	4	M	C
			Maximum ΔH = 400 ft Head			
48			6,300	17,700	3,700	790
60			8,850	24,700	3,700	1100
66			11,500	30,400	8,500	1100
96			46,300	95,200	28,000	2100

[a]Material codes:

	Flanged 125 psig		Weld End	
	1	2	3	4
Body	Cast iron	Cast iron	Carbon steel	316 SS
Disk	Cast iron	Ni-Resist	Carbon steel	316 SS
Shaft trim	316 SS	316 SS	316 SS	316 SS

All valves to have resilient seats for bubble-tight shutoff at rated pressures. Seats available for temperatures to 300°F. Maximum line velocity assumed to be 8 fps.

[b]Operator code: M = manual operator, C = cylinder operator – 2000 psi minimum pressure.

Table 28. All Types: Pattern, Engineering, Test Equipment, and Valve Charges[a]

Valve Size (in.)	Charges by Material Types for All Heads (dollars)		
	1 and 2	3	4
24	500	700	1,200
30	500	700	1,200
36	500	700	1,200
42	500	700	1,200
48	500	700	1,200 (6000 ΔH = 400 ft)
54	500	700	1,200
60	500	700	1,200 (7000 ΔH = 400 ft)
66	500	900	1,400 (8000 ΔH = 400 ft)
84	500	900	1,400
96	6700	1,500	2,000
108		8,600	8,600
120		8,600	8,600
144		16,000	16,000
150		18,000	18,000
162		22,000	22,000
184		36,000	36,000

[a]These one-time charges are to be added to the valve costs, regardless of the quantities involved.

[b]Materials defined in Table 27.

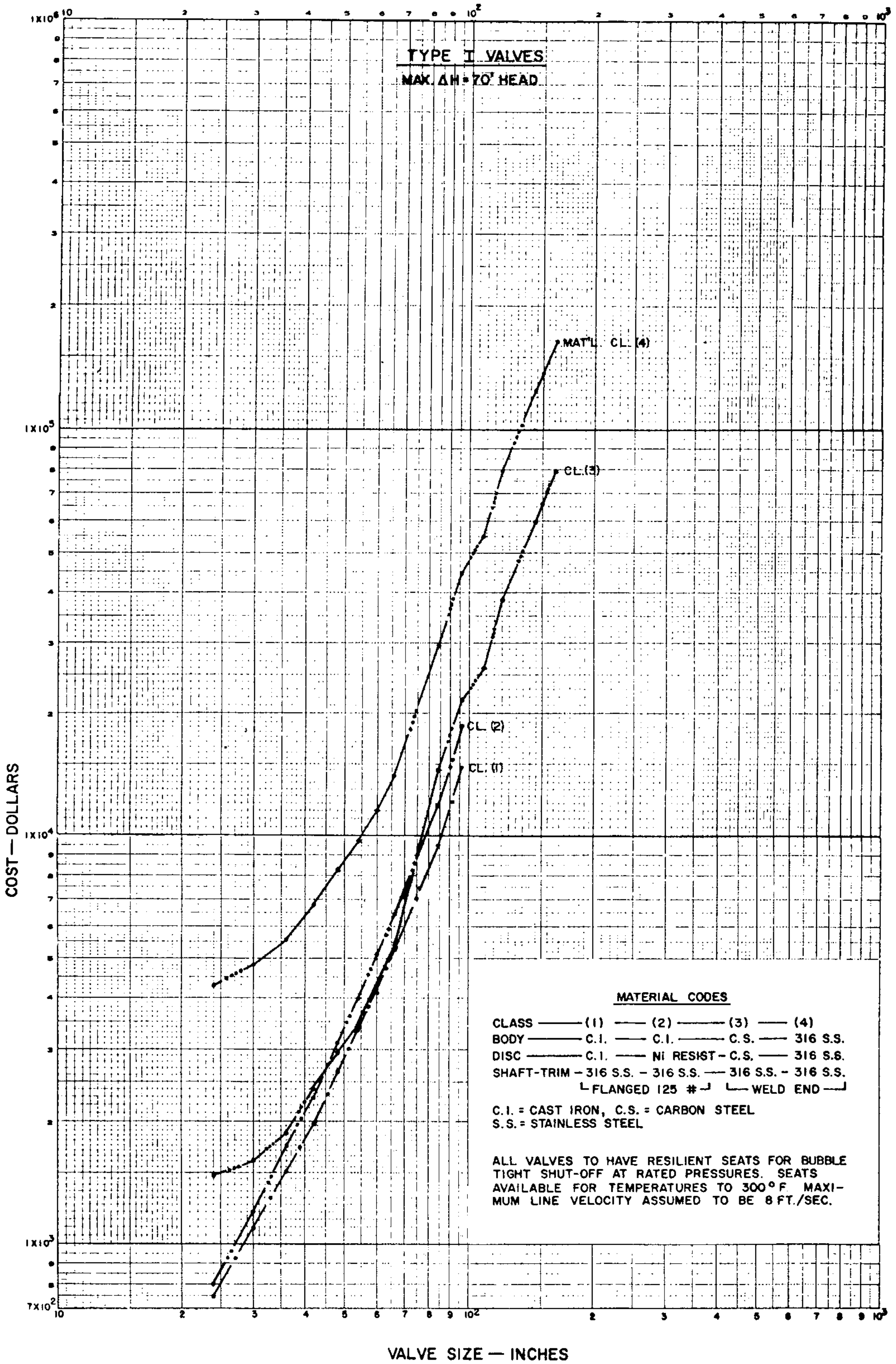

Fig. 29. Valve Costs as a Function of Size. Type I valves. Maximum $\Delta H = 70$ ft head.

