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COMPACT HEAT EXCHANGERS FOR SEA THERMAL POWER PLANTS

Final Technical Progress Report, May 1, 1977-March 31, 1978

March 31, 1978

Work Performed Under Contract No. EG-77-C-02-4300

Sea Solar Power, Inc.
York, Pennsylvania

MASTER

U.S. Department of Energy

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FINAL TECHNICAL PROGRESS REPORT

DATE: 3/31/78

PERIOD: 5/1/77 - 3/31/78

PROJECT

Title: Compact Heat Exchangers for Sea
Thermal Power Plants

Report No. COO/4300-7

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CONTRACT

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Start Date: 5/1/77
Completion Date: 6/30/78
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TABLE I
LIST OF SYMBOLS

<u>SYMBOL</u>	<u>DEFINITION</u>
A	Heat Transfer Area sq. ft./ (MM Btu/Hr)
A1	Water Passage Cross-Section Area Sq. ft./ (MM Btu/Hr)
B	Water Passage Height
Cp	Specific Heat of Water Btu/lb/F
<u>BOILER</u>	
C1	Heat Transfer Surface Cost/Unit Area $\$/\text{Ft}^2$
C2	Pumping Cost/HP $\$/\text{HP}$
C3	Total Heat Transfer Cost $\$/(\text{MM Btu/Hr})$
C4	Total Pumping Cost $\$/(\text{MM Btu/Hr})$
C6	Cost of Water $\$/(\text{MM Btu/Hr})$
C7	Total Boiler Cost $\$/(\text{MM Btu/Hr})$
C8	Corrected Boiler Cost $\$/(\text{MM Btu/Hr})$
C9	Cost of Boiler Surface/KW $\$/\text{KW}$
<u>CONDENSER</u>	
C	Heat Transfer Surface Cost/Unit Area $\$/\text{Ft}^2$
C1	Pumping Cost/HP $\$/\text{HP}$
C2	Total Heat Transfer Cost $\$/(\text{MM Btu/Hr})$
C3	Total Pumping Cost $\$/(\text{MM Btu/Hr})$
C4	Cost of Water $\$/(\text{MM Btu/Hr})$
C5	Total Condenser Cost $\$/(\text{MM Btu/Hr})$
C6	Corrected Condenser Cost $\$/(\text{MM Btu/Hr})$
C7	Cost of Condenser Surface/KW $\$/\text{KW}$
F	Friction Factor
F1	Fouling Factor $\text{Ft}^2\text{Hr.F/Btu}$
g	Acceleration Due to Gravity $\text{Ft/Sec}^2 = 32 \text{ Ft/Sec}^2$
G	Mass Velocity lbs/Hr/ft^2

TABLE I (Cont.)

<u>SYMBOL</u>	<u>DEFINITION</u>
H	Heat Transfer Coefficient, Fluid Side Btu/[Ft ² Hr°F]
H _l	Heat Transfer Coefficient, Water Side Btu/[Ft ² Hr°F]
h	Height of Heat Exchanger Ft.
K	Heat Conductance [Btu-In]/[Ft ² Hr°F]
L	Length of Heat Exchanger Ft.
LMTD	Log Mean Temperature Difference
N _{st}	Stanton Number
N _{pr}	Prandtl Number
N	Number of Passages/Heat Exchanger
P	Pressure Loss P.S.F.
P _l	Pumping Power HP
Q	Heat Rate Btu/Hr
R	Reynolds Number, Water Side
R _l	Overall Resistance Hr-F-Ft ² /Btu
r _h	Hydraulic Radius Ft.
t	Thickness in.
T	Water Temp Drop Across Boiler (Condenser) F
T ₁ = 79.74	Effective Temp of Water Entering Boiler F
T ₂ (T ₁)	Boiling (Condensing) Temp leaving last Chamber F
T ₃ (T ₂)	LMTD Boiler (Condenser) F
T ₄ (T ₃)	Mean Boiling (Condensing) Temp F
T ₅ (T ₄)	Temp Diff. Across Wall, Boiler (Condenser) F
U	Overall Conductance Btu/[Ft ² Hr°F]
V	Water Velocity Ft/Sec.
V ₁	First Water Velocity In Computer Calculation
V ₂	Last Water Velocity In Computer Calculation
W	Width of Water Passage in.
ρ	Density, Water lb/Ft ³
μ	Viscosity, Water lb/Hr/Ft

REPORT SUMMARY

This report covers the investigative studies required to develop a low cost, high performance flat plate heat exchanger design suitable for sea thermal power plants.

The studies indicate that higher boiling transfer coefficients quoted in the literature can be attained in a flat plate exchanger. High condensing coefficients are also obtainable.

Higher water side heat transfer coefficients are also obtainable in cleanable open passages.

The study has examined the effect of various fouling rates. A cleaning concept is proposed that is simple, high speed, and easy to operate without plant or exchanger shutdown during the cleaning process.

The exchanger concept for both boilers and condensers is a low profile, mechanical assembled structure with no welding, soldering, or brazing required. This makes it possible to choose a variety of materials best suited for the working fluid and sea water compatibility. While steel and aluminum are the favored materials for low cost and good performance, it is also possible to use materials such as cupro-nickel or titanium, should it prove advisable.

The exchanger design has been evaluated for manufacturability and ease of shipping, installation, and maintenance in a power plant. A preliminary arrangement for exchangers of this construction in a full scale power plant has been examined to provide confidence that the design would be usable in such a plant.

Computer programs have been written to optimize design conditions for boilers and condensers in a power plant, recognizing costs of exchangers, cost of warm and cold water, water pumping power and effect of cycle efficiency.

After optimizing design conditions a conceptual design has been prepared for a test boiler and a test condenser, suitable for testing in the Argonne test facility.

A detailed design for a smaller test unit has been prepared. It is decided that, since these exchangers have many new features in performance and construction, we should first build and test a much smaller model, before building units for test at Argonne. This work is in progress.

The Argonne test element will use ammonia as the working fluid. The smaller test unit will use R-22.

1.0 CONTRACT OBJECTIVE

- 1.1 The objective of this contract was to develop and evaluate a design for a plate-fin heat exchanger assembly that is manufacturable, and is capable of being core tested at a capacity of 3.2 million Btu per hour plus or minus 25%. The general design concept was to be directed toward OTEC applications.
- 1.2 Exact operating conditions were not specified, but the requirement for OTEC applications provided guidelines for the general area of operating conditions. The detailed design conditions needed to be defined by the contractor, based on a general knowledge of potential OTEC operating conditions.
- 1.3 The contract did not specify whether the heat exchanger design should be for a boiler or for a condenser. However, since both are required in an OTEC plant, we construed the objective to be that the design study should include both boilers and condensers.
- 1.4 The objective was to design the heat exchanger for operation with ammonia and sea water as the heat exchange fluids. The problem of cleaning sea water passages was also to be considered. The design was also to be directed to a full scale size, with the test core design to be suitable for closely predicting performance typical of a full scale size, suitable for use in a power plant.
- 1.5 Conceptual design drawings with material specifications were to be furnished, with detailed performance calculations on heat transfer, pressure drops, flow rates, vapor quality and mechanical stresses.
- 1.6 Manufacturers of compact heat exchangers were to be contacted and the potential for using their existing designs for OTEC plants was to be evaluated.
- 1.7 Sea Solar Power, Inc. was to demonstrate the operating OTEC model at approximately six sites in the continental U.S.A. Sites to be selected by ERDA.
- 1.8 The following data is required to be delivered:

DATA

- a. Drawings showing a conceptual design as described in Section 1.5.
- b. A written plan describing the recommended manufacturing procedure.
- c. A bill of material.
- d. A list of firms capable of manufacturing the heat exchangers.

- e. Detailed calculations of performance.
- f. Cost estimates from at least two manufacturers.

1.9 Since no existing heat exchanger designs were finally considered suitable for the original objective, it was decided that the contract should be changed to include experimental manufacture of a smaller model of the design than the size called for in the original contract. This could serve for preliminary testing prior to building one or more test cores with a capacity of 3.2 million Btus. per hour.

2.0 CONTRACT TASKS

2.1 Table III shows a list of heat exchanger manufacturers contacted to see if their designs would be suitable for plate and fin heat exchangers usable in OTEC plants.

2.2 The survey of manufacturers showed that three general types of compact plate and fin heat exchangers are presently being manufactured:

- a. Soft soldered heat exchanger cores. These exchangers are limited to small sizes, not potentially suitable for large cores that will be required for OTEC plants.
- b. Brazed aluminum cores. These cores are being used in many cryogenic applications. They do, however, have certain disadvantages for use in OTEC plants:
 - 1. They are limited in size, because aluminum brazing must be done within an extremely small temperature range. This is almost an impossible task in very large sizes, such as those desired for OTEC plants.
 - 2. Wall thicknesses are critical. It is practically impossible to combine very thin surfaces with thicker surfaces.
 - 3. Specially treated surfaces, such as those required for high transfer rates with nucleate boiling, are difficult to construct in brazed construction.
 - 4. Brazing limits the alloys that can be used to a few aluminum alloys, and the combination of brazing material and sheet materials is likely to promote corrosion.
 - 5. Brazing construction has generally been limited to small passage sizes, not very easily cleanable.
- c. Pressed plate exchangers, mechanically bolted together with gaskets to seal plates, and shaped passages formed by pressed out projections or dimples in the plates. These exchangers are limited in size but are widely used for many purposes. They have certain disadvantages for OTEC service:

1. They are rather limited in size.
2. Since passages must be rather complex they are difficult to clean mechanically, unless the exchanger is disassembled.
3. There appears to be little possibility of including high efficiency nucleating boiling surfaces in the design.
4. There appears to be little possibility of producing long straight passages that are desirable for easily cleanable sea water passages.

2.2.1 Letters were sent to manufacturers listed in Table III. It was felt that these firms might have experience in manufacturing plate heat exchangers and might in fact have "on the shelf" heat exchangers which would be useful as a test article. If a heat exchanger design could be found which was sufficiently similar and could provide relevant test data then it would be logical to purchase a small sample and have it tested. This data would then be helpful in the detailed design of our own plate heat exchanger.

