THERMOECONOMIC ANALYSIS OF ENVIRONMENTAL VAPOR POWER SYSTEMS

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THESIS

THERMOECONOMIC ANALYSIS OF ENVIRONMENTAL VAPOR POWER SYSTEMS

by

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ABSTRACT

A method is presented for determining the relationships between the costs and technical performance of environmental vapor power systems in a manner which permits fundamental design specifications to be made optimally with respect to overall system lifetime costs. Means of applying optimization techniques for large scale systems to the thermoeconomic analysis of environmental vapor power systems are described and demonstrated with a simplified sample model. A sequential unconstrained minimization algorithm is employed for overall system design optimization.

TABLE OF CONTENTS

LIST OF TABLES

Appendix

LIST OF FIGURES

LIST OF SYMBOLS

Material Properties		Units
ρ	working fluid density	lbm/ft^3
ρ_H	seawater density	lbm/ft^3
C_p	working fluid heat capacity	Btu/lbm-°F
C_{pH}	seawater heat capacity	Btu/lbm-°F
μ	working fluid viscosity	lbm/ft-hr
μ_H	seawater viscosity	lbm/ft-hr
k	working fluid conductivity	Btu/hr-ft-°F
k_H	seawater conductivity	Btu/hr-ft-°F
k_w	tube wall conductivity	Btu/hr-ft-°F
P_r	working fluid Prandtl number	dimensionless
P_{rH}	seawater Prandtl number	dimensionless

Dimensions		
d	tube inside diameter	ft
t	tube wall thickness	ft
S_T	tube bank transverse spacing	ft
S_L	tube bank longitudinal spacing	ft
ℓ	heat exchanger length	ft
w	heat exchanger width	ft
a	heat exchanger height	ft
A_o	tube outside area	ft^2
A_i	tube inside area	ft^2
A_{mw}	tube wall median area	ft^2

Heat Exchanger Characteristics		Units
A_H	area	ft^2
U_H	overall heat transfer coefficient	$Btu/ft^2\text{-}hr\text{-}°F$
N_H	number of thermal units	dimensionless
Γ_H	intermediate performance variable	dimensionless
θ_H	rate/capacity ratio	dimensionless
h_o	outside heat transfer coefficient	$Btu/ft^2\text{-}hr\text{-}°F$
h_i	inside heat transfer coefficient	$Btu/ft^2\text{-}hr\text{-}°F$
R_F	fouling resistance	$ft^2\text{-}hr\text{-}°F/Btu$
R_w	tube wall resistance	$ft^2\text{-}hr\text{-}°F/Btu$
R_i	inside resistance	$ft^2\text{-}hr\text{-}°F/Btu$
R_o	outside resistance	$ft^2\text{-}hr\text{-}°F/Btu$
N'	number of tube rows	dimensionless
N_T	total number of tubes	dimensionless
N_u	working fluid Nusselt number	dimensionless
N_{uH}	seawater Nusselt number	dimensionless

Temperatures

T_H	working fluid: exchanger outlet	°F
T_c	working fluid: exchanger inlet	°F
T_{HE}	hot seawater	°F
T_{CE}	cold seawater	°F

Power

$\dot{G}$	gross power output	MW
E	net power output	MW
W_p	working fluid pumping power	KW
W_H	seawater pumping power	KW

Flow parameters

$\dot{m}$	working fluid flow rate	lbm/hr
$\dot{m}_H$	seawater flow rate	lbm/hr
G_H	working fluid mass flux	$lbm/hr\text{-}ft^2$
U_H	seawater velocity	ft/hr

Flow parameters (cont'd)		Units
R_{eH}	seawater Reynolds number	dimensionless
R_{ec}	working fluid Reynolds number	dimensionless
DP	friction head, working fluid	lbf/ft^2
DP_H	friction head, seawater	lbf/ft^2
f	friction factor, working fluid	dimensionless
f_H	friction factor, seawater	dimensionless

Costing Factors

π	profit	\$
z_1	capital cost, heat exchanger	\$
z_{21}	capital cost	\$
z_{22}	capital cost	\$
z_2	capital cost, all pumps	\$
COST	total zone 1 cost	\$
p_o	market price of power	\$-hr/ft-lbf
C_{H1}	capacity-head product	$gal\text{-}lbf/min\text{-}in^2$
C_{H2}	capacity-head product	$gal\text{-}lbf/min\text{-}in^2$
B_{C1}	base cost seawater pump	\$
B_{C2}	base cost working fluid pump	\$
P_{C1}	intermediate cost variable	\$
P_{C2}	intermediate cost variable	\$
F_{M1}	material factor	dimensionless
F_{M2}	material factor	dimensionless
D_{eX}	price index	dimensionless

Miscellaneous

ψ	energy conversion factor	MW-hr/lbm-°F
η	working fluid pump efficiency	dimensionless
η_H	seawater pump efficiency	dimensionless
X	vector of decision variables	various

I. INTRODUCTION

A. BACKGROUND

The motivation for developing new energy technology, long forecast by such researchers as Putnam [1], has now become so widely understood as to require no elaboration here. Research and development efforts are proceeding on a broad front in search of alternatives to the conventional non-renewable fossil fuels and potentially hazardous fission processes.

One class of proposals seeks to extract useful energy from sources existing in nature. With the exception of geothermal energy, virtually all of these rely ultimately on some phenomenon associated with receipt by the earth of energy radiated from the sun. A sub-class of these "environmental" energy systems utilize vapor cycles in mechanizing the conversion from the diffuse heat sources found in nature into the more concentrated and transportable energy forms required in many applications.

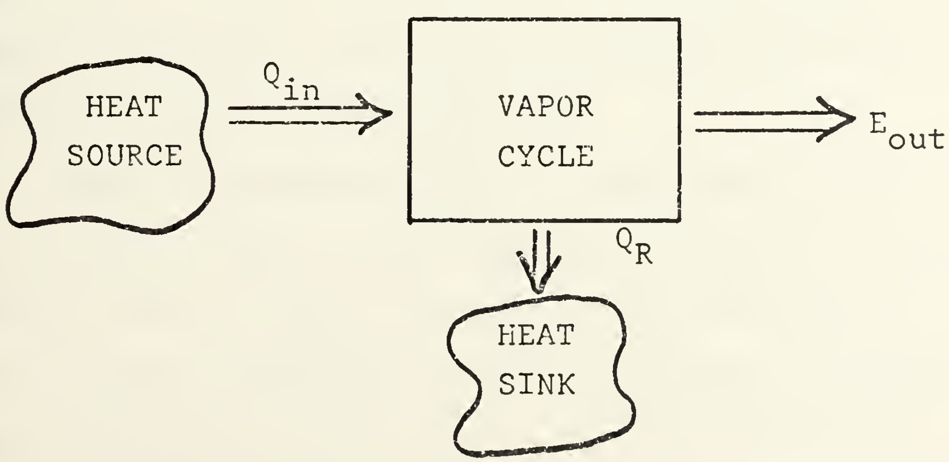

FIGURE 1.

Figure 1 diagrams the fundamental concept upon which these environmental vapor power systems depend. Heat is extracted from some natural source of elevated temperature (geothermal wells, direct solar collectors, hot seawater, etc.) and transferred to a working fluid. Devices suitable to the application convert the thermodynamically available portion of the available heat into other forms of energy, and the unavailable energy is rejected to a thermal sink. Both open and closed vapor cycles are possible [2], [3] and the products of conversion can take many forms, including electricity and fuels such as hydrogen, methanol, and ammonia [4].

One widely used vapor cycle is the closed Rankine cycle [5], shown schematically in Figure 2.

In addition to the conservation of chemical fuels, certain of the proposed methods of harnessing energy hold promise of significant additional advantages. It is expected that their effects on the environment will be relatively benign [6], particularly in terms of atmospheric and thermal pollution. They would cause no addition to the total heat burden at the earth's atmosphere and rely on sources which are continuously renewed by natural processes [7].

There is another class of vapor power cycles which, although not always exploiting environmental energy sources, shares enough of the operating characteristics of those that do to warrant mention as a group amenable to the type of analysis discussed in this paper. These are the "bottoming" cycles for extracting power from the still energetic discharges of

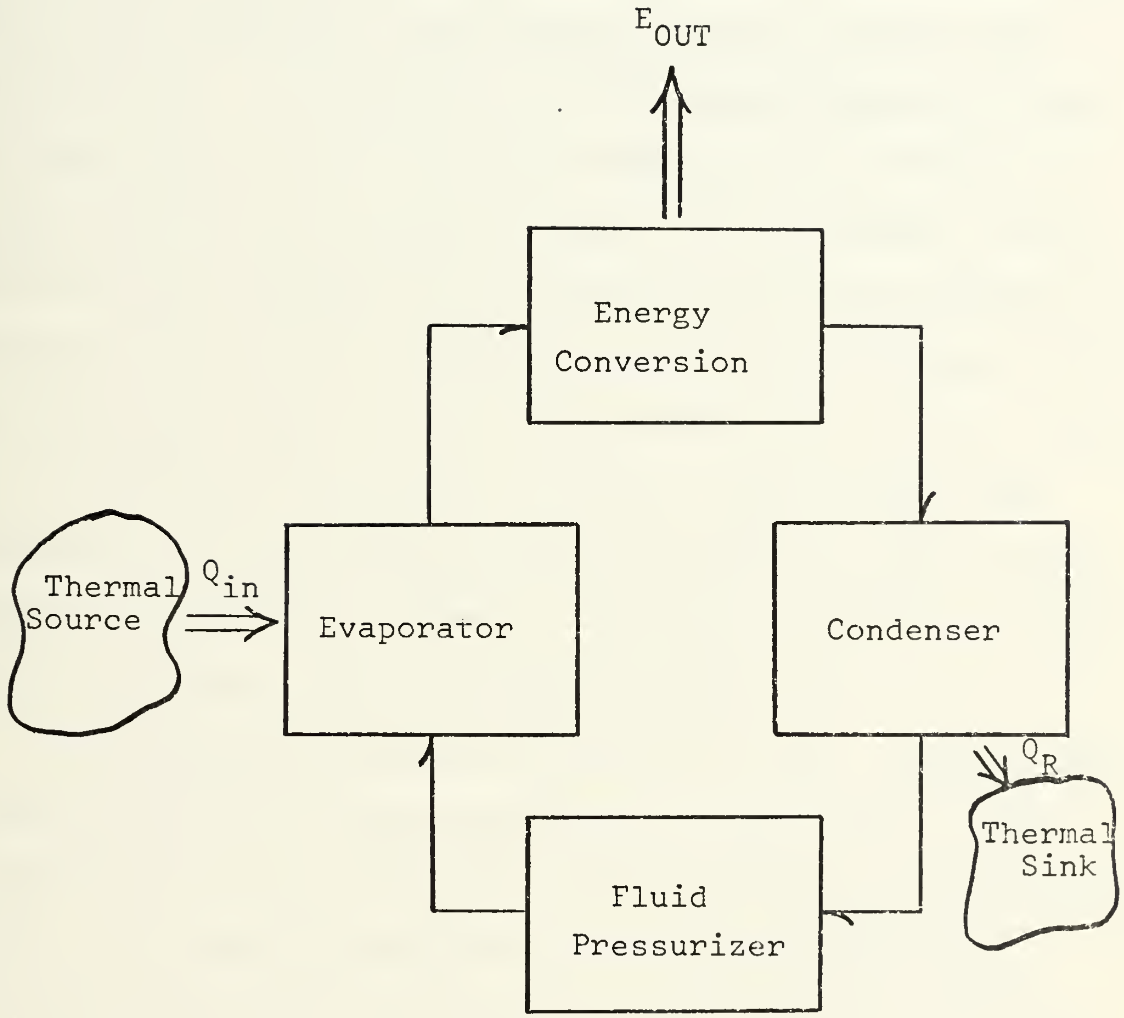

FIGURE 2. Closed Rankine Cycle

geothermal, nuclear, and chemically fueled plants. Figure 3 shows two general types of these "waste heat"cycles.

The essential feature which differentiates both the environmental and waste heat vapor systems from conventional ones is the relatively low thermal potential within which they operate. A consequence of this characteristic is that the size scale of all the cycle components is increased in comparison with conventional plants. Heat exchange surface areas must be enlarged for sufficient heat to be transferred through small driving potentials, and with less energy available from each unit of fluid circulated, a far greater volume rate of working fluid must be cycled. Pumps, pipes, and conversion devices such as turbines all grow in size as the temperature difference between the source and sink is reduced while the energy product is held constant.

Viewed fundamentally, environmental power systems employ technology which has been available for many years. Many concepts have been tested with working models or demonstration plants, and some are employed presently on a small scale. Although significant technical problems arise in connection with specific applications, these do not appear to be permanent obstacles. Net energy asessments appear favorable and questions of material availability and local adverse environmental effects seem amenable to solution [6].

The primary question which will determine when environmental energy sources will be exploited on a scale large enough to significantly affect the energy market is that of

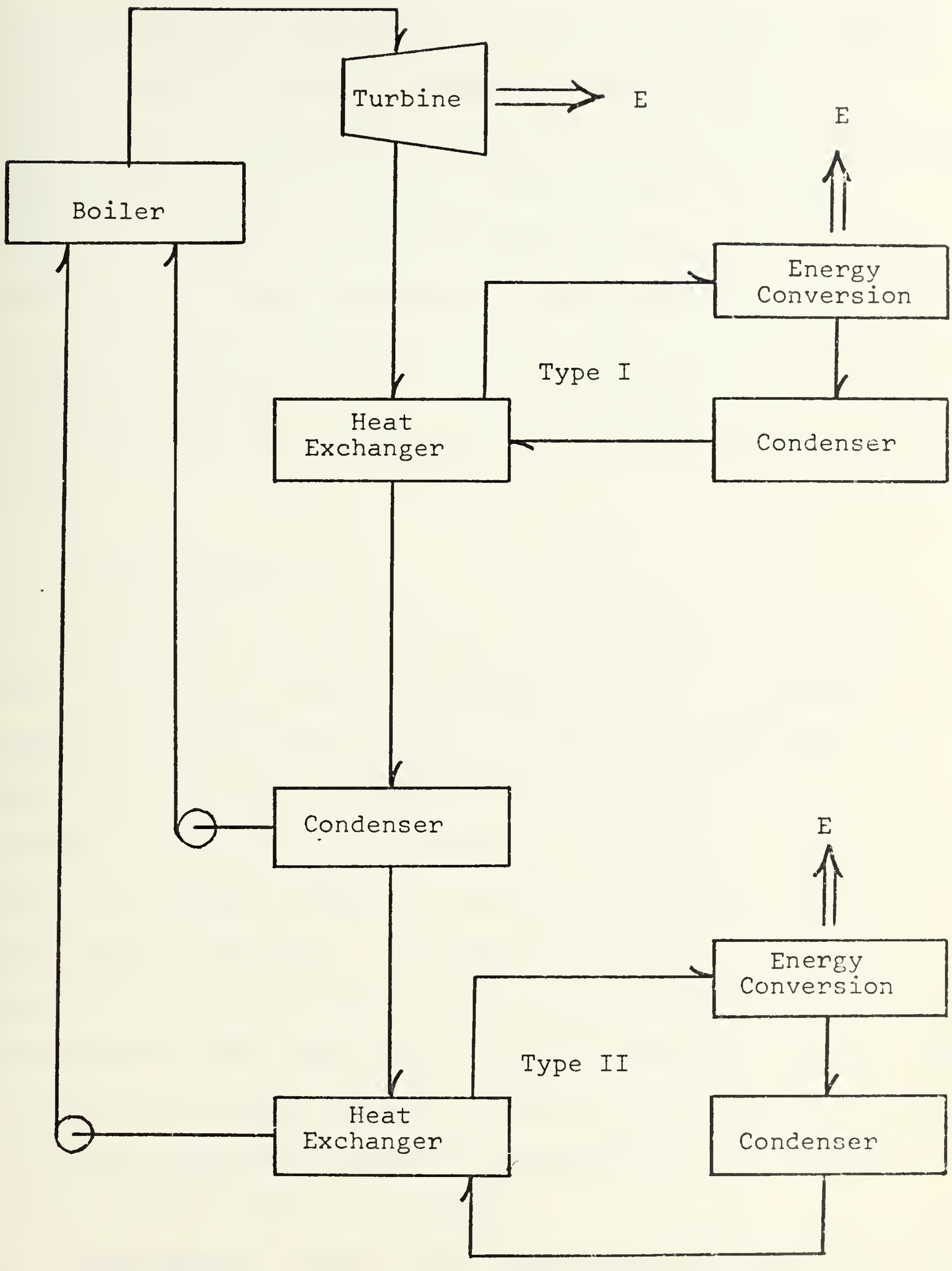

FIGURE 3. <u>Waste Heat Cycles</u>

system economics. Although some researchers predict plant costs which are currently competitive with conventional methods, [9] uncertainties arising from the lack of operating experience weaken the claims of these proponents. Until economic viability can be conclusively shown, risk aversity will act as a strong deterrent to attracting the very large amounts of venture capital required. With the private sector presently unconvinced, the federal government is undertaking the expenditures required for research and development efforts [10].

