ESSERGY OPTIMIZATION OF REGENERATIVE FEEDWATER HEATERS

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ESSERGY OPTIMIZATION OF REGENERATIVE

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NOMENCLATURE

Latin Letters

A	area, ft^2
A_c	cross-sectional area, ft^2
A_t	cross-sectional area of a tube, ft^2
$\dot{C}$	amortized capital cost flow, \$/hr
C_{CAP}	capital cost, \$
C_p	specific heat, Btu/lb°F
$\dot{C}$	total feedwater heater cost flow (based on second law analysis), \$/hr
$\dot{c}_A$	unit area cost, \$/yr-ft^2
$\dot{c}_A'$	unit area cost including head loss essergy dissipation, \$/yr-ft^2
c_e	unit essergy cost, \$/mm-Btu
c_m	unit material cost, e.g., \$/lb, \$/mole, etc.
c_t	unit cost of an essergy transport, \$/mm-Btu
$\dot{c}_z'$	cost per transfer unit, \$/hr
d	diameter, ft
d_t	tube diameter, ft
E	energy, Btu
$\dot{E}$	energy flow, Btu/hr
f	Fanning friction factor
h	convective heat transfer coefficient, Btu/hr-ft^2°R
h_ℓ	head loss, psig

$\dot{K}$	turbine stage cost constant (including maintenance costs, insurance, taxes, interest, etc.), \$/Btu
k	thermal conductivity, Btu/hr-ft°R
L	length, ft
$\mathcal{L}$	life, years
$\dot{M}$	material flow, e.g., lb/hr, moles/hr, etc.
$\dot{m}$	mass flow, lb/hr
N	number of tubes
N_c	quantity of material c, e.g., lbs, moles, etc.
$\dot{N}_c$	flow of material c, e.g., lbs/hr, moles/hr, etc.
P_o	datum state pressure
P_w	wetted perimeter, ft
P_r	Prandtl number dimensionless
$\dot{Q}$	heat flow, Btu/hr
R	radius, ft
Re	Reynolds number, dimensionless
S	entropy, Btu/°R
$\dot{S}$	entropy flow, Btu/hr°R
s	specific entropy, Btu/lb°R
$\dot{\mathcal{S}}$	entropy creation, Btu/hr°R
$\mathcal{s}$	entropy creation per unit mass, Btu/lb°R
T	temperature, °F or °R
$T_{C,B}$	bleed steam condensing temperature, °F
T_f	film temperature, °F
T_m	mean temperature, °F

T_o	datum state temperature
T_w	wall temperature, °F
t_w	tube wall thickness, ft
U	overall heat transfer coefficient, Btu/hr-ft^2°R
V	volume, ft^3
$\dot{V}$	volume change (with time), ft^3/hr
V	velocity, ft/sec
$\dot{W}$	shaft work flow, Btu/hr
x	material fraction, e.g. mole fraction or mass fraction
$\dot{Y}$	time derivative dY/dt where Y is an arbitrary property
z	temperature effectiveness, dimensionless

Greek Letters

ε	essergy, Btu
$\dot{\varepsilon}$	essergy flow, Btu/hr
η_{II}	effectiveness or second law efficiency, dimensionless
η_t	isentropic turbine efficiency, dimensionless
ϕ	maintenance cost, insurance, tax, interest factor, dimensionless
ρ	density, lb/ft^3
μ	viscosity, lb/hr-ft
χ	number of transfer units, dimensionless

Subscripts

A	air
b	boundary area
c	component
cw	cooling water
F	fuel
FB	furnace-boiler
FH	feedwater heater
FG	flue gas
FW	feedwater
FWe	feedwater exit
FWi	feedwater inlet
HP	high pressure
HPBFP	high pressure boiler feed pump
HPFB	high pressure section of furnace-boiler
HPT	high pressure turbine stage
IPT	intermediate pressure turbine stage
in	input
LPT	low pressure turbine stage
m	mean
0	datum state
opt	optimum
out	output
r,R	region or zone
RH	reheat

RHFB	reheat section of furnace-boiler
s	shaft work
ss	shellside
T	throttle
ts	tubeside

Superscripts

d	diffusional flow
f	hydrodynamic flow
fC	flow cell
fM	flow mechanical
fT	flow thermal
fTM	flow thermomechanical
m	material
q	heat transfer
t	transport
w	work

SUMMARY

This study demonstrates the power of essergy analysis for solving power plant operating and design problems. An effective method is developed for analyzing the economic value of flows of the commodity which the modern day power plant transforms and consumes (dissipates)--that commodity being essergy (essential energy via the second law) and not energy. Using this method, unit essergy costs are calculated for various points in an actual power plant operated by Wisconsin Electric Power Company. It is established that these unit essergy costs will remain constant regardless of any changes in the power cycle (i.e., the power cycle is linearized) and thus after being calculated once for design conditions can be used throughout the life of the power plant for making economic decisions. Analysis of this type have already led to significant savings in construction and operating costs at the Wisconsin Electric plant; see Fehring and Gaggioli (1978).

A practical example of the utility of analyzing power cycles in the above manner is demonstrated by using the unit essergy costs in an economic analysis of the repair or replacement of a feedwater heater which is operating in a deteriorated condition. This analysis includes determination

of the profitability of replacement of the feedwater heater and the maximum time that the heater can be left down for retubing before replacement becomes more economical.

Linearization of the power cycle also leads to decentralization so that optimum design of each zone in the power plant optimizes the design of the entire power cycle. It is in this spirit that the second part of this study concerns design of feedwater heaters. A simple essergy consumption model is developed for the feedwater heater in which total cost is made up of the sum of capital cost and essergy dissipation cost. Fundamental and well-known expressions which describe the momentum and heat transfer processes occuring within the feedwater heater along with a known capital cost relation are used to develop a total cost equation in terms of basic operating and design parameters. Minimization of the total cost equation with respect to feedwater velocity and heat transfer area using ordinary differential calculus results in optimum expressions for these two parameters. This analysis is equivalent to obtaining the optimum number and length of heat transfer tubes for the feedwater heater.

Using the unit essergy costs determined in the earlier essergy analysis, the optimum feedwater velocity and heat transfer area are calculated for a feedwater heater with the same operating conditions as feedwater heater number 6 from the power cycle under consideration in this study. The design equations are then generalized to some extent for application to the design of certain other feedwater heaters within the same or different power cycle.

The essergy analysis methods developed within this study are shown to be effective for solving power plant operating problems and design optimization. These methods prove more reliable than first law analysis and time-honored "rules-of-thumb." In summary, essergy analysis provides powerful and useful fundamental tools for the practicing power plant engineer.

CHAPTER I

INTRODUCTION

The process of preheating air and water to improve overall cycle efficiency is used extensively in power plants. Steam is bled from various turbine stages through exchangers such as regenerative feedwater heaters to preheat feedwater entering the boiler and air heaters to preheat combustion air entering the furnace. Exchangers are also placed in the furnace stack to reclaim heat normally rejected with the flue gases for additional preheating of combustion air and feedwater. Therefore, the power plant engineer is often faced with the problem of determining optimum operating and design parameters for preheating equipment and with operational and design decisions which involve these parameters.

Fehring and Gaggioli (1977) have demonstrated a second law analysis method for making economic decisions concerning the repair or replacement of a deteriorated regenerative feedwater heater within an actual power plant operated by Wisconsin Electric Power Company. While it is a significant contribution to the field, their method appears to be deficient in that the feedwater heating costs that they calculated are not linear with changes in operation of the power cycle. Linearity of the feedwater heating costs is a requirement in order for the feedwater heater economic analysis to be performed in the manner that they have done.

It is the intent of this study to develop a more general and accurate second law method suitable for analyzing the entire steam power cycle. The Fehring and Gaggioli method will serve as a starting point by way of review and extension of their previous work concerning the feedwater heating system. The second law method developed within this study will be used to analyze the same power cycle and the feedwater heating costs generated by this analysis used to make the same feedwater heater economic analysis. The results of this study will be compared with the results of the Fehring and Gaggioli study to assess the effect of the assumptions used within each and to determine their relative value for optimizing the operation of feedwater heating systems.

The second part of this study will concern optimum design of feedwater heaters. A cost equation for the feedwater heater which is based on capital cost and the Second Law will be developed. This cost equation will be related to basic feedwater heater parameters and then minimized with respect to two of these parameters to demonstrate the optimization process. Application of the optimum design equations to a feedwater heater with operating conditions taken from the actual Wisconsin Electric plant will serve to illustrate their practicality.

An exact but complex method for optimizing the operation and design of a power cycle requires treatment by LaGranges Method of Undetermined Multipliers in a manner similar to that illustrated by El-Sayed and Evans (1970) in a paper concerning the design of heat systems. It is hoped that this study will lead to a method that is simpler to understand and easier to use. Enough generality will be retained within the development to allow extension (following the examples presented in this paper) of the second law analysis method to optimize operation or design at any point within a steam power cycle.

As Fehring and Gaggioli point out, power plant operating decisions can be based on first law analysis, "rules of thumb" and measured heat rate (unit efficiency) tests. Design decisions are usually based on first law analysis or "rules of thumb." Use of heat rate tests is inherently inaccurate because of the difficulty in making an interpretation of the results. It is often difficult or impossible to attribute what portion of an increased heat rate is due to the equipment in question and what portion is due to other system components and variables. "Rules of thumb" are subject to the inaccuracies of applying generalized "rules" to specific situations.

First law analysis is sometimes arduous (depending on the size of the system being analyzed) and often fails to reveal the true nature of the physical process that is being studied. The second law analysis method developed within this study will be shown to be more powerful and more reliable than the above mentioned methods.

A. A Brief Discussion of First Law Analysis

An energy balance (first law analysis) around a component or group of components can be used by the plant engineer to help him assess the effect of changes in equipment or operating procedures and for specifying design parameters for new equipment. But the energy balance is deficient for evaluating physical processes and its use for designing power cycles can result in misleading conclusions.

It is well known that energy is never consumed in any physical process, as this would violate the First Law, but is merely transformed in its ability to do work. Since all physical processes are irreversible to some extent (i.e., the entropy of the system wherein the physical process is occuring increases), the work available from the energy outputs is always less than the work available from the energy inputs for a particular process. The First Law does not distinguish between ability to do work for different energy streams and thus, the energy balance is not an effective measure for evaluating physical processes.

Fehring and Gaggioli (1977) provide an excellent illustration of the difficiencies of first law analysis by considering the throttling process (i.e., the expansion of a fluid through a pressure drop). The throttling process occurs at constant enthalpy since no work is produced and heat transfer effects may be neglected. Therefore, the fluid has the same amount of energy at the exit of the process as it does at the entrance; i.e., the fluid has no loss of energy and first law (thermal) efficiency is unity. However, due to irreversibilities in the throttling process the fluid has lost ability to do work, but first law analysis gives no information concerning this important fact.

Kadaba (1977) cites yet another example of the deficiency of first law analysis by considering a heat exchanger that is operating adiabatically. In this case, as in the case of throttling, the heat exchange process occurs at constant enthalpy and first law efficiency is again equal to unity (i.e., indicating that the heat exchange process has been 100 percent efficient). Due to the irreversibilities associated with heat transfer across a finite temperature difference, the heat that left the higher temperature stream and entered the lower temperature stream is reduced in its ability to do work. As before, first law analysis gives no information concerning this important fact.

B. A Brief Discussion of Essergy Analysis (Second Law Analysis)

If a particular energy flow is to be used to obtain a change from equilibrium in the physical world, then the property of prime importance that is associated with that energy flow is its ability to do work. The commodity that so-called energy companies (electric companies, gas companies, etc.) actually sell is not energy per se but the ability to do work that is associated with the energy. It is obvious from the example of the throttling and heat transfer process described in the previous section that an energy balance (first law analysis) gives no information about changes in this important property.

Since Evans (1969,1977) and El-Sayed and Evans (1970) have shown that essergy (essential energy via the second law) is a direct quantitative measure of an energy flow's ability to do work for any chemical system, an essergy balance (second law analysis) around the throttling process or heat exchanger will yield the desired information. The essergy balance will immediately identify the fluid stream's loss in ability to do work by assessment of the amount of essergy that is consumed (dissipated)--i.e., by assessment of the irreversibility of the process.

An essergy balance around any physical process (including regenerative feedwater heating) will lead to an assessment of the amount of essergy consumed by the process which in turn will lead to an evaluation of the second law efficiency of the process in transferring or transforming essergy. If the unit costs of essergy flows to and from a process are determined, they can be used in conjunction with the essergy balance to make effective and accurate economic decisions concerning operation and design of the process equipment. In this manner, the power of essergy analysis (second law analysis) and its superiority over conventional methods for analyzing power cycles can be soundly demonstrated.

C. A Brief Discussion of Regenerative Feedwater Heating

Using bleed steam from the turbines to preheat feedwater (regenerative feedwater heating) before it returns to the boiler has been used for years as a means for improving overall power cycle efficiency. Intuitively, it might seem that removing steam from the turbines which otherwise might be used to produce work would be detrimental to the overall cycle efficiency. The reason why regenerative feedwater heating works to improve overall power cycle efficiency has as its basis the concept of efficient utilization of essergy by minimization of the essergy dissipated--i.e., by minimization of irreversibility.

Recall that as the temperature difference across which heat is being transferred increases, the irreversibility or essergy dissipation increases. Therefore, any process within power cycle that increases the temperature at which the working fluid receives heat from the heat source (the products of combustion) or decreases the flow of heat from the heat source which working fluid receives at the lowest temperatures will work to improve the overall cycle efficiency; after Keenan (1941). That is, any process that reduces the temperature difference across which heat must be transferred from the heat source to the working fluid will serve to reduce the irreversibility of the heat transfer and hence increase the overall power cycle efficiency.

Regenerative feedwater heating is a process which decreases the flow of heat from the heat source which the working fluid receives at the lowest temperatures. Since the feedwater is preheated by bleed steam, it enters the boiler at a higher temperature and consequently less heat is needed from the combustion products in order for the boiler to deliver steam to the turbines at design conditions thereby effectively reducing the flow of heat from the combustion products which the feedwater receives when it is at its lowest temperatures.

Another way of viewing regenerative feedwater heating is that the essergy flows within the power cycle are more appropriately matched to the requirements of the processes being performed. Whereas the creation of steam in the boiler requires a source with a high value of essergy, the preheating of feedwater may be accomplished with a source that has a much lower value of essergy such as bleed steam from the turbines. Therefore, if the feedwater is preheated with bleed steam, then it will not be necessary to use "high essergy" fuel (with a concurrent large irreversibility) to accomplish this task and consequently overall power cycle efficiency will be improved.

D. Literature Survey

The concept of maximum potential work for a system or process has been of interest since man first started dealing with power systems. As early as the work of von Helmholtz and Gibbs (1873), references to maximum potential work expressions (free energy and available energy functions) have been made. More recently, other writers such as Darrieus (1930) and Keenan (1932,1941,1951) formulated and discussed the concept of availability; a measure of the maximum potential work for systems and processes. Rant (1956) introduced yet another name for the measure of maximum potential work of processes in 1956--he called his measure exergy, but for all practical purposes it is the same as steady flow availability. Gaggioli (1962) made further contributions to the availability concept in the early sixties.

Evans (1968,1969) formulated and proved a completely general measure for the potential work for chemical systems. He called this measure essergy and showed that all of the earlier developed measures for potential work (such as free energy, availability, available energy, useful energy, exergy, etc.) are all special cases of this one unique measure. It is with this study in mind that essergy will be used as the measure of maximum potential work for processes in this study. Haywood (1974) has recently provided a critical review of essergy and all of its special cases.

Application of essergy or special cases of essergy for analysis of systems which deal with power have appeared as early as the work of Darrieus (1930) and Keenan (1932) concerning steam power cycles, but it is only recently that significant contributions to this field have been made.

Evans, et al. (1966) and El-Sayed and Aplenc (1970) have applied essergy analysis to a vapor compression sea-water desalination system. El-Sayed and Evans (1970) demonstrated a general development for the application of essergy analysis to the design of heat systems.

Following the work of Rant, workers in Europe and South Africa have applied exergy concepts to the evaluation of chemical processes. These workers include Boberg (1971), Fratzscher and Eckert (1974) and Louw (1975). Another foreign worker who utilizes availability concepts for analyzing industrial processes is Cozzi (1975).

Gaggioli, et al. (1975) and Fehring and Gaggioli (1977,1978) have applied available energy analysis to a steam power cycle. The work of Fehring and Gaggioli specifically involves the use of available energy analysis to make operational decisions concerning boiler feed pump drives and feedwater heaters for an actual power plant operated by Wisconsin Electric Power Company. Their study concerning feedwater heaters will provide the starting point for this study.

Recently, second law analysis has been extended into the field of fuel conversion with the work of Gaggioli and Petit (1975) and Jhawar, et al. (1977).

Second law analysis has also been utilized for evaluating energy systems with the work of Reistad, et al. (1970), Hamel and Brown (1972) and Lee and McCulloch (1972).

A useful application of second law analysis which is now being utilized is the evaluation of potential areas for energy conservation programs and for deciding national energy policy. Work in this field includes that of Berg (1975), Reistad (1975), Gyltopoulos, et al. (1975), Hall (1975), Hall, et al. (1975) and Rotty and Van Artsdalen (1977).

CHAPTER II

THEORETICAL DEVELOPMENT FOR ESSERGY AND ECONOMIC ANALYSIS

A. Essergy

The essergy ε of a system may be defined as the minimum work necessary to create the system from its environment or conversely as the maximum work attainable by allowing the system to come to complete equilibrium with its environment.[1] Essergy is a measure of departure from equilibrium and is the driving force for all physical processes.

Any quantity of matter, any fixed region of space (evan a vacuum) or any flux across a boundary can have essergy. The essergy associated with a quantity of matter, space or flux is a measure of its work equivalent (where work equivalent is by definition the maximum amount of work that

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[1]The environment is defined here as a surroundings of such an extent that its intensive properties (i.e., T_O, P_O, etc.) remain unchanged after an interaction with the system. A more general definition of essergy ε_i is that it is the information about proposition "i" with respect to some reference level "0":

$$\varepsilon_i \equiv \log P_{io}^{-1} - \log P_i^{-1}$$

where $\log P_i^{-1} \equiv$ information content of proposition "i" and P_i represents the probability of proposition "i"; after Evans (1977). A more concise discussion of essergy is presented in a paper by El-Sayed and Evans (1970) concerning the design of heat systems.

can be obtained by allowing the matter, space or flux to come to complete equilibrium with its environment); after Evans, et al. (1966) and Evans (1969). For example, the essergy of a pound of fuel entering a boiler is the maximum work that can be produced by bringing the fuel to complete, stable, chemical equilibrium with its environment for the simple case where the fuel is at the same temperature T_o as its environment.[2] This type essergy is known as chemical essergy and is equal simply to the Gibbs-free-energy of the fuel. It should be obvious that essergy is a property of the system and its relation to the surroundings.

Evans, et al. (1966) illustrates this fact by considering that the essergy of an evacuated vessel transported from outer space to the earth will be the maximum work obtainable by allowing the vacuum in the vessel to come to pressure and temperature equilibrium with the atmosphere.[3] Conversely, the essergy of an air filled vessel transported to outer space will be the maximum work obtainable by allowing the air in the vessel to come to pressure and temperature equilibrium with outer space.

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[2]Complete, stable, chemical equilibrium occurs when all chemical species making up the fuel (i.e., carbon, hydrogen, sulfur, etc.) are in their most stable chemical configuration found in the environment (i.e., H_2O, CO_2, $CaSO_4 \cdot 2H_2O$, etc.). That is, all species are at the Gibbs chemical potential of the environment.

[3]Note that the first law would be totally useless in evaluating this situation since a vacuum has no matter and therefore <u>no</u> <u>energy</u>.

The essergy function ε of a system can be expressed by the following:

$$\varepsilon = E+P_oV-T_oS - \sum_c \mu_{co}N_c \qquad (1)$$

Proof of equation (1) will not be discussed here as it has been demonstrated rigorously by Evans (1969).

In order to handle essergy flows one may introduce the notation $\dot{Y} = dY/dt$ where Y is an arbitrary property and t denotes time; after Evans (1969). Rewriting equation (1) in this notation will give,

$$\dot{\varepsilon} = \dot{E}+P_o\dot{V}-T_o\dot{S} - \sum_c \mu_{co}\dot{N}_c \qquad (2)$$

Equation (2) holds for any open chemical system in any given environment.

Many special cases of equation (1) have been developed within the framework of classical thermodynamics by other workers and the resulting functions called by such names as available energy, availability, useful energy, free energy, exergy, etc. (see Appendix K). It is noteworthy that essergy ε is an extensive property of a system for any given datum level (environment) and will never be negative.

B. Essergy Balances

A balance of essergy around any system and process is represented by the following equation; after Evans, et al. (1966):[4]

$$\dot{\varepsilon} = \dot{\varepsilon}^q + \dot{\varepsilon}^w + \dot{\varepsilon}^f + \dot{\varepsilon}^d - T_o\dot{S} \qquad (3)$$

$\dot{\varepsilon}$ ≡ the rate of essergy storage in the system

$\dot{\varepsilon}^q$ ≡ thermal essergy, net rate that essergy enters or leaves the system with heat transfer

$\dot{\varepsilon}^w$ ≡ work essergy, net rate that essergy enters or leaves the system as mechanical work

$\dot{\varepsilon}^f$ ≡ flow essergy, net rate that essergy enters or leaves the system with material streams (i.e., hydrodynamic flow)

$\dot{\varepsilon}^d$ ≡ diffusional essergy, net rate that essergy enters or leaves the system with mass transfer (i.e., diffusion)

$\dot{S}$ ≡ rate that entropy is created within the system

T_o ≡ environment temperature

The quantity $T_o\dot{S}$ represents the dissipation or consumption of essergy within the system--i.e., the rate at which essergy disappears from the system plus environment. It is useful to

[4]External effects on the energy of the system such as electricity, magnetism, gravity, stress and capillarity have been neglected here for simplicity. In addition, equation (2) assumes that the environment has a uniform pressure P_o throughout. Equation (2) may be extended to include all contributions to the energy of the system, as well as the case of non-uniform environment pressure using methods illustrated by Evans (1966).

note that in view of the Second Law of Thermodynamics, $T_o\dot{S}$ is always greater than or equal to zero. For r regions and b boundary areas (ports) we may derive the general essergy balance by extending equation (2) to give the following expression,

$$\sum_r \dot{\varepsilon}_r = \sum_b (\dot{\varepsilon}_b^q + \dot{\varepsilon}_b^w + \dot{\varepsilon}_b^f + \dot{\varepsilon}_b^d) - \sum_r T_o\dot{S}_r \tag{4}$$

where $\dot{\varepsilon}_r$ is the essergy stored in region r, $T_o\dot{S}_r$ is the essergy dissipated in region r (i.e., a measure of the irreversibility of the process occurring within zone r) and $\dot{\varepsilon}_b^q, \dot{\varepsilon}_b^w, \dot{\varepsilon}_b^f$ and $\dot{\varepsilon}_b^d$ are all the ways in which essergy may enter the system at boundary region b.

The flow essergy term may be divided into three separate and distinguishable essergy forms (see Appendices B and C). The first form is flow cell essergy $\dot{\varepsilon}_b^{fC}$ and represents the maximum work that can be obtained from a change in composition from $x_{c,b}$ (material fraction of component c in the flowing stream at the conditions of boundary region b) to $x_{c,o}$ (material fraction of component c in the flowing stream at environment conditions T_o and P_o). This type essergy obtains its name due to the fact that a cell of some kind may usually be used to harness power from changes in composition at fixed temperature and pressure. The second form is flow mechanical essergy $\dot{\varepsilon}_b^{fM}$ and represents the mechanical work that would be produced by a material stream flowing reversibly from conditions T and P at boundary area b to environment conditions T_o and P_o at fixed composition. The third form is flow thermal essergy $\dot{\varepsilon}_b^{fT}$ and represents the essergy associated with the thermal energy flowing with the material stream at boundary area b.

