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OFFICE OF SALINE WATER

**CONCEPTUAL DESIGN OF A
250-MGD VERTICAL TUBE EVAPORATOR
FOR SEAWATER DESALINATION**

**PHASE I
SELECTION OF FLOWSHEET, LAYOUT
AND DESIGN CRITERIA**

R. P. Hammond, Program Director

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R. P. Hammond, Program Director

Report Compiled By

E. C. Hise

SEPTEMBER 1967

OAK RIDGE NATIONAL LABORATORY
Oak Ridge, Tennessee
Operated by
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for the
UNITED STATES ATOMIC ENERGY COMMISSION

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TABLE OF CONTENTS

	<u>Page</u>
ACKNOWLEDGEMENT	v
PHASE I SUMMARY	vii
INTRODUCTION	1
BASIS OF STUDY	2
Objectives and Scope	2
Ground Rules	2
OSW Ground Rules and Assumptions	3
ORNL Design Criteria	4
1. EXPLORATORY SHELL DESIGN AND EVALUATION	4
1.1 Preliminary Flowsheet	6
1.1.1 Purpose	6
1.1.2 Basis	6
1.1.3 Parametric Survey	6
1.2 Preliminary VTE Tube Bundle Design	8
1.3 Rectilinear Plant Design	15
1.3.1 Preliminary Studies	15
1.3.2 Plant with Rectangular Tube Bundles	25
1.3.3 Plant with S _e micircular Tube Bundles	30
1.4 Concentric Ring Plant Design	33
1.4.1 Preliminary Studies	33
1.4.2 Plant Design	36
1.4.3 Structural Design	36
1.5 Vertical Plant Design	38
1.5.1 Preliminary Studies	38
1.5.2 Plant Design	41
1.5.3 Structural Design	44
1.6 Cost Analyses of the Structural Variations	46
1.6.1 Summary	46
1.6.2 Basis of Cost Information and Method of Analysis ...	46
1.6.3 Rectilinear Structures	49
1.6.4 Concentric Ring Structure with 10-ft Tubes	53

1.6.5	Vertical Structure with 5-ft Tubes	<u>Page</u> 53
2.	EXPLORATORY PROCESS DESIGN AND EVALUATION	54
2.1	Comparison of Conventional and Enhanced Tubing	54
2.2	Comparison of Multistage-per-Effect Feed Heating with Single-Stage-per-Effect Feed Heating	56
2.3	Parametric Studies	60
2.3.1	Evaporator Optimization	60
2.3.2	Feed Heater Optimization	67
3.	DESIGN OF PLANT COMPONENTS AND DETERMINATION OF DESIGN VALUES	81
3.1	Tube Bundle Design	81
3.1.1	Heat Transfer Coefficients	82
3.1.2	Friction Factor Measurements on Tube Banks Composed of 3-in.-OD Smooth and Fluted Tubes on Triangular Pitches of 1.15 and 1.75	90
3.1.3	Tube Sheet Stress Analysis	92
3.1.4	Bundle Mechanical Design	94
3.1.5	Tube Sheet Layout	94
3.1.6	Tube Sheet Drilling	100
3.1.7	Tube to Tube Sheet Joint	101
3.1.8	Seal Welding	103
3.1.9	Nozzles	104
3.1.10	Tube Sheet Corrosion Protection	106
3.1.11	Tube Fabrication Cost	107
3.1.12	Fabricated Bundle Cost Estimates	108
3.2	Deaerator Design	109
3.3	Entrainment Separators	111
4.	SELECTION	114
	REFERENCES	117

ACKNOWLEDGEMENT

In the course of this study we have investigated and used ideas and information from conceptual design studies and technical reports prepared for the Office of Saline Water, U.S. Department of the Interior, by various contractors and we have benefitted from discussions with representatives of these organizations. The final selection of a plant configuration and process is, in our opinion, the most economical for the ground rules established. Another set of conditions could show some other arrangement to be advantageous.

We wish to acknowledge particularly the General Electric Company's development of fluted tubes, the work of the W. L. Badger Company on multistage feed heating, the C. F. Braun Company's conceptual design of the multistage feed-heating process and the concentric-ring plant, the Stearns-Roger Manufacturing Company's operating data from the Freeport Plant, and the contributions of Ferris Standiford of the W. L. Badger Manufacturing Company and Professor Bromley of the University of California at Los Angeles with respect to the vertical plant concept, and the many companies that supplied cost information for various components.

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PHASE I SUMMARY

This Phase I report on the conceptual design of a 250-Mgd vertical-tube evaporator describes the preliminary engineering steps leading to selection of the flowsheet, plant layout, and other basic criteria for the design concept which will be developed in more detail as Phase II of the conceptual design study. The design and analytical comparative evaluations reported in Sections 1 and 2 of this report led in some cases to clear cut selections for the final plant design and in other cases indicated that there is a considerable latitude for a designer to exercise engineering judgement. The most striking of the comparisons was that of a plant with conventional smooth tubes in the vertical evaporator to one with high-performance vertical tubes of the doubly fluted type developed by General Electric. The fluting increases the heat transfer coefficient 200% to 300% while increasing the tubing cost only 5% to 10%. The comparison of the multistage-per-effect feed heating with single-stage-per-effect was less dramatic but clearly favored the multistage feed heating. The rectilinear plant design was marginally cheaper than the concentric ring or vertical designs and was best suited to the multistage feed heating process. The parametric studies which examined a range of such variables as number of effects, concentration ratio, feed heater performance ratio, and several terminal temperatures indicated that there is a fairly large region in which changes of capital cost are compensated by changes in operating cost in such a way that the cost of water is virtually unaffected. Some of the final design values were chosen therefore for reasons other than economic optimum and some were chosen arbitrarily.

The experimental measurements discussed in Section 3 led to considerable confidence in the validity of the design heat transfer coefficients and of the design steam pressure loss through the bundles of three-inch fluted tubes. Cost estimates by the staff and by appropriate manufacturers were used to establish materials and fabrication costs.

The heat costs specified in the ground rules and the developed plant costs for the chosen design were balanced to obtain a plant performance ratio that would yield near minimum cost water.

The plant selected for the final conceptual design is a 15-effect, falling film, doubly-fluted-tube evaporator with a 50-stage flash feed heater-product cooler. The overall performance ratio is 13. The four-train shell structure is rectilinear having the evaporator integrally housed and directly above the feed heater-product cooler. The preliminary cost estimates of this design indicate a capital cost of about \$0.30 per daily gallon or a water cost of about \$0.12 per 1000 gal.

INTRODUCTION

Conceptual design studies of 50-Mgd plants have indicated that, in comparison with flash evaporators, multieffect vertical-tube evaporators have advantages which provide a potential for reduced cost of product water. Design and cost information on the vertical-tube evaporator (VTE) is at present much less detailed than corresponding information on flash evaporators. A thorough study, investigation, and conceptual design of vertical-tube evaporator plants will contribute considerably to the knowledge of desalination and will serve as an aid in making decisions concerning the type of large capacity plants which should be developed. The Office of Saline Water is undertaking the development of vertical tube evaporators to the point where their potential can be defined clearly. This design will be related closely to the output of the overall program.

In planning the work necessary to develop a conceptual design of a 250-Mgd vertical-tube evaporator for desalting seawater, it was provided that Phase I would be concerned with determining what design and process alternatives appeared most promising and which of these should form the basis for the actual conceptual design. Phase II of the planned work is concerned with the development of the conceptual design incorporating the optimum features indicated by the Phase I studies.

Since there are virtually no independent variables in the engineering-economic problem of evaporator design, the exploration and evaluation of various shell and process alternatives must proceed as a series of iterations. A process description is assumed and a design-cost estimate made. On the basis of these costs, new values for the affected process parameters are calculated, a new process description written, and a more detailed design-cost estimate made. Therefore, Phase I represents several cycles of iterations reported generally in chronological order. It is important that the reader bear this in mind since succeeding cost estimates and flowsheets will reflect changes from preceding ones.

Section 1 is concerned primarily with the investigation of various arrangements of the shell and is composed of design studies based on first and second generation design values and costs. The usefulness of the cost analyses made in Section 1 then lies in their relative magnitude rather than the absolute values, which may shift as more information is developed.

Section 2 is composed of design and analytical studies of variations in process and heat transfer surface based on first-through-third-generation design values and costs, so that again the cost analyses are useful primarily for comparison purposes.

Section 3 describes the development for the final design of those costs and design values which the work of Sections 1 and 2 indicated to be needed. These are for items peculiar to the VTE, since for those items common to the MSF, the values of that design study¹ were used. The design values were measured experimentally and the costs were obtained from appropriate manufacturers and/or developed by standard cost estimating procedures.

In Section 4 the selection of the final design is discussed. It is pointed out that the final design does not necessarily represent the optimum but that it is one of a family of plant designs whose costs are close to the minimum for the ground rules of this study.

BASIS OF STUDY

Objectives and Scope

The objective of the study is to develop feasible VTE designs for large dual-purpose plants on a basis which will permit economic comparison with the multistage flash plant previously studied by ORNL. It is intended that the designs will incorporate advanced technology available from the OSW program.

The scope of Phase I of this study was defined by OSW as follows:

1. Evaluate the design and cost of alternative VTE shell configurations, including the rectangular layout, the concentric ring design and the silo design. Information on these designs will be obtained from the 50-Mgd conceptual design reports* and from proponents of the various configurations.
2. Evaluate and compare the costs of alternative process designs in the falling-film configuration, and various feed arrangements.
3. Develop ground rules and design criteria comparable to those used in the 250-Mgd MSF design.
4. Evaluate alternative materials.
5. Select the best configurations of process and shell configuration for a falling-film system for Phase II, a more detailed conceptual design.

Ground Rules

The ground rules for the study are given below and are, in general, those used for the multistage flash conceptual design study¹ so that a comparison of the two plants will be as straightforward as possible. A few variations were necessitated by the physical differences of the two plants, the most notable differences being in the heat transfer surface. Two types of vertical tube surfaces are considered in this study, one being the 3-in.-OD doubly fluted tube with the characteristics described in the General Electric conceptual design report,² and the other being the 2-in.-OD smooth tube having the characteristics described in the report of the OSW demonstration plant in Freeport.³

*For the list of 50-Mgd conceptual design report, see "1965 Saline Water Conversion Report," United States Department of the Interior, Office of Saline Water, page 251, U.S. Govt. Printing Office, Washington, 1965.

OSW Ground Rules and Assumptions

The following design criteria were supplied to ORNL by OSW:

1. The plant is to be located in Southern California.
2. All costs are to be expressed in terms of 1965 dollars.
3. California labor rates are to be used for estimating the cost of the evaporators.
4. Capital charges are to include 5.185% as an amortization charge (3-1/8% interest for 30 years), 0.25% for insurance, and a figure for interim replacement costs (to be determined by ORNL).
5. The plant load factor is to be 90%.
6. Land costs are to be supplied by OSW.
7. Interim replacement cost is to be evaluated for each plant component. The choice of materials for components is to be analyzed on the basis of replacement costs as well as other capital and operating costs..
8. The energy source for the plants is to be a heavy-water-moderated organic-cooled reactor of the appropriate size. The evaporator designs, however, are to be useful with other nuclear or fossil energy sources.
9. The seawater temperature is to be taken as 65°F.
10. The seawater intake system design is to be similar to that of the 250-Mgd MSF design.
11. Makeup water is to be treated with sulfuric acid for scale control.
12. The product water is to be delivered at 25 psig at the plant battery limits. An evaluation is to be made of cooling the product water to 90°F vs allowing the water to cool by evaporation in the reservoir.
13. Component designs developed by OSW and its contractors are to be used to the extent feasible.
14. An evaluation is to be made of tubing and shell materials.
15. Pumps (product and brine reject) are to be provided with a barometric leg to supply the appropriate suction pressure.
16. An evaluation is to be made of a minimum number of large pumps vs multiples of smaller pumps.
17. Hydraulically operated butterfly-type valves are to be used for shutoff and control.

18. The instrumentation system is to be described.
19. Possibilities of improving vapor disengagement, design of condensing bundles, removal of noncondensables, and system hydraulics are to be investigated.
20. Operating and maintenance costs are to be estimated based on a single operating staff for both power and water plants.
21. Operating control for the plant is to be based on a single control room for the evaporator area.
22. Designs are, in general, to be based on current technology.
23. The heat transfer coefficients and operating requirements for the enhanced vertical-tube surfaces are to be determined by a review of experimental data.
24. The heat transfer coefficients and operating requirements for the conventional vertical-tube surfaces are to be in accord with the experience at Freeport.

ORNL Design Criteria

1. Reference energy costs assumed for a 3300-Mw(t) heavy-water-moderated organic-cooled reactor are:

Prime steam at 677°F and 800 psia	13.0¢/MBtu
Exhaust steam at 295°F, saturated	6.8¢/MBtu
Exhaust steam at 254°F, saturated	5.5¢/Mbtu
Electric power	2.0 mills/kwhr
2. The maximum tube velocity specified is chosen to avoid severe erosion for the tube materials selected.
3. The product purity is defined in relation to entrainment separator design.
4. The ground elevation at the heat reject portion of the plant is 35 ft above mean sea level.

1. EXPLORATORY SHELL DESIGN AND EVALUATION

The objectives of this portion of the work were to (1) compare the economics of three different evaporator shell configurations, (2) define the nature and extent of the problems that require solution prior to performing a conceptual design, (3) develop cost data for further iterations of the process flowsheet.

These objectives were accomplished by making partial designs of the three shell configurations based on a common process flowsheet. This reference flowsheet was not intended to be optimum, since the information necessary for optimization had not been developed during initial phases of this study; however, a common process flowsheet, even without optimization and without the best values for all variables, still provided an adequate basis for comparative evaluations. The partial design of each configuration consisted of a layout of the complete plant plus detail designs of three or four effects and of the plant section where the transition from ten-ft-long tubes to five-ft-long tubes occurs, when applicable, between effects VII and VIII. These three or four effects were chosen to enable an analysis of effect cost as a function of condensing temperature and to examine the effect on plant design at the transition from ten-ft- to five-ft-long.

The conclusions reached in this work were that, at least within this set of ground rules, there were no advantages found in the concentric ring nor in the vertical plant designs, that the use of ten-ft-long doubly fluted tubes throughout was more economical than changing to five-ft-long tubes at the low temperature end, and that continuous, rectangular vertical-tube bundles were more attractive than separate semicircular bundles.

The concentric ring plant design has the apparent advantages of being contained in a cylindrical vessel and of having increasing size vapor flow passages and bundle entry face as the specific volume of the vapor increases. However, the diameter of the vessel required for a plant of this capacity is such that the outer shell differs little from a flat wall. Although the vapor passages do increase, the reference concentric ring design still required entrainment separators; and the partial design that had sufficient disengagement volume for gravity separation was excessively large and expensive. The tube bundles for the outer effects became quite long and shallow so that there did not appear to be a satisfactory way to accumulate and vent non-condensables.

The vertical plant has the apparent advantage of not requiring intereffect pumps since the feed can be pumped to the first effect which is located at the top and then proceed through the rest of the effects by gravity. Partial designs of several arrangements of such a plant were made and costed. The additional structural cost offset a good bit of the savings in pumps and, because this design required all of the feed to be lifted to the top of the plant whereas the feed requirement of the doubly fluted tube requires only a portion of the total feed to pass through any given effect, the increased pumping power offset the rest.

It is recognized that these investigations were not exhaustive, that they were based on a specific set of conditions, and that some other set of conditions or a more ingenious arrangement may show a configuration in a more favorable light; but it is also felt that these investigations were adequate to indicate that there is no strong incentive to pursue either of these configurations which are, at least, unusual and which offer serious problems in adjusting to an efficient feed heating process.

Although it was not a major objective of this program, some effort was spent on the question of the best length for the evaporator tubes. The heat transfer data from the General Electric pilot plant experience at Wrightsville Beach indicated that the performance of the ten-ft-long doubly fluted tube was inferior to the five-ft-length and that this penalty increased with decreasing temperature. This difference in performance is a term in the expression for the design heat transfer coefficients so that the penalty can be readily calculated. Designs were made and costed for plants having five-ft-long tubes in effects condensing below 190° and for plants having ten-ft-long tubes throughout. The cost of the added surface for the long tubes was more than offset by the cost of the added tube sheets, nozzles, and fabrication for the five-ft tubes. Again, it is recognized that this is not a definitive study and that the possibility exists of further reducing the plant cost through a better understanding of the effect of the L/D ratio.

Various tube bundle configurations were designed and costed. The rectangular unit bundle with a folded non-condensable cooler and vent section was the most attractive from the standpoints of steam pressure drop and velocity, physical strength, and cost.

1.1 Preliminary Flowsheet

1.1.1 Purpose

A preliminary flowsheet was selected to serve as a base to develop engineering designs and costs for a rectilinear, concentric, and vertical type plant. The engineering designs and costs were then used as input to computer codes which calculated plant capital and operating costs for these basic layouts and various modifications. Parametric studies of the economic effects of variables in the flowsheets and layouts were conducted to determine the selections which would yield lowest cost of product water.

1.1.2 Basis

As a guide to flowsheet selection, a parametric study was performed, based upon preliminary cost and design information. The most important assumptions were as follows: (1) fluted tube heat transfer coefficients are computed from the equation presented in the General Electric 50-Mgd design study²; (2) installed tube bundle cost as a function of effect number, as given in Fig. 1; (3) evaporator tube density decreasing with effect number, as shown in Fig. 1; (4) effect cost (including tube bundles) increasing with effect number (due to volume increase), as depicted in Fig. 2; (5) the heat cost of $5.5\phi/\text{MBtu}$ from the ground rules; and (6) preheater and condenser costs of $\$2.80/\text{ft}^2$. The maximum brine design temperature was 260°F and the temperature of the last effect was arbitrarily set at 103.7°F .

1.1.3 Parametric Survey

The range of variables studied was limited because of the preliminary nature of the cost estimates. As a starting point, a thermoeconomic

Fig. 1. Tube Density and Installed Cost of 3-in.-OD Double-Fluted, Aluminum-Brass Tubes with 90-10 Cupro-nickel-clad Tubesheets (Preliminary information based on tentative design and cost data from first-generation studies)

Fig. 2. Structural Cost of VTE Effects (less entrainment separators). Thirteen Effect Evaporator (Case 251A1). Concrete structure with Steel Roof. (Preliminary information based on tentative design and cost data from first-generation studies)

study of an evaporator plant was performed. Included as variable costs were heat transfer area, tube bundle density, and shell volume. The temperature gradient obtained is denoted in Table 1 as the fixed temperature gradient. Perturbations of this case were made using the ORVE-5 computer code⁴ and systematically varying the approach temperatures of the feedheaters.

A similar series, in which the evaporator areas and feed-heater areas for all effects are held constant, is also summarized in Table 1. Each fixed temperature gradient case (feed-heater and evaporator areas are different for different effects) had a slightly lower product cost and a higher performance ratio than the corresponding constant area case. The cost advantage resulted primarily from the lower cost of heat; the cost of heat transfer surfaces was essentially the same or slightly higher for the fixed temperature gradient case.

Table 1 also shows a small cost advantage in using 5- and 10-ft tubes rather than only 10-ft tubes. Although the dual-tube-length-plants appeared to give product costs that are lower by about 0.2¢/1000 gal, later studies showed that 10-ft tubes could be used throughout.

These case studies indicated a fairly flat cost response to the variations in process. For the reference process flowsheet to be used in the Section 1 studies, shown in Fig. 3, 10-ft-long tubes were selected for the high-temperature end of the plant and 5-ft-long tubes for the low temperature end. This follows the General Electric recommendation. Constant evaporator and feed-heater areas were selected to simplify the design, and an 8°F feed-heater TTD was chosen. While this flowsheet may not represent the optimum, it provides a reasonable basic process for which three alternative shell designs can be compared.

More accurate cost data resulting from a study of these shell designs will aid in the description of a better process.

1.2 Preliminary VTE Tube Bundle Design

Following the selection of the reference flowsheet, the next controlling item in the design was the vertical tube evaporator bundle. The total number of tubes per effect specified by the flowsheet and an estimated tube pitch or S/D ratio gave the approximate total tubesheet area per effect. A rough physical layout of the plant was made, using this approximate tubesheet area per effect, and from this layout a bundle depth was specified. Then the tube pitch and tubesheet area were calculated for bundles of the specified depth, condensing temperature, steam loading, and pressure drop. The ORVE-5 computer code⁴ restrained the temperature loss due to pressure drop through the bundle to 0.25°F for effects operating at condensing temperatures above 220°. As the condensing temperature dropped from 220° to 120°, the restraint was raised linearly to 1.50°.

In describing the shape and dimensions of vertical tube bundles the following terms will be used, with the meaning indicated:

Table 1. Cost Study for Fluted Tube VTE Process Flow Diagram Selection^a

Vertical Tube Length (ft)	Feed Heater TTD ^b (°F)	Performance Ratio ^c	Total Feed- Heater and Condenser Area (ft ²)	Total Evaporator Area (ft ²)	Product Cost (cents per 1000 gal)			
					Heat	Feed-Heater and Condenser Area	Evaporator Area	Total
Fixed Temperature Gradients in Feed Heaters and in Evaporators								
5 and 10	6.0	10.8	4.1 x 10 ⁶	4.0 x 10 ⁶	4.18	0.89	1.98	7.06
5 and 10	8.0	10.5	3.6	4.0	4.32	0.77	1.99	7.08
5 and 10	10.0	10.2	3.2	4.0	4.45	0.68	1.99	7.13
5 and 10	12.0	9.7	2.9	4.0	4.58	0.62	1.99	7.20
10	6.0	10.8	4.1	4.3	4.18	0.89	2.20	7.27
10	8.0	10.5	3.6	4.3	4.32	0.77	2.20	7.34
10	10.0	10.2	3.2	4.3	4.45	0.68	2.20	7.34
10	12.0	9.7	2.9	4.3	4.58	0.62	2.21	7.41
Constant Evaporator Areas and Constant Feed Heater Areas								
5 and 10	6.0	10.4	3.1 x 10 ⁶	4.0 x 10 ⁶	4.33	0.76	2.01	7.10
5 and 10	8.0	10.1	2.8	4.0	4.47	0.68	2.02	7.17
5 and 10	10.0	9.8	2.5	4.1	4.61	0.62	2.02	7.26
5 and 10	12.0	9.6	2.4	4.1	4.73	0.59	2.03	7.36
10	6.0	10.7	3.0	4.4	4.37	0.74	2.18	7.29
10	8.0	10.0	2.7	4.4	4.52	0.66	2.18	7.37
10	10.0	9.7	2.5	4.4	4.67	0.61	2.19	7.48
10	12.0	9.4	2.4	4.4	4.80	0.58	2.20	7.58

^aPreliminary information based on tentative design and cost data from first-generation studies.^bTTD = Terminal temperature difference.^cPerformance Ratio = lb water evaporated per 1000 Btu.

Fig. 3. VTE Conceptual Design VTE Process Flowsheet
250-Mgd Capacity; Case No. 251A1

- Length - For vertical tubes or tube bundles, length refers to the height of the active heat transfer surface and is either 10 ft or 5 ft.
- Width - The tube bundle width is the horizontal dimension in the direction of the steam flow through the bundle.
- Depth - The tube bundle depth is the horizontal dimension in the direction of the steam flow through the bundle.
- S/D ratio - The distance between tube centers divided by the tube diameter gives the S/D ratio.
- Pitch - Same as S/D ratio.
- Row - In a rectangular bundle a row is a line of tubes across the bundle width, perpendicular to the direction of vapor flow. In a semicircular bundle, a row is a line of tubes on the circumference of the semicircle, perpendicular to the direction of vapor flow. In the mathematical model used for computer analysis the rows were assumed to be straight and the pie-shaped section of the semicircle was represented by a trapezoid.

Two versions of this code were prepared, one for analyzing rectangular shaped bundles and another for semicircular bundles. In the rectangular bundle code, the length of the tube rows was kept constant and equal to the face width as the row-by-row calculations were performed. To maintain constant vapor velocity within the bundle, the spacing between tubes was decreased so that the number of tubes per row increased until the minimum tube spacing was reached. After this point, the pitch was held at the minimum value of 1.15 and the vapor velocity was allowed to decrease. With the allowable temperature drops through the tube bundles fixed for each effect as previously explained, iterations were performed with this code to determine the tube spacing which will meet this temperature drop requirement for a predetermined bundle depth.

For semicircular tube bundles, a trapezoidal sector was assumed in which the length of each tube row varied from the first row to the last row to follow this shape, but the number of tubes per row was held constant so that the tube spacing decreased from row to row until the minimum tube spacing was attained. With this type of layout, the vapor velocity within the initial rows of the tube bundle actually increased due to the decreasing area available for flow until the minimum tube spacing was reached. In this region of increasing vapor velocity, it was found desirable to limit the maximum vapor velocity for some of the cases in order to prevent this velocity from becoming excessive. For rows in which excessive vapor velocity would otherwise be encountered, the number of tubes and the tube spacing were both allowed to vary so that the vapor velocity could be held at the fixed maximum value. After the tube spacing decreased to the minimum allowable value, the computer program retained the minimum spacing and decreased the number of tubes per row thereafter. The vapor velocity decreased as flow progressed through the bundle. However, due to the tapered shape of the bundle, the decrease in vapor velocity was

not as rapid as in the rectangular bundle discussed above. Here again, iterations were performed to meet pressure drop and bundle depth specifications, as in the rectangular bundle code. The semicircular bundle was then composed of the appropriate number of these sections.

Calculations were performed for tube bundles having 3-in.-diam, GE-type fluted tubes for which the friction factor was assumed to be 1.5 times that of smooth tubes. (Later measurements on actual tube arrays showed the friction factor for fluted tubes to be no different from that for smooth tubes. However, the use of the incorrect friction factor does not affect the validity of the comparison of the three designs.) A summary of the tube spacing and the number of rows for rectangular bundles 10 ft deep for the conditions of effects numbers II, VII, VIII, and XIII is shown in Table 2. A set of results based on a friction factor ratio of 3 is given in Table 3 to indicate the effect of this parameter on tube density. In Table 4, similar results are given for semicircular bundles which have a radius of 12.7 ft and a friction factor ratio of 1.5.