B. THERMOECONOMICS

Typically, system design and cost studies are conducted in a two-step or, at best, iterative process. Designers assemble specifications based on technically achievable and desirable functional characteristics. They are, of course, guided in their design decisions by some measure of intuition as to the economic impacts, usually based on prior experience with similar programs. The degree of detail in the initial specifications presented, in fact, often reflects the confidence held by the engineers in their economic appraisals. The system and component specifications are then subjected to cost analysis, prime cost factors are identified, and technical-economic trade-offs are suggested.

The problems arising from this partial separation of the design and costing steps are more or less severe according to the application. There is a fundamental difficulty in communicating the two groups' understandings in a meaningful

way, a difficulty which increases as the novelty of the design situation and hence number of unconstrained design choices increases. In their quest for a sophisticated design product, engineers may design around some apparently desirable parameter, such as a high heat transfer coefficient. Cost analysts may take this figure as fixed and address themselves to questions of material selection, maintainability, or manufacturing tolerances without recognizing that adjusting the heat transfer coefficient itself could produce the most rewarding cost effects.

This type of difficulty is most severe when little experience is available to guide the engineer's fundamental choices, as is true in the case of environmental vapor power systems. The small available thermal potential drastically distinguishes these systems from their high temperature counterparts. In a fuel-fired generating plant, for example, the power required to pump the working fluid can be neglected in a first approximation, and a variation of 1°F temperature difference across a boiler tube is insignificant. As is demonstrated later in this study, such considerations can have a profound influence on the overall economics of an environmental plant.

To some extent, the engineer's problem can be viewed as being where to start. Recognizing that pumping power is going to be substantial, he might decide to assign 10 or 20 percent of the plant's overall output to pumping requirements and build much of the rest of the design about this choice. Or

he might choose to utilize 30 percent of the total temperature difference for heat transfer, leaving the remaining 70 percent available for enthalpy drop across the turbine. He might establish a dimensional constraint, based on nothing much more than the feeling that a 100 foot diameter pipe is a very big pipe.

Unfortunatly, all these basic choices involve performance and cost tradeoffs. If flow rates are increased to enhance heat transfer, drag coefficients increase as well. How much improvement in heat transfer is worth how large an increase in pump head, and hence pump work? Pump work is also influenced by heat exchanger tube diameter, spacing, and surface characteristics, which also affect space and material requirements. How much should one be willing to pay to reduce fouling heat resistance? If heat exchange is dominated by fouling resistance, is it worth the extra temperature drop necessary to shift to a different boiling regime? Unless the cost analyst is knowledgeable about the thermodynamic consequences of costing factors he is in as poor a position as the engineer to make the tradeoffs in dollars per millimeter of fouling organisms.

C. OBJECTIVE

What is needed is an analytical method whereby overall economic effects may be integrated into engineering design in such a way that the designer's intuition may be enhanced in trading off the costs and benefits of parameter selection at the margin. A means is required for mapping the large number

of interrelated engineering variables into their individual and collective effects in the marketplace, where the ultimate design appraisal will take place.

The research reported on in this paper is intended to develop and evaluate one method of integrating marginal cost/benefit analysis into engineering design and to show the kinds of information which could thus be gained. In this initial investigation, no effort has been made to apply the method to any particular practical design problem or to produce analytical insights into existing systems. The intent has been to show how thermoeconomic analysis can be performed and what value it can have when applied to a specific real case.

II. THERMOECONOMIC ANALYSIS

A. CONCEPT

Profit is the difference between benefits and costs, both broadly considered. When these can be related over a common set of decision variables, $\tilde{X}$, one may write

$$\pi(\tilde{X}) = B(\tilde{X}) - C(\tilde{X})$$

with $B(\tilde{X})$ representing the sum of all benefits, and $C(\tilde{X})$ the sum of all costs:

$$B(\tilde{X}) = \sum_{i}^{N} B_i(\tilde{X})$$

$$C(\tilde{X}) = \sum_{i}^{N} C_i(\tilde{X})$$

If all the relevant B_i and C_i can be defined functionally over $\tilde{X}$, performing

maximize: π
subject to: a required level of performance (A)

would produce the desired optimization.

B. PROBLEM REDUCTION

Attempting a global optimization directly with all possible costs and benefits considered, although theoretically possible, encounters many practical difficulties [11]. It is possible, however, to achieve considerable reduction of the problem without sacrificing many of the benefits of the analysis.

First, although the impacts of general (and difficult to quantify) externalities, such as independence from foreign control of energy sources, are important and should not be excluded from the final analysis, many interior decisions suffer not at all from excluding externalities such as these from most of the study. Many other factors are not related to the decision variables (X_i) and therefore do not affect marginal design choices. For example, personnel training expenses are not close functions of tube diameter. Since the solution to

$$\begin{aligned} &\text{maximize } \pi = B(\tilde{X}) - C(\tilde{X}) - D \\ &\text{subject to } \quad g(\tilde{X}) = 0 \end{aligned}$$

where D is constant with respect to $\tilde{X}$ is identical to the solution of

$$\begin{aligned} &\text{maximize } \pi = B(\tilde{X}) - C(\tilde{X}) \\ &\text{subject to } \quad g(\tilde{X}) = 0 \end{aligned}$$

any factor which acts only as an additive constant may be excluded from the analysis without affecting the results.

Even with invariants over the decision variables excluded, there are other serious impediments to seeking global solutions to (A). Convexity of the optimization problem is not assured by the physical relations modeled.[1] When all

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[1] A convex optimization problem is defined as one with a convex objective function, to be minimized, concave $\geq$ inequality constraints, and linear equality constraints [12]. The conditions on the constraints assure that the feasible region is a convex set, i.e., for every λ, $0 \leq \lambda \leq 1$, and any two points X_1, $X_2 \in T$, a convex set, $[\lambda X_1 + (1-\lambda)X_2] \in T$.

decision variables are considered at once in a global assault it is increasingly difficult to test for uniqueness of the solution. Secondly, since design variables in one system component often are only distantly related to those in another, insights are obscured when they are varied simultaneously within one code. Thirdly, the model can never be exact. It is important for the designer to keep track of the effects of his modeling choices in detail. This is more easily achieved by putting the pieces together sequentially than all at once. Finally, the designer often has adequate information available to intelligently fix some of the variables. It is unnecessary to complicate the analysis by including as free variables factors which are closely constrained by other considerations.

For these reasons, it appeared desirable to follow the usual procedure for the optimization of large scale systems by decomposing the problem into coherent interrelated zones and achieving global optimality through one of the available zone coordination methods. The next section contains an outline of the general procedure.

C. PROBLEM COORDINATION

The general theory for optimizing large scale systems through coordination of smaller subsystems can be found in references such as Wismer [11]. The following discussion of the two basic approaches is greatly particularized in that the terminology and composition of the examples reflect the structure of the sample analysis which is presented in section III.

The first approach, called the model coordination method, can be understood through consideration of the decomposed system shown in Figure 4.

Define $\tilde{y} = (T_H, T_C)$ as a vector of coordinating variables and $\tilde{X}_1$ and $\tilde{X}_2$ as vectors of design variables in zones 1 and 2. Then construct the zone subproblems:

$$\begin{aligned} &\underset{\tilde{X}_i, \tilde{y}}{\text{minimize:}} \quad f_i(\tilde{X}_i, \tilde{y}) \qquad\qquad i = 1,2 \\ &\text{subject to: } g_i(\tilde{X}_i, \tilde{y}) \geq 0 \\ &\qquad\qquad\quad h_i(\tilde{X}_i, y) = 0. \end{aligned}$$

The first level of analysis is conducted by setting $\tilde{y} = \tilde{y}^o$, a feasible value of $\tilde{y}$. Then solve

$$\begin{aligned} &\underset{\tilde{X}_i}{\text{minimize:}} \quad f_i(\tilde{X}_i, \tilde{y}^o) \qquad\qquad i = 1,2 \\ &\text{subject to: } g_i(\tilde{X}_i, \tilde{y}^o) \geq 0 \\ &\qquad\qquad\quad h_i(\tilde{X}_i, \tilde{y}^o) = 0. \end{aligned}$$

The solutions are designated $\tilde{X}_1^1$ and $\tilde{X}_2^1$. The second level of analysis seeks to find the value of $\tilde{y}$ which produces the minimum value of

$$\begin{aligned} &F(\tilde{X}_1^1, \tilde{X}_2^1, \tilde{y}) = f_1(\tilde{X}_1^1, \tilde{y}) + f_2(\tilde{X}_2^1, \tilde{y}) \\ &\text{subject to:} \qquad g_i(\tilde{X}_i^1, \tilde{y}) \geq 0 \qquad\qquad i = 1,2 \\ &\qquad\qquad\qquad\quad h_i(\tilde{X}_i^1, \tilde{y}) = 0. \end{aligned}$$

Designate the solution by $\tilde{y}^1$. An iterative sequence is now established by replacing $\tilde{y}^o$ by $\tilde{y}^1$ in the first level problems and resolving; using the resulting $\tilde{X}_i^2$ to find $\tilde{y}^2$

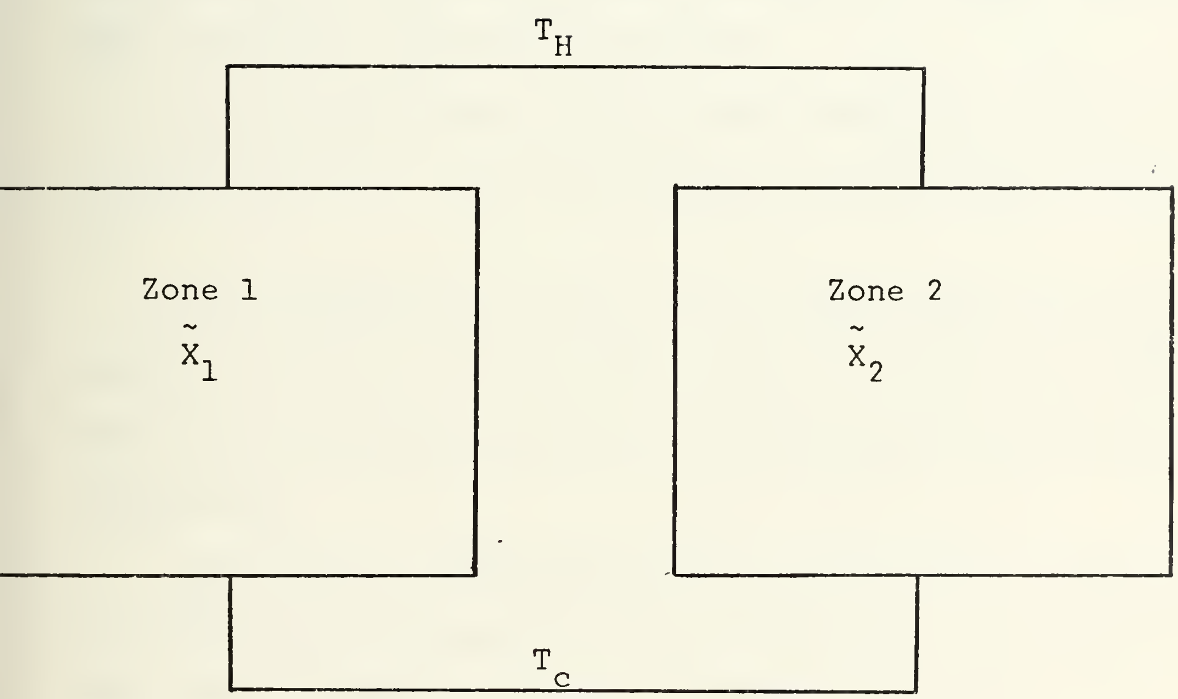

FIGURE 4. Model Coordination

in the second level problem, and so forth until the improvement achieved with each iteration is less than a specified tolerance.

This approach is called the model coordination method because the task of the second level control is to choose the linking variables in such a way that the independent first level systems are forced to choose solutions which in fact correspond to an overall system optimum. In some references, this is called the feasible method.

The second method, called goal coordination or the dual feasible method, views the decomposed system as in Figure 5.

It is important to note that in this formulation of the problem, $\tilde{y}$ does not necessarily equal $\tilde{z}$. The interactions have been literally removed by "cutting" all links between subsystems.

The physical requirement that, in the end, $\tilde{y}$ must equal $\tilde{z}$, termed the interaction-balance principle, is satisfied in the course of the analysis as follows.

In the first level analysis, let $\tilde{\lambda} = \tilde{\lambda}^o$. Then solve

$$\begin{aligned} \text{minimize:} \quad & L_1(\tilde{X}_1, T_H, \mathcal{T}_c, \tilde{\lambda}^o) = f_1(\tilde{X}_1) + \lambda_1^o T_H - \lambda_2^o \mathcal{T}_c \\ \text{subject to:} \quad & g_1(\tilde{X}_1, T_H, \mathcal{T}_c) \geq 0 \\ & h_i(X_1, T_H, \mathcal{T}_c) = 0 \end{aligned}$$

and

$$\begin{aligned} \text{minimize:} \quad & L_2(\tilde{X}_2, \mathcal{T}_H, T_c, \tilde{\lambda}^o) = f_2(\tilde{X}_2) - \lambda_1^o \mathcal{T}_H + \lambda_2^o T_c . \\ \text{subject to:} \quad & g_2(\tilde{X}_2, \mathcal{T}_H, T_c) \geq 0 \\ & h_2(X_2, \mathcal{T}_H, T_c) = 0 \end{aligned}$$

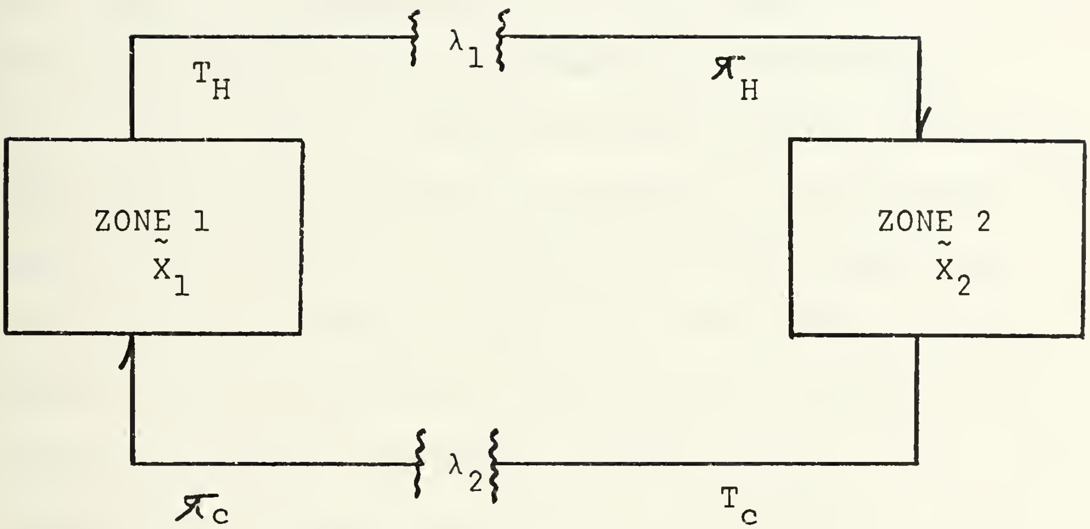

FIGURE 5. Goal Coordination

Define:

$$\tilde{y} = \{T_H, T_C\}$$

$$\tilde{z} = \{\pi_H, \pi_C\}$$

$$\tilde{\lambda} = \{\lambda_1, \lambda_2\}$$

yielding: $\tilde{X}_1^1$, $\tilde{X}_2^1$, and $\tilde{y}^1$.

The second level problem then becomes choosing $\tilde{\lambda}$ such that solutions to the first level problems result in satisfaction of the interaction balance principle. This is a well behaved optimization in its own right and is solved with the usual techniques of mathematical programming.

Notice that in this method the coordination effect of the second level analysis is effected by manipulating the goals of the first level analysis through adjustment of the $\tilde{\lambda}$ coordinating variables, hence the term goal coordination method. The $\tilde{\lambda}$ multipliers enter the individual first-level problem objective functions linearly and act like prices, adding to or subtracting from the performance function of each subproblem in direct proportion (with proper sign) to the amount of z_i demanded and the amount of y_i produced. Thus the second-level goal coordination can be interpreted as modifying "prices" of the interacting variables in order to force the independent first-level problems to select consistent values of the linking variables and hence the correct overall system optimum.

Much additional information is available in the results of the steps of the solution when cast in this format, and the interested reader is referred to the considerable literature on the subject, [13, 14, 15 and 16, for instance].