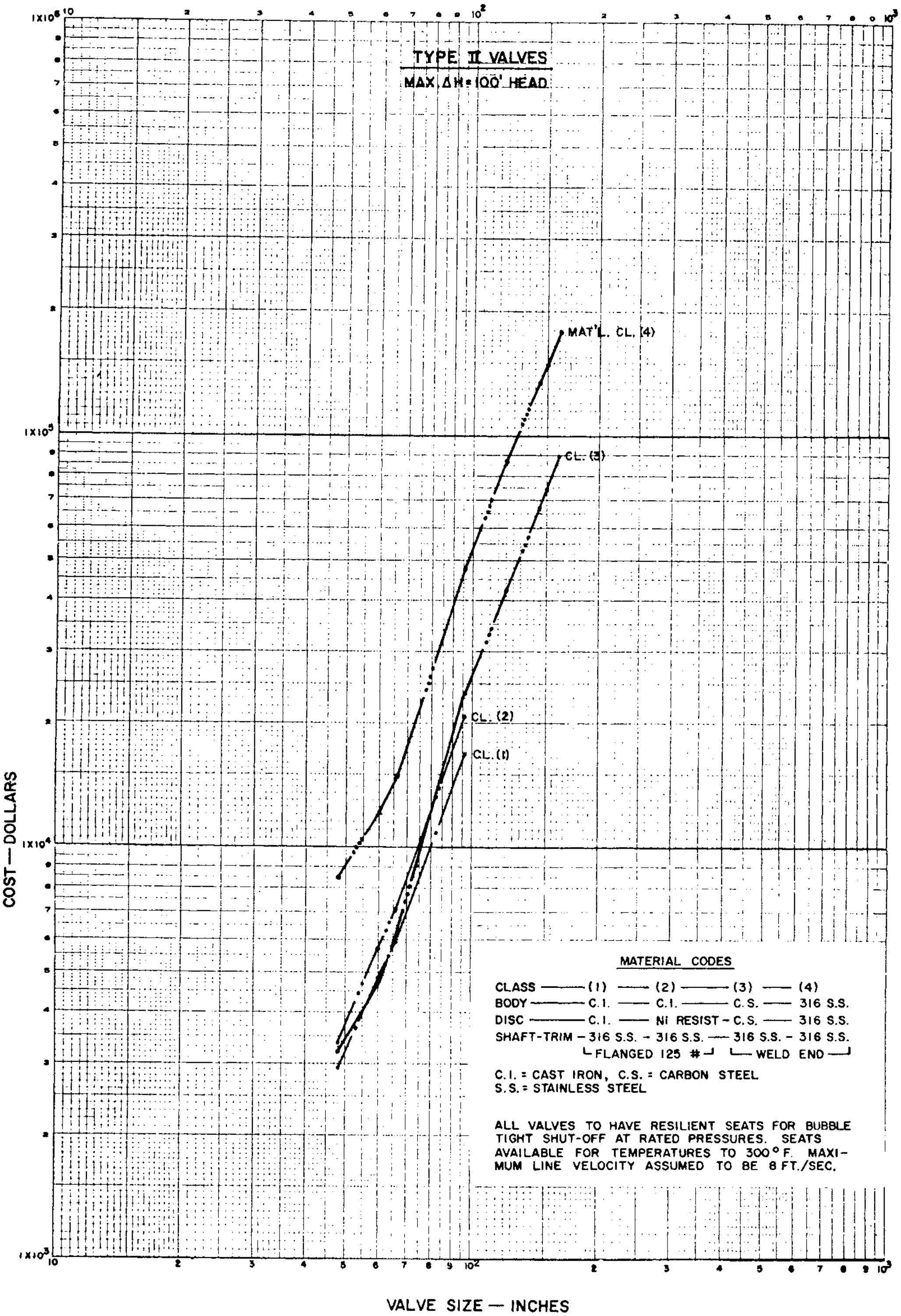

Fig. 30. Valve Costs as a Function of Size. Type II valves. Maximum ΔH = 100 ft head.