Table IV lists those manufacturers with which we were already familiar and those which responded to our inquiry, and which exhibit a product line, and/ or manufacturing capability conducive to flat plate heat exchangers. Table V separates plate heat exchanger manufacturing types into several generic classes. Mechanical Plate types are all similar in that they consist of press formed plates clamped together in a stack with gaskets at their edges. The plates are machine pressed and formed to provide channels of flow for two fluids in either parallel or counter-current flow. These exchangers are simple to manufacture and assemble. They are held together by clamping bolts and can be disassembled simply by loosening the bolts. Mechanical plate heat exchangers produce turbulence in both fluids and therefore, enhance heat transfer. The tortuous paths between the channeled plates do inflict a relatively large pressure drop on the fluids. Pressure drop on the water side is a particularly serious problem in a sea thermal plant, because the water flow per net kilowatt generated is extremely large. The present conventional plate designs are not well suited to boiling or condensing duty since plate spacing and channel widths are uniform for each flow. Cleaning may be accomplished by chemical or slurry flushing. Commonly, plates are disassembled in conventional service for inspection and scrubbing.

A second generic type of plate heat exchanger is the flat plate brazed heat exchanger. These are constructed of flat plates separated by corrugated fin sheets. Parallel, cross-flow, and counterflow arrangements are possible.

Variable channel widths are available. Several manufacturers make these units to special order. Individual units are relatively small because they must be assembled and furnace brazed. Thus the entire unit must be brought up to brazing temperature uniformly and likewise cooled slowly.

Panel coil heat exchangers are pre-formed plates which are soldered, brazed, or welded together. The two pieces are so formed that when they are mated together a tubular circuit results. In other words, each plate has a tubular circuit pressed on its face and when mated with its mirror image plate, a sealed tubular circuit results. These commercial off-the-shelf units do not accurately simulate the cross-flow plate design proposed for SSP plants.

Several heat exchanger manufacturers or metal fabricators have expressed interest in working with us to build prototype modules. We have built a small wooden heat exchanger module to be followed by an actual metal module on which the mechanical sealing and assembly can be evaluated. In addition, we propose to have the unit thermally tested. This would provide valuable data prior to constructing the full size test unit for the Argonne test facility.

2.3 Design Requirements

2.3.1 Study of Nucleating Surfaces

Since sea thermal power plants utilize a relatively small temperature difference, it will be advantageous to use a nucleating surface that can also improve transfer coefficients.

By studying the literature on nucleating enhancement, it seems best, for fluids such as refrigerants, to use surfaces with small interconnecting holes. By using such materials as sintered metal particles, fine screen wire, perforated metal, foamed metal, or expanded metal, it is possible to increase the area for heat transfer as well as increasing nucleation activity.

We have had numerous discussions with people in the metal industry and are working with them to develop a possible test program for these materials. We are planning to incorporate these nucleating surfaces in our small test heat exchanger and hopefully will be able to test various surfaces for their heat transfer and nucleating capabilities.

2.3.2 Boiler Water-Side Heat Transfer Surfaces

We have studied the possibilities of increasing heat transfer on both the water side and the boiling side. On the water side, we have looked into the possibility of using shallow diagonal grooves in the plates which would serve to help break up the boundary layer but would still be easily cleanable. Even though the diagonal grooves will increase the friction loss, thus increase pumping cost, the larger decrease in cost due to an improvement in heat transfer coefficient could result in a net decrease in cost. However, in reality, this may not always be the case because the friction loss with grooved surface will cause the optimum system cost to occur at a lower velocity. There is indication in the literature that biological slime film thickness decreases with increasing water velocity. Hence, the lower optimum water velocity derived for the grooved configuration may result in a larger fouling factor than is the case for a smooth surface. Clearly an increase in fouling factor is not favored. We have studied the effect of diagonal grooves on the water side in terms of boiler performance. Examples of our results are shown in Fig. 10. Smooth surface results are shown in Fig. 9.

2.3.3 Types of Boiling Contact

The boiling of the fluid in a sea thermal plant is most nearly like that in a refrigerant evaporator. The temperature differences are also similar in a typical refrigerant evaporator. Since the fluids being evaporated are commonly used refrigerants, the data on refrigerant evaporation is probably most nearly characteristic to the boilers we would want to use here. Since refrigerant evaporators are highly developed and in wide usage in many designs, this industry probably provides our most useful and accurate information on boiling of refrigerants. There are four broad classes of evaporators used in refrigeration work.

- a. The spray type. In this the liquid is sprayed over the evaporating surface. This is used very little today, partially because it is difficult to insure complete wetting of the surface, and partially because spraying requires power. This type does not appear very feasible for a sea thermal boiler.
- b. DX or direct expansion. In this type the fluid expands and boils as it flows thru a channel. This scheme operates very well on a refrigerant evaporator where there is vapor at the channel inlet and high velocity can be maintained throughout the channel length. It is not generally used in large evaporators, because it is difficult to keep the flow uniform in

all channels. The high velocity in this type permits development of high heat transfer coefficients, but flow paths tend to become long. It is doubtful if this type can be successful in a sea thermal boiler, because the pure liquid state at inlet would have very low velocity and poor heat transfer. If velocity in the liquid section is increased, then the flow path gets longer, and velocity in the pure vapor region becomes so high that pressure loss will be excessive. The problem of equalizing flow rate between channels is also almost an impossible one. In refrigerant evaporators this has proved to be very difficult, requiring individual controls to small groups of tubes.

- c. Flooded evaporation. In this type the heat exchange surface is literally submerged in the liquid and the vapor bubbles rise to the surface thru the liquid, much as they do in an ordinary teakettle. There is violent agitation caused by flashing at the bottom from the expansion valve, thereby promoting good heat transfer. There are some problems with flow distribution, but they are not as serious as those in the DX type. This is the most commonly used evaporator in large refrigeration systems. In a sea thermal boiler it has the defect that there is no agitation, and therefore, poor heat transfer is likely in the liquid region.
- d. Flooded evaporation with liquid recirculation. This is similar to the plain flooded evaporator, but some of the liquid is allowed to overflow at the top of the evaporator and is pumped into the bottom to be recirculated. This insures complete wetting of all surfaces for good heat transfer. It also increases liquid velocity at inlet, so that heat transfer in the liquid heating region can be improved to an acceptable level. It also helps to minimize the problem of bad flow distribution, caused by the natural tendency for flow to be somewhat unstable in the boiling region. This type evaporator has the disadvantage of a slight increase in pumping power, and requires many pumps. Despite this, it is most probably the best type system for a sea thermal boiler. Moderate recirculation without pumps is also possible with certain arrangements.

We have designed a recirculation system that will not require pumping and thus will not increase the pumping cost as with the conventional system. This type of flooded evaporator is easily adapted for use with multiple fluid circuits.

2.3.4. Mechanical Construction

In considering the possibilities for the structural design of the exchanger modules, we feel that a completely mechanical construction of the exchanger has a decided advantage over brazed construction.

Mechanical construction would allow the use of several different alloys in the construction. This would allow for better control of corrosion, by concentrating corrosion on the larger parts and minimizing corrosion on the critical heat transfer surface.

Brazing requires cleaning after manufacture and the overall process can be quite costly. Thus in attempting to find a method of construction with low cost and dependability, it appears that mechanical construction is the method to use.

In the module design, it is necessary to put the maximum amount of heat transfer surface into the minimum amount of space. The heat exchanger must also be manufacturable and shippable as well as easy to remove for repair or replacement. The design that we are considering should meet all these requirements.

We have found that it will be advisable to include, in the module, a mechanism to remove liquid from the vapor, thus preventing it from flowing through the turbine. It would be best to remove the liquid before it leaves the boiler, thus an independant eliminator would not be required. For this reason we have decided to put the eliminator right above the boiler surface. This is also well established refrigeration practice. For our proposed boiler design, a working fluid flow control will be required. Simple float and pilot valve systems have been useful and can be employed here.

Considering possible materials for the heat exchangers; aluminum alloys, copper-nickel alloys, and titanium are the most probable. Copper-nickel alloys are very resistant to fouling but cannot be used with ammonia. Aluminum alloys have high heat transfer properties and low cost but are subject to corrosion and fouling. Titanium is very corrosion resistant but is very expensive and is subject to fouling. It also has rather low thermal conductivity.

Since we would like to use mechanical construction rather than brazing for assembling the heat exchanger, suitable sealants will have to be used.

These sealants must withstand sea water and working fluids, and must be compatible with the heat exchanger materials. Critical joints will occur between flat surfaces and between perpendicular surfaces such as where the metal heat exchanger plates meet the upper and lower housings.

We have talked with people in the sealant industry and have gathered data and samples of potential sealants which could be tested in the test heat exchanger. A list of these samples and their properties is shown in Table I. Then the material's compatibility with various metals (aluminum, copper-nickel alloys, steel, and titanium), various working fluids (ammonia, halo-carbons, and hydrocarbons), and sea water is noted. An X signifies that this product is compatible, while a NO means that it is definitely not compatible. A blank signifies that this particular information was not available. This does not necessarily mean that the product will not work, but it's successful use with these materials remains questionable.

As seen from the Table I, only a few products are definitely compatible with all the possible heat exchanger materials and fluids. Gore-Tex and Episeal Universal Resin can be used with all the metals, working fluids, and sea water. B. F. Goodrich's A610B and A821B and Dural's product #103 are compatible with all the metals and working fluids, but have not been specifically tested for long exposure to sea water. A number of the other products look like they may be possible choices if further information can be obtained.

It should be noted that viscosity will be a major factor in the selection of a possible sealant, since the material may have to be injected into the joint. This will have to be tested for each specific case. Most of the products can be modified to suit our needs. It would be advantageous to use mechanical sealing devices such as o-rings or gaskets, where possible. We have talked with people in the industry about these products and we hope to test various combinations in the small test heat exchanger.

2.3.5 Fouling Factor Models

An extensive study was made of fouling and of possible functions relating it to velocity and to water side heat transfer coefficients. A graph of 4 fouling factor models is shown in Fig. 1. All of these models are asymptotic in time. That is they assume round tubes or annulus flows which have been in service for many weeks and have developed a full fouling layer which is only

being controlled by changes in velocity and not by any cleaning methods.

The first model assumes simply that the fouling factor, F_1 , is inversely proportional to velocity, V .

$$F_1 = .004/V \quad (\text{Curve A})$$

The constant .004 is based on arbitrarily choosing a value of $F_1 = .0005$ at $V = 8$ ft./sec.

As a next approximation, we make the assumption that the heat transfer coefficient is inversely proportional to the thickness of the boundary layer and thus the fouling factor becomes inversely proportional to the water side heat transfer coefficient, H_1 :

$$F_1 = .672/H_1 \quad (\text{Curve B})$$

The constant .672 is arrived at by the following assumptions. Arbitrarily assume $F_1 = .0005$ at 6 ft./sec. water velocity (V) in a passage width of .75 inches.