Because of its more straight forward formulation, the sample analysis in the following section is cast in the model coordination format.

III. SAMPLE ANALYSIS

A. PRELIMINARIES

The methodology of thermoeconomic analysis can best be described through demonstration with a sample analysis. For this purpose, an extremely simplified thermal system was selected; one which contains the essential features of a realistic system but avoids a number of complications which would tend to obscure the technique. It should be well understood that with lumped component representations and several significant losses neglected, the model chosen can not be treated as representing a practical plant, nor can the results of the analysis be taken as having implications for a real system. The model does have many similarities with ocean thermal energy conversion plants as presently conceived, and in Appendix B a discussion is presented as to what refinements would be necessary to extend the sample model into one of a functional ocean thermal system. For ease of exposition, the model will be discussed without repeated references to these departures from realism.

B. BASIC SYSTEM DESCRIPTION

Figure 6 diagrams the basic system considered. The thermal source consists of an infinite supply of seawater at a temperature of T_{HE} = 85°F. The thermal sink is a similarly limitless supply of seawater at T_{CE} = 45°F. In the energy extraction component the ammonia working fluid is heated as

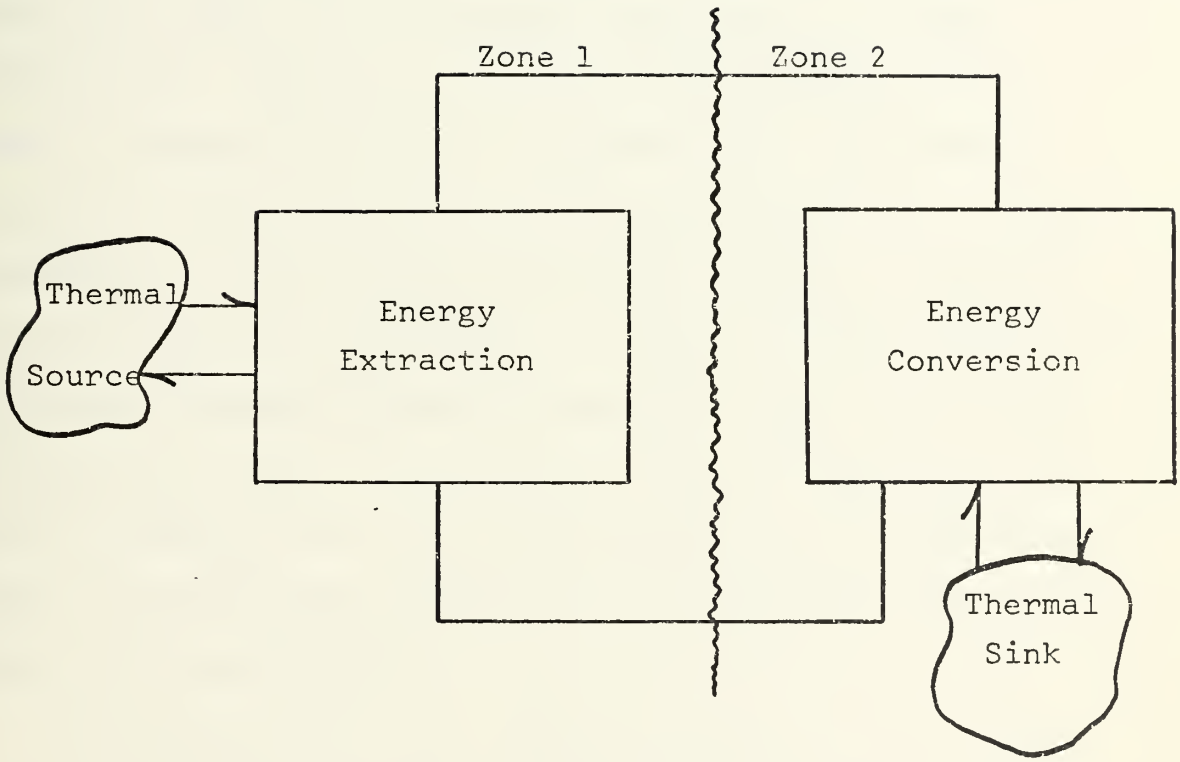

FIGURE 6. Basic System

it flows through the shell side of a rectangular crossflow shell and tube heat exchanger with smooth staggered tubes. The heat is provided by relatively hot seawater flowing through the tubes as shown in Figure 7.

Single phase heat exchange takes place in the device, with both fluids remaining compressed liquids.

The energy conversion component receives hot ammonia liquid from the heater, accomplishes energy conversion to electrical form, and discharges the liquid at a lower temperature. The manner in which the conversion takes place is unspecified and not necessary for this sample analysis. The conversion process is described by a single parameter, ψ, which measures how much energy is converted to electricity per pound mass of ammonia flowing through the device per degree Fahrenheit temperature drop. The value selected for ψ was 0.5 Btu/lbm °F, which is approximately half the specific heat for liquid ammonia and, incidentally about the same energy available to a turbine with saturated vapor inlet conditions.[2]

The fluid pressurizer consists simply of one or more standard centrifugal pumps, sufficient to drive the working fluid through the system at the required rate. Both the hot and cold seawater are similarily pumped.

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[2] Saturated ammonia vapor at 80°F has enthalpy of 630 Btu/lbm. Isentropic expansion to 50°F results in enthalpy of 615 Btu/lbm, or 0.5 Btu/lbm per °F [17].

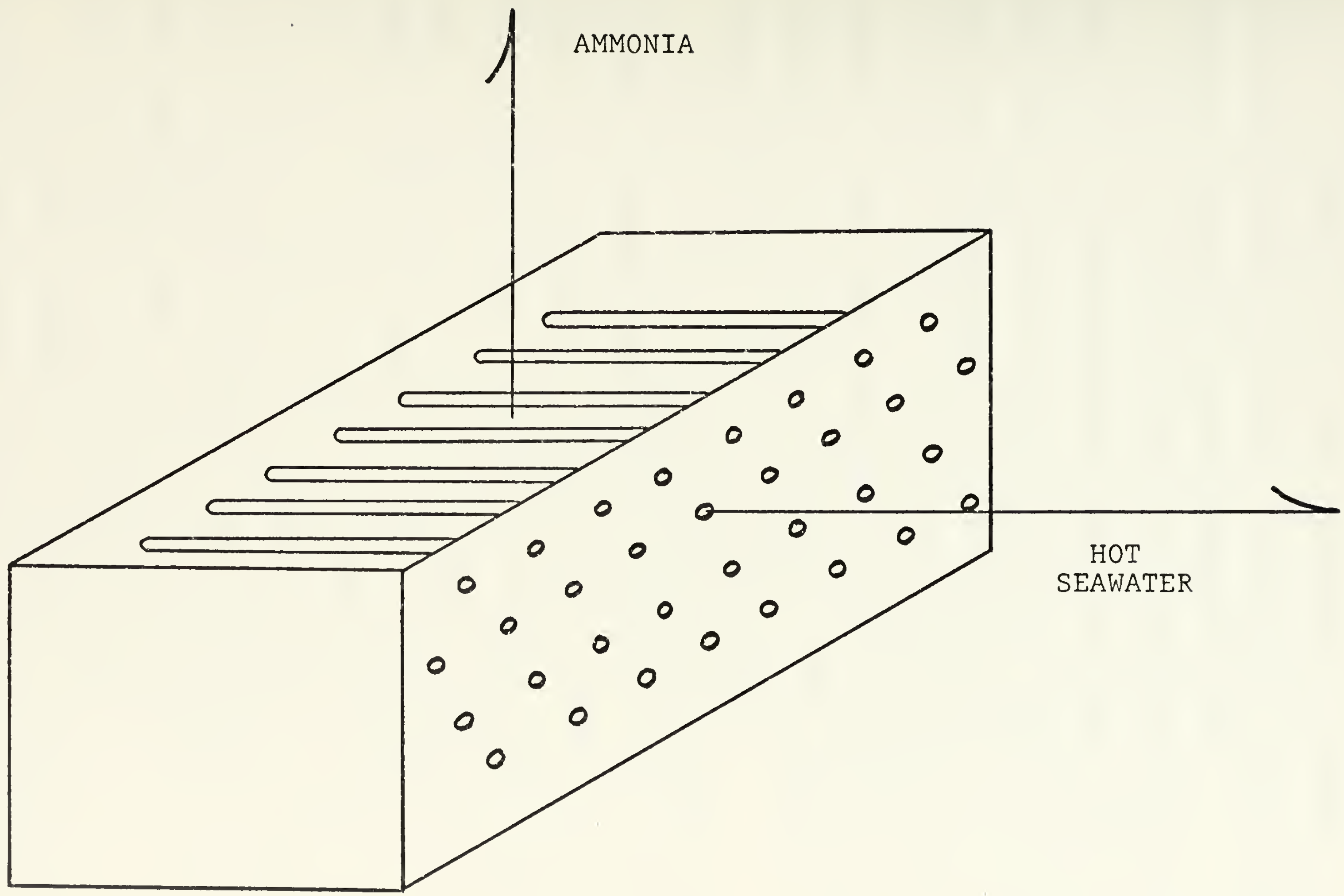

FIGURE 7. Heat Exchanger

The system was zoned as depicted in Figure 6, with zone 1 consisting of the heat exchanger and working fluid and hot seawater pumps, and zone 2 consisting of the energy conversion device and the cold seawater pumps.

Before proceeding further, it should be made clear that none of the above assumptions nor those which follow constitute final arbitrary design selections. Each parameter, fluid, and configuration is eventually fixed as an output of the analysis itself. Their initial specification should be regarded as tentative, pending further information to be developed in the course of the study. This preliminary configuration acts only as a starting point.

The next step is to characterize zonal inputs and outputs in terms of appropriate physical variables which are descriptive of the transactions taking place at zone boundaries. The principal feature of the hot seawater is its temperature, T_{HE}, so this was chosen as the input to zone 1 from the thermal source. The other input to zone 1 is the ammonia discharge from zone 2, which is again described by its temperature, T_C. The output of zone 1 is hot ammonia liquid at temperature T_H. The only remaining variables which cross zone boundaries are the electrical output of zone 2, G, and the cold seawater from the thermal sink at temperature T_{CE}.

The global problem is to maximize the profit obtainable by selling the system's electrical output at market prices. Translated into zone terms, this implies that each zone should produce the required level of output at minimum cost, given the inputs it has to work with.

C. DETAILED DESCRIPTION OF ZONE 1

The heat exchanger has length in the direction of seawater flow (ℓ), height in the direction of working fluid flow (a), and width transverse to each (w). Tubes have inside diameter (d), wall thickness t, and have transverse and longitudinal spacing S_T and S_L. The inside and outside heat flow resistances due to chemical and biological fouling are combined into one fouling resistance, R_F.

Selection of ammonia as the working fluid and seawater as the heat source leads to the following table of approximate physical properties, all considered constant.

Table III-1.
FLUID PROPERTIES

FLUID	DENSITY (lbm/ft3)	VISCOCITY (lbm/ft-hr)	CONDUCTIVITY (Btu/hr-ft-°F)	SPECIFIC HEAT (Btu/lbm°F)
Ammonia	$\rho = 40$	$\mu = 0.5616$	$K = 0.307$	$C_p = 1.135$
Seawater	$\rho_H = 64$	$\mu_H = 2.37$	$K_H = 0.349$	$C_p = 1.0$

The working fluid mass flowrate ($\dot{m}$) and hot seawater mass flowrate ($\dot{m}_H$) are provided by centrifugal pumps, which deliver the required flows against the head created by frictional and form losses in the heat exchanger, (minor losses were neglected but could easily be included). The pumping power for these pumps, W_p and W_H, is a parasitic deduction from the gross plant electrical output, G_i.

Fixing the input, output, and linking variables (T_{HE}, T_{CE}, T_H, T_c, and G) temporarily helps to focus an understanding of zone 1 objectives and constraints. T_{HE} and T_{CE} were set previously at 85°F and 45°F, and these define the thermal potential available to the system. If half of this potential is assigned to drive heat through the exchanger surfaces, T_H and T_c selections of 75°F and 55°F result. If overall power output is set at 25 mw (a frequently encountered figure for prototype ocean thermal plants), the required working fluid flow rate can now be determined from the relationship

$$G = \dot{m}\ \psi(T_H - T_c). \tag{1}$$

It now becomes evident that the task of zone 1 is to receive T_{HE} and T_c and produce the required $\dot{m}$ at T_H with minimum cost. The next major task is to select the design variables to be used. To do this, we first look at the governing physical and cost relationships.

As discussed in [18] the performance of a heat exchanger can be described in terms of its effectiveness:

$$\epsilon = \frac{T_H - T_c}{T_{HE} - T_c} = 1 - e^{-\Gamma\Theta}$$

where

$$\Gamma = 1 - e^{-N\Theta}$$

and

$$N = \frac{A_H U_H}{\dot{m}C_p}, \quad \Theta = \frac{\dot{m}_H C_{pH}}{\dot{m}C_p}.$$

Alternatively, the amount of heat transferred can be found from

$$Q = U_H A_H \Delta T_{LM}.$$

In either case, the fundamental process description is in terms of heat exchange surface area, heat exchange coefficients, flowrates, temperatures, and fluid properties.

The major costs in zone 1 are the capital costs of the heat exchanger, z_1, the pumps, z_2, and the cost of the pumping power. This latter can be considered as an opportunity cost and valued at the amount which could have been realized had the parasitic pumping power, W_p and W_H, been sold at the prevailing rate, p_o, instead of being used internally. Correlations are available [20] which give capital costs of heat exchangers as functions of heat exchanger area and capital costs of pumps in terms of the product of flowrate and head. The frictional head is usually determined empirically and related to flow velocities, exchanger configuration, and fluid properties in terms of Reynolds numbers.[3]

It might initially appear attractive to choose the design variables to be $\dot{m}_H$, U_H, A_H, and the friction heads

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[3] The Reynolds number is a dimensionless grouping of physical variables which indicates the ratio of inertia forces to viscous forces. It is formed from the product of a characteristic velocity times a characteristic dimension, divided by the fluid kinematic viscocity [21].

The Prandtl number is also dimensionless, being a measure of the ratio of the diffusivity of momentum to the diffusivity of heat. It is formed by multiplying the fluid's specific heat times its viscocity and dividing by its conductivity [21].

DP and DP_H, but it is at this point that a key feature of thermoeconomic analysis comes into force. Recall that what is desired is a way to discover the tradeoffs between cost and performance. Whatever variables are chosen, it must be possible to establish this balance through the functional relations over the domain of the variable set. As an example of what happens otherwise, consider working with the variables suggested just above. Performance can always be improved by increasing U_H, and it doesn't cost anything to do so since U_H is absent from the cost correlation. Costs can always be decreased by dropping the DP's, and performance would seem to be unaffected because the heat exchange equations do not contain pressure drop terms. Any sensible computer code would therefore drive U_H and DP as high and low, respectively, as is allowed. Setting a constraint on these parameters is, in effect, arbitrarily choosing them, and no information has been gained in the process. Achieving high heat transfer with low pressure drops is known to be desirable <u>a priori</u>.

Besides not permitting a cost-performance balance to be weighed, there is a second problem with the variable list suggested, namely that U_H and the DP's are inextricably linked through the Reynolds numbers. With a given working fluid, heat transfer can only be improved by raising the Reynolds numbers with the concurrent result that the pressure drops are increased simultaneously. This is the core concern of the zone 1 analysis: U_H is made up of the inside coefficient,

h_i, the outside coefficient, h_o, the tube wall conductivity and the fouling resistance. There are two pressure drops of concern, one in the seawater and one in the working fluid. What combination of these parameters will produce the heat exchange required at minimum cost?

To identify an appropriate variable set over which to find this cost-performance balance, one must look to the next level. Besides physical fluid properties, Reynolds numbers depend upon flow rates and spatial dimensions. Surface area depends on spatial dimensions. Pump work depends on flow rate and pressure drop, which vary as functions of Reynolds number.

Clearly, then, all the cost and performance calculations can be built up in terms of flow rates and spatial dimensions, and this is the highest level of variable with which the desired tradeoff can be made with the functional relationships available. Note, however, that if other valid relationships could be found which gave the information required in terms of other quantities, other variables might be able to be used.

With the tools at hand, though, it was decided to perform the analysis in terms of dimensions and flowrates. But variable selection is not yet complete; one has the option of which parameters will be allowed to vary independently in the code and which will be controlled externally. This question is resolved as a matter of judgment, and depends upon the confidence the designer has in his preliminary intuition and what specific information he seeks from the analysis. Hope-

fully, this matter will be clarified as the analysis proceeds. For the present investigation, it was decided to permit four variables to "float": seawater flow rate, and the length, height and width of the heat exchanger. The following parameter selections were made. Included are those which have been discussed previously.