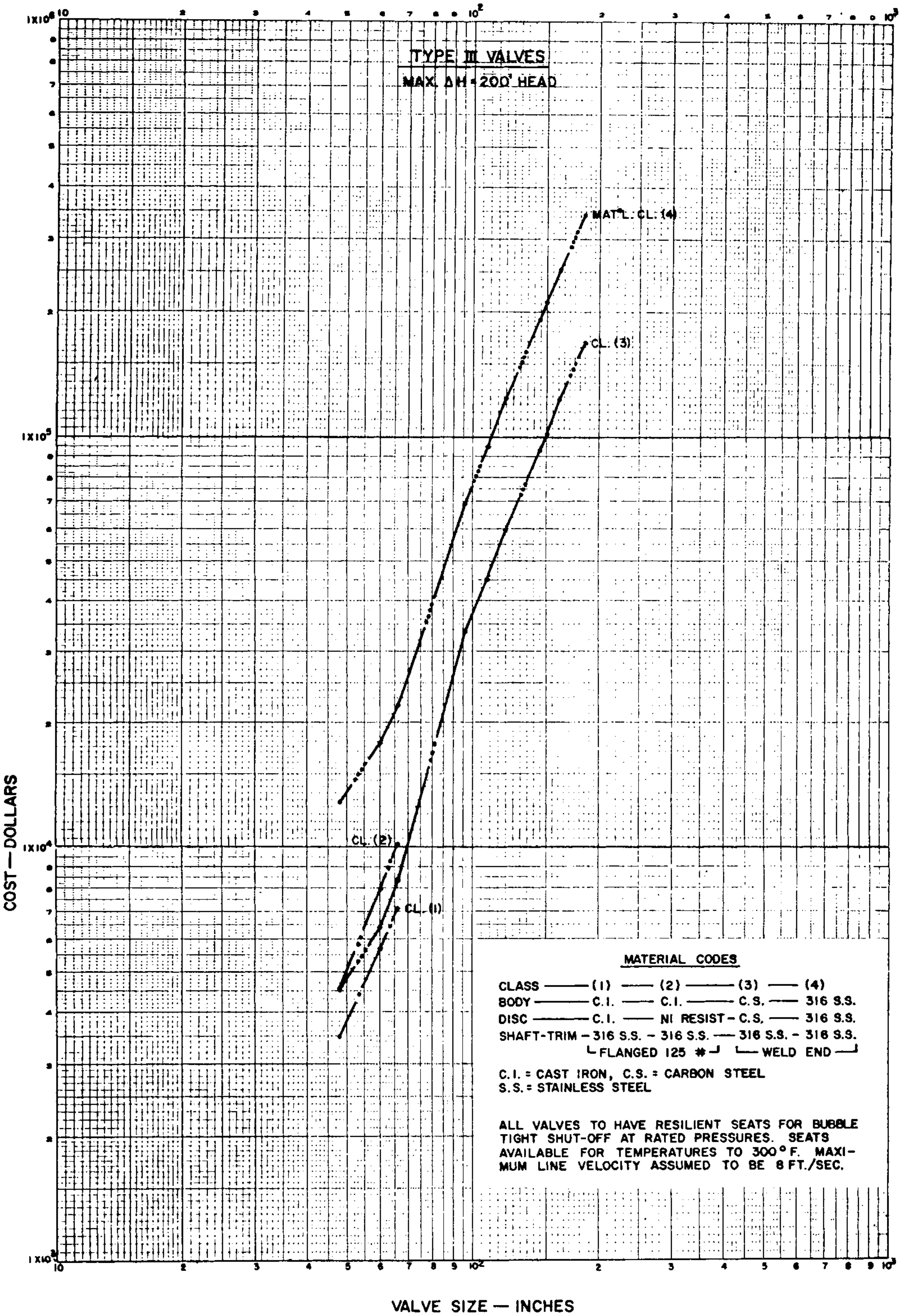

Fig. 31. Valve Costs as a Function of Size. Type III valves. Maximum $\Delta H = 200$ ft head.

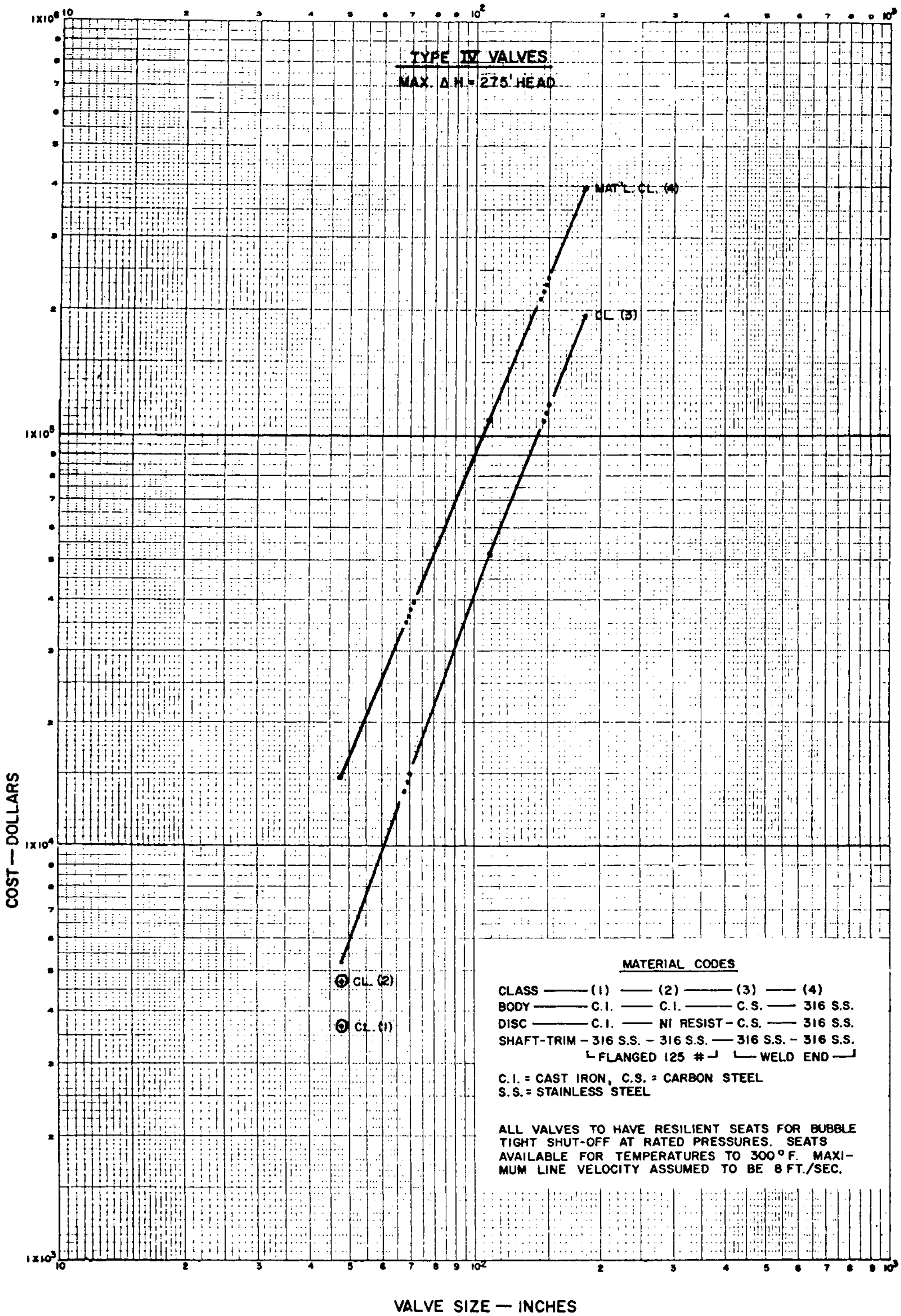

Fig. 32. Valve Costs as a Function of Size. Type IV valves. Maximum ΔH = 275 ft head.