Table 2. Rectangular Tube Bundle Dimensions, Case 251A1

Friction Factor Ratio, $\frac{\text{Rough}}{\text{Smooth}} = 1.5$

	Effect Number			
	II	VII	VIII	XIII
Steam Temperature, °F	258.8	199.9	186.9	120.5
Bundle Depth, ft	10.03	9.98	10.11	9.92
Bundle Height, ft	10.0	10.0	5.0	5.0
Face Velocity, fps	9.06	21.25	25.94	62.50
Velocity, First Row	37.4	65.5	74.9	99.3
Temperature Loss, °F	0.24	0.71	0.90	1.47
Number of Rows	40.0	38.0	38.0	28.0
S/D, First Row	1.32	1.48	1.53	2.70
S/D, Row n	1.15	1.15	1.15	1.15
n	21.0	26.0	26.0	25.0
Number of Tubes/10-ft Face	1346.0	1222.0	1208.0	695.7

Table 3. Rectangular Tube Bundle Dimensions, Case 251A1
Friction Factor Ratio, $\frac{\text{Rough}}{\text{Smooth}} = 3.0$

	Effect Number			
	II	VII	VIII	XIII
Steam Temperature, °F	258.8	199.9	186.9	120.5
Bundle Depth, ft	9.97	9.99	9.96	9.93
Bundle Height, ft	10.0	10.0	5.0	5.0
Face Velocity, fps	8.13	18.75	22.66	52.34
Velocity, First Row	25.0	46.3	52.1	78.5
Temperature Loss, °F	0.23	0.70	0.88	1.44
Number of Rows	38.0	36.0	35.0	26.0
S/D, First Row	1.48	1.68	1.77	3.00
S/D, Row n	1.15	1.15	1.15	1.43
n	25.0	27.0	27.0	18.0
Number of Tubes/10-ft Face	1224.0	1101.0	1047.0	588.4

Table 4. Semicircular Tube Bundle Dimensions, Case 251A1
Friction Factor Ratio, $\frac{\text{Rough}}{\text{Smooth}} = 1.5$

	Effect Number			
	II	VII	VIII	XIII
Steam Temperature, °F	258.8	199.9	186.9	120.5
Bundle Outer Radius, ft	12.66	12.77	12.76	12.78
Bundle Inner Radius, ft	1.05	3.54	3.43	3.52
Bundle Height, ft	10.0	10.0	5.0	5.0
Face Velocity, fps	5.76	13.54	16.61	44.73
Velocity, First Row	36.1	52.6	64.1	104.4
Temperature Loss, °F	0.25	0.71	0.89	1.45
Number of Rows	45.0	36.0	36.0	29.0
S/D, First Row	11.19	1.30	1.35	1.75
S/D, Row n	1.15	1.15	1.15	1.40
n	3.0	14.0	18.0	9.0
Number of Tubes/Semicircle	3337.4	3111.1	3028.9	1997.2

The tube densities for the rectangular and semicircular shapes were about the same, except in effect number XIII. Here the semicircular shape had a tube density 12% more than the tube density for the rectangular shape. In all cases, the exit velocity was greater for the semicircular bundle than for the rectangular bundle, and from the standpoint of noncondensable removal, this was a definite advantage with the semicircular shape. These bundle shapes were each incorporated in rectilinear plant designs. The overall plant design will influence bundle selection. This study showed that the shape of the vertical tube bundle is not an important factor in determining plant performance. The decision as to tube bundle shape in the final design can be made on the basis of layout, structural, or other considerations with assurance that the decision will not impose a significant penalty in plant performance.

The concentric ring plant required a different bundle geometry and a slightly different approach to the bundle design. As in the case of the rectilinear plant the evaporator bundles were considered to be the controlling component and sizing the plant depended upon a calculation of the size of the bundles. A trail S/D ratio was assumed and a total tube sheet area estimated for each effect. Then a trail plant layout was made

Table 5. Tube Bundles with 3-in.-diam Fluted Tubes
for Concentric Plant, Case 261-C

	Effect Number			
	II	VII	VIII	XIII
Steam Temperature, °F	259.5	203.5	191.5	123.6
$\bar{U}$, Btu/hr.ft ² .°F	1909.0	1583.0	1491.0	844.0
Bundle Depth, ft	10.03	4.52	4.02	2.88
Bundle Height, ft	10.0	10.0	10.0	10.0
Face Velocity, fps	8.95	11.2	12.5	32.0
Velocity - 1st Row, fps	29.3	86.2	95.8	133.0
Temperature Loss, °F	0.24	0.53	0.60	1.47
Steam to Exit, %	17.3	20.5	23.8	19.1
Number of Rows	37.0	18.0	16.0	11.0
S/D, 1st Row	1.44	1.15	1.15	1.32
S/D, row n	1.15	1.15	1.15	1.15
n	27.0	1.0	1.0	8.0
Tube Density, number of tubes/ft ²	11.87	13.85	13.84	12.75

to obtain an estimated bundle width (circumference) and depth (thickness). The concentric ring layout naturally leads, in most effects, to very wide shallow bundles whose rear faces constitute stagnant zones. This necessitated one departure from the assumptions on which the bundle pressure drop calculations for the rectilinear plant were made. That is, the noncondensable sweep stream exiting from the rear face of the bundle was increased from about 4% to about 20% of the entering stream. This constitutes the entire supply for the feed heater bundles. Table 5 shows the results of the pressure drop calculations. In the design of a concentric ring plant, the structure and flow pattern were found to be greatly complicated when both 5- and 10-ft-tube lengths are used. It was believed that the use of 10-ft tubes through-out the plant at a lower cost per square foot, with less tube-sheet area, and a simpler design would more than compensate for the larger surface area required due to the lower heat transfer coefficients of the 10-ft tubes. Therefore, the flowsheet shown in Fig. 4 (case 261-C) was developed. This is identical to the previous flowsheet except for the tube lengths which are 10-ft in all effects. (This later proved to be a more economical design for the rectilinear as well as the concentric ring structure.)

1.3 Rectilinear Plant Design

This exploratory shell design and evaluation was the first of comparative assessments of the rectilinear, concentric, and vertical (silo-type) structures. In order to accomplish the comparison with a minimum amount of effort, it was decided to make a layout of the evaporator and to develop costs on only four effects. The flowsheet developed in Section 1.1 would be used throughout. This flowsheet (Fig.3) has 10- and 5-ft-long tubes with equal surface areas in each effect. Effects 2, 7, 8, and 13 were selected for developing cost information since they represented extremes of vapor density for the 10- and 5-ft tubes. With the developed costs, curves could be plotted and unit costs could be used to (1) determine with a fair amount of accuracy the total evaporator cost and (2) provide information input to the computer program for future optimization work.

1.3.1 Preliminary Studies

Layout sketches were made to seek the best solution to the problems of component placement and materials handling. The component placement problem involved the disposition of evaporator bundles, feed heaters, pumps and associated piping, entrainment separators, and noncondensable removal systems. The materials handling problem involved the transport of large quantities of brine, vapor, noncondensables, and product withing stipulated ranges of velocity, depth, and pressure drop.

Layout A. In this layout the flashing brine and product streams were located on the floor of the plant and were specified as having a constant width to accommodate a stream flow of 500,000 lb/hr.ft width. (Fig. 5)

The vertical tube bundles consisted of 10-ft-long, 3-in.-OD fluted tubes in the high temperature effects and 5-ft-long, 3-in.-OD fluted tubes in the

Fig. 4. VTE Conceptual Design VTE Process Flowsheet
250-Mgd Capacity; Case No. 261-C

Fig. 5. Layout "A"; 250-Mgd VTE Plant

low temperature effects. The amount of tube surface area per effect was held constant and the bundle depth (in the direction of vapor flow) was set at 10-ft.

To maintain a constant tube surface area per effect and a constant bundle depth, as decided upon, it was necessary to accommodate changes in tube pitch by varying the tube bundle width. Each effect was designed with eight tube bundles arranged so that vapor flowed into the bundles at right angles to the main brine stream. All bundles in a single effect had the same tube spacing. The variables from effect to effect were the tube spacing and the bundle width.

The vertical tube bundles were arranged in pairs, one on each side of the open vapor passages so that each effect had four pairs of bundles across the full width of the plant. The vapor passage between the bundles of each pair was designed with a horizontal entrainment separator, York Style 421, through which vapor flows upward into an open vapor space. Vapor enters the bundles on either side of the vapor space and passes through the vertical tube bundles, after which the remaining vapor is condensed on the feed heater tube bundles at the back of the vertical tube bundles.

Vapor velocities in the tube bundles were calculated by the computer codes, as described in Section 1.2.1. Vapor velocities through the entrainment separators were computed and found to be within the acceptable operating range indicated by the demister performance study reported by ORNL.²

In a typical effect, steam from the previous effect flows through an opening in the effect wall and is distributed under the entrainment separator by a duct which keeps it separated from steam being generated in the effect under consideration. About 4% of the steam entering the vertical tube bundles exits from its rear face to sweep the noncondensable gases into the feed heater bundles. The feed heater bundles condense the sweep steam from the gas along with the steam from the flashing product stream below. The noncondensable gases are removed at points within the feed heater bundles.

Brine feed for the vertical tubes is picked up from the brine stream on the evaporator floor and pumped to a brine chest over each vertical tube bundle.

It was readily apparent that horizontal demisters take excessive space and make the plant larger than it needs to be. Therefore Layout A was not studied in detail.

Layout B. Layout A was modified to use vertical entrainment separators installed along the face of the bundles. (Fig. 6). This enabled narrowing the vapor space between the rows of vertical tube bundles. Since the plant width was the same, due to flashing requirements, the space gained could be utilized by installing additional bundles. Each effect in this layout consisted of 6 pairs of vertical tube bundles rather than 4, as in Layout A. Further study of this layout suggested that the location of the feed heater

Fig. 6. Layout "B"; 250-Mgd VTE Plant

tube bundles beside the vertical tube bundles with separate product water trays below them made the plant larger, more complex, and more costly than necessary.

Layout C. This layout differs from layout B in that the feed heater tube bundles were moved below the vertical tube bundles, permitting the space between the rear faces of adjacent bundles to be reduced to that required for exit vapor flow. (Figs. 7 and 8) This allowed space for two more bundles across the plant width, making a total of fourteen bundles per effect. By adding more bundles, the plant was again shortened.

A detailed study of vapor flow requirements showed that in order to best utilize available space for vapor flow, elaborate separating walls were required to separate vapor being generated in an effect from vapor which is to be condensed in that effect. These walls, which must withstand the intereffect pressure difference and do not run parallel to this axis of the plant, would be costly to build. It appeared that a simpler and less costly arrangement could be devised.

Layout D. The arrangement of the vertical tube bundles in this layout differs from the previous layouts discussed in that the plant width is not held constant and the vapor flow through the bundles is parallel to the direction of the main brine flow on the floor below. (Fig. 9) The bundle depth (in the direction of vapor flow) is 10 ft and the total effect depth is only that required for the tube bundle plus a 10-ft vapor flow and disengagement space between bundles. The width of the bundle (now in the direction of plant width) varies to accommodate changes in tube spacing.

Vapor exits from the lower end of the vertical tubes immediately over the flashing brine stream, turns and flows from under the bundle and upward and turns into the face of the bundle for the next effect. Vapor velocities are much lower than in layout C, being within the range of 10 to 60 ft/sec. Vapor flow passages are thus not only shorter and more direct, but vapor pressure drop is less than for layout C. No elaborate vapor ducts are required for vapor flow.

To minimize the plant width and to allow for vapor flow in the transition area between the last 10-ft-tube effect and the first 5-ft-tube effect, the 5-ft-tube bundles are arranged in two levels with vapor flowing to both levels from the same vapor flow space in front of the bundles.

The vertical type entrainment separators are located at the face of the bundles, and the superficial vapor velocity through the separators can again be as low as the face velocity on the tube bundle.

The increase in tube spacing as vapor density decreased results in a plant which becomes larger toward the lower temperature end. On the basis of previous experience with pumps and other equipment associated with plants of this size and with structural problems in using common wall construction, it was decided to divide the plant into four identical trapezoidal trains with the feed heater bundles in the center.

Fig. 7. Layout "C"; 250-Mgd VTE Plant

Fig. 8. Layout "C"; 250-Mgd VTE Plant; Vapor Duct
at Effect X

Fig. 9. Layout "D"; 250-Mgd VTE Plant

A duct at the rear face of each vertical tube bundle carries uncondensed water vapor and noncondensables across to the feed heater bundles, where condensing is completed and gases are vented.

Product water is collected at the rear of the vertical tube bundles and ducted to the evaporator floor where it is collected under the feed heater bundles. The product water stream flashes from effect to effect and widens as flow increases, maintaining a constant flow rate per foot of width. A roof is provided over the product water stream to prevent contamination from falling brine.

Entrainment separators are located at the edge of the product water tray to prevent brine being carried in by vapor from the flashing brine stream.

Choice of Layout. The choice of a basic layout for the rectilinear plant became a choice between layout C, which evolved from layouts A and B, and layout D. In order to make the choice, advantages and disadvantages of each layout were considered.

Layout C.

- Advantages:
1. Constant plant width.
 2. Readily adaptable to either horizontal or vertical type entrainment separators.
 3. True rectangular structure.
- Disadvantages:
1. Feed heater-tube lengths in effects are unequal.
 2. Not as well suited to modular construction.
 3. More complicated structurally due to elaborate internal baffles required.
 4. Vapor flow paths are longer and vapor velocities higher.

Layout D.

- Advantages:
1. No elaborate internal baffles are required. Structure is, therefore, simpler.
 2. Vapor flow is shorter, more direct; vapor velocities are lower.
 3. Five-foot tube bundles can be stacked more easily.
 4. Better suited to central location of feed-heater bundles in module.
 5. Feed-heater tube lengths are same in all effects.
- Disadvantages:
1. Horizontal entrainment separators would be more difficult to use in the two-level, five-foot-tube effects.
 2. A small amount of waste space results from the nonlinear variation of tube bundle width.
 3. Buildings are trapezoidal rather than rectangular.

Advantages 1 and 3 of layout C are not too important. Feed-water and vapor flow considerations favor layout D. Waste space due to vertical tube bundle requirements amounts to only about 2% of the total volume of the plant.

Preliminary information indicates that vertical entrainment separators can be used and may actually be more economical. More information on vertical entrainment separators is being obtained. Horizontal separators could be used in layout D, but building volume would have to increase to accommodate baffles to prevent flooding.

Layout C requires walls within each effect to separate vapor being generated within the effect from vapor coming in from the previous effect. These walls must withstand the intereffect pressure differences and do not run parallel to the longitudinal axis of the plant. The separating walls would, therefore, be costly to build.

From these considerations, layout D was chosen as the basis for the rectilinear plant design.

1.3.2 Plant with Rectangular Tube Bundles

Plant Design. Layout D, previously described, was used with the flowsheet case 251A1 (Fig. 3) as the basis for plant design. The flashing brine and product water trays were located on the floor level, with the vertical tube bundles above, arranged so that the flow of vapor into the bundles is parallel to the brine and product water streams. The tube bundles were laid out in accordance with data from the computer program for tube-bundle vapor-pressure loss.

The plant width was determined from the tube-bundle width required, with allowance made for structural requirements. After the width was determined, brine and product water flow rates were calculated. In the first effect, the flow rate was found to be 650,000 lb/hr.ft. Extrapolation of the Westgarth equation⁶ to this flow indicated that a chamber length of 7.44 ft is required for equilibration. Since the effect length, as determined by tube-bundle and vapor-flow space requirements, is 22 ft, there is evidently an adequate safety factor for the extrapolation to 650,000 lb/hr.ft.

The variation of tube spacing with effect vapor density resulted in a plant which increased in width from the high temperature end. (Fig. 10) In order to achieve a compact design, the vertical tube bundles in the effects with 5-ft tubes were stacked in two levels. (Fig. 11). The design provided that brine falling from the tubes in the upper level would drop onto the top of the lower-level brine chest and then be channeled to a central drop point to prevent interference with vapor flowing from the lower-level tubes. The brine falls directly from the central drop point to the brine stream on the evaporator floor.

The plant was divided into four trains with the feed-heater bundles centered in the trains. Each effect in a train, therefore, contained

Fig. 10. Rectilinear Plant with Rectangular Bundles
Site Plan

Fig. 11. Rectilinear Plant with Rectangular Bundles
Cross Sections

two discrete vertical-tube bundles in the 10-ft tube effects and four bundles in the 5-ft tube effects.

The feed-heater tubes were arranged in a vertical bank of three bundles, with condensate baffles below the upper bundles to prevent flooding of the lowest bundles. (Fig. 12) Product water would be collected on the evaporator floor under the feed-heater bundles and would flow from effect to effect. As product water accumulates and the flow rate tends to increase, the product stream would be widened into the area under the vertical tubes to maintain proper product-water flow rate. To prevent mixing of the brine and product, a cover was provided over the product-water stream in such a way that the cover will not interfere with vapor flow from the vertical tubes or with flashing of the product stream. The product stream and feed-heater tubes were thus enclosed in a chamber which was separated from the rest of the process. Vapor entered through vertical entrainment separators located between the effect bundle and the inter-effect wall. This is shown in Fig. 12.

Condensate from the vertical-tube bundles would drain to the rear of the bundle and collect in a trough which runs lengthwise along the rear face. (Fig. 12). Each trough has a drain pipe which carries the product water down to the floor level.

Noncondensable gases pass through the bundle and exit at the rear face along with 4% of the water vapor which entered. The noncondensables and water vapor were then ducted along the rear of the bundle to the center of the module where they flow into the feed heater bundles. Noncondensable gases would be withdrawn from the feed-heater bundles through appropriately placed vent pipes.

Since the product-water tray was located in the center of the module, the flashing brine was divided into two separate streams. A pickup point was provided in each stream for the brine feed pump. Due to structural arrangements for thermal expansion, these pickup points were fixed in location, and an expansion joint was provided at each point to take up lateral expansion of the pipe.

The modules were set up in a paired arrangement with an open space between the module pairs. This allowed the brine feed pumps for each module to be located outside and adjacent to the module. The brine was then pumped into the end of the brine chest, which runs the full length of the tube bundle. Entering brine velocity in the brine chest was computed to be about 2-ft/sec. Since the velocity will decrease continuously, brine pressure drop along the length of the brine chest should not affect distribution to the tubes.

The brine feed pumps are of the mixed flow type and have two stages to develop sufficient head. To provide a NPSH of 18 ft, the pumps were submerged in a tank with brine feed at the top. The pumps would be driven by electric motors mounted above the tank. The pumps are rated at 15,000 gpm and 7500 gpm at a head of 40 ft.

Structural Design. The evaporator structure was designed of reinforced, cast-in-place concrete, even where lining it with thin steel sheet is required to contain pressure, provides economy in design. The steel sheet and the small channels which tie the sheet to the concrete were considered as part of the reinforcing required in the shell. Composite action of the concrete and the liner tends to keep the concrete thickness to a minimum. The liner is also a form for pouring wall sections and reduces the amount of shoring required in construction. Structural steel usage was limited to the arched tops, which provide access for installation and removal of tube bundles.

The fan shape of the evaporator was dictated by the required increase in tube sheet area from the high pressure to the low pressure ends. Stage lengths (22 ft) and column spacing (11 ft) in the longitudinal direction remain constant. From effect I at the high pressure end of the plant through effect IX, ten interior column rows are required in each module, which means that interior column spacing will vary from approximately 6-ft OC to 9-ft OC in the transverse direction. From effect X through effect XIII intermediate columns are inserted (making 20 column rows).

Longitudinal expansion joints were located under the feed-heater bundles and at the separation walls between modules. Sixteen-gauge Hastelloy strips were used to seal the outer pressure shell joints.

Brine return pits on each side of the feed-heater bundles act as anchor points to control the direction of expansion. The fan shape of the evaporator is conducive to this expansion joint system, as the distance between joints increases as the internal temperature decreases, and the differential temperature between ambient conditions and internal plant conditions becomes less.

Transverse expansion joints occur at each stage in the tube bundle section. The arch roof will adjust automatically without joints, while sliding seals at the ends and one side of the tube sheets permit movement. The concrete bottom and end walls have neoprene fillers with polypropylene hinge seals at each joint.

Barriers between the product water trays and the brine are made of polypropylene and polypropylene-covered steel struts.

Polypropylene is also used extensively as protective liner for both steel and concrete wherever there will be contact with brine, product water, or splash from either of these. Surfaces exposed to vapor only are coated with a phenolic coating.

1.3.3 Plant with Semicircular Tube Bundles

Plant Design. This plant was designed primarily to study the economic effect of using the semicircular tube bundles described under Bundle Design. Another design feature included in the study was a continuous floor beneath the vertical tube bundle region separating it from the flashing brine stream below. (Fig. 13) This separation was not required

Fig. 13. 250-Mgd Rectilinear Plant; Semicircular Bundles Plan & Sections

for the use of semicircular bundles but was included to develop cost information for the planned study of multistage feed heating.

The determination of the number of tubes per bundle and of the bundle diameter was discussed previously. Full semicircles or quarter circles were used rather than the odd angles between 180 and 0 degrees, with adjustments of number of rows per bundle to provide the correct number of tubes per effect.

Four feed-heater tube bundles per module were run below the floor which divides the vertical bundles from the flashing brine. The decision to use four smaller bundles was based on considerations of plant geometry. The product-water stream runs beneath the feed bundles, increasing in width toward the low pressure end of the plant as the product flow increases. Product water from the vertical tube bundles is piped from the center space of the semicircular bundles to pits in the bottom of the product tray. Noncondensables and water vapor which pass through the vertical bundles are collected in the center space and piped to the feed-heater tube bundles containing the noncondensable removal system.

Vapor flashing from the brine stream on the floor of the evaporator passes through vertical entrainment separators along each side of the product tray where it is condensed on the feed-heater tubes.

Brine is pumped from two pickup points in the evaporator floor to the water-boxes. There is one pump for each 10-ft-tube effect module and two pumps for each 5-ft-tube effect module. The pumps are essentially identical to those in the plant with rectangular tube bundles.

Structural Design. The semicircular tube bundle evaporator structure is similar in construction to the rectangular bundle evaporator with a few minor exceptions. The plant area of each evaporator module is approximately the same, although the semicircular tube bundle evaporator is 4 ft higher in the 10-ft-tube section, and 5-1/2-ft higher in the 5-ft-tube section. Effect lengths remain 22 ft, but since the radius of the semicircular tube sheet is 13 ft, a three-arch roof structure (two spans of 7 ft and one of 8 ft) is used to keep the arch steel thickness to a minimum. A beam, supported at the ends and by 3-in. round bar tie columns at the third points, spans the semicircular bundle and provides edge support for one side of the pair of 7-ft arches. The 3-in. tie columns, where required, replace tubes in the bundle. A similar mid-beam is required in the lower water box for the 5-ft-long tube section.

Use of an additional floor slab for brine collection replaces the polypropylene baffles at the product water trays of the rectangular bundle plant. Expansion joints are similar to those of the rectangular plant in detail. However, the longitudinal joints in the semicircular bundle evaporator will occur only at the longitudinal walls, and the transverse joints are placed in the vapor distribution section of each effect instead of under tube bundles.

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If the outside walls of this evaporator had been straight as in the rectangular tube bundle evaporator, the crane-way and crane for handling the pumps in the middle pump canyon would be much less costly. The irregular shape used provides overall plant economical design, but the crane span is much longer and also requires the addition of special crane columns where the pump canyon becomes extremely wide.

1.4 Concentric Ring Plant Design

The basic layout of the concentric ring plant arranges the vertical tube bundles and vapor space of the effects in concentric rings with the highest temperature effect at the center where it requires a relatively small pressure vessel and the lowest temperature effect at the outer ring where the low density steam has the greatest flow area. To be comparable with the other designs, this plant is also divided into four trains each comprising one quadrant of the circle. The placement of the other major components - feed heater, bundles, pumps, pipes - and the transport of the materials - brine, vapor, product - had to be fitted to this configuration.

1.4.1 Preliminary Studies

Three feed heater bundle arrangements were examined (1) continuous tubes from the outside to the center, (2) individual vertical bundles in each effect, and (3) individual horizontal bundles in each effect.

The problem with the continuous tube was the mismatch between the tube length, required for feed heater surface area, and the available effect length. Two solutions were proposed. The first was to run the tubes in on a spiral such that the appropriate feed heater tube length was maintained regardless of the radial dimension of the effect. (Fig. 14) The tube sheets would be mounted at right angles to the effect wall rather than parallel to it. Such an arrangement complicated the placement of the product stream, which would normally be located below the feed heater bundles, and there is some question concerning the ability to economically install continuous tubes in a curve. The second solution was to run the feed-heater tubes radially and make the radial dimension of the effects equal to the required feed heater length. This arrangement, of course, resulted in increased building volume.

A vertical U-tube bundle with the inlet and outlet water boxes located at the top was investigated. It has the advantage of relative ease of replacement; however it did not fit well into the vapor space and did require that the feed heating vapor, which exits from the rear face of the vertical tube bundle, travel a relatively long distance.

The horizontal bundles were shown as circumferential segments located in the vapor space between effect brine chests. (Fig. 15) These occupy a relatively high percentage of the available annuli so that the majority of the steam leaving the rear face of the vertical tube bundles can travel directly up to the feed heater bundles. The condensate can run down the

Fig. 14. 250-Mgd Concentric Ring Plant Continuous Feed Heaters; One Quadrant Plan

Fig. 15. 250-Mgd Concentric Ring Plant; Circumferential Feed Heaters; One Quadrant Plan

sloping intereffect partition and join the condensate from the vertical bundles.

A more detailed analysis would be required to establish if one of the feed-heater proposals was superior to the others. However, since continuous bundles were estimated in the rectilinear plant, it was decided to estimate individual bundles in this plant to obtain the cost information. Of the various types of separate bundles, the horizontal circumferential segments were chosen because they were more easily accommodated in the design.

Three entrainment separating methods were considered - gravity, horizontal mesh, and vertical plate - and sketches of the three alternatives were prepared. The vapor disengagement spaces for gravity separation were sized according to data from the Freeport demonstration plant. The horizontal-mesh separators were sized in accordance with the York formulas. The vertical plate separators were designed to occupy the entry face of the vertical bundles. A comparison of the cost of the shell volume for gravity separation, the mesh separator, and the vertical-plate separator led to the selection of the vertical plate separator.

1.4.2 Plant Design

The design used for estimating costs, shown in Figs 15 and 16, is a cylindrical shell 360 ft in diameter by about 23 ft high and divided into four quadrants each capable of operating as an independent train. The vertical tube bundles range in depth from 15 ft to 2-1/2 ft. The vapor emerging from the evaporator bundles and from the flashing brine arises and enters the entrainment separator of the succeeding effect, drives through the evaporator bundle where about 80% of it is condensed, and the remaining 20% with the noncondensable load exits the rear face of the bundle and rises to the feed heater bundles where the remainder of the steam is condensed and the noncondensables are removed. The brine from each effect drains to two brine trays which have intereffect orifices for flashing the brine downstream. The product from the evaporator and feed heater bundles of each effect drains to two product trays similarly equipped for flashing the product downstream.

The effect feed pumps for each quadrant are located in a radial tunnel centered below the quadrant floor. The pumps are fed from the two brine trays and deliver to the brine chest.

1.4.3 Structural Design

Construction is of structural steel and concrete. All ring walls, both exterior and between effects, columns, slabs, and beams are of conventionally reinforced concrete. The floor and pump trench are also reinforced concrete.

The roof is insulated arches of steel plate which varies in thickness as determined from the ASME Pressure Vessel Code for the spans and the start-up and/or operating pressures involved. A 1/8-in. corrosion

Fig. 16. 250-Mgd Concentric Ring Plant; Circumferential Feed Heaters Sections

allowance is used in the roof design. Minimum plate thickness throughout the evaporator is 1/4-in.