Table III-2.
SELECTED VALUES OF PARAMETERS

working fluid:	ammonia
tube diameter:	1/2 inch
tube wall thickness:	0.035 inch
tube spacing, S_T:	1.5 d inches
tube spacing, S_L:	1.5 d inches
hot seawater temperature:	85°F
hot working fluid temperature:	75°F
cold working fluid temperature:	55°F
Gross plant power output:	25 MW
energy conversion factor, ψ:	0.5 Btu/lbm-°F
market price of energy:	0.03/Kw-hr
pump efficiencies:	0.9
fouling heat resistance:	0.005 $ft^2{}^\circ F^{-hr}$/Btu
tube wall conductivity:	30 Btu/ft-hr-°F

The functional relationships over both costs and performance are developed in detail in Appendix A and summarized below.

Pump Work, Working Fluid

$$WP = \frac{\dot{m}DP}{\rho\eta} \qquad \left(\frac{f_t lb_f}{hr}\right)$$

where

$$DP = \frac{2f}{\rho}\left[\frac{\dot{m}S_T}{\ell(S_T - d)w}\right]^2 \frac{a}{S_L} \qquad \left(\frac{lb_f}{ft^2}\right)$$

and

$$f = 0.75\ Rec^{-0.2};\ Rec = \frac{\dot{m}S_T}{\mu \ell W}$$

Pump Work, Hot Seawater

$$W_H = \frac{\dot{m}_H DP_H}{\rho_H N_H} \qquad \left(\frac{f_T lb_f}{hr}\right)$$

where

$$DP_H = f_H\left(\frac{8}{\pi^2}\right)\left(\frac{\ell}{d}\right)\left(\frac{\dot{m}_H^{\,2}}{\rho_H}\right)\left(\frac{S_T}{d}\right)^4\left(\frac{1}{w_a}\right)^2 \qquad \left(\frac{lb_f}{f_T^{\,2}}\right)$$

and

$$f_H = 0.316\ R_{eH}^{\ -1/4};\quad R_{eH} = \frac{4\dot{m}_H\ S_T^{\,2}}{\pi\mu_H d w_a}$$

Capital Cost, Heat Exchanger

$$z_1 = 103\ 2\ A_H^{\,0.627} \qquad (\$)$$

Capital Cost, Pumps

1. Working Fluid

$$z_{21} = 488(c/H)^{0.602} \qquad (\$)$$

2. Hot seawater

$$z_{22} = 814\ (c/H)_H^{\,0.602} \qquad (\$)$$

Heat Exchange Coefficients

1. Inside coefficient

$$\frac{h_i d}{k_H} = 0.036\ R_{eH}^{\ 0.8}\ P_{rH}^{\ 1/3}\ \left(\frac{d}{\ell}\right)^{0.055}$$

2. Outside coefficient

$$\frac{h_o d}{k} = 0.511\ R_{ec}^{\ 0.562} P_r^{\ 1/3}$$

3. Tube wall resistance

$$R_T = \frac{A_i t}{A_{mw} k_w} \qquad \left(\frac{f_t^{\ 2}\text{-hr-}°F}{Btu}\right)$$

4. Overall heat transfer coefficient

$$\frac{1}{U_H} = \frac{1}{h_i} + \frac{A_i}{A_o h_o} + R_T + R_F \qquad \left(\frac{f_t^{\ 2}\text{-hr-}°F}{Btu}\right)$$

So far, the sample model has been zoned, input and output variables chosen, design variables selected, and the relevant functional relationships formulated. The next step is to define explicitly the objective and constraints applicable to zone 1 and proceed with computation. The zone objective which supports the global objective (maximum profit) most directly is minimum zone cost. With $\dot{m}$ fixed by (1), the fundamental zone constraint is to raise the required ammonia temperature from T_c to T_H, given the heat source at T_{HE}. By writing this constraint in terms of heat exchanger effectiveness, the zone 1 optimization problem may be cast as:

$$\underset{\tilde{X}}{\text{minimize:}} \quad z = z_1 + z_2 + p_o(W_p + W_H) \qquad \text{(A)}$$

$$\text{subject to:} \quad \epsilon = \frac{T_H - T_c}{T_{HE} - T_c}$$

where

$$\tilde{X} = (\dot{m}_H,\ \ell,\ w,\ a).$$

The reader may wish to test his own intuition at this point. Should the heat exchanger be particularly long, high, or wide, or should the dimensions be approximately equal? Would seawater flow be expected to be about the same as ammonia flow, or more, or less? A very compact heat exchanger would reduce the capital costs for that component, give high fluid velocities and hence good heat exchange, but would give higher pressure drops and hence more pumping costs than a larger exchanger.

D. SOLUTION ALGORITHM

Numerous computational methods exist for solving (A). Since most of the equations are in the Cobb-Douglas form $f(\tilde{X}) = C\, X_1^{\alpha_1}\, X_2^{\alpha_2}\, X_3^{\alpha_3} \ldots$ it is possible to use geometric programming [22] or a number of search techniques. The algorithm utilized in this analysis was SUMT4, a complete description of which is contained in [23]. A full treatment of the underlying theory is given in [12] and only a brief explanation of the essential computational sequence is given here.

SUMT4 was chosen partly because of the generality of the optimization problem which it can solve. The solution techniques do not depend on any special features of the problem structure, and the conditions for the existence of a solution are not overly restrictive. The only absolute requirements are that the feasible space be non-empty and that local minima occur at points short of infinity. In order to assure uniqueness of the solution, the objective function and all inequality

constraints must be continuous and convex/concave respectively, but many problems have been solved when these conditions were not met [17].

This generality is achieved by transforming the constrained minimization problem into a sequence of unconstrained problems, the solutions of which converge to the solution of the original problem. The basic idea is that the objective and constraints are formed into an auxiliary function (penalty function, generalized Lagrangian) as follows:

given: minimize: $f(x)$

subject to: $g_j(\tilde{X}) \geq 0$, $j = 1,2,\ldots, m$

$h_j(\tilde{X}) = 0$, $j = m+1, \ldots, m+p$

where $\tilde{X}$ is an n-dimensional column vector, form:

$$P(\tilde{X},r) = f(\tilde{X}) - r \sum_{j=1}^{m} \ln g_j(\tilde{X}) + \sum_{j=m+1}^{m+p} [h_j(\tilde{X})]^2/r.$$

Look first at the term involving the inequality constraints, $g_j(X) \geq 0$. Because of the shape of the logarithm function, a very large amount is added to the P function as the boundary is approached, while the term is relatively flat in the interior of the solution space. Conversely, departure in any direction from the equality constraints, h_j, invokes increasing penalties. The size of the penalty is controlled by the parameter r. Given a starting point, X_o, and an initial r, the algorithm searches for a minimum of the P function. The search technique is specified by the user, and can be either of two modifications of the generalized Newton-Raphson method, the method of

steepest decent, or McCormick's modification of the Fletcher-Powell method. When a minimum has been found to a tolerance specified by the user, the parameter r is reduced by a specified ratio and a new minimum is located. This process is repeated until no significant improvement in the P value is obtained, as indicated by one of a number of available tests. As each successive point is generated, first and second order extrapolation are used for convergence acceleration.

Other attractive features of SUMT4 besides its generality, have to do with the flexibility it makes available to the user. The ability to specify 12 options and 4 tolerances afford great freedom in adapting the program to the specific application. In addition, the program will produce for itself much of the information normally required to be supplied. If a feasible starting point is not known, one will be calculated. An initial r value may be prescribed, or the program will find a good one on its own. If the derivatives of the f, g, and h functions are not continuous or not explicitly available SUMT4 will compute them with central differencing procedures. If the problem does have a special structure, this can be exploited. A very helpful feature is that the mixed interior-exterior penalty function makes it unnecessary for the starting point to strictly satisfy the equality constraints or be interior to the solution space. This makes it possible to avoid much advance manual computation.

Finally, SUMT4 is written in a modular structure to facilitate changes in logic, options, problems, and input-output. This fact was appreciated in the early stages of the present investigation, as a few minor modifications were necessary to help avoid programming problems. As an example, note that the form of the modeling equations requires that the X values be raised to fractional powers. Since Fortran accomplishes this through logarithms, negative values of X can not be handled. Although SUMT4 will include non-negativity constraints automatically, if desired, the differencing and extrapolation subroutines still selected negative values during the computations. Small changes in the subroutines which handle these phases avoided this problem, making the errors leading to negative X values easier to locate.

E. INITIAL COMPUTATIONAL CONSIDERATIONS

The penalty function formed from problem A is:

$$P(\tilde{X},r) = z + r l_n \left[\epsilon - \left(\frac{T_H - T_c}{T_{HE} - T_c} \right) \right].$$

Observe that z represents the costs over the lifetime of the plant and is therefore a very large number, while ϵ and $\frac{T_H - T_c}{T_{HE} - T_c}$ are of the order of one half to one. Without scaling, the algorithm can accomplish major reductions in the P function by merely calculating moves which reduce the z values without regard to how well the equality constraint is satisfied. This distortion is easily avoided by scaling the cost equation to bring it to approximately the same size as the constraint

equation. One preliminary computational run is all that is necessary to identify an appropriate scaling factor.

Secondly, note that the constraint function consists of a nested pair of exponentials which is asymptotic to the value 1.0. If the initial $\tilde{X}$ guess produces an ϵ value which is well out on the flat portion of the graph, the algorithm is unable to determine which direction will produce improvement, since the gradient vector is essentially zero. Because of this particular shape of the constraint function, it is necessary to provide a starting point that is "sufficiently feasible" to provide for some gradient in the constraint function despite the fact that, in general, SUMT4 does not require a feasible initial point. This requirement is not particularly burdensome, however, because a simple hand calculation can quickly locate an $\tilde{X}_o$ that corresponds with a mid-range ϵ.

F. FIRST ZONE 1 SOLUTION

With the preliminary parameter selections listed in Table 2 and a starting point of $\tilde{X}$ = (10, 10, 10, 10), the computational sequence was initiated, and the first interesting piece of information was developed. With the specified weighting of capital costs and costs over lifetime pumping charges, the calculation determined that overall costs could be continuously reduced by extending the heat exchanger width indefinitely. Since this dimension is normal to both fluid flow paths, both flow velocities are reduced as w is increased, resulting in progressively lower pressure drop losses and hence pumping power requirements. The corresponding increase

in exchanger capital cost was not sufficient to offset the savings in pump capital cost and power charges.

This is the first category of information which thermoeconomics provides to the designer: which parameters should be set as large or as small as other practical considerations permit. The preferred method of handling instances of this sort is to explicitly determine the costs associated with allowing the parameter in question to assume increasing values. In this case, it should be possible to estimate the increase in hull or platform costs required to accommodate increasingly wide heat exchangers. If no functional relationship is available, however, the parameter can simply be constrained to some limit which seems reasonable, and the sensitivity of the resulting design to this constraint determined. In order to proceed, it was necessary to include a size limit in the list of preliminary specifications, so a dimensional constraint of 20 feet was chosen, based loosely on the arrangement considerations in [16].

With this addition, the following solution was obtained:

$$\dot{m}_H = 21.1 \cdot 10^6 \text{ lbm/hr}$$
$$\ell = 6.1 \text{ ft}$$
$$w = 20.0 \text{ ft}$$
$$a = 6.1 \text{ ft.}$$

This is an example of the second category of information obtainable from the method: a point design based on the supporting assumptions and specifications. It is useful to look a bit closer at the results and implications of this first

run. With the use of a separate small Fortran program, the relations in Table III-3 were calculated, using the above solution.

It is important at this stage to insure that none of these variables is out of range with respect to the modeling equations. In particular, the Reynolds numbers must be within the scope of the friction loss and heat transfer correlations and the elements of the cost equations must lie within the span of the cost correlations. If not, the correlations must be altered to restore applicability. Comparison with the ranges of validity specified in Appendix A shows that all such constraints were satisfied in this case.

Looking first at the physical dimensions which have been computed, it is seen that an exchanger has been specified which is approximately three tenths as long in the directions of flow as in the breadth direction. The immediate implication is that the cost tradeoffs specified have dictated that flow velocities be reduced, suggesting that pressure losses incur greater penalties than the rewards of enhanced heat transfer.

A possible explanation for this fact can be found through an examination of the constituent elements of the overall heat transfer coefficient. Note that this number is calculated by taking the reciprocal of the sum of the heat flow path resistances:

$$U_H = \frac{1}{R_i + R_o + R_F + R_W} = \frac{1}{.0016+.0013+.005+.00009}$$

It is clear that the resistance due to fouling, R_F, is

Table III-3.

FIRST ZONE 1 SOLUTION

Heat exchanger length, ℓ	6.1 ft
Heat exchanger width, w	20.0 ft
Heat exchanger height, a	6.1 ft
Heat exchanger surface area, A_H	24,924 ft^2
Number of tubes, N_T	31,368
Heat exchanger capital cost, z_1	$656,711
Seawater	
Flow rate, $\dot{m}_H$	$21.1 \cdot 10^6$ lbm/hr
Reynolds number, R_{eH}	8672
Prandtl number, P_{rH}	6.79
Nusselt number, N_{uH}	73.3
Heat transfer resistance, R_i	.00162ft^2-hr-°F/Btu
Friction factor, f_H	.033
Friction pressure drop, DP_H	21.7 lbf/ft^2
Friction pump power, WP_H	3 KW
Pump capital cost, z_{21}	$382,834
Working Fluid	
Flow rate, $\dot{m}$	$8.5 \cdot 10^6$ lbm/hr
Reynolds number, R_{ec}	7823
Prandtl number, P_r	2.08
Nusselt number, N_u	100.5
Heat transfer resistance, R_o	.00126 ft^2-hr-°F/Btu
Friction factor, f	.125
Friction pressure drop, DP	65 lb_f/ft^2
Friction pump power, WP	5.8 KW
Pump capital cost, z_{22}	$341,155
Fouling heat transfer resistance, R_F	.005 ft^2-hr-°F/Btu
Tube wall heat transfer resistance, R_w	.00009 ft^2-hr-°F/Btu
Overall heat transfer coefficient, U_H	125 Btu/hr-ft^2-°F
Total zone 1 cost, z	$1,427,067

dominating the overall heat transfer coefficient. Even doubling both of the convective coefficients would only achieve an 18 percent improvement in overall transfer. To achieve this small improvement, the Reynolds numbers would have to be more than doubled, which would increase pumping power requirements by almost a factor of 5.

It should be pointed out that this particular performance trade-off constitutes one of the major artificialities of the simplified sample model. In a more realistic system the working fluid would be vaporized and the difference between the saturation pressures of the hot and cold working fluid would represent the bulk of the pump head. Under these circumstances, friction losses would not be nearly so controlling, and the heat transfer/pressure loss balance might reverse. If the system were as described, however, there is an inexorable logic in the dimensional relationships.

Much additional information can be identified within the computed data, some of which will be pointed out later. For now, just observe the remarkable balance among the cost elements. Not only are the capital costs of the seawater and working fluid pumps about the same, but their total is of about the same order as the capital cost of the heat exchanger. The ratio of three between seawater and working fluid flow rates begins to make sense when related to these balanced costs.

G. TESTING THE SOLUTION

The fact that a solution was obtained to the initial zone problem was most encouraging.

The possibility remained, however, that the minimum found was not unique and hence did not represent a global solution.[4] Although consideration of the smoothness of the functions suggested that the local solution was also a global one, further tests were necessary to strengthen that conviction. A variety of starting points was accordingly devised in order to find out if the same solution resulted. Each of the $\tilde{X}_o$ vectors in Table III-4 led to the identical solution.

A number of other computational controls were also varied without altering the resulting solution. Tolerances, completion criteria, differencing step sizes, scaling factors, and minimization methods were specified over wide ranges, and the identical solution resulted each time.

Additionally, SUMT4 contains its own tests for convexity. If the matrix of second partial derivatives of the P function is not positive definite at any step in the computation a warning message is printed to alert the user that the problem is probably not convex (the algorithm proceeds despite this, with an orthogonal move).

This warning message was occasionally received once or twice at the beginning of a computational sequence when the initial vector was radically different from the eventual

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[4]In this section, the terms local and global are used in relation to the zone 1 optimization problem.

Table III-4.

INITIAL $\underset{\sim}{X}_o$ VECTORS

X_1	X_2	X_3	X_4
1	18	18	18
18	1	18	18
18	18	1	18
18	18	18	1
5	18	18	18
18	5	18	18
18	18	5	18
18	18	18	5
10	18	18	18
18	10	18	18
18	18	10	18
18	18	18	10
10	5	5	5
10	5	10	10
10	5	5	10
10	10	5	5
10	10	10	10
20	1	18	18
25	5	1	8
50	18	1	18
50	1	18	18

solution, but any reasonable starting point proceeded through the sequence without orthogonal moves being required.