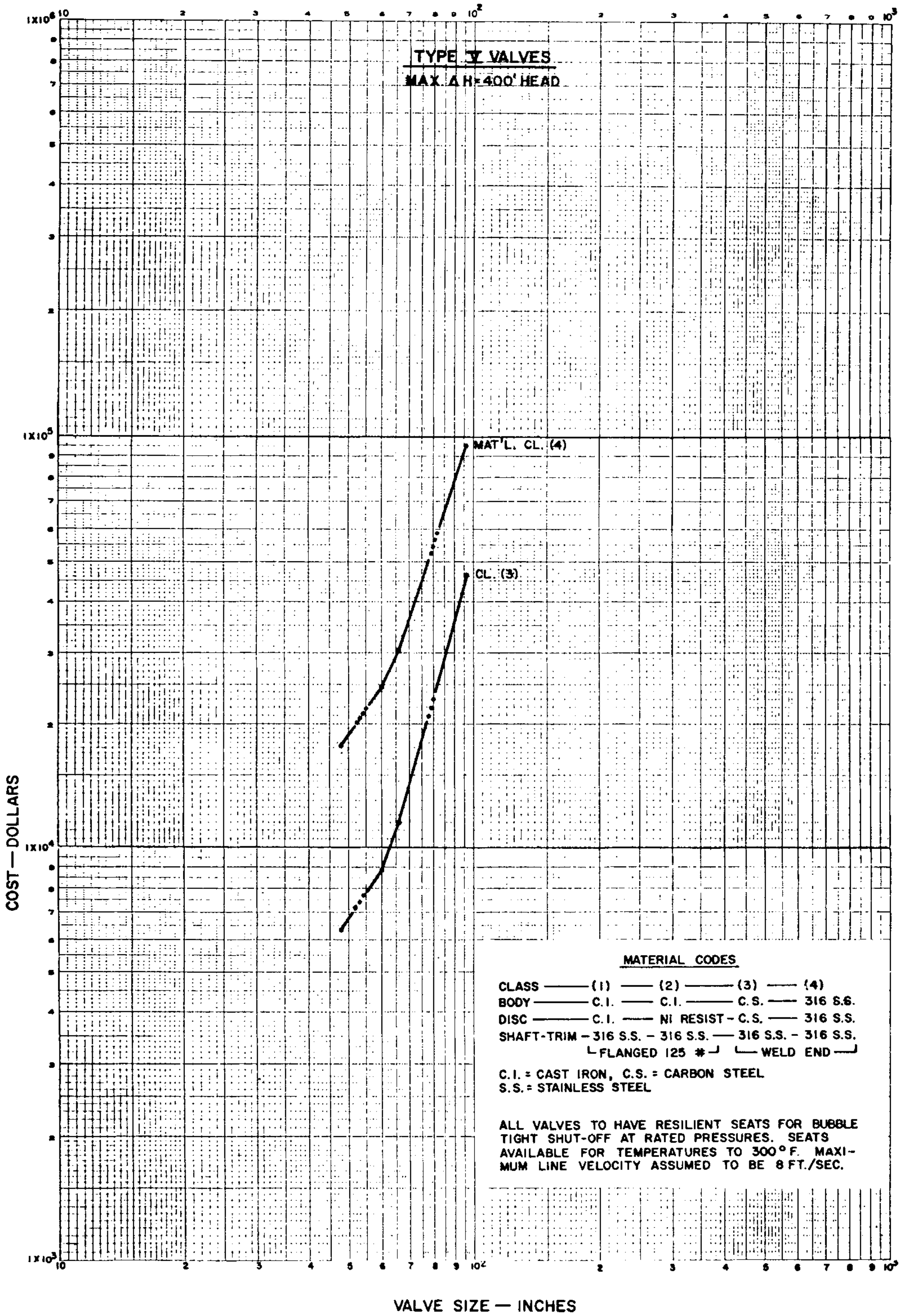

Fig. 33. Valve Costs as a Function of Size. Type V valves. Maximum ΔH = 400 ft head.