All concrete exterior walls, walls between modules, and the floor slab are lined with 1/4-in. plate for a gas pressure seal. In addition, all surfaces, both steel and concrete, where exposed to brine or product-water immersion or splash are protected with 1/8-in. polypropylene sheet. Surfaces which are exposed only to vapor have a phenolic coating.

A radial pump trench is provided under each module (four required). Each trench is 16-ft wide and extends 19 ft below the evaporator floor level. A five-ton underhung crane in each trench will handle pump components for maintenance.

1.5 Vertical Plant Design

In the falling film, vertical structure plant, the effects are placed one above the other, the highest temperature effect being at the top. Brine then feeds from effect to effect by gravity and the need for brine feed pumps at each effect is eliminated. In the overall picture the pumping cost is not eliminated, however, as the makeup water pump must supply the head required to pump to the top of the plant.

1.5.1 Preliminary Studies

Three basically different evaporator arrangements were considered. In the first, shown on Figs. 17 and 18, the effect bundles are essentially the same as those of the rectilinear plant with rectangular bundles. The building thus has a rectangular base and is stepped vertically to fit the smaller areas of the upper effects. A brine tank is located over the top of each bundle to collect brine and feed it to the vertical tubes. Since only a portion of the brine flowing into an effect flows through the vertical tubes, bypass lines are provided to carry brine from the tank to the next effect. As vapor and brine discharge from the tubes of a typical effect, the brine falls into the brine tank for the next effect while the vapor moves horizontally out from under the tube bundle, makes a 90° turn, flows downward and then turns back to flow into the face of the vertical tube bundle. Vertical entrainment separators are located at the face of the tube bundles. Noncondensable gases are swept through the vertical tube bundle and carried into the feed heater bundles where they are cooled and removed. As with the previous designs, the plant is divided into four modules. A typical effect in a module contains two vertical tube bundles with the feed heater tube bundles located between. Vapor flows into the two bundles from opposite directions with the non-condensables and remaining water vapor from the two bundles converging on the feed heater bundles.

In the early stages of the study, the decision between 5-ft-long and 10-ft-long vertical tubes had not been made; for this reason, two versions of the rectangular layout were drawn, one with 10-ft tubes (Fig. 17) and one with 5-ft tubes (Fig. 18). A bill of material was prepared for items which would differ in cost between the 5-ft and 10-ft tubed plants. With

Fig. 17. 250-Mgd VTE Vertical Structure; Layout "A";
10-ft Tubes

Fig. 18. [250-Mgd VTE Vertical Structure; Layout "A";
5-ft Tubes

operating cost for the makeup pump considered, the 5-ft tube version proved to be more economical. The plant is made lower but wider by using 5-ft tubes.

The second basic layout was a vertical cylinder with each of the four modules occupying one quadrant of the cylinder. The feed heater tube bundles are located at the inner corner of each quadrant. The vertical tube bundles are arc shaped; vapor flows from the outer periphery and velocity is maintained by the converging flow path. A drawing of this scheme was not prepared; since serious thermal stress problems were foreseen in operating the plant with one or more modules shut down, we decided not to pursue the design.

The third basic arrangement considered was to put each of the four modules in a separate vertical cylinder. The vertical tubes are arranged in annular bundles with the feed heater tubes located at the center. Vapor flows inward from the periphery of the bundle with noncondensibles swept into the feed heater tubes at the center. Layouts of this evaporator with 5-ft and 10-ft tubes are shown in Figs. 19 and 20.

The layouts of the two arrangements under consideration were studied and the design having separate cylindrical shells for each module was chosen.

Another idea considered was putting the evaporator in two separate structures, the higher temperature effects being in one building and the lower temperature effects in the other. The purpose of this arrangement would be to avoid high structures. The extra piping required for makeup water connections and the vapor duct required to interconnect the two evaporators would tend to offset any advantages which could be realized. For this reason, this idea was not pursued to the point of making a cost comparison.

1.5.2 Plant Design

The design chosen for cost estimating, shown in Fig. 20, consists of four independent modules each housed in a vertical cylindrical shell. The basic shell structure is 212-ft 8-in. high and has a maximum diameter of 70 ft. The GE-type fluted tubes are arranged in annular bundles with feed heater tubes located at the center. Tube density in the bundle is based on data for the rectangular bundle used in the rectilinear plant. This is an approximation which should give a basis for refined tube bundle calculations.

Brine and vapor flow are essentially as described previously for the rectangular bundle layout. Vapor flows horizontally outward from the area under the tube bundles. Maximum velocity of vapor as it passes from under the tube bundle ranges from 23 fps in Effect II to 145 fps in Effect XIII. A space is provided outside the face of the vertical tube bundle for collection of carried-over brine and runoff from the entrainment separators located at the face of the bundle. This brine is then allowed to drain into the brine collection tank of the next effect.

Fig. 20. 250-Mgd VTE Vertical Structure Cylindrical Modules Plan & Elevation

The flow of product water and brine from effect to effect is controlled by float valves. The float valve, which is based on a design presented by Dow Chemical Company in their 50-Mgd conceptual design report, is detailed in Fig. 20. A nozzle is provided at the end of each bypass and drain line to prevent flashing in the line. Product water is collected in the area between the vertical tubes and feed heater bundles.

A nozzle at the top of each of the fluted tubes requires 5-psia drop to assure proper brine distribution. The 5 psia is provided by intereffect pressure differential plus brine depth in the tank over the tubes. This brine depth requirement, which adds to the building height, is given for each effect in Fig. 20.

Turbine exhaust steam is supplied through an 18-ft-diam duct which runs upward between the four modules. Each module then connects to the main duct and can be valved off.

1.5.3 Structural Design

The vertical VTE structure, or silo type evaporator, consists of 13 effects and a condenser stacked vertically. This structure with the falling film evaporation system requires a high-head pump to raise the brine to the top and then utilizes gravity for process flow downward through the effects.

Three shapes of structures plus two lengths of vertical tube bundles were considered. Four modules, or independently operating units, are required for the plant, each with a capacity of 62.5 Mgd.

A single large cylindrical structure, divided by radial walls into four equal pie-shaped units was considered. A preliminary study eliminated this shape rather quickly. One or more of the modules may be in a non-operating condition at any time and the stresses across the module separating walls, plus thermal stresses and expansion in a tapering shape, make this a structure which is expensive and unnecessarily complicated. Even the exterior walls of the cylinder, which are economical when acting under uniform pressure, lose all their shape advantages when one or more modules are nonoperable.

A rectangular structure was then considered. One longitudinal and one transverse common wall separated this evaporator into four modules. This design also presents an expansion and thermal stress problem when one or more modules are not operating, but complications are less severe than in the first study since each module is rectangular in plan. However, both the separation walls and the exterior walls are extremely thick since they span between horizontal beam and strut systems which occur at the bottom of each effect.

The pumps required to raise the brine to the top of the evaporator are important in a vertical type plant, both from initial cost and operating cost. To determine the relative importance of plant height on evaporator costs a comparison was developed between evaporators containing 10-ft and 5-ft vertical tubes. The rectangular shaped plant was used in this

comparison as being typical, although perhaps not the most economical shape for a vertical structure.

The following comparison is based on cost of units varying because of tube bundle length change only, and does not represent total evaporator cost:

Evaporator with 10-ft tube bundles

Structure	\$ 3,803,669
Mechanical and Piping	16,706,400
Total	<u>\$20,510,069</u>

Evaporator with 5-ft tube bundles

Structure	\$ 4,078,642
Mechanical and Piping	16,790,400
Total	<u>\$20,869,042</u>

Since the cost differential between pumping the brine in evaporator structures using 10-ft instead of 5-ft tube bundles is approximately \$114,000 per year, the shorter tube bundle evaporator proved to be the most economical. The third shape considered and later developed as the best of the vertical structure designs, consisted of four separate cylindrical silos using 5-ft tube bundles.

The cylindrical shape provides many advantages in pressure vessel design. In this evaporator the vertical walls of the top effects, which have positive internal pressures, are designed of prestressed concrete. The cylindrical shape plus prestressing considerably reduces the thickness of these walls when compared with the rectangular plant. The lower-effect walls, in the positive external pressure area, are conventionally reinforced concrete, but again will have their comparative thickness reduced because of arch action.

To keep the exterior wall acting as a true ring, the internal structure and the roof are constructed independently of this wall, with flexible seals at joints as required for pressure tightness. Differential expansion problems of the internal and external structures are eliminated. No expansion joints are provided in the external wall since the structure is free to expand vertically and the internal effect temperature differences are so small that differential ring expansion is of minor importance. Internal stress in the wall caused by large temperature differences between the interior and exterior surfaces has been considered and will present a minor problem.

The exterior concrete wall and ground floor slab are lined with 1/4-in. carbon steel plate for pressure tightness. The top is constructed of arched steel plate which combines economy of material with ease of removal if required for maintenance. This top plate is insulated to limit heat loss.

Internal construction is of concrete, conventionally reinforced, with the concrete structure kept as small and/or as thin as possible to limit internal temperature stresses during startup or shutdown of the evaporator. The bottom footing is the combined type, supporting both the exterior wall and the interior columns. This footing, or pile cap, is supported on 50-ton capacity, steel shell, concrete-filled tapered piles 45-ft long.

For corrosion control, all surfaces in contact with brine or product water, liquid or splash, are covered with 1/8-in. polypropylene sheet. Surfaces exposed to vapor only are protected with a phenolic coating.

This evaporator structure was presented to a specialist in earthquake engineering for comment. He felt the structure, as designed, would be considered to have very high earthquake resistance. An earthquake analysis would be required on any final design which possibly could modify some minor details but he foresees no change which would greatly affect costs.

1.6 Cost Analyses of the Structural Variations

1.6.1 Summary

Although the original intent of the exploratory shell design and evaluation program was to examine only the rectilinear plant, the concentric ring plant, and the vertical plant other important variations suggested themselves during the course of the investigations. There were, therefore, as described in Sections 1.3, 1.4, and 1.5, partial designs made and costed to examine the economics of tube length, bundle configuration, and the structural costs associated with multistage feed heating. The results of these investigations are shown in Table 6.

Table 7 is a rough comparison of the VTE with the MSF Conceptual Design reference plant which was made at this time to determine the incentive to continue the pursuit of the VTE conceptual design program.

1.6.2 Basis of Cost Information and Method of Analysis

Cost estimates were developed by preparing detailed bills of materials from the design drawings for each plant. In order to develop realistic figures, quotations were obtained from three vendors, where possible, on all major items of equipment.

With the exception of tubing, all material costs represent 1965 prices. Feed heater tubing costs were projected at 1970 prices based on the use of an on-site tube mill and no inflation of 1965 dollars. Estimates of tubing costs do not reflect price increases announced by the manufacturers in November 1965. The 10.5% increase in the basic cost for 70-30 copper-nickel alloy tubing would add 18¢/ft² to the costs included in the estimates for tubing.

Table 6. 250-Mgd Vertical Tube Evaporator - Structure Analysis
Comparison of Cost Information for Various Designs
(3-in.-OD-diam; Type: Fluted)

	Type Structure					
	Rectilinear				Concentric Ring	Vertical
Tube Conditions:)						
Case number	251A1	251A1	251A1	261C	261C	261-C7
Length, ft	10 & 5	10 & 5	10 & 5	10	10	5
Type bundle	Rectangular	Semicircular	Semicircular	Rectangular	Rectangular, & Circumferential	Rectangular, & Circumferential
Other			Separate flash chamber			
Cost Data						
Effect tubes & nozzles ¹	\$14,677,506	\$14,680,417	\$14,680,417	\$15,222,144	\$15,222,984	\$15,720,680
Effect tubesheets ² and tube support plates	2,884,499	2,929,103	2,929,103	2,057,348	1,953,644	4,632,175
Feed heater tubes	5,621,915	5,621,213	5,621,213	5,254,541	5,254,541	4,921,588
Feed heater tubesheets and support plates	117,364	107,575	107,575	118,781	763,619	123,760
Effect pumps & motors	2,193,800	2,193,300	2,193,800	2,032,800	2,201,280	
Effect piping	1,509,557	1,736,058	1,779,961	1,446,439	1,794,197	2,142,910
Structural costs	4,593,837	5,487,447	5,567,411	3,959,325	4,460,877	5,210,324
Differential in distri- bution piping					418,651	- 476,730
Total of Items Estimated	\$31,598,478	\$32,755,613	\$32,879,480	\$30,091,378	\$32,074,793	\$32,274,707

¹General Electric fluted tubes procured at \$2.70 and \$3.20 per square foot for 10- and 5-ft tubes with nozzles. Two-inch smooth tubes procured at \$2/ft². Installed cost of metal nozzles in Effects I through III is \$6.50 each. Installed cost of plastic nozzles in Effects IV through XIII is \$1.80 each. General Electric tube costs include nozzles.

²Tubesheet costs include \$1.06/ft² for a plastic coating.

Table 7. Comparison of Rectangular VTE and Reference MSF Plants, 250-Mgd
(Flowsheet 251A1, Fig. 3)

	Evaporator Size Without Deaerator	Shell Cost Without Tubes, Deaerator, and Weirs	Cost of Weirs	Surface Area (Less MSF Erine Heater and VTE Final Condenser)	Bundle Cost	$\bar{U}$	
MSF	$4.429 \times 10^6 \text{ ft}^3$	\$11,119,435	\$1,837,000	Recovery 12,444,000 ft^2 Reject 1,279,000 ft^2 <u>13,723,000 ft^2</u>	\$31,706,110 3,747,000 <u>35,453,110</u>	640 454	
VTE	$3.255 \times 10^6 \text{ ft}^3$	\$4,279,500	\$86,200	VT 3,930,000 ft^2 Feed Htr. 2,110,000 ft^2 <u>6,040,000 ft^2</u>	\$17,562,000 5,739,280 <u>\$23,301,280</u>	1626 710	$\frac{\text{ft}}{\text{oo}}$
Ratio, VTE/MSF	0.735	0.385	0.047	0.44	0.657		
Comments	344,760 ft^3 subtracted from MSF design figure for deaerator	MSF = \$2.55/ ft^3 VTE = 1.31/ ft^3	Interstage Flow = 21.4% Ratio No. stages to No. effects = 31%	Recovery = \$2.55/ ft^2 Reject = 2.93/ ft^2 MSF Totals 2.58/ ft^2 VT \$4.47/ ft^2 Feed Htr. = 2.72/ ft^2 VTE Total = 3.86/ ft^2	Phenolic coated tube sheets used in bundle costs.		

The cost of the 3-in.-OD double-fluted, 0.049-in. nominal wall thickness, aluminum-brass tubes (with nozzles) is the material cost given in the General Electric 50-Mgd conceptual design report.² These costs were \$2.70/ft² for 10-ft tubes and \$3.20/ft² for 5-ft tubes. A 15-yr life is predicted for these tubes. The cost of replacement is cost included in the preliminary estimates which are for comparison purposes only.

The cost estimates are based on the engineering, procurement, and construction being performed by one general contractor under favorable conditions of project scheduling, contracts, etc. Labor costs are 1965 rates for a Southern California site.

Originally, the intent was to use the same flowsheet for all structure analysis work. For this purpose, a preliminary flowsheet was developed (Fig. 3) with 3-in.-OD, doubly fluted, 10- and 5-ft-long tubes with equal surface areas in each effect. However the interest in examining the use of 10-ft long tubes throughout the vertical plant necessitated the development of two additional flowsheets. These three flowsheets were made as comparable as possible.

For comparison purposes it was considered adequate to estimate only four effects in each design and to assume a straight line relationship for the cost of the other effects. Table 7 is shown as a typical work sheet for the development of detail costs and Figs. 21 and 22 as typical plots of bundle cost and effect cost for this phase of the study. Similar material was prepared for each of the studies in this section.

1.6.3 Rectilinear Structures

Rectangular Tube Bundle with 10-ft and 5-ft Tubes. This is the design described in Section 1.3.2 and the detailed costing of it is shown in the examples of Table 8 and Figs. 21 and 22. Since it was the first plant costed, the succeeding rectilinear plants are referenced to it.

Semicircular Tube Bundle with 10- and 5-ft Tubes. The direct capital cost of this structure is about \$1,400,000 or 4.4% greater than a similar design using rectangular tube bundles. The main increase is in the structural costs of the building to accommodate the semicircular bundles. Since the noncondensables flow to the center of the circle, they are collected and transported to the feed heater by piping rather than by concrete ducts. This increases the piping costs about \$227,000 and eliminates the concrete duct cost of \$233,000.

Semicircular Tube Bundle with Separate Flash Chamber. This design is similar to the preceding one with the exception that a floor is added to provide a separate flash chamber physically separated from the vertical tube portion. The cost of adding this feature increased the shell costs by only \$124,000 (0.38%). By adding inexpensive stage walls plus weirs, additional flash stages can be provided for each vertical effect. Placing the feed-heater tube bundles under the floor in the flash chamber increased the building height, but decreased the width. This arrangement permitted

Fig. 21. Typical Plot of Cost of Installed Tube Bundles

Fig. 22. Typical Plot of Cost of Effects

Table 8. Summary of Effect Cost Information
Rectilinear, 250 Mgd, VTE Plant with Rectangular Tube Bundles
(Typical Work Sheet)

	Effect II	Effect VII	Effect VIII	Effect XIII
A. Mechanical				
Effect Tube Bundles				
Vertical Tubes	\$1,046,994	\$1,046,994	\$1,224,758	\$1,224,758
Tube Sheets (Phenolic Coated) (Refer to C.)	118,523	127,175	257,812	417,039
Feed Heater Tube Bundles				
Tubes	432,455	432,455	432,455	432,455
Tube Sheets & Support Plates	9,028	9,028	9,028	9,028
Effect Pumps and Motors	169,400	169,400	168,000	168,000
Effect Brine Piping and Valves (with insulation)	104,024	104,558	131,082	128,757
Subtotal	\$1,880,424	\$1,889,610	\$2,223,135	\$2,380,037
B. Structural				
Steel	\$ 104,019	\$ 100,559	\$ 121,694	\$ 158,077
Concrete	93,308	118,908	134,408	170,508
Phenolic Coating	27,308	38,933	51,533	62,263
Polypropylene Lining	17,039	20,976	35,101	38,440
Expansion Joints	3,289	3,289	3,499	3,499
Adjustable Effect Weirs	6,632	6,632	6,632	6,632
Roof Insulation	5,183	6,555	6,803	8,205
Subtotal	\$ 256,778	\$ 295,852	\$ 359,670	\$ 447,624

Table 8. (Continued)

	Effect II	Effect VII	Effect VIII	Effect XIII
Effect Cost (without Entrainment Separators)	\$2,137,202	\$ 2,185,462	\$2,582,805	\$2,827,661
C. Optional Tube Sheet Material				
For Cupronickel Cladding Add	\$ 130,140	\$ 141,360	\$ 287,207	\$ 493,638
For Bare Steel Subtract	4,970	5,397	10,968	18,850
D. Additional Shell Costs Not Assignable to Effects				
Low Temperature End Wall (Steel)	\$ 54,000			
High Temperature End Wall (Steel)	25,500			
End Walls - Concrete	117,000			
Bridge Crane, Rail and Girders	41,250			
Total Additional Costs	\$ 237,750			
E. Total Evaporator Cost (without Entrainment Separators)				
With Phenolic Coated Tube Sheets				
Average Cost of Effects 1-7	\$1,719,849			
Cost of 1-7 Effects		\$12,038,943		
Average Cost of Effects 8-13	\$2,263,751			
Cost of Effects 8-13		\$13,582,506		
Subtotal		\$25,621,449		
Feed Heater Costs		5,739,279		
Additional Shell Costs (Item D)		237,750		
Evaporator Cost		\$31,598,478		
With Cupronickel Clad Tube Sheet		\$34,882,195		

a reduction in the noncondensable piping but this was counterbalanced by the increased effect piping due to additional elevation (4 ft in the section with 10-ft tubes and 5-1/2 ft in the section with 5-ft tubes). The slight increase in pumping cost due to the additional elevation was not calculated.

Rectangular Tube Bundle with 10-ft Tubes. This design was similar to the same plant with 10- and 5-ft tubes. The direct capital costs decreased 4.8% to \$30,091,000. More surface was required because of the lower heat transfer coefficients for the 10-ft tubes as compared with 5-ft tubes. This is compensated by the lower price per sq ft of surface and by reduction in the number of tube sheet requirements. The installed tube costs increased \$545,000 and the installed tube sheet costs decreased \$827,000. By eliminating the two-level feature of the 5-ft tubes, a reduction in structural costs of \$634,500 was achieved.

1.6.4 Concentric Ring Structure with 10-ft Tubes

The evaporator costs for this design, \$31,656,142, are \$1,565,000 or 5.2% higher than for a rectilinear structure with 10-ft tubes and rectangular tube bundles. The addition of the differential cost (\$418,651) of distribution piping around the circular structure and a steam-supply line to the center increase the differential costs to \$1,983,000 or a 6.6% increase. The use of numerous circumferential tube bundles with associated water boxes and tube sheets increases the feed heater cost by \$645,000. The effect piping and piping between water boxes adds \$348,000 and the structure plus the underground pump pits add an additional \$502,000.

1.6.5 Vertical Structure with 5-ft Tubes

The evaporator costs for this design are \$32,275,000 which is 2.9% higher than the rectilinear plant with 10- and 5-ft tubes and 7-1/4% higher than the rectilinear plant with 10-ft tubes throughout.

The cost of the effect tube bundles increased from \$17,280,000 for 4,395,000 ft² of surface in the rectilinear structure with 10-ft-long tubes to \$20,353,000 for 3,880,000 ft² of surface in the vertical structure with 5-ft-long tubes. The installed cost of the 10-ft-long tubes was \$3.93/ft² of surface and the installed cost of the 5-ft-long tubes was \$5.25/ft². The tube cost increased \$499,000 and the tube sheets increased \$2,575,000 (primarily due to almost doubling the number of tube sheets required). The feed-heater tube bundle costs were \$2.72/ft² for 1,978,000 ft² in the rectilinear structure with 10-ft tubes and \$2.75/ft² for 1,833,500 ft² in the vertical structure with 5-ft tubes; a decrease of \$328,000 for the vertical structure.

The vertical structure cost was \$2,033,000 less due to the elimination of effect pumps and motors. However, the increased feed-pump power required to pump the brine to the top of a vertical structure was not included in the cost. In both the vertical and horizontal-type plants the entire brine feed is pumped to the first effect (via the feed heaters) but in the vertical structure this is several hundred feet high. The interstage brine pumps

in the horizontal structure do not equal this in power requirements since only part of the brine flow is pumped to the top of succeeding effects. Assuming a 6% interest, 30-yr plant life, 90% load factor, and electrical power at 2.0 mills/kwhr, the increase in power required for a vertical structure would be equivalent to a capital cost of \$1,000,000. The effect piping increased \$697,000 due entirely to the float valves between effects. There was a decrease in distribution piping external to the shell amounting to \$477,000. The structure costs increased by \$1,250,000.

2. EXPLORATORY PROCESS DESIGN AND EVALUATION

2.1 Comparison of Conventional and Enhanced Tubing

In order to develop cost information for the comparison of conventional tubing with enhanced tubing, the rectilinear plant was modified for evaporator bundles of 2-in.-diam, 24-ft-long smooth tubes and the reference process of Chapter 1 was recalculated using the heat transfer coefficients and brine feed rates indicated by the experience of the OSW demonstration plant in Freeport. The developed flowsheet (Case 281) is shown in Fig. 23.

The vertical tube bundles were calculated for Case 281 using the bundle configuration and pressure drop allowances of Section 1.2 but the lower heat transfer coefficients and friction factor assumed for the smooth tubes. The minimum tube pitch of 1.15 was found to be permissible for Effects II, VII, and VIII but Effect XIII required some relaxation to meet the pressure drop requirements. Table 9 is a tabulation of the results.

Table 9. Rectangular Tube Bundles with 2-in.-diam Smooth Tubes, Case Number 281

	Effect Number			
	II	VII	VIII	XIII
Steam temperature, °F	258.9	200.3	188.1	121.1
$\bar{U}$, Btu/hr.ft ² .°F	688.0	613.0	593.0	405.0
Bundle depth, ft	9.99	9.99	9.98	9.72
Bundle height, ft	24.0	24.0	24.0	24.0
Face velocity, fps	4.22	9.49	12.50	37.50
Temperature loss, °F	0.10	0.41	0.80	1.48
Number of rows	60.0	60.0	60.0	50.0
S/D, First row	1.15	1.15	1.15	1.72
S/D, Row n	1.15	1.15	1.15	1.15
n	1.0	1.0	1.0	38.0
Number of tubes/10-ft face	3130.0	3130.0	3130.0	2330.0
Tube density, number/ft ²	31.33	31.35	31.35	23.31

Fig. 23. VTE Conceptual Design; VTE Process Flowsheet;
250-Mgd Capacity; Case No. 281

The plant layout was modified to accept these bundles. The structural design for the evaporator using 2-in.-diam by 24-ft-long smooth tubes is very similar to the portion of the rectilinear design using 10-ft GE fluted tubes. The additional length of tubing raises the height of the plant from 23 ft to an average of 36 ft. The additional tubing length also requires that the effect and noncondensable duct walls be increased from 6- to 8-in. thick.

However, the plan dimensions of this plant are considerably less. Each module of the 10-ft-tube plant expands in width from 70 ft at Effect I to 120 ft at Effect XIII. The smooth-tube plant module is 64-ft wide from Effect I through Effect VII and then increases uniformly in width to 84 ft at Effect XIII. Both plants have a total length of 308 ft.

The additional height of this plant requires the use of larger pumps, which dictates a minimum pump canyon width of 30 ft. The variation of effect width further widens the canyon at Effect VII and establishes a crane span of 40 ft which is twice that used in the 10-ft-tube plant.

Conventional brine distribution methods were considered to be impractical for large bundles with small S/D ratios. This plant is therefore designed to operate with filled brine chests and to use nozzles in each tube.

Detailed bills of material and cost estimates were made for Effect II, VII, VIII, and XIII, the total evaporator cost was calculated from these and is shown in Table 10 on a basis comparable to Table 6 of Section 1.

2.2 Comparison of Multistage-per-Effect Feed Heating with Single-Stage-per-Effect Feed Heating

The process flowsheets considered up to this point in the conceptual design program have had a single stage of feed heating related to each of their 13 effects. This created a feed heater circuit comparable to a 13 stage flash evaporator which necessarily has a large interstage flashdown and is thereby limited to a performance ratio of less than 6. The performance ratio of the vertical tube effect circuit of a 13 effect evaporator is approximately 13 since each effect adds a performance ratio of approximately 1. The sum of the performance ratios of these two circuits gives an overall plant performance ratio of approximately 10. Increasing the number of stages in the feed heater circuit decreases the availability loss. This increase in availability can be used either to decrease the heat transfer surface requirements or increase the performance ratio of the feed heater circuit. If one chooses to increase the performance ratio of the feed heater this increase can be used either to increase the overall plant performance ratio or to decrease the surface requirements of the vertical tube effect circuit by decreasing the number of effects.