Finally, belief in the uniqueness of the computed solution was supported by the consistent and rational way the solution shifted in response to variations in the supporting assumptions. In later sections, these variations are reported and analyzed.

Although the above arguments do not constitute a mathematical proof that the prerequisites for solution uniqueness exist in the sample model, they do provide strong support for such a conclusion.

H. ACCELERATION PROCEDURES

Besides the built-in acceleration features of the SUMT4 algorithm, the user may save substantial calculation time with a few simple steps once confidence is gained that the local solution is unique. Recall that SUMT4 solves a series of minimization problems controlled by the parameter r. Both the starting r and the ratio by which r is reduced in each step are under the control of the user. Typically, a fairly large initial r (like 1.0) is used when the solution is not known even approximately, and the first trial vector may be far from optimality. The reduction ratio brings r down gradually, with the result that the code can spend considerable time solving the wrong problem; i.e., a long series of large r subproblems, before arriving in the vicinity of the desired solution. Frequently in the course of the study however, the analyst wishes to observe the effects of

only small perturbations in the baseline configuration, e.g., the response to a 10 percent change in a material price. Using the old solution as a starting point, the initial r can be chosen small, resulting in only a few minimization subproblems before the new solution is obtained. With a little experience, the user can usually select r^o and the r reduction ratio such that exactly three subproblems are run, yielding second-order extrapolation results in the least amount of computer time.

A second control which the user has over computational time requirements is exercised through specification of subproblem tolerances and completion criteria. These should be carefully selected so that computer time is not wasted in producing more precision in the solution than is needed.

With judicious use of the available controls, the experienced user can achieve solutions to problems such as analyzed here in less than fifteen seconds of computer time. Even more savings in computer time may be realized by modifying the SUMT4 algorithm for the particular type of problem analyzed. Depending on the circumstances, some subroutines may be deleted altogether, others shortened, and dimension statements reduced.

I. LINKING VARIABLE BEHAVIOR

Before proceeding with the zone 1 analysis, and without conducting a rigorous second-level analysis as described in II C, a measure of preliminary intuition may be gained concerning the linking variables T_H and T_C while remaining in the

present format.

Recall that in the first zone 1 solution, the requirement placed on the optimization program was to produce T_H = 75°F at the minimum cost. This T_H was chosen merely on the basis of using half of the available thermal difference for heat exchange, evenly divided between the heater and the condenser. As far as zone 2 is concerned, the higher T_H goes the better. Raising the enthalpy drop per unit mass of working fluid taken across the energy conversion device can do nothing but good in zone 2, both from a performance and cost viewpoint. In zone 1, on the other hand, the T_H tradeoffs become apparent. To bring T_H closer and closer to the ultimate limit, T_{HE}, requires increasing heat exchanger effectiveness, either through increased exchanger area, better heat transfer coefficients, or increased flow rates. Costs attach directly or indirectly to each of these improvements, leading to the expectation that at some point it will cost more to raise T_H than it is worth in terms of performance. For the first clue to optimal T_H selection, therefore, attention was focused in zone 1. Leaving all other temporary parameters unchanged, the zone 1 analysis was repeated for a range of T_H values. At each T_H point, optimum zone 1 X vectors were computed, along with zone costs and other intermediate variables. The results are given in Table 1, Appendix C. The most important relationship developed was that between T_H and zone 1 cost, shown in Figure 8.

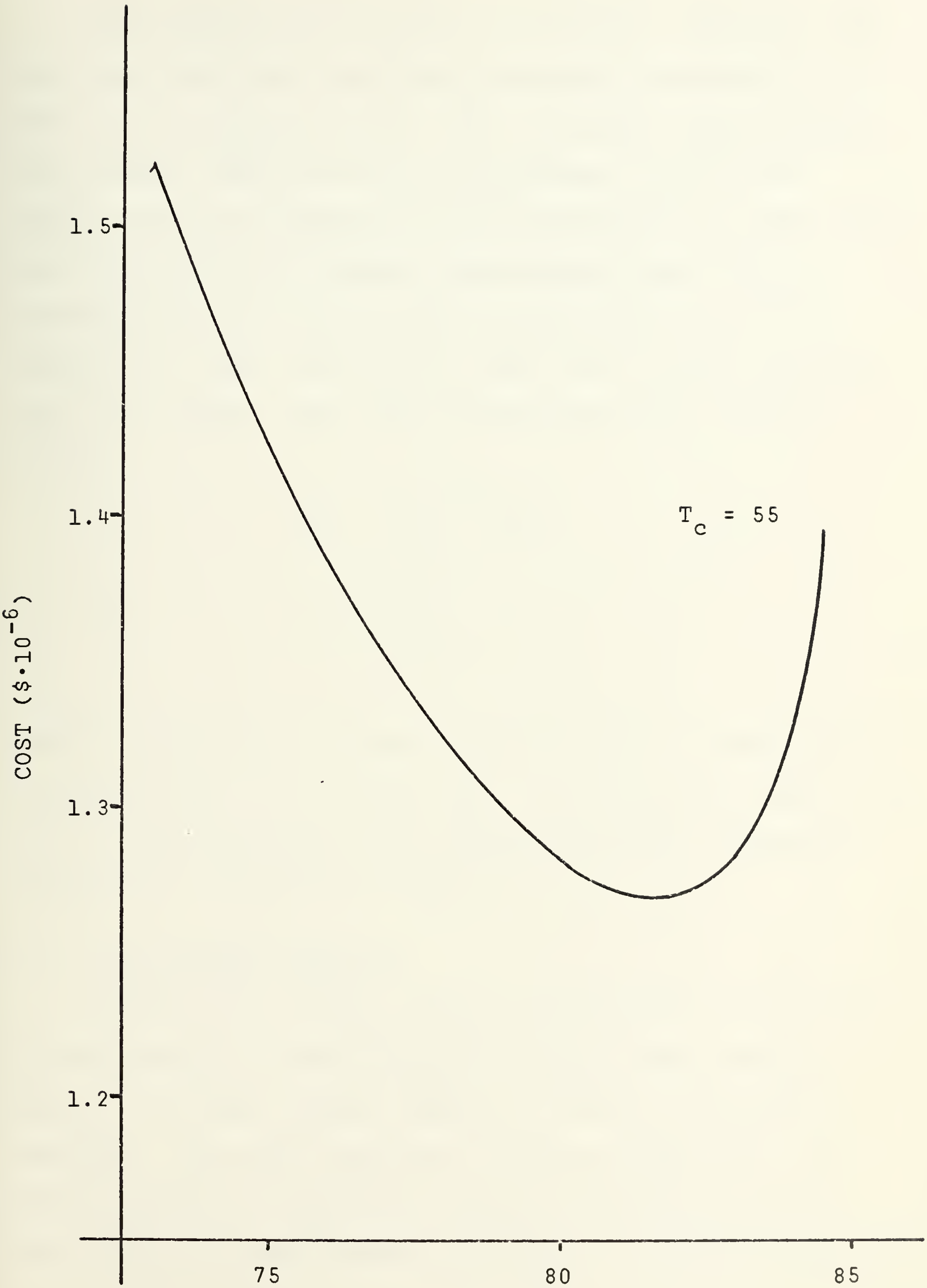

FIGURE 8. Cost vs. T_H

The implications of Figure 8 are clear. Certainly, at least with the other preliminary parameter assignments, T_H = 82°F is far superior to T_H = 75°F. As other specifications are altered, this will have to be checked, but for the present purposes, shifting the T_H specification to 82°F seems entirely justified. Without even checking zone 2 for a corresponding T_c^*, one can also surmise that T_c = 55°F is probably too high, and a T_c of, say, 50°F will be closer to the final figure. With these modifications the solution became

$$\dot{m}_H = 22 \cdot 10^6 \text{ lbm/hr}$$

$$\ell = 3.6 \text{ ft}$$

$$w = 20 \text{ ft}$$

$$a = 6.1 \text{ ft}$$

and the resulting cost figure was $1,075,991, a reduction of 25 percent from the baseline case. Rerunning cost against T_H at T_c = 50°F showed that the minimum cost point occurred at the same T_H^*, as shown in Figure 9.

J. ZONE VARIABLE BEHAVIOR

All the zone 1 input, output, and linking variables have by now been set at least to better figures than arbitrary guesses. The next logical step is to refine the interior variables. The most arbitrarily chosen of these was d, so the next study was concerned with gaining insight over the cost effects of tube diameter selection. A series of problems

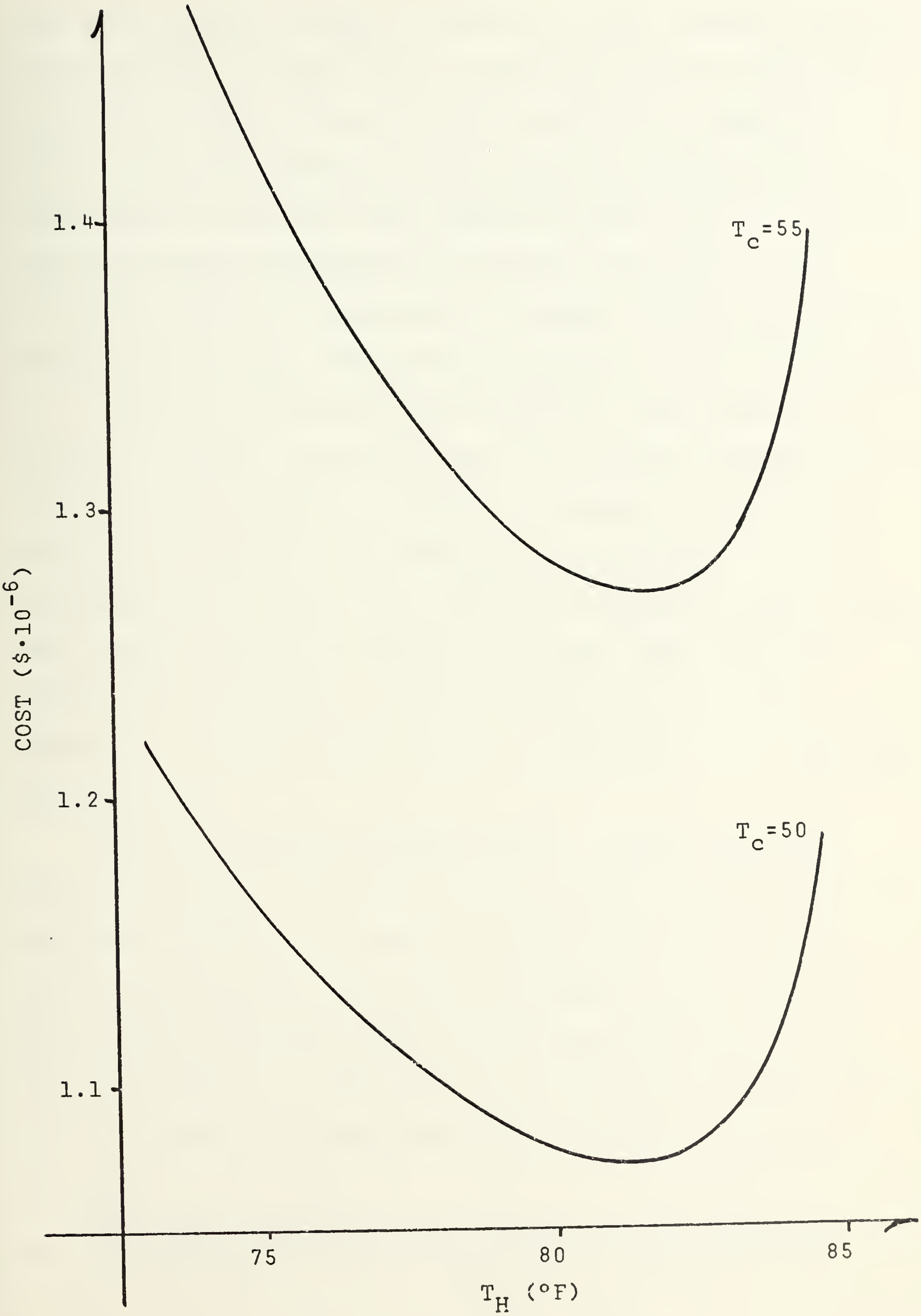

FIGURE 9. Cost vs. T_H

was solved using a range of possible d and t values[5] with the specifications of Table III-2 remaining constant except for T_H and T_C, which were set at 82°F and 50°F, as discussed above. At each d point, the corresponding optimum dimensions and seawater flowrates were computed along with the resulting costs and other intermediate variables. The results are tabulated in Table 2, Appendix C. Again, the most significant graph occurs in the cost-diameter plane, as shown in Figure 10.

In effect, thermoeconomics calls for tube diameters as large as possible (another example of type one information), requiring the designer to specify a maximum size based on practical information not contained in the model. Without conjecturing as to what the practical constraint might be, it was assumed that good reasons existed for using tubes no larger than 2.5 inches. System costs have by now been reduced an additional 55 percent, or 66 percent less than the baseline case.

The next parameter studied was R_F. Surely lower R_F values are better than higher ones, so the R_F analysis was not designed to lead to discovery of which way to go. Instead, this is the kind of study which develops the third type of information recoverable from thermoeconomic analysis: how much it is worth to achieve a given technological improvement. In other words, if an engineer could bring R_F from .005 to

---

[5]Tubewall thicknesses were related to tube diameters with the guidance of Table 2, Appendix C, reference [19].

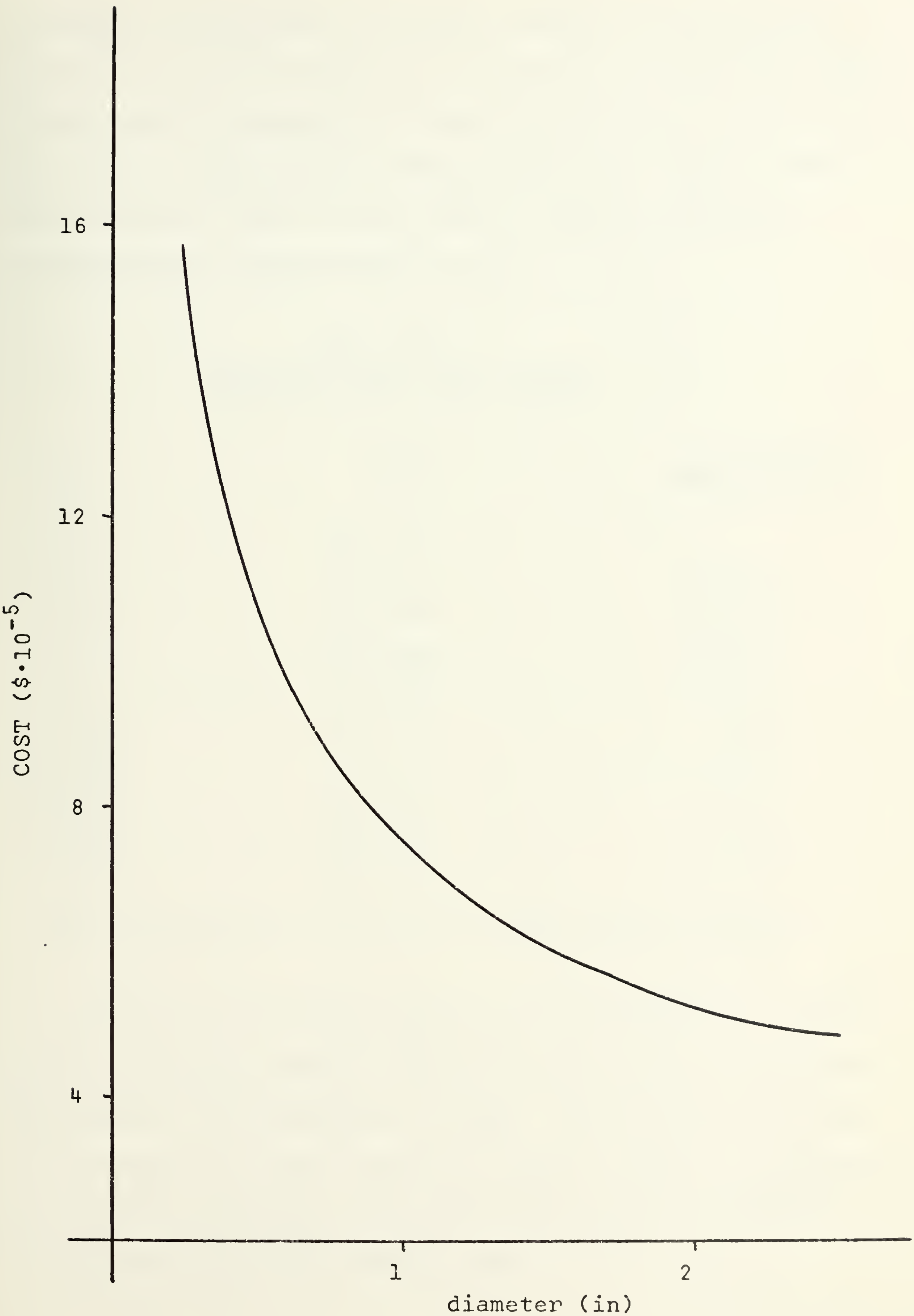

FIGURE 10. Cost vs. Tube Diameter

.0008 by the addition of special devices or non-fouling materials, how much should you be willing to spend for these technological advances? The data from the R_F runs are given in Table 3, Appendix C. Working with these figures leads to the following table of total zone 1 costs improvements achievable by the reduction of R_F.