In view of equation (C-15) from Appendix C, it is clear that essergy can enter or leave a system with hydrodynamic material flow at boundary area b in three different forms.

$$\dot{\varepsilon}_b^{f} = \dot{\varepsilon}_b^{fM} + \dot{\varepsilon}_b^{fT} + \dot{\varepsilon}_b^{fC} \tag{C-15}$$

Substitution of equation (C-15) into equation (4) yields the following:

$$\sum_r \dot{\varepsilon}_r = \sum_b (\dot{\varepsilon}_b^{q}+\dot{\varepsilon}_b^{w}+\dot{\varepsilon}_b^{fM}+\dot{\varepsilon}_b^{fT}+\dot{\varepsilon}_b^{fC}+\dot{\varepsilon}_b^{d}) - \sum_r T_o\dot{S}_r \tag{5}$$

One may now observe that essergy can enter or leave a system at boundary area b in six distinguishable ways--viz., heat transfer, work transport, diffusion, and in three different forms with hydrodynamic material flow. For convenience, we may define an essergy transport $\dot{\varepsilon}_b^{t}$ at boundary area b by,

$$\dot{\varepsilon}_b^{t} = \dot{\varepsilon}_b^{q}+\dot{\varepsilon}_b^{w}+\dot{\varepsilon}_b^{fM}+\dot{\varepsilon}_b^{fT}+\dot{\varepsilon}_b^{fC}+\dot{\varepsilon}_b^{d} \tag{6}$$

Substituting equation (6) into equation (5) and rearranging yields the following simplified form of the general essergy balance:

$$\sum_b \dot{\varepsilon}_b^t = \sum_r (\dot{\varepsilon}_r + T_o \dot{S}_r) \tag{7}$$

Therefore, one can see that the sum of all essergy transports to and from a system is equal to the essergy stored in the system plus the essergy dissipated in the system.

If the system is at steady state so that $\sum_r \dot{\varepsilon}_r = 0$, the general essergy balance becomes,

$$\sum_b \dot{\varepsilon}_b^t = \sum_r T_o \dot{S}_r \tag{8}$$

For this case the sum of all essergy transports to and from the system equals the essergy dissipated in the system.

In view of the fact that equation (6) represents all of the ways that essergy can enter a system at boundary area b, we see that equation (6) and (7) taken together constitute a completely general balance of essergy for any system while equations (6) and (8) taken together represent the same for any system at steady state.

For the purpose of analyzing power cycles, equations (7) and (8) can be simplified to be more readily applicable to the various components included within a power cycle. Since all forms of essergy are equivalent in a thermodynamic sense we may identify essergy transports at all boundary areas only as inputs or outputs. In view of this simplification, one may rewrite equations (7) and (8) for a zone R of N essergy inputs and M essergy outputs to yield the following equations:

$$\sum_{i}^{N} \dot{\varepsilon}_{in,i} + \sum_{j}^{M} \dot{\varepsilon}_{out,j} = \dot{\varepsilon}_R + T_o \dot{S}_R \tag{9}$$

$$\sum_{i}^{N} \dot{\varepsilon}_{in,i} + \sum_{j}^{M} \dot{\varepsilon}_{out,j} = T_o \dot{S}_R \tag{10}$$

Equations (9) and (10) are simplified forms of equations (7) and (8), respectively, and thus convey precisely the same information. For a transient system the sum of all essergy inputs and outputs equals the essergy stored in the system plus the essergy dissipated in the system. For a steady state system the sum of all essergy inputs and outputs equals just the essergy dissipated in the system.

C. Effectiveness

In order to determine "effectiveness" or "second law efficiency" of a zone, the zone must be viewed as a "simple essergy consumption system." For "simple essergy consumption systems," essergy inputs and outputs due to work and heat transfer effects have the same meaning as in the general essergy balance, but the essergy inputs and outputs due to hydrodynamic and diffusional flows must be viewed in a different manner. An essergy contribution due to a hydrodynamic or diffusional flow is calculated as a net difference between the amount of essergy at the flow entrance and exit. Furthermore, the contribution is classified as an input or output according to whether the specific purpose of the flow is to deliver essergy to or receive essergy from a zone. For example, the difference between the essergy entering and leaving a turbine with steam flow constitutes a net essergy input contribution due to hydrodynamic flow. On the other hand, the difference between the essergy entering and leaving a feedwater heater with feedwater flow constitutes a net essergy output contribution due to hydrodynamic flow. Essergy contributions due to diffusional flows are handled similarly.

In determining effectiveness we also require that any essergy output that is exhausted and allowed to dissipate for no useful purpose (i.e., essergy that is "thrown away") must be assigned a value of zero. Flue gas from a furnace exhausted to the atmosphere or cooling water exhausted to a river from a power plant are examples of this type of essergy output. One is, in effect, causing the actual essergy lost with the output stream to be counted in the same manner as the essergy dissipated in the zone. This viewpoint represents sound rationale since the output stream is normally a physical or economic necessity in order for a zone to perform a useful purpose. One is, therefore, assured that the effectiveness of a zone reflects not only that the actual irreversibilities in the zone but also the essergy that is by necessity "thrown away" in order for the zone to perform a useful purpose.[5]

The effectiveness of a zone R is defined by Kadaba (1977) as,

$$\eta_{II,R} = \frac{\Sigma\ \dot{\varepsilon}_{out}}{\Sigma\ \dot{\varepsilon}_{in}} \tag{11}$$

where $\dot{\varepsilon}_{in}$ and $\dot{\varepsilon}_{out}$ can represent essergy inputs and outputs (except those dissipated without useful purpose) due to heat, work, and net hydrodynamic and diffusional flow contributions. As noted earlier, effectiveness is a measure of the essergy dissipated or "thrown away" by the zone. The closer the effectiveness is to unity the less the essergy being dissipated or "thrown away" in the zone or the more effectively essergy is being transferred or transformed in the zone.

The effectiveness of components which utilize energy is an important and useful measure. It can be used to assess the deviation from the ideal for different components and therefore may indicate differences or changes in performance, deterioration with time and areas for improved design. The heat exchanger mentioned earlier is a good example of where the effectiveness concept might be well utilized to compare

[5]See Appendix M.

different designs, different modes of operation and assess the effect of maintenance problems such as fouling. In fact, as Kadaba (1977) points out, temperature effectiveness has been defined within the framework of heat exchanger technology to provide a measure similar to the "thermodynamic effectiveness" defined above.

D. Internal Economy

For the purpose of formulating an internal economy (internal cost balances) for a power cycle, we will take the view that any region or zone within the power cycle has three distinct modes of cash flow associated with it. These cash flows include the cost of creating and maintaining the zone, the cost of essergy and material "bought" by the zone and the value of essergy and material "sold" by the zone. The cost of creating and maintaining the zone represents the amortized capital cost (including interest, taxes, insurance, maintenance, etc.) of the equipment in the zone. The cost of essergy and material "bought" by the zone represents the continuous operating expense for the zone while the value of essergy and material "sold" by the zone represents the gross income for the zone. For example, if the boiler is viewed as a zone, the fuel, feedwater and preheated combustion air "bought" would represent the continuous operating expense while the steam "sold" would represent the gross income.

For a given region or zone R, the steady state economic balance is defined by Evans, et al. (1966):[6]

[6]After the "brac-ket" notation introduced by Dirac.

$$\sum_b [c_{t,b}](\dot{\varepsilon}_b^t) + \dot{C}_R + \sum_b [c_{m,b}](\dot{M}_b^m) = 0 \tag{12}$$

where:

$\dot{\varepsilon}_b^t$ = essergy transport t at boundary area b of zone R

$c_{t,b}$ = unit cost of essergy transport t at boundary area b of zone R (e.g., $c_{q,b}$)

$\dot{C}_R$ = amortized cost of capital equipment (including interest, taxes, insurance, maintenance, etc.) in zone R

$\dot{M}_b^m$ = material transport m at boundary area b of zone R

$c_{m,b}$ = unit cost for material transport m at boundary area b of zone R

In equation (12), $[c_{t,b}]$ is a row vector of unit essergy costs,

$$[c_{t,b}] = c_{q,b} c_{w,b} c_{fM,b} c_{fT,b} c_{fC,b} c_{d,b} \tag{13}$$

and $(\dot{\varepsilon}_b^t)$ is a column vector of essergy fluxes,

$$(\dot{\varepsilon}_b^t) = \begin{pmatrix} \dot{\varepsilon}_b^q \\ \dot{\varepsilon}_b^w \\ \dot{\varepsilon}_b^{fM} \\ \dot{\varepsilon}_b^{fT} \\ \dot{\varepsilon}_b^{fC} \\ \dot{\varepsilon}_b^d \end{pmatrix} \tag{14}$$

Similarly, $[c_{m,b}]$ is a row vector of unit material costs and $(\dot{M}_b^m)$ is a column vector of material fluxes.

A given zone is not allowed to operate at a profit (i.e., it must operate at such a rate as to "break even"). Therefore, the sum of amortized capital cost, continuous operating expenses and net income must equal zero as is reflected by equation (12). This restriction guarantees that the unit cost of a material or essergy stream flowing to or from a zone has the same value irrespective of whether you are viewing it from inside or outside the boundary of the zone. That is, one is assured that each zone cannot sell any material or essergy stream at a rate higher than is necessary to operate at a "break even" point and therefore each zone is protected from profit taking at its boundaries by adjacent zones. In view of this requirement, if equation (12) is written for all zones of an entire plant and all of these equations added together, all internal transactions will cancel and the sum of all zonal equations will represent the same equation that would have occured had equation (12) been written considering the entire plant to be a single zone.

Each zone of a plant is included in the design for a technical purpose. One or more of the material or essergy streams "sold" by each zone represents the technical purpose or principal output of the zone. For example, the technical purpose of a turbine is its work output. On the other hand, the technical purpose of a distillation column is its product streams. In some cases, the technical purpose is represented by a combined material and essergy stream (e.g., chemical process compressor).

The subdivision of a plant into zones may not be done in an arbitrary manner, but must satisfy the following requirements. The costs of all essergy or material streams except the principal output or technical purpose of a zone are determined by the state of affairs in the adjacent zones that they are "bought" from or "sold" to. The net difference between these costs represents the continuous operating expense for the zone. The cash flow for the principal output of the zone is adjusted to pay for the amortized capital cost of the equipment in the zone and the continuous operating expense for the zone so as to satisfy the zone's economic balance given by equation (12). Therefore, each subdivision of a plant must have a principal product or technical purpose whose associated cash flow is used to amortize the capital equipment and pay for the operating costs.

As when calculating effectiveness for a zone, an essergy output which is thrown away (exhausted and dissipated for no useful purpose) must be viewed differently. This type essergy output is given zero economic value (i.e., it cannot be considered as income for the zone). This viewpoint assures that the value of the essergy thrown away is charged against the principal product of the zone. For example, setting the value of the flue gas stream from the boiler equal to zero assures that the cost of steam (the principal product) from the boiler will reflect the value of the essergy that is by necessity being thrown away with the flue gas.[7]

While all forms of essergy are equivalent in a thermodynamic sense, they are not equivalent in an economic sense. For example, mechanical essergy flowing with a mass stream may be more valuable than the thermal essergy flowing with the same mass stream. The relative value of any two essergy flows depends strictly upon the technical purpose of the two zones exchanging the essergy flows. For simplicity in this analysis, all forms of essergy will be considered to be economically equivalent. This simplification can lead to problems, as will later be seen, but a method for resolving them will be developed.

If all forms of essergy are viewed as economically

[7] See Appendix M.

equal then they may be identified only as inputs or outputs and equation (12) becomes (for N essergy inputs and M essergy outputs),

$$\sum_{j}^{N} c_{in,j}\dot{\varepsilon}_{in,j} + \sum_{k}^{M} c_{out,k}\dot{\varepsilon}_{out,k} + \dot{C}_R + \sum_{b} [c_{m,b}](\dot{M}_b^m) = 0 \quad (15)$$

One may simplify equation (15) by realizing that for any zone in a steam power plant, the terms representing the materials "bought" and "sold" will in most cases cancel. One is, in effect, assuming that no steam or water flow into a zone is used up or lost in that zone (at steady state), but simply exits as a steam or water flow to some adjacent zone. Therefore, the steam and water flows in a steam power plant have economic value only for the essergy they are carrying.[8]

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[8]This is true for the case where the environment is the earth and its atmosphere but for other types of environments we may have an entirely different situation. For example, consider the hypothetical case of a power plant condenser with outer space (vacuum) as its environment versus one with earth atmosphere as its environment. The technical purpose of the condenser which has the earth's atmosphere as its environment is to supply vacuum (mechanical energy) to the turbines. Therefore, the mechanical essergy flowing from the condenser has high value while material streams (steam or water) have little or no value. On the other hand, for the case of the condenser in outer space the technical purpose would be to collect the steam from the turbines and keep it from escaping (the condenser is not supplying a vacuum to the turbines since vacuum is supplied by the outer space and does not represent a departure from equilibrium). Because of the high procurement cost of make-up water in outer space, the material streams (steam and water) entering or leaving the condenser have high value relative to the essergy flowing with them.

The above viewpoint is not strictly true for zones where water or steam is lost (e.g., feedwater makeup). These lost and gained streams certainly have value due to their procurement and treatment costs, but the size of these streams is small compared to most flows in the power plant and thus their "material value" (and essergy value if they have any) can be neglected. In view of the above, the fourth term $\sum_b [c_{m,b}](\dot{M}_b^m)$ in equation (15) is set equal to zero to yield,

$$\sum_j^n c_{in,j}\dot{\varepsilon}_{in,j} + \sum_k^m c_{out,k}\dot{\varepsilon}_{out,k} + \dot{C}_R = 0 \qquad (16)$$

Now if the cost rate of any particular essergy flow (e.g., $c_{in,1}\dot{\varepsilon}_{in,1}$) is desired it may be had by simply isolating it on one side of the equal sign in equation (16). The unit cost of any essergy flow (e.g. $c_{in,1}$) may be determined by dividing through by the amount of the essergy flowing (i.e., $\dot{\varepsilon}_{in,1}$).

CHAPTER III

ESSERGY AND ECONOMIC ANALYSIS OF A FEEDWATER HEATING SYSTEM

A. Description of the Power Plant

This study considers an actual power plant operated by Wisconsin Electric Power Company. This same power cycle has been studied previously by Gaggioli, et al. (1975) and Fehring and Gaggioli (1977) and is depicted in Figure A-1 in Appendix A. The cycle is rated at slightly under 308,000 kilowatts and is fairly typical with eight turbine stages and seven points of extraction to feedwater heating. Tables A-1, A-2 and A-3 in Appendix A present steam or water properties and flow rates at various points in the power cycle; Case A represents the properties at design conditions, while Cases B and C represent properties at two different stages of deterioration of feedwater heater number 5 which will be described presently. The deteriorated conditions of Cases B and C will be used to assess the usefulness of our analysis in making operational decisions at a later point in this study. Figure A-1 is a schematic presentation of the power cycle depicting reference points. Details of the calculations necessary to obtain the power cycle data which could not be taken directly from the above mentioned studies are also presented in Appendix A.

A primary maintenance problem with feedwater heaters as they age is the occurence of tube leaks. In order to avoid the possibility of water backing up into the turbine and causing damage, the leaking tubes are plugged. Early in the life of a heater, the number of plugged tubes is small and performance of the heater is unaffected. Eventually, however, the number of plugged tubes increases to a point where the heater performance is significantly affected and consequently the overall cycle efficiency decreases. At some point in time, the deterioration in heater performance becomes so great that it must be either retubed or replaced. Either solution to this problem requires that the heater be taken out of service, but retubing usually requires more downtime than replacement.

For Case B it is assumed that approximately twenty percent of the tubes in feedwater heater number 5 are plugged and the terminal temperature difference (the difference between temperature of the bleed steam at saturation and the feedwater outlet temperature) has decreased by approximately five degrees. Feedwater heater number 6 is assumed to be in good enough condition to pick up the load that heater number 5 fails to carry by drawing more bleed steam (additional essergy flow) and the overall cycle efficiency remains unchanged (see Tables A-2 and A-3 in Appendix A).

For Case C it is assumed that the performance of heater number 5 has deteriorated to such an extent that it must be taken out of service to be retubed or replaced. The loss of heater number 5 causes the temperature of the feedwater entering the boiler economizer to be lower. This upset to the feedwater heating system is rectified by increased fuel flow to the boiler and increased bleed steam flows to the other heaters (see Tables A-2 and A-3 in Appendix A).

In their analysis of this power cycle Fehring and Gaggioli assume that all components in the power cycle except for feedwater heaters 4 through 7 in cases B and C operate at design conditions. In reality, changes in the operation of any one component affects the operation of all of the other components in the power cycle. Therefore, the deterioration of feedwater heater number 5 causes changes not only in feedwater heaters 4, 6, and 7 but also in every other component in the power cycle. In addition, the various components of the power cycle all deteriorate and at different rates which also contributes to deviations from design conditions as the power plant ages.

If the unit costs of the essergy flows in the power cycle (operating at design conditions) are found that are constant over the life of the plant (i.e., independent of the amounts of essergy flowing), then the above assumptions will not affect any operational decisions which utilize these unit essergy costs since these costs determined for design conditions will be valid at any point in time and stage of deterioration of the power plant. Constant unit costs for all essergy flows in the power cycle will occur if the power cycle can be analyzed as a linear essergy utilization system.

B. Calculation Methods for Essergy Flows

For simplicity in calculating the essergy flows from zone to zone in this analysis, the power plant will be subdivided such that all zones may be assumed to be operating adiabatically and free of diffusion at their boundaries (i.e., heat transfer and diffusion at the boundaries may be neglected, $\dot{\varepsilon}^q = 0$, $\dot{\varepsilon}^d = 0$).

The cell essergy form of the hydrodynamic flow essergy associated with the steam and water flows in the plant is small in all cases and will be neglected. The other two forms of hydrodynamic flow essergy associated with the steam and water flows will be grouped together into the one form known as flow thermomechanical essergy, $\dot{\varepsilon}^{fTM}$. The hydrodynamic flow essergy associated with the fuel flow to the plant has no thermomechanical essergy since it is at environment conditions of T_o and P_o, but consists totally of flow cell essergy $\dot{\varepsilon}^{fC}$. In view of these assumptions, the only type essergy flows of interest at any boundary are work essergy, flow cell essergy and flow thermomechanical essergy and they may be calculated by the following three equations:[9]

$$\dot{\varepsilon}^W = -\dot{W}_b \tag{17}$$

[9]See Appendices C and D.

$$\dot{\varepsilon}^{fC} = \dot{m}[\sum_{c} x_{c,b}(\mu_{c,b}-\mu_{c,o})] \quad (18)$$

$$\dot{\varepsilon}^{fTM} = \dot{\varepsilon}^{fM} + \dot{\varepsilon}^{fT} = \dot{m}[(h_b-h_o)-T_o(s_b-s_o)] \quad (19)$$

In order to calculate all essergy flows to or from a zone in the power cycle it is necessary only that one know the value of work inputs or outputs and the thermodynamic properties and mass flow rates of all material streams at the conditions of the boundary that each stream is crossing plus the thermodynamic properties of the datum state (environment) for the power cycle.[10] After all essergy flows for a zone have been calculated, they may be used in determining the effectiveness, essergy balance and economic balance for the zone.

C. Calculation Methods for Essergy Balances, Effectiveness and Economic Balances

Figures 1 through 7 will serve to illustrate the methods by which equations (10), (11) and (16) may be used to calculate the essergy balance, effectiveness and economic

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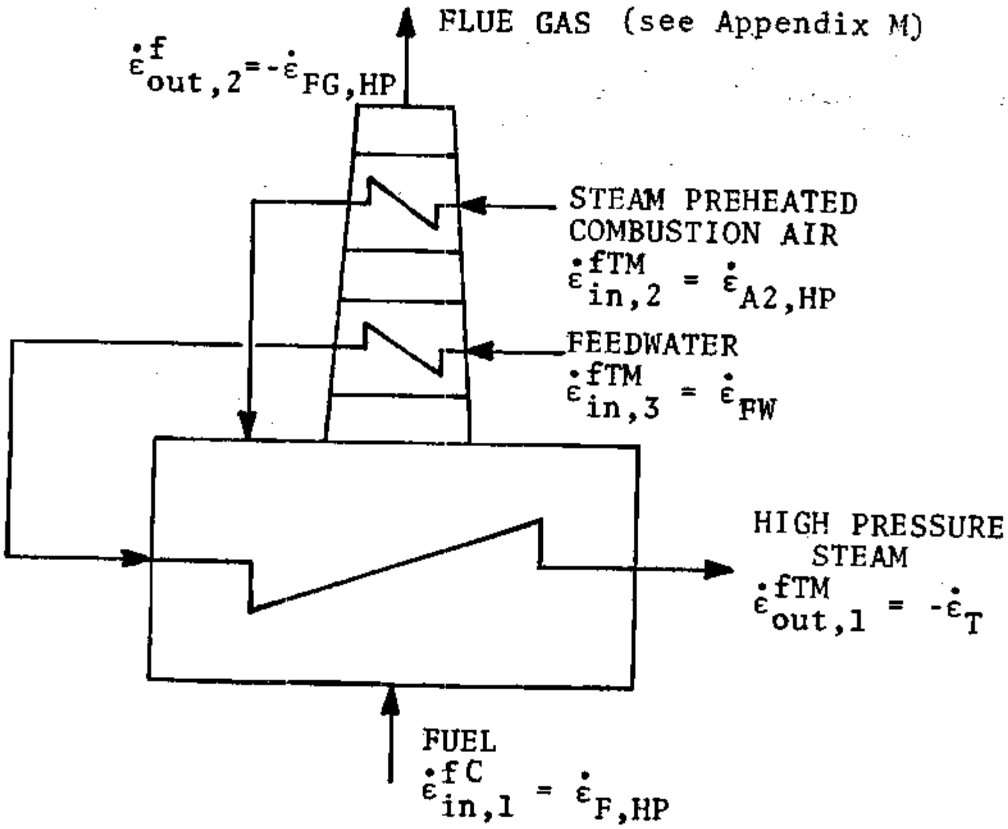
[10]Note that the term in brackets in equation (19) is the specific flow thermomechanical essergy at the boundary region $e = \dot{\varepsilon}/\dot{m} = [h-h_o)-T_o(s-s_o)]$. The work of expansion (turbine work output) or compression (pump work input) in equation (17) is calculated by the First Law; i.e., $W = \dot{m}(h_{in}-h_{out})$. The quantity in the brackets in the expression for flow cell essergy is equal simply to the Gibbs-free-energy difference between the material flow and the datum state.