This results in $H_1 = 1344$ Btu/ft.²hr.°F (see p.13)

Curve C in Fig. 1 shows a fouling factor function that is based on information developed by the Heat Transfer Research Institute (Ref.14).

$$F_1 = \frac{\text{constant}}{V^{.5} e^{(V/6)^2}} \quad (\text{Curve C})$$

Assuming $F_1 = .0005$ at $V = 6$ ft./sec. the constant = .003329 and

$$F_1 = \frac{.003329}{V^{.5} e^{(V/6)^2}}$$

This relation is based on a great deal of test data and may be considered to be the closest approximation of the effect of velocity on the fouling factor. It should be noted that the data were not specifically for the open ocean and thus do not simulate true conditions for a Sea Solar Power plant.

Also, Heat Transfer Research Inst. suggests that while this function is suitable for tubular exchangers, values for flat plate exchangers can be expected to be less than this. The reasons given are that in general there can be more dead spaces not subject to turbulent action in tubular exchangers than in flat plate exchangers.

Curve D shows the function:

$$F_1 = 16.9/V^{1.8}/H_1 \quad (\text{Curve D})$$

This model is assumed in an attempt to approximate the effect of both the water side heat transfer coefficient and velocity on fouling factor. As seen in Fig. 1, this function gives values of F1 closer to the Heat Transfer Research, Inst. curve at higher velocities than Curve A or B.

The above models for fouling were examined to see their variation with velocity and how they compare to each other. They correlate very well indeed. The HTRI data is probably the most extensive and the best prediction for asymptotic fouling rates but even the simple hyperbolic relationship of F1 to V as in Curve A follows the shape of the HTRI data quite well at the higher velocities.

In our latest heat exchanger performance calculations we have used the model

$$F1 = \frac{.0003}{V}$$

This formulation used the same inverse proportionality between F1 and V as in Curve A but now assumes a cleaning cycle of at least once in 4 weeks. In other words if we clean the water sides once in 4 weeks, we can expect a lower average fouling rate than the asymptotic values of Fig. 1.

As a check on this model at 6 ft./sec. water velocity

$$F1 = \frac{.0003}{6} = .00005$$

Comparing this to recent test data at Keahole Point in Hawaii (Ref. 8) and at St. Croix on aluminum tubes in open sea water, we see that after 4 weeks F1 is .00004 to .00006. Therefore, our assumed model is probably a good starting point to attempt to optimize on heat exchanger design. It should be kept in mind that in the computer program we can use a constant or any other model we like for F1 but it is preferable if it relates known data such as the fouling tests at Keahole Point, St. Croix and HTRI.

Reference 12 shows some results of long time fouling factors using cooling tower water. While this data was not taken on sea water tests, it was done under actual heat transfer operating conditions, and is therefore useful. It illustrates the great effect of velocity on the fouling rate. These tests apply to chemical precipitation rather than to biofouling, but the trends shown should be of interest.

2.3.6 Staging Effects

As pointed out in our early papers on this subject, it is extremely desirable to stage the heat exchangers, so that the water in passing thru the exchangers boils the working fluid at several different temperatures, likewise in passing thru the condensers the water condenses the fluid at several different temperatures. This permits more BTU/lb. of water pumped thru the plant. A detailed discussion of the effect of staging on plant performance is given in Appendix II. The effect of staging condensers and boilers is shown on Figures 3, 4, 5. Fig. 3 shows relative performance for a log mean temperature difference of 6°F, Figure 4 for a log mean temperature difference of 4°F, and Figure 5 for a log mean temperature difference of 2°F. The curves are plotted to show the mean temperature difference available between boiling and condensing temperatures, for various values of heat extracted or delivered per lb. of water, or water temperature drop thru the boiler or rise thru the condenser. The curves assume that the temperature difference in boiler and condenser are the same, and that the staging effect in the boiler and condenser is the same.

For example, as shown in Fig. 5 using a value of 12 BTU/lb. for heat extracted from the water in the boiler and 12 BTU/lb. heat delivered to the water in the condenser, the use of a one stage heat exchanger results in a temperature difference of 15.7°F across the turbine. Using a 2-stage heat exchanger results in a 21.1°F temperature difference. This 5.4°F increase in temperature across the turbine by going from a one stage heat exchanger, results in a large increase in plant output. As can be noted, the effect of staging is very marked between going from one to two stages, and the effect becomes less as more stages are added. The effect of staging is also greater at lower mean temperature differences, and is greater as the heat obtained per pound of water is increased. These factors have an obvious effect on water required per unit of net power output.

2.3.7 Construction Design of Heat Exchangers

In developing a heat exchanger design it is important to fit it into a plant arrangement that is practical for construction, maintenance, cleanability, and efficiency of water and flow circulation.

The design variables for a heat exchanger module are the dimensions of the working fluid passage and the dimensions of the water passages. Since we selected a flooded recirculation type of boiler, this dictated that

the boiling passages be vertical, and be low enough in height so that the static liquid head penalty would be low. Low passages also made it easy to keep gas velocities low enough so that elimination of liquid droplets would not be difficult, and that friction pressure drop in the passages would be negligible.

Clearly the advantages of low short passages must be examined against manifolding difficulties and losses. It is likewise important to include module assembly shape and dimension from a transportation standpoint. It is with these considerations that for purposes of this study we decided on a 2 ft. high working fluid passage, an 8 ft. overall exchanger width. The static head penalty for a 2 ft. working fluid height will be .11°F on ammonia or .26°F on R-22. Certainly, these dimensions can be altered as plant arrangements and logistics for a full scale plant are made. However, these dimensions serve as a good starting point for plant design.

The water passages are made 0.75" wide so as to be cleanable by a variety of methods. The height of each horizontal flow passage would be about 1.52 inches high. Therefore, there would be 15 water flow passages in a 2 ft. high heat exchanger. The hydraulic diameter of each .75 x 1.52 inch passage is

$$\frac{4[.75 \times 1.52]}{2[.75 + 1.52]} = 1.004''$$

Having fixed for this study the height of the heat exchanger plates and the width of the water passages, the length of the water passages would be calculated from the assumed heat transfer data and the overall temperature drop chosen for the water. This has been calculated to be approximately 50 ft. for the boiler and 50 ft. for the condenser.

The boilers, 8 ft. x 2 ft. in area and approximately 50 ft. in length, would have a thermal rating of about 80,000,000 BTU/HR. The condenser will have the same cross-sectional area as the boilers and a thermal rating of about 95,000,000 BTU/HR. A sketch of this heat exchanger unit is shown in Fig. 20.

The heat exchangers could be put into modules of 8 heat exchangers each, 4 units high and 2 units wide. The size of this module would then be about 24 ft. wide, 18 ft. high, and 50 ft. long.

Since the greatest parasitic loss in the plant would be water pumping power it is obviously important to make water passages as simple and straight as possible, and to avoid all high velocity turns. This convinced us that the water passages should be straight thru, single pass.

It was also considered important to eliminate any high velocity turns entering or leaving the heat exchanger. As an example of how serious such losses can be, Reference 13 cites a proposed heat exchanger arrangement where the inlet and outlet elbow losses would require 6000 pumping horsepower for a 100 MW power plant.

2.3.8 Design Condition Selection For Boilers

Cost optimization for the boiler and condenser becomes a matter of balancing the cost of supplying the water and the cost of transmitting the heat thru the heat exchangers, and at the same time obtaining the economic maximum power or temperature difference across the turbine.

In the boiler, an increased water flow will deliver less heat per pound of water to the working fluid (with a higher mean water temperature) thus requiring less heat transfer surface. However, the cost of the warm water will increase to offset the savings in the cost of the heat transfer materials.

If warm water inlet and outlet temperatures and flow are fixed, a reduction in the log mean temperature difference (LMTD) between the water and boiling temperature means that the boiling temperature has increased. By the simple equation

$$Q = UA (LMTD),$$

where Q = heat flux,

and U = overall heat transfer coefficient.

Clearly an increase in A or in U can result in a decrease in LMTD with Q remaining the same. So if we increase heat transfer surface A keeping U and Q constant LMTD is reduced. Then mean boiling temperature is raised causing a corresponding increase in temperature difference across the turbine which means a greater turbine power output.

A computer program has been set up to evaluate these different factors and to determine what values of water velocity, water temperature drop, log mean temperature difference, and boiling temperature will give minimum cost.

Copies of the computer programs for both smooth and grooved surfaces are shown in Figures 6 and 7. A table of symbols and units used in these programs and this report is listed in Table I.

The boiler water inlet temperature is assumed to be 80F in all cases. From this we subtracted 0.26°F to allow for static head penalty in the two foot high boiler passages. The effective boiler water inlet temperature, used for heat transfer calculations, and designated as T1 in the table of symbols, is then 79.74F in all cases. The condenser water inlet temperature is assumed to be 40F in all cases.

In determining cost optimization, values of T, T1, W, C1, C2, V1, V2 are input, and computations are made for successive values of velocity at 1 ft./sec. increments from V = V1 to V = V2.

- T = Water temp drop across boiler (condenser)
- T1 = 79.74 = Effective temp of water entering boiler
- W = Width of water passage
- C1 = Heat transfer surface cost/unit area \$/ft.²
- C2 = Pumping cost/HP \$/HP
- T4 = Mean boiling temperature

To generate a curve such as shown in our cost curves, different values of T4 are computed for a constant T and then cost is plotted versus T4 corresponding to each T at the velocity giving the lowest cost.

The Reynolds number R, for water flow is calculated from

$$R = \frac{4r_h V \rho \times 3600 \text{ sec.}}{\mu \text{ hr.}} \quad (\text{Ref. 15})$$

$$\text{and } r_h = \frac{1}{12} \left(\frac{BW}{2(B+W)} \right) \text{ in ft.}$$

where r_h is the hydraulic radius in ft., V is water velocity in ft./sec., ρ is water density in lb./ft.³, μ is water viscosity in lb./HR./FT., B is passage height and W is passage width in inches.

For our exchanger configuration the 2 ft. passage height is divided into 1.52" high passages separated by spacers. Therefore, $B = 1.52"$

$$\text{and } r_h = \frac{1.52 W}{24 (1.52 + W)} \quad (\text{FT.})$$

Using a mean water temperature of 78.5°F , $\frac{\rho}{\mu} = 29.296 \frac{\text{HR}}{\text{ft.}^2}$

$$R = 26,718 V \left(\frac{W}{W + 1.52} \right)$$

Next, the water side heat transfer coefficient, H_1 , is determined from:

$H_1 = N_{st} G c_p$ where N_{st} is the Stanton Number, G , the water mass velocity, c_p the specific heat of water.