It was decided that this economic analysis of multistage feed heating could be more easily done and understood solely on the basis of capital cost comparisons, in other words all of the plants considered would have the same overall performance ratio and therefore the same heat cost.

Table 10. Comparison of Cost of 250-Mgd
Vertical Tube Evaporator with Smooth
Tubes versus Reference Design with Fluted Tubes

	Rectilinear Structure	
Tube Conditions:		
Case number	281	261C
Length, ft	24	10
Diameter, in. OD	2	3
Tube type	Smooth	Fluted
Type bundle	Rectangular	Rectangular
Cost Data		
Effect tubes & nozzles*	\$29,316,976T 2,507,000N	\$15,222,144
Effect tubesheets and support plates	1,610,752	2,057,348
Feed heater tubes	5,621,473	5,254,541
Feed heater tubesheets and support plates	128,390	118,781
Effect pumps & motors	5,611,950	2,032,800
Effect piping	2,287,927	1,446,439
Structural costs	4,182,495	3,959,325
Total of Items Estimated	\$51,266,963	\$30,091,378

* GE fluted tubes procured at \$2.70 and \$3.20/ft²
for 10- and 5-ft tubes with nozzles. Two-inch smooth
tubes procured at \$2.00/ft².

By this time in the design program the VTE computer program contained all of the detail cost information developed in section one and specifically the study of section 1.3.3 which provided the costs for physically separating the feed heater circuit from the VTE circuit and the incremental cost of interstage walls. Although these costs were neither complete nor final, they were adequate for this analysis.

This program and cost data were used to compute the matrix of plants described in Table 11. The first two plants have single stage per effect feed heating but differ by having in one case equal feed heater surface in each stage and in the other case equal feed heater surface in each stage. The other three plants have multistage feed heating per effect and utilize the increased availability to reduce the feed heater surface, to eliminate two effects, and to eliminate one effect and reduce the evaporating temperature range. Table 12 gives the results of the calculations and indicates no incentive to reduce feed heater surface but a strong incentive to increase feed heater efficiency. The arbitrary limitations placed on this investigation in order to keep it simple forced the comparison of plants of varying degrees of nonoptimum such that a

Table 11. Descriptive Characteristics of Plants for Investigation of Preheaters
(10-ft-long fluted tubes)

Type System	Number of Effects	Performance Ratio	Product Temperature (°F)	Input Variables	Area or Temperature Gradient Set By	Remarks
Conventional VTE reference	13	9.95	103.7	Evaporator areas Preheater areas	Equal evaporator and preheater areas	Preliminary reference case; Preheater approach temperatures vary
Conventional VTE	13	10.46	103.7	Evaporator temperatures Preheater temperatures	Same evaporator temperature gradient as reference	Preheater approach temperature constant at 8°F
VTE + MSF preheat, MSF area reduced	13	9.95	103.7	Evaporator temperatures Performance ratio	Same evaporator temperature gradient as reference	Same as reference plant but with MSF preheater
VTE + MSF preheat, number effects reduced	11	9.95	103.7	Evaporator areas Performance ratio	Equal evaporator areas	Eleven equal area effects and MSF preheat
VTE + MSF preheat, number effects reduced and evaporating range reduced	12	9.95	123.6	Evaporator temperatures Performance ratio	Same evaporator temperature gradient as reference	Same VE conditions as reference except bottom effect eliminated

Table 12. Major Variable Capital Cost for Plants for Investigation of Preheaters

Cost Category	Reference Plant	Preheater Approach Temperature Equal (8°F)	MSF Advantage Used to Reduce MSF Area	MSF Advantage Used to Reduce VE Effects from 13 to 11	MSF Advantage Used to Reduce Evaporating Temperature Range and Number of Effects
Capital Cost, 10 ⁶ Dollars					
Fluted area + shell	30.6760	30.8109	30.9600	25.6874	31.2859
Preheaters					
Surface	5.3343	7.4715	4.8095	7.4541	5.0823
Credit for no flash chambers	-0.1223	-0.1223	0.	0.	0.
Flash chamber interstage partitions	0.	0.	0.4638	0.6571	0.4252
Subtotal	5.2120	7.3492	5.2733	8.1112	5.5075
Condenser	2.1967	2.0345	1.5330	1.9050	1.1851
MSF brine heater	0.	0.	0.4806	0.3705	0.4138
Seawater intake system	4.9878	4.7385	4.9025	4.9360	3.4420
Delivery pumps	0.5850	0.5464	0.5718	0.5770	0.3976
Totals					
Major variable direct capital cost	43.6575	45.4795	43.7212	41.5871	42.2319
Directs plus indirects (higher cost factor = 1.20)	52.3890	54.5754	52.4654	49.9045	50.6783
Annual Costs, 10 ⁶ dollars/year					
Partial capital cost	2.9338	3.0562	2.9381	2.7949	2.8380
Heat cost	4.1426	4.1219	4.1426	4.1426	4.1426
Total	7.0764	7.1781	7.0807	6.9375	6.9806
Product Cost, ¢/1000 gal					
Partial unit water cost	8.617	8.737	8.622	8.447	8.500

more sophisticated analysis would show an even stronger incentive to increase the feed heater efficiency.

2.3 Parametric Studies

2.3.1 Evaporator Optimization

The preceding work in Sections 1 and 2 led to the selection of the major design and process features for the final plant design and provided detail costs for those items peculiar to the vertical tube evaporator. A computer program subroutine was developed for calculating the cost of vertical tube evaporator plants based in part on the costs and in part, for example the chemical system, on information borrowed from the multistage flash plant cost subroutine. The plant model is as follows. The plant floor plan is trapezoidal in shape, narrower at the high-temperature end than the cold end. There are three levels, each approximately 10-ft high; the lower level houses the multistage flash preheater; the second level provides space for vapor-liquid disengagement for the vertical tubes, and the top level contains the doubly fluted-tube bundles.

This subroutine was installed in the ORVE Code and used to enable the selection of the remaining evaporator design and process features through a parametric study varying the number of effects from 12 to 16, concentration ratio 2 to 3.25, temperature of the final effect from 94 to 117°F, performance ratio of the MSF preheater from 6 to 8, and the condenser approach temperature from 5 to 15°F. The results of this study are given in Tables 13 through 15. Table 13 summarizes the plant costs, Table 14 the plant temperatures and flows, and Table 15 the plant geometry.

The overall results of each parametric variation are given in Fig. 24-28. These figures indicate that the variation in water cost as a result of changing the design parameters was really quite small. If, in some cases, an over-riding consideration were to lower the capital cost at the expense of increased operating cost it could easily be done by varying the number of effects. The total annual cost increase would be minimal. The maximum changes in costs found were:

<u>Parameter</u>	<u>Change in Water Cost</u>	<u>Change in Capital Cost</u>
Performance ratio MSF	0.9%	3.2%
Condenser approach temperature	0.2%	0.2%
Number of effects	1.8%	18.8%
Concentration ratio	2.6%	3.0%
Heat effect temperature	1.4%	2.7%

The minimum water cost was achieved with 14 effects, a final effect temperature of 107°F, a concentration ratio of 3, an MSF performance ratio of 6.29, and a condenser approach temperature of 5°F.

TABLE 13
CAPITAL COSTS OF MAJOR EQUIPMENT ITEMS
AND UNIT WATER COSTS
FOR VE PARAMETRIC STUDY

	Reference	12 Effects	13 Effects	14 Effects	16 Effects	17 Effects	Concentration Ratio of 2.0	Concentration Ratio of 2.75	Concentration Ratio of 3.0	Performance Ratio of 6.91, MSF	Performance Ratio of 7.38 MSF	Performance Ratio of 8.52 MSF	Condenser Approach Temperature, 10°F	Condenser Approach Temperature, 15°F	Brine Temperature in Bottom Effect, 117°F	Brine Temperature in Bottom Effect, 107°F	Brine Temperature in Bottom Effect, 97°F	Brine Temperature in Bottom Effect, 94°F
<u>Direct Capital Investment, millions of dollars</u>																		
Shell total	4.07	3.17	3.47	3.77	4.40	4.74	3.76	4.19	4.30	4.07	4.07	4.07	4.07	4.07	4.13	4.01	3.93	3.91
Effect tubes	14.75	10.93	12.12	13.38	16.14	17.58	13.16	15.39	16.03	14.74	14.75	14.75	14.74	14.75	15.26	14.28	13.41	13.17
MSF tubes	7.93	9.31	8.76	8.33	7.61	7.36	9.02	7.70	7.57	6.42	7.09	9.02	7.93	7.93	7.95	7.93	8.04	8.12
Brine heater	0.61	0.53	0.55	0.58	0.65	0.68	0.70	0.59	0.58	0.66	0.64	0.59	0.61	0.61	0.63	0.60	0.58	0.57
Condenser	1.26	1.81	1.59	1.41	1.14	1.05	1.07	1.32	1.37	1.22	1.24	1.29	1.09	1.00	0.93	1.10	1.38	1.52
Deaerator	2.33	2.33	2.33	2.33	2.33	2.33	2.74	2.21	2.12	2.33	2.33	2.33	2.33	2.33	2.33	2.33	2.33	2.33
MSF shell	3.17	2.56	2.77	2.94	3.48	3.75	2.90	3.27	3.36	3.20	3.18	3.16	3.17	3.17	3.22	3.14	3.11	3.11
Site work	0.29	0.25	0.27	0.28	0.31	0.34	0.27	0.30	0.30	0.29	0.29	0.29	0.29	0.29	0.29	0.29	0.29	0.29
Seawater intake	4.84	4.14	4.87	4.86	4.83	4.35	4.87	4.84	4.83	4.81	4.86	4.83	4.86	4.89	4.86	4.26	4.65	4.97
Pumps and drives	6.51	5.85	6.04	6.25	6.78	7.07	6.63	6.49	6.50	6.77	6.63	6.40	6.58	6.68	6.66	6.75	7.17	7.47
Electrical equipment	0.94	0.86	0.88	0.91	0.93	1.02	1.01	0.92	0.91	1.00	0.97	0.91	0.95	0.95	0.95	0.96	0.98	1.01
Instruments, buildings, cranes	3.47	3.17	3.27	3.37	3.56	3.65	3.47	3.47	3.47	3.47	3.47	3.47	3.47	3.47	3.47	3.47	3.47	3.47
Valves and piping	4.58	4.39	4.47	4.53	4.74	4.92	4.50	4.63	4.68	4.68	4.63	4.54	4.63	4.70	4.67	4.77	5.10	5.32
Total Direct Plus Indirect Capital Investment Cost, millions of dollars	69.01	62.49	64.99	66.85	71.66	73.93	68.46	69.62	70.44	67.83	68.28	70.10	69.00	69.16	69.77	68.00	68.75	69.79
<u>Unit Water Cost, \$/1000 gal</u>																		
Capital	0.0528	0.0469	0.0491	0.05082	0.0552	0.0573	0.0518	0.0535	0.0543	0.052	0.0523	0.0536	0.0528	0.0529	0.0535	0.0520	0.0521	0.0527
Heat	0.0376	0.0449	0.0420	0.0396	0.0358	0.0342	0.0385	0.0372	0.0370	0.0391	0.0384	0.0368	0.0376	0.0376	0.0375	0.0376	0.0377	0.0377
Chemicals	0.0118	0.0118	0.0118	0.0118	0.0117	0.0117	0.0137	0.0112	0.0108	0.0118	0.0118	0.0118	0.0118	0.0118	0.0118	0.0118	0.0120	0.0121
Power	0.0056	0.0049	0.0051	0.0053	0.0059	0.0062	0.0062	0.0054	0.0053	0.0057	0.0056	0.0055	0.0056	0.0057	0.0057	0.0057	0.0059	0.0061
Operating	0.0069	0.0078	0.0069	0.0069	0.0069	0.0069	0.0069	0.0069	0.0069	0.0069	0.0069	0.0069	0.0069	0.0069	0.0069	0.0078	0.0078	0.0078
Maintenance and supplies	0.0014	0.0013	0.0013	0.0014	0.0015	0.0015	0.0014	0.0014	0.0015	0.0014	0.0014	0.0015	0.0014	0.0014	0.0014	0.0014	0.0014	0.0014
Total	0.11608	0.11764	0.11625	0.11583	0.11698	0.11789	0.11860	0.11568	0.11569	0.11696	0.11640	0.11594	0.11614	0.11635	0.11691	0.11633	0.11694	0.11789

TABLE 15
PLANT GEOMETRY FOR PARAMETRIC STUDY

	Reference	12 Effects	13 Effects	14 Effects	16 Effects	Concentration Ratio of 2.0	Concentration Ratio of 2.5	Concentration Ratio of 2.75	Concentration Ratio of 3.0	Performance Ratio of 6.91, MSF	Performance Ratio of 7.38, MSF	Performance Ratio of 8.52, MSF	Condenser Approach Temperature, 10°F	Condenser Approach Temperature, 15°F	Brine Temperature in Bottom Effect, 117°F	Brine Temperature in Bottom Effect, 107°F	Brine Temperature in Bottom Effect, 97°F	Brine Temperature in Bottom Effect 94°F
VTE plant length, ft	348	282	304	326	370	392	348	348	348	348	348	348	348	348	348	348	348	348
MSF plant length, ft	348	282	304	326	370	392	348	348	348	348	348	348	348	348	348	348	348	348
Brine heater length, ft	19.8	12.6	14.7	17.1	22.7	25.5	20.6	19.4	19.0	25.4	22.6	17.2	13.07	19.8	20.1	19.4	18.5	18.2
Condenser length, ft*	68.2	53.7	58.2	63.0	73.4	78.2	71.1	66.8	65.2	79.2	73.8	62.6	47.7	35.4	38.6	31.6	21.9	18.3
Preheater tube OD,** in.	0.7500	0.7500	0.7500	0.7500	0.7500	0.7500	0.7500	0.7500	0.7500	0.7500	0.7500	0.7500	0.7500	0.7500	0.7500	0.7500	0.7500	0.7500
Length of fluted tubes, ft	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10	10
Plant height, ft	31	31	31	31	31	31	31	31	31	31	31	31	31	31	31	31	31	31

* Based on reject stream.
** Tube wall thickness is 0.042 inches.

Fig. 24. Varying Final Effect Evaporator Temperature

Fig. 25. Varying MSF Performance Ratio

Fig. 26. Varying Concentration Ratio

Fig. 27. Varying Condenser Approach Temperature

Fig. 28. Varying the Number of Evaporator Effects

2.3.2 Feed Heater Optimization

A second parametric study was made using the ORVE Code to enable the selection of the feed heater process features and to answer the following questions:

1. What is the effect of varying linear temperature decrements throughout the MSF preheat stages (without regard to the stage-to-stage pressure driving force which is needed to drive the brine and product streams down the cascade)?
2. What is the penalty for deviating from the optimum number of MSF preheat stages
 - a. in the temperature range spanned by the vertical evaporators;
 - b. in the temperature range below the bottom evaporator evaporating temperature?
3. What is the gain or loss associated with changing from the reference case (nonuniform MSF stage decrements) to an equal temperature decrement configuration in the MSF preheater?
4. What is the penalty when the temperature of the MSF feed to the VTE nozzles is different from the evaporating temperature by an amount not to exceed the design MSF stage temperature decrement?
5. What capital cost change is involved in making the VTE evaporating temperature match the MSF brine temperature by altering the VTE effect areas?

Each of the above questions will be discussed in order.

1. Linear Temperature Decrement in MSF Stages. In this study only those MSF stages which operate over the temperature range spanned by the VTE system were under consideration. The VTE reference flow diagram has a flashing range of 148°F; the flashing range and the number of stages were held constant in this work. It was assumed that the brine feed can be taken from two adjacent MSF stages in a proper ratio to feed the VTE effects at the desired temperature. Furthermore, it was assumed that the cost of this method of matching temperature would be approximately constant and not a factor in this study. This study then comes down to determining the particular linear MSF stage temperature decrement taper which yields the minimum MSF preheater area.

The results of the study are presented in Table 16 and Figure 29. Four cases were examined which spanned the optimum. The linearly varying stage temperature decrement, the flashing temperature, the thermal driving force, and the stage area are all tabulated versus stage position for each case in Table 16. Entries are made for every fifth stage in order to keep the table manageable.

Table 16. MSF Area Requirement as a Function of Changes in Linear Distribution in Stage Temperature Decrement
(Study Based on 311-A (2/20/67) Reference Case with 45 Stages Operating Over VTE Evaporator Temperature Range; 148° Flashing Range)

Stage Position	Case 1				Case 2				Case 3			
	Stage Temp. Decrement (°F)	Flashing Temp. (°F)	LMTD (°F)	Area (10 ⁴ ft ²)	Stage Temp. Decrement (°F)	Flashing Temp. (°F)	LMTD (°F)	Area (10 ⁴ ft ²)	Stage Temp. Decrement (°F)	Flashing Temp. (°F)	LMTD (°F)	Area (10 ⁴ ft ²)
1	2.980	257.02	10.68	4.9848	2.235	257.76	11.00	3.652	4.471	255.53	9.95	7.988
2	2.980	254.04	10.72	4.9821	2.235	255.53	11.03	3.651	4.471	251.06	10.03	7.975
10	2.928	230.75	10.97	4.9222	2.177	238.03	11.23	3.565	2.970	218.95	11.07	5.036
15	2.970	215.98	11.11	5.0381	2.928	224.89	11.05	4.951	3.113	203.77	11.10	5.383
20	3.113	200.66	11.15	5.3898	3.031	209.92	11.11	5.207	3.222	187.88	11.20	5.691
25	3.371	184.51	11.11	6.0276	3.222	194.32	11.11	5.678	3.559	170.65	11.10	6.562
30	3.559	167.09	11.14	6.5921	3.371	177.77	11.18	6.106	2.867	155.62	11.43	5.341
35	3.155	149.31	11.23	6.0559	3.823	159.44	10.99	7.324	2.524	143.00	11.52	4.826
40	3.610	132.17	10.90	7.4759	4.814	138.18	10.45	0.208	2.888	128.56	11.16	5.935
45	4.139	112.00	10.65	9.766	5.518	112.00	9.82	4.073	3.311	112.00	11.12	7.522
Total MSF Area This Section, 10 ⁶ ft ²				2.7521								
								2.7974				
												2.7612

Fig. 29. MSF Area vs Temperature Decrement in First Stage -
Stage Decrement Taper Approximately Linear

In Fig. 29 the total MSF area is plotted against the temperature decrement in stage 1. It is seen that the optimum taper would be one having a temperature decrement of about 3.5°F on stage 1. Deviations from this optimum taper to as high as 4°F and as low as 3°F , however, will result in a cost penalty of less than 0.2% of the total MSF area cost.

2. Optimum Number of Stages.

a. In Temperature Range Spanned by the VTE system. Here again, based on the reference flow diagram, the flashing range is 148°F . It was assumed that the temperature decrement is constant for each of the four cases studied, and again assumed that the cost of matching feed temperatures to VTE evaporating temperatures would not be a factor in the cost comparison.

The area calculations and results are presented in Table 17. Table 18 presents the cost balance in which the area savings computed in Table 17 are weighed against the cost of additional stage partitions. Finally, in Fig. 30 the net savings (in terms of percent of design area) are plotted against the number of stages. It may be seen that the optimum number of stages is 44, however, deviations of ± 3 stages will result in a cost penalty of less than 0.2% of the total design MSF area.

b. In Temperature Range Below the Evaporating Temperature of the Bottom Effect. The flashing range is held constant at the 20°F range obtained from the reference diagram. The temperature decrement is varied for each case studied as above, but is the same for each stage within a case.

The results of the area calculations are presented in Table 19, and Table 20 contains the cost analysis for this section of MSF preheater.

In Fig. 31 the net savings are plotted versus the number of stages. The optimum number of stages is 6 and represents a savings of 7.2% of the section area relative to our reference diagram which has 3 stages in this section. The stage-to-stage pressure difference available for brine flow and flashing is quite low in this area so that the designer may wish to trade heat transfer surface for improved hydraulics. It is, therefore, important to know that one may deviate from 6 stages (3.33°F temperature decrement) to 4 stages (5°F temperature decrement) and lose only one-seventh of the gain achieved with the optimum number of stages for heat transfer.

3. Penalty Associated with the MSF Temperature Decrement Array in the Reference Flowsheet Relative to Constant Temperature Decrement. In the reference flow diagram, the portion of the MSF system operating over the VTE temperature range contains 40 stages (an average temperature decrement of 3.7°F). The temperature decrement of a group of stages associated with a given effect is constant, but the decrement varies from one effect to another. This is done in order to match MSF feed temperature with VTE evaporating temperature. The question arises as to how much is being paid in preheater area because this matching criteria imposes a nonoptimum stage decrement array on the MSF preheater. To answer this question, the reference case is compared to a uniform decrement case which is reasonably close to optimum. To satisfy the matching criteria, it was assumed that

Table 17. MSF Area Requirement as a Function of Number
of Stages Operating Over VTE Temperature Range
(Constant MSF Stage Temperature Decrement for Each Case)

Stage Number	Temperature Decrement, °F											
	3.700			3.364			3.217			3.054		
	Flashing Temp. (°F)	LMTD (°F)	Area (10 ⁴ ft ²)	Flashing Temp. (°F)	LMTD (°F)	Area (10 ⁴ ft ²)	Flashing Temp. (°F)	LMTD (°F)	Area (10 ⁴ ft ²)	Flashing Temp. (°F)	LMTD (°F)	Area (10 ⁴ ft ²)
1	256.3	11.14	6.4168	256.6	11.48	5.661	256.8	11.38	5.462	257.0	11.46	5.148
5	241.5	11.21	6.4992	243.2	11.37	5.805	243.9	11.44	5.517	244.7	11.52	5.201
10	223.0	11.37	6.5897	226.4	11.49	5.861	227.8	11.53	5.580	229.5	11.64	5.219
15	204.5	11.57	6.6773	209.5	11.67	5.930	211.7	11.72	5.627	214.2	11.78	5.301
20	186.0	11.66	6.8288	192.7	11.80	6.044	195.7	11.86	5.712	198.9	11.94	5.360
25	167.5	11.84	6.9969	175.9	11.90	6.174	179.6	11.95	5.836	183.6	11.98	5.501
30	149.0	11.95	7.2700	159.1	12.02	6.339	163.5	12.09	5.974	168.4	12.10	5.603
35	130.5	12.09	7.5469	142.3	12.15	6.598	147.4	12.21	6.197	153.1	12.23	5.755
40	112.0	12.12	8.3707	125.5	12.29	7.032	131.3	12.32	6.400	137.8	12.30	5.980
44				112.0	12.29	7.505	118.4	12.36	6.962	125.6	12.46	6.180
46							112.0	12.36	7.137	112.0	12.44	6.567
48.5											12.46	3.387
Total MSF Area this Sec- tion, 10 ⁶ ft ²			2.80073			2.7495			2.7275			2.7165

Table 18. Determination of Optimum Number of MSF Stages
Operating Over VTE Temperature Range

Unit MSF Area Cost = \$2.476/ft²
Cost of Adding A Stage Partition = \$28,589

	Number of Stages			
	40	44	46	48.5
MSF Area, ^a 10 ⁶ ft ²	2.8007	2.7495	2.7275	2.7165
(Base Area-Area), 10 ⁶ ft ²	(Base)	0.0512	0.0732	0.0842
Area Savings, \$		126,770	181,240	208,480
Incremental Stage Wall Cost, \$		114,356	171,534	241,860
Net Savings, \$		12,414	9,706	-33,380
Net Savings, \$ of MSF Area Cost this Section		0.18%	0.14%	-0.33%

^aThese figures are developed in Table 17.

Fig. 30. MSF Stages Operating Over VTE Temperature Range

Table 19. MSF Area Requirement as a Function of the Number of Stages Operating
in Temperature Range Below Bottom VTE Evaporating Temperature
(Constant MSF Stage Temperature Decrement for Each Case)

Stage Number	Temperature Decrement, °F														
	Flashing Temp. (°F)	LMTD (°F)	Area (10 ⁶ ft ²)	Flashing Temp. (°F)	LMTD (°F)	Area (10 ⁶ ft ²)	Flashing Temp. (°F)	LMTD (°F)	Area (10 ⁶ ft ²)	Flashing Temp. (°F)	LMTD (°F)	Area (10 ⁶ ft ²)	Flashing Temp. (°F)	LMTD (°F)	Area (10 ⁶ ft ²)
1	105.3	10.40	0.1647	107.0	11.22	0.11386	108.0	11.71	0.08700	108.7	12.04	0.07032	109.5	12.46	0.05082
2	98.7	10.44	0.1678	102.0	11.25	0.11556	104.0	11.74	0.08797	105.3	12.06	0.07101	107.0	12.47	0.05122
3	92.0	10.49	0.1712	97.0	11.28	0.11347	100.0	11.76	0.08905	102.0	12.08	0.07174	104.5	12.49	0.05158
4				92.0	11.32	0.11899	96.0	11.79	0.09018	98.7	12.10	0.07245	102.0	12.50	0.05200
5							92.0	11.82	0.09117	95.3	12.13	0.07323	99.5	12.51	0.05246
6										92.0	12.15	0.07391	97.0	12.53	0.05287
7													94.5	12.55	0.05323
8													92.0	12.57	0.05358
Total MSF Area this Section, 10 ⁶ ft ²			0.5037			0.4619			0.4454			0.4327			0.4178

Table 20. Determination of Optimum Number of MSF Stages
Operating Below the Bottom VTE Evaporating Temperature

Unit MSF Area Cost = \$2.476/ft²
Cost of Adding A Stage Partition = \$28,589

	Number of Stages				
	3	4	5	6	8
Stage Temperature Decrement, °F	6.667	5.000	4.000	3.333	2.500
MSF Area, ^a 10 ⁶ ft ²	0.5037	0.46188	0.44537	0.43266	0.41776
(Base Area-Area), 10 ⁶ ft ²	Base	0.04182	0.05833	0.07104	0.08594
Area Savings, \$		103,500	144,400	175,895	212,700
Incremental Stage Wall Cost, \$		28,589	57,178	85,767	142,945
Net Savings, \$		74,911	87,222	90,128	69,835
Net Savings, % of MSF Area Cost, This Section		6.00%	6.99%	7.23%	5.60%

^aThese figures are developed in Table 19.

Fig. 31. Area Requirement versus Number of Stages for that
Portion of the MSF System Operating Between Final
Evaporating Temperature and Blowdown Temperature

the brine feed can be withdrawn from two adjacent MSF stages in a ratio which will result in a feed temperature which matches the evaporating temperature. It was further assumed that this can be accomplished at low or no extra cost. The results of the calculations are:

MSF Area, Reference Case, ft ²	2.80078 x 10 ⁶
MSF Area, Constant Decrement, ft ²	2.80073 x 10 ⁶

The net savings in area associated with an equal temperature decrement system is equivalent to a capital cost of \$125, which is insignificant.