Effectiveness:

$$\eta_{II,HPFB} = \frac{\dot{\varepsilon}_T - \dot{\varepsilon}_{FW}}{\dot{\varepsilon}_{F,HP} + \dot{\varepsilon}_{A2,HP}}$$

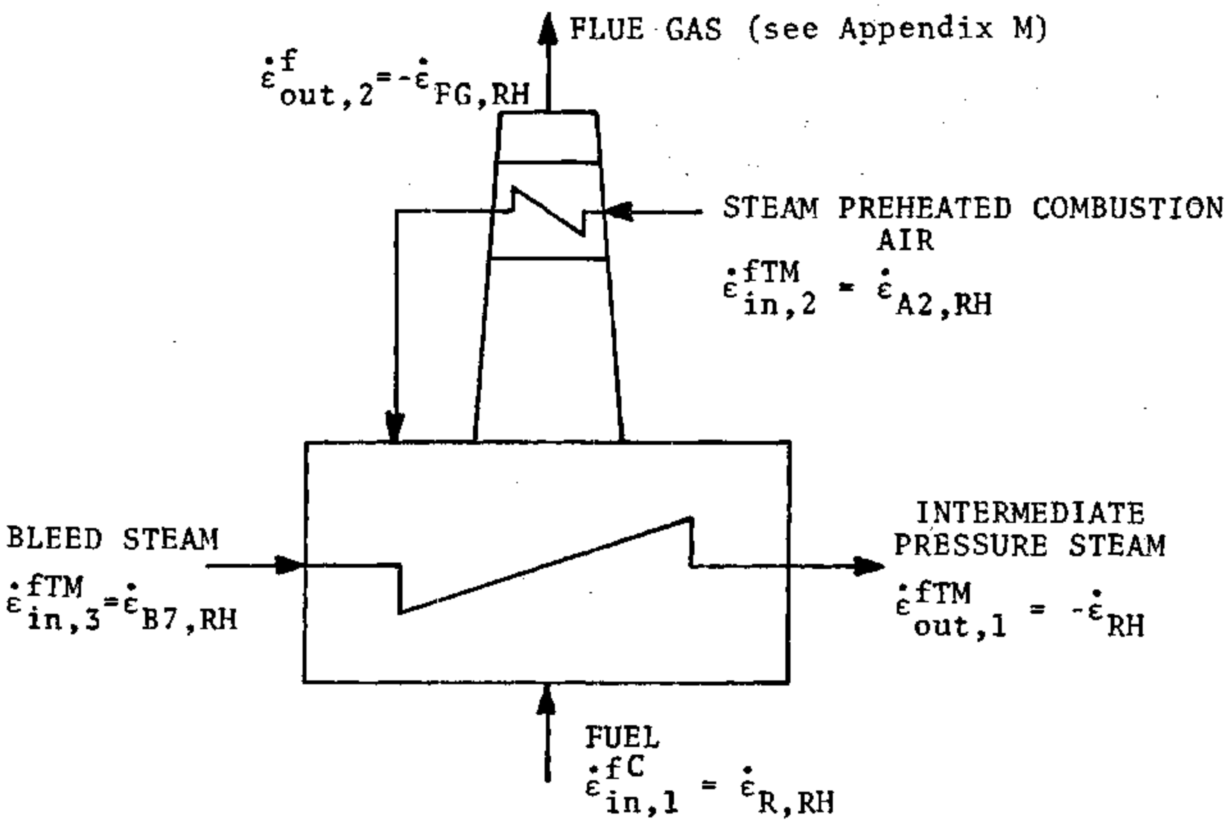
Essergy Balance:

$$\dot{\varepsilon}_{F,HP} + \dot{\varepsilon}_{A2,HP} + \dot{\varepsilon}_{FW} - \dot{\varepsilon}_T - \dot{\varepsilon}_{FG,HP} = T_o\dot{S}_{HPFB}$$

Economic Balance:

$$c_{e,F}\dot{\varepsilon}_{F,HP} + c_{e,A2}\dot{\varepsilon}_{A2,HP} + c_{e,FW}\dot{\varepsilon}_{FW} - c_{e,T}\dot{\varepsilon}_T + \dot{C}_{HPFB} = 0$$

Figure 1. Schematic of the Essergy Flows to the High Pressure Section of the Boiler



Effectiveness:

$$\eta_{II,RHFB} = \frac{\dot\varepsilon_{RH} - \dot\varepsilon_{B7,RH}}{\dot\varepsilon_{F,RH} + \dot\varepsilon_{A2,RH}}$$

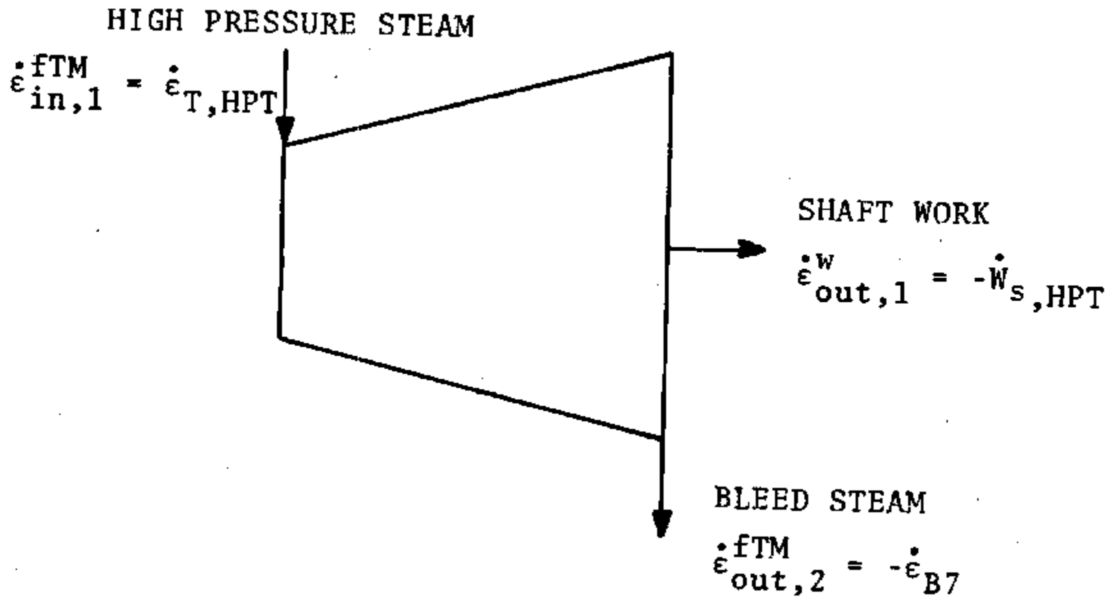
Essergy Balance:

$$\dot\varepsilon_{F,RH} + \dot\varepsilon_{A2,RH} + \dot\varepsilon_{B7,RH} - \dot\varepsilon_{RH} - \dot\varepsilon_{FG,RH} = T_o\dot S_{RHFB}$$

Economic Balance:

$$c_{e,F}\dot\varepsilon_{F,RH} + c_{e,A2}\dot\varepsilon_{A2,RH} + c_{e,B7}\dot\varepsilon_{B7,RH} - c_{e,RH}\dot\varepsilon_{RH} + \dot C_{RHFB} = 0$$

Figure 2. Schematic of the Essergy Flows to the Reheat Section of the Boiler



Effectiveness:

$$\eta_{II,HPT} = \frac{\dot W_{s,HPT}}{\dot\varepsilon_{T,HPT} - \dot\varepsilon_{B7}}$$

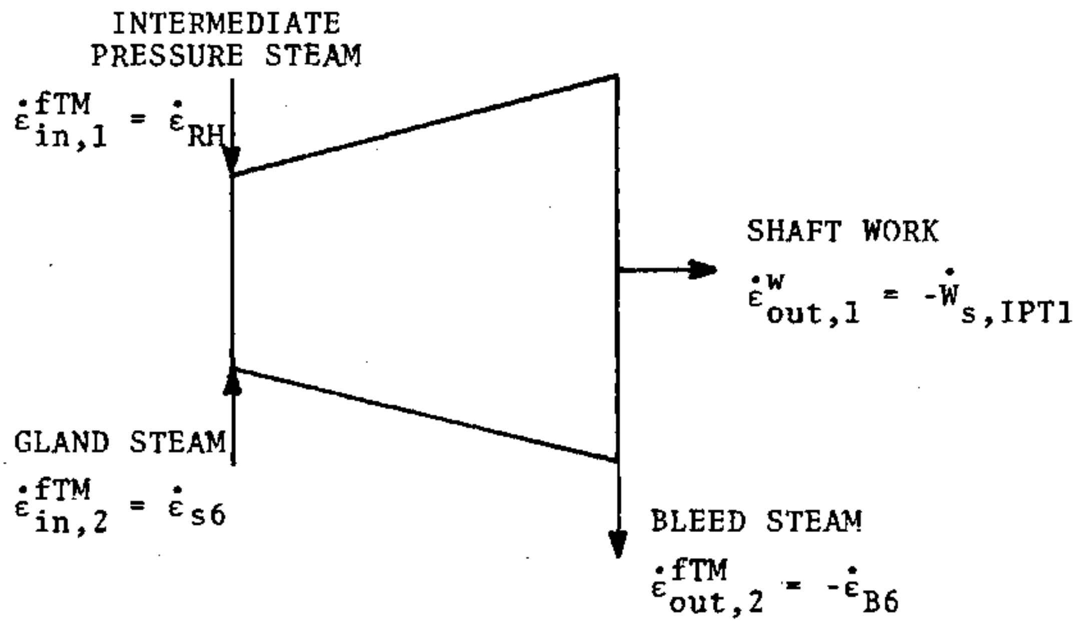
Essergy Balance:

$$\dot\varepsilon_{T,HPT} - \dot\varepsilon_{B7} - \dot W_{s,HPT} = T_o \dot S_{HPT}$$

Economic Balance:

$$c_{e,T}\dot\varepsilon_{T,HPT} - c_{e,B7}\dot\varepsilon_{B7} - c_{e,s}\dot W_{s,HPT} + \dot C_{HPT} = 0$$

Figure 3. Schematic of the Essergy Flows to the High Pressure Turbine



Effectiveness:

$$\eta_{II,IPT1} = \frac{\dot{W}_{s,IPT1}}{\dot{\varepsilon}_{RH} + \dot{\varepsilon}_{s6} - \dot{\varepsilon}_{B6}}$$

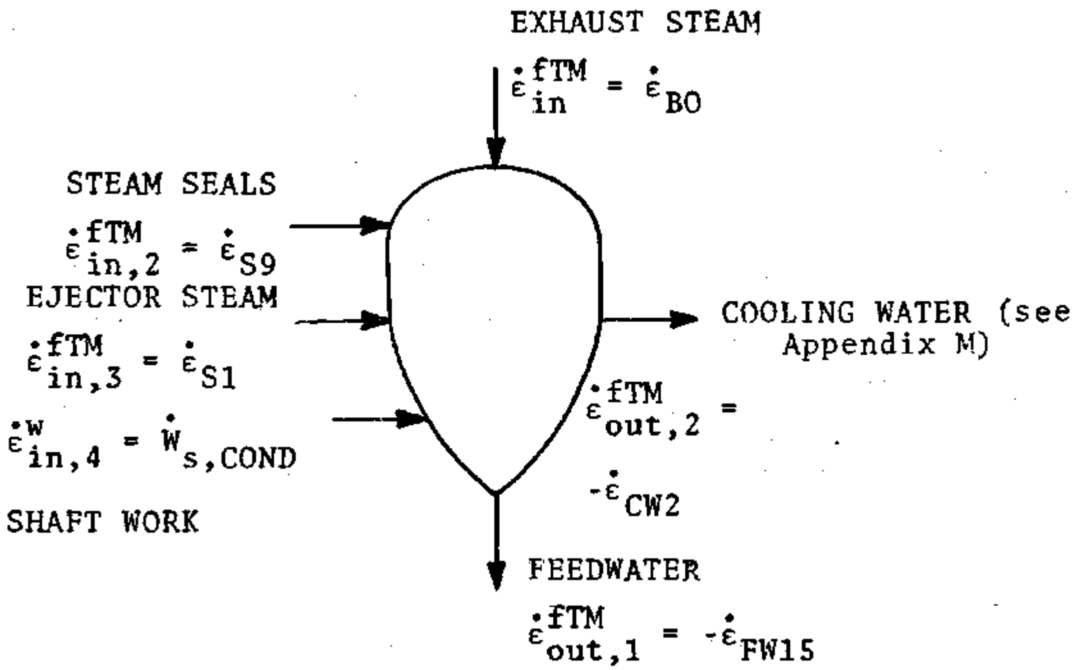
Essergy Balance:

$$\dot{\varepsilon}_{RH} + \dot{\varepsilon}_{S6} - \dot{\varepsilon}_{B6} - \dot{W}_{s,IPT1} = T_o\dot{S}_{IPT1}$$

Economic Balance:

$$c_{e,RH}\dot{\varepsilon}_{RH} + c_{e,s6}\dot{\varepsilon}_{s6} - c_{e,B6}\dot{\varepsilon}_{B6} - c_{e,s}\dot{W}_{s,IPT1} + \dot{C}_{IPT1} = 0$$

Figure 4. Schematic of the Essergy Flows to the First Stage of the Intermediate Pressure Turbine



Effectiveness:

$$\eta_{II,COND} = \frac{\dot{\varepsilon}_{FW15}}{\dot{\varepsilon}_{BO} + \dot{\varepsilon}_{s9} + \dot{\varepsilon}_{s1} + \dot{W}_{s,COND}}$$

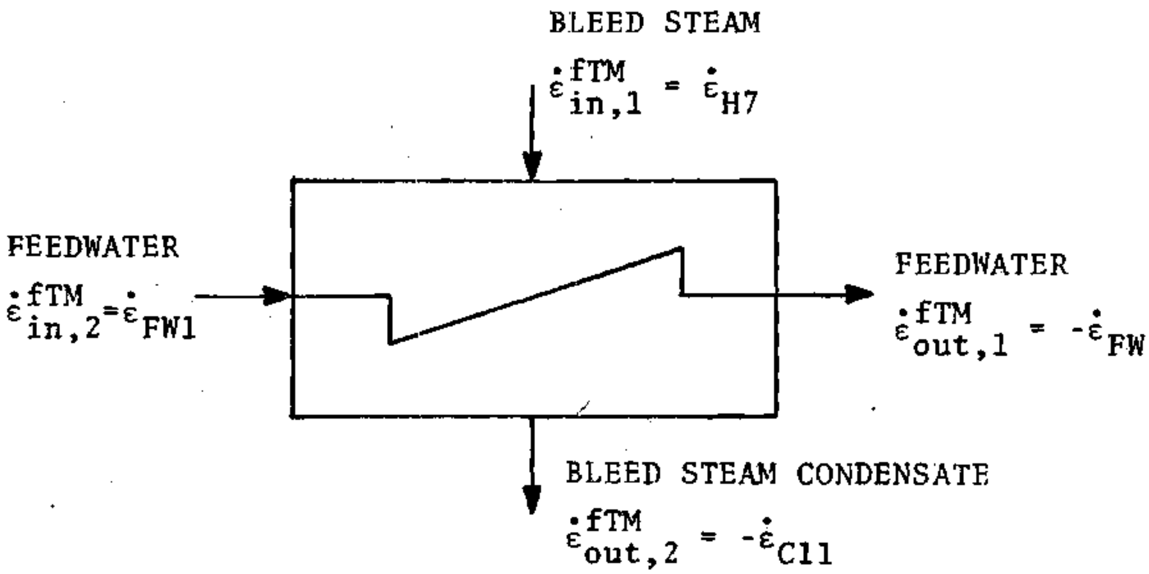
Essergy Balance:

$$\dot{\varepsilon}_{BO} + \dot{\varepsilon}_{s9} + \dot{\varepsilon}_{s1} + \dot{W}_{s,COND} - \dot{\varepsilon}_{FW15} - \dot{\varepsilon}_{CW2} = T_o \dot{S}_{COND}$$

Economic Balance:

$$c_{e,BO}\dot{\varepsilon}_{BO} + c_{e,s9}\dot{\varepsilon}_{s9} + c_{e,s1}\dot{\varepsilon}_{s1} + c_{e,s}\dot{W}_{s,COND} - c_{e,FW15}\dot{\varepsilon}_{FW15} + \dot{C}_{COND} = 0$$

Figure 5. Schematic of the Essergy Flows to the Condenser and Auxiliaries



Effectiveness:

$$\eta_{II,FH7} = \frac{\dot{\varepsilon}_{FW} - \dot{\varepsilon}_{FW1}}{\dot{\varepsilon}_{H7} - \dot{\varepsilon}_{C11}}$$

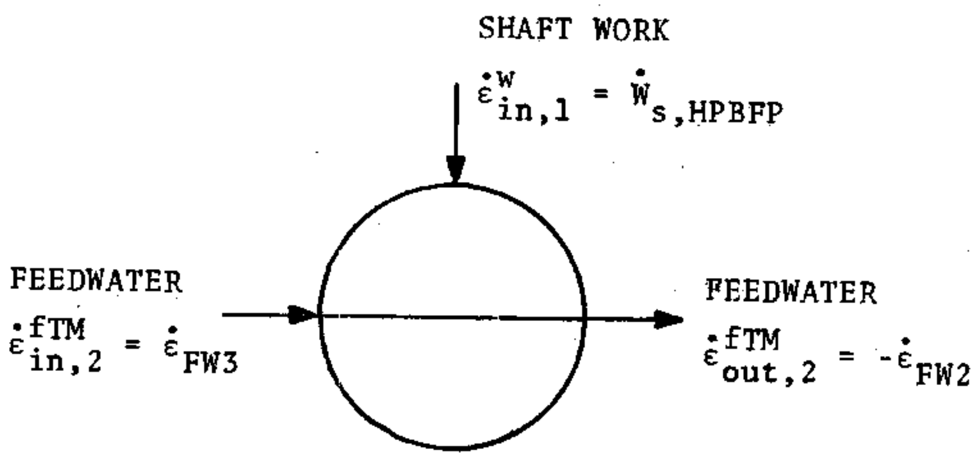
Essergy Balance

$$\dot{\varepsilon}_{FW1} + \dot{\varepsilon}_{H7} - \dot{\varepsilon}_{FW} - \dot{\varepsilon}_{C11} = T_o \dot{S}_{FH7}$$

Economic Balance:

$$c_{e,FW1}\dot{\varepsilon}_{FW1} + c_{e,H7}\dot{\varepsilon}_{H7} - c_{e,FW}\dot{\varepsilon}_{FW} - c_{e,C11}\dot{\varepsilon}_{C11} + \dot{C}_{FH7} = 0$$

Figure 6. Schematic of the Essergy Flows to Feedwater Heater Number 7



Effectiveness:

$$\eta_{II,HPBFP} = \frac{\dot{\varepsilon}_{FW2} - \dot{\varepsilon}_{FW3}}{\dot{W}_{s,HPBFP}}$$

Essergy Balance:

$$\dot{\varepsilon}_{FW3} + \dot{W}_{s,HPBFB} - \dot{\varepsilon}_{FW2} = T_o\dot{S}_{HPBFP}$$

Economic Balance:

$$c_{e,FW3}\dot{\varepsilon}_{FW3} + c_{e,s}\dot{W}_{s,HPBFP} - c_{e,FW2}\dot{\varepsilon}_{FW2} + \dot{C}_{HPBFP} = 0$$

Figure 7. Schematic of the Essergy Flows to the High Pressure Boiler Feed Pump

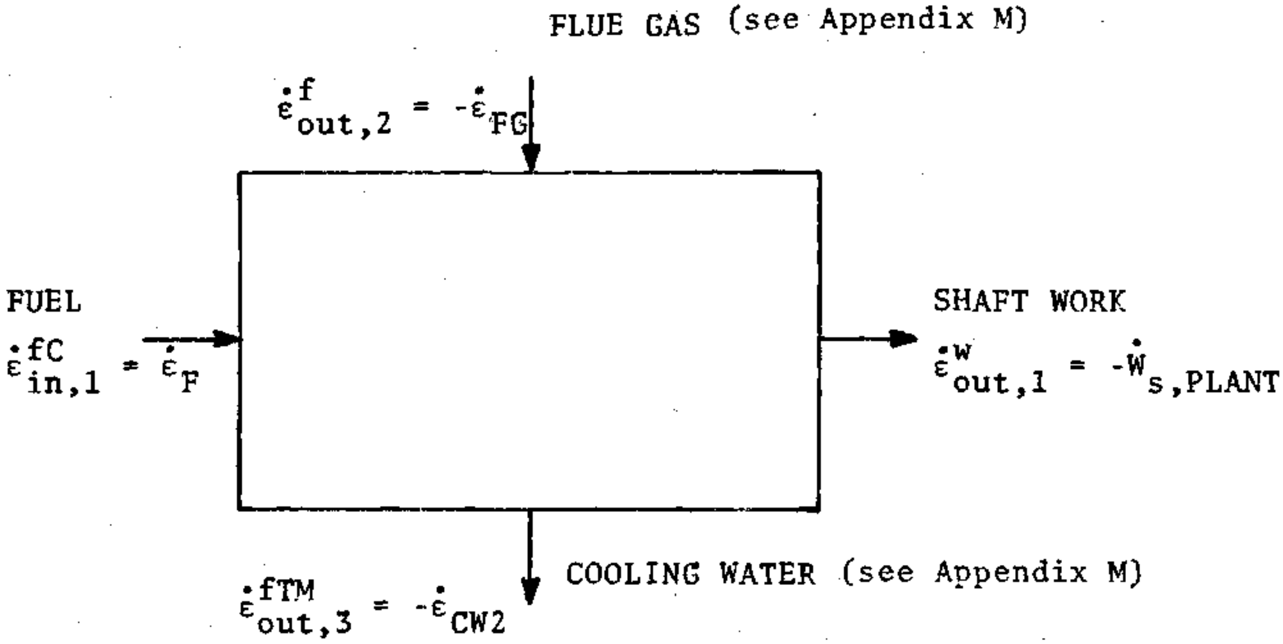
balance, respectively, for various zones in the power plant. Figure 8 illustrates the same methods applied to the entire power plant. For this study, all of the zones are assumed to be operating at steady state.

All other zones of the plant can be analyzed by using the illustrations provided by Figures 1 through 8. For example, the second intermediate pressure turbine stage (second reheat turbine stage) and all five low pressure turbine stages are similar to the high pressure turbine so that it will serve as an example for setting up their essergy balance, effectiveness and economic balance equations. In like manner, feedwater heater number 7 may be used as an example for analyzing all of the other feedwater heaters plus the two steam air preheaters.

For this analysis, power consuming devices in the plant such as pumps are allowed to buy power at the same unit cost that applies at the buss bars. This assumption allows the unit cost of power to remain constant for all variations to be considered in the power plant which is consistent with linearization of the power cycle.

D. Capital Costs of Power Plant Components

Total power plant capital cost was quoted at $150 per kilowatt by Evans (1976). For this study, capital costs are assumed to be escalated to just under $200 per kilowatt to set total plant capital cost at $60,000,000. The breakdown



Effectiveness:

$$\eta_{II,PLANT} = \frac{\dot{W}_{s,PLANT}}{\dot{\varepsilon}_{F}}$$

Essergy Balance:

$$\dot{\varepsilon}_{F} - \dot{\varepsilon}_{FG} - \dot{\varepsilon}_{CW2} - \dot{W}_{s,PLANT} = T_{o}\dot{S}_{PLANT}$$

Economic Balance:

$$c_{e,F}\dot{\varepsilon}_{F} - c_{e,s}\dot{W}_{s,PLANT} + \dot{C}_{PLANT} = 0$$

Figure 8. Schematic of the Essergy Flows to the Power Plant

of the total plant capital cost into capital costs for the individual major components is displayed in Table 1. The amortized cost rates for all of the major zones in the power plant, except for the turbines, were determined using the following expression:

$$\dot{C}_R = \frac{\phi_R C_{CAP,R}}{\ell_R} \tag{20}$$

where:

$\dot{C}_R$ = amortized capital cost rate for equipment in zone R

ϕ_R = dimensionless factor which includes maintenance costs, insurance, taxes and the time value of money; typically between 3.0 and 5.0 from Evans, et al. (1966)

ℓ_R = life of equipment in zone R

$C_{CAP,R}$ = initial installed capital cost of equipment in zone R.