Table III-5.
FOULING FACTOR COST EFFECTS

R_F	COST	COST SAVINGS*
.005	482,566	
.004	472,352	10,214
.003	461,306	11,046
.002	448,838	12,468
.001	434,592	14,246
.0009	433,044	1,548
.0008	431,470	1,574
.0007	429,869	1,601
.0006	428,239	1,630
.0005	426,579	1,660

*Cost Savings for reduction to next lower R_F figure.

An R_F of 0.0008 was assumed to be all that was technically achievable without prohibitive cost and was therefore specified in the developing design. This particular value also implied that heat transfer would no longer be dominated by fouling resistance, since the heat transfer resistance values for this latest design were:

$R_F = 0.0008$

$R_W = 0.0004$

$R_I = 0.0017$

$R_o = 0.0023.$

With new d and R_F values, it seemed prudent to recheck T_H^*, but, again, the minimum cost occurred at $T_H^* = 82°F$, and the cost - T_H graph had the same shape as in Figures 8 and 9.

The next variable examined was the arbitrary dimensional limitation of 20 feet, with the results shown in the following table, III-6 and Figure 11. Complete data are given in Table 4, Appendix C.

Table III-6.

DIMENSIONAL CONSTRAINT COST EFFECTS

CONSTRAINT	COST	COST SAVING*
20	431,470	--
25	391,303	40,177
30	361,406	29,897
35	337,994	23,412
40	318,995	18,999
45	303,159	15,836
50	289,686	13,473

*Cost Saving achievable by extending the constraint by five ft.

For the purpose of the sample analysis, it was assumed that the twenty foot constraint remained limiting.

Finally, the solution must be tested for sensitivity to the modeling equations themselves. The following table gives

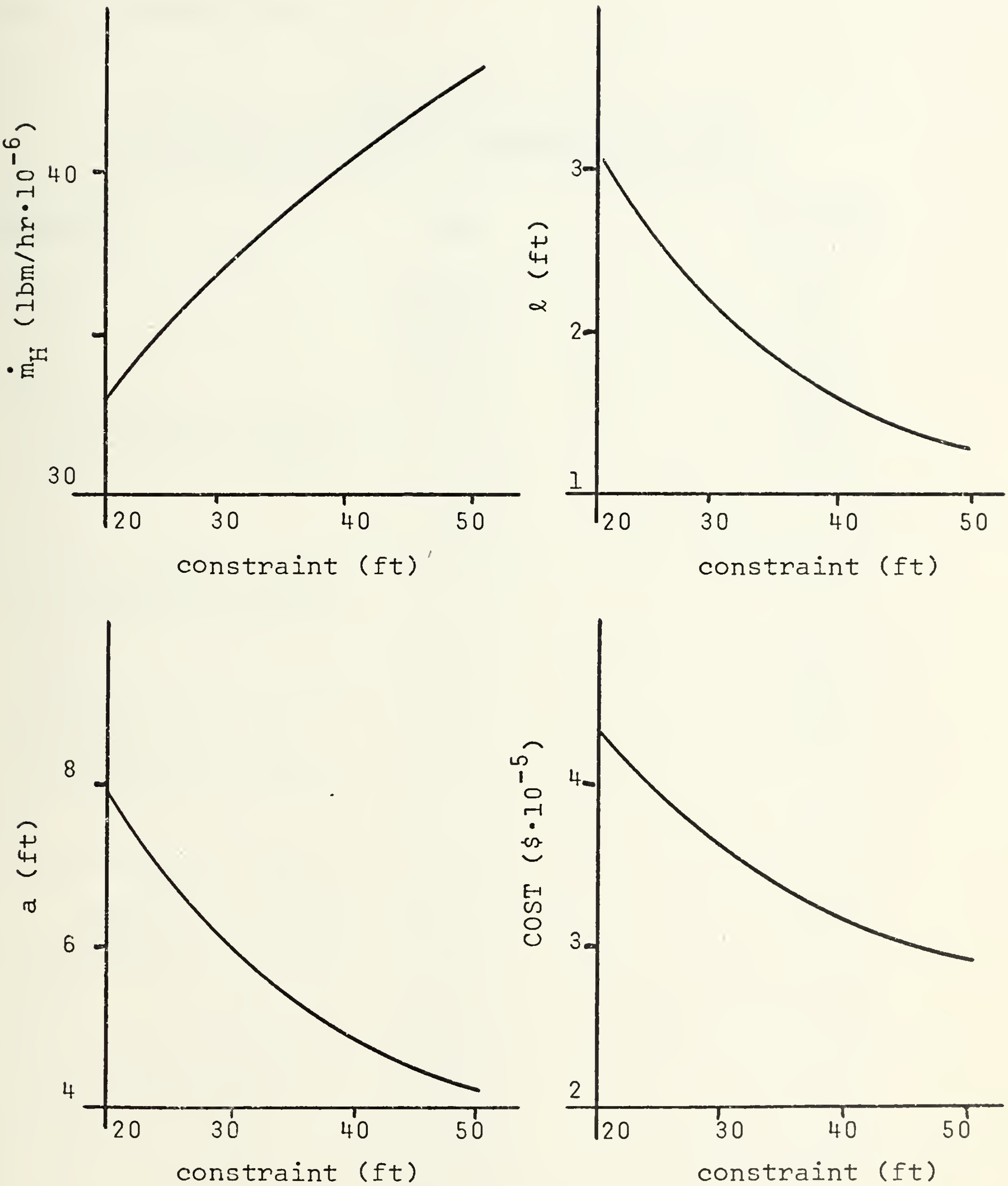

FIGURE 11. Dimensional Constraint Effects

the percent cost change in response to a 10% increase in selected parameters.

Table III-7.

COST SENSITIVITY TO MODELING EQUATIONS

PARAMETER	COST	% CHANGE
K_H	329,049	-23.7
μ	331,850	-23.1
ρ_H	335,976	-22.1
z_{22} equation exponent	486,293	+12.7
z_{21} equation exponent	482,772	+11.9
ψ	390,087	- 9.6
C_p	456,549	+ 5.8
f_H equation exponent	411,139	- 4.7
N_u equation exponent	412,371	- 4.4
C_{pH}	413,127	- 4.3
z_1 equation constant	449,470	+ 4.2
N_{uH} equation exp.	414,351	- 4.0
f equation exponent	415,449	- 3.7
ρ	417,366	- 3.3
z_{21} equation const.	443,359	+ 2.7
f equation constant	438,715	+ 1.7
f_H equation constant	438,801	+ 1.7
N_H	424,278	- 1.7
N	424,455	- 1.6
N_u equations const.	427,934	- 0.8
μ_H	434,518	+ 0.7
k	429,088	- 0.6
Po	432,221	+ 0.2
k_w	431,350	-0.03

It is interesting to note that three modeling parameters produce percentage cost shifts more than double the percentage change in the quantities themselves, and that they are all fluid properties. Besides suggesting that care be used in selecting the proper figures, these results indicate that an important refinement would be the use of variable properties as fluid state conditions change.

The least sensitive modeling parameter is tube wall conductivity. With the other heat flow path resistances being so much higher than that through the wall, this is a predictable result. If fouling resistance can be reduced, and boiling heat transfer included, however, this may well no longer be the case.

The strong sensitivity to certain fluid properties suggested a short side excursion into the cost effects of using alternate working fluids. Five of the leading contenders [37] were analyzed with the results shown in Table III-8.

Table III-8.
ALTERNATE WORKING FLUIDS

FLUID	SEAWATER FLOW RATE lbm/hr	LENGTH ft	HEIGHT ft	COST $
Ammonia	$33.1 \cdot 10^6$	3.1	7.9	431,470
R-12/31	$29.1 \cdot 10^6$	7.9	7.3	720,723
R-500	$30.6 \cdot 10^6$	6.8	7.6	680,112
R-31/114	$29.5 \cdot 10^6$	7.3	7.3	692,839
Propane	$30.8 \cdot 10^6$	5.4	7.6	591,571
Isobutane	$30.9 \cdot 10^6$	5.4	7.6	591,916

Recall, however, that these are strictly zone 1 costs, and that the working fluid significantly affects the performance of the turbine, cycle efficiency, and plant arrangements. [24,38] Still, the superiority in zone 1 of ammonia as a working fluid is clear.

K. ZONE ONE SUMMARY

At its present stage of development, the system design parameters are summarized in Table III-9.

Table III-9.
IMPROVED PARAMETER SELECTIONS

Gross plant power output	25 MW
Working fluid	ammonia
Hot seawater temperature	85°F
Cold seawater temperature	45°F
Hot working fluid temperature	82°F
Cold working fluid temperature	50°F
Tube diameter	2.5 inches
Tube wall thickness	0.148 inches
Tube spacing	3.75 inches
Fouling heat resistance	0.0008 ft^2-°F-hr/Btu
Tube wall conductivity	30 Btu/ft-hr-°F
Energy conversion factor	0.5 Btu/lbm-°F
Market price of energy	$0.03/kw-hr

The resulting design is summarized in Table III-10.

L. ANALYSIS OF ZONE TWO

Zone 2 is analyzed in exactly the same manner as zone 1, and the information developed is of the same form. Since the

Table III-10.

IMPROVED ZONE 1 SOLUTION

Heat exchanger length, ℓ	3.11 ft
Heat exchanger width, w	20.0 ft
Heat exchanger height, a	7.92 ft
Heat exchanger surface area, A_H	3298 ft^2
Number of tubes, N_T	1623
Heat exchanger capital cost, z_1	$184,757
Seawater	
Flow rate, $\dot{m}_H$	$33.1 \cdot 10^6$ lbm/hr
Reynolds number, R_{eH}	52,648
Prandtl number, P_{rH}	6.79
Nusselt number, N_{uH}	352
Heat transfer resistance, R_i	.0017 ft^2-hr-°F/Btu
Friction factor, f_H	.021
Friction pressure drop, DPH	2.1 lbf/ft^2
Friction pump power, W_{PH}	0.5 kw
Pump capital cost, z_{21}	$122,606
Working Fluid	
Flow rate, $\dot{m}$	$5.3 \cdot 10^6$ lbm/hr
Reynolds number, R_{ec}	47,796
Prandtl number, P_r	2.08
Nusselt number, N_u	278
Heat transfer resistance, R_o	.0023
Friction factor, f	.087
Friction pressure drop, DP	17.5 lbf/ft^2
Friction pump power, WP	1.0 kw
Pump capital cost, z_{22}	$116,585
Fouling heat transfer resistance, R_F	.0008 ft^2-hr-°F/Btu
Tube wall heat transfer resistance, R_w	.0004 ft^2-hr-°F/Btu
Overall heat transfer coefficient, U_H	192 Btu/ft^2-hr-°F
Total zone 1 cost, z	$431,470

sample model lacks sufficient realism to make the final actual figures meaningful, it was considered redundant to repeat these developments pending the introduction of more realistic two-phase flow conditions into both zones.

M. COMPLETION OF THE ANALYSIS

With $\tilde{X}_1^1$ fixed by the zone 1 analysis, $\tilde{X}_2^1$ determined similarly in zone 2, and preliminary information available as to the range of T_H^* and T_C^*, the sample problem could most easily be completed through application of the model coordination method described in II C. Had the user desired to employ the goal coordination method, the zone objective functions would have had to be cast in the form required by that approach from the outset. In either case, one can see that the resulting final design would precisely meet the original objective of balancing costs and benefits at the margin everywhere in the design. Believing this to be apparent, and considering the remaining time available, it was concluded that little would be added to the primary goal of this paper by presenting the details of the computation. Conclusive analysis will be much more valuable if applied to a realistic model as described in Appendix B.

If this is done, a conclusion of great interest will be a conclusive determination of an optimum plant output power level, G. With increasing returns to scale likely, this assignment will probably have to be reached on the basis of limitations to the feasibility of commercial manufacture of

turbines, platforms, or mooring systems and will accordingly not be arrived at trivially.

Even if this one parameter is selected arbitrarily, however, the savings achieved through setting the remaining design decision variables at their thermoeconomic optima should be considerably in excess of the cost of the analysis.

IV. SUMMARY AND CONCLUSIONS

The development in section III does not exhaust the possibilities for investigation of the thermoeconomic behavior of the sample model. It does, however, indicate the four types of information which can be derived:

1. Which parameters should be fixed as high or as low as practical considerations permit. Commercial availability or unit costs may provide the practical limit, or the constraint may be based on technical achievability, as in R_F, or base platform size limitations and hull arrangements (as in the overall heat exchanger size limit).
2. A point design of the system, based on a given specification of parameters which are not design variables.
3. What cost savings could be achieved if the practical limits discussed in 1 above are extended by a given amount.
4. The sensitivity of the design to variations in the modeling equations themselves.

Examples of each of these general classes of information are contained in section III K. In addition, Table IV-1 lists the major thermoeconomics results of the four design stages analyzed.

Certain lessons learned in the process of developing these data are considered important enough to warrant emphasis through further comment:

Table IV-1.

SUMMARY OF DESIGN EVALUATION

VARIABLE	1	2	3	4
$\dot{m}$ (lbm/hr)	8,500,000	5,334,000	5,334,000	5,334,000
$\dot{m}_H$(lbm/hr)	21,100,000	21,964,000	38,383,000	33,129,000
ℓ (ft)	6.07	3.61	3.34	3.11
w (ft)	20.0	20.0	20.0	20.0
a (ft)	6.13	6.12	9.83	7.92
A_H(ft^2)	24,924	14,815	4,403	3,297
$_{eH}$(non-dim)	8,672	8,217	49,086	52,648
$_{ec}$(non-dim)	7,823	9,038	44,484	47,796
U_H(Btu/ft^2hr°F)	125	127	104	192
o_p(psf)	65	71	19	17
o_{pH}(psf)	22	14	2	2
z_1 (\$)	657,000	474,000	221,000	185,000
z_{21}(\$)	383,000	300,000	130,000	123,000
z_{22}(\$)	341,000	271,000	123,000	117,000
z_2 (\$)	724,000	571,000	253,000	239,000
Cost (\$)	1,427,000	1,076,000	483,000	431,000

1 - Original design as per table III-2.

2 - T_H = 82°F, T_c = 50°F

3 - d = 2.5", t = 0.148"

4 - R_F = 0.008

Realistic, large scale systems (such as discussed in Appendix B) are undoubtedly best handled through a zone approach. Analytical tools are available to insure that the solutions to the zone sub-problems are coordinated such that an overall system optimum is achieved. Vapor power systems lend themselves readily to the zone approach in that the functions of the major components are distinct and the linking relationships are clear.

There is a simple rule for deciding which zone variables should be free to vary in the optimizing algorithm and which should be externally controlled. If the cost effects of varying a parameter can be readily included in the statement of the objective function, the parameter may be included in the vector of decision variables. If not, the parameter should be fixed during the minimization search and investigated separately. Sensitivity analysis reveals which of these affect the design sufficiently that they require careful selection. Those that do should be examined further, either to develop approximate costing relationships or to find other valid means for establishing their final specification.

The SUMT4 algorithm was found to be convenient and effective in finding solutions to the zone optimization problems. Although not demonstrated, it is expected that it would be equally capable of handling the second level coordination problem. Used as a subroutine, it should be straightforward to mechanize SUMT into a master program for conducting the iterative first level-second level procedure

discussed in section III E. Substantial economies in computation time appear possible of achievement by adapting the general program to the specific application.

The principal conclusion of this study is that thermoeconomic analysis of the kind suggested is capable of producing insights which are of considerable value to the system designer. This is particularly true when the system in question is such that the long run cost effects of technical design specifications are not well known on the basis of extensive previous experience with similar systems. Environmental power systems are such a class, and the financial imperatives which result from their considerable size, coupled with the recognized need for their early success, strongly suggest the wisdom of carefully applying the integrated cost/performance analysis techniques presented in this paper.

APPENDIX A

DEVELOPMENT OF THE THERMOECONOMIC MODEL

I. FLUID FLOW AND HEAT EXCHANGE EQUATIONS

A. Basic Model Description

The basic model is described in sections II B and C of the text. Ammonia was chosen as the working fluid because of its favorable heat exchange properties [24], but also note the practical objections to ammonia contained in [25]. From among the candidate thermal sources and sinks, those involved in the ocean thermal energy conversion process [26] provide the low thermal potential typical of environmental energy sources and were therefore chosen for the sample model.