4. Penalty Associated with Feed Temperature Not Matching VTE Evaporating Temperature. In this study, an alternate of Reference Case 301-C supplies the base VTE system. The base MSF preheat system contains 40 stages with a constant 3.7°F stage temperature decrement. The base case matches the evaporating and feed temperatures by an ideal free pump system. The alternate design has the same average temperature rise per stage, however, the flash chamber design is special: When a subcooled brine stream is fed to a VTE effect, the return brine is fed to the same flash stage. The MSF feed stage has a below average flash down because the feed is returned superheated; it is a high flow, low flash down stage. The situation is exactly reversed where the MSF feed stage furnishes brine which flashes. The return stream goes to a downstream stage which is a low flow high flash down stage. In Table 21 the figures are tabulated to show the cost changes in the VTE portion of the plant as a result of temperature mismatch, and in Table 22 the changes in MSF area requirements are developed and presented.

The numbers in Table 21 were obtained as follows: Column (a), effect evaporating temperature, was obtained from reference flow diagram 301-C. Column (b) is the feed temperature which most closely matches the required evaporating temperature. Judgement was used in selecting these temperatures to try to balance the amount of feed flashing and feed subcooling. Column (c) is the MSF stage from which feed is withdrawn. Column (d) states whether the feed enters the effect subcooled, flashing, or matched. Column (e) indicated degrees (°F) of feed mismatch (+ indicates flashing, - indicates subcooled). Column (f) is the algebraic sum of the degrees of mismatch occurring. These temperature decrements, when multiplied by the product of the feed flow and the heat capacity, will yield the change in the heat generated in the effect relative to design. A positive decrement indicates a decrease. Column (g) is the change in the amount of heat transferred in the effect. Column (h) is the incremental area required to condense the incremental increase or decrease in steam flow from the previous effect and column (i) is the incremental change in VTE product rate.

As a result of the temperature mismatch, 14,000 ft² less area was required and 0.4 Mgd less product made in the VTE section. In Table 22, stage calculations are summarized which lead to a savings of 12,774 ft² MSF area requirements resulting from the mismatch between feed and evaporating temperature. In the model here, it was assumed that the tube-side temperature gradient and flow remains at design. The amount of preheating and,

Table 21. VTE System Changes Resulting When MSF Feed Stream to VTE Nozzles Does Not Match VTE Evaporating Temperature

	(a)	(b)	(c)	(d)	(e)	(f)	(g)	(h)	(i)
VTE Effect Number	VTE Evaporating Temperature (°F)	Feed Temperature from MSF (°F)	MSF Stage Supplying Feed	Feed Enters VTE	Feed Temperature Less Evaporating Temperature (°F)	ΔQ Generated in VTE in Terms of °F of Feed Stream*	($Q - Q_{\text{Design}}$) in Effect (10^{10} Btu/hr)	Additional VTE Area Required (ft ²)	VTE Product Less VTE Design Product (lb/hr)
I	260.00	260.00	Heater	Matched	0.00	0.00	0	0	0
II	251.92	252.60	2	Flashing	+0.68	+0.68	+0.00306	0	0
III	243.80	241.83	5	Subcooled	-1.97	-1.29	-0.00581	+ 3,060	+ 30,600
IV	235.38	238.13	6	Flashing	+2.75	+1.46	+0.00657	- 5,806	- 58,060
V	226.63	226.30	9	Subcooled	-0.33	+1.13	+0.00509	+ 6,570	+ 65,700
VI	217.53	215.74	12	Subcooled	-1.79	-0.66	-0.00297	+ 5,090	+ 50,900
VII	208.06	208.34	14	Flashing	+0.28	-0.38	-0.00171	- 2,970	- 29,700
VIII	198.17	200.86	16	Flashing	+2.69	+2.31	+0.01040	- 1,710	- 17,100
IX	187.82	185.85	20	Subcooled	-1.97	+0.34	+0.00153	+10,400	+104,000
X	176.96	175.26	23	Subcooled	-1.70	-1.36	-0.00612	+15,530	+ 15,300
XI	165.51	164.47	26	Subcooled	-1.40	-2.40	-0.01080	- 6,120	- 61,200
XII	153.37	153.37	29	Matched	0.00	-2.40	-0.01080	-10,800	-108,000
XIII	140.41	142.27	32	Flashing	+1.86	-0.54	-0.00243	-10,800	-108,000
XIV	126.36	126.91	36	Flashing	+0.55	+0.01	0.00	- 2,431	- 24,310
XV	112.00	112.00	40	Matched	0.00	0.00	0.00	0	0
Net Change								-14,000	-140,000

*This quantity is more precisely defined as $(Q_{\text{Actual}} - Q_{\text{Design}}) / (w_{\text{Feed}} \times C_p)$

Table 22. MSF System Area Changes Resulting When
MSF Feed Stream to VTE Nozzles Does Not
Match VTE Evaporating Temperature

MSF Stage Number	MSF Flashing Temperature (°F)	Design LMTD (°F)	Change in LMTD* (°F)	Incremental MSF Area Required (ft ²)
1	256.30	11.14	0	0
2	252.60	11.31	0	0
3	248.70	11.15	-0.10	+ 636
4	245.00	11.20	-0.20	+ 1,272
5	241.83	11.21	+0.07	- 413
6	238.13	11.27	+0.33	- 2,100
7	233.60	11.26	-0.08	+ 509
8	229.90	11.32	-0.50	+ 3,180
9	226.30	11.31	-0.45	+ 2,860
10	222.60	11.37	-0.40	+ 2,540
11	218.90	11.46	-0.40	+ 2,540
12	215.74	11.51	-0.13	+ 826
13	212.04	11.55	+0.14	- 890
14	208.34	11.45	+0.14	- 890
15	204.56	11.57	+0.10	- 636
16	200.86	11.62	+0.06	- 381
17	196.35	11.60	-0.34	+ 2,160
18	192.65	11.65	-0.75	+ 4,770
19	188.95	11.70	-0.75	+ 4,770
20	185.85	11.66	-0.45	+ 2,860
21	182.15	11.71	-0.15	+ 950
22	178.45	11.76	-0.15	+ 950
23	175.26	11.72	+0.10	- 636
24	171.56	11.78	+0.36	- 2,290
25	167.86	11.84	+0.36	- 2,290
26	164.47	11.78	+0.50	- 3,180
27	160.77	11.85	+0.67	- 4,261
28	157.07	11.91	+0.67	- 4,261
29	153.37	11.90	+0.67	- 4,261
30	149.67	11.95	+0.67	- 4,261
31	145.97	12.00	+0.67	- 4,261
32	142.27	12.05	+0.67	- 4,261
33	138.01	11.98	+0.39	- 2,480
34	134.31	12.03	+0.11	- 700
35	130.61	12.09	+0.11	- 700
36	126.91	12.14	+0.11	- 700
37	123.05	12.06	+0.03	- 190
38	119.35	12.12	-0.05	+ 318
39	115.65	12.17	-0.05	+ 318
40	112.00	12.12	-0.05	+ 318
Net Area, ft ²				-12,774

* Resulting from temperature perturbations caused by
temperature mismatch.

thus, the amount of steam condensed in the MSF stages also remains at design. Since 140,000 lb/hr less product is made in the VTE system (Table 21), it follows that this increment of flow will be introduced into the brine tray (rather than the product tray as in the reference plant), and a small increase in MSF product of 14,000 lb/hr will result.

Below, the incremental cost associated with this method of dealing with feed temperature mismatch is developed. The cost figures are based on the following unit costs, charge rate, and load factor:

Unit Cost of MSF Area, \$/ft	2.476
Unit Cost of VTE Surface (including shell), \$/ft	4.49
Unit Cost of Product Water, \$/Kgal	0.10
Annual Charge Rate, 1/yr	0.056
Plant Load Factor	0.90

The incremental change in cost is:

VTE Area Savings, = 14,000 x 4.49	+\$62,860
MSF Area Savings, = 12,775 x 2.476	+\$31,620

Cost of lost product over life of plant, \$

Product Lost in VTE	=	140,000 lb/hr
Product Gained in MSF	=	<u>14,000 lb/hr</u>
Net Product Lost	=	126,000 lb/hr

$$= \frac{126,000 \times 8760 \times 0.9 \times 0.10}{8.3 \times 1000 \times 0.056} \quad \text{e } \$213,720$$

Net Loss	\$119,240
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The net loss is then \$119,240, or about 0.2% of the capital cost of the plant.

It is believed that this study indicates the insensitivity of plant capital cost to small deviations in feed stream temperature mismatch rather than a precise evaluation of the loss associated with this mode of operation. It has been shown (though the figures are not presented here) that it is possible to choose a matching array slightly different from that reflected in Table 21 and thus to obtain a very small decrease in capital cost. This results from the fact that, although the process flow diagram is good and near optimum, it is not a true minimum cost plant. It is, thus, not surprising that small perturbations in some of the process variables may result in a cost savings.

5. Penalty Associated in Adjusting VTE Evaporating Temperature to Match MSF Feed Temperature. Here again, the 40-stage constant decrement (Table 16, Case 3) MSF system is used as a basis. The feed mismatch in the base case

is eliminated by an ideal free pumping system. The mismatch in the comparable plant is eliminated by shifting VTE area so that the VTE feed temperature required is that which the MSF can provide.

The vital parameters leading to the solution of this problem and the VTE area adjustment resulting are presented in Table 23. Effect-by-effect area changes are in the last column. It is seen that the particular VTE effect to MSF stage matching array which was chosen actually results in a savings in VTE area of 5,860 ft². This is slightly less than 0.1% of the total investment in VTE area.

Here, as in the previous study, it is believed that the significant conclusion is that the system cost is quite insensitive to this method of matching temperatures. Since the area distribution is near optimum rather than optimum, half of the random changes of this kind might be expected to gain slightly and half lose.

Conclusions. The following conclusions are considered to be the most significant conclusions to be drawn from the study.

1. The optimum MSF linear temperature decrement taper (40 stages with 148°F flashing range) is one having a decrement of 3.5°F on stage 1. Deviations from this optimum to a decrement as high as 4° or as low as 3° will, however, result in an increase in cost of only about 0.2% of the total MSF area.
2. The optimum number of MSF stages operating over the temperature range spanned by the VTE effects is 44. Deviations of ±3 stages will result in a cost penalty of only about 0.2% of the design MSF area cost in this section.
3. The optimum number of stages operating in the temperature range below the bottom effect VTE evaporating temperature is 6. A net savings amounting to 7.2% of the design area cost in this section can be obtained relative to the 3-stage section in our reference flow diagram. The use of 4 stages, however, yields 6% savings. Since a 4-stage section will give a 5°F stage temperature decrement which has a decided hydraulic advantage over the 3.33 F stage decrement of the 6-stage section, the 4-stage section may be the best design choice.
4. There is no significant gain associated with a constant decrement MSF system over that of the 301-C reference flowsheet. In the reference plant, the temperature decrement is constant for stages associated with each effect; but the value changes by a small amount from one effect to the other as required for matching of feed and evaporating temperatures.
5. The plant cost is quite insensitive to small amounts of mismatching (0 - 3°F) of feed temperature to evaporating temperature. For the particular matching scheme investigated, a cost penalty of only about 0.2% of the plant cost resulted.

Table 23. VTE System Cost Changes Resulting When VTE Evaporating Temperature is Altered to Match MSF Feed Temperature
(40 Stage Constant Temperature Decrement MSF System)

Effect Number	Design Evaporating Temperature (°F)	MSF Stage Supplying Feed	Feed Temperature (°F)	Design VTE Area (10 ⁶ ft ²)	Design VTE Driving Force (°F)	VTE Driving Force After Matching (°F)	Change in VTE Driving Force (°F)	VTE Area Change Required (10 ⁶ ft ²)
I	260.00	Heater	260.0	0.40924	6.17			
II	251.92	2	252.6	0.40924	6.33	5.65	+0.68	+0.04397
III	243.80	4	245.2	0.40924	6.50	5.78	+0.72	+0.04532
IV	235.38	7	234.1	0.40924	6.69	9.37	-2.68	-0.16440
V	226.63	9	226.7	0.40924	6.90	5.55	+1.35	+0.08006
VI	217.53	12	215.6	0.40924	7.13	9.13	-2.00	-0.11480
VII	208.06	14	208.2	0.40924	7.39	5.32	+2.07	+0.11460
VIII	198.17	17	197.1	0.40924	7.68	8.89	-1.21	-0.06447
IX	187.82	20	186.0	0.40924	8.01	8.76	-0.75	-0.03831
X	176.96	22	178.6	0.40924	8.39	4.93	+3.46	+0.16870
XI	165.51	26	163.8	0.40924	8.85	12.20	-3.35	-0.15490
XII	153.37	29	152.7	0.40924	9.39	8.36	+1.03	+0.04489
XIII	140.41	32	141.6	0.40924	10.09	8.22	+1.87	+0.07584
XIV	126.36	36	126.8	0.40924	11.02	11.77	-0.75	-0.02785
XV	112.00	40	112.0	0.40924	12.41	12.85	-0.44	-0.01451
Total				6.1386				(0.1% of Total VTE Area)

6. The plant cost is quite insensitive to adjustments in VTE evaporating temperatures as a means of matching feed temperatures. For the particular case studied, a change in cost of less than 0.1% of the plant cost resulted.

7. In general, this system exhibits a considerable degree of stability with respect to cost changes resulting from rather large deviations of process variables from design. The designer, thus, might be given considerable freedom in making deviations in the areas encompassed by this study, with the expectations of possibly ingenious cost saving features resulting.

3. DESIGN OF PLANT COMPONENTS AND DETERMINATION OF DESIGN VALUES

Most components used in this design are commercial items or have been the subject of previous OSW-sponsored design reports. The components that are peculiar to this design and that represent an appreciable investment or design uncertainty were investigated in some depth independently of the overall shell and process studies. The outstanding item in this category is the vertical tube bundle with its many subcomponents and questions of performance, strength, and cost. On the basis of the meager information available, the hook and vane entrainment separator appeared quite early in the study to be attractive in performance, cost, and especially physical compatibility. A considerable effort was required to obtain design estimates applicable to evaporator conditions; additional development appears to be needed to achieve the desired low cost design. The deaerator design based on information from the Freeport deaerator test program, strives to utilize the process reject energy to achieve maximum deaeration at minimum cost.

3.1 Tube Bundle Design

The vertical tube evaporator tube bundles represent about 30% of the total capital cost and as such are the largest single item of cost in the evaporator plant. The bundles are composed of sheets, tubes, sheet corrosion protection, and nozzles or brine distribution devices. Their fabrication requires the operations and costs of drilling the sheets, installing the tubes, sealing the tubes, installing the protective covering, and installing the nozzles. The design requires solution to the problems of heat transfer coefficients, brine distribution, and noncondensable removal; of bending, shear, and rolling stresses in the sheets; and of supporting and sealing the bundle units in the shell structure. The bundle is therefore the subject of a component study intended to determine the materials, methods, and measurements that will provide a satisfactory, low cost bundle.

Although there were a number of shell configurations studied, rectangular and semicircular tube bundles have been the only variation in tube bundle shape. For the purpose of costing and design the rectangular bundle is considered in this study.

The preliminary investigations of Phase I provided the following guide lines for this study:

1. The tubes will be 3-in. OD, 10-ft long, General Electric profile, Olin No. 194 copper alloy.
2. The bundles will be 10-ft deep in the direction of steam flow.
3. The tube spacing shall be calculated to give a pressure loss ranging from 0.25°F for effects above 220°F to 1.50°F in the last effect.
4. The unit bundle will be 20-ft wide.
5. The bundles will be fabricated in a shop, possibly on-site, and installed in the shell structure.
6. The tubing is considered to have a 15-year life. Retubing will be accomplished in the same fashion and at the same cost as the initial tubing.
7. The tube sheets will be carbon steel with corrosion protection provided only in the brine chest area (top of the top tube sheet).
8. The study is based on flowsheet Case 261-C (Fig. 32). Necessary modifications will be made to suit the final flowsheet.

The cost summary of the assembled tube bundles has been shown on a "per tube" basis. This presents a truer picture of the unit price with varying tube spacing than other methods, since the greater portion of the total expense is in tube cost. The bundle cost, based on the above criteria, with ceramic flow nozzles which also act as top tube sheet protection, polypropylene baffles, and including the contractor's overhead and profit, is \$20.91 per tube.

Fabrication of these bundles is based on the use of automatic equipment; and the size of the bundles has been held within rail shipment limitations so that assembly could be accomplished at plants remote from the evaporator site. Final "in place" costs will be higher since the expenses of packaging for shipment, transportation, erection, etc., must be added.

3.1.1 Heat Transfer Coefficients

Philosophy Underlying the Choice of Heat Transfer Coefficients. In the 250-Mgd MSF design,¹ the condenser bundle design was based on an evaluation of the Point Loma performance with no safety factors. Similarly, in the present design it was desired to derive heat transfer rates from the General Electric 37,000-gpd pilot plant,⁹ with confirmation by testing in ORNL facilities. No additional factors are provided for fouling.

Operating Principles and Heat Transfer Coefficients. The basic mechanisms by which thin-films operate to give enhanced heat transfer performance are described by Carnavos¹⁰ as follows:

Fig. 32. VTE Conceptual Design; VTE Process Flowsheet
250-Mgd Capacity; Case No. 261-C

"Tube orientation is vertical, with steam condensing on the outside, and a falling film of feed on the inside. Figure 33 is a pictorial representation of the tube and liquid films.

Fig. 33. Thin-Film Double-Flute Heat Transfer

The operating principle of the double flute is such that the condensate film which forms has surface tension forces acting to drain it from the crests into the grooves. This results in the major portion of each crest having a very thin film of condensate, which greatly reduces the resistance to heat flow through the crest area. The condensate in the grooves is channeled off by gravity with the heat flow through this area somewhat less. The falling film of feed on the boiling side also tends to drain into the grooves by surface tension forces. Boiling takes place in the thin film on each crest as well as in the grooves. The crests are kept constantly wet by the splash from the boiling action in the grooves. Figure 33 shows this boiling action during an experiment which was designed to observe this phenomenon. The vapor which is generated travels concurrently with the feed. The flutes perform a secondary but very important function in organizing and controlling the falling film to assure uniform distribution down the length of the tube."

Carnavos also presents two plots which show how the condensing coefficients and evaporating coefficients vary with ΔT or heat flux. These are shown in Figs. 34 and 35, which are presented here to contrast the effect of ΔT or heat flux on the condensing and boiling coefficients. The condensing coefficients are seen to decrease as heat flux increases, from nearly 20,000 Btu/hr.ft².°F to about 8000 as heat flux in a 12-in.-long tube increases from about 17,000 Btu/hr.ft².°F to about 45,000, while the boiling coefficient increases from about 3000 to 7000 Btu/hr.ft².°F as heat Flux varies

Fig. 34. Fluted Tube Condensing Heat Transfer

Fig. 35. Fluted Tube Boiling Heat Transfer

from about 10,000 to 80,000. It can be seen that the boiling film is the major resistance to heat transfer.

These figures are intended to illustrate trends. The absolute values of individual coefficients are a function of the shape of the fluted profile, the type of flow nozzle, the length of the tube and other variables. The understanding of the individual resistances to heat transfer is much inferior to the understanding of the horizontal condenser bundles typical of MSF plants.

Methods by Which Heat Transfer Coefficients were Assigned for the Design Study.

1. Heat transfer coefficients given in the various plots of heat transfer coefficient vs time in the GE pilot plant report⁹ were tabulated along with other pertinent information such as operating conditions, nozzles used and dates.
2. The apparent heat transfer coefficients tabulated in step 1 above were based upon values of ΔT obtained from the difference in saturated vapor temperatures at the observed absolute pressures in the heating steam jacket and in the brine vapor chest below the tube bundles. The area basis was the total development outside surface area, which was given as 1.28 times the "rubber band" area of smooth 3-in.-OD tubes. Use of ΔT defined in this manner gives what is called an "apparent" heat transfer coefficient which is lower than a "true" heat transfer coefficient would be because of thermodynamic inefficiencies such as BPE of the brine and pressure drop of the vapor on both sides of the tube bundle which make the true ΔT less than that value based on pressure differences. In order to place all heat transfer coefficients obtained with different ΔT 's, temperatures and brine concentrations on a basis that would be closer to a "true" ΔT , corrections were made for BPE, by multiplying the "apparent" coefficient by the ratio $\Delta T/(\Delta T - \text{BPE})$. Values for BPE were obtained at the estimated average brine concentration and temperature within the tubes using the OSW chart for BPE.¹¹ This gives an apparent coefficient corrected for BPE that still contains thermodynamic inefficiencies due to pressure drop.
3. All of the BPE-corrected heat transfer coefficients were then plotted as a function of evaporator temperature using symbols and notes which permit ready identification of tube length, ΔT , nozzle type and concentration. This plot is shown in Fig. 36.
4. By means of cross-plots through these curves, a table of values of heat transfer coefficients at various temperatures and concentrations was established for 5-ft copper tubes.
5. All of the data published by GE indicate that the heat transfer coefficients in 10-ft-long tubes are lower than those for 5-ft tubes, presumably because of pressure drop on the inside of the tube and because of the thicker brine films occasioned by higher feed rates in the longer tubes. A temperature-dependent relationship was given in the GE conceptual design report^c for predicting heat transfer coefficients in 10-ft tubes from "known"

Fig. 36. Heat Transfer Coefficients; 3-in.-OD Double-Fluted Copper Tubes GE Pilot Plant Data at Wrightsville Beach, Reported in GE Report 65APE-38 and OSW-181

values for 5-ft tubes. This relationship was applied to the table of values derived in step 4 above for 5-ft bundles.

6. Heat transfer coefficients obtained in step 5 were for 10-ft-long copper tubes. A similar set of values for 10-ft-long tubes made of Olin alloy was obtained by adding the extra resistance of Olin alloy over that of copper to each of the values in step 5. It was assumed that the extra resistance in going from pure copper to Olin alloy amounted to the difference in thermal resistance between a flat copper plate and an Olin alloy plate each of 0.062-in. thickness, which is the assumed thickness of the smooth tubes from which the fluted tubes are fabricated.

7. It was further assumed that for design purposes, heat transfer coefficients would be independent of ΔT , so that the table of values described in step 6 above would be used with no correction of ΔT 's different from 10° . The length correction factor is assumed to have accounted for the higher brine flow rates required for the longer tubes.

8. Application of the temperature-dependent length correction factor to the curve for 5-ft tubes operated once-through gives a similar curve of lower values for 10-ft lengths.

9. Use of the table of values described in step 6 at the operating conditions of temperature and concentration in each of the 15 effects of the reference design flowsheet gave the values for heat transfer coefficient that were used to calculate required heat transfer areas. These values are shown in Fig. 37. It will be noticed that the overall coefficients of the reference design are all less than the coefficients reported for the 10-ft bundle of the pilot plant, with the exception of the pilot plant result at 250°F .

10. Heat transfer tests were carried out in a 10-ft-long double-fluted copper tube in an ORNL test loop over the temperature range of the conceptual design. The results, plotted in Fig. 37, fall generally above the design values.

Major Uncertainties. Although the heat transfer relationships used in the conceptual design appear to be reasonable, more experience and testing are required before a plant such as the 250-Mgd Conceptual Design should be built. The major uncertainties are believed to be the following:

1. Can the heat transfer coefficients be maintained without diminution for indefinitely long period of operation, which means for thousands of hours rather than for the maximum periods of 500 hours that were demonstrated in three pilot plant runs?
2. Can a 10-ft-long double-fluted tube operating with a ΔT of about 7°F at 260°F and 2 to 4 gpm feed per 3-in. tube be operated free of anhydrite scale?

Fig. 37. Effect of Temperature on Overall Heat Transfer Coefficient for Various Tubes

3. In a tube bundle being fed steam in cross-flow, that is, with the steam progressing horizontally through the vertical tube bundle, are high velocities of heating steam deleterious to the condensing heat transfer coefficients of double-fluted tubes?
4. Tests should be run on tubing of the same alloy used in the design, since there are no good correlations for adjusting for wall conductivity.
5. Many of the experimental results fall well above the design correlation. It may be possible to refine the flute profile, nozzle, feed flow, and ΔT as a function of feed concentration and effect temperature to improve performance further.

Work which proves that high performance can be maintained over long periods should precede attempts to develop even higher performance.

3.1.2 Friction Factor Measurements on Tube Banks Composed of 3-in.-OD Smooth and Fluted Tubes on Triangular Pitches of 1.15 and 1.75

Available literature does not contain friction factor data for flow across banks of 3-in. tubes nor any indication of the relative drop across fluted and smooth tubes. Measurements were made to obtain this information and the results are summarized here.

Tube Bank Configuration. There were four cases tested. The first with new, smooth aluminum tubes and the second with used double-fluted tubes spaced on a triangular pitch of $S/D = 1.15$. The third and fourth were again the smooth and fluted tubes respectively spaced on a triangular pitch of $S/D = 1.75$. The tubes were arranged eight rows wide and ten rows deep in a box with a frontal opening of 46-in. high and 27-5/8-in. or 42-in. wide for the 1.15 and 1.75 pitches respectively.

Flow Distortions. During the first tests on smooth tubes having an S/D of 1.15, it was noticed that the flow field downstream of the bundle was badly distorted and had an unexpected velocity profile. The tests of the tube bundle with an S/D of 1.75 were more normal or less surprising. Since this phenomenon might occur in any heat exchanger having a similar geometry this effect appeared to be important and was investigated further. A small water table and elementary flow visualization techniques were used to explore the phenomenon. Preliminary work indicates that variations in tube to tube spacing can cause stall regions which can account for effect observed. It also appears that orienting the triangular pitch array with the flow parallel to one side will produce a more uniform and desirable flow pattern than that with the flow perpendicular to one side of the triangles.

Friction Factors. The results of the friction factor tests are shown in Fig. 38. Referring to Fig. 38, there are three important contributions that can be cited for this work:

1. There was found to be no significant difference between the friction factors of the smooth and fluted tubes.
2. The smooth tube data for the larger spacing agree reasonably well with the data reported in the literature.¹²

Fig. 38. Friction Factor as a Function of Reynolds Number for Smooth and Fluted Tubes

3. The grimson correlation has been extended from an $S/D = 1.25$ to a more densely packed $S/D = 1.15$.

Also of interest are the shapes of the fluted tube curves. For each spacing for fluted tubes the friction factors tended to level off or rise at high Reynolds numbers. This characteristic has been observed with single cylinders in cross flow. While of interest and use in development work, this behavior is not of immediate concern for the conceptual design because the design Reynolds numbers are below 40,000.

Application to Design. As a result of these tests the Grimson correlation is used for estimating losses inside tube bundles for both fluted and smooth tubes and the tubes are oriented in a triangular spacing with flow parallel to a side of the characteristic triangle.