From Reference 4, Fig. 28, N_{st} , the Stanton No. is given by:

$$N_{st} = \frac{E}{R^{.18}} \left(\frac{\mu_w}{\mu_m} \right)^n \times 10^{-4}$$

μ_w is viscosity at the wall, μ_m is mean viscosity in the water channel.

In our case $E = 75.5$ at N_{pr} (Prandtl Number) = 6.00 and $n = .20$ for cooling water.

Also $G = 3600 \times 62.3 V$ and $c_p = .998 \text{ BTU/lb.}^\circ\text{F}$.

Combining equations

$$H_1 = 1679V/R^{.18}$$

The log mean temperature difference is calculated for 4 stages of boiling, as described in the diagram below.

Here $D = LTD' = T1 - T - T2$ where T is water temperature drop across the boiler, $T1$ is the effective temperature of water entering the boiler, and $T2$ is boiling temperature leaving the last chamber.

The log mean temperature difference, $LMTD = T3$ is given by:

$$LMTD = \frac{(T1 - t_a) - (T1 - T/4 - t_a)}{\ln \left(\frac{T1 - t_a}{T1 - T/4 - t_a} \right)}$$

Substituting for $t_a = T1 - D - T/4$ gives:

$$LMTD = T3 = [T/4] / \left[\ln \left(\frac{D + T/4}{D} \right) \right]$$

The mean boiling temperature, $T4$, is equal to:

$$T4 = \frac{t_a + t_b + t_c + t_d}{4}$$

where:

$$\begin{aligned} t_d &= T1 - T - D \\ t_c &= T1 - 3 \times T/4 - D \\ t_b &= T1 - 2 \times T/4 - D \\ t_a &= T1 - T/4 - D \end{aligned}$$

thus:

$$T4 = T1 - D - 5 \times T/8$$

since

$$D = LTD = T1 - T - T2$$

$$T4 = T2 + 3 \times T/8$$

The model for the boiling heat transfer coefficient on the fluid side is

$$H = 11,349 \times T5^{(.3028)}$$

The use of this formulation comes as a result of much in depth study of nucleate boiling as researched and recorded in the refrigeration industry and other sources such as Ref. 2. Frankly, we cannot pretend to know exactly what to expect for boiling coefficients. Therefore, we have taken the conservative approach using the lowest boiling coefficients which could be expected for enhanced boiling surfaces. This equation is based on data from the propane curve in Fig. 8, Ref.11, Fig. 10. Even though, in the plate configuration, R-22 should develop higher coefficients than ammonia or propane (Ref.2), we used the lower estimate of the propane coefficient in our program.

We considered this function to be a typical coefficient for nucleate boiling surfaces. However, since this represents a rather low resistance compared to the overall resistance, considerable variations in this value will not affect the overall coefficient very much.

The overall resistance per unit area, R_1 , is the sum of the resistance due to the water film, the dirt film on the water side, the metal wall, the dirt film on the fluid side, and the fluid film.

As shown in Reference 1, the overall resistance per unit area, R_1 , is:

$$R_1 = 1/H_1 + \frac{t_2}{K_2} + \frac{t_3}{K_3} + \frac{A_1 t_4}{A_4 K_4} + \frac{A_1}{A_4 H}$$

where:

- t_2 = thickness of the dirt film on water side
- K_2 = conductivity of the dirt film on water side
- t_3 = thickness of metal wall
- K_3 = conductivity of metal wall
- A_1 = water side area
- A_4 = fluid side area
- t_4 = thickness of dirt film on fluid side
- K_4 = conductivity of film on fluid side

If the thickness of the dirt film on the fluid side is assumed to be zero and the second term on the right is called the fouling factor (F_1):

$$F_1 = t_2/K_2 = .0003/V$$

with 0.080" aluminum wall, $K_3 = 1224$, and using $A_1/A_4 = 1$:

$$R_1 = 1/H_1 + F_1 + .0000654 + 1/H$$

Since H in this equation depends on T_5 , we must estimate the temperature difference across the wall (T_5). An iterative technique was used where, first 1/10 of the LMTD (T_3) was used to calculate a first guess of the overall resistance R_1 . The ΔT (T_5) was recalculated using $T_5 = T_3 (R_1/H)$ and this value was used to compute H .

The equation for the overall heat transfer coefficient is given by:

$$U = 1/R_1$$

$$\text{Using } A = Q/[LMTD \times U] = [Q \times R_1]/T_3$$

The area of heat transfer area needed for 1,000,000 BTU/HR is:

$$A = 1,000,000 \times R1/T3$$

The necessary water channel cross-sectional area is determined by:

$$A1 = Q/G/C_p/T$$

$$A1 = Q/(V \rho \times 3600)/C_p/T$$

with $Q = 1,000,000$ BTU/HR,

and $\rho = 62.3$ lb/Ft³

$$A1 = 4.459/V/T$$

The number of passages required will be determined by the ratio of the total area to the area per passage.

$$N = \text{Total Area/Area per passage}$$

In terms of passage width (inches) W , and passage hgt. of 2ft

$$N = A1/[2 \times \frac{W}{12}] = \frac{6 \times A1}{W}$$

The length of the heat exchanger will be determined by:

$$A = 2 N h L$$

for $h = 2$ ft. $L = A/4N$

To determine the pressure loss through the heat exchanger, it will be necessary to calculate the friction factor.

From Reference 4

$$F = .03875/R^{.193}$$

The friction pressure drop through the heat exchanger is calculated from (Reference 4):

$$P = \frac{F \rho V^2}{2g} \times A/A1 \left(\frac{1b}{Ft^2} \right)$$

$$\text{where } A/A1 = \frac{L}{r_h}$$

with $L =$ Passage Length

and $r_h =$ hydraulic radius

For inlet and exit losses, due to contraction and expansion, the resistance coefficient is given by (Reference 15):

$$K = \left(1 - \frac{W}{W + .255}\right)^2$$

Here .255 is the overall width of the boiling channel, including plates, and

$\frac{W}{W + .255}$ is the ratio of velocity ahead of channel to velocity in channel.

The sum of inlet and exit losses is then $2K \left(\frac{v^2}{2g}\right)$.

Combining equations for friction loss plus inlet and exit losses, the total pressure loss is equivalent to:

$$P = .97xV^2x \left[FxA/A1 + .13/(W + .255)^2 \right] \left(\frac{1b}{Ft^2} \right)$$

Then pressure loss is converted into pumping power.

$$P1 = \frac{P \times V \times A1}{550 \times .85} \quad (\text{Horsepower})$$

where .85 is pumping efficiency.

To calculate the cost for heat transfer surface,

$$C3 = (\text{Cost/Area}) \times \text{Area} = C1 \times A$$

For pumping cost

$$C4 = (\text{Cost/HP}) \times \text{HP} = C2 \times P1$$

Calculation of the cost of water is based on assumed cost figures of warm water system components:

Trash rack and inlet screen	\$.84 million
Warm water piping	.40 million
Warm water pumps and drives	3.48 million
Chlorination Equipment	2.54 million
Total	\$ 7.26 million

With warm water flow of 63.5×10^8 lb/hr:

$$\text{Cost/million lb./hr.} = \$1143$$

Since the number of BTU of heat per pound of water delivered to the working fluid is approximately equal to T:

$$\text{BTU/lb.} = T$$

$$\text{The cost/million BTU/Hr.} = \frac{\$1143}{T}$$

To this must be added the pumping cost outside of the boilers. For this we will use an assumed head loss of 1.43 ft. Pumping HP required will be.

$$\text{HP} = \frac{1.43 \times 63.5 \times 10^8}{3600 \times 550 \times .85} = 5407 \text{ HP}$$

$$\text{HP/[MM lb/Hr]} = \frac{5407 \times 10^6}{63.5 \times 10^8} = .8515$$

$$\text{Since BTU/lb} = T$$

$$\text{HP/[MM BTU/HR]} = \frac{.8515}{T}$$

To convert this into a value of cost, it must be multiplied by C2

$$\text{Cost/[MM BTU/HR]} = \frac{.8515 \times C2}{T}$$

Thus the total cost of water, C6, is

$$C6 = 1143/T + .8515 \times C2/T$$

C7 is the total cost, i.e., the sum of C3, C4, and C6.

A correction must be made to C7 for cases where the temperature difference across the turbine varies from optimum.

Using an assumed optimum temperature difference of 22°F, a mean condensing temperature of 49°, a temperature head penalty loss of 4.4°F, and raising the ratio to the power of 1.75, the corrected cost is:

$$C8 = C7 \times \left(\frac{17.6}{T4 - 53.4} \right)^{1.75}$$

The correction for temperature drop across the turbine is based on considerations discussed in Reference 13. The power output from the main turbine is very close to being proportional to the temperature drop thru the turbine or the difference between boiling and condensing temperatures. Since water pumping power must be subtracted from gross power this can be approximated by subtracting 4.4°F from the turbine temperature drop. The net power output per lb. of water supplied is then proportional to the quantity (turbine temperature drop - temperature drop equivalent to water pumping).

While this proportion provides a rough evaluation of effect on net power output, it must be recognized that the turbines and pipes, and associated plant structure must become larger as turbine temperature drop becomes smaller. Therefore, the effect of temperature drop on cost is greater than a first power relationship. We have assumed the effect to be proportional to the 1.75 power. This power relationship probably lies between an exponent of 1.25 and 2.25. It is too early to place this with precision. However, a change in exponent is not likely to change optimum conditions very greatly.

To determine the cost of the boiler surface/KWH, the quantity of BTU/KWH must be calculated.

The theoretical cycle efficiency is approximated by:

$$\frac{22^{\circ}\text{F} - 4.4^{\circ}\text{F}}{460^{\circ}\text{R} + 71^{\circ}\text{F}} = .0331$$

Multiplying this by the turbo-generator efficiency results in Cycle efficiency = $.0331 \times .97 \times .93 = .0299$ BTU/BTU Input. Where generator eff. = .97 and turbine eff. = .93.

BTU Input/Kwh is given by:

$$\text{BTU Input/Kwh} = 3413 / .0299 = 114,100 \text{ BTU/Kwh}$$

The cost of the boiler surface/Kw is thus:

$$C9 = C8/C7 \times C3 \times .1141$$

which includes the correction factor for variation in temperature difference across the turbine.