For the amortized capital cost rates presented in Table 1, values of 3.5 for ϕ_R and 20 years for ℓ_R were used for all zones (except the turbines). The division of the initial total power plant capital cost into reasonable initial installed capital costs for each of the major zones was performed by the author using "best" engineering judgment. Since this study is meant to present parametric examples only, these assumed capital costs should suffice as long as they are reasonable.

According to Tribus (1978), the following equation represents capital cost for a power plant turbine stage.

$$\dot{C}_{\substack{\text{turbine}\\ \text{stage}}} = \dot{K}f(T_{in})[1-\eta_t]^{-a}\dot{m}\ \ln(P_{in}/P_{out}) \tag{21}$$

The exponent "a" is an empirically determined constant which represents the effect of isentropic turbine efficiency η_t. The constant $\dot{K}$ includes maintenance costs, insurance, taxes and the time value of money. The function $f(T_{in})$ represents the effect of maximum turbine operating temperature and is a severely steep when turbine inlet temperature T_{in} approaches some limiting temperature imposed by metallurgical considerations. In view of the term $\dot{m}\ln(P_{in}/P_{out})$, it is seen that the capital cost per turbine stage is proportional to the isotropic work of expansion.

For this study, it was felt that linearization of the power cycle would be more closely approached if each turbine stage was viewed as a simple essergy consumption system for the purpose of setting capital cost and the capital cost for each stage set proportional to its essergy input $\dot{\varepsilon}_{in}$. Therefore, the total capital cost for all turbine stages ($28,800,000 from Table 1) was allocated among each of the eight turbine stages according to the following equation:

$$\dot{C}_{\substack{\text{turbine}\\ \text{stage}}} = \dot{K}f(T_{in})[1-\eta_t]^{-a}\ \dot{\varepsilon}_{in} \tag{22}$$

where:

$a = 3.0$ (assumed)

$\dot{K} = 1.923125 \times 10^{-4} \frac{\$}{Btu}$ (based on the total capital cost of all turbine stages, \$28,800,000).

The dimensionless function $f(T_{in})$ is assumed to be given by,

$$f(T_{in}) = 1 + \left(\frac{T_{in}-T_o}{T_R-T_o}\right)^B \tag{23}$$

where:

$T_R = 1050°F$ (assumed)

$T_o = 50.4°F$

$B = 14.159$ (assumed)

$T_{in} \geq T_o$

The turbine stage capital costs given in Table 1 and the plot of $f(T_{in})$ which appears in Figure 9 were both generated by computer program BH1 presented in Appendix L.

E. Results of the Power Cycle Essergy Analysis

Using the equations (18) and (19) given in Section B of this chapter, the specific essergy and essergy flow associated with steam, water, air or fuel mass flows at various points in the power plant have been calculated for design conditions and are presented in Table E-1 in Appendix E. Table E-2, also in Appendix E, shows the change in essergy flows due to deterioration in feedwater heater number 5. Utilizing the data from Table E-1 and the methods demonstrated

Table 1. Capital Cost of Power Plant Equipment

	Installed Cost,$	Amortized Cost Rate,$/Hr	Total Number Installed	Total Installed Cost,$	Per Cent of Total Plant Cost
Furnace-Boiler	10,800,000	215.753	1	10,800,000	18
(High Pressure Section)					
Furnace-Boiler	10,800,000	215.753	1	10,800,000	18
(Reheat Section)					
Condenser and Auxilliaries	7,200,000	143.836	1	7,200,000	12
Feedwater Heater	255,000	5.094	7	1,785,000	
Boiler Feed Pump	270,000	5.394	1	270,000	
(High Pressure)					4
Boiler Feed Pump	115,000	2.297	1	115,000	
(Low Pressure)					
Air Preheater	115,000	2.297	2	230,000	
High Pressure Turbine		135.510			
Intermediate Pressure Turbines					
Stage 1		44.098			
Stage 2		55.028			
Low Pressure Turbines			8	28,800,000	48
Stage 1		81.312			
Stage 2		112.673			
Stage 3		56.935			
Stage 4		51.842			
Stage 5		37.941			
Plant		1198.630		60,000,000	100

Note: The furnace-boiler includes the economizer and the stack air preheater.

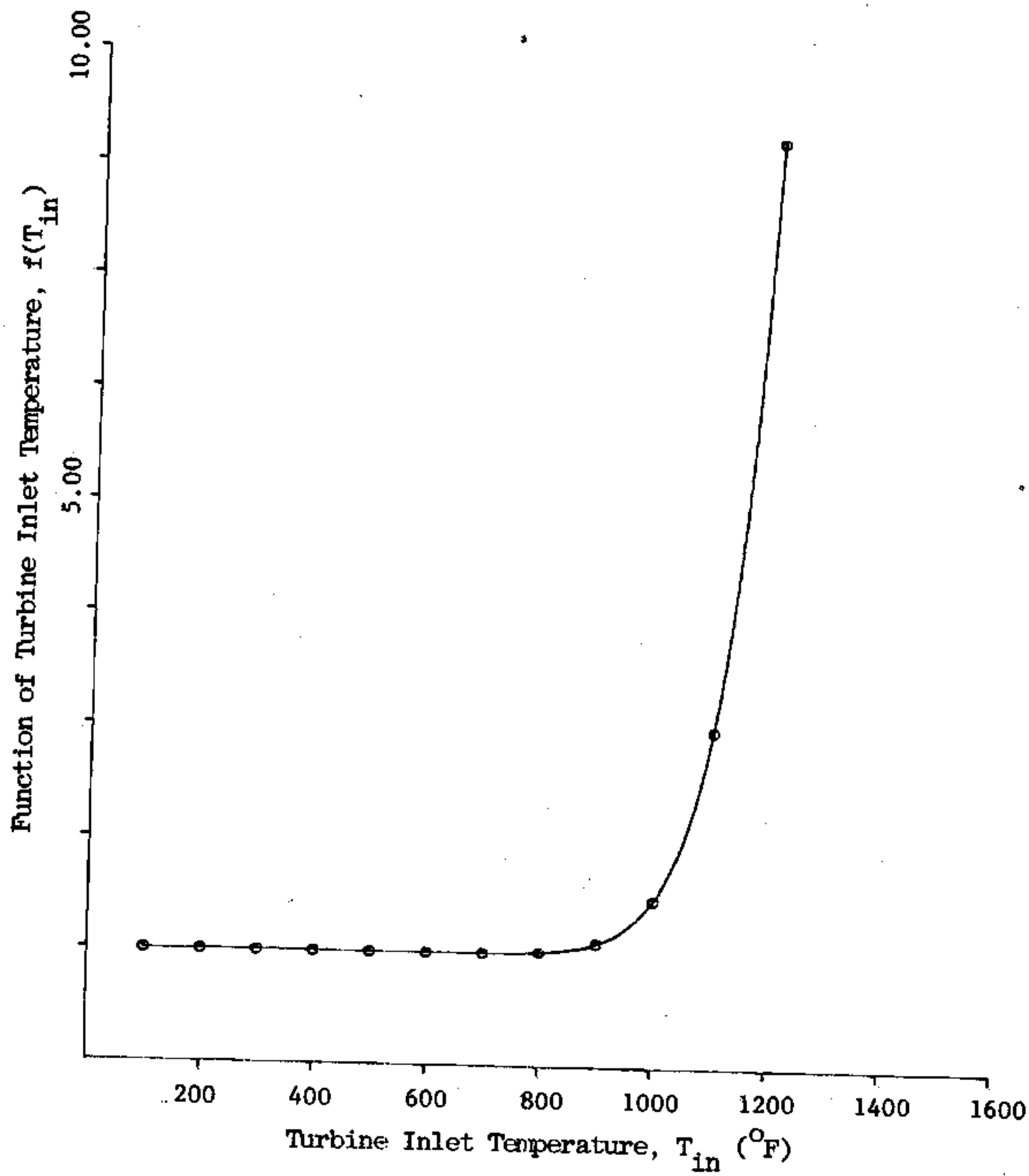


Figure 9. Dimensionless Function of Turbine Inlet Temperature

in Section C of this chapter, the essergy dissipation and effectiveness for the major zones in the plant at design conditions have been calculated and are shown in Table 2.

F. On the Solution of the Power Cycle Economic Balance Equations

The major difficulty with making an economic analysis of the power plant based on internal essergy flows is that the economic balance equations produced by applying the methods outlined in Section C of this chapter are not independent. After setting the independent, external, economic constraints for the power cycle, one is then allowed only a single arbitrary degree of freedom for setting internal cash flows.[11] In other words, once a particular internal cash flow is arbitrarily set, all other internal cash flows depend directly upon it and the external constraints.[12] For example, if the cost of the feedwater entering the economizer section of the boiler is set at a particular value, then all other internal cash flows will be adjusted by simultaneous solution of the internal cost balance equations to yield exactly the

[11] Independent, external, economic constraints include the cost of fuel and the cost of equipment (amortized capital cost including interest, maintenance, taxes, insurance, etc.).

[12] The external cash flow associated with the power sold is independent of the internal cash flows, but is not an independent external variable. That is, once the other external constraints have been set, the cash flow associated with the power output depends directly upon them.

Table 2: Essergy Dissipation and Effectiveness Power Plant Zones

	Essergy Dissipation, Million BTU/hr	Per Cent of Total Plant Essergy Dissipation	Effectiveness
Furnace-Boiler (High Pressure Section)	1048.726	74.63	0.506
Furnace-Boiler (Reheat Section)	182.88	13.01	0.495
High Pressure Turbine	25.579	1.82	0.922
Intermediate Pressure Turbines			
Stage 1	13.503	0.96	0.922
Stage 2	10.634	0.76	0.932
Low Pressure Turbines			
Stage 1	7.375	0.52	0.925
Stage 2	12.735	0.91	0.922
Stage 3	6.763	0.48	0.915
Stage 4	6.808	0.48	0.909
Stage 5	18.859	1.34	0.863
Condenser and Auxilliaries	34.785	2.48	0.033

Table 2 (cont.)

	Essergy Dissipation, Million BTU/hr	Per Cent of Total Plant Essergy Dissipation	Effectiveness
Feedwater Heater Number 1	2.30	0.16	0.677
Feedwater Heater Number 2	1.406	0.10	0.832
Feedwater Heater Number 3	1.579	0.11	0.872
Feedwater Heater Number 4	4.959	0.35	0.851
Feedwater Heater Number 5	2.302	0.16	0.905
Feedwater Heater Number 6	3.660	0.26	0.922
Feedwater Heater Number 7	4.856	0.35	0.938
Boiler Feed Pump (High Pressure)	4.716	0.34	0.791
Boiler Feed Pump (Low Pressure)	0.570	0.04	0.354

Table 2 (cont.)

	Essergy Dissipation, Million BTU/hr	Per Cent of Total Plant Essergy Dissipation	Effectiveness
Air Preheater Number 1	1.927	0.14	0.666
Air Preheater Number 2	2.472	0.18	0.760
Miscellaneous	5.869	0.42	-
Plant	1405.274	100.00	0.432

Note: The essergy dissipation for both the high pressure and reheat sections of the furnace-boiler (including the economizer and stack preheater) includes the essergy that is "thrown away" with the flue gas. The essergy dissipation for the condenser and auxilliaries also includes the essergy that is "thrown away" with the cooling water. The essergy dissipation for the entire plant includes the essergy that is "thrown away" with both the flue gas and the cooling water.

particular value that the feedwater cost was set at initially.

One is therefore faced with the question of which internal cash flow to set and at what value to set it. Before developing a method for setting the internal cash flows properly, a review will be made of some of the difficulties that were encountered in attempting to solve the set of dependent internal cost balance equations in hopes that it will shed some light on the nature of the problem. The resolution of these difficulties lead directly to an understanding of the concept of a single arbitrary degree of freedom for setting internal cash flows for the power cycle.

Due to the complexity of the power cycle, its set of simultaneous cost balance equations was programmed for solution on digital computer (see program WAH2 in Appendix L). The cash flows of interest were those associated with the bleed steam flowing to each feedwater heater. Initial trials at solution on the computer involved setting the value of the unit cost for the essergy entering with the feedwater into the economizer at some arbitrary value and solving all cost balance equations simultaneously. These trials indicated that the unit essergy cost associated with the feedwater flow to the economizer would always iterate to a unique value for a particular set of external economic constraints, regardless of what value it was set at initially. In addition, the two low pressure turbine stages nearest the condenser exhibited negative unit essergy costs for the steam being bled for feedwater heating or exhausted to the condenser.

This apparent lack of arbitrariness in setting one internal cash flow ran counter to what was intuitively expected and the negative unit essergy costs were unexpected. Therefore, in order to fully understand the economics of the system, it would be necessary to determine if the solution was truly iterative and if the negative unit essergy costs were real.

The first attempt at understanding the economics of the power cycle involved studying the value of the cash flows associated with the zone capital costs. It was felt that the behavior of the system was possibly due to improper allocation of plant capital costs. Accordingly, all zone capital costs were set at several different yet realistic values with turbine stage capital costs always being determined via equation (22). The varying of zone capital costs was found to have no effect on the iterative nature of the economics of the power cycle and from one to three of the low pressure turbine stages nearest the condenser still exhibited negative unit essergy costs depending on the relative magnitude of the zone capital costs.

Faced with the fact that the problems were not arising due to improper zone capital cost allocation, it was postulated that the principle of a single arbitrary degree of freedom for setting the internal cash flows of the power cycle was destroyed by the bleeding of steam from the various turbine stages for feedwater heating at a constant, unique, unit essergy cost for each turbine stage. Because of the complexity of the power cycle under consideration in this study, it was decided to investigate the above hypothesis using simpler power cycles. Accordingly, sets of internal cost balance equations were written for several different simple power cycles and nature of the simultaneous solution of each investigated. Figures F-1 through F-6 in Appendix F give a profile of the different simple power cycles that were studied. Investigation of these simple power cycles served to illustrate the effect of particular operations such as reheating, regenerative feedwater heating and air preheating on the economics of the complex power cycle.

In all cases studied, the principle of a single arbitrary degree of freedom for setting internal cash flows held and the unit essergy cost associated with the turbine exhaust to the condenser was negative. For the case illustrating the effect of air preheating, it was found that the value of the essergy associated with the preheated combustion air entering the boiler and the bleed steam used to preheat the combustion air are dependent on each other. Thus, the cash flow associated with the preheated combustion air had to be found by iteration before any other cash flow could be set arbitrarily. It was noted that the converse would also hold; in order to set the combustion air cash flow arbitrarily requires that the bleed steam essergy cost be found by iteration.

The fact that the principle held for the simple power cycles implied that it should also hold for the complex power cycle. A closer examination of the computer solution of the set of internal cost balance equations for the complex power cycle revealed that the system was indeed exhibiting the principle of a single arbitrary degree of freedom for setting internal cash flows, but the fact was being covered up by the bleed steam and combustion air dependency. A slight modification in the program allowed proof of the principle (see Table F-1 in Appendix F).

It was also obvious that the negative unit essergy costs associated with the bleed steam flows from the low pressure turbine stages near the condenser were real. The reason for the negative cash flows is directly related to a simplification made earlier in the analysis; that all forms of essergy would be viewed as economically equivalent.

In order to illustrate the error caused by this simplification, one must consider the essergy inputs and outputs for a turbine stage operating at less than atmospheric pressure (e.g., low pressure turbine stages 2, 3, 4, and 5) as illustrated in Figure 10. The flow thermomechanical essergy associated with the mass flow of steam through the turbine is divided into its two different forms; flow thermal essergy and flow mechanical essergy (flow cell essergy being neglected). The direction for the flow thermal essergy is

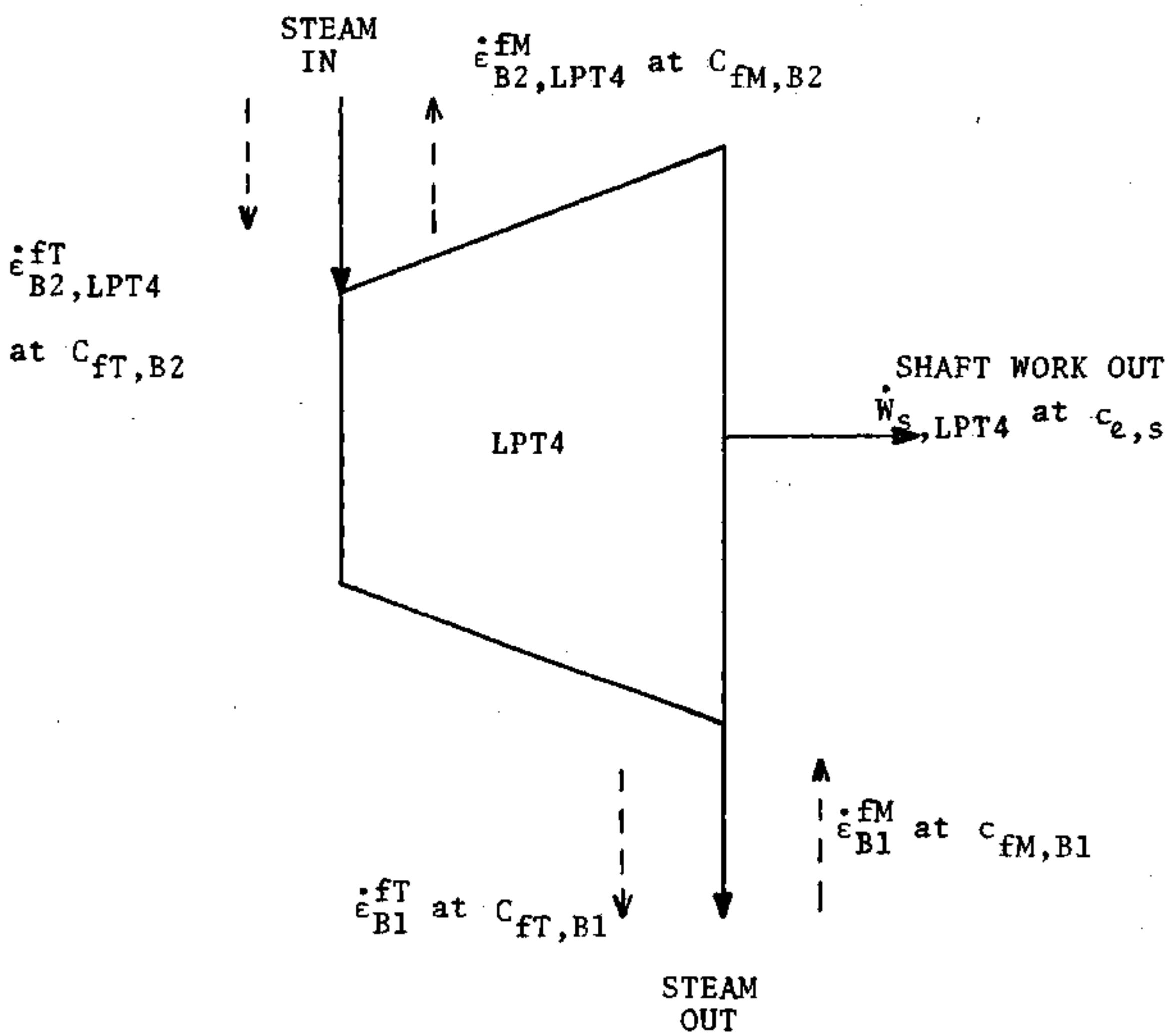


Figure 10. Low Pressure Turbine Stage Operating at Sub-Atmospheric Pressure

the same as the steam mass flow while the flow mechanical essergy moves in the opposite direction. The direction of the flow mechanical essergy for this case is not surprising since it is well known that the mechanical essergy of a vacuum always flows out of the vacuum as mass flows in.

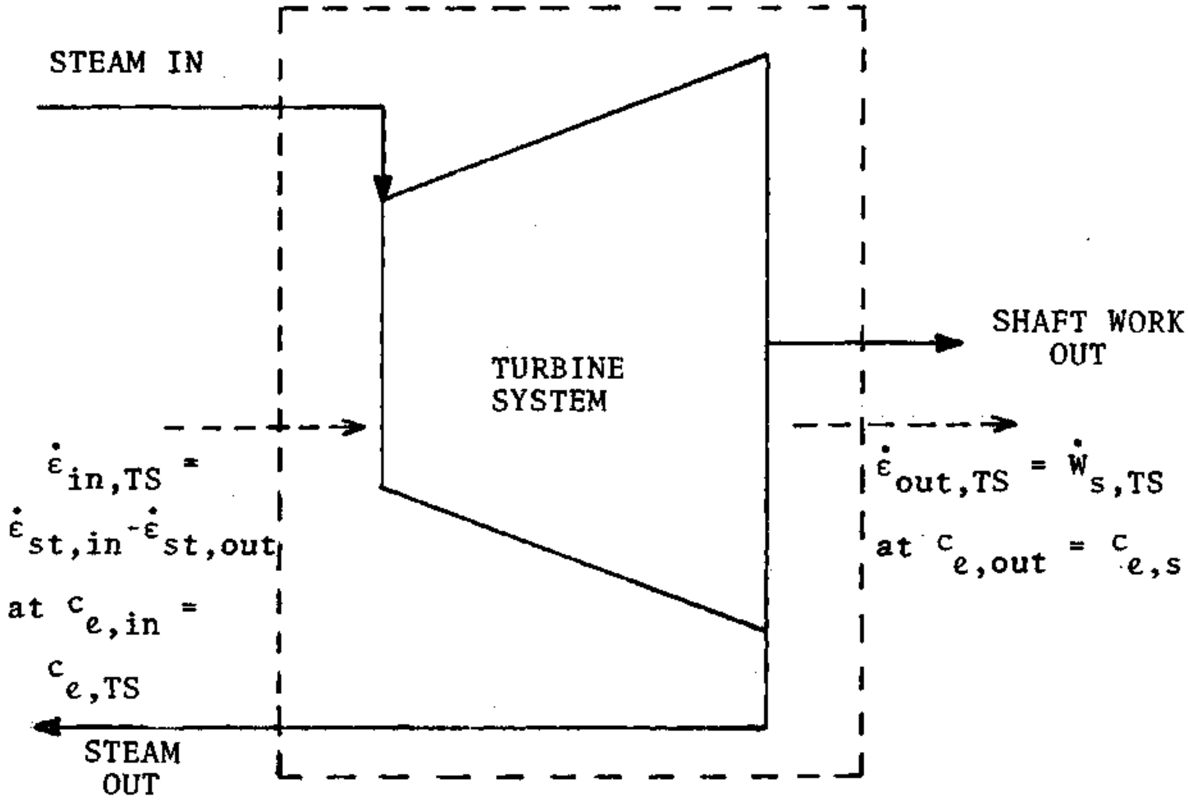
Due to the large contribution of the latent heat of vaporization for steam to the flow thermal essergy, its value is greater than the value of the flow mechanical essergy at both the inlet and outlet of the turbine (i.e., $\dot{\varepsilon}^{fT}_{B2,LPT4} > \dot{\varepsilon}^{fM}_{B2,LPT}$ and $\dot{\varepsilon}^{fT}_{B1} > \dot{\varepsilon}^{fM}_{B1}$). Therefore, the net value of the hydrodynamic flow essergy at both the turbine inlet and outlet is positive. Because these two forms of hydrodynamic flow essergy are not economically equivalent and the unit cost of the flow mechanical essergy can be greater than that of the flow thermal essergy at one or both points (i.e., $c_{fM,B1} > c_{fT,B1}$ and $c_{fM,B2} > c_{fT,B2}$), the unit cost of flow thermo-mechanical essergy at one or both points can be negative. Hence, the negative unit essergy costs are real and are an indication of the true state of affairs at the inlet and outlet of a low pressure stage operating at less than atmospheric pressure.

Now, unless another scheme is devised, the economic value of the essergy at various points in the power cycle cannot be accurately assessed unless each is split into its flow thermal and flow mechanical forms. This type analysis would not be a desirable approach since it would lead to a great deal of complexity. The next task, then, is to devise a simpler approach for analyzing the economics of the power cycle essergy flows.