3.1.3 Tube Sheet Stress Analysis

An analysis of the tube bundle has been made to determine maximum tube stress, and tube sheet deflection and stresses. Figure 39 shows the loads applied to tube sheet and tubes. Deflection of the tube sheet from the pressure and weight causes the bending of the tube which in turn applies a resisting moment to the tube sheet. The analysis is based on the principle of superposition of stresses. The first step was to treat the tube sheet as a simple beam loaded only by weight and pressure. A deflection from this loading was computed. The actual tube sheet deflection depends on the resisting moment from each tube so that a trial and error solution was necessary. To simplify the problem, the deflection of the tube sheet was assumed to be a sine curve, the only unknown being the maximum deflection. To complete the analysis, an iterative solution is necessary. An actual tube sheet deflection was assumed from which all the tube moments could be computed. The upward deflection from each tube moment acting alone on the tube sheet was computed. These deflections were then summed and subtracted from the previously computed deflection due to weight and pressure alone, the result being a computed value for tube sheet deflection. This computed deflection is then compared with the assumed deflection. If the two do not agree within a desired accuracy, a new deflection value is assumed and the process is repeated until the desired accuracy is obtained. Once the maximum tube sheet deflection is thus obtained, the tube moments are known on the basis of the assumed sine curve deflection and tube stresses can be computed.

A computer program was written to carry out the computations. The program was written with a large number of input variables to give greater efficiency in use of the program for plant designs.

A typical tube bundle from the rectilinear plant was analyzed using the computer program.

The analytical bundle has forty rows of 3-in.-OD tubes on a triangular spacing. The average center to center tube spacing is 3.56 in. ($S/D = 1.18$). Cases were run for tube sheets from one inch to two inches in thickness.

Fig. 39. Typical Tube Bundle

For the cases considered, the maximum deflection and maximum tube stress did not vary with tube sheet thickness. A complete analysis of the complex tube sheet stresses from the applied loads was not made, but it appears that the minimum tube sheet thickness consistent with good rolling practice is the limitation rather than tube sheet stress. This thickness was determined to be 1-1/2 in. from extrapolated minimum thicknesses recommended by the Tubular Exchanger Manufacturers Association.

Shear and bending stresses in the ligaments between tubes were checked and found to be well below the maximum allowable stress for all tube sheet thicknesses considered. Tube sheet shear stress is approximately 2120 psi under the worst loading conditions. Under maximum loading conditions tube bearing stress (tube to tube sheet) is calculated as being 442 psi.

3.1.4 Bundle Mechanical Design

A typical tube bundle has approximately 40 rows of 3-in.-OD tubes on a triangular spacing. The free span of the tube bundle is 10 ft, 3 in. with 1/2 in. allowed between the outer tube and evaporator structure mounting flange. The tube sheet is 1-1/2-in. thick. The bundle is supported by the upper tube sheet along the edge normal to the vapor flow. The bottom tube sheet has a guide to allow vertical movement only and a seal to prevent vapor and condensate leakage. (See Fig. 40)

The tube bundles are shop fabricated in sections approximately 20-ft wide and are installed by sections during construction of the evaporator. Retubing of entire bundles will also be done in a shop but replacing a small number of bad tubes may be done in place. Bad tubes would be removed without dismounting the tube sheet and new tubes would be inserted and rolled-in by hand.

3.1.5 Tube Sheet Layout

The initial calculations of tube spacing (Sect. 1.2.1) were performed for the conditions of the preliminary flowsheet and with inadequate friction factor information. Since that time the final flowsheet (Case 311A, Fig. 44) has been established and pressure drop measurements of air flow over banks of 3-in. diameter fluted tubes on a triangular pitch have been made (Section 3.1.3). The bundle for the final design is rectangular with a folded noncondensable cooler and removal section as shown in Fig. 41. Steam velocity is maintained by varying spacing rather than by laning. Between 5 and 10% of the steam entering the main bundle exits from its rear face and is redirected by appropriate baffles to the cooler bundle. Between 1 and 1.5% of the steam along with the noncondensable is removed from the cooler section.

Table 24 gives some design values for effects II, VIII, and XV. Tables 25, 26 and 27 are detailed, row-by-row tabulations of row width, tube spacing, and number of tubes per row for these same effects. The calculations were made with a mathematical model permitting constant varying spacing and

CONNECTING DETAILS AT ENDS
OF 20' LONG BUNDLES

NO. BY APPD. DATE				DESCRIPTION				CIVIL	MECH.	ELEC.	INST.	MACH.
REFERENCES:				REVISIONS				SQUAD CHECK				
SCALE: 1" = 1'-0" UNLESS NOTED				DATE								
DESIGN BY: J. V. GREGORY								UNION CARBIDE CORPORATION				
DRAWN BY: R. E. FEHLING								NUCLEAR DIVISION				
CHECKED BY:								ONE RIDGE GASTRO DIFFUSION PLANT				
DEPT.								250 MGD VTE				
OPER.								TUBE BUNDLE				
DIV.												
SAFETY												
E.S.O. NO.								BLDG. NO.				
CHANGE								LOCATION				
R.M. NO.								D- REV.				

Fig. 40. 250-Mgd VTE Tube Bundle

fractional tubes. In practice, each main bundle will be divided into not more than three zones each running the width of the bundle. Each zone will use one of the five spacings shown in Table 28.

Table 24.

	Effect Number					
	II		VIII		XV	
	Main	Cooler	Main	Cooler	Main	Cooler
Steam temp., °F	258.84	258.84	204.08	204.98	119.52	119.52
Bundle depth, ft	10.25	7.83	10.36	7.65	10.30	6.86
Bundle height, ft	10.0	10.0	10.0	10.0	10.0	10.0
Face velocity, fps	5.7	3.2	13.2	6.1	54.7	20.1
Temp. loss, °F	0.20	0.05	0.50	0.06	1.26	0.14
∅ F Rows	41	30	40	26	31	17
S/D, 1st row	1.15	1.33	1.30	1.61	1.61	2.60
S/D, Row N	1.15	1.17	1.15	1.27	1.46	1.44
N	1	29	23	23	29	17
Tube density, #/ft ²	13.9	11.7	12.9	8.9	7.9	4.2
Steam uncondensed, %	8.3	1.6	7.5	1.4	4.7	0.9

Fig. 41. Main Bundle and Noncondensable Cooler

Table 25. Bundle Layout for Effect II

Main Bundle				Noncondensable Cooler			
Row No.	Bundle Width (ft)	S/D	Tubes per Row	Row No.	Bundle Width (ft)	S/D	Tubes per Row
1	19.95	1.150	69.4	1	3.93	1.437	10.9
3	19.76	1.150	68.7	3	3.70	1.354	10.9
5	19.58	1.150	68.1	5	3.49	1.307	10.7
7	19.39	1.150	67.4	7	3.28	1.294	10.1
9	19.21	1.150	66.8	9	3.07	1.281	9.6
11	19.02	1.150	66.2	11	2.87	1.268	9.0
13	18.84	1.150	65.5	13	2.66	1.256	8.5
15	18.65	1.150	64.9	15	2.46	1.244	7.9
17	18.47	1.150	64.2	17	2.26	1.233	7.3
19	18.28	1.150	63.6	19	2.06	1.224	6.7
21	18.10	1.150	62.9	21	1.87	1.215	6.2
23	17.91	1.150	62.3	23	1.67	1.208	5.5
25	17.73	1.150	61.7	25	1.48	1.204	4.9
27	17.54	1.150	61.0	27	1.28	1.204	4.3
29	17.36	1.150	60.4	28	1.19	1.206	3.9
31	17.17	1.150	59.7	Cooler Total			222.0*
33	16.99	1.150	59.1				
35	16.80	1.150	58.4				
37	16.62	1.150	57.8	Total number of tubes			2803.0
39	16.43	1.150	57.2	Bundle Width			20.18
41	16.25	1.150	56.5	Bundle Depth			10.25
Main Tube Bundle Total			2581.0*	Overall Tube Density			13.55

* Including rows not tabulated above.

Table 26. Bundle Layout for Effect VIII

Main Bundle				Noncondensable Cooler			
Row No.	Width of Row (ft)	S/D	Tubes per Row	Row No.	Width of Row (ft)	S/D	Tubes per Row
1	19.94	1.297	61.5	1	3.92	1.590	9.9
3	19.73	1.283	61.5	3	3.67	1.488	9.9
5	19.52	1.270	61.5	5	3.44	1.403	9.8
7	19.32	1.257	61.5	7	3.21	1.383	9.3
9	19.12	1.244	61.5	9	2.99	1.364	8.8
11	18.92	1.231	61.5	11	2.77	1.345	8.2
13	18.72	1.218	61.5	13	2.56	1.327	7.7
15	18.53	1.205	61.5	15	2.34	1.310	7.2
17	18.33	1.193	61.5	17	2.14	1.294	6.6
19	18.14	1.180	61.5	19	1.93	1.279	6.0
21	17.95	1.168	61.5	21	1.72	1.267	5.4
23	17.77	1.156	61.5	23	1.52	1.251	4.8
25	17.58	1.150	61.2	25	1.32	1.251	4.2
27	17.40	1.150	60.5	26	1.22	1.251	3.9
29	17.21	1.150	59.9	Cooler Total			193.0*
31	17.03	1.150	59.2				
33	16.84	1.150	58.6				
35	16.66	1.150	57.9	Total Number of Tubes 2609.0			
37	16.47	1.150	57.3	Bundle Width 20.11			
39	16.29	1.150	56.6	Bundle Depth 10.36			
40	16.19	1.150	56.3	Tube Density 12.52			
Main Bundle Total			2416.0*				

* Including rows not tabulated above.

Table 27. Bundle Layout for Effect XV

Main Bundle				Noncondensable Cooler			
Row No.	Width of Row (ft)	S/D	Tubes per Row	Row No.	Width of Row (ft)	S/D	Tubes per Row
1	19.92	1.787	44.6	1	3.90	2.117	7.4
3	19.63	1.762	44.6	3	3.57	1.935	7.4
5	19.35	1.736	44.6	5	3.26	1.771	7.4
7	19.07	1.711	44.6	7	2.98	1.663	7.2
9	18.79	1.687	44.6	9	2.72	1.616	6.7
11	18.52	1.663	44.6	11	2.46	1.571	6.3
13	18.26	1.639	44.6	13	2.21	1.526	5.8
15	18.00	1.615	44.6	15	1.97	1.483	5.3
17	17.74	1.592	44.6	17	1.73	1.441	4.8
19	17.48	1.569	44.6	19	1.50	1.402	4.3
21	17.23	1.546	44.6	20	1.39	1.384	4.0
23	16.98	1.524	44.6	Cooler Total			123.0
25	16.74	1.502	44.6	Total number of tubes			2803.0
27	16.50	1.481	44.6	Bundle Width			20.18
29	16.26	1.459	44.6	Bundle Depth			10.25
30	16.14	1.449	44.6	Overall Tube Density			13.55
Main Bundle Total			1337.0*				

* Including rows not tabulated above.

Table 28. Standard S/D Spacings
and Hole Distribution for Vertical
Tube Bundles

Tube Spacing, S/D*	Number of Holes
1.15	391,000
1.19	122,000
1.31	122,000
1.49	122,000
1.73	122,000

* There will be an average of three different tube spacing zones in each 20-ft long x 10-ft deep tube bundle.

3.1.6 Tube Sheet Drilling

Estimates of tube sheet drilling costs have been made for several earlier conceptual designs^{1,13}, however, the requirements of this design departed sufficiently from the previous to warrant reinvestigation of machining processes and equipment.

The chosen process is that of bore-drilling which trepans holes greater than one inch diameter and thus removes the bulk of the metal as a core rather than as chips. The bore-drill tools and coolant systems will be bounted on and driven by a gantry type machine which is available from a major manufacturer. This machine is an integral unit capable of supporting up to eight 50-hp heads, servo-driven equipment, hydraulic power unit and numerical controls, operating from a table 12-ft wide and 76-ft long.

The cost estimate is based on the capability to drill the five different tube spacings which were shown in Table 28. The elements of cost making up the total estimate for drilling tube sheet holes are given in Table 29.

Tube Sheet Drilling Cost Factors:

1. Tube sheets: 396; 1.5-in. thick; 10 x 20 ft; mild steel
2. Holes: 2220 Avg/Sheet; 3-in. diam; total 879,000
3. Drilling depth: 3-in. ($396 \times 1.5/3 = 198$ stacks)
4. Tolerance on diameter; +0.005, -0.000 in.
5. Labor, overhead and profit; \$10/hr (contracted, off-site - 2-shift operation)

6. Tooling: Gantry-type machine, 5 heads required for 5 S/D changes, 8 spindles per head, 40 total, 50-hp drive heads, 12-ft-wide table, numerically controlled gantry and heads. Reverse coolant flow pressure head and multiple spindle units adapted. Bore-drill tools with carbide inserts used for trepanning.

Amortization includes payment for the job duration of 6 months plus interest for that period based on a 15-year write-off. The special drilling heads and associated special equipment were assumed to be fully amortized on this one desalting plant job.

Table 29. Tube Sheet Drilling Cost Estimate

Drill Inserts	\$ 21,975
Tool Grinding	1,428
Handling and Setup	10,500
Drilling	41,780
Deburring	5,280
Special Tooling Amortization	50,000
Partial Amortization of Standard Equipment	11,588
Scrap Allowance	2,000
Contingency (@ 15% of other costs)	<u>21,383</u>
Total Drilling Cost	\$165,933
Drilling cost per hole	<u>\$0.1887</u>

3.1.7 Tube to Tube Sheet Joint

For assembly of the tubes to the carbon steel tube sheets, two methods of fastening have been considered, brazing and mechanically expanding (rolling-in) the tubes.

Brazing. Manufacturers of furnace brazing equipment agree that it is practical to braze tubes to tube sheets. However, the furnace must be very large to accommodate the size tube bundle required, and would cost about \$150,000. The cycle time is long, about three hours. A luminous-wall type brazing furnace to contain one end of the tube bundle at a time would cost about \$40,000 and would have only a 15-min cycle time. High frequency induction brazing using three coils should be effective for brazing tubes to tube sheets without requiring a large furnace. The equipment would braze three tubes per minute with manual operation. The three-coil system was estimated at \$40,000 with considerable additional cost for programming the equipment for automatic operation, the exact amount of which the manufacturers did not estimate. The fitting problems for satisfactory brazing are very difficult. The outside diameter tolerance for 3-in.-OD copper alloy tube according to ASTM is ± 0.005 in., but satisfactory brazing requires maintaining a joint clearance of 0.002 in. Considering the variations in

tube diameter plus the tube sheet hole inaccuracies of drilled holes, brazing appears impractical.

Rolling. Joining the tubes to tube sheets will be very low in cost and may be accomplished with programmed machines. Tube rolling is the most common procedure presently used in tube bundle assembly since it produces leak tight joints with very satisfactory structural properties. There have been no complete "off the shelf" tube rolling machines located; however, by purchasing standard components and designing a few special components, an adequate machine can be built. Several manufacturers build automatic rolling heads. These heads generally consist of a drive motor, power feed and retract, the rolling tool and a control unit. The published cycle time for these heads varies somewhat, but is in the range of 5 seconds per tube. Also available from various manufacturers are two-axis tape control and drive units. The control function is capable of controlling rolling heads on both ends of the tube bundle simultaneously, provided the support structure and drive systems are properly designed. The cost of a tape control and drive units varies considerably, depending on speed of operation, accuracy required, control area, machine inertia and other factors, with prices starting at about \$25,000. The indexing or cycle time for this equipment varies according to the equipment itself and the machine being controlled but can reasonably be expected to be in the order of 5 seconds. The support part of the machine must be specially designed and built to cover the area of the tube sheet. This support would consist of a set of vertical rails, a set of horizontal rails, and a frame. The rolling heads would be supported on the horizontal rails, which would be supported by the vertical rails which, in turn, would be supported by the frame. The cost of this equipment would depend on the accuracy required, the total number of rolling heads, etc. An automatic tube rolling machine consisting of two rolling heads at each end of the tube bundle would be capable of rolling approximately 20 tube ends per minute. It is estimated that this machine would cost approximately \$142,000 including engineering.

Cost Factors for Joining Tubes to Tube Sheets:

1. Tubes: 0.049 wall x 3-in. OD x 10-ft 3-in. long 439,500 total number of tubes
2. Tube bundles: 198
3. Labor, overhead and profit: \$10/hr (contracted off-site)
4. Tooling: Automatic rolling heads, tape controlled, two each end of tube for simultaneous rolling, speed 10 tubes per minute
5. Equipment cost: \$126,000 plus 13% of equipment cost for special engineering
6. Tooling Cost: Material at \$25 per bundle; labor at 1 manhour per bundle

Table 30. Tube Installation and Rolling Cost Estimate

Handling and setup	\$ 14,850
Tube installation	183,125
Tube rolling	8,620
Tooling	6,930
Engineering (@ 13% of machine tool cost)	16,380
Special equipment amortization	86,000
Partial amortization of standard equipment	96,293
Contingency (@ 15% of other costs)	48,930
Total tube installation and rolling cost	\$375,128
Installation and Rolling cost per tube	\$0.8535

Explosive Expansion. Investigation was also made of explosively securing tubes into tube sheets. Although this method is not presently used industrially, chiefly because it is not competitive in cost, it produces very satisfactory joining. The cost of the detonating cap is more than the entire cost of rolling a tube in place. An advantage of this process is that it may be possible to secure serrated tubes into the tube sheet holes, but considerable development effort would be required.

3.1.8 Seal Welding

In the event that a copper alloy tube sheet protection is required, methods and costs of welding the tubes to the cladding were studied. An adequate seal by hand welding is a slow and expensive process and could not compete, either in cost or in workmanship, with automatic attachment methods. Contacts have been made with manufacturers of welding equipment and a numerically controlled system for fusion welding tubes to tube sheets is available for approximately \$150,000. Speed of this machine for 3-in.-OD tubes is five minutes per tube. The labor involved makes the cost of this welding method extremely high. A semiautomatic welder by Revere Copper and Brass Company, which requires hand positioning of the welder, but automatically completes the weld cycle in approximately five minutes, appears to be the best of the welding methods. Costs of each welding gun is \$5,500, with operator labor time being the same as with the automatic system. A cost breakdown of using this welding system for attaching a protective sheet to the tubes for all effects of an evaporator designed for flowsheet No. 261-C follows.

Bases for Sealing a Cladding Material to the Tubes:

1. Semiautomatic fusion welding equipment will be used.
2. Five-minute includes hole-to-hole transfer. Two revolutions of gun electrode required: (1) welding, (2) tapering off (decay) current to eliminate cratering and provide trailing shielding gas.

3. Protective sheet: Alloy 194; 0.093-in. thick x 10 ft x 20 ft; 198 total
4. Tube sheet: Steel; 1.5-in. thick x 10 ft x 20 ft
5. Tubes: 0.049-in. wall x 3-in. OD x 10 ft long; 439,500 total
6. Rolling of welded tube end required as well as unwelded end to ensure psotitive contact during fusion welding
7. Labor, overhead and profit @ \$6.50/hr plus 20% (on-site) for tube-to-protective sheet welding; \$10/hr (off-site) for sheet-to-sheet welding before drilling

Table 31. Cost of Welding Alloy 194
Protective Sheet to Tubes and Tube Sheet

Material - Alloy 194	\$174,720
Welding	584,317
Shielding gas, electrodes and brazing rods	25,670
Handling (includes outer seal installation)	9,900
Equipment	33,000
Subtotal	\$652,887
Contingency @ 10%	65,289
Total Welding Cost	\$718,176
Welding Cost per Tube:	
$\$718,176 / 439,500 \text{ tubes} = \1.64 per tube	

3.1.9 Nozzles

In order to assure proper operation of the doubly fluted tube and to prevent dryout with resulting scale formation, it is essential that the feed brine be properly distributed to and within the some 600,000 vertical tubes of the plant. Experience to date with doubly fluted tube indicates that distribution within the tube is best accomplished with a hollow cone spray nozzle. The problem of distributing the brine to the many tubes in a tube sheet measuring ten feet in one dimension and sixty or more feet in the other seems best solved by operating with a completely filled brine chest and a metering orifice in each tube. The penalties for a combined spray nozzle and metering orifice as opposed to the conventional wier in each tube are higher pumping cost and the hazard of corrosion products plugging the orifices. An increase in effect pumping head of 5 psia increases water cost less than 1 mil per 1000 gal. The corrosion product problem is attacked by placing protective coating over all steel surfaces and by the nozzle design.

The proposed unit for this conceptual design consists of a ceramic hollow-cone spray nozzle and a housing as shown in Fig. 42. The purposes of the

Fig. 42. Nozzle Assembly

housing are to support the nozzle and position it within the tube, to raise it several inches above the tube sheet thereby reducing the possibility of plugging, and to provide an impingement zone which protects the entry region of the tube from erosion.

Simple hollow cone spray nozzles, in large quantities, were estimated to cost 50 cents each fabricated of monel, or about 25 cents each of high purity Al_2O_3 ceramic. Housings of cupronickel, molded plastic, and molded ceramic were investigated and estimated. There was little question of the suitability of the cupronickel but the estimated unit cost of about \$6 seemed prohibitive. "Lexan," a moldable polycarbonate was used in the Wrightsville Beach pilot plant and appeared satisfactory to 230°F. Housings of this material or of more recent plastics such as polyphenylene oxide or polysulfone with higher heat deflection points were estimated to cost about \$1.50 each. A nozzle-housing assembly of porcelain ceramic was fabricated in experimental quantities by Knox Porcelain Corporation and appears satisfactory after limited testing. The fabricator estimates this assembly to cost 38¢ in quantity production. This cost is well below any other combination of nozzle and housing; and the glazed porcelain eliminates temperature problems as well as being highly resistant to corrosion and erosion. This assembly has been chosen for the reference design.

3.1.10 Tube Sheet Corrosion Protection

The protective coating for the upper tube sheet presents a more difficult problem than the other steel surfaces in the plant because of its complicated geometry. Two general approaches to protection were considered. In one, the protective coating would be sealed to the tube sheet and tube ends and the nozzle assemblies would be simply inserted in the tubes. In the other, the nozzle housing, or more specifically the flange of the housing, was considered to become a part of the protective coating by welding, cementing, or grouting the flanges to each other or to a protective sheet.

A discussion and cost estimate of the first approach using a copper alloy sheet with welded sheet-to-tube joint is given in section 3.1.9.

Within the second approach several materials were considered. Using plastic nozzle housings with hexagonal flanges would permit the flanges to be welded or cemented to each other or to a match-drilled plastic sheet laid over the steel tube sheet. Equipment is available for fusion or filler rod welding thermoplastics and the manufacturers feel this equipment could be adapted to the semiautomatic methods common for welding tubes to tube sheets. Using porcelain nozzle housings, the flanges could be grouted or cemented to the steel sheet and/or to each other. Possible materials are mono-calcium Saureisen Company, and silicone rubber compounds such as proposed in the General Electric 50-Mgd Conceptual design report. Sections of tube sheets have been coated with these materials and are being tested.

The cost of copper-alloy cladding is such as to offer considerable incentive to develop other methods. Since the porcelain nozzle is so attractive both in cost and serviceability it is desirable to develop a method compatible with it. Therefore the fabricated bundle cost estimate for the final design,

section 3.1.3, is based on the use of the porcelain nozzle cemented to the tube sheet. Further testing is required to determine which of the proposed materials is best suited to the service but the installed cost will be about the same as for the Saureisen cement which was selected only as being representative of the cost.

3.1.11 Tube Fabrication Cost

Costs for the GE profile double fluted tubes (3-in. OD x 0.046-in. wall, 10-1/2 ft long) are based on the use of Olin 194 alloy and are summarized in Tables 32 and 33. These costs provide for a prime contractor to procure the strip material, and to hire a subcontractor for welding the strip into tubes and fluting these tubes in a small shop (low-overhead operation).

Table 32. Tube Fluting Cost Analysis^a
GE Profile, 3-in. OD, 450,000 10.5-ft Tubes

Capital Cost - Equipment	\$182,000
Interest (7% annual rate, 1.25 yrs)	15,925
Fluting Operations	
Start-up ^b	4,293
Operating cost ^b	27,503
Handling, cleaning, testing ^b	82,508
Auxiliaries	5,820
Maintenance	
Tooling	60,750
Machine repair	5,000
Contingency, 10%	<u>38,380</u>
Total Fluting Cost	\$422,179
Cost per Tube, 450,000 Tubes	\$ 0.9382
Cost per Square Foot	\$ 0.0938

^aBased on prime contractor procuring material and subcontracting fluting. Therefore, subcontractor applies overhead and profit to labor only.

^bSouthern California labor rates plus 100% allowance for overhead and profit.

Table 33. Estimated Fluted Tube Cost
Olin Alloy No. 194 - Strip-Welded and Fluted

Material Cost (24.402 lbs/10.5-ft)	\$13.793
Welding Cost ^a	0.683
Fluting Cost (from Table 32)	0.9382
Shipping Cost to Bundle Assembly Plant ^b	0.500
Sub-Total	\$15.914
Scrap Allowance (4-1/2% fluting rejects, 5% welding rejects)	0.808
Total Cost per Fluted Tube	\$16.722
Cost of Fluted Surface, per ft ²	\$ 1.67

^aIncludes subcontractor overhead and profit.

^bTubing from New Jersey to California.

3.1.12 Fabricated Bundle Cost Estimates

The following breakdown of costs are based on the quantities of tubes and tube sheets as required by flowsheet 261-C. Materials and fabrication techniques used for cost purposes are as follows:

Tube Sheet Drilling - Tape controlled multiple spindle, multiple head, gantry supported drill

Tube Sheets - 1-1/2-in. thick carbon steel

Tube Roller - Tape controlled, four heads (a pair for each end of tubes with simultaneous rolling)

Tubes - 3-in.-OD Olin copper alloy No. 194, GE fluted type - 10 ft 4-in. total length

Tube Nozzles and Housing - One piece ceramic

Nozzle Assembly - Cementing between nozzle housings and tube sheets with No. 40 Saureisen Cement

Since there are 439,500 tubes, the assembled tube bundle cost would be \$20.91 per tube. Installation cost of the tube bundles in the evaporators plus overhead and profit must be added for an "in place" figure.

Table 34. Summary of Fabricated Bundle Cost Estimates

Material or Operation	Unit	Number of Units	Unit Costs, \$	
Tube sheet drilling	Each hole	879,000	0.1887	\$ 165,867
Tube sheets	sq ft	85,239	12.2400	1,043,325
Tube installation & rolling	Each tube	439,500	0.8535	375,113
Tubes	Each	439,500	16,722	7,349,319
Tube nozzle & housing	Each	439,500	0.3835	168,548
Tube nozzle cementing	Each	439,500	0.1358	59,684
1/8-in. polypropylene baffles	Each bundle	396	68.36	27,070
				<u>\$9,188,926</u>

3.2 Deaerator Design

A new concept was utilized in developing the design for the deaerator. The design incorporates both the flash-cocurrent and the stripping-counter-current principles. The deaerator neither uses steam from an external source nor rejects any steam for external condensation. The latter feature results in considerable savings in equipment cost due to elimination of expensive ducts and a vent condenser. In addition, the deaerator is designed to remove also the necessary amount of CO_2 thus eliminating use of a separate decarbonator. A flowsheet for the deaerator is shown in Fig. 43.