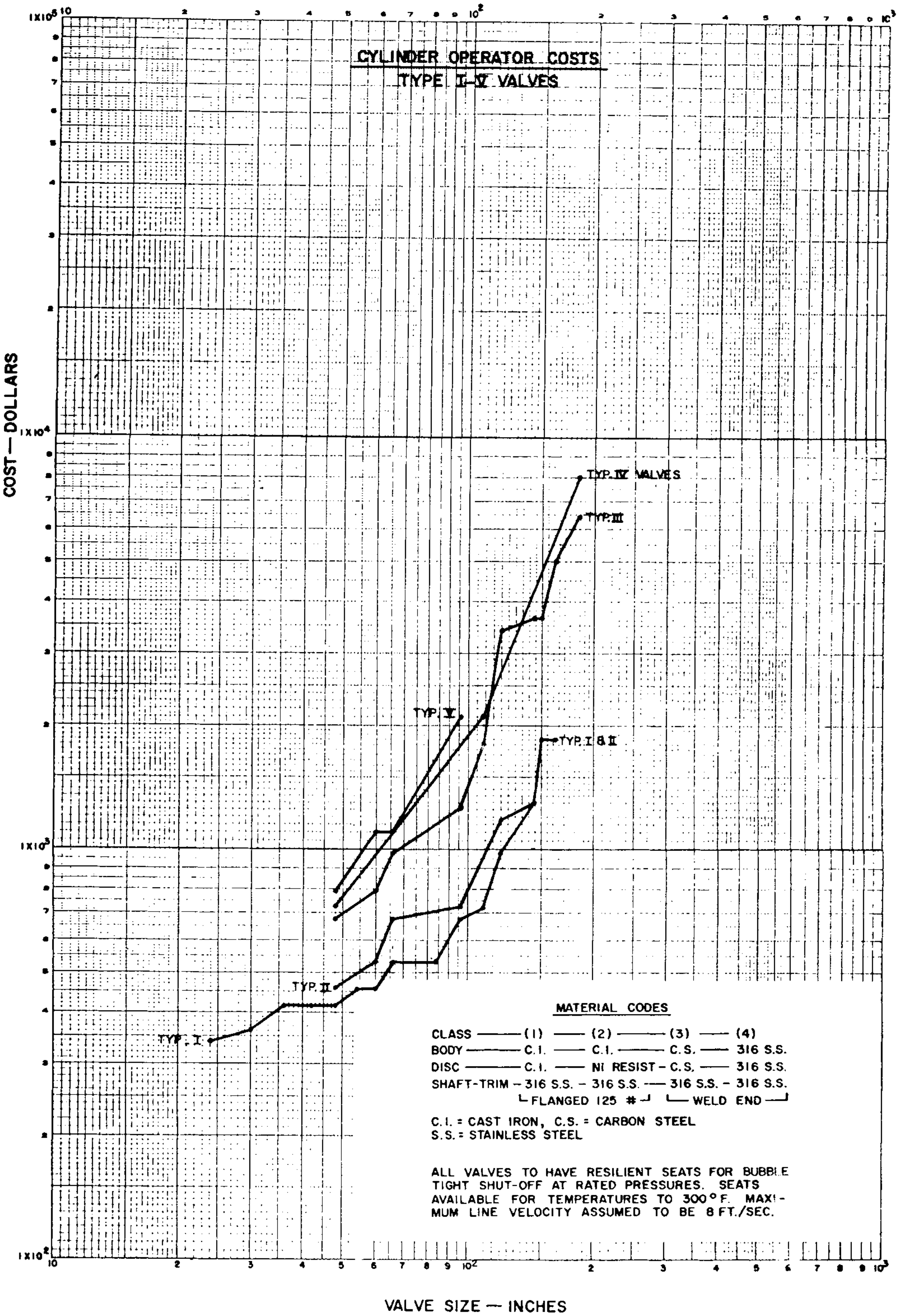

Fig. 34. Cylinder Operator as a Function of Size. Types I–V valves.