If a grooved surface is used for enhancement of the heat transfer coefficient, then the value of H1 will have to be re-evaluated.

From Figure 1, P.9 of the Bergles & Jensen report (Ref. 3) with spiral grooves, a ratio for H1 of 3 to 1 over the smooth surface is shown. Thus assuming the same ratio for diagonal up and down grooves in a rectangular passage, the value of H1 would be:

$$H1 = 3 \times 1679 \text{ V/Re}^{.18}$$

$$H1 = 5037 \text{ V/Re}^{.18}$$

Use of diagonal grooves will change the friction factor F. From Figure 10. (Reference 3).

F for smooth tubes is:

$$F = .0704/Re^{.237}$$

For spiral grooves:

$$F = 2.166/Re^{.474}$$

If it is assumed that the Kays and London friction factor for flat plate will be increased in the same ratio as spiral groove to smooth from Bergles & Jensen then F for diagonal grooves will be:

$$F = \frac{.03875}{Re^{.193}} \times \frac{2.166}{Re^{.474}} \times \frac{Re^{.237}}{.0704}$$

$$\text{or } F = 1.192/Re^{.43}$$

Figure 9 shows our latest results for cost optimization of the boiler based on the computer program described above. A sample calculation using this analytical technique and corresponding to data in Figure 9 is shown in Appendix I. For this case the minimum boiler cost of \$990.00/[MM BTU/Hr] occurs at a water velocity of 7 feet per second, $T = 6^\circ\text{F}$, and a mean boiling temperature of approximately 69.8°F . This corresponds to a boiler surface cost of about \$61.00 per KW and a pumping cost of \$150/[MM BTU/HR] and the pressure loss through the boiler is about 240 pounds per square foot. LMTD is 6.96F and U is $1217 \text{ BTU/Ft}^2\text{HR.F}$.

Figure 10 shows a graph of cost vs. mean boiling temperature for data as in Figure 9 except a grooved surface is used instead of a smooth surface. In this case the minimum cost, \$690.00/[MM BTU/HR] occurs at a water velocity of 5 F.P.S., and an optimum value of T between 6° and 7°F . The mean boiling temperature is approximately 70°F . The corresponding surface cost, pumping cost, and water cost is about \$37/KW, \$93/[MM BTU/HR], and \$230/[MM BTU/HR] respectively. The pressure loss through the boiler is about 160 p.s.f. LMTD is 6.06F and U is $2080 \text{ BTU/Ft}^2\text{Hr.F}$.

Figure 11 shows a graph of cost vs. mean boiling temperature for data as in Figure 9 except $H = 750 \text{ Btu/Hr/Ft}^2/\text{F}$. This represents the worst case or the lowest value of H that we would expect. All other equations in the program were kept the same. The minimum cost of \$1885/[MM Btu/Hr] occurs at a lower velocity, 5 F.P.S., higher mean boiling temperature, 70.3°F , and a smaller value of $T = 4\text{F}$. The corresponding surface cost, C_9 , pumping cost, C_4 , and water cost, C_6 , is approximately \$151/KW,

\$163/[MM Btu/HR] and \$370/[MM Btu/HR] respectively. The corresponding pressure loss is about 170 p.s.f. LMTD is 7.22 and $U = 439$.

The heat transfer surface area per [MM Btu/Hr] is plotted on Figures 12, 13, and 14, corresponding to the computations for Figures 9, 10 and 11. While these curves define specific heat transfer area, they provide no clue as to what the optimum cost operating conditions are. This is why it was important to apply cost evaluations for, water cost, pumping cost, and effect of net power output as was done to calculate and plot the values of C_8 on Figures 9, 10, and 11.

2.3.9 Design Condition Selection for Condensers

For the condenser, the same factors must be taken into consideration. Greater cold water flow will decrease the need for heat transfer material and vice-versa. These, in turn, will affect the mean condensing temperature and thus the temperature difference across the turbine.

The same basic program is used for the condenser with corrections made due to colder water and condensing instead of boiling. A copy of the program is shown in Figure 15. For the condenser, D , the difference between the condensing temperature and the water temperature leaving each stage, will be changed:

$$D = T_1 - T - 40$$

Here 40° is the temperature of water entering the condenser. The same procedure is used to derive the equation for LMTD, T_2 , and mean condensing temperature, T_3 , in the condenser as it was done for the boiler.

$$T_2 = \frac{T/4}{\ln \left(\frac{D + T/4}{D} \right)}$$

$$T_3 = T_1 - 3 \times T/8$$

where T is water temperature drop through condenser

T_1 is condensing temperature in last chamber

The Reynolds number will be changed due to a different water temperature. With $\rho = 62.4 \text{ lb/ft}^3$ and $\mu = 3.394 \text{ lb/hr/ft}$.

$$R = 16,767 \times V \times W / (W + 1.52)$$

The heat transfer coefficient on the water side for the condenser will also be modified. As before:

$$H_1 = N_{st} G c_p$$

From Reference 4, Figure 28, N_{st} , the Stanton number is given by:

$$N_{st} = \frac{E}{R^{.17}} \left(\frac{\mu}{m} \right)^n \times 10^{-4}$$

In our case $E = 47.7$ at $N_{Pr} = 10.38$ and $n = .36$ for warming the water. Also $G = 3600 \times 62.4 V$ and $c_p = 1.002 \text{ Btu/lb-}^\circ\text{F}$

Combining equations:

$$H_1 = 1100 V/R^{.17}$$

In developing condensing transfer coefficients we found Reference 2 to be most useful in describing the possibilities for improved condensing transfer coefficients. As a base figure we have a direct test experience on halocarbon refrigerants indicating that condensing coefficients of slightly over 2000 BTU's/sq. ft. hr. °F are easily practical with finned tube surfaces. This coefficient is based on projected area. While Reference 2 deals entirely with water coefficients it gives much information on the effect of various designs of surfaces on potential condensing coefficients. The design which appears to be most clearly beneficial for improving condensing coefficients is the fluted or grooved vertical tube design. Reference 6 clearly points out that similar improvements in condensing coefficients can be obtained by applying vertical wires loosely to the condensing surface to act as drain channels. In this reference, Thomas clearly explains why this improvement occurs and shows that the improvements are of various grooved tube designs. It appears to us that the vertical channels added between plates in the condenser passages would be equivalent to the wires as suggested by Thomas, and this is the design that we would propose to use for the condensing surfaces. While it is not possible to predict with any degree of accuracy what condensing coefficients are obtainable we have plotted on Figure 16. the coefficients that we have estimated to be conservatively obtainable.

Figure 16 shows curves taken from Reference 2. The curve marked Gregorig is replotted directly from Reference 2, as is also the curve marked ORNL. We have plotted these curves on log log paper in order to get a clearer and more convenient mathematical definition of values. The curve marked ORNL corresponds to the lowest curve found by experiment in Reference 2. Based on this, we used a value for our condensing coefficients as shown by the straight line and marked "assumed value". This value is considerably below either the Gregorig value or the ORNL value, and we believe should be quite possible of attainment. The mathematical formulation for this curve is:

$$H = 7000/T^{.576}$$

The water channel cross-sectional area, A_1 , will be modified slightly due to change in mass velocity and specific heat.

$$\text{with } \rho = 62.4 \text{ lb/ft}^3$$

$$c_p = 1.00 \text{ Btu/lb-}^\circ\text{F}$$

$$\text{and } Q = 1,000,000 \text{ Btu/Hr}$$

$$\text{and } A_1 = Q / (V \rho \times 3600) / c_p / T$$

$$A_1 = 4.451 / V / T$$

The pressure loss through the heat exchanger will have to be modified due to an increase in density as well as a change in inlet and outlet losses caused by a different working fluid overall passage width.

With $\rho = 62.4 \text{ lb/ft}^3$ and an overall working fluid passage width of .188"

$$P = .97 V^2 \left[F \times A / A_1 + \frac{.0707}{(W + .188)^2} \right]$$

To calculate the cost for heat transfer surface

$$C_2 = (\text{cost/area}) \times \text{Area} = C \times A$$

For pumping cost

$$C_3 = (\text{cost/HP}) \times \text{HP} = C_1 \times P_1$$

Calculation of the cost of cold water is based on assumed cost figures of cold water system components:

Cold Water Pumps & Drive	\$ 2.55 million
Cold Water Pipe	26.25 million
	<u>\$28.80 million</u>

With cold water flow of $29.5 \times 10^8 \text{ lb/hr.}$

$$\text{Cost/[Million lb/hr]} = \$9759$$

with $\text{Btu/lb} = T$

$$\text{Cost/[Million Btu/Hr]} = \frac{\$9759}{T}$$

Since head losses/lb of water tend to increase as the flow decreases, the cost/Btu will increase at a greater rate as T increases and flow decreases. Therefore, T is assumed to be raised to the .7 power and the coefficient will be modified.

$$\text{Cost/[MM Btu/Hr]} = \frac{6211}{T^{.7}}$$

To the cost of cold water must be added the pumping cost outside of the condensers. For this we use an assumed head loss of 4.60 ft.

Pumping HP required will be:

$$HP = \frac{4.60 \times 29.5 \times 10^8}{3600 \times 550 \times .85} = 8063 \text{ HP}$$

$$HP/[MM \text{ lb/Hr}] = \frac{8063 \times 10^6}{29.5 \times 10^8} = 2.73$$

$$HP/[MM \text{ BTU/HR}] = \frac{2.73}{T}$$

$$\text{Cost}/[MM \text{ BTU/HR}] = \frac{2.73 \times C1}{T}$$

The total cost of cold water is thus:

$$C4 = \text{Cost}/[MM \text{ BTU/HR}] = \frac{6211}{T^{.7}} + \frac{2.73 \text{ C1}}{T}$$

C5 is the total cost, the sum of C2, C3, and C4.

$$C5 = C2 + C3 + C4$$

This will again have to be corrected as in the boiler.

$$C6 = C5 \left(\frac{22 - 4.4}{71 - T3 - 4.4} \right)^{1.75} = C5 \left(\frac{17.6}{66.6 - T3} \right)^{1.75}$$

For the condenser, the BTU rejected/Kwh is:

$$\text{BTU/Kwh} = \frac{3413 \times .975}{.0299} = 111,293$$

The cost of the condenser surface/Kwh is thus:

$$C7 = .1113 \times C2 \times C6/C5$$

Figure 17 shows our latest results for cost optimization of the condenser based on the computer program described above. For this case the minimum condenser cost, \$2500/[MM BTU/HR], occurs at a water velocity of 6 F.P.S., a mean condensing temperature of about 48.8°F, and T = 9F. The surface cost, C7; is \$94/Kw, pumping cost, C3, is \$261/[MM BTU/HR], and water cost, C4 is \$1455/[MM BTU/HR]. The pressure loss through the condenser is about 615 pound per square foot. LMTD = 4.02F, and U = 900 BTU/Ft.²hr.F.