It is obvious that if the value of the flow thermal essergy is high enough at the sub-atmospheric points, the negative unit essergy costs will be eliminated. One way in which this situation could be achieved would be by arbitrarily assigning a large value to the unit cost of the essergy flowing at some point in the cycle which has a large flow thermal essergy value associated with it. For example, such a point would be the feedwater entrance or throttle steam exit from the high pressure section of the boiler. However, there is no justification for arbitrarily setting the unit essergy costs at any point in the power cycle.

If unit costs for essergy flows to and from the turbines can be found which are independent of the amount of essergy flowing (i.e., if the turbine system is linearized), the negative costs will not occur for the low pressure turbine stages operating at sub-atmospheric conditions. A method for linearizing the turbine system may be had as follows. First, one must consider all of the turbines to be acting together as a simple essergy consumption system (see Figure 11). Equation (25) may be rearranged as follows:

$$c_{e,TS} = \frac{c_{e,s}\dot{W}_{s,TS} - \dot{C}_{TS}}{\dot{\varepsilon}_{st,in} - \dot{\varepsilon}_{st,out}} \qquad (26)$$



Economic Balance:

$$c_{e,in}\dot{\varepsilon}_{in,TS} + \dot{C}_{TS} = c_{e,out}\dot{\varepsilon}_{out,TS} \tag{24}$$

$$c_{e,TS}(\dot{\varepsilon}_{st,in} - \dot{\varepsilon}_{st,out}) + \dot{C}_{TS} = c_{e,s}\dot{W}_{s,TS} \tag{25}$$

Figure 11. All Turbines Acting as a Simple Essergy Consumption System

Equation (26) is used to calculate a value for $c_{e,TS}$ and then this value is used to calculate a value for $c_{e,TS}\dot{\varepsilon}_{in,TS}$. Setting the sum of all cash flows associated with the total essergy input to all of the turbine stages equal to this calculated value for $c_{e,TS}\dot{\varepsilon}_{in,TS}$ in the computer program for simultaneous solution of the internal cost balance equations (see program WAH2 in Appendix L) will cause the unit essergy costs for all steam flows to and from the turbines to be approximately equal to $c_{e,TS}$. If the effectiveness of every turbine stage were exactly equal and if the capital cost of each turbine stage is exactly proportional to its net essergy input, then the unit cost of all essergy flows to and from the turbine stages would be exactly equal to $c_{e,TS}$.

Table 3 presents the unit essergy costs for the turbine system for three different trials. In Trial 1, the unit essergy cost for the feedwater entering the economizer and the combustion air entering the stack air preheater were arbitrarily set at zero (an assumption made by Fehring and Gaggioli (1977) in their analysis of this power cycle) and the pertinent internal cost balance equations solved simultaneously to yield values for the unit essergy costs for the turbine system. The unit essergy costs in Trial 2 were calculated using the method described above for linearizing the turbine system except that capital costs were neglected. The unit essergy costs in Trial 3 were calculated using the linearizing method described above with turbine stage

Table 3. Unit Costs of Essergy Associated With Steam Flows To and From the Turbine Stages

Point	Type of Flow	Turbine Stage	Effectiveness	Specific Essergy of Flows, BTU/Lb	Unit Costs, $/Million BTU		
					Trial 1	Trial 2	Trial 3
TS	Input	Turbine System	0.915	-	-	1.696	2.248
T	Input	HPT	0.922	700.917	1.293	1.702	2.213
RH	Input	IPT1	0.922	631.828	1.233	1.689	2.292
S6	Input	IPT1	0.922	679.720	1.334	1.755	2.282
B7	Output	HPT	0.922	522.741	1.152	1.700	2.173
H6	Output	IPT1	0.922	529.458	1.143	1.687	2.254
H5	Output	IPT2	0.932	430.225	1.008	1.678	2.215
H4	Output	LPT1	0.925	365.979	0.884	1.672	2.266
H3	Output	LPT2	0.922	251.601	0.509	1.655	2.365
H2	Output	LPT3	0.915	192.151	0.142	1.643	2.476
H1	Output	LPT4	0.909	131.742	-0.565	1.642	2.689
B0	Output	LPT5	0.863	15.225	-17.126	1.817	5.735

capital cost calculated as in Section D of this chapter.

Each unit essergy cost calculated in Trial 1 depends strongly on the specific essergy for the point at which it is determined. The unit essergy costs decrease with decreasing specific essergy and become negative for the last two low pressure turbine stages. The unit essergy costs calculated by the turbine system linearizing method in Trials 2 and 3 are all approximately equal to the value of $c_{e,TS}$ with the exception of $c_{e,BO}$. In Trial 2 the deviation of $c_{e,BO}$ from $c_{e,TS}$ is probably caused by the low value for the effectiveness of the last low pressure turbine stage and in Trial 3 by a combination of the low value for effectiveness for the last low pressure turbine stage and the fact that the capital cost allocated to each turbine stage was not exactly proportional to its net essergy input.[13]

Further in-depth study of the behavior of a system which includes the last low pressure turbine stage and the condenser will be required in order to properly set the unit essergy costs associated with these components of the power cycle. A treatment of this problem will not be performed for this paper.

If it is assumed that the effectiveness of the turbines remains constant throughout the life of the power plant, the unit essergy costs for the turbine system will remain constant

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[13]See Equation (22) in Section D of this chapter.

and approximately equal to $c_{e,TS}$, as long as the overall power cycle effectiveness and turbine system work output remain unchanged. This fact may be shown by rearranging equation (26) as follows:

$$c_{e,TS} = \frac{c_{e,s}\dot{W}_{s,TS}}{\dot{\varepsilon}_{st,in} - \dot{\varepsilon}_{st,out}} - \frac{\dot{C}_{TS}}{\dot{\varepsilon}_{st,in} - \dot{\varepsilon}_{st,out}} \qquad (27)$$

The effectiveness of the turbine system is given by,

$$\eta_{II,TS} = \frac{\dot{W}_{s,TS}}{\dot{\varepsilon}_{st,in} - \dot{\varepsilon}_{st,out}} \qquad (28)$$

In view of equation (28), equation (27) may be rewritten to obtain,

$$c_{e,TS} = c_{e,s}\eta_{II,TS} - \frac{\eta_{II,TS}\,\dot{C}_{TS}}{\dot{W}_{s,TS}} \qquad (29)$$

Since all of the terms on the right hand side of equation (29) will remain constant if the above assumptions of constant turbine system effectiveness, power cycle effectiveness and turbine system work output all hold, $c_{e,TS}$ will remain constant over the entire life of the power plant.[14] The

[14] For all practical purposes for a given power plant, the capital cost of the turbine system $\dot{C}_{TS}$ can be assumed to remain constant always.

analysis in which the value for $c_{e,TS}$ is being used. An illustration of this type of treatment will be shown later in this paper in a problem involving the economic analysis of repairing or replacing a deteriorated feedwater heater.

As turbines age they have various parts that wear out or fail mechanically. The change in effectiveness of a turbine due to aging is very likely negligible so that the assumption of constant turbine system effectiveness is probably justified.

As was indicated earlier, in their analysis of the power plant that is being considered in this study, Fehring and Gaggioli arbitrarily assigned the unit essergy cost associated with both the feedwater entering the economizer and the combustion air entering the stack air preheater a value of zero. In addition, they also neglected all capital cost contributions to the zone economic balances, indicating that they were irrelevant since they were already sunk. It has been observed, however, that there is no justification for arbitrarily setting unit essergy costs at any point within the power cycle.

Unit essergy costs calculated by the Fehring and Gaggioli method are highly dependent on the magnitude of the essergy flow and therefore are not linear. Fehring and Gaggioli calculated the unit essergy costs for only the first four turbine stages (high pressure through the first low pressure stage) and therefore did not observe the problem unit essergy costs for the turbine system may always be set approximately equal to $c_{e,TS}$ by this turbine system linearizing method and thereby power cycle internal cash flows properly set.

It is obvious that changes in turbine system effectiveness (i.e., $\eta_{II,TS}$ not constant), power cycle effectiveness (i.e., $c_{e,s}$ not constant) or turbine system work output (i.e., $\dot{W}_{s,TS}$ not constant) will be reflected by a change in the value of $c_{e,TS}$. For this reason, changes in the value of $c_{e,TS}$, as $\eta_{II,TS}$, $c_{e,s}$ or $\dot{W}_{s,TS}$ is varied, may be used as a measure of the effect of changes in these parameters.

All of the assumptions used in the linearizing the turbine system (i.e., $\eta_{II,TS}$, $c_{e,s}$ and $\dot{W}_{s,TS}$ all remaining constant over the life of the power plant) do not appear to be unreasonable. Power plants are designed to be operated at or near maximum capacity at a fixed heat rate and are usually operated in just this manner which will justify the assumptions of constant values for $c_{e,s}$ and $\dot{W}_{s,TS}$.

As the price of fuel escalates or if the fuel flow to the boiler changes, it will undoubtedly cause the value of $c_{e,s}$ to change which runs counter to our assumption of constant power cycle effectiveness. The error introduced by assuming that $c_{e,s}$ remains constant when in fact it is changing because of the escalating price of fuel or changes in the fuel flow rate to the boiler may be rectified by treating the escalation or changed fuel flow in the economic with negative unit essergy costs for the low pressure turbine stages near the condenser. Had they continued their analysis to include all of the turbine stages, they would have observed these negative unit essergy costs and suspected that they might be nonlinear.

Fehring and Gaggioli assumed that the unit essergy costs calculated by their method would remain constant over the life of the power plant and therefore could be used to make economic decisions concerning the repair or replacement of a feedwater heater that had deteriorated. As a feedwater heater deteriorates, the condensing temperature of the bleed steam must go up in order for the feedwater heater to continue to carry its design load. Obviously, as the condensing temperature of the bleed steam rises, the exit temperature of the turbine from which the steam is being bled must also go up and therefore the essergy flowing with the bleed steam must change. Since the unit essergy costs calculated by the Fehring and Gaggioli method are dependent on the amount of essergy flowing, their assumption of a constant unit essergy cost for bleed steam flowing to a feedwater heater that has deteriorated is invalid. Unit essergy costs for the turbine system calculated using the method that was developed earlier in this study are linear with amount of essergy flowing and therefore will work very effectively for making economic decisions, regardless of the condition of the various components within the power cycle. Provided the assumptions that have been made in developing the method all hold, the unit essergy costs for the bleed steam from the turbines will remain constant even though the essergy flowing with the bleed steam may vary.

G. Results of the Power Cycle Economic Analysis

Fehring and Gaggioli used the feedwater heater unit essergy costs that they calculated to make an analysis of the economic feasibility of repairing or replacing feedwater heater number 5 which is operating with deteriorated performance due to plugged tubes. In this study, the validity of the unit essergy costs determined by their method and the unit essergy costs calculated using the turbine system linearizing method developed in Section F will be investigated by using both sets of costs to make the same economic analysis of feedwater heater number 5 and comparing results.

Hourly essergy costs associated with steam, water, fuel and air flows at various points in the cycle are calculated for three different trials and are presented in Table G-1 of Appendix G. Table G-2, also in Appendix G, shows the essergy and cash flows associated with the shaft work flowing to or from various components in the plant. Trial 1 calculations were made by solving the plant economic balance equations on digital computer (program WAH2 in Appendix L) using the Fehring and Gaggioli assumptions. Calculations for Trials 2 and 3 were also performed using digital computation (program WAH2 in Appendix L) with the unit essergy costs associated with the turbine stages set approximately equal to $c_{\varepsilon,TS}$ using the linearizing method developed earlier. Trial 2 neglects zone capital cost contributions while Trial 3 uses the zone capital costs calculated in Section D of this chapter. For simplicity in all three trials, the unit essergy costs associated with the condensate flows (drips) from the feedwater heaters have been neglected.

The hourly essergy costs calculated in Trials 1, 2 and 3 will be used in the next chapter to calculate the costs of feedwater heating for Cases A, B and C described earlier in this study. These feedwater heating costs will be used to determine the economic feasibility of repairing or replacing feedwater heater number 5. In this manner, the relative effect of the various assumptions made in Trials 1, 2 and 3 may be determined and the usefulness of the numbers from each trial may be evaluated for making economic operational decisions.

CHAPTER IV

ECONOMIC ANALYSIS FOR REPAIR OR REPLACEMENT OF FEEDWATER HEATER NUMBER 5

The total hourly feedwater heating cost for heaters 4 through 7 for Cases A, B, and C described earlier in this paper may be had by summing the hourly cost of the essergy flowing with the bleed steam to each of the heaters for each case.[15] This approach is equivalent to assuming the feedwater heaters are simple essergy consumption systems. Since it is desired that Trials 1, 2, and 3 be compared for their relative value in making economic decisions, the total hourly feedwater heating cost for all three cases must be determined for each trial.

The total hourly feedwater heating cost for heaters 4 through 7 for Case A may be had directly for all three trials by summing the hourly bleed steam essergy costs calculated for each in Section G of Chapter III and given in Table G-1 of Appendix G. The unit bleed steam essergy costs for heaters 4 through 7 for each of the trials may be had by dividing the hourly bleed steam essergy cost to each heater at design operation by the essergy associated with the corresponding bleed steam flow at design conditions. These unit essergy costs are assumed to be constant over the

[15]The hourly costs of heaters 1, 2, and 3 are considered constant for all three cases and therefore are irrelevant.

life of the power cycle and may be used in conjunction with the essergy flows to feedwater heaters 4 through 7 for Cases B and C (from Table E-2) to calculate hourly costs to these heaters for each trial. These hourly costs may be summed for each case and each trial to give total hourly feedwater heating cost.[15] The individual hourly feedwater heating costs and total hourly feedwater heating costs for heaters 4 through 7 for Cases A, B, and C and Trials 1, 2, and 3 are presented in Table 4.

For the repair or replacement analysis, it is assumed that the power plant operates 8000 hours per year at an average of 70 percent of capacity with feedwater heater number 5 down for three weeks per year for plugging of 15 leaks. Maintenance costs are determined at 28 man-hours per leak with $10.07 charged for each man-hour. Using this data in conjunction with the calculated total feedwater heating costs for each of the cases, the annual fuel and maintenance expenditure for operating feedwater heater number 5 in a deteriorated condition can be calculated for Trials 1, 2, and 3.

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[15]In Case C additional fuel is needed since the temperature of feedwater entering the boiler economizer is lower than design conditions. Since the unit bleed steam essergy costs were calculated from design conditions, and considered constant they do not reflect the cost of this decreased fuel flow (a point that was discussed earlier in Section F of Chapter III). Therefore, the cost of additional fuel essergy needed must be added to bleed steam essergy costs to obtain true total hourly feedwater heating cost for Case C.

Table 4. Essergy Costs for Feedwater Heaters 4 Through 7

Point	Unit Cost of Essergy, $/MMBTU			Essergy Costs for Feedwater Heating, $/Hr								
				Trial 1			Trial 2			Trial 3		
	Trial 1	Trial 2	Trial 3	Case A	Case B	Case C	Case A	Case B	Case C	Case A	Case B	Case C
H7	1.154	1.709	2.185	108.522	108.522	112.129	160.709	160.709	166.055	205.484	205.484	212.306
H6	1.147	1.702	2.273	56.142	60.674	81.316	83.305	90.033	120.662	111.268	120.238	161.143
H5	1.065	1.746	2.292	29.508	26.334	0	48.365	43.173	0	63.484	56.673	0
H4	0.887	1.693	2.295	33.123	33.123	33.123	33.496	63.225	63.933	85.706	85.706	86.666
Fuel Increase	0.800	0.800	0.800	0	0	21.659	0	0	21.659	0	0	21.659
			Totals	227.295	228.653	248.600	355.604	357.140	372.309	465.942	468.101	481.774

Note: The unit essergy costs and hourly feedwater heating costs calculated in Trial 1 by the Fehring and Gaggioli method differ from those presented in their paper because of arithmetic errors contained in the paper.

Assuming that the fuel and maintenance expenditures escalate at a rate of six percent per year and that after tax cost of capital is nine percent, the annual cash flow due to the deterioration of heater number 5 can be calculated for each year that it is left in service.[16] If heater number 5 is replaced then the fuel and maintenance expenditure which would have resulted had it been left to operate in a deteriorated condition will represent a cost saving against which the cost of a new heater can be amortized. Using this viewpoint, the method for determining if replacement can be economically justified is to calculate the uniform annual fuel and maintenance savings for each year over the life of a new heater and compare this savings to the uniform annual cost of paying for the new heater. The repair analysis involves calculation of the maximum time that heater number 5 may be left down for retubing before replacement would become more economical.

The new heater is assumed to have a life of 20 years, a replacement cost of \$235,000 and a salvage value of \$18,000. Retubing of the old heater will cost \$185,000. The economics of replacing or retubing feedwater heater number 5 have been calculated for Trials 1, 2, and 3, and are presented in

[16]Allowing the cost of fuel to escalate in the economic analysis corrects the error introduced by neglecting escalation when calculating unit bleed steam essergy costs from design conditions considering them as constant throughout the life of the power plant (a point that was discussed earlier in Section F of Chapter III).

Tables 5 and 6, respectively. Details of all calculations are given in Appendix H.

Since the benefit-to-cost ratio for Trials 1 and 2 are less than unity, they both indicate that the operation of feedwater heater number 5 must deteriorate further before it will be profitable to replace it with a new heater. The benefit-to-cost ratio in Trial 3 is slightly greater than unity indicating that would be profitable to replace feedwater heater number 5. In reality, the decision that a company would make concerning replacement of feedwater heater number 5 based on the results of Trial 3 would depend on what rate of return on its investments that the company requires.

It is interesting to note that Trial 1 would lead to an economic decision different from that of Trial 3 for a power plant whose capital cost is not sunk. Trials 1 and 2 (for a power plant whose capital cost is sunk) both lead to the same decision for this particular example even though the essergy costs used in Trial 1 were considered invalid due to reasons discussed earlier in this paper. If a feedwater heater closer to the condenser had been selected for economic analysis, the effect of the non-linear unit essergy costs calculated by the Fehring and Gaggioli method (see Table 3) would be more pronounced and the economic decisions arrived upon in comparing the methods would have been radically different.

Table 5. Economic Evaluation for Repair of Feedwater Heater Number 5

Trial	Additional Essergy Cost Million Btu/hr	Maximum Allowable Downtime Hours (weeks)
1	21.305	2347 (14.0)
2	16.705	2993 (17.8)
3	15.832	3158 (18.8)

Table 6. Economic Evaluation for Replacement of Feedwater Heater Number 5

Trial	Uniform Annual Savings,$	Uniform Annual Cost,$	Benefit to Cost Ratio
1	22122	22817	0.9695
2	21096	22817	0.9246
3	23431	22817	1.0269

The repair analysis for feedwater heater number 5 indicates that it may be profitably left out of service for retubing the longest period of time in Trial 3, the next longest in Trial 2 and the least amount of time in Trial 1. The difference between the maximum profitable downtimes calculated for Trials 2 and 3 is due to the fact that zone capital costs were neglected in determining the essergy costs used in Trial 2. The decision on whether to use the Trial 2 or Trial 3 result would depend on whether or not the power plant capital cost was sunk. The difference between the maximum profitable downtime calculated for Trial 1 and those calculated in Trials 2 and 3 is a direct reflection of the non-linearity of the unit essergy costs used in Trial 1.

CHAPTER V

OPTIMUM DESIGN OF A FEEDWATER HEATER

Linearization of the power cycle by essergy analysis is important for design purposes since it leads to decentralization. Once decentralization is achieved, one is assured that optimization of the design of each zone within the power cycle will lead to an optimum design for the entire power cycle. Usually a system as complex as a power cycle requires treatment by LaGranges Method of Undetermined Multipliers for decentralization. Since decentralization may be achieved by the linearization method demonstrated in Section F of Chapter III, one may proceed directly with optimizing the design of the various components within the power cycle. It is in this spirit that an approach to the design of a feedwater heater will be developed in this chapter. This development will serve as an example for additional design optimization of feedwater heaters and for the design of other power cycle components.

A. Theoretical Development

The total cost of a feedwater heater is dependent on the sum of capital cost and essergy dissipation cost. If capital cost and essergy dissipation cost can be expressed in terms of number of transfer units or area available for heat transfer, the minimum cost for the heater occurs at minimum cost per transfer unit as long as the cost per transfer unit is not a function of the number of transfer units. Use of ordinary differential calculus to minimize the cost per transfer unit and total feedwater heater cost will lead to expressions for determining optimum velocity and optimum heat transfer area, respectively.

If a feedwater heater is considered as a simple essergy consumption system, the following equations may be written for the essergy and economic balances:

$$\dot{\varepsilon}_{FH,in} = \dot{\varepsilon}_{FH,out} + T_o\dot{S}_{FH} \tag{30}$$

$$c_{e,out}\dot{\varepsilon}_{FH,out} = \dot{C}_{FH} + c_{e,in}\dot{\varepsilon}_{FH,in} \tag{31}$$

Multiplying both sides of equation (30) by $c_{e,in}$ will yield,

$$c_{e,in}\dot{\varepsilon}_{FH,in} = c_{e,in}\dot{\varepsilon}_{FH,out} + c_{e,in}T_o\dot{S}_{FH} \tag{32}$$

Substitution of equation (32) into equation (31) and rearranging will give,

$$(c_{e,out}-c_{e,in})\dot{\varepsilon}_{FH,out} = \dot{C}_{FH} + c_{e,in}T_o\dot{S}_{FH} \tag{33}$$

One may easily recognize that the term $\dot{\varepsilon}_{FH,out}$ represents that part of the essergy input $\dot{\varepsilon}_{FH,in}$ that was transmitted by the feedwater heater from the bleed steam to the feedwater with the remainder being dissipated. Therefore, the quantity in parentheses in equation (33) represents the increase in the unit cost of the essergy transmitted by the feedwater heater which is necessary to pay for the amortized capital cost of the feedwater heater (including interest, taxes, insurance, maintenance costs, etc.) plus the cost of the essergy dissipation in the feedwater heater. Thus, the left hand side of equation (33) represents the increased charge for its product that must be made by the feedwater heater in order to satisfy its economic balance. This increased charge made by the feedwater heater may be viewed as the "net cost" $\dot{C}$ of the feedwater heater and equation (33) may be expressed by,

$$\dot{C} = \dot{C}_{FH} + c_{\varepsilon,in} T_o \dot{S}_{FH} \tag{34}$$

with $\dot{C}_{FH}$ being the amortized capital cost of the feedwater heater and $c_{\varepsilon,in} T_o \dot{S}_{FH}$ being the cost of essergy dissipation in the feedwater heater.