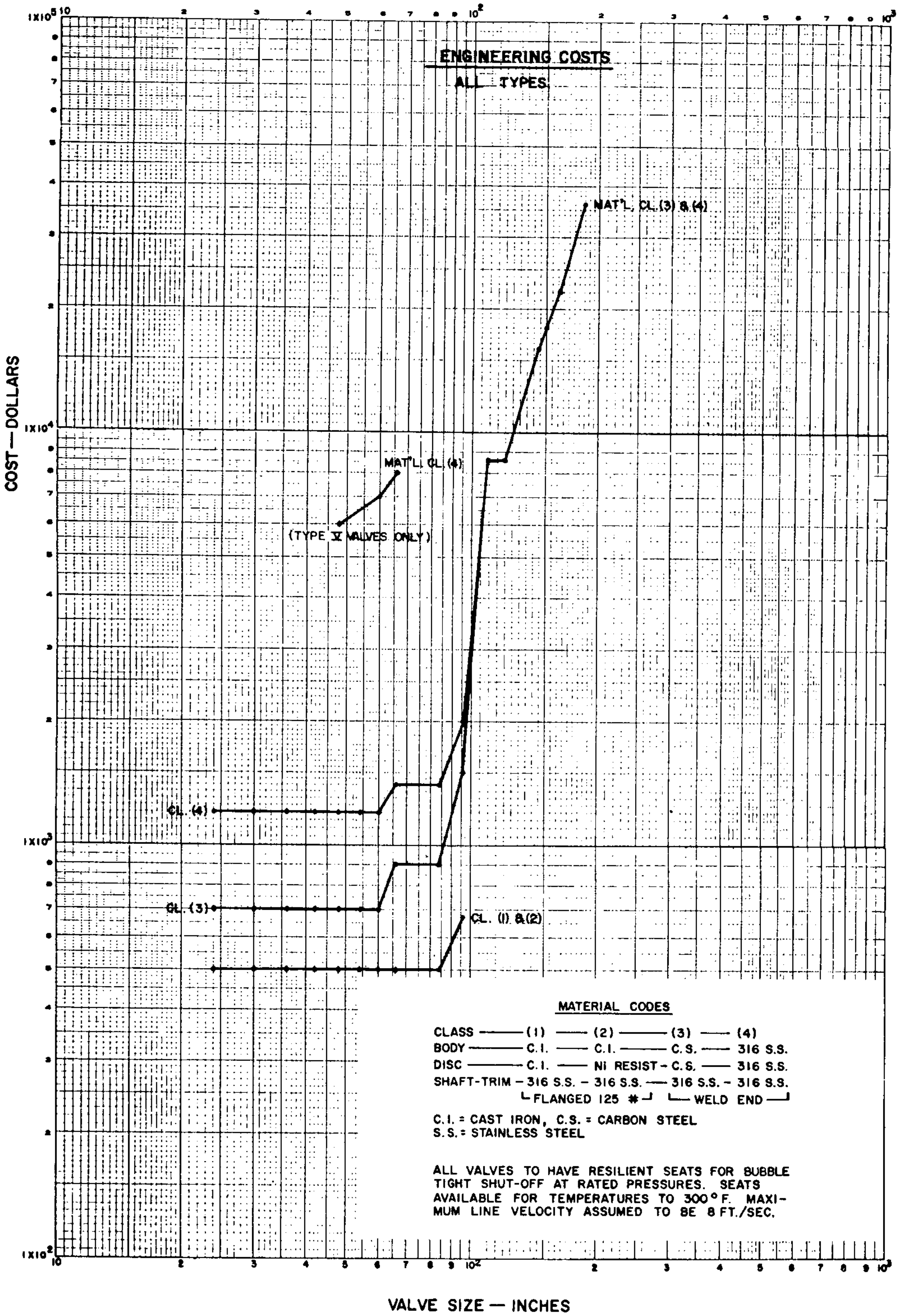

Fig. 35. Engineering Costs as a Function of Size. Types I–V valves.

Cost Comparison of Gate Valves and Butterfly Valves. – A comparison of prices for several large-size carbon steel butterfly and gate valves is presented in the following tabulation.

Size (in.)	150-psig Butterfly Valve with Hydraulic Operator	150-psig Gate Valve with Hydraulic Operator
48	$ 5,500	$15,250
54	8,000	22,000
60	10,500	29,000
66	14,000	37,000
72	18,000	47,000
84	27,000	62,000
96	40,000	73,000

An economic analysis was made to determine the most favorable type of valve, considering capital charges on first cost and the energy cost associated with pressure-drop losses in the fully open position. The butterfly valve was found to be the most economical, as shown by Table 29 for 72- and 96-in.-size butterfly and gate valves.

Table 29. Comparison of Butterfly and Gate Valves (72- and 96-in. Size)

	72-in. Seawater Intake		96-in. Brine Recycle	
	Gate	Butterfly	Gate	Butterfly
Flow, gpm	142,000	142,000	248,000	248,000
Velocity, fps	10	10	10	10
Fluid	Seawater, 65°F	Seawater, 65°F	Brine, 92°F	Brine, 92°F
Pressure drop, ft, fully open	0.231	0.625	0.231	0.625
Losses, hp	10	27	17.3	46.8
Losses, dollars/year at $0.002/kwhr	119	319	204	554
Cost, initial, dollars	47,000	18,000	73,000	40,000
Cost, dollars/year, fixed capital charge at 5.6%	2630	1010	4090	2240
Cost advantage of butterfly over gate, dollars/year	1420		1500	

3.3.5 Conclusions

Butterfly valves appear to be the best type of valve for desalination plant service, considering costs and previous experience. It would be desirable for additional information to be developed on valve seating and trim materials in desalting plant service.

3.4 VAPOR COMPRESSORS

The largest vapor compressors constructed to date have a capacity of the order of 1,500,000 cfm. For vapor-compression distillation plants of 50- to 250-Mgd capacity, larger compressors will be required. It will be necessary to have available designs and price estimates for compressors of capacities suitable for these plants before it will be possible to make realistic conceptual designs and cost estimates for the distillation plants themselves.

The objectives of the study were as follows:

1. to ascertain the state of the art of large axial-flow compressor design and construction and to determine from this whether compressors of the size needed are currently feasible,
2. to develop design and cost information.

3.4.1 Vapor Compressor Applications

A typical simple vapor compressor flowsheet is shown in Fig. 36. Steam is taken from a low-pressure evaporator effect and compressed into the steam chest of a high-pressure effect. The vapor compressor acts as a heat pump, transforming mechanical energy to thermal energy (the reverse of a steam turbine).

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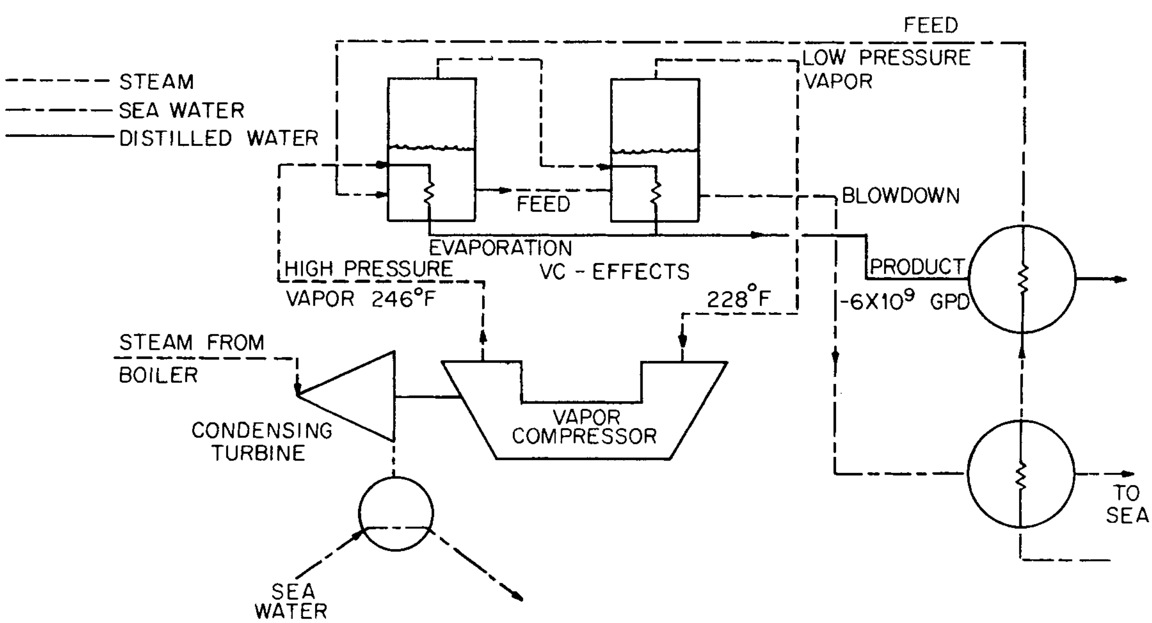