The deaerator is composed of two sections, a hot side and a cold side. In the hot side an acidulated portion of the plant heat reject water is flashed in two consecutive stages, each stage having a packed bed to provide ample residence time and free surface. A water seal between the beds separates the top stage from the bottom. The flashed steam from each stage goes to the cold side of the deaerator and the resulting fully deaerated water falls into the reservoir below. The cold side of the deaerator uses cold acidulated seawater. It also has two packing beds, one above the other. Steam from the hot side enters below each bed of packing and flows upward counter-current to the falling water. The stripping action of the steam plus the reduced pressure is expected to accomplish the required deaeration. Since the water is considerably colder than the steam and since the quantities of each can be regulated, most of the steam can be condensed by the time it reaches the top of the deaerator. Vent gases containing only the minimum water vapor go to the evacuation system.

Fig. 43. Deaerator and Evacuation System for One Train (62-1/2 Mgd) of VTE Evaporator

Both the cold side and the hot side design approach are based on ORNL's extensive tests on a packed column deaerator at the OSW's Freeport Desalting Plant. The Freeport deaerator contained the same types of packing and spray nozzle as recommended for the subject deaerator. The tests indicated that condensing of the stripping steam on the water did not have a detrimental effect on the deaeration efficiency of a countercurrent packed deaerator. The tests also demonstrated that a considerable amount of deaeration can be accomplished in a good spray nozzle, which reduces the height of the packing bed required for such deaerators.

The exact behavior of a process, like the one under consideration, cannot be very accurately anticipated by the presently available theoretical means. Therefore, a feature of this design is the incorporation of an internal damper which will help obtain the optimum interstage pressure in the hot side under all operating conditions. The control of interstage pressure will also govern the distribution of steam between the upper and the lower beds in the cold side. Also, the flowrates of the hot and cold brine streams can be regulated. Thus, any operational deviation from the design conditions can be easily handled without jeopardizing the deaeration efficiency.

3.3 Entrainment Separators

The design of the entrainment separators requires both estimation of expected entrainment as well as capacity and efficiency of the particular separator. The disengagement chamber geometry and flow conditions affect the final choice of separator. Information developed to date suggests the choice of a "hook and vane" type entrainment separator, which is commercially available. Other mist eliminators considered were wire mesh, packed beds, gravity and the many types of centrifugal separators, which were ruled out because of either lower capacity, higher cost or noncompatible geometry.

Three methods of estimating the expected incident entrainment on the separators were examined. All gave values within the same order of magnitude, although they were quite different in approach. The application of any of the three methods to the particular geometry under consideration is rather speculative without some experimental data for the chamber geometry under consideration. However, since the main source of liquid entrainment is thought to be the down flow of steam and brine from the vertical tube bundles in the VTE section, the above three methods are probably conservative for at least this section of the plant since they were derived for evaporators and boilers from which the vapor and entrained liquid are ejected upward from the boiling liquid surface. For the estimation of separation stages required the method developed by D. S. Ullock¹⁴ was used. Since the entrainment was found to be a strong function of the vapor velocity, the duty of the separators increased with decreasing temperature, pressure and vapor density from the first to the last effect. The difficulty of eliminating liquid droplets is a compounding one, since increasing brine concentration requires more complete entrainment separation in the low-temperature effects where vapor velocities are higher.

The performance of the "hook and vane" separator was examined, and from limited data, a correlation for predicting performance was developed. It must be stressed that the correlation was based on a very limited amount of information on a particular metal hook and vane type and at higher pressures and lower velocities than will be experienced in the plant under consideration. A limited experimental program would seem to be prudent since, because of materials of construction and cost, the suggested hook and vane separators would either be plastic coated or extruded from plastic. It is not known how the difference between metal and plastic surfaces will affect the performance of the separators. Materials of construction have proven to be a problem not yet resolved because of the separator shape, methods of fabrication and the special corrosion resistance requirements dictated by an environment of steam and hot concentrated brine.

The use of 90-10 Cu-Ni alloy was ruled out because of a quoted estimate of \$168 per ft² of face area (not installed). It was at first believed that some high speed roll forming operation could reduce the quoted cost significantly. However, it was found that the cost of materials alone was approximately \$120 per ft² of face thus the use of 90-10 Cu-Ni was excluded.

Carbon steel separators (approximately \$35/ft² uncoated, not installed) were thought to be attractive if they could be coated with a protective surface of plastic or possibly porcelain enamel. At our request, standard commercial separators were sent to a vendor for coating with Pentone, a chlorinated polyether made by Hercules. The vendor was unable to provide a continuous coating as well as keep the hooks from filling with the coating material. It was felt that the same problems would result if porcelain enameling were tried. The Porcelain Institute was contacted and would not recommend the material for this service.

Aluminum vanes are produced at the present time by both manufacturers contacted. A price of \$17/ft² face was quoted for extruded aluminum vanes. Because of its low cost, this material of construction should be seriously considered in the final design. The New Kensington, Pennsylvania research department of Alcoa was contacted for information as to the applicability of aluminum in the environment under consideration. Very limited experimental information is known to have been developed to date for aluminum in an environment of steam-brine with 3-10 wt% NaCl at 120-260°F containing traces of heavy metal ions such as copper. A recommendation of aluminum for extended life in the desalination plant design under consideration cannot be made as yet. However, since some corrosion would be tolerable, almost to the point of mechanical failure, aluminum hook and vanes should not be ruled out at this time.

Extruded plastic separators seem also to offer an economical solution to the problem of materials of construction. Several low cost plastics are under study including glass and asbestos filled polypropylene, polyphenylene oxide, and polysulfone. Estimated materials cost per sq. ft face for the above plastics are \$3.70 for filled polypropylene, \$13.50 for polyphenylene oxide and \$13.80 for polysulfone. Attempts are being made to

extrude these plastics at the present time in the complete or partial shape of the separator. It is believed that the polysulfone would have the best mechanical and degradation resistant properties in the desalination plant environment. The possibilities considered to date are summarized below and are based on 1 sq ft of face area for an eight (8) stage separator:

Table 35. Estimated Cost of Entrainment Separators
Fabricated of Various Materials

Material	Material Cost		Fabricated Cost (\$/ft ² face)	Installed Cost (\$/ft ² face)
	(\$/lb)	(\$/ft ² face)		
90-10 Cu-Ni	1.00	120.0	168	
Carbon steel			35	
Aluminum		4.63	17	24
Polypropylene	0.26	3.67	15	21
Polyphenylene oxide	0.98	13.50		
Polysulfone	1.00	13.80		

It therefore seems reasonable that hook and vane type separator can be developed in the near future at an installed cost of not more than \$30/ft² of face area. The development of a satisfactory unit should take about a year.

A comparison between the proposed hook and vane type and the second most attractive type, wire mesh, provided the basis for the choice of the former. A simplified relationship between the maximum capacities of the two types can be derived from relationships for maximum vapor velocities.

$$\text{For wire mesh } V_{\max} \approx 3.00 \sqrt{v_g}$$

$$\text{For hook and vane } V_{\max} \approx 5.00 \sqrt{v_g}$$

where

$V_{\max}$ = maximum allowable vapor velocity in ft/sec.

v_g = specific volume of the vapor in ft³/lb.

Therefore for the same duty the minimum area required would be inversely proportional to the coefficients in the above relationships, or area of hook and vane required would be about 0.6 of that required for wire mesh demisters.

A previously estimated cost for the 250 Mgd-VTE plant was \$25/ft² for monel wire mesh and a range of values for polypropylene hook and vane from \$19 to \$26/ft² of face installed (depending on number of stages required). The installed cost for mesh was estimated at \$1.8 x 10⁶ and for hook and vane at \$1.2 x 10⁶. Neglected was the increase in plant size for the lower temperature effects due to added space requirements for the mesh demister. Adding to the attractiveness of the hook and vanes is the fact that the pressure drop through the mesh is about 10 times greater than through the hook and vanes under the same conditions of vapor density and collection efficiency.

At this point, the most pertinent assumptions which affect the final cost should be restated.

1. Estimates of incident entrainment for the chamber geometry considered are order of magnitude figures based on relationships developed for standard types of vessels previously used in the process industries.
2. Performance of the separators, both hook and vane and mesh types, have been extrapolated over wide ranges not backed by experimental data.
3. The development of an economical hook and vane separator fabricated from low cost materials can be reasonably expected to take about a year.

Based on the information now available, a hook and vane separator would be the choice for the 250 Mgd plant design.

4. SELECTION

The configuration and values selected for or assigned to the final design come in part from the study ground rules, in part from clearly drawn comparisons in the applicable sections of this report, and in part from compromises or engineering judgement choices between apparent near equals.

The shell is a rectilinear, reinforced concrete structure divided by longitudinal walls into four independent trains; by a horizontal partition into the multistage feed heater on the lower floor and the vertical tube effects on the upper floor; and by transverse walls into 50 stages and 15 effects. To accumulate the increasing vapor flow area required with decreasing temperature each train is trapezoidal in plan view. The amount of surface assigned to each effect was adjusted slightly (away from equal area per effect) to fit the physical linear taper. Forcing a physical match for length and width of the feed heater section and the vertical tube effects causes them to be individually less economical but collectively more economical. The design will withstand full vacuum throughout and meet the unfired pressure vessel code. The brine heater is integral and a continuation of the feed heater. This method was chosen over using the first effect for feed heating because of the uncertainty of the heat transfer coefficients to be expected for the doubly fluted tubes at feed rates of about 8 gpm per tube. The final condenser and deaerator are also integral.

The tube bundles will be constructed in modules 10-ft-deep by 20-ft-wide with doubly-fluted tubes 10-ft long and 3-in. in diameter. Each module has its own noncondensable removal section. Hook and vane entrainment separators will be mounted at the face of the bundles.

It was decided to cool the product to 90° in the flash feed heater. A feed heater performance ratio of 7.8 was computed to be near optimum for this set of circumstances. These two conditions established the feed temperature of 77° . Other methods have been suggested for product cooling which will permit higher feed temperatures. An economic analysis of these will be presented in the final report.

The final effect evaporating temperature of 109° and the final condenser approach temperature of 5° were computed to be near optimum balances of the cost of surface against the cost of coolant.

The optimum concentration ratio appeared to be somewhere between 2.75 and 3.0, however, the curve was quite flat in the region from 2.5 to 3.25. A concentration ratio of 2.5 was selected on the basis that the added cost was small but that the change in the operating curve with respect to the solubility limits was attractive. This parameter will be reinvestigated in the final report.

The process flowsheet (Fig. 44) and computer printout for case 311A are shown on the following pages. The detail design and costing of this case will not only provide a cost estimate for this specific plant but will provide the cost subroutine with sufficiently accurate and detailed information to permit confident extrapolation over a relatively wide range.

Fig. 44. VTE Conceptual Design Process Flowsheet;
250-Mgd Capacity; Case 311A

DESIGN AND COST DATA COMPUTED FOR THE REFERENCE CASE OF A
250-MGD VTE PLANT WITH MSF FEED HEATING

Computer Code Input Specifications for 250-Mgd Plant Design

SEA WATER SALT CONCENTRATION	0.03360	PREHEATER+CONDENSER TUBE ID (IN)	0.6660
PRODUCT SALT CONCENTRATION	0.00002	PREHEATER+CONDENSER TUBE OD (IN)	0.7500
CONCENTRATION RATIO	2.50	PREHEATER+CONDENSER TUBE VELOCITY (FT/SEC)	5.7986
FIRST EFFECT DELTA T TS-TB	7.39	PREHEATER+CONDENSER TUBE FLOODING FACTOR	16.00
APPROACH TEMP-CONDENSER TN-TC	5.00	PREHEATER FOULING FACTOR	0.00030
		CONDENSER FOULING FACTOR	0.00050
OCEAN TEMPERATURE (DEG F)	65.00	VTE PLANT LENGTH (FEET)	0.348000E 03
CONDENSER EXIT BRINE TEMP (DEG F)	104.10	PRODUCT FLOW (LBS/HR)	0.864159E 08
PRODUCT TEMPERATURE (DEG F)	88.40	BLOWDOWN FLOW (LBS/HR)	0.575677E 08
BLOWDOWN TEMPERATURE (DEG F)	92.82	STEAM FLOW VTE (LBS/HR)	0.499999E 07
STEAM TEMPERATURE (DEG F)	267.39	FEED FLOW (LBS/HR)	0.143853E 09
MSF FEED TEMPERATURE (DEG F)	77.00		

NUMBER OF EFFECTS 15.00

VTE Effect No	Effect Characteristics									
	UBAR	WB	CB	TB	HB	T	E	EV		Q
1	1828.0	0.1439E 09	0.03363	260.00	221.97	267.39	0.4972E 07	0.4972E 07		0.4668E 10
2	1782.7	0.1389E 09	0.03484	260.00	221.73	258.63	0.6231E 07	0.4934E 07		0.4668E 10
3	1736.3	0.1327E 09	0.03648	251.06	212.63	249.92	0.6094E 07	0.4903E 07		0.4668E 10
4	1688.7	0.1266E 09	0.03823	242.38	203.82	241.22	0.6003E 07	0.4874E 07		0.4668E 10
5	1639.7	0.1206E 09	0.04014	233.68	195.00	232.49	0.5920E 07	0.4845E 07		0.4668E 10
6	1589.3	0.1146E 09	0.04222	224.89	186.13	223.68	0.5843E 07	0.4816E 07		0.4668E 10
7	1537.1	0.1088E 09	0.04449	215.98	177.16	214.74	0.5772E 07	0.4788E 07		0.4668E 10
8	1482.8	0.1030E 09	0.04699	206.89	168.03	205.61	0.5707E 07	0.4759E 07		0.4668E 10
9	1426.0	0.9731E 08	0.04975	197.55	158.70	196.24	0.5647E 07	0.4730E 07		0.4668E 10
10	1365.8	0.9168E 08	0.05282	187.88	149.08	186.53	0.5594E 07	0.4700E 07		0.4668E 10
11	1301.1	0.8607E 08	0.05626	177.77	139.08	176.38	0.5546E 07	0.4670E 07		0.4668E 10
12	1229.8	0.8052E 08	0.06015	167.09	128.59	165.65	0.5507E 07	0.4638E 07		0.4668E 10
13	1148.3	0.7502E 08	0.06457	155.52	117.41	154.13	0.5482E 07	0.4604E 07		0.4668E 10
14	1049.9	0.6953E 08	0.06968	143.00	105.23	141.45	0.5483E 07	0.4565E 07		0.4668E 10
15	920.0	0.6405E 08	0.07566	128.56	91.47	126.94	0.5474E 07	0.4517E 07		0.4668E 10
		0.5858E 08	0.08260	112.00	75.94	109.10				0.4668E 10

UBAR	#EFFECT UBAR, BTU/HR-FT2-DEG F	WB#BRINE FLOW OUT, LBS/HR	CB#SALT CONC. ,WT. FRACTION(AFTER MIXING)
TB	#BRINE TEMP OF WB, DEG F	HB#BRINE ENTHALPY OF WB,BTU/LB	T #SATURATED VAPOR TEMP, DEG F
E-TOTAL#	EVAPORATION RATE, LBS/HR	EV#VAPOR FLOW TO VTE CHEST, LBS/HR	
		Q #EVAPORATOR HEAT RATE, BTU/HR	

Effect Characteristics

VTE Effect No.	ACTUAL		TUBE		TUBE		Chemical Boiling Point Elevation	Delta T. From Bundle Pressure Losses
	AREA (FT2)	UBAR	LENGTH	NUMBER	DELTA-T *			
1	0.3484136E 06	0.1827955E 04	10.00	0.3484E 05	7.33	1.184592	0.250000	
2	0.3653814E 06	0.1782673E 04	10.00	0.3654E 05	7.17	1.200848	0.339286	
3	0.3815478E 06	0.1736295E 04	10.00	0.3815E 05	7.05	1.219834	0.428571	
4	0.3967386E 06	0.1688682E 04	10.00	0.3967E 05	6.97	1.241806	0.517857	
5	0.4106055E 06	0.1639721E 04	10.00	0.4106E 05	6.94	1.266937	0.607143	
6	0.4229045E 06	0.1589256E 04	10.00	0.4229E 05	6.95	1.295419	0.696429	
7	0.4332872E 06	0.1537064E 04	10.00	0.4333E 05	7.01	1.327527	0.785714	
8	0.4413007E 06	0.1482814E 04	10.00	0.4413E 05	7.14	1.363617	0.875000	
9	0.4462830E 06	0.1425998E 04	10.00	0.4463E 05	7.34	1.404150	0.964286	
10	0.4467708E 06	0.1365765E 04	10.00	0.4468E 05	7.65	1.449816	1.053571	
11	0.4441577E 06	0.1301077E 04	10.00	0.4442E 05	8.08	1.500671	1.142857	
12	0.4348202E 06	0.1229780E 04	10.00	0.4348E 05	8.73	1.558044	1.232143	
13	0.4176783E 06	0.1148329E 04	10.00	0.4177E 05	9.74	1.622439	1.321428	
14	0.3901884E 06	0.1049908E 04	10.00	0.3902E 05	11.40	1.694206	1.410714	
15	0.3484136E 06	0.9200361E 03	10.00	0.3484E 05	14.57	1.769590	1.500000	

*Thermal Driving Force Across Vertical Evaporator Tubes

Effect Characteristics

N	EBF(N)	ECP(N)	OBF(N)	QCF(N)	CI(N)
1	0.	0.	0.	0.	0.336305E-01
2	0.129700E 07	0.112031E 08	0.122566E 10	0.440086E 08	0.351652E-01
3	0.119104E 07	0.172975E 08	0.113244E 10	0.988046E 08	0.368060E-01
4	0.112942E 07	0.233009E 08	0.108040E 10	0.152823E 09	0.385792E-01
5	0.107515E 07	0.292210E 08	0.103456E 10	0.207371E 09	0.405034E-01
6	0.102711E 07	0.350642E 08	0.994203E 09	0.263439E 09	0.426006E-01
7	0.984836E 06	0.408367E 08	0.958955E 09	0.322150E 09	0.448971E-01
8	0.948125E 06	0.465438E 08	0.928744E 09	0.384884E 09	0.474249E-01
9	0.917240E 06	0.521910E 08	0.903965E 09	0.453543E 09	0.502232E-01
10	0.893891E 06	0.577853E 08	0.886469E 09	0.531474E 09	0.533415E-01
11	0.875779E 06	0.633309E 08	0.874146E 09	0.620607E 09	0.568407E-01
12	0.869237E 06	0.688381E 08	0.873563E 09	0.730038E 09	0.608027E-01
13	0.878657E 06	0.743205E 08	0.889774E 09	0.872359E 09	0.653385E-01
14	0.917506E 06	0.798034E 08	0.936629E 09	0.107646E 10	0.706098E-01
15	0.956625E 06	0.852774E 08	0.985754E 09	0.120733E 10	0.768083E-01

Steam Formed From The Flashing Brine At Effect (F), lb/hr	
Steam Formed From Flashing Product at Effect (F), lb/hr	
Heat From Brine Flashdown, Dtu/hr	
Heat Contained in Steam From Flashing Product at Effect (N), Btu/hr	
Concentration Entering Evaporator, weight fraction	

Total heat load of deaerator condenser product, Btu/hr 0.873598E 10

Total heat load of deaerator condenser brine, Btu/hr 0.147835E 11

Heat and Material Balance Calculations

INPUT NO OF EFFECTS 15					
WF	0.14385318E 09	THF	260.000	TCON	104.10271
TD	65.00	CD	0.03363	TP	107.60271
TS	267.3947	WBS	0.49999907E 07	QCON	0.46679236E 10
HPRDD	0.49922381E 10	HBLDW	0.33678165E 10	RHD	3.28889
IR	CJ	C	QBF		
260.0000	0.033630	0.034837	0.		
260.0000	0.035165	0.036476	0.12256552E 10		
251.0588	0.036806	0.038235	0.11324358E 10		
242.3819	0.038579	0.040142	0.10804034E 10		
233.6751	0.040503	0.042219	0.10345609E 10		
224.8908	0.042601	0.044491	0.99420284E 09		
215.9796	0.044897	0.046988	0.95895510E 09		
206.8864	0.047425	0.049750	0.92874413E 09		
197.5468	0.050223	0.052821	0.90396486E 09		
187.8797	0.053342	0.056262	0.88646939E 09		
177.7667	0.056841	0.060146	0.87414638E 09		
167.0903	0.060803	0.064573	0.87356267E 09		
155.6203	0.065338	0.069678	0.88977352E 09		
142.9985	0.070610	0.075660	0.93662862E 09		
128.5571	0.076808	0.082598	0.98575377E 09		
QCF	0.71786917E 10	0.13705257E 11	0.15572891E 10	0.10782092E 10	0.23519447E 11
TCF	0.24652382E 03				
NO OF TUBES IN MSF PLANT 0.47205262E 05					
BRINE HEATER					
LMID	0.12987967E 02	HIH	0.25359076E 04	MOH	0.26105501E 04
UBARH	0.78746239E 03	QBR	0.19058474E 10	AREA	0.18634475E 06
LENGTH	0.20104682E 02	TUBES	0.47205262E 05		
OVERALL HEAT BALANCE					
WF	0.14385318E 09	WCON	0.77563164E 08	HFEED	0.48140268E 10
HCON	0.54848775E 10	HSEA	0.25956406E 10	QCON	0.46679236E 10
QIN	0.13983433E 11	QOUT	0.13844932E 11	QBAL	-0.21919536E-02
TERMINAL CONDENSER					
OVERALL					
Q	0.46679236E 10	UBAR	0.49684944E 03	AREA	0.47191139E 06
TUBES	0.69282667E 05				
FEED STREAM					
Q	0.17786867E 10	AREA	0.10025238E 06	LMTD	0.35709189E 02
LENGTH	0.11339474E 02	TUBES	0.45026918E 05		
REJECT STREAM					
Q	0.28892369E 10	AREA	0.37165901E 06	LMTD	0.15646373E 02

Note: See Next Page
for
Nomenclature

LENGTH 0.78035980E 02 TUBES 0.24255749E 05

DEAERATOR

CO	0.33630466E-01	CNQ	0.33600000E-01	WDE	0.13042891E 06	WFF	0.14398361E 09
WF	0.14385318E 09	DEFR	0.10000000E 01	TDD	0.77000000E 02	TDD	0.78000000E 02
TDEARJ	0.70443973E 02	TSTM	0.75443974E 02	WDECN	0.26500251E 08	UDEAC	0.42348304E 03
OMDCN	0.73908065E 01	QDECN	0.13699240E 09	ADECN	0.43769202E 05		

ADECN - Area of deaerator condenser, ft².

C(I) - Concentration of mixed stream after boiling or concentration stream to MSF chamber, weight fraction.

CI(I) - Concentration entering evaporator, weight fraction.

CØ - Concentration entering tube side MSF reject, weight fraction.

CØØ - Seawater concentration, weight fraction.

IEFR - Flashing range of deaerator flash tank (TØD-TØØ), °F.

HBLØW - Blowdown stream heat rate for the overall plant, Btu/hr.

HCØN - Heat dumped in condenser reject stream, Btu/hr.

HFEED - Heat entering system in feed stream, Btu/hr.

HICØN - Inside film coefficient terminal condenser.

HIH - Inside film coefficient of brine heater.

HØCØN - Outside film coefficient, terminal condenser.

HØH - Outside film coefficient of brine heater.

HPRØD - Heat leaving system in product stream, Btu/hr.

HSEA - Heat entering system in condenser reject stream, Btu/hr.

LMTD - Log mean temperature difference of terminal condenser, °F.

ØMDCN - Log mean temperature difference of deaerator condenser, °F.

QBAL - Heat balance.

QBF(I) - Heat from brine flashdown, Btu/hr.

QBR - Prime steam rate to MSF brine heater, Btu/hr.

QCØN - Heat load of terminal condenser, Btu/hr.

QDECN - Heat load of deaerator condenser, Btu/hr.

QIN - Total heat entering system, Btu/hr.

QØUT - Total heat leaving system, Btu/hr.

QSTEAM - Prime steam heat rate, Btu/hr.

RHØ - MSF stage temperature decrement.

RP - Performance ratio for overall system.

TB(I) - Temperature of boiling brine in evaporator, °F.

TCF - Maximum tube side brine temperature, °F.

TCØN - Heat load on terminal condenser, Btu/hr.

TDEARJ - Total heat load of deaerator condenser in product, Btu/hr.

THF - Maximum brine temperature, °F.

TØØ - Temperature entering MSF reject, tube side, °F.

TØ - Ocean temperature, °F.

TØD - Temperature entering deaerator flash tank, °F.

TP - Product temperature, °F.

TS - Prime steam temperature, °F.

TSTM - Condensing temperature of steam to deaerator condenser, °F.

UDEAC - Overall heat transfer coefficient of deaerator condenser, (Btu/hr)/ft²-°F.

WBS - Prime steam flow to vertical evaporator, lb/hr.

WCØN - Condenser reject flow in lb/hr.

WDE - Steam leaving deaerator flash tank, lb/hr.

WDECN - Coolant water stream to deaerator condenser, lb/hr.

WF - System feed flow, lb/hr.

WFF - Feed flow to deaerator flash tank, lb/hr.

HEAT RATES, BTU/HR

Q, VTE STEAM	0.46679183E 10
Q, MSF STEAM	0.19058474E 10
Q, STEAM TOTAL	0.65737657E 10
Q, FLASHING BRINE	0.14783466E 11
Q, FLASHING PRODUCT	0.87359808E 10
Q, MSF TO VTE	0.53120000E 04

FLOW RATES LBS/HR

TOTAL PRODUCT	0.86415885E 08	
VTE PRODUCT	0.71315490E 08	(PRODUCT FROM BOILING BRINE
MSF PRODUCT	0.14938683E 08	(PRODUCT FROM FLASH-DOWN OF BRINE)
DEAER. PRODUCT	0.13042891E 06	

BLOWDOWN	0.57567724E 08
TOTAL INTAKE	0.25323043E 09
MAKEUP	0.14398361E 09
FEED TO MSF	0.14385318E 09
CONDENSER REJECT	0.77563164E 08
DEAERATOR COOLANT	0.26500251E 08
VENT SYSTEM	0.51834098E 07

PERFORMANCE RATIOS

OVERALL	0.13145568E 02
VTE	0.15277793E 02
MSF, (ACTUAL)	0.78383416E 01
MSF, (SIMILAR CONVENTIONAL PLANT)	0.11659123E 02

Nomenclature for Feed Heater Characteristics
(following page)

ALPHA	- Boiling point elevation MSF stage (chemical + other effects), °F.
AS	- Area of MSF stage, ft^2 .
C1	- Concentration entering MSF stage group associated with an evaporator (shell side), weight fraction.
C2	- Concentration leaving the MSF stage (shell side), weight fraction.
CT	- Tube side concentration, MSF plant.
HI	- Inside film coefficient, $(\text{Btu/hr})/\text{ft}^2 \times ^\circ\text{F}$.
HØ	- Outside film coefficient, $(\text{Btu/hr})/\text{ft}^2 \times ^\circ\text{F}$.
LS	- Stage length of MSF stage, ft.
N	- Number of flash stages associated with a particular evaporator effect.
ØM	- Log mean temperature difference of MSF stage.
QN	- Heat load of MSF stage, Btu/hr.
UBAR	- Overall heat transfer coefficient of MSF stages, $(\text{Btu/hr})/\text{ft}^2 \times ^\circ\text{F}$.