The cost per unit area for the feedwater heater $\dot{c}_A$ is a marginal cost and is given by,[17]

[17] Marginal costs are defined by $\partial c/\partial x$ where c is the cost and x is some system parameter which affects the cost.

$$\dot{c}_A = \frac{\partial \dot{C}_{FH}}{\partial A_{FH}} \tag{35}$$

If the system is linear as has been assumed, then the unit area cost is expressed by,

$$\dot{c}_A = \frac{\dot{C}_{FH}}{A_{FH}} \tag{36}$$

so that,

$$\dot{C}_{FH} = \dot{c}_A A_{FH} \tag{37}$$

Hence, equation (34) becomes,

$$\dot{C} = \dot{c}_A A_{FH} + c_{e,in} T_o \dot{S}_{FH} \tag{38}$$

The entropy creation in the feedwater heater is due not only to heat transfer across a finite temperature difference but also to fluid friction (i.e., head loss) on both the tubeside and shellside. Therefore, one may write for equation (38),

$$\dot{C} = \dot{c}_A A_{FH} + c_{e,ts} T_o \dot{S}_{ts} + c_{e,ss} T_o \dot{S}_{ss} + c_{e,ht} T_o \dot{S}_{ht} \tag{39}$$

where $T_o\dot{S}_{ts}$ is the essergy dissipation due to tubeside head loss, $T_o\dot{S}_{ss}$ is the essergy dissipation due to shellside head loss and $T_o\dot{S}_{ht}$ is the essergy dissipation due to heat transfer between the two fluid streams. Since the essergy dissipation due to head loss on both the tubeside and shellside of the feedwater heater is proportional to the area available for heat transfer (i.e., $T_o\dot{S}_{ts} \propto A_{FH}$ and $T_o\dot{S}_{ss} \propto A_{FH}$ where it is assumed that $A_{FH,ts} \cong A_{FH,ss} = A_{FH}$), one may write,

$$c_{e,ts}T_o\dot{S}_{ts} = \dot{c}_{A,ts}A_{FH} \tag{40}$$

and

$$c_{e,ss}T_o\dot{S}_{ss} = \dot{c}_{A,ss}A_{FH} \tag{41}$$

Now, equation (39) may be written in the following form:

$$\dot{C} = (\dot{c}_A+\dot{c}_{A,ts}+\dot{c}_{A,ss})A_{FH} + c_{e,ht}T_o\dot{S}_{ht} \tag{42}$$

If one makes the following definition,

$$\dot{c}_A' = \dot{c}_A + \dot{c}_{A,ts} + \dot{c}_{A,ss} \tag{43}$$

or

$$\dot{c}_A' = \dot{c}_A + \frac{c_{e,ts}T_o\dot{S}_{ts}}{A_{FH}} + \frac{c_{e,ss}T_o\dot{S}_{ss}}{A_{FH}} \tag{44}$$

equation (42) may be expressed by,

$$\dot{C} = \dot{c}_A' \, A_{FH} + c_{e,ht} T_o \dot{S}_{ht} \tag{45}$$

Treating the feedwater heater as a condenser and writing the entropy balance for heat transfer across a differential section of the tubing wall dL and integrating over the entire length L of the tubing, one will obtain the following expression for entropy creation due to heat transfer across a finite temperature difference (see Appendix I):

$$\dot{S}_{ht} = \dot{m}_{FW} C_{p,FW} (T_{FWe} - T_{FWi}) \left[\frac{\ln \frac{T_{FWe}}{T_{FWi}}}{T_{FWe} - T_{FWi}} - \frac{1}{T_{C,B}} \right] \tag{46}$$

or

$$\dot{S}_{ht} = \dot{Q}_{FH} \left[\frac{\ln \frac{T_{FWe}}{T_{FWi}}}{T_{FWe} - T_{FWi}} - \frac{1}{T_{C,B}} \right] \tag{47}$$

where $T_{C,B}$ is the condensing temperature of the bleed steam entering the feedwater heater and $\dot{Q}_{FH} = \dot{m}_{FW} C_{p,FW} \, (T_{FWe} - T_{FWi})$ is the total heat transferred to the feedwater stream.

Kays and London (1964) have shown that the temperature effectiveness z for a condenser is given by,

$$z = \frac{T_{C,out} - T_{C,in}}{T_{H,out} - T_{C,in}} \tag{48}$$

where the subscripts C and H in this case refer to the cold and hot streams, respectively. They have also shown that the temperature effectiveness for a condenser is related to the number of heat transfer units χ by,

$$z = 1 - e^{-\chi} \tag{49}$$

where the number of transfer units χ for a feedwater heater is given by,

$$\chi = \frac{UA_{FH}}{\dot{m}_{FW} C_{p,FW}} \tag{50}$$

with U being the overall conductance for the feedwater heater.

Applying equation (48) to a feedwater heater will give,

$$z = \frac{T_{FWe} - T_{FWi}}{T_{C,B} - T_{FWi}} \tag{51}$$

so that,

$$\frac{T_{FWe}-T_{FWi}}{T_{C,B}-T_{FWi}} = 1-e^{-X} \tag{52}$$

Solving equation (52) for $T_{C,B}$ one obtains,

$$T_{C,B} = \frac{T_{FWe}-T_{FWi}e^{-X}}{1-e^{-X}} \tag{53}$$

Substituting equation (53) into equation (47) will yield,

$$\dot{S}_{ht} = \dot{Q}_{FH}\left\{\frac{\ln\frac{T_{FWe}}{T_{FWi}}}{T_{FWe}-T_{FWi}} - \frac{1-e^{-X}}{T_{FWe}-T_{FWi}\ e^{-X}}\right\} \tag{54}$$

Rewriting equation (54) one has,

$$\frac{T_{FWi}\dot{S}_{ht}}{\dot{Q}_{FH}} = \frac{\ln\frac{T_{FWe}}{T_{FWi}}}{\frac{T_{FWe}}{T_{FWi}}-1} - \frac{1-e^{-X}}{\frac{T_{FWe}}{T_{FWi}}-e^{-X}} \tag{55}$$

Rearranging equation (50) for the number of transfer units will give,

$$A_{FH} = \frac{\dot{m}_{FW}C_{p,FW}X}{U} \tag{56}$$

Multiplying equation (56) by $\dot{c}_A'$, one may define c_z' by,

$$\dot{c}_A' \, A_{FH} = \frac{\dot{c}_A' \dot{m}_{FW} C_{p,FW} \chi}{U} = \dot{c}_z' \chi \tag{57}$$

Rearranging equation (57) one obtains for $\dot{c}_z'$,

$$\dot{c}_z' = \dot{m}_{FW} C_{p,FW} \left(\frac{\dot{c}_A'}{U}\right) \tag{58}$$

The term $\dot{c}_A'/U$ may be recognized as the cost per transfer unit.

Substituting equations (55) and (57) into equation (45) one obtains for the cost of the feedwater heater,

$$\dot{C} = \dot{c}_z' \chi + \frac{c_{e,ht} T_o \dot{Q}_{FH}}{T_{FWi}} \left\{ \frac{1 - e^{-\chi}}{\frac{T_{FWe}}{T_{FWi}} - e^{-\chi}} \right\} + \frac{c_{e,ht} T_o Q_{FH}}{T_{FWi}} \left\{ \frac{\ln \frac{T_{FWe}}{T_{FWi}}}{\frac{T_{FWe}}{T_{FWi}} - 1} \right\} \tag{59}$$

For this analysis, it will be assumed that the value of the essergy in the bleed steam condensate can be neglected so that $c_{e,ht}$ is approximately equal to the unit cost of the essergy in the bleed steam $c_{e,B}$. In view of this assumption, one may write,

$$\dot{C} = \dot{c}_z' \chi + \frac{c_{e,B} T_o \dot{Q}_{FH}}{T_{FWi}} \left(\frac{1 - e^{-\chi}}{\frac{T_{FWe}}{T_{FWi}} - e^{-\chi}} \right) + \frac{c_{e,B} T_o \dot{Q}_{FH}}{T_{FWi}} \left(\frac{\ln \frac{T_{FWe}}{T_{FWi}}}{\frac{T_{FWe}}{T_{FWi}} - 1} \right) \tag{60}$$

One can easily see that $\dot{C}$ is a function of both $\dot{c}_z'$ and χ. It was stipulated earlier that $\dot{c}_z'$ is not a function of χ. That is, $\dot{C} = g(\dot{c}_z',\chi)$ with $\dot{c}_z' \neq f(\chi)$. Determining the minimum feedwater heater cost requires that equation (60) be minimized with respect to χ at constant $\dot{c}_z'$ and with respect to $\dot{c}_z'$ at constant χ.

Ordinary differential calculus will allow minimization of $\dot{C}$ with respect to χ at constant $\dot{c}_z'$, since the minimum occurs at $(\partial C/\partial\chi)_{\dot{c}z'} = 0$. Minimization of $\dot{C}$ with respect to $\dot{c}_z'$ at constant χ, however, requires a different approach.

Since the feedwater heater cost equation is of the form $\dot{C} = \dot{c}_z'\chi - bf(\chi) + d$, it is obvious that its derivative with respect to $\dot{c}_z'$ at constant χ is equal to zero at χ equal to zero and is positive for all positive values of χ (note that a negative value for χ is undefined) and therefore cannot be used to minimize the value of $\dot{C}$ in $\{\dot{c}_z'\}$ for constant χ.

One may note that for any given χ (including any given optimum χ), feedwater heater cost $\dot{C}$ increases linearly with $\dot{c}_z'$ so that the smallest $\dot{c}_z'$ will yield the smallest $\dot{C}$. In fact, one can easily see that $\dot{c}_z' = 0$ would yield the smallest value of $\dot{C}$ for any given χ. A value of zero for $\dot{c}_z'$ is not realistic, however, since the various feedwater heater parameters upon which $\dot{c}_z'$ depends require that it have a value greater than zero. Therefore, in order to minimize $\dot{C}$ with respect to $\dot{c}_z'$ at constant χ, it is required that $\dot{c}_z'$ be minimized with respect to the feedwater heater parameters upon which it depends. That is, for a linear area cost system where $\dot{c}_z'$ is not a function of χ (i.e., $\dot{c}_A'$ is not a function of the feedwater heater area A_{FH}), the minimum feedwater cost $\dot{C}_{min}$ occurs at the minimum cost per transfer unit $(\dot{c}_A'/U)_{min}$.

For this analysis, the shellside (condensate velocity) will be considered constant and the cost per transfer unit will be minimized with respect to the tubeside (feedwater) velocity. Recall the expression for $\dot{c}_z'$ given in equation (58),

$$\dot{c}_z' = \dot{m}_{FW} C_{p,FW} (\dot{c}_A'/U) \tag{58}$$

Obviously, minimizing $\dot{c}_A'/U$ will be equivalent to minimizing $\dot{c}_z'$ if $\dot{m}_{FW} C_{p,FW}$ is considered to be a specified constant.

The head loss for flow in cylindrical tubes is given by the Fanning formula,

$$\text{Head Loss} = h_\ell = f \, \frac{L}{R} \, \frac{V^2}{2g_c} \tag{61}$$

where:

f = Fanning friction factor

L = tube length

R = tube radius

V = fluid velocity

gc = gravitational constant, 32.174 ft-lb_m/lb_f-sec^2

The entropy created by head loss may be approximated by the following expression:

$$\frac{\dot{S}_{h\ell}}{\dot{m}} = s = \frac{h_\ell}{T_m} = \frac{f}{T_m}\,\frac{L}{R}\,\frac{V^2}{2g_c} \tag{62}$$

where:

T_m = mean value for the temperature range over which the friction occurs

Recall that mass flow rate in a duct is given by:

$$\dot{m} = \rho V A_c \tag{63}$$

where:

ρ = density

A_c = cross-sectional area available for flow

Substituting equation (63) into equation (62) will give,

$$\dot{S}_{h\ell} = \frac{f}{T_m}\,\rho\frac{L}{R}\,\frac{V^3 A_c}{2g_c} \tag{64}$$

By definition, the hydraulic radius for any conduit is,

$$A_c = RP_w \tag{65}$$

where:

P_w = wetted perimeter

The lateral area available for heat transfer is given by,

$$A = LP_w \tag{66}$$

Eliminating P_w from equation (65) using equation (66) will give,

$$A_c = \frac{AR}{L} \tag{67}$$

Substituting equation (67) into equation (64) gives the following expression for entropy creation due to head loss,

$$\dot{S}_{h\ell} = \frac{\rho f A V^3}{2g_c T_m} \tag{68}$$

Multiplying both sides of equation (68) by T_o/A will yield,

$$\frac{T_o \dot{S}_{h\ell}}{A} = \frac{T_o}{T_m}\,\rho f\,\frac{V^3}{2g_c} \tag{69}$$

Therefore, in view of equation (69), one can easily see that entropy creation due to head loss is a function of fluid velocity alone if fluid properties are assumed to be constant. For the entropy creation due to tubeside head loss one may write,

$$\frac{T_o\dot{S}_{ts}}{A_{FH}} = \frac{T_o}{T_{m,ts}} \rho_{ts} f_{ts} \frac{V_{ts}^3}{2g_c} \tag{70}$$

Similarly, for the entropy creation due to shellside head loss one has,

$$\frac{T_o\dot{S}_{ss}}{A_{FH}} = \frac{T_o}{T_{m,ss}} \rho_{ss} f_{ss} \frac{V_{ss}^3}{2g_c} \tag{71}$$

Substituting equations (70) and (71) into equation (44), the expression for $\dot{c}_A'$, one obtains,

$$\dot{c}_A' = \dot{c}_A + c_{e,ts} \frac{T_o}{T_{m,ts}} \rho_{ts} f_{ts} \frac{V_{ts}^3}{2g_c} + c_{e,ss} \frac{T_o}{T_{m,ss}} \rho_{ss} f_{ss} \frac{V_{ss}^3}{2g_c} \tag{72}$$

Therefore $\dot{c}_A'$ is seen to be a function of feedwater heater unit area cost $\dot{c}_A$ and tubeside and shellside fluid velocities for constant fluid properties and unit costs for essergy dissipation due to fluid friction.

According to Giedt (1957), the experimentally determined expression for the friction factor for flow in smooth tubes is given by

$$f = 0.046\ Re^{-0.2}$$

$$5000 < Re < 200000 \tag{73}$$

Substituting equation (73) into equation (69) will give,

$$\frac{T_o S_{h\ell}}{A} = 0.023 \frac{T_o}{T_m} \frac{\rho}{g_c} V^3 \, Re^{-0.2} \tag{74}$$

The Reynolds number Re is given by,

$$Re = \frac{\rho V d}{\mu} \tag{75}$$

Therefore, equation (74) becomes,

$$\frac{T_o S_{h\ell}}{A} = 0.023 \frac{T_o}{T_m} \frac{\rho^{0.8}}{g_c} \left(\frac{\mu}{d}\right)^{0.2} V^{2.8} \tag{76}$$

Hence, for the cost of essergy dissipation due to tubeside head loss one obtains,

$$c_{e,ts} \frac{T_o \dot{S}_{ts}}{A_{FH}} = K_{V,ts} V_{ts}^{2.8} \tag{77}$$

where:

$$K_{V,ts} = 0.023 \, c_{e,ts} \frac{T_o}{T_{m,ts}} \frac{\rho_{ts}^{0.8}}{g_c} \left(\frac{\mu_{ts}}{d_t}\right)^{0.2} \tag{77a}$$

In view of equation (77), equation (44) becomes,

$$\dot{c}_A' = \dot{c}_A + K_{V,ts} V_{ts}^{2.8} + \frac{c_{e,ss} T_o \dot{S}_{ss}}{A_{FH}} \tag{78}$$

The overall conductance U for the feedwater heater is defined by,

$$\frac{1}{U} = \frac{1}{h_{ts}} + \frac{1}{h_{ss}} + \frac{1}{k/t_w} \tag{79}$$

where:

h_{ts} = convective heat transfer coefficient on the tubeside

h_{ss} = convective heat transfer coefficient on the shellside

k = thermal conductivity of the tube wall

t_w = thickness of tube wall

Note that k/t_w may represent a complex wall

Kreith (1973) gives the following expression for the Nusselt number Nu for turbulent flow in smooth tubes,

$$Nu = 0.023\ Re^{0.8}\ Pr^{0.33} = \frac{hd}{k}\,\cancel{d}_t \tag{80}$$

$$Re > 6000,\ Pr > 0.7$$

All physical properties evaluated at the mean film temperature $(T_w + T_b)/2$.

Substituting the expression for Reynolds number given in equation (75) into equation (80) and rearranging one obtains,

$$h = 0.023\ Pr^{0.33}\ (\frac{\rho V}{\mu})^{0.08}\ kd_t^{-0.2} \qquad (81)$$

Therefore, one may write for the convective heat transfer coefficient on the tubeside,

$$\frac{1}{h_{ts}} = \frac{V_{ts}^{-0.8}}{K_{h,ts}} \qquad (82)$$

where:

$$K_{h,ts} = 0.023\ Pr_{ts}^{0.33}\ (\frac{\rho_{ts}}{\mu_{ts}})^{0.8}\ kd_t^{-0.2} \qquad (82a)$$

Substituting equation (82) into equation (79) and rearranging one obtains,

$$\frac{1}{U} = \frac{V_{ts}^{-0.8}}{K_{h,ts}} + K_U \qquad (83)$$

where:

$$K_U = \frac{1}{h_{ss}} + \frac{1}{k/t_w} \qquad (83a)$$

Multiplying equations (78) and (83) together will yield,

$$\frac{\dot{c}_A'}{U} = [\dot{c}_A + K_{V,ts}V_{ts}^{2.8} + \frac{c_{e,ss}T_o\dot{S}_{ss}}{A_{FH}}][\frac{V_{ts}^{-0.8}}{K_{h,ts}} + K_U] \qquad (84)$$

Since equation (84) is being minimized with respect to the tubeside fluid velocity V_{ts} with the shellside fluid velocity V_{ss} considered constant, one may define,

$$K_A = \dot{c}_A + \frac{c_{e,ss}T_o\dot{S}_{ss}}{A_{FH}} \qquad (85)$$

so that equation (84) becomes,

$$\frac{\dot{c}_A'}{U} = (K_A + K_{V,ts}V_{ts}^{2.8})(\frac{V_{ts}^{-0.8}}{K_{h,ts}} + K_U) \qquad (86)$$

Since the film convective heat transfer coefficient on the shellside is very large due to condensation, one may assume $1/h_{ss} \cong 0$. Since k/t_w is usually large in unfouled tubing one may also assume $\frac{1}{k/t_w} \cong 0$. In view of these assumptions one obtains,

$$K_U \cong 0 \qquad (87)$$

Again since the film convective heat transfer coefficient and the shellside is very large due to condensation, a desired heat transfer rate may be achieved using low fluid velocity. Therefore, the head loss on the shellside will be small and the entropy creation due to head loss on the shellside may be neglected (i.e., $\dot{S}_{ss} \cong 0$ and thus $T_o\dot{S}_{ss} \cong 0$). Equation (85) may now be represented by,

$$K_A \cong \dot{c}_A \tag{88}$$

Substituting equations (87) and (88) into equation (86) one obtains,

$$\frac{\dot{c}_A'}{U} = (\dot{c}_A + K_{V,ts}V_{ts}^{2.8})(\frac{V_{ts}^{-0.8}}{K_{h,ts}}) \tag{89}$$

so that,

$$\dot{c}_A' = \dot{c}_A + K_{V,ts}V_{ts}^{2.8} \tag{90}$$

and

$$\frac{1}{U} = \frac{V_{ts}^{-0.8}}{K_{h,ts}} \tag{91}$$

In order to determine the optimum tubeside velocity which corresponds to the minimum of $\dot{c}_A'/U$ in $\{V_{ts}\}$, differentiate equation (89) with respect to V_{ts}, equate the result to zero and solve for $V_{ts,opt}$ to get,

$$V_{ts,opt} = \left[\frac{0.4\ \dot{c}_A}{K_{V,ts}}\right]^{1/2.8} \tag{92}$$

where:

$$K_{V,ts} = 0.023\ c_{e,ts}\ \frac{T_o}{T_{m,ts}}\ \frac{\rho_{ts}^{0.8}}{g_c}\ \left(\frac{\mu_{ts}}{d_t}\right)^{0.2} \tag{77a}$$

Rearranging equation (92) will give

$$K_{V,ts} V_{ts,opt}^{2.8} = 0.4\ \dot{c}_A \tag{93}$$

In view of equation (93), one obtains for equations (90) and (89),

$$\dot{c}'_{A,opt} = 1.4\ \dot{c}_A \tag{94}$$

and

$$\left(\frac{\dot{c}'_A}{U}\right)_{opt} = \frac{1.4\ c_A V_{ts,opt}^{-0.8}}{K_{h,ts}} \tag{95}$$

where:

$$K_{h,ts} = 0.023 \, Pr_{ts}^{0.33} \left(\frac{\rho_{ts}}{\mu_{ts}}\right)^{0.8} k d_t^{-0.2} \tag{82a}$$

From equations (90), (93) and (94) it is easily seen that the essergy dissipation cost associated with the head loss in the feedwater heater is represented by 40 percent of the unit area cost. Expressions such as equation (94), derived by "brute force" from years of practical experience, are sometimes used as a "rule of thumb" for design purposes.

Once the optimum tubeside velocity for a feedwater heater has been determined it is very easy to obtain the optimum number of tubes of a given diameter. In general, the number of tubes is given by,

$$N_{FH} = \frac{\dot{m}_{FW}}{V_{ts}\rho_{ts}A_t} \tag{96}$$

where A_t is the cross-sectional area of each tube and is given by,

$$A_t = \frac{\pi d_t^2}{4} \tag{97}$$

Thus, the optimum number of feedwater heater tubes N_{opt} of a given diameter for a given feedwater mass flow rate is obtained by,

$$N_{opt} = \frac{4\dot{m}_{FW}}{\pi \rho_{ts} d_t^2 V_{ts,opt}} \tag{98}$$

In view of equation (95), the expression for the optimum cost per transfer unit $(\dot{c}_A'/U)_{opt}$, one may write for equation (58),

$$\dot{c}_{z,opt}' = \frac{1.4\ \dot{c}_A \dot{m}_{FW} C_{p,FW} V_{ts,opt}^{-0.8}}{K_{h,ts}} \tag{99}$$

and for equation (60),

$$\dot{C} = \dot{c}_{z,opt}'\chi - \frac{c_{e,B} T_o \dot{Q}_{FH}}{T_{FWi}} \left[\frac{1 - e^{-\chi}}{\frac{T_{FWe}}{T_{FWi}} - e^{-\chi}} \right] + \frac{c_{e,B} T_o \dot{Q}_{FH}}{T_{FWi}} \left[\frac{\ln \frac{T_{FWe}}{T_{FWi}}}{\frac{T_{FWe}}{T_{FWi}} - 1} \right] \tag{100}$$

One can now determine the optimum number of transfer units which corresponds to minimum feedwater cost $\dot{C}_{min}$ in $\{\chi\}$. As discussed earlier, one needs only to differentiate equation (100) with respect to χ at constant $\dot{c}_z'$, equate to zero and solve for χ_{opt}. Performing these operations will yield the following expression for χ_{opt} (see Appendix J):

$$\chi_{opt} = -\ln \left| \frac{\frac{c_{e,B}T_o\dot{Q}_{FH}}{T_{FWi}}\left(\frac{T_{FWe}}{T_{FWi}} - 1\right) - \frac{T_{FWe}}{T_{FWi}}\left\{\left[\frac{c_{e,B}T_o\dot{Q}_{FH}}{T_{FWe}}\left(\frac{T_{FWe}}{T_{FWi}} - 1\right)\right]^2 + 4\dot{c}'_{z,opt}\left[\frac{c_{e,B}T_o\dot{Q}_{FH}}{T_{FWe}}\left(\frac{T_{FWe}}{T_{FWi}} - 1\right)\right]\right\}^{1/2}}{2\dot{c}'_{z,opt}} + \frac{T_{FWe}}{T_{FWi}} \right| \tag{101}$$

For,

$$\frac{(T_{FWe} - T_{FWi})^2}{T_{FWe}T_{FWi}} < \frac{c_{e,B}T_o\dot{Q}_{FH}}{\dot{c}'_{z,opt}T_{FWe}}\left(\frac{T_{FWe}}{T_{FWi}} - 1\right) < \infty$$

or

$$c'_{e,B} > \frac{1.4\ \dot{c}_A}{T_o U_{opt}}$$

The range of validity for the expression for χ_{opt} has definite physical interpretation with respect to feedwater heater design. Since $\dot{c}'_{z,opt}$ is given by,

$$\dot{c}'_{z,opt} = \frac{1.4\ \dot{c}_A \dot{m}_{FW} C_{p,FW} \nu_{ts,opt}^{-0.8}}{K_{h,ts}} \tag{99}$$

the upper limit for the range of validity is represented by,

$$\frac{c_{e,B}T_o\dot{Q}_{FH}K_{h,ts}}{1.4\ \dot{c}_A T_{FWe}\dot{m}_{FW}C_{p,FW}\nu_{ts,opt}^{-0.8}}\left(\frac{T_{FWe}}{T_{FWi}} - 1\right) \rightarrow \infty$$

This upper limit can be satisfied by the following three cases:

(i) $\frac{T_{FWe}}{T_{FWi}} \to \infty$

(ii) $c_{e,B} \to \infty$

(iii) $\dot{c}_A \to 0$

Note that if any of the above three cases hold, then $\chi_{opt} \to \infty$ also holds (i.e., $A_{opt} \to \infty$ also holds).