Fig. 36. Simple Vapor Compressor Flowsheet.

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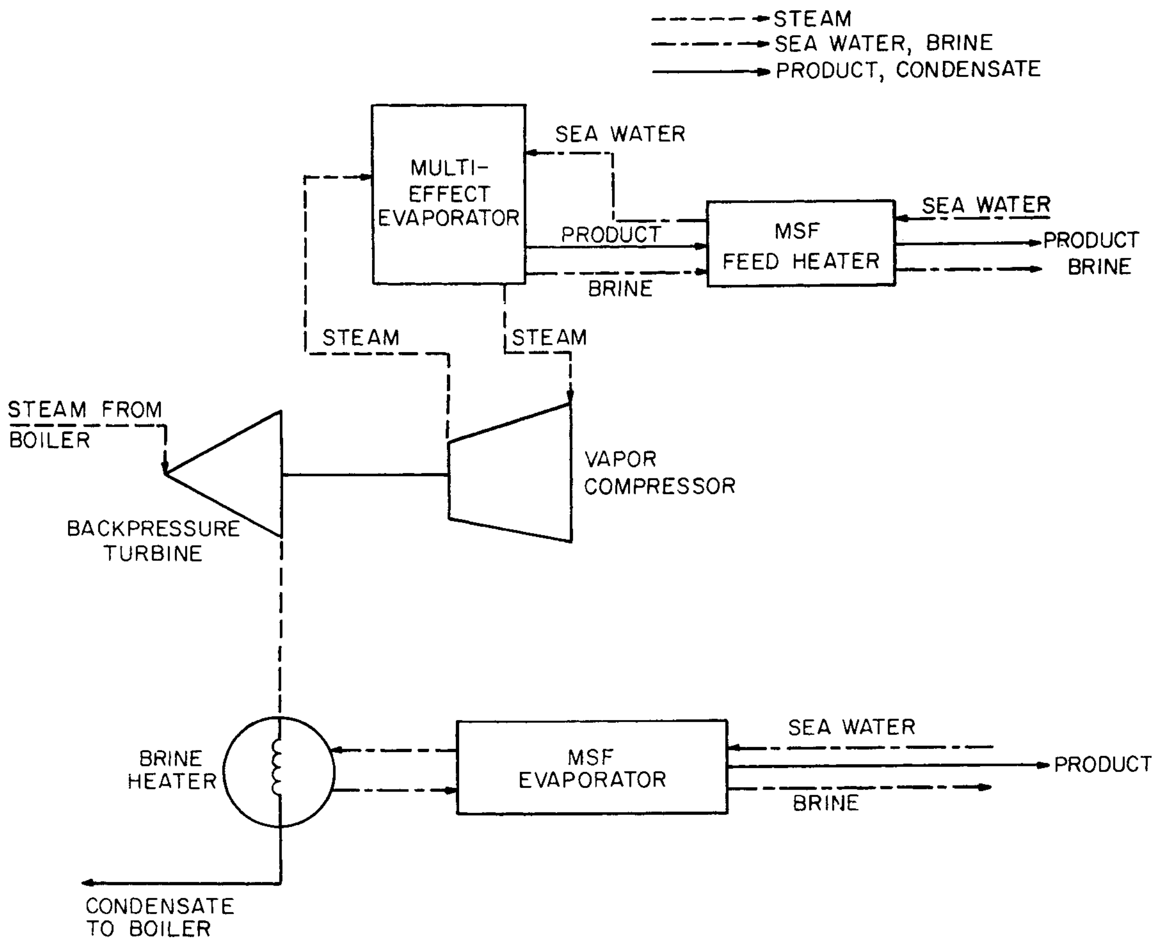

Fig. 37. Hybrid Flowsheet Showing Vapor Compression Combined with Flash Evaporators.

A typical vapor compressor system of this type could have the following characteristics:

Basis	50 Mgd of product
Shaft horsepower	115,000 hp
Suction pressure	17 psia
Suction flow	3,500,000 cfm
Vapor compressor efficiency	84%
Compression ratio	1.4
Evaporator ΔT per effect	8°
Performance ratio	58 lb of product per 1000 Btu of shaft work

The vapor compressor fits into hybrid systems as illustrated in Fig. 37. In that example flash evaporators are combined with vapor compressor evaporators to provide feed heating and primary evaporation with exhaust steam. Many variations of this type of system are possible, including the use of vertical tube evaporators.

The ratio of vapor compressor water production to MSF water production varies from about 1 to 5 in hybrid water-only seawater evaporators. If it is desired to produce electric power as well (dual-purpose plant), the vapor compression component is reduced. In the most common dual-purpose plant the vapor compression component is eliminated.