B. Pump Work, Working Fluid (See Knudsen and Katz [27])

The work required to pump the working fluid against friction losses in the heat exchanger is

$$W_p = \frac{\dot{m}DP}{\rho\ N} \qquad \frac{ft - lb_f}{hr}$$

where DP depends on a friction factor which is measured empirically for the given configuration:

$$DP = \frac{2\ f\ G_H^2\ N'}{\rho\ g_c} \qquad \frac{lb_f}{ft^2}\ .$$

For the specific application considered,

$$G_H = \frac{\dot{m}\ S_t}{w\ell(S_t - d)\ 3600} \qquad \frac{lbm}{ft^2 sec}$$

$$N' = \frac{a}{S_L} \qquad \text{dimensionless}$$

and

$$f = 0.75\, R_{ec}^{-0.2} \qquad \text{dimensionless}$$

$$(100 < R_{ec} < 20{,}000)$$

where

$$R_{ec} = \frac{\dot{m}\, S_T}{\mu \ell w} \qquad \text{dimensionless}$$

C. Pump Work, Hot Seawater (See Streeter [28])

The pump work required to pump the hot seawater against friction losses through the heat exchanger tubes is

$$W_H = \frac{\dot{m}_H\, DP_H}{\rho_H\, N_H}$$

with

$$DP_H = f_H\, \frac{\ell \rho_H\, U_s^2}{2d}$$

$$U_s = \frac{4\, \dot{m}_H\, S_T S_L}{\pi\, \rho_H d^2\, wa}$$

and

$$f_H = 0.316\, R_{eH}^{-1/4} \qquad (R_{eH} < 10^5)$$

where

$$R_{eH} = \frac{4\, \dot{m}_H\, S_T S_L}{\pi\, \mu_H\, dwa} .$$

D. Heat Exchange Equations

The overall heat exchange coefficient is given by:

$$U_H = \frac{1}{R_i + R_o + R_w + R_F}$$

where:

1. R_F is the thermal resistance due to fouling and is an externally controlled parameter in the analysis.

2. R_w is the thermal resistance due to the conduction path through the tube walls, and is computed from

$$R_w = \frac{A_i t}{A_{mw} k_w} .$$

3. R_i is the thermal resistance at the seawater/tube interface, $\frac{1}{h_i}$. The inner film coefficient, h_i is given by [29]

$$\frac{h_i d}{k_H} = Nu_H = 0.036\ R_{eH}^{0.8} Pr_H^{1/3} \left(\frac{d}{\ell}\right)^{0.055}$$

where

$$R_{eH} = \frac{4\ \dot{m}_H\ S_T S_L}{\pi\ \mu_H\ dwa} \text{, as before,}$$

and

$$Pr_H = \frac{C_{pH}\ \mu_H}{k_H} .$$

4. R_o is the thermal resistance at the ammonia/tube interface, $\frac{A_i}{A_o h_o}$. The outer film coefficient, h_o, is given by [29]

$$\frac{h_o d}{k} = Nu = 0.511\ R_{ec}^{0.562} Pr^{1/3}$$

where

$$R_{ec} = \frac{\dot{m}\ S_t}{\mu \ell w} \text{, as before,}$$

and

$$Pr = \frac{C_p \mu}{k} .$$

II. COST EQUATIONS

The costs accounted for in sample model were the capital cost of the heat exchanger and pumps and the opportunity cost of the parasitic pumping power. Rather than encumber the model with a detailed accounting of discounted cash flows, a simple financing method was assumed which was believed to

provide cost assignments which were adequate for demonstration purposes. Specifically, amortization and interest expense were assigned as if the capital investment were funded with a twenty year, ten percent mortgage. The twenty year life was based on the consensus of the working group for economics, ocean thermal energy conversion workshop, held under the auspices of the National Science Foundation in Washington, D.C. in September, 1974 [30]. The ten percent interest rate corresponds to the discount rate specified for all public programs by current government directives. All costs were expressed in terms of the total cost over the twenty year life of the plant. Capital costs were estimated from the data contained in [20] and multiplied by a factor of 2.3 to approximate current dollars.

A. Capital Cost, Heat Exchanger

The base cost (BC) graph for shell and tube heat exchangers given in the reference was converted into functional form as

$$BC = 111\ A_H^{0.627}, \text{ where } A_H = \frac{\pi d\ \ell a w}{S_T S_L}.$$

Applying appropriate adjustment factors in accordance with the instructions contained in the reference, the cost estimating relationship chosen to represent the capital cost of the heat exchanger became

$$\text{Exchanger Cost} = z_1 = 500\ A_H^{0.627}.$$

B. Capital Cost, Pumps

The base cost graph for centrifugal pumps and drivers translates into

$$BC = 84\ (C \cdot H)^{0.602} .$$

Applying material adjustment factors for monel (seawater pumps) and stainless steel (ammonia pumps) the costs become

$$z_2 = z_{21} + z_{22}$$
$$z_{21} = 271\ (CH_1)^{0.602}$$
$$z_{22} = 162\ (CH_2)^{0.602}$$

where the CH terms are the product of the flow rate in gallons per minute and the pump head in pounds per square inch.

C. Cost of Pumping Power

Since the total system cost was formulated as

$$COST = p_o(W_p + W_H) + z_1 + z_2$$

it was necessary to use a p_o factor which brought the power cost term into the same units as the z terms; i.e., cost over a twenty year period. Applying the necessary unit conversion factors, p_o was used as

$$p_o = 0.0019795 \quad \frac{\$ - hr}{ft - lb_f} \quad 20\ yrs.$$

APPENDIX B
EXTENSION TO A REALISTIC SYSTEM

The sample analysis of section III indicates that valuable information could be obtained if the thermoeconomic method of anlysis were applied to a realistic system model. This Appendix outlines suggestions for modifying and extending the sample model so that the insights developed will be meaningful in the design of an actual working prototype.

Several groups have already conducted extensive study of the ocean thermal energy conversion process, leading to a number of preliminary designs [31, 32]. Probably the most complete design from the viewpoint of detailed system modeling is that of the University of Massachusetts (Amherst) [31]. With the engineering relationships already developed to the point of a comprehensive, coordinated, feasible design, this model is a very attractive candidate for thermoeconomic analysis.

A. ENGINEERING MODIFICATIONS

The most dramatic modification to the sample model results when two-phase heat transfer is included. The difference in saturation pressures at T_H = 82°F and T_C = 50°F amounts to almost 10,000 lbf/ft^2, dwarfing the 17 lbf/ft^2 friction pressure drop in the sample model. The obvious result is that reduction in flow velocity will no longer be the dominant objective of the optimization search and it may well result

that higher velocities are more advantageous than lower ones.

The second major change resulting from the inclusion of two-phase flow is in the area of heat exchange calculations. It is well known that boiling heat transfer coefficients are much higher than those for compressed liquids. With the working fluid heat transfer resistance becoming negligible in comparison with the other path resistances, further major alterations in the optimum design can be anticipated. Tube wall resistance may become a significant contributer and will have to be carefully specified.

Again, much of the background engineering in this area has already been done. The Amherst team has adapted the Chen [33] and Chawla [34] correlations for boiling heat transfer to the plate-fin exchanger configuration they recommend.

A second refinement which may strongly influence the thermoeconomic balance is the inclusion of the multiple fluid path flow losses commonly referred to as minor losses. Preliminary calculations show that these may well substantially exceed the friction and form losses which were the only flow losses accounted for in the sample model. Added to the substantial power required to pump the cold seawater from a depth of some 2000-3000 ft, pumping costs are certain to shift the zone costing relationships radically.

The sensitivity of the sample model to the physical characteristics of the working fluid suggests that greater attention be given to its selection. As reported by [35], these properties also have a strong influence on turbine design.

B. COSTING MODIFICATIONS

First, the tube and shell exchanger correlation used in the sample analysis will have to be modified for applicability to the plate-fin configuration. Second, note that the pump cost estimating relationship employed was for centrifugal pumps in ordinary industrial use. The pumps in an ocean thermal plant, which are required to drive very high volume rates of flow against very small pump heads, will necessarily be quite different. Something on the order of shrouded propellors in axial flow will be better suited to the application, and these will have different cost estimating formulas altogether.

In both cases, much greater care will be required in selecting appropriate materials and insuring commonly valued dollars than was taken in the model of section III.

But probably the most difficult costing relationships to estimate will be those associated with the size of the hull or platform required to support the plant. These costs control the component dimensions which can be chosen, and will probably dictate the very important ultimate selection of optimum plant power output.

C. FINANCING CONSIDERATIONS

Rather than assuming a single financing method, the realistic analysis should apply the general methods for evaluating investment decisions. The overall cash flow inherent in the project should be estimated as to its time dimension and discounted to its net present value. The

assumptions recommended by [36] should be utilized in performing this step.

D. COMPUTATIONAL CONSIDERATIONS

It is recommended that any realistic analysis be decomposed into at least three zones grouped about the three major functional components: the boiler, turbine, and condenser. The linking variables should consist of the working fluid state variables at the entrance and exit to these components. In addition to the temperatures, it will probably be necessary to include vapor quality as a linking variable, in that the performance of all three components is affected by the moisture content.

Either the model or goal method may be used in establishing problem coordination. Model coordination is the more straightforward of the two, but the auxiliary information produced in the process of goal coordination should be considered in making the selection.

In either case, the iterative procedure required should be mechanized into a controlling program, with SUMT or some other appropriate optimization algorithm being applied sequentially to the first and second level problems.

APPENDIX C

TABLE 1. RESULTS OF T_H INVESTIGATION

Input

G = 25MW

ψ = 389 ft-lbf/lbm°F

T_H = various

T_{HE} = 85°F

N = 0.9

P_r = 2.08

d = 0.5 inches

R_F = 0.005

P_o = 0.03/kw-hr

$S_T = S_L$ = 1.5d

T_c = 55°F

T_{CE} = 45°F

N_H = 0.9

P_{rH} = 6.79

t = 0.035 inches

Output

T_H (°F)	$\dot{m}$ (lbm/hr·10^6)	$\dot{m}_H$ (lbm/hr·10^6)	ℓ (ft)	w (ft)	a (ft)
73	9.5	21.0	6.8	20	6.1
74	9.0	21.0	6.4	20	6.1
75	8.5	21.1	6.1	20	6.1
76	8.1	21.2	5.8	20	6.1
77	7.8	21.4	5.5	20	6.2
78	7.4	21.7	5.2	20	6.2
79	7.1	22.0	5.0	20	6.3
80	6.8	22.5	4.7	20	6.4
81	6.6	23.2	4.5	20	6.5
82	6.3	24.2	4.3	20	6.7
82.5	6.2	24.8	4.2	20	6.8
83	6.1	25.7	4.1	20	7.0
83.5	6.0	26.9	3.9	20	7.2
84	5.9	28.5	3.8	20	7.6
84.5	5.8	31.7	3.6	20	8.1

T_H (°F)	R_{ec} (non-dim)	f (non-dim)	o_p (psf)	W_p (kw)	N_u (non-dim)
73	7762	.125	64	6.4	100
74	7792	.125	65	6.1	100
75	7824	.125	65	5.8	101
76	7858	.125	66	5.6	101
77	7895	.125	67	5.4	101
78	7937	.124	68	5.3	101
79	7985	.124	69	5.2	102
80	8042	.124	71	5.1	102
81	8113	.124	74	5.1	103
82	8206	.124	78	5.2	103
82.5	8267	.123	80	5.2	104
83	8344	.123	84	5.3	104
83.5	8476	.123	89	5.6	105
84	8599	.123	95	5.9	106
84.5	8881	.122	109	6.6	108

T_H (°F)	R_{eH} (non-dim)	f_H (non-dim)	o_{pH} (psf)	W_H (kw)	N_{uH} (non-dim)
73	8620	.033	24	3.3	73
74	8645	.033	23	3.1	73
75	8672	.033	22	3.0	73
76	8703	.033	21	2.9	74
77	8738	.033	20	2.8	74
78	8779	.033	19	2.7	75
79	8827	.033	18	2.6	75
80	8885	.033	18	2.6	76
81	8960	.032	17	2.6	76
82	9060	.032	17	2.6	77
82.5	9128	.032	16	2.7	78
83	9213	.032	16	2.7	79
83.5	9353	.032	16	2.8	80
84	9499	.032	16	3.0	81
84.5	9820	.032	16	3.3	83

T_H (°F)	A_H (ft^2)	N' (non-dim)	N_{ToT} (non-dim)
73	27,971	98	31,433
74	26,342	98	31,368
75	24,924	98	31,369
76	23,688	98	31,441
77	22,612	99	31,591
78	21,680	99	31,834
79	20,881	101	32,187
80	20,212	102	32,685
81	19,676	104	33,382
82	19,294	107	34,383
82.5	19,174	110	35,061
83	19,116	112	35,922
83.5	19,051	116	37,018
84	19,312	121	38,737
84.5	19,753	130	41,624

T_H (°F)	R_o	R_i	R_w	R_F	U_H (Btu/ft^2hr°F)
	←	(ft^2 hr °F/Btu)		→	
73	.0013	.0016	.00009	.005	125
74	.0013	.0016	.00009	.005	125
75	.0013	.0016	.00009	.005	125
76	.0013	.0016	.00009	.005	125
77	.0013	.0016	.00009	.005	126
78	.0013	.0016	.00009	.005	126
79	.0012	.0016	.00009	.005	126
80	.0012	.0016	.00009	.005	126
81	.0012	.0016	.00009	.005	127
82	.0012	.0015	.00009	.005	127
82.5	.0012	.0015	.00009	.005	127
83	.0012	.0015	.00009	.005	128
83.5	.0012	.0015	.00009	.005	128
84	.0012	.0015	.00009	.005	129
84.5	.0012	.0014	.00009	.005	130

T_H (°F)	z_1 ($\$\cdot 10^{-3}$)	z_{21} ($\$\cdot 10^{-3}$)	z_{22} ($\$\cdot 10^{-3}$)	z_2 ($\$\cdot 10^{-3}$)	COST ($\$\cdot 10^{-3}$)
73	706	406	361	767	1,524
74	680	394	350	744	1,473
75	657	383	341	724	1,427
76	636	373	333	707	1,387
77	618	366	327	692	1,353
78	602	359	321	681	1,324
79	588	354	317	672	1,301
80	576	351	315	666	1,283
81	566	350	315	665	1,272
82	559	353	317	670	1,270
82.5	557	356	320	675	1,275
83	556	361	324	684	1,283
83.5	555	369	332	701	1,300
84	560	382	343	725	1,331
84.5	568	409	367	776	1,396

TABLE 2. RESULTS OF TUBE DIAMETER INVESTIGATION

Input

$G = 25MW$

$\psi = 389$ ft-lbf/lbm-°F

$T_H = 82°F$

$T_{HE} = 85°F$

$N = 0.9$

$P_r = 2.08$

$d =$ various

$p_o = 0.03$/Kw-hr

$S_T = S_L = 1.5d$

$T_c = 50°F$

$T_{CE} = 45°F$

$N_H = 0.9$

$P_{rH} = 6.79$

$t =$ various

Output

d (in)	t (in)	$\dot{m}$ (lbm/hr·10^{-6})	$\dot{m}_H$ (lb/hr·10^{-6})	ℓ (ft)	w (ft)	a (ft)
1/4	.00183	5.334	18.276	3.61	20.0	5.10
3/8	.00233	5.334	20.223	3.63	20.0	5.67
1/2	.00242	5.334	21.961	3.61	20.0	6.12
5/8	.00350	5.334	23.520	3.59	20.0	6.51
3/4	.00408	5.334	24.942	3.57	20.0	6.85
1.0	.00408	5.334	27.473	3.52	20.0	7.44
1.25	.00408	5.334	29.703	3.48	20.0	7.95
1.5	.00542	5.334	31.715	3.44	20.0	8.40
2.0	.00792	5.334	35.270	3.38	20.0	9.18
2.5	.01233	5.334	38.383	3.34	20.0	9.85

d (in)	R_{ec} (non-dim)	f (non-dim)	o_p (psf)	W_P (KW)	N_u (non-dim)
1/4	4,109	.142	136	7.6	70
3/8	6,131	.131	92	5.1	88
1/2	8,213	.124	71	4.0	103
5/8	10,334	.118	59	3.3	118
3/4	12,486	.114	50	2.8	131
1.0	16,864	.107	39	2.2	155
1.25	21,326	.102	33	1.8	177
1.5	25,856	.098	28	1.6	197
2.0	35,079	.092	23	1.3	234
2.5	44,484	.088	19	1.1	267

d (in)	R_{eH} (non-dim)	f_H (non-dim)	o_{pH} (psf)	W_H (KW)	N_{uH} (non-dim)
1/4	4,509	.039	33	3.9	43
3/8	6,738	.035	20	2.6	60
1/2	9,035	.032	14	2.0	78
5/8	11,375	.031	11	1.6	95
3/4	13,751	.029	8	1.4	112
1.0	18,585	.027	6	1.1	144
1.25	23,512	.026	5	.9	177
1.5	28,515	.024	4	.8	208
2.0	38,700	.023	3	.6	270
2.5	49,086	.021	2	.5	331

d (in)	A_{H_2} (ft^2)	N' (non-dim)	N_{TOT} (non-dim)
1/4	24,703	163	104,520
3/8	18,391	120	51,595
1/2	14,823	98	31,337
5/8	12,528	83	21,327
3/4	10,915	73	15,590
1.0	8,781	60	9,529
1.25	7,419	51	6,315
1.5	6,465	45	4,780
2.0	5,206	37	2,938
2.5	4,403	32	2,016

d	R_e	R_i	R_w	R_F	U_H
(in)	<———	ft^2 - °F - hr/Btu		———>	Btu/ft^2hr°F
1/4	.0009	.0014	.00006	.005	136
3/8	.0011	.0015	.00007	.005	131
1/2	.0012	.0015	.00009	.005	127
5/8	.0014	.0016	.00011	.005	124
3/4	.0015	.0016	.00013	.005	122
1.0	.0017	.0017	.00013	.005	118
1.25	.0018	.0017	.00013	.005	115
1.5	.0020	.0017	.00018	.005	113
2.0	.0020	.0018	.00026	.005	108
2.5	.0024	.0018	.00040	.005	104

d (in)	z_1 $\$\cdot10^{-3}$	z_{21} $\$\cdot10^{-3}$	z_{22} $\$\cdot10^{-3}$	z_2 $\$\cdot10^{-3}$	COST $\$\cdot10^{-3}$
1/4	653	451	401	852	1566
3/8	543	353	317	670	1253
1/2	474	300	271	571	1076
5/8	427	265	241	506	958
3/4	391	240	219	460	873
1.0	341	206	190	396	755
1.25	307	184	170	354	676
1.5	282	168	156	324	618
2.0	246	145	136	281	537
2.5	221	130	123	253	483