Feed Heater Characteristics

STAGE	N	CT	C	G2	QN	UBAR	HI	HO	QM	AS	LS	ALPHA
1	3	0.03363	0.03484	0.03495	0.42322160E 09	793.72	2486.52	2745.47	10.68	0.499311E 05	5.387	0.125881E 01
2	0	0.03363	0.	0.03506	0.42322160E 09	790.69	2467.51	2735.43	10.73	0.499034E 05	5.384	0.125879E 01
3	0	0.03363	0.	0.03517	0.42322160E 09	787.60	2448.27	2725.16	10.77	0.498828E 05	5.382	0.125904E 01
4	3	0.03363	0.03648	0.03659	0.41041347E 09	784.50	2429.09	2714.94	10.82	0.483694E 05	5.219	0.130993E 01
5	0	0.03363	0.	0.03670	0.41041347E 09	781.37	2409.99	2704.49	10.86	0.483624E 05	5.218	0.131051E 01
6	0	0.03363	0.	0.03681	0.41041347E 09	778.19	2390.67	2693.81	10.91	0.483621E 05	5.218	0.131137E 01
7	3	0.03363	0.03823	0.03835	0.41107530E 09	775.22	2371.09	2686.37	10.90	0.486465E 05	5.248	0.135583E 01
8	0	0.03363	0.	0.03846	0.41107530E 09	771.89	2351.30	2675.15	10.94	0.486577E 05	5.250	0.135694E 01
9	0	0.03363	0.	0.03858	0.41107530E 09	768.50	2331.32	2663.70	10.99	0.486760E 05	5.252	0.135837E 01
10	3	0.03363	0.04014	0.04026	0.41397716E 09	765.37	2311.01	2656.09	10.97	0.492881E 05	5.318	0.140487E 01
11	0	0.03363	0.	0.04038	0.41397716E 09	761.82	2290.46	2644.03	11.02	0.493190E 05	5.321	0.140661E 01
12	0	0.03363	0.	0.04050	0.41397716E 09	758.21	2269.72	2631.76	11.06	0.493577E 05	5.325	0.140872E 01
13	3	0.03363	0.04222	0.04235	0.41921399E 09	754.89	2248.60	2623.92	11.04	0.503206E 05	5.429	0.145763E 01
14	0	0.03363	0.	0.04247	0.41921399E 09	751.09	2227.18	2610.95	11.08	0.503742E 05	5.435	0.146016E 01
15	0	0.03363	0.	0.04260	0.41921399E 09	747.22	2205.59	2597.77	11.12	0.504363E 05	5.442	0.146314E 01
16	3	0.03363	0.04449	0.04463	0.42703512E 09	743.69	2183.55	2589.66	11.09	0.517995E 05	5.589	0.151484E 01
17	0	0.03363	0.	0.04476	0.42703512E 09	739.60	2161.16	2575.68	11.13	0.518801E 05	5.597	0.151840E 01
18	0	0.03363	0.	0.04490	0.42703512E 09	735.44	2138.60	2561.49	11.17	0.519706E 05	5.607	0.152254E 01
19	3	0.03363	0.04699	0.04713	0.43787610E 09	731.65	2115.50	2553.09	11.12	0.536172E 05	5.806	0.157741E 01
20	0	0.03363	0.	0.04728	0.43787610E 09	727.23	2091.98	2537.99	11.16	0.539317E 05	5.819	0.158238E 01
21	0	0.03363	0.	0.04742	0.43787610E 09	722.73	2068.31	2522.67	11.21	0.540582E 05	5.832	0.158810E 01
22	3	0.03363	0.04975	0.04991	0.45250277E 09	718.65	2043.98	2513.98	11.14	0.565304E 05	6.099	0.164645E 01
23	0	0.03363	0.	0.05007	0.45250277E 09	713.84	2019.15	2497.58	11.18	0.566897E 05	6.116	0.165343E 01
24	0	0.03363	0.	0.05022	0.45250277E 09	708.94	1994.16	2480.97	11.22	0.568642E 05	6.135	0.166141E 01
25	3	0.03363	0.05282	0.05299	0.47264783E 09	704.53	1968.35	2472.13	11.13	0.602657E 05	6.502	0.172298E 01
26	0	0.03363	0.	0.05317	0.47264783E 09	699.23	1941.91	2454.18	11.17	0.604885E 05	6.526	0.173293E 01
27	0	0.03363	0.	0.05334	0.47264783E 09	693.85	1915.32	2436.05	11.22	0.607315E 05	6.552	0.174433E 01
28	3	0.03363	0.05626	0.05646	0.49825113E 09	689.00	1887.73	2426.77	11.10	0.651538E 05	7.029	0.181078E 01
29	0	0.03363	0.	0.05665	0.49825113E 09	683.12	1859.38	2407.03	11.14	0.654701E 05	7.064	0.182535E 01
30	0	0.03363	0.	0.05684	0.49825113E 09	677.16	1830.89	2387.12	11.18	0.658155E 05	7.101	0.184214E 01
31	3	0.03363	0.06015	0.06037	0.53453339E 09	671.80	1801.07	2378.01	11.02	0.722056E 05	7.790	0.191095E 01
32	0	0.03363	0.	0.06058	0.53453339E 09	665.17	1770.21	2356.02	11.06	0.726756E 05	7.841	0.193323E 01
33	0	0.03363	0.	0.06080	0.53453339E 09	658.45	1739.23	2333.91	11.09	0.731924E 05	7.897	0.195921E 01
34	4	0.03363	0.06457	0.06475	0.44053317E 09	651.44	1711.02	2303.45	11.30	0.598642E 05	6.459	0.218007E 01
35	0	0.03363	0.	0.06496	0.44053317E 09	645.76	1685.31	2285.19	11.32	0.602772E 05	6.503	0.220814E 01
36	0	0.03363	0.	0.06515	0.44053317E 09	640.01	1659.55	2266.90	11.34	0.607238E 05	6.551	0.223989E 01
37	0	0.03363	0.	0.06534	0.44053317E 09	634.21	1633.75	2248.61	11.35	0.612075E 05	6.604	0.227581E 01
38	4	0.03363	0.06968	0.06991	0.50327216E 09	629.14	1605.80	2245.09	11.06	0.723160E 05	7.802	0.236234E 01
39	0	0.03363	0.	0.07014	0.50327216E 09	622.33	1576.19	2223.98	11.07	0.730810E 05	7.885	0.241467E 01
40	0	0.03363	0.	0.07038	0.50327216E 09	615.44	1546.55	2203.02	11.06	0.739287E 05	7.976	0.247553E 01
41	0	0.03363	0.	0.07061	0.50327216E 09	608.50	1516.90	2182.29	11.05	0.748724E 05	8.078	0.254650E 01
42	4	0.03363	0.07566	0.07595	0.60162108E 09	602.14	1484.82	2176.08	10.73	0.931400E 05	10.049	0.262629E 01
43	0	0.03363	0.	0.07623	0.60162108E 09	594.01	1450.83	2152.84	10.68	0.948125E 05	10.229	0.273530E 01
44	0	0.03363	0.	0.07652	0.60162108E 09	585.84	1416.85	2130.46	10.61	0.957626E 05	10.440	0.286822E 01
45	0	0.03363	0.	0.07681	0.60162108E 09	577.64	1382.91	2109.28	10.51	0.990692E 05	10.689	0.303149E 01
46	5	0.03363	0.00000	0.08288	0.52772194E 09	570.21	1350.35	2096.37	10.27	0.901260E 05	9.724	0.348136E 01
47	0	0.03363	0.00000	0.08316	0.52729021E 09	563.03	1319.35	2084.30	10.04	0.932638E 05	10.062	0.371364E 01
48	0	0.03363	0.00000	0.08344	0.52685861E 09	555.55	1288.47	2069.30	9.85	0.962803E 05	10.388	0.391030E 01
49	0	0.03363	0.00000	0.08372	0.52650878E 09	548.09	1257.66	2056.22	9.62	0.998645E 05	10.774	0.414542E 01
50	0	0.03363	0.00000	0.08400	0.52590892E 09	540.83	1226.90	2047.52	9.31	0.104454E 06	11.270	0.445778E 01
TOTAL MSF AREA										0.322747E 07		

Note: See Previous
Page for
Nomenclature

Feed Heater Characteristics

STAGE	DEL	RHO	TV2	TS1	TS2	T1	T2	DTPDP	SUBMERG	
1	3	0.1094975E-03	0.2980397E 01	255.76	260.00	257.02	243.50	246.52	0.602746E-02	0.153165E 00
2	0	0.	0.2980397E 01	252.78	257.02	254.04	240.47	243.50	0.641153E-02	0.159774E 00
3	0	0.	0.2980397E 01	249.80	254.04	251.06	237.44	240.47	0.682510E-02	0.166613E 00
4	3	0.1101775E-03	0.2892302E 01	246.86	251.06	248.17	234.51	237.44	0.726498E-02	0.180331E 00
5	0	0.	0.2892302E 01	243.96	248.17	245.27	231.57	234.51	0.773064E-02	0.187490E 00
6	0	0.	0.2892302E 01	241.07	245.27	242.38	228.63	231.57	0.823227E-02	0.194882E 00
7	3	0.1147864E-03	0.2902283E 01	238.12	242.38	239.48	225.68	228.63	0.878349E-02	0.199360E 00
8	0	0.	0.2902283E 01	235.22	239.48	236.58	222.74	225.68	0.937014E-02	0.207211E 00
9	0	0.	0.2902283E 01	232.32	236.58	233.68	219.79	222.74	0.100040E-01	0.215325E 00
10	3	0.1204200E-03	0.2928081E 01	229.34	233.68	230.75	216.81	219.79	0.107070E-01	0.218658E 00
11	0	0.	0.2928081E 01	226.41	230.75	227.82	213.84	216.81	0.114577E-01	0.227327E 00
12	0	0.	0.2928081E 01	223.48	227.82	224.89	210.87	213.84	0.122718E-01	0.236297E 00
13	3	0.1272378E-03	0.2970404E 01	220.46	224.89	221.92	207.85	210.87	0.131838E-01	0.238435E 00
14	0	0.	0.2970404E 01	217.49	221.92	218.95	204.83	207.85	0.141617E-01	0.248096E 00
15	0	0.	0.2970404E 01	214.52	218.95	215.98	201.82	204.83	0.152273E-01	0.258111E 00
16	3	0.1354775E-03	0.3031068E 01	211.43	215.98	212.95	198.74	201.82	0.164343E-01	0.258920E 00
17	0	0.	0.3031068E 01	208.40	212.95	209.92	195.66	198.74	0.177353E-01	0.269814E 00
18	0	0.	0.3031068E 01	205.36	209.92	206.89	192.58	195.66	0.191610E-01	0.281138E 00
19	3	0.1454828E-03	0.3113213E 01	202.20	206.89	203.77	189.42	192.58	0.207970E-01	0.280363E 00
20	0	0.	0.3113213E 01	199.08	203.77	200.66	186.26	189.42	0.225721E-01	0.292829E 00
21	0	0.	0.3113213E 01	195.96	200.66	197.55	183.10	186.26	0.245308E-01	0.305837E 00
22	3	0.1577809E-03	0.3222362E 01	192.68	197.55	194.32	179.82	183.10	0.268136E-01	0.302972E 00
23	0	0.	0.3222362E 01	189.45	194.32	191.10	176.55	179.82	0.293116E-01	0.317505E 00
24	0	0.	0.3222362E 01	186.22	191.10	187.88	173.28	176.55	0.320923E-01	0.332746E 00
25	3	0.1733645E-03	0.3370991E 01	182.79	187.88	184.51	169.85	173.28	0.353984E-01	0.326456E 00
26	0	0.	0.3370991E 01	179.40	184.51	181.14	166.43	169.85	0.390583E-01	0.343810E 00
27	0	0.	0.3370991E 01	176.02	181.14	177.77	163.01	166.43	0.431793E-01	0.362135E 00
28	3	0.1927492E-03	0.3558795E 01	172.40	177.77	174.21	159.39	163.01	0.481844E-01	0.352570E 00
29	0	0.	0.3558795E 01	168.82	174.21	170.65	155.78	159.39	0.538075E-01	0.373848E 00
30	0	0.	0.3558795E 01	165.25	170.65	167.09	152.16	155.78	0.602333E-01	0.396518E 00
31	3	0.2187315E-03	0.3823327E 01	161.36	167.09	163.27	148.28	152.16	0.682954E-01	0.378892E 00
32	0	0.	0.3823327E 01	157.51	163.27	159.44	144.40	148.28	0.775531E-01	0.405955E 00
33	0	0.	0.3823327E 01	153.66	159.44	155.62	140.52	144.40	0.883542E-01	0.435153E 00
34	4	0.1913228E-03	0.3155446E 01	150.28	155.62	152.46	137.31	140.52	0.993290E-01	0.554210E 00
35	0	0.	0.3155446E 01	147.10	152.46	149.31	134.11	137.31	0.111194E 00	0.582774E 00
36	0	0.	0.3155446E 01	143.91	149.31	146.15	130.90	134.11	0.124800E 00	0.613256E 00
37	0	0.	0.3155446E 01	140.72	146.15	143.00	127.70	130.90	0.140456E 00	0.645852E 00
38	4	0.2328905E-03	0.3610362E 01	137.03	143.00	139.39	124.03	127.70	0.161611E 00	0.610365E 00
39	0	0.	0.3610362E 01	133.36	139.39	135.78	120.36	124.03	0.186415E 00	0.653043E 00
40	0	0.	0.3610362E 01	129.69	135.78	132.17	116.70	120.36	0.215957E 00	0.699496E 00
41	0	0.	0.3610362E 01	126.01	132.17	128.56	113.03	116.70	0.251347E 00	0.750187E 00
42	4	0.2870392E-03	0.4138756E 01	121.79	128.56	124.42	108.83	113.03	0.300750E 00	0.667961E 00
43	0	0.	0.4138756E 01	117.54	124.42	120.28	104.62	108.83	0.362623E 00	0.733807E 00
44	0	0.	0.4138756E 01	113.27	120.28	116.14	100.42	104.62	0.440734E 00	0.807287E 00
45	0	0.	0.4138756E 01	108.97	116.14	112.00	96.22	100.42	0.540505E 00	0.889443E 00
46	5	0.	0.3836000E 01	104.68	112.00	108.16	92.37	96.22	0.667843E 00	0.108529E 01
47	0	0.	0.3836000E 01	100.61	108.16	104.33	88.53	92.37	0.822998E 00	0.118136E 01
48	0	0.	0.3836000E 01	96.58	104.33	100.49	84.69	88.53	0.102099E 01	0.119897E 01
49	0	0.	0.3836000E 01	92.51	100.49	96.66	80.84	84.69	0.128111E 01	0.119287E 01
50	0	0.	0.3836000E 01	88.36	96.66	92.82	77.00	80.84	0.163149E 01	0.117373E 01

TS2 = Shell side temperature leaving MSF stage, °F
T1 = Tube side temperature entering flash stage, °F
T2 = Tube side temperature leaving flash stage, °F
DTPDP = Delta T associated with MSF pressure losses, °F
SUBMERG = Delta T associated with MSF submergence, °F

NF = Number of flash stages for a particular evaporator effect
DEL = Shell side concentration decrement in MSF stage block
RHO = MSF stage temperature decrement, °F
TV2 = MSF stage condensing temperature, °F
TS1 = Shell side temperature entering MSF stage, °F

IGROUP		AS	SLS	U BAR
2	3	0.14971729E 06	0.16152956E 02	0.79066606E 03
3	3	0.14504385E 06	0.15654134E 02	0.78135227E 03
4	3	0.14598027E 06	0.15749771E 02	0.77187196E 03
5	3	0.14796482E 06	0.15963883E 02	0.76180314E 03
6	3	0.15113108E 06	0.16305489E 02	0.75106773E 03
7	3	0.15565024E 06	0.16793060E 02	0.73957256E 03
8	3	0.16180712E 06	0.17457324E 02	0.72720267E 03
9	3	0.17008429E 06	0.18350347E 02	0.71381171E 03
10	3	0.18148572E 06	0.19580443E 02	0.69920345E 03
11	3	0.19643938E 06	0.21193789E 02	0.68309340E 03
12	3	0.21807361E 06	0.23527901E 02	0.66514040E 03
13	4	0.24207276E 06	0.26117161E 02	0.64285374E 03
14	4	0.29419806E 06	0.31740946E 02	0.61885242E 03
15	4	0.38378432E 06	0.41406384E 02	0.58990771E 03
16	5	0.48398902E 06	0.52217440E 02	0.55554168E 03
TOTAL		0.32274717E 07	0.34821102E 03	

- AS - Heat Transfer Surface Area of the group of MSF associated with a given effect, ft².
- SLS - MSF preheater tube length associated with a particular evaporator effect, ft.
- UBAR - Average overall heat transfer coefficient of preheater tubes associated with an effect, (Btu/hr)/ft² x °F.

Calculated Costs for VTE Rectilinear Plant with Rectangular Bundles

COMPONENTS	COST-\$/YEAR	COST-\$/KGAL	COST-\$/DIRECTS		
Capital Costs					
RDDF	0.48009415E 05	0.58458953E-03	0.70431339E 06		
HIGH TEMP ENDWALL	0.38327316E 04	0.46669486E-04	0.56227391E 05		
LOW TEMP ENDWALL	0.63210003E 04	0.76968040E-04	0.92731084E 05		
LONG SEPARATION WALLS	0.28124159E 05	0.34245551E-03	0.41259036E 06		
TOP BEAM ENCASUREMENT	0.85061605E 04	0.10357577E-03	0.12478808E 06	HIGHER COST FACTOR#	0.12172295E 01
INTERNAL BRINE CHESTS	0.	0.	0.		
FLOOR	0.64441124E 05	0.78467122E-03	0.94537178E 06	VTE RETUB CHARGE#	0.32111007E-01
PUMP ALLEY AND SLABS	0.31860248E 04	0.38794822E-04	0.46739996E 05		
STAGE WALLS	0.20260404E 05	0.24670202E-03	0.29722656E 06	MSF RETUB CHARGE#	0.
COLUMNS	0.95248578E 04	0.11598000E-03	0.13973269E 06		
BEAMS	0.18930174E 05	0.23050441E-03	0.27771169E 06		
EXPANS JOINT-HASTELLOY	0.71841484E 03	0.87478214E-05	0.10539374E 05		
EXPANS JOINT-NEDPRENE	0.22289406E 04	0.27140829E-04	0.32699267E 05		
PHENOLIC COATING	0.34657190E 05	0.42200536E-03	0.50843201E 06		
LINER-1/8 POLYPROP	0.13278489E 05	0.16168631E-03	0.19479965E 05		
LINER-1/4 POLYPROP	0.	0.	0.		
DRAIN PITS	0.81797818E 04	0.99601604E-04	0.12000000E 05		
RDDF INSULATION	0.71389723E 04	0.86928126E-04	0.10473100E 06		
CRANE-13T BRIDGE	0.31850025E 04	0.38782374E-04	0.46725000E 05		
SHELL TOTAL	0.28052283E 06	0.34158031E-02	0.41153591E 07	\$/FT3 VTE SHELL#	0.17134259E 01
EFFECT TUBES	0.16480008E 07	0.20066981E-01	0.16434672E 08		
EFFECT TUBE SHEETS	0.15274266E 06	0.18598802E-02	0.22407834E 07		
EFFECT TUBES+SHEETS	0.18007434E 07	0.21926861E-01	0.18675455E 08	\$/FT2 VTE AREA#	0.30473170E 01
FEED HTR TUBES	0.50793623E 06	0.61849161E-02	0.74515861E 07		
FEED HTR TUBF SHEETS	0.20296791E 05	0.24714509E-03	0.29776038E 06		
FEED HTR INTER TUBE SHEETS	0.65019444E 04	0.79171317E-04	0.95385593E 05		
FEED HTR TUBES+SHEETS	0.53473496E 06	0.65112324E-02	0.78447321E 07	\$/FT2 MSF AREA#	0.24306121E 01
BRINE HTR TOTAL	0.42324245E 05	0.51536371E-03	0.62091013E 06		
CONDENSER TOTAL	0.94278446E 05	0.11479872E-02	0.13830947E 07		
DEAERATOR	0.15859821E 06	0.19311806E-02	0.23266861E 07	\$/GPM MAKEUP FLOW#	0.82797695E 01
CHEMICAL CAPITAL COST	0.13064745E 06	0.15908366E-02	0.19166397E 07		
MSF SHELL COST	0.21954711E 06	0.26733286E-02	0.32208259E 07	\$/FT3 MSF SHELL#	0.24768146E 01
SIIE WORK	0.19878890E 05	0.24205650E-03	0.29162965E 06	PER CENT TOTAL COST#	0.47956882E 00
SEA-WATER INTAKE	0.33014391E 06	0.40200171E-02	0.48433161E 07	\$/GPM INTAKE FLOW#	0.97926143E 01
PUMPS AND DRIVES	0.44598263E 06	0.54305343E-02	0.65427070E 07	\$/HP#	0.19326161E 03
ELECTRICAL EQUIP.	0.64300679E 05	0.78296108E-03	0.94331141E 06	\$/HP#	0.27863983E 02
INST. BLDG. CRANES	0.23622620E 06	0.28764225E-02	0.34655135E 07	PER CENT TOTAL COST#	0.56988452E 01
VALVES AND PIPING	0.31496409E 06	0.38351791E-02	0.46206233E 07	PER CENT TOTAL COST#	0.75983594E 01
				COST-\$/DIR+IND	\$/GPD
TOTAL CAPITAL COST		0.56899761E-01	0.60810801E 03	0.74020698E 08	0.29608279E 00
Operating Costs					
HEAT	0.30646456E 07	0.37316842E-01			
CHEMICAL	0.96456296E 06	0.11769656E-01			
POWER	0.45907159E 06	0.55899129E-02			
OPERATING	0.56779730E 06	0.69138181E-02			
MAINT+SUPPLIES	0.12583519E 06	0.15322397E-02			
TOTAL OPERATING COST		0.63122467E-01			
TOTAL WATER COST, CAPITAL PLUS OPERATING		0.12002222E 00			

Structural Characteristics

NUMBER OF EFFECTS	0.15000000E 02	VTE SHELL FT3	0.24018308E 07
NUMBER OF COLUMNS (1)	0.15925238E 02	VTE AREA FT2	0.61284911E 07
NUMBER OF COLUMNS (N)	0.25018410E 02	MSF SHELL FT3	0.13003903E 07
NUMBER OF CRANES	0.10000000E 01	MSF AREA FT2	0.32274717E 07
NUMBER OF TRAINS	0.40000000E 01		
BRINE HEATER WIDTH	0.51898126E 01		
BUNDLE HEIGHT	0.13000000E 02		
VERTICAL ROWS	0.16000000E 03		
HORIZONTAL ROWS	0.73758221E 02		

	Width of One Train (ft)	VTE Height (ft)	Bundle Width, One Train, ft	Tubes per ft ²	Bundle Depth (ft)	Total No. of Tubes per Effect
1	0.69553923E 02	0.21500000E 02	0.62553923E 02	0.13924532E 02	0.10000000E 02	0.34341364E 05
2	0.73017215E 02	0.21500000E 02	0.66017215E 02	0.13836595E 02	0.10000000E 02	0.36538138E 05
3	0.76497576E 02	0.21500000E 02	0.69497576E 02	0.13725218E 02	0.10000000E 02	0.38154777E 05
4	0.79993400E 02	0.21500000E 02	0.72993400E 02	0.13588167E 02	0.10000000E 02	0.39573861E 05
5	0.83476922E 02	0.21500000E 02	0.76476922E 02	0.13422528E 02	0.10000000E 02	0.41060547E 05
6	0.86947094E 02	0.21500000E 02	0.79947094E 02	0.13224510E 02	0.10000000E 02	0.42290447E 05
7	0.90393456E 02	0.21500000E 02	0.83393456E 02	0.12989244E 02	0.10000000E 02	0.43328720E 05
8	0.93799042E 02	0.21500000E 02	0.86799042E 02	0.12710414E 02	0.10000000E 02	0.44130071E 05
9	0.97124024E 02	0.21500000E 02	0.90124024E 02	0.12379691E 02	0.10000000E 02	0.44528303E 05
10	0.10018786E 03	0.21500000E 02	0.93187834E 02	0.11985758E 02	0.10000000E 02	0.44577081E 05
11	0.10345415E 03	0.21500000E 02	0.96454148E 02	0.11512147E 02	0.10000000E 02	0.44415771E 05
12	0.10639093E 03	0.21500000E 02	0.99390928E 02	0.10937120E 02	0.10000000E 02	0.43482022E 05
13	0.10912819E 03	0.21500000E 02	0.10212819E 03	0.10224363E 02	0.10000000E 02	0.41767826E 05
14	0.11174014E 03	0.21500000E 02	0.10474014E 03	0.93132501E 01	0.10000000E 02	0.39018843E 05
15	0.11470941E 03	0.21500000E 02	0.10770941E 03	0.80868896E 01	0.10000000E 02	0.34841364E 05

Cost Factors					
HIGHER COST	1.21723	INTEREST RATE	0.04000	PLANT LIFE	30.00
VTE TUBING LIFE, YR	15.00	PLANT LOAD FACTOR	0.900	ANNUAL CAP 1/YR	0.05600
PHIR+COND TUB LIFE, YR	30.00				
VTE RETUBING CHARGE	0.03211	POWER COST (\$/KW-HR)	0.00197	HEAT COST (\$/MMBTU)	0.0591
PHIR+COND RETUB CHARGE	0.				

HUEL	55.00000	HDEA	128.39411	HPR	128.57507
HD	35.00000	HEADE	74.23997	MBL	40.00000
HEADR	-0.				

CONDENSER+PREHEATER TUBE DATA

TUBE CONDUCTIVITY	26.00000	TUBE OD	0.75000	FLOODING FACTOR	16.00000
VELOCITY	5.79864	TUBE ID	0.66600		

NUMBER OF CONDENSER TUBES	69282	NUMBER OF PREHEATER TUBES	47205
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POBLD	0.98589	POPRD	4.75709	PODEA	7.90780
PODEL	5.96307	PD2	5.64135	PO1	25.25519
PO3	4.22747	PU	29.48266		

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