The heat transfer area for the condensers is plotted on Figure 18. This corresponds to the data plotted on Figure 17. As might be expected, area increases as condensing temperature range increases. However, turbine power per unit of water flow increases as condensing temperature decreases, and parasitic pumping power decreases as water flow decreases. Figure 17 takes these factors into account to optimize heat transfer and water flow conditions for lowest cost net power output.

2.3.10 Cleaning System

In attempting to find a cleaning system which is simple as well as effective, we have considered mechanical systems such as brush cleaning, rubber-ball cleaning, and variations of these. Also, chemical cleaning systems have been considered.

We have decided that a cleaning system based on methods dating back over a century, that of keeping boiler surfaces clean with the use of sand is most practical for this application.

We have studied the literature and talked with people in the field and we feel that a 5 to 10% sand slurry mixture, run through the heat exchanger, would effectively clean the heat transfer surfaces.

We have discussed the system with Perry Ocean Systems, Inc. and they feel that this should be a very feasible system to design and build.

If it is found that sand is too abrasive and removes the oxide layer from the aluminum, and if this layer cannot be replenished by oxidation, then less abrasive substances could be circulated through the system.

With this type of system it would be possible to add detergents or other chemicals to the sand, if it is found advisable.

2.3.11 Test Heat Exchanger

The design of the test boiler is made using smooth surface on the water side and using the conservative estimate of fouling; $FF = .672/H1$. The preliminary selection of design conditions for the test exchanger are listed in Table VI. In order to meet the space requirements of Argonne, we decided to make the test model approximately one half the length of the design length or 24.5 ft. The height of the test model core is set at 16" and the width will be approximately 16". The required water flow for that configuration is about 4200 GPM. The housing for the exchanger has been designed as well as the jet pump needed to recirculate the liquid and to keep it above saturation pressure in the liquid heating passages. Two liquid heating passages and twelve boiling passages will be required, as well as two passages for liquid recirculation. The eliminator screen for removing liquid from the vapor will be placed above the boiler surface. The liquid will drain off this surface into channels leading back through the boiler; the jet pump being used to recirculate it. This heat exchanger

will be divided into two stages. As discussed previously, this permits the use of less water for a specified heat flow than does a one stage boiler.

Since it appears that grooved surfaces on the water side would definitely aid heat transfer and thus lower costs, provisions could be made so that diagonal grooves could be rolled into the metal during the final rolling process. However, we do not expect grooved surfaces to be available for the first model exchanger.

As explained before, mechanical construction is preferred over brazed construction. We have designed the required construction procedure and necessary hardware for the assembly of the heat exchanger.

The same basic arrangement will be used for the test condenser. Both the width of the working fluid and water passages will probably be changed. The construction of the working fluid passage will be simpler and the eliminator screen and liquid recirculation system will not be required.

There will be two vapor entrance ports on the top of the exchanger and two liquid outlet ports on the bottom instead of two vapor and liquid ports on the top as in the boiler. The working fluid will be flowing top to bottom in the condenser. A sketch of this test heat exchanger is shown in Figure 19.

2.3.12 Small Test Heat Exchanger

It is advantageous to build a small heat exchanger before construction of the larger one for the Argonne facility. We are in the process of designing a 1/2 ft. by 1 ft. by 2 ft. heat exchanger core for this purpose. This will allow us to optimize combinations of materials in the exchanger, construction methods for best sealing and pressure testing, and would be ideal for future heat exchange testing. We have constructed a wooden model as a first step to analyzing joint geometry and assembly procedures.

FIGURE 1

FIGURE 2

THERMAL RESISTANCE vs TIME
A1 6061-T6 PIPES

○ Data for nominal flow = 3 ft/sec

△ Data for nominal flow = 6 ft/sec

FIGURE 3
BOILING MINUS CONDENSING
TEMPERATURE VS. STAGES, LMTD=6F

FIGURE 4
BOILING MINUS CONDENSING
TEMPERATURE VS. STAGES, LMTD=4F

FIGURE 6

```

10 REM BOILER 15 FEB 78 JHA 415
20 REM SAVE DCR $ (1)"BOILER"
30 REM LOAD DCR "BOILER"
40 REM SCRATCH R "BOILER"
50 REM SAVE DCR $ ("BOILER")"BOILER"
60 SELECT R
70 INPUT T, T1, T2, W, C1, C2, V1, V2
80 FOR V=V1 TO V2
90 R=26718*V*W/(W+1.52)
100 H1=1679*V/R+.18
110 D=T1-T-T2
120 T4=T2+.3*T/D
130 T3=T/4/LOC((D+T/4)/D)
140 H=11349*(T3/10)+.3028
150 F1=.0003/V
160 R1=1/H1+F1+.0000654+1/H
170 T5=T3/H/R1
180 H=11349*T5+.3028
190 R1=1/H1+F1+.0000654+1/H
200 U=1/R1
210 A=1000000*R1/T3
220 A1=4.459/V/T
230 N=A1/W*.6
240 L=A/4/N
250 F=.03875/R+.193
260 P=.97*V+2*(F*A/A1+.13/(W+.255)+2)
270 P1=P*U*A1/467.5
280 C3=C1*A
290 C4=C2*P1
300 C6=1143/T+.0515*C2/T
310 C7=C3+C4+C6
320 C8=C7*(17.6/(T4-53.4))+1.75
330 C9=C8/C7*.1141*C3
340 SELECT PRINT 215(80)
350 PRINT " V T T1 T2 T3 T4 H W R H1 F1 U
      A A1"
360 PRINT USING 370, V, T, T1, T2, T3, T4, H, W, R, H1, F1, U, A, A1
370 X## # ##.## ## ##.## ##.## ##### #.## ##### ###.##### ### ##
#.#
380 PRINT " N L P P1 C1 C2 C3 C4 C6 C7 C8
      C9"
390 PRINT USING 400, N, L, P, P1, C1, C2, C3, C4, C6, C7, C8, C9
400 X#.# ##.# ###.# ##.## ##.### ##.### ##### ##### ##### ##### ##
#.#
410 PRINT
420 NEXT V
430 STOP
440 GOTO 70

```

Computer Program • Boiler (Smooth Surface)

FIGURE 7

```

10 REM BOILER 15 FEB 78 JHA 430
20 REM SAVE DCR $ (1)*"BOILER"
30 REM LOAD DCR "BOILER"
40 REM SCRATCH R "BOILER"
50 REM SAVE DCR $ ("BOILER")*"BOILER"
60 SELECT R
70 INPUT T, T1, T2, W, C1, C2, V1, V2
80 FOR V=V1 TO V2
90 R=26710*V*W/(W+1.52)
100 H1=5037*V/R+.18
110 D=T1-T-T2
120 T4=T2+.3*T/B
130 T3=T/4/LOG((D+T/4)/D)
140 H=11349*(T3/10)+.3028
150 F1=.0003/V
160 R1=1/H1+F1+.0000654+1/H
170 T5=T3/H/R1
180 H=11349*T5+.3028
190 R1=1/H1+F1+.0000654+1/H
200 U=1/R1
210 A=1000000*R1/T3
220 A1=4.459/U/T
230 N=A1/W*.6
240 L=A/4/N
250 F=1.192/R+.43
260 P=.97*V+.2*(F*A/A1+.13/(W+.255)+2)
270 P1=P*V*A1/467.5
280 C3=C1*A
290 C4=C2*P1
300 C6=1143/T+.8515*C2/T
310 C7=C3+C4+C6
320 C8=C7*(17.6/(T4-53.4))+1.75
330 C9=C8/C7*.1141*C3
340 SELECT PRINT 215(80)
350 PRINT " V  T  T1  T2  T3  T4  H  W  R  H1  F1  U
      A  A1"
360 PRINT USING 370, V, T, T1, T2, T3, T4, H, W, R, H1, F1, U, A, A1
370 Z##  # ##.##  ##  ##.##  ##.##  ##.##  #####  ##.##  #####  #####  #####  #####
      #.##
380 PRINT "  N      L      P      P1  C1  C2  C3  C4  C6  C7  C8  C9"
390 PRINT USING 400, N, L, P, P1, C1, C2, C3, C4, C6, C7, C8, C9
400 Z#.##  ###.##  #####.##  ##.##  ##  #####  #####  #####  #####  #####  #####  ##
      #.##
410 PRINT
420 NEXT V
430 STOP
440 GOTO 70

```

Computer Program - Boiler (Grooved Surface)

FIGURE 8

Surface Boiling Heat Transfer Coefficient vs. Temperature (Linde)

From FIGURE 10, (Reference 11)

FIGURE 9

7/11/11
7/20/12
7/21/12

FIGURE 10

FIGURE 11

PROGRAM P. 1034
 DATA PP. 1034-1070
 R=22
 SMOOTH SURFACE
 $H=11349 * T^{1.3028}$
 $C1=4.00/SQ-FT$
 $C2=400/HP$
 WATER TEMP=80F
 $WFO=75$
 $V=OPTIMUM=7 F.P.S.$
 $FF=0003/V$
 $C8=C7*(17.6/(T4-53.4))^{1.75}$
 $T4$ = Water Temperature Drop thru Boiler °F

HEAT TRANSFER AREA (MMBTU/HR) - (SQ.FT.)