TABLE 3. RESULTS OF FOULING FACTOR INVESTIGATION

Input

G = 25MW

ψ = 389 ft-lbf/lbm°F

T_H = 82°F

T_{HE} = 85°F

N = 0.9

P_r = 2.08

d = 2.5 inches

R_F = various

P_o = 0.03/Kw-hr

$S_T = S_L$ = 1.5 d

T_c = 50°F

T_{CE} = 45°F

N_H = 0.9

P_{rH} = 6.79

t = 0.148 inches

Output

R_F (ft^2°Fhr/Btu)	$\dot{m}$ ($lbm/hr \cdot 10^{-6}$)	$\dot{m}_H$ ($lbm/hr \cdot 10^{-6}$)	ℓ (ft)	w (ft)	a (ft)
.005	5.3	38.4	3.3	20	9.8
.004	5.3	37.3	3.3	20	9.5
.003	5.3	36.1	3.3	20	9.0
.002	5.3	34.8	3.2	20	8.6
.001	5.3	33.4	3.1	20	8.0
.0009	5.3	33.3	3.1	20	8.0
.0008	5.3	33.1	3.1	20	7.9
.0007	5.3	33.0	3.1	20	7.9
.0006	5.3	32.8	3.1	20	7.8
.0005	5.3	32.7	3.1	20	7.7

R_F (ft^2°Fhr/Btu)	R_{ec} (non-dim)	f (non-dim)	o_p (psf)	W_p (KW)	N_u (non-dim)
.005	44,486	.088	19	1.1	267
.004	44,969	.088	19	1.0	269
.003	45,583	.088	18	1.0	271
.002	46,392	.088	18	1.0	273
.001	47,515	.088	18	1.0	277
.0009	47,651	.087	18	1.0	277
.0008	47,796	.087	18	1.0	278
.0007	47,944	.087	17	1.0	278
.0006	48,103	.087	17	1.0	279
.0005	48,268	.087	17	1.0	280

R_F (ft^2hr°F/Btu)	R_{eH} (non-dim)	f_H (non-dim)	o_{pH} (psf)	W_H (KW)	N_u (non-dim)
.005	49,086	.021	2.0	.5	331
.004	49,585	.021	2.0	.5	334
.003	50,267	.021	2.0	.5	338
.002	51,138	.021	2.0	.5	343
.001	52,346	.021	2.1	.5	350
.0009	52,493	.021	2.1	.5	351
.0008	52,648	.021	2.1	.5	352
.0007	52,807	.021	2.1	.5	353
.0006	52,978	.021	2.1	.4	354
.0005	53,155	.021	2.1	.4	355

R_F (ft^{2}hr°F/Btu)	A_H (ft^2)	N' (non-dim)	N_{TOT} (non-dim)
.005	4402	32	2016
.004	4188	30	1939
.003	3949	29	1853
.002	3679	27	1757
.001	3367	26	1647
.0009	3333	26	1635
.0008	3298	25	1623
.0007	3262	25	1610
.0006	3225	25	1597
.0005	3188	25	1584

R_F (ft^{2}hr°F/Btu)	R_o	R_i	R_w	U_{H_2} (Btu/ft^{2}hr°F)
.005	.002	.002	.0004	104
.004	.002	.002	.0004	116
.003	.002	.002	.0004	133
.002	.002	.002	.0004	154
.001	.002	.002	.0004	185
.0009	.002	.002	.0004	188
.0008	.002	.002	.0004	192
.0007	.002	.002	.0004	196
.0006	.002	.002	.0004	201
.0005	.002	.002	.0004	226

R_F ($ft^2 hr°F/Btu$)	z_1 ($\$ \cdot 10^{-3}$)	z_{21} ($\$ \cdot 10^{-3}$)	z_{22} ($\$ \cdot 10^{-3}$)	z_2 ($\$ \cdot 10^{-3}$)	COST ($\$ \cdot 10^{-3}$)
.005	221	130	123	253	483
.004	215	128	121	250	472
.003	207	127	120	247	461
.002	198	125	118	243	449
.001	187	123	117	240	435
.0009	186	123	117	240	433
.0008	185	123	117	240	431
.0007	184	122	116	239	430
.0006	182	122	116	239	428
.0005	181	122	116	238	427

TABLE 4. RESULTS OF DIMENSIONAL CONSTRAINT RUN

Input

G = 25MW
ψ = 389 ft-lbf/lbm°F
T_H = 82°F
T_{HE} = 85°F
N = 0.9
P_r = 2.08
d = 2.5 inches
LMT = various

p_o = 0.03/Kw-hr
$S_T = S_L$ = 1.5d
T_c = 50°F
T_{CE} = 45°F
N_H = 0.9
P_{rH} = 6.79
t = 0.148 inches

Output

LMT (ft)	$\dot{m}$ ($lbm/hr\cdot 10^{-6}$)	$\dot{m}_H$ ($lbm/hr\cdot 10^{-6}$)	ℓ (ft)	w (ft)	a (ft)
20	5.33	33.1	3.1	20	7.9
25	5.33	35.3	2.5	25	6.8
30	5.33	37.2	2.1	30	6.0
35	5.33	38.9	1.8	35	5.4
40	5.33	40.4	1.6	40	4.9
45	5.33	41.9	1.4	45	4.5
50	5.33	43.2	1.3	50	4.2

LMT (ft)	R_{ec} (non-dim)	f (non-dim)	σ_p (psf)	W_p (KW)	N_u (non-dim)
20	47,798	.087	17.5	1.0	352
25	47,623	.087	14.9	0.8	354
30	47,492	.087	13.1	0.7	356
35	47,384	.087	11.7	0.7	358
40	47,296	.087	10.7	0.6	360
45	47,220	.087	9.8	0.5	362
50	47,161	.087	9.1	0.5	363

LMT (ft)	R_{eH} (non-dim)	f_H (non-dim)	σ_{pH} (psf)	W_H (KW)	N_{uH} (non-dim)
20	52,650	.021	2.1	0.5	352
25	52,349	.021	1.7	0.4	354
30	52,122	.021	1.4	0.3	357
35	51,935	.021	1.2	0.3	359
40	51,780	.021	1.0	0.3	360
45	51,647	.021	0.9	0.2	362
50	51,535	.021	0.8	0.2	363

LMT (ft)	A_H (ft^2)	N' (non-dim)	N_{TOT} (non-dim)
20	3297	25	1623
25	2836	22	1739
30	2509	19	1840
35	2262	17	1931
40	2067	16	2013
45	1911	15	2089
50	1780	14	2160

LMT (ft)	R_o	R_i	R_w	R_F	U_H ($Btu/ft^2{}^\circ Fhr$)
	<———— $ft^2{}^\circ F$ hr/Btu			————>	
20.	.0023	.0017	.0004	.0008	192
25	.0023	.0017	.0004	.0008	193
30	.0023	.0017	.0004	.0008	193
35	.0023	.0017	.0004	.0008	193
40	.0023	.0017	.0004	.0008	193
45	.0023	.0017	.0004	.0008	193
50	.0023	.0017	.0004	.0008	194

LMT (ft)	z_1 ($\$\cdot 10^{-3}$)	z_{21} ($\$\cdot 10^{-3}$)	z_{22} ($\$\cdot 10^{-3}$)	z_2 ($\$\cdot 10^{-3}$)	COST ($\$\cdot 10^{-3}$)
20	185	123	117	239	431
25	168	111	106	217	391
30	156	102	98	200	361
35	146	95	92	187	338
40	138	90	86	177	319
45	131	86	82	168	303
50	126	82	79	160	290

```
//SEAPWR  JOB (2014,0758,NG24),'SHEPPARD'
// EXEC FORTCLG,REGION.GO=180K
//FORT.SYSIN DD *
 THIS SUBROUTINE ESTABLISHES THE INITIAL VALUES OF THE
 MODELING PARAMETERS.
       SUBROUTINE READIN
       IMPLICIT REAL*8 (A-H,L-Z)
 STORAGE AREA SHARE IS IN COMMON WITH SUMT4.
       COMMON/SHARE/X(100),DEL(100),A(100,100),N,M,MN,NP1,NM1
 STORAGE AREA PROPS IS IN COMMON WITH SUBROUTINE RSTNT
       COMMON/PROPS/ST,SL,PI,D,MDOT,MU,MUH,PR,PRH,DK,DKH,C1,
      1C2,C3,C4,CP,CPH,PO,RHO,RHOH,ETA,ETAH,MDOTH,LMT
 BEGIN BY READING THOSE VALUES WHICH ARE DISTINCTIVE TO
 THIS RUN.
       READ (5,1000) TH
 1000 FORMAT ( F12.6)
 THEN ASSIGN THOSE VARIABLES WHICH ARE TO REMAIN CONSTANT
 AS THE ANALYZED VARIABLES  CHANGE.
       G=25.
       PO=.0019795
       PSI=232.
       LMT=20.
       D=2.5
       T=0.148
       TC=50.
       THE=85.
       TCE=45.
       PI=3.14159
       ETA=.9
       ETAH=.9
       RF=0.0008
       D=D/12.
       T=0.035/12.
       SL=1.5*D
       ST=1.5*D
       AO=D+T
       AI=D
       AMW=D+T/2.
       RHO=35.5
       RHOH=64.
       CP=.596
       CPH=1.
       MU=.400
       MUH=2.37
       DK=.0625
       DKH=0.349
       DKW=30.
       PR=CP*MU/DK
       PRH=CPH*MUH/DKH
       C1=AI*T/(AMW*DKW)
       C2=RF
       C3=AI/AO
       C4=(TH-TC)/(THE-TC)
       G=G*778000000./.29293
       MDOT=G/(PSI*(TH-TC))
 WRITE ANY MESSAGES DESIRED AT THE START OF EACH RUN.
       WRITE (6,3000) TH
 3000 FORMAT ('0','TH FOR THIS RUN WAS',F12.6)
       RETURN
       END
```

```
      SUBROUTINE  RESTNT(I,VAL)
C  SUBROUTIN RESTNT ESTABLISHES THE BASIC MODELING RELATION-
C  SHIPS.
      IMPLICIT REAL*8 (A-H,L-Z)
      COMMON/SHARE/X(100),DEL(100),A(100,100),N,M,MN,NP1,NM1
      COMMON/PROPS/ST,SL,PI,D,MDOT,MU,MUH,PR,PRH,DK,DKH,C1,
     1C2,C3,C4,CP,CPH,PO,RHO,RHOH,ETA,ETAH,MDOTH,LMT
C  FIRST COMPUTE THOSE NUMBERS WHICH WILL CHANGE WITH EACH
C  CALL TO RESTNT.
      MDOTH=X(1)*1000000.
      L=X(2)
      W=X(3)
      DA=X(4)
      NSLNT=DA/SL
      NTOT=(W*DA)/(ST*SL)
      AH=PI*D*L*NTOT
      REC=(MDOT*ST)/(MU*L*W)
      REH=(4.*MDOTH*ST*SL)/(PI*MUH*D*W*DA)
C  THE REMAINDER OF THE SUBROUTINE IS STRUCTURED SUCH THAT
C  ONLY THOSE COMPUTATIONS REQUIRED FOR THIS PARTICULAR CALL
C  ARE PERFORMED.
      IF (I.EQ.0) GO TO 3
      GO TO (5,6,7),I
      NU=0.511*(REC**0.562)*(PR**0.33333333)
      HO=NU*DK/D
      A2=C3/HO
      NUH=0.036*(REH**0.8)*(PRH**0.3333333)*((D/L)**0.055)
      HI=NUH*DKH/D
      A1=1./HI
      UH=1./(A1+A2+C1+C2)
      NH=(AH*UH)/(MDOT*CP)
      THETAH=(MDOTH*CPH)/(MDOT*CP)
      GAMMAH=1.-1./(DEXP(NH*THETAH))
      EFF=1.-1./(DEXP(GAMMAH*THETAH))
      CONSTR=C4-EFF
      VAL=CONSTR*100.
      GO TO 4
    3 DF=0.75/(REC**0.2)
      GH=MDOT*ST/(L*W*(ST-D))
      DP=2.*DF*GH*GH*NSLNT/RHO
      WP2=(MDOT*DP)/(RHO*ETA*3600.*3600.*32.2)
      DFH=0.316/(REH**0.25)
      VH=(4.*MDOTH)/(PI*RHOH*D*D*NTOT)
      DPH=(DFH*L*RHOH*VH*VH)/(2.*D)
      WH=(MDOTH*DPH)/(RHOH*ETAH*3600.*3600.*32.2)
      DEX=2.3
      Z1=500.*(AH**0.627)*DEX
      CH1=7.4805*WH/8640.
      FM1=3.23
      BC1=84.*(CH1**.602)
      PC1=BC1*FM1
      CH2=7.4805*WP2/8640.
      FM2=1.93
      BC2=84.*(CH2**0.602)
      PC2=BC2*FM2
      Z21=PC1*DEX*3.
      Z22=PC2*DEX*3.
      Z2=Z21+Z22
      SCALE=1000000.
      COST=PO*(WP2+WH)+Z1+Z2
      VAL=COST/SCALE
      GO TO 4
    5 CON2=(LMT-X(2))/LMT
      VAL=CON2
      GO TO 4
    6 CON3=(LMT-X(3))/LMT
      VAL=CON3
      GO TO 4
    7 CON4=(LMT-X(4))/LMT
      VAL=CON4
    4 RETURN
      END
```

```
      SUBROUTINE GRAD1 (I)
 IN GRAD1, THE FIRST DERIVATIVES MAY BE PROVIDED IN
 FUNCTIONAL FORM, OR, AS USED HERE, CENTRAL DIFFERENCING
 MAY BE SPECIFIED.
      IMPLICIT REAL*8 (A-H,M-Z)
      COMMON/SHARE/X(100),DEL(100),A(100,100),N,M,MN,NP1,NM1
      CALL DIFF1 (I)
      RETURN
      END
      SUBROUTINE MATRIX (J,L)
 CENTRAL DIFFERENCING IS SPECIFIED FOR THE SECOND DERIVA-
 TIVES, ALSO, ALTHOUGH THE FUCTIONAL FORM COULD HAVE BEEN
 SPECIFIED AS IN GRAD1
      IMPLICIT REAL*8 (A-H,M-Z)
      COMMON/SHARE/X(100),DEL(100),A(100,100),N,M,MN,NP1,NM1
      CALL DIFF2(J)
      RETURN
      END
 THE FOLLOWING STATEMENTS ACCOMPLISH THE LINK TO THE BULK OF
 THE ALGORITHM, WHICH IS IN STORAGE.
//LINK.USDD DD DSNAME=S2014.SEAPWR,UNIT=2321,
//             DISP=SHR,VOLUME=SER=CEL006
//LINK.SYSIN DD *
 INCLUDE USDD(SHEP)
 ENTRY MAIN
//GO.FT07F001 DD DUMMY
//GO.SYSIN DD *
```

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