Case (i) represents the situation where the feedwater heater outlet temperature is much greater than the feedwater heater inlet temperature. It is very easy to see that if this case holds, a very large optimum heat transfer area will be required (i.e., a large number of transfer units will be required).

Case (ii) represents the situation where the essergy in the bleed steam to the feedwater heater has a very large unit cost. If this case holds, then a large optimum heat transfer area will be required to minimize the amount of essergy that is dissipated (i.e., minimize the entropy creation) and hence minimize feedwater heater cost.

Case (iii) represents the situation where the unit area cost for the feedwater heater is very small. If heat transfer area is very cheap, then optimum heat transfer area can be very large in order to again minimize essergy dissipation and thus minimize feedwater heater cost.

The lower limit of validity for the expression for χ_{opt} requires $c_{e,B} > 1.4\ \dot{c}_A/T_oU_{opt}$. The essergy in the bleed steam fed to the feedwater heater may take on any value depending on the particular power cycle that it is calculated for. Therefore, a lower limit on the value of $c_{e,B}$ does not make sense unless one considers the nature of the feedwater heater cost equation. The feedwater heater cost equation is of the form $\dot{C} = a\chi - bf(\chi) + d$ where $a\chi$ represents the capital cost and $bf(\chi)+d$ represents the essergy dissipation cost. For the case where $c_{e,B} \leq 1.4\ \dot{c}_A/T_oU_{opt}$, the slope of the capital cost term is greater than the slope of the essergy dissipation cost term for all $\chi \leq 0$, so that the slope of the feedwater heater cost equation is always increasing for all $\chi \leq 0$ (see Figure 12). For this case, the mathematical minimum occurs at $\chi = 0$ (since $\chi<0$ is not defined). Since $\chi = 0$ corresponds to an infinite condensing temperature for the bleed steam, $T_{C,B} \to \infty$, this minimum cannot hold for realistic power plants. For actual feedwater heater design, the condensing temperature of the bleed steam $T_{C,B}$ should be set at some maximum possible value $T_{C,max}$ which corresponds to some $\chi_{opt} > 0$ (see Figure 12).

Once the optimum number of transfer units χ_{opt} for a feedwater heater has been determined it is very easy to obtain the optimum heat transfer area. Since minimization of feedwater heater cost with respect to $\dot{c}_z'$ leads to an optimum number of tubes, determining optimum heat transfer

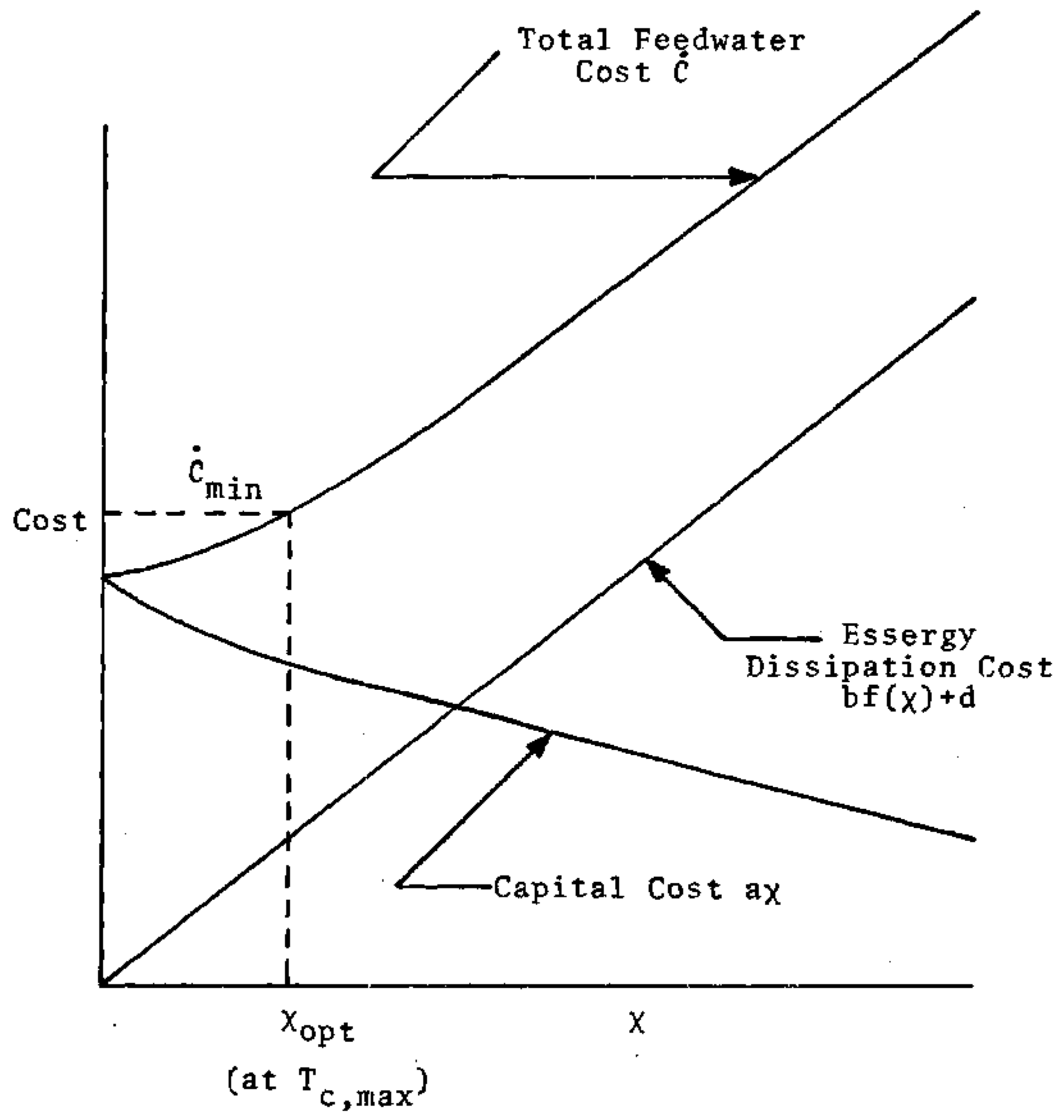


Figure 12. Feedwater Heater Cost Which Illustrates the Lower Limit of Validity for χ_{opt}

area corresponding to the optimum number of tubes will be equivalent to obtaining the optimum length of these tubes. In general, the total area available for heat transfer may be found by,

$$A_{FH} = \frac{\dot{m}_{FW} C_{p,FW} \chi}{U} \tag{56}$$

Therefore, the optimum heat transfer area for a feedwater heater A_{opt} corresponding to an optimum tubeside velocity $V_{ts,opt}$ (i.e., optimum number of tubes N_{opt}) is obtained by:

$$A_{opt} = \frac{\dot{m}_{FW} C_{p,FW} \chi_{opt}}{U_{opt}} \tag{102}$$

It was determined earlier that,

$$\frac{1}{U} = \frac{V_{ts}^{-0.8}}{K_{h,ts}} \tag{91}$$

In view of equation (91), the optimum overall heat transfer coefficient is given by,

$$U_{opt} = K_{h,ts} V_{ts,opt}^{0.8} \tag{103}$$

Substituting equations (101) and (103) into equation (102) will give for the optimum heat transfer area for a feedwater heater A_{opt},

$$A_{opt} = \frac{-\dot{m}_{FW} C_{p,FW}}{K_{h,ts} v_{ts,opt}^{0.8}} \ln \left| \frac{\frac{c_{e,B} T_o \dot{Q}_{FH}}{T_{FWi}} \left(\frac{T_{FWe}}{T_{FWi}} - 1\right) - \frac{T_{FWe}}{T_{FWi}} - \left\{ \left[\frac{c_{e,B} T_o \dot{Q}_{FH}}{T_{FWe}} \left(\frac{T_{FWe}}{T_{FWi}} - 1\right)\right]^2 + 4\dot{c}'_{t,opt} \left[\frac{c_{e,B} T_o \dot{Q}_{FH}}{T_{FWe}} \left(\frac{T_{FWe}}{T_{FWi}} - 1\right)\right]\right\}^{\frac{1}{2}}}{2\dot{c}'_{t,opt}} + \frac{T_{FWe}}{T_{FWi}} \right| \tag{104}$$

Once the optimum heat transfer area for an optimum number of tubes is known, the optimum length for the tubes L_{opt}, can be found by,

$$L_{opt} = \frac{(1-F_{ea})A_{opt}}{\pi d_t N_{opt}} \tag{105}$$

$$0 \le F_{ea} < 1.0$$

where, for want of better words, F_{ea} will be known as the extended area factor. This factor represents that fraction of the total heat transfer area which is supplied by extended area surface.[18] Obviously, a value of zero for F_{ea} corresponds to the absence of any extended area surface while increasing values for F_{ea} correspond to increasing amounts of extended

[18] Fins are not presently used in feedwater heaters, but as the cost of energy continues to rise it may become necessary to consider their inclusion in designs.

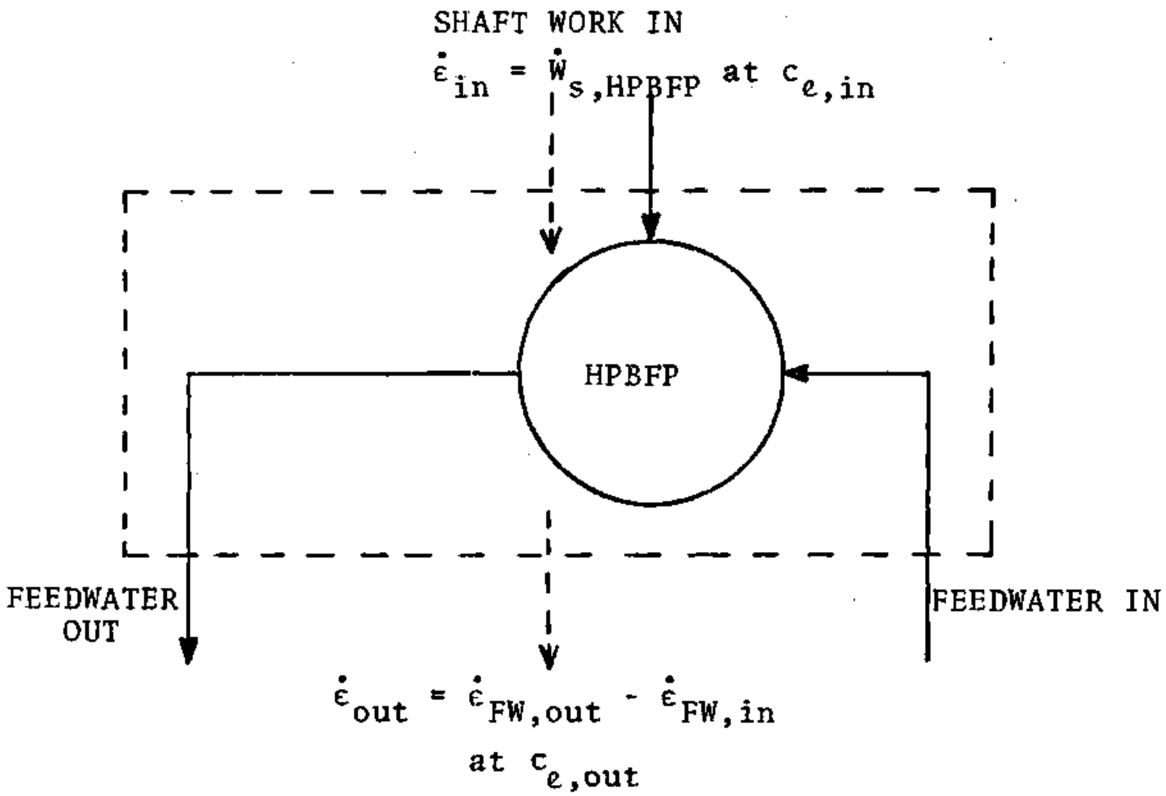
area surface.

The use of equations (98) and (105) determines two important feedwater heater design parameters, namely, optimum number and length of given diameter tubes for constant tubeside mass flowrate and shellside fluid velocity.

B. Application of the Design Optimization Equations

In the previous section the equations for determining the optimum velocity and optimum number of transfer units for a feedwater heater were developed. These equations were in turn used to develop expressions for determining optimum number of tubes and optimum heat transfer area (i.e., length of tubes). In this section, the utility of these expressions will be illustrated by applying them to a design of a feedwater heater which has the same operating conditions as heater number 6 of the power plant used earlier in this study.

Before the design equations may be used, it is first necessary that the unit cost at which essergy is being dissipated by the tubeside head loss $c_{\varepsilon,ts}$ be determined. This unit essergy cost can be determined by treating the high pressure boiler feed pump as a simple essergy consumption system (see Figure 13). Rearrangement of equation (107) will yield the following expression for the unit essergy cost at which head is delivered to the feedwater mass stream by the high pressure boiler feed pump:



Economic Balance:

$$c_{e,in}\dot{\varepsilon}_{in} + \dot{C}_{CAP} = c_{e,out}\dot{\varepsilon}_{out} \tag{106}$$

$$c_{e,s}\dot{W}_{s,HPBFP} + \dot{C}_{HPBFP} = c_{e,out}\,(\dot{\varepsilon}_{FW,out} - \dot{\varepsilon}_{FW,in}) \tag{107}$$

Figure 13. High Pressure Boiler Feed Pump Considered as a Simple Essergy Consumption System

$$c_{e,out} = \frac{c_{e,s}\dot{W}_{s,HPBFP} + \dot{C}_{HPBFP}}{\dot{\varepsilon}_{FW,out} - \dot{\varepsilon}_{FW,in}} \tag{108}$$

Essergy dissipation due to friction loss (head loss) at all points between the high pressure boiler feed pump outlet and the high pressure turbine will have the same unit cost as the unit cost of the essergy transmitted by the high pressure boiler feed pump to the feedwater mass stream in producing its head increase. Therefore, one obtains the following expression for the unit cost at which essergy is being dissipated by tubeside head loss in the feedwater heater:

$$c_{e,ts} = c_{e,out} = \frac{c_{e,s}\dot{W}_{s,HPBFP} + \dot{C}_{HPBFP}}{\dot{\varepsilon}_{FW,out} - \dot{\varepsilon}_{FW,in}} \tag{109}$$

Three design plots were generated for the design of the hypothetical feedwater heater using equations (89) and (100) for the following constraints:

$$\dot{c}_A = 2.00 \frac{\$}{\text{yr-ft}^2}$$

$$c_{e,B} = 2.2732 \text{ \$/million Btu}$$

$$c_{e,s} = 2.9765 \text{ \$/million Btu}$$

$$c_{e,ts} = 4.0623 \text{ \$/million Btu}$$

$$T_{FWi} = 785.1°R$$

$$T_{FWe} = 845.0°R$$

$$h_{FWi} = 300.9 \text{ Btu/lb}$$

$$h_{FWe} = 362.8 \text{ Btu/lb}$$

$$\dot{W}_{s,HPBFP} = 22.610 \text{ milltion Btu/hr}$$

$$\dot{C}_{HPBFP} = 6.625 \text{ \$/hr}$$

These constraints were obtained either from the power cycle design operating conditions or from the economic analysis performed in Section G of Chapter III.

Figure 14 is a plot of feedwater velocity against cost per transfer unit $\dot{c}_A'/U$, Figure 15 is a plot of feedwater velocity V_{ts} against feedwater heater cost $\dot{C}$ and Figure 16 is a plot of number of transfer units χ against feedwater heater cost $\dot{C}$. Equations (95), (92) and (101) were used to determine the values for optimum cost per transfer unit, optimum tubeside velocity and optimum number of transfer units, respectively, which are listed below.