T4 = MEAN BOILING TEMP (°F)

FIGURE 15

8/11/74
TJB

$CB = C7 * ((T4 - 53.4) / (T1 - 75))^{1.75}$
 PROGRAM P-1156
 DATA PF-1156-1191
 R-22
 GROOVED SURFACE
 $HE = 1349 * T1^{.3028}$
 $C1 = 4.00 / SQ\ FT$
 $C2 = 400 / HP$
 WATER TEMP = 80F
 $W = 0.75$
 $VEOPTIMUM = 5\ F.P.S.$
 $FE = .0003 / V$
 T = Water Temperature Drop thru
 Boiler, F

FIGURE 14

TLD

$C8 = C7 * (17.6 / (T - 53.4))^{1.75}$
 PROGRAM P. 1120
 DATA P. 1120 1155
 R. 122
 SMOOTH SURFACE
 $h = 750 \text{ BTU/SQ. FT. HR. F}$
 $C1 = 4.00/\text{SQ. FT.}$
 $C2 = 400/\text{HP}$
 WATER TEMP = 80F
 $W = 0.75$
 $V = \text{OPTIMUM} = 5 \text{ F. P. S.}$
 $FF = 0.003/V$
 C. = Water Temperature Drop thru
 Boiler F

85/

FIGURE 15

```

10 REM CONDEN 15 JAN 78 JHA 415
20 REM SAVE DCR $ (1) "CONDEN"
30 REM LOAD DCR "CONDEN"
40 REM SCRATCH R "CONDEN"
50 REM SAVE DCR $ ("CONDEN")"CONDEN"
60 SELECT R
70 INPUT T, T1, W, C, C1, V1, V2
80 FOR V=V1 TO V2
90 D=T1-T-40
100 T2=T/4/LOG((D+T/4)/D)
110 T3=T1-3*T/8
120 R=16767*V*W/(W+1.52)
130 H1=1100*V/R+.17
140 F1=.0003/V
150 H=7000/(T2/9)+.576
160 R1=1/H1+F1+.0000654+1/H
170 T4=T2/H/R1
180 H=7000/(T4)+.576
190 R1=1/H1+F1+.0000654+1/H
200 U=1/R1
210 A=1000000*R1/T2
220 A1=4.451/V/T
230 N=6*A1/W
240 L=A/4/N
250 F=.03875/R+.193
260 P=.97*V+2*(F*A/A1+.0707/(W+.188)+2)
270 P1=P*V*A1/467.5
280 C2=C*A
290 C3=P1*C1
300 C4=6211/T+.7+2.73*C1/T
310 C5=C2+C3+C4
320 C6=C5*(17.6/(66.6-T3))+1.75
330 C7=.1107*C2*C6/C5
340 SELECT PRINT 215(80)
350 PRINT " V      T      T1      W      T2      T3      H      R      H1      F1
      A      A1"
360 PRINT USING 370, V, T, T1, W, T2, T3, H, R, H1, F1, U, A, A1
370 Z##.##  ##.##  ###.##  ##.##  ##.##  #####  #####  ###.##.#####  ###
###
380 PRINT " N      L      P      P1      C      C1      C2      C3      C4      C5
      7"
390 PRINT USING 400, N, L, P, P1, C, C1, C2, C3, C4, C5, C6, C7
400 Z#.##  ##.##  ###.##  ##.##  ##.##  ##.##  #####  #####  #####  #####  i
410 PRINT
420 NEXT V
430 STOP
440 GOTO 70

```

Computer Program - Condenser

K&E LOGARITHMIC 477923
3 X 3 CYCLES 33114949
REUSFEL • 001111 20

FIGURE 17

FIGURE 18

PROGRAM P.367

DATA PF.367-416

R=22

SMOOTH SURFACE

H=7000/T^{1.576}

C=3.50/SQ.FT.

C1=400/HP

WATER TEMP=40F

VE=375

V=OPTIMUM=6 F.P.S.

FP=2003/V

C5=C5*((17.6/(65.6-T3))^{1.75}

T = Water Temperature

Drop Thru Condenser °F

HEAT TRANSFER AREA (MMBTU/HR) (SQ.FT.)

800
700
600
500
400
300
200
100
000

Condenser Surface Area vs. Mean
Condensing Temperature
4 Stages

MEAN CONDENSING TEMP (F)

FIGURE 19

TEST HEAT EXCHANGER
ARGONNE NATIONAL LABORATORY

MANUFACTURER	PRODUCT	METAL				WORKING FLUID			SEAWATER
		AL	CU-NI	ST	TI	AM	HA	HY	
Chem Seal	CS3201A	X	X	X	X			X	
Bonded Products	Episeal-Universal Resin	X	X	X	X	X	X	X	X
Calbar	Dow Corning Silicone Sealants	X	NO	X	X				
Babbitt	Plas-Dux		X	X					X
Goal Chemical	6C-401B1, GC408					X	X	X	X
Palmer Product	699M	X		X					
Palmer Product	P-752		X						
3M	801	X		X				X	X
B.F. Goodrich	AG10B, A821B	X	X	X	X	X	X	X	
Goodyear	Plionail								
Dural	Dural Seal							X	X
Dural	Dural 103	X	X	X	X	X	X	X	
National Adhesives	Duribbon 1082	X	X	X	X				
National Adhesives	JS141A	X							
Norton	V-384, V-414-VS34	X	X	X	X				
Uniseal	SE420, SE424	X	X	X	X	X		X	X
Woodhill Permatex	F.A.G. #6								X
Dev Con	EZ Gasket 10	X	X	X	X	NO	NO	X	
Dev Con	Devcon F	X		X		X	X	X	
Loctite	504, 510, 515	X	X	X	X		X	X	X
Loctite	262, RC-40(640)	X	X	X	X		X	X	X
Loctite	360	X	X	X	X		X		X
Oil Center Research, Inc.	Lubon #404								
Thiem Corp.	Thiemite 990, 441.3 02-266.5								
W. L. Gore	Gore-Tex	X	X	X	X	X	X	X	X

TABLE II

TABLE III

HEAT EXCHANGER MANUFACTURERS
AND INTERESTED MANUFACTURERS

APV Company Inc.	Thermxchanger, Inc.
Cleaver Brooks	David S. Ude11, Inc.
Dean Products Inc.	Whitlock Mfg. Co.
Kintex, Inc.	Sub. of Ametek Inc.
Niagara Blower Co.	Standard Refrigeration Co.
Tranter Inc.	Alpha Electric Refrigeration Co.
CTI-NUCLEAR, INC.	Koolant Koolers, Inc.
Edwards Engineering Corp.	Killebrew Engineering Corp.
C.B. Thaw Associates Inc.	Paul Mueller Co.
Happy Division	Atlas Industrial Manufacturing Co.
Therma Technology Inc.	Agric Machinery Corp.
Device Engineering Co.	Lawler Company
H & H Associates	M & T Chemicals Inc.
Patterson-Kelley Co. Inc.	Electro Impulse, Inc.
Valad Electric Heating Inc.	Kearney Industries, Inc.
Bemco Inc.	Heat Pipe Corp. of America
Stewart Warner	Sen-Dure Products, Inc.
South Wind Div.	Yula Corporation
Union Carbide Corp.	Basco Div.
American Heat Reclaiming Corp.	American Precision Industries, Inc.
Chester-Jensen Co.	Bos-Hatten, Inc.
American Standard	Hughes-Treitler Mfg. Corp.
Power-Control Group	L.I. Metal Inc.
Richmond Engineering Co. Inc.	Graham Manufacturing Co. Inc.
Kobe Steel, LTD.	Crane Co.
Chamberlain Manufacturing Corp.	Doyle & Roth Mfg. Co., Inc.
Conseco Div.	De Laval Separator Co.
Western Precipitation Div.	Brighton Corporation
Joy Manufacturing Co.	Ellis & Watts Co.
Hex Industries, Inc.	
Subsidiary of Hoyt Mfg. Corp.	

TABLE IV
HEAT EXCHANGER MANUFACTURERS

Vulcan Manufacturing Co.
A Sub. of Zurn Industries, Inc.

Nixon Ribcoil Panel Corp.

Carborundum Co.
The Process Equipment Branch
Graphite Products Div.

Astro Metallurgical Corp.

Brown Fintube Co.

Harris Thermal Transfer Products Inc.

Ionics Inc.

Schutte and Koerting Co.
A Sub. of Ametek Inc.

The Sims Company Inc.

Andale Co.

Thermal Transfer Co.

The American Schack Co. Inc.

Dormont Manufacturing Co.

Edwin L. Wiegand, Div.
Emerson Electric Co.

Fabricating Engineering Co.
Div. of Fecor Industries Ltd.

Heatron Inc.

Aerofin Corp.

Nixon Plate Coils Ltd.

Long Manufacturing Div.
Borg-Warner Ltd.

Bohn Heat Transfer Div.
Gulf & Western Industries, Inc.

Dunham-Bush Inc.

Marlo Coil Products

Coupling Corporation of America

TABLE V
Respondees To Survey

NAME	Mechanical Plate	Brazed Plate	Panel Coil	Fabricator
1. American Standard	x			
2. Chester-Jensen	x			
3. Heatron				
4. Carborundum				
5. Brown Fintube Co.				
6. Heat Pipe Corp. of Am.				
7. Linco Manufacturing Corp.		x		
8. Bemcoil			x	
9. Nixon Plate Coils			x	
10. Tranter	x			
11. Kintex, Inc.		x		
12. Patterson-Kelley Co.	x			
13. APV	x			
14. American Heat Reclaim- ing Corp.	x			
15. Dean Products, Inc.			x	
16. Long Division of Borg Warner		x		
17. Electro Impulse, Inc.				
18. Niagra Blower				
19. Stewart-Warner South Wind Div.		x		
20. Kobe		x		
21. CTI-Nuclear, Inc.				x
22. Edwards Engineering				x
23. Young Radiator				
24. Coupling Corporation of America				x

TABLE VI
TEST HEAT EXCHANGER DESIGN CONDITIONS

V	Water Velocity	7 ft. per sec.
T	Heat Extracted from water	5 BTU/HR
	Water Inlet temperature	80°F
	Static heat penalty	.26°F
T1	Effective water inlet temperature	79.74°F
T4	Mean boiling temperature for 4 stages	70.8°F
T3	LMTD	6.34°F
H	Boiling heat transfer coefficient	9,338 BTU/HR/Ft. ² /°F
H1	Water side heat transfer coefficient	1525 BTU/HR/Ft. ² /°F
F1	Fouling Factor	.000440 Ft. ² HR°F/BTU
U	Overall heat transfer coefficient	814 BTU/HR/Ft. ² /°F
L	Length of water passage	47.4 ft.
P	Pressure drop thru water passage	317 lbs/Ft. ²
A	Unit area of surface	193 Sq.Ft./[MM BTU/HR]

APPENDIX I

Sample Calculations - Design Condition Selection For Boilers

To generate a set of curves for cost vs. mean boiling temperature as shown in Figure 9, values of T from T = 3 to T = 7 and values of T2 from T2 = 63 to T2 = 71 are used. In this sample calculation, T = 6 and T2 = 68. T1, the effective water temperature entering the boiler is 79.74°F.