The vapor compression (VC) system has several advantages: namely, it can be operated economically as a water-only plant since it makes maximum use of the high-pressure steam; it requires less cooling water, and, hence, the site work and pumping power are reduced considerably; it appears to produce lower-cost water than the other types of water-only distillation plants; and a generator can be connected to the turbine compressor drive and by-product power can be made, if desired. Since it can be a water-only plant, it can be used in regions where there is a need for water but no need for power. Also, the VC system can be operated in conjunction with a dual-purpose plant to produce water during power off-peak conditions.

3.4.2 Present State of the Art

A considerable number of vapor compressors have been built to date with capacities ranging up to 1,500,000 cfm. Specialized compressors with capacities from 4,500,000 to 13,000,000 cfm have been built. The largest compressor built to date is the 13,000,000-cfm compressor located at the United States Air Force Propulsion Wind Tunnel at Tullahoma, Tennessee. Flows in this compressor system range from 7,000,000 to 13,000,000 cfm at the inlets with Mach numbers from 1.4 to 3.5 at the supersonic test section. The system includes one transonic compressor with three stages and four supersonic compressors. Pressure ratios on this system vary up to 7. While this machine is appropriately sized, it is a very advanced design and has specialized features which would not be considered in the vapor compressors to be studied here.

While there is no doubt about the feasibility of large vapor compressors, the optimization of any particular unit is likely to require some model testing so that the proper flow configuration and efficiency are obtained. Methods of injecting moisture to reduce superheat and improve efficiency should be developed. The development of low-cost materials for blades, such as epoxy-impregnated fiber glass, would also be desirable to reduce the cost of the equipment.

3.4.3 Cost Trends

Several manufacturers were approached for cost information on large vapor compressors. Replies were received from two manufacturers. The following cost relationship was derived for machines of compression ratio 1.4 to 1.7, suction pressures of 15 to 30 psia, and power requirements of 100,000 to 1,100,000 hp:

$$\text{compressor cost, dollars,} = 102{,}000 + 1392 \left(\frac{\text{hp}}{1000}\right)^{0.855} (V_g)^{0.746}\,(\text{C.R.})^{-0.177}\,,$$

where

hp = horsepower requirement for machine,

V_g = steam specific volume at suction, ft^3/lb, and

C.R. = compression ratio.

The cost of steam turbine drives is given in Fig. 38.

On this basis a typical 50-Mgd vapor compressor would cost $900,000 and its gearing and drive $3,000,000. If the compression ratio were 1.4, it would be about a four-stage machine.

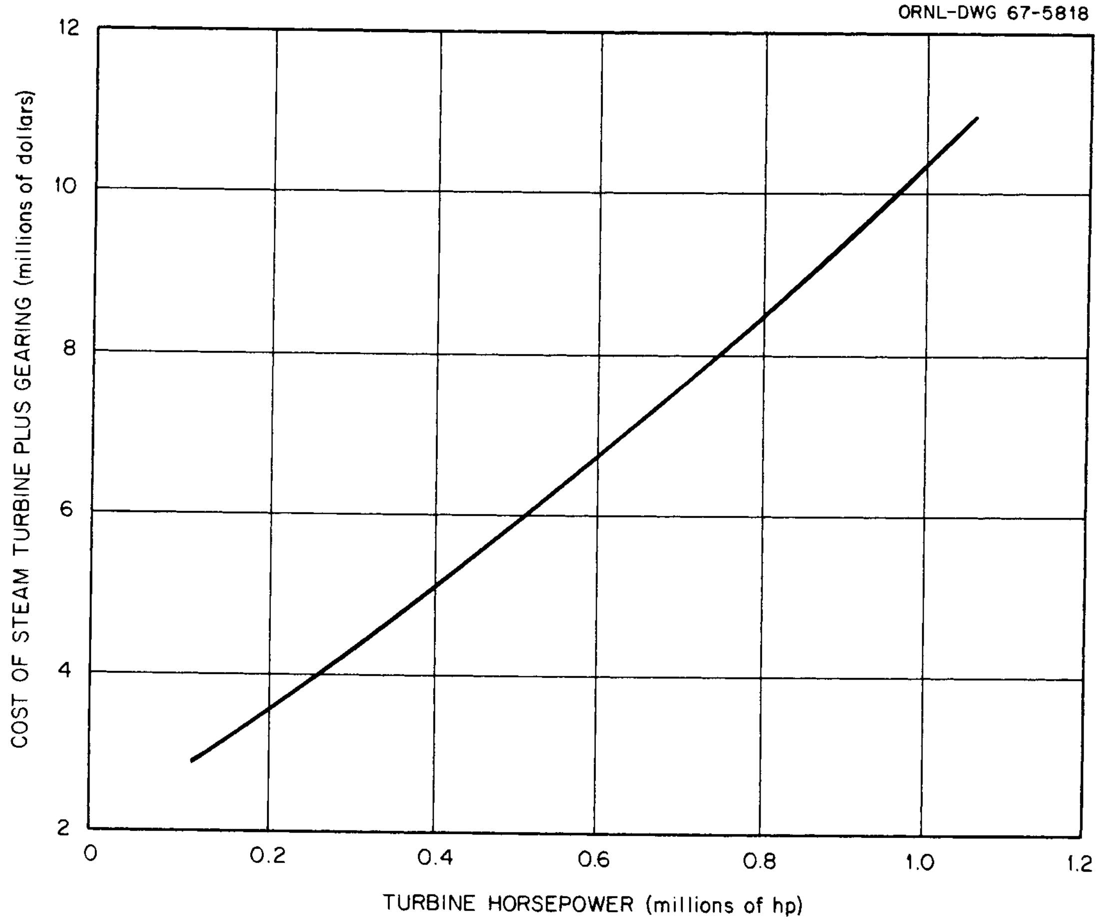

Fig. 38. Compressor Drive System Cost.

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