W, the passage width in inches is set at .75". The estimated cost for heat transfer surface, C1, is \$4.00/Sq.Ft. and the pumping cost is \$400/HP. The computations are made for values of velocity, V, from V1 = 4 Ft./Sec to V2 = 10 Ft./Sec. at 1 Ft./Sec. increments. For this calculation, V = 7 Ft./Sec. is used. The Reynolds number, R, with V = 7 Ft./Sec. and W = .75" is:

$$R = 26,718 (7) \times \left(\frac{.75}{.75 + 1.52} \right) = 61792$$

Using the value of R calculated above and V = 7 Ft./Sect.

$$H1 = 1679 (7) / (61792)^{.18} = 1613 \text{ BTU/HR/Ft.}^2/^{\circ}\text{F}$$

with T1 = 79.74F, T = 6F, T2 = 68F

$$D = 79.74 - 6 - 68 = 5.74^{\circ}\text{F}$$

$$T4 = 68 + 3(6)/8 = 70.2^{\circ}\text{F}$$

Then LMTD = T3 is given by:

$$T3 = 6/4/\ln \left(\frac{5.74 + 6/4}{5.74} \right) = 6.46^{\circ}\text{F}$$

Fouling Factor, F1, is:

$$F1 = .0003/7 = .000042$$

The iterative technique is used to calculate H,

$$H = 11,349 \times (T3/10)^{.3028}$$

$$\text{or } H = 11,349 \times (6.46/10)^{.3028} = 9942 \text{ BTU/HR.FT.}^2/^{\circ}\text{F}$$

$$\text{Then } R1 = (1/1613) + .000042 + .0000654 + (1/9942)$$

$$\text{or } R1 = .00082 \text{ HR.FT.}^2/^{\circ}\text{F/BTU}$$

T5 is then calculated from T5 = T3/R/H

$$T5 = 6.46 / (.00082 \times 9942) = .79^{\circ}\text{F}$$

$$\text{Finally, } H = 11,349 (.79)^{.3028} = 10,567 \text{ BTU/HR.FT.}^2/^{\circ}\text{F}$$

Now a final value of R1 can be computed.

$$R1 = 1/H1 + F1 + .0000654 + 1/H$$

$$\text{or } R1 = (1/1613) + .000042 + .0000654 + (1/10,567)$$

$$R1 = .00082 \text{ HR.FT.}^2\text{F/BTU}$$

The overall conductance, U, is given by:

$$U = 1/R1 = 1/.00082 = 1215 \text{ BTU/HR.FT.}^2\text{F}$$

The required heat transfer area for 1 million BTU/HR, A, is:

$$A = (1,000,000 \times R1)/T3 = 1,000,000 \times .00082/6.46^\circ\text{F}$$

$$A = 127 \text{ FT.}^2$$

The necessary water passage cross sectional area per million BTU/HR, A1, is:

$$A1 = 4.459/V/T = 4.459/7/6 = .10 \text{ Ft.}^2$$

The number of passages needed, N, is given by:

$$N = 6xA1/W = 6(.10)/.75 = .84$$

The passage length, L, is then:

$$L = A/4/N = 127/4/.84 = 37.5 \text{ FT.}$$

The friction factor, F, is a function of the Reynolds number, R, and is given by:

$$F = .03875/R^{.193} = .03875/61792^{.193}$$

$$F = .0042$$

With this the pressure loss, P, can be computed:

$$P = .97 V^2 \left[\frac{F \times A}{A1} + \frac{.13}{(W + .255)^2} \right]$$

$$P = .97 (7^2) \left[\frac{.0042 \times 127}{.10} + \frac{.13}{(.75 + .255)^2} \right]$$

$$P = 263 \text{ lbs/Ft.}^2$$

The pumping power required, P1, is:

$$P1 = P \times V \times A1/467.5$$

$$P1 = (263) (7) (.10)/467.5$$

$$P1 = .39 \text{ HP}$$

The total heat transfer cost per million BTU/HR, C3, is:

$$C3 = C1 \times A = \$4.00/\text{FT.}^2 \times 127 \text{ FT.}^2 = \$508$$

The total pumping cost is, C4:

$$C4 = C2 \times P1 = \$400/\text{HR} \times .39 \text{ HP} = \$160$$

The cost of water/million BTU/HR, C6, is computed by:

$$C6 = 1143/T + .8515 \times C2/T$$

$$C6 = 1143/6 + .8515 \times 400/6 = \$247$$

C7, the total boiler/cost million BTU/HR

$$C7 = C3 + C4 + C6 = \$508 + \$160 + \$247$$

$$C7 = \$920$$

The corrected boiler cost/million BTU/HR, C8, is given by:

$$C8 = C7 \times \left(\frac{17.6}{T4 - 53.4} \right)^{1.75}$$

$$C8 = \$920 \left(\frac{17.6}{70.2 - 53.4} \right)^{1.75} = \$996$$

The cost of the boiler surface/KW, C9, is:

$$C9 = C8/C7 \times .1141 \times C3$$

$$C9 = 996/920 \times .1141 \times 508 = \$62.7$$

The same calculations are made for each value of V from V1= 4 FT/Sec. Then, the velocity giving the lowest costs is considered as the optimum velocity and the costs at this velocity are plotted versus the mean boiling temperature, T4.

As seen in Figure 9, the value of C8 computed above, \$996, is plotted versus the mean boiling temperature, 70.2F, for V = 7 FT/Sec and T = 6F. To generate a complete curve, costs are plotted versus various mean boiling temperatures T4 = 65.2F to T4 = 72.2F.

A similar procedure is used for condenser cost optimization.

APPENDIX II

EFFECT OF HEAT EXCHANGER STAGING ON PLANT PERFORMANCE

SYMBOLS USED IN APPENDIX II

D_b, D_c	Temperature difference between the working fluid and the water leaving each stage of the boiler (condenser)
e	Cycle efficiency
LMTD	Log mean temperature difference
$\bar{T}_b$	Mean boiling temperature
$\bar{T}_c$	Mean condensing temperature
T_{f1b}, T_{f1c}	Temperature of working fluid in first stage of boiler (condenser)
T_{f2b}, T_{f2c}	Temperature of working fluid in last stage of boiler (condenser)
T_b, T_c	Temperature drop of water thru boiler (condenser)
T_{wb}, T_{wc}	Temperature of water entering the boiler (condenser)

One important way to increase the effective temperature difference, or LMTD is to use multiple fluid circuits in parallel. We can illustrate this by calculating LMTD for two fluid circuits as illustrated on Fig. A-1. The mean boiling temperature is 72 F but by using two fluid circuits in the boiler we get two boiler temperatures of 72.75 F and 71.25 F. Under these conditions we get

$$LMTD = \frac{(82 - 72.75) - (80.5 - 72.75)}{\ln \frac{(9.25)}{(7.75)}} = 8.478$$

For one circuit.

$$LMTD = \frac{(82 - 72) - (79 - 72)}{\ln \frac{(82 - 72)}{(79 - 72)}} = 8.411$$

The ratio of LMTD's for two circuits to one circuit is

$$\frac{8.478}{8.411} = 1.0080$$

In the condenser we can use two condensing temperatures of 53.5F and 50.5 F instead of the mean temperature of 52 F.

$$\text{Then } LMTD = \frac{(50.5 - 43) - (50.5 - 46)}{\ln \frac{(7.5)}{(4.5)}} = 5.873$$

For one circuit

$$LMTD = \frac{(52 - 43) - (52 - 49)}{\ln \frac{(52 - 43)}{(52 - 49)}} = 5.461$$

The ratio of LMTD's for two circuits to one circuit is

$$\frac{5.873}{5.461} = 1.0754$$

There is also a very slight improvement in cycle efficiency attained by the use of multiple fluid circuits, as illustrated by the following calculation of ideal cycle efficiency.

For the single circuit

$$e = \frac{T_1 - T_2}{T_1} = \frac{72 - 52}{460 + 72} = .0375940$$

For the dual circuit

$$e = 1/2 \left(\frac{72.75 - 53.5}{532.75} + \frac{71.25 - 50.50}{531.25} \right) = .0375960$$

$$\text{Ratio} = 1.000053$$

Because of the efficiency improvement there is also a very slight reduction in heat rejection load as illustrated by the following calculation:

For the single circuit $\frac{1}{e} - 1 = 25.599989$

For the dual circuit $\frac{1}{e} - 1 = 25.598574$

Ratio = 1.000055

While the latter two effects are negligible, they illustrate that there are three benefits from using multiple fluid circuits.

First, it increases mean temperature difference across the heat exchanger.

Second, it improves cycle efficiency.

Third, it reduces heat rejection load.

The effect of staging on plant power output can also be seen by comparing the temperature heads across the turbine for one and two stage heat exchangers.

The temperature distribution using a one stage heat exchanger is shown in Figure A-2.

The temperature distribution using two stage heat exchangers is shown in Figure A-3.

ΔT , the temperature difference across the turbine in a single stage heat exchanger, is given by:

$$\Delta T = T_{f_{2b}} - T_{f_{2c}}$$

$$\text{with } T_{f_{2b}} = T_{w_b} - T_b - D_b$$

$$\text{and } T_{f_{2c}} = T_{w_c} + T_c + D_c,$$

$$\Delta T = T_{w_b} - T_b - D_b - T_{w_c} - T_c - D_c$$

In a two stage heat exchanger:

$$T_{f_{1b}} = T_{w_b} - T_{b/2} - D_b$$

$$T_{f_{2c}} = T_{w_c} + T_c + D_c$$

Thus ΔT_A , the temperature difference across the first stage turbine is:

$$\Delta T_A = T_{f_{1b}} - T_{f_{2c}} = T_{w_b} - T_{b/2} - D_b - T_{w_c} - T_c - D_c$$

Clearly $\Delta T_A > \Delta T$

with $T_{f_{2b}} = T_{w_b} - T_b - D_b$

and $T_{f_{1c}} = T_{w_c} + T_{c/2} + D_c$

ΔT_B , the temperature difference across the second stage turbine is:

$$\Delta T_B = T_{f_{2b}} - T_{f_{1c}} = T_{w_b} - T_b - D_b - T_{w_c} - T_{c/2} - D_c$$

Thus $\Delta T_B > \Delta T$

Therefore, in a two stage system, there are two turbines each operating on higher temperature heads than the single turbine in a single stage system. This increased temperature head across the turbines will lead to an increase in overall plant output.

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TEMPERATURE DISTRIBUTION
ONE STAGE

FIGURE A-2

TEMPERATURE DISTRIBUTION
TWO STAGES

FIGURE A-3

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