$$(\dot{c}_A/U)_{opt} = 6.5931 \times 10^{-8} \frac{\$F}{Btu}$$

$$V_{ts,opt} = 13.855 \text{ ft/sec}$$

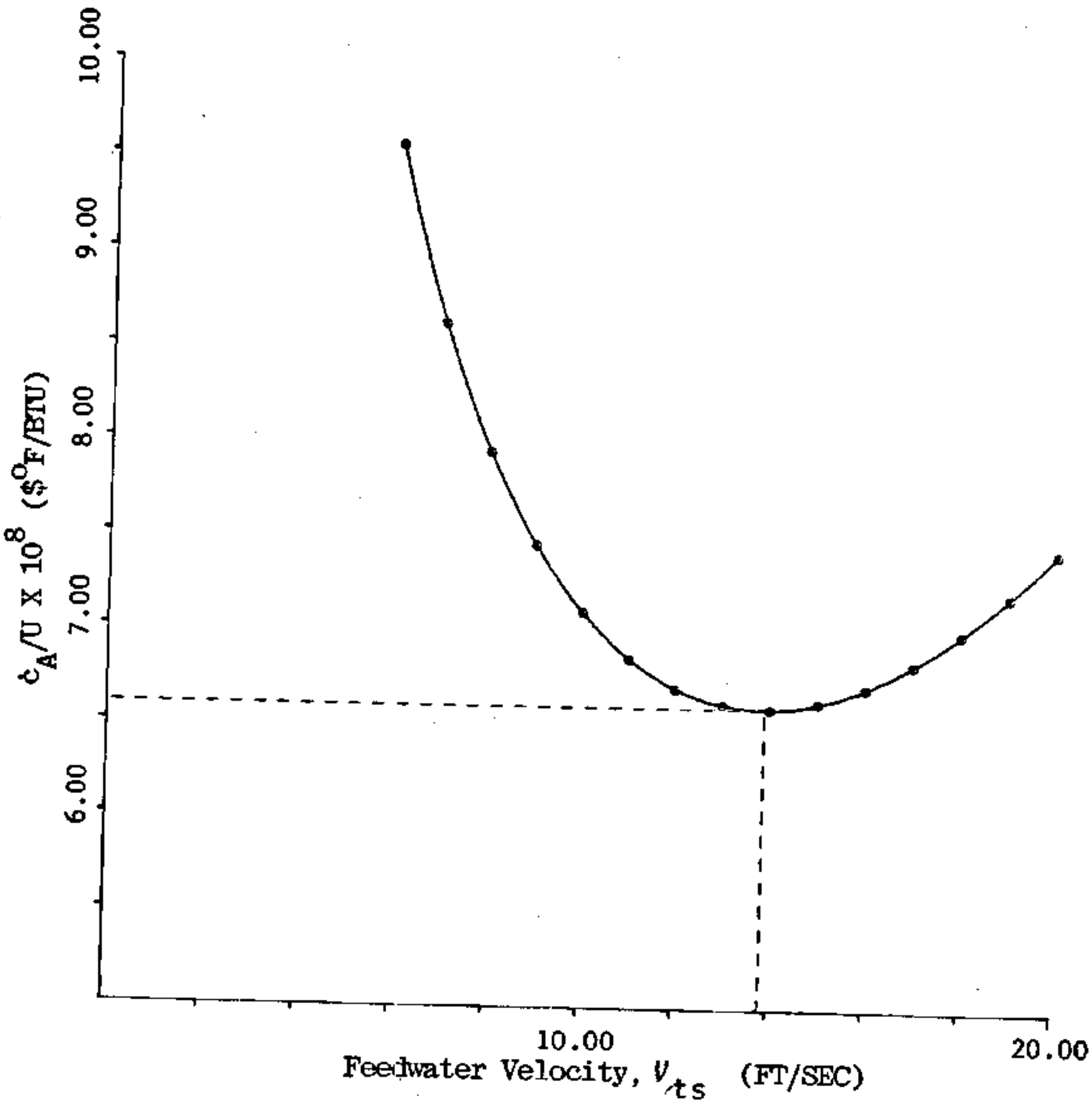


Figure 14. Plot of Velocity Against Cost per Transfer Unit

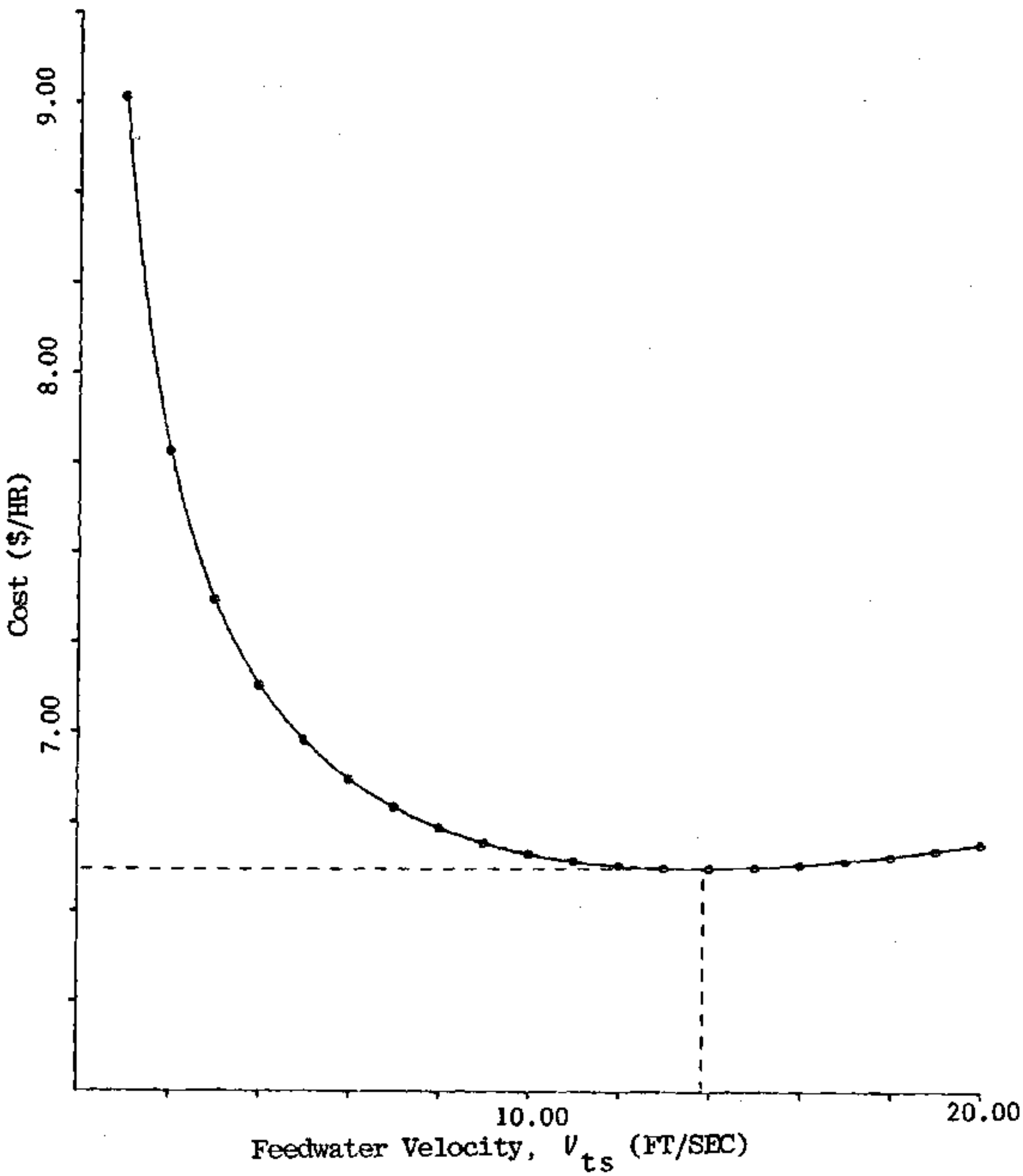


Figure 15. Plot of Feedwater Velocity Against Feedwater Heater Cost

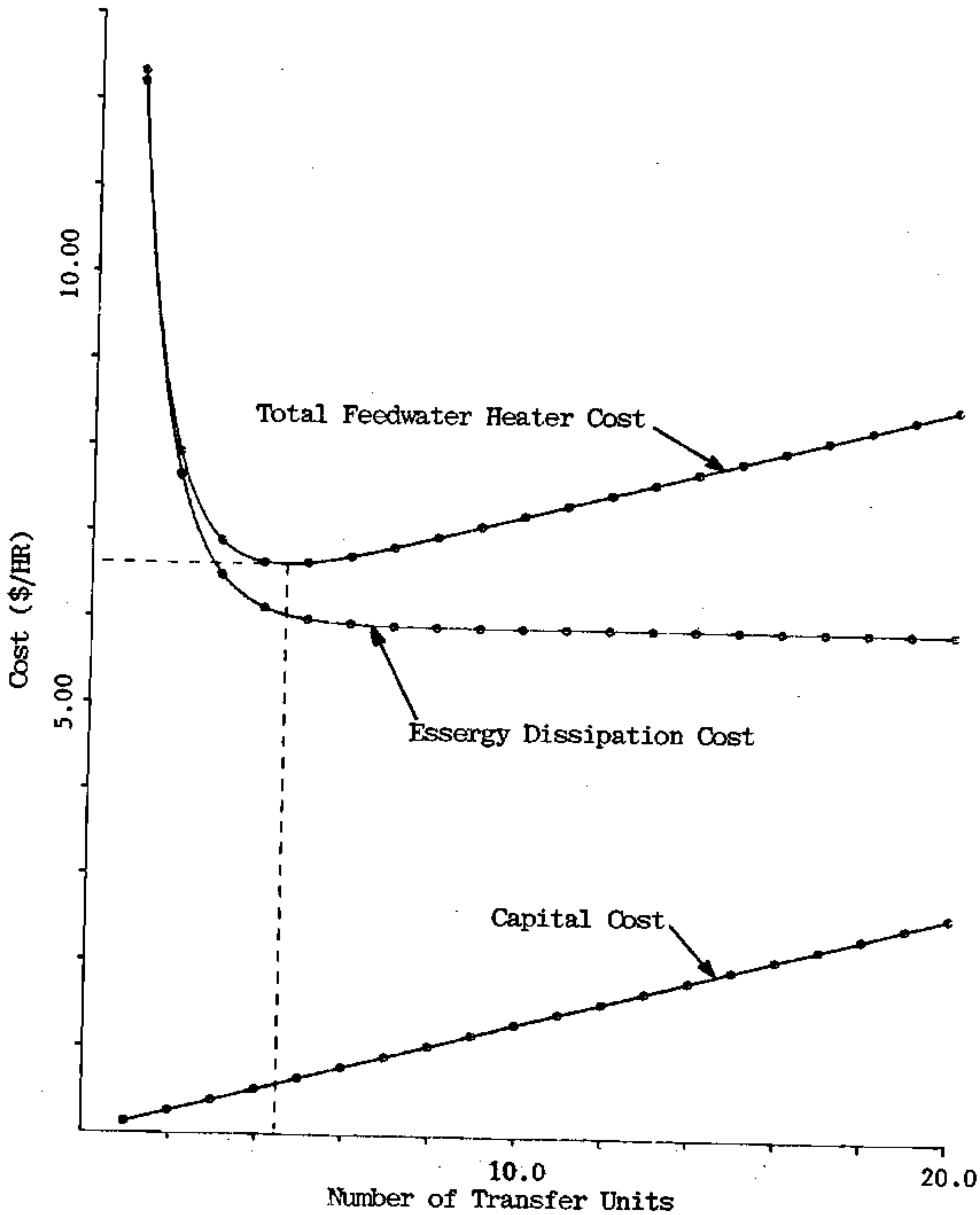


Figure 16. Plot of Number of Transfer Units Against Feedwater Heater Cost

$$\chi_{opt} = 4.481$$

$$\dot{C}_{min} = \$6.620/hr$$

One may easily see by examination of these three plots that the minimum feedwater cost occurs at minimum cost per transfer unit.

C. Generalizing Feedwater Heater Design

The results of Section A may be used to formulate a general set of design plots for the optimum number of tubes and the optimum heat transfer area (i.e., length of tubes). Examination of equations (77a), (92) and (98) reveals that for a given feedwater mass flowrate and feedwater heater tube diameter, the optimum number of feedwater heater tubes N_{opt} is function only of the ratios $T_{m,ts}/T_o$ and $\dot{c}_A/c_{e,ts}$. Figure 17 presents a plot of N_{opt} against the ratio $\dot{c}_A/c_{e,ts}$ for various values of the ratio $T_{m,ts}/T_o$ (with $d_t = 0.0625$ ft and $\dot{m}_{FW} = 1.869086 \times 10^6$ lbs/hr). Note that this plot accurately predicts the value of N_{opt} calculated by the equation (98) for the hypothetical feedwater heater studied in Section B of this chapter.

$$T_m/T_o = 1.6$$

$$\dot{c}_A/c_{e,ts} = 0.492 \ \frac{\text{million Btu}}{\text{yr-ft}^2}$$

$$N_{opt} = 221$$

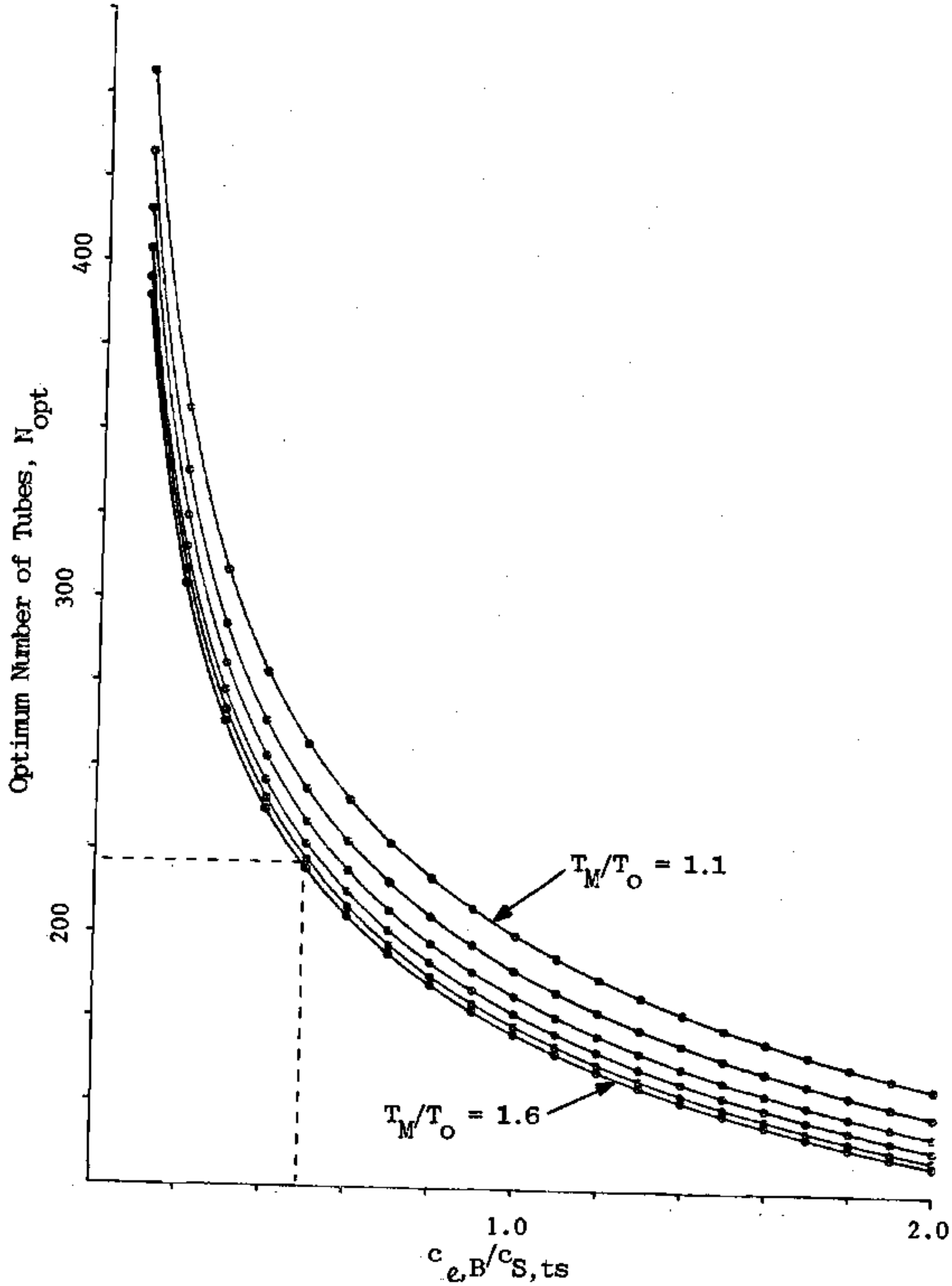


Figure 17. Generalized Plot of Optimum Number of Tubes

Examination of equations (77a), (82a), (92), (99) and (104) reveals that for a given power plant (i.e., $c_{e,s}$ is given) in a given environment (i.e., T_o is given) with specified feedwater mass flow rates and feedwater heater tube diameters and entrance and exit temperatures, the optimum heat transfer area A_{opt} is a function only of the unit cost of the essergy in the bleed steam $c_{e,B}$ and the feedwater heater unit area cost $\dot{c}_A$. Figure 18 presents a plot of A_{opt} against $c_{e,B}$ for various values of $\dot{c}_A$ with $c_{e,s}$ = 2.9765 \$/million Btu, T_o = 510.1°R, $\dot{m}_{FW}$ = 1.869086 x 10^6 lbs/hr, T_{FWi} = 325.4°F, T_{FWe} = 385.3°F and d_t = 0.0625 ft. Note that this plot accurately predicts the value for A_{opt} calculated by equation (104) for the hypothetical feedwater heater studied in Section B of this chapter.

$$\dot{c}_A = 2.00 \frac{\$}{\text{yr-ft}^2}$$

$$c_{e,B} = 2.2732 \frac{\$}{\text{million Btu}}$$

$$A_{opt} = 1825 \text{ ft}^2$$

$$L_{opt} = 42.1 \text{ ft } (F_{ea} = 0)$$

Use of Figure 18 as a design plot is limited in that it is valid only for a feedwater heater with the exact same operating conditions as the hypothetical feedwater heater. Figure 17 has a wider utility in that it is valid for any

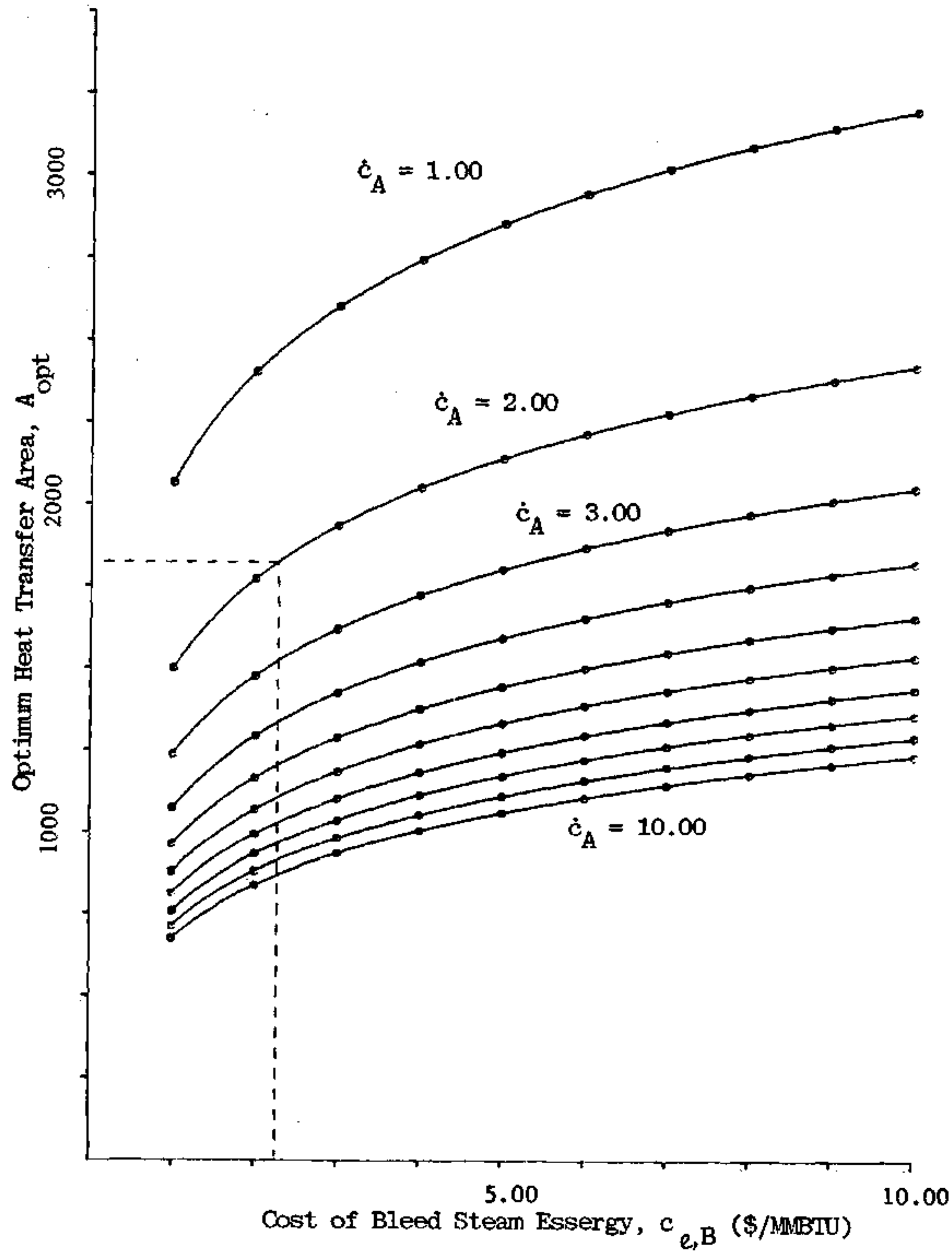


Figure 18. Generalized Plot of Optimum Heat Transfer Area

feedwater heater in any plant for the given feedwater mass flow rate and tube diameter.

CHAPTER VI

CONCLUSIONS

This study demonstrates an effective method for economic analysis of the value of flows of the commodity which the modern day power plant transforms and consumes (dissipates)--that commodity being essergy and not energy. This method allows development of unit economic values for essergy which permit a correct reflection of the relative monetary value of the essergy flows at various points in power cycle but are independent of the corresponding essergy balances. In other words, the analysis allows calculation of unit costs for essergy flows at various points in power cycle which are independent of the specific essergy at the points.

The importance of developing unit costs for essergy that are linear with changes in the essergy flow (i.e., independent of the amount of essergy flowing), is that it is necessary that these unit costs be calculated only one time during the life of the power plant. Once these unit essergy costs have been determined for the various junctures in the power cycle operating at design conditions, they can be used without recalculation for operational analysis throughout the life of the power plant regardless of any changes in internal essergy flows.

The solution of an actual power plant problem in this study by essergy analysis is a sound demonstration of its practicality. Once a power cycle has been analyzed by the methods presented in this study, any operational problem requiring assessment of the relative monetary value of various essergy flows to and from a zone may be solved in a manner similar to the solution of the feedwater heater problem.

Essergy analysis works equally well for design purposes as it does for solving plant operating problems by isolating areas within the cycle (decentralization) which are in need of design improvement--i.e., for design optimization. In this study the feedwater heater was modeled as a simple essergy consumption system where its total cost is made up of the sum of capital cost and essergy dissipation cost. Using fundamental and well-known expressions which describe the momentum and heat transfer processes that are occuring within the feedwater heater along with a known capital cost relation, a total cost equation in terms of basic operating and design parameters was developed.[19] For

[19]These parameters include unit heat transfer area cost, heat transfer area, heat transfer tube diameter, bleed steam unit essergy cost, fluid friction unit essergy cost, feedwater mass flow rate, feedwater velocity, feedwater inlet and exit temperatures, feedwater physical properties and datum state (environment) temperature, etc.

the purpose of demonstrating the optimization method, the cost equation was minimized by use of ordinary differential calculus to obtain expressions for optimum feedwater velocity (with bleed steam velocity considered constant) and optimum heat transfer area for a feedwater heater. This analysis is equivalent to obtaining the optimum number and length of heat transfer tubes for a feedwater heater. The optimization need not be restricted to just these parameters, but may be extended to include all important feedwater heater design parameters.

The practical utility of the design equations that were developed was demonstrated by optimizing the number and length of tubes for a feedwater heater which has the same operating conditions as feedwater heater number 6 of the same power cycle examined earlier in this study by essergy analysis. Then, the design-equations were generalized to some extent for application to the design of certain other feedwater heaters within the same or different (but similar) power cycle. The generalization lead to three dimensional design plots of N_{opt} against $\dot{c}_A/c_{e,s}$ for various values of T_M/T_o and A_{opt} against $c_{e,B}$ for various values of $\dot{c}_A$.[20]

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[20]The restriction of the use of the generalized design plots for the other feedwater heaters is that certain of their operating conditions must be the same or nearly the same as those of feedwater heater number 6.

In the same manner that essergy analysis may be used to solve many plant operating problems, so may the design optimization be extended to apply to any other zone in the power plant. One needs only to model the total cost of the zone in question as a sum of capital cost and essergy dissipation cost, relate this equation to the zone parameters by known economic or physical expressions, and proceed with the optimization.

The essergy analysis methods developed within this study have been proven to be effective for solving power plant operating problems and design optimization. These methods are more reliable than first law analysis and time-honored "rules-of-thumb." These methods have also been shown to be more accurate than an earlier performed second law analysis of a power cycle. In summary, the second law or essergy analysis methods contained within this report provide powerful and useful fundamental tools for the practicing power plant engineer.

CHAPTER VII

RECOMMENDATIONS

The analysis presented in this thesis points to several areas which are in need of additional study.

For example, one might study the effect of changes (with time) of turbine system effectiveness, power cycle effectiveness or turbine system work output on the value of $c_{e,TS}$, the linearized unit cost of the essergy inputs and outputs to the turbine system. Once it has been determined what effect changes in the above parameters have on the value of $c_{e,TS}$, the investigation may be broadened to cover how economic analyses that utilize the value of $c_{e,TS}$ might be affected.

Another area for further investigation involves developing a method for linearizing the system containing the last low pressure turbine stage and the condenser. This investigation would probably require a more high-powered mathematical treatment in the form of LaGranges Method of Undetermined Multipliers to determine unit essergy costs.[21]

There are certainly many other devices in the power cycle besides the feedwater heaters and condenser that could

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[21] Demonstrated by El-Sayed and Evans (1970).

be analyzed by using an approach similar to the one used in this study. The analysis of some of these devices such as the economizer or the stack air preheater would be more complex than that of the feedwater heater since the entropy creation (and consequently essergy dissipation) within them is not independent of all other zones. Analysis of these type devices would almost certainly require treatment by LaGranges Method for formulating the internal economy.

Determination of expressions based on zone design parameters which properly allocate capital costs to the zones would be an interesting area of study.[22] For example, equation (22) which was used for allocating turbine stage capital costs worked adequately but could certainly use some refinement.

Additional design optimization study might be performed for the feedwater heaters. Expressions for optimizing other feedwater heater parameters such as tube diameter, materials of construction, etc., could be developed. Extension of these optimization expressions to formulate generalized design plots suitable for analyzing a wide variety of feedwater heater operating conditions would prove valuable.

Finally, other devices within the power cycle could in turn be optimized with respect to their various operating

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[22] Zone design parameters might include operating temperatures, effectiveness, mass flow rates, etc.

parameters. In this manner, a comprehensive text for efficient and accurate design of all devices within the power cycle could be developed.

APPENDICES

APPENDIX A

POWER PLANT DATA

This appendix presents thermodynamic property and flow data for the power cycle under consideration in this study for various operating modes. Most of the information presented in Figure A-2 and Tables A-1, A-2, and A-3 was taken from Gaggioli, et al. (1975) and Fehring and Gaggioli (1977). Data not taken directly from the references were calculated by the following methods.

Steam and Water Flow Rates

With the exception of the feedwater flow rate to the boiler which is 1,869,086 lb/hr at design conditions, all other major flow rates were calculated using energy balances. For example, consider the steam and feedwater flow rates to feedwater heater number 5 (refer to Figures A-1 and A-2).

$$\dot{m}_{FW5} \Delta h_{FW5} = \dot{m}_{H5} \Delta h_{H5}$$

$$\dot{m}_{H5} = \dot{m}_{FW5} \frac{\Delta h_{FW5}}{\Delta h_{H5}} = 0.042355\ \dot{m}_{FW5} \qquad \text{(A-1)}$$

$$\dot{m}_{FW5} = \dot{m}_{FW7} - \dot{m}_{H7} - \dot{m}_{H6} - \dot{m}_{H5}$$

$$\dot{m}_{FW5} = 1,595,105 - \dot{m}_{H5} \qquad \text{(A-2)}$$

Solving equations (A-1) and (A-2) simultaneously we get:

$$\dot{m}_{H5} = 64,815 \text{ lb/hr}$$

$$\dot{m}_{FW5} = 1,530,290 \text{ lb/hr}$$

Steam Heated Combustion Air Preheater Exit Temperatures

Gaggioli, et al. (1975) gives the mass flow rate of the combustion air as 2.6×10^6 lb/hr and the essergy gained by the combustion air as 5.54×10^6 Btu/hr for an environment temperature of 500°F. The essergy increase of the combustion air, as it passes through air steam preheater number 1, may be calculated by the following expression (see Appendix E):

$$\dot{\varepsilon}_{A1} = \dot{m}_{A1} \; [C_{p,m} \; (T_{A1} - T_{A0}) - C_{p,m} \; T_o \; \ln(T_{A1}/T_{A0})]$$

where:

$C_{p,m}$ = specific heat of the fluid (air) evaluated at the mean temperature between T_{A1} and T_{A0}.

$$5.54\text{x}10^6 = 2.6\text{x}10^6[0.24(T_{A1}-500)-0.24(500)\ln(T_{A1}/500)]$$

$$508.872 = T_{A1}-500\ \ln(T_{A1}/500)$$

Solving the above equation by trial and error we obtain:

$$T_{A1} = 600°R$$

In a similar manner, the exit temperature of the combustion air leaving steam air preheater number 2 may be calculated as

$$T_{A2} = 661°R$$

For lack of better information on the power cycle being analyzed in this study, it will be assumed that the combustion air flow rate and temperatures at the steam air preheater exits are the same in an environment at 510.1°R as they are in an environment at 500°R.

Fuel Flow Rate at Design Conditions

Gaggioli, et al. (1975) gives the thermal efficiency of the boiler, furnace, economizer and **stack** air preheater combined as 0.916. Thermal efficiency is defined by the following expressions:

$$\eta_{I,FB} = \frac{\dot{Q}_T + \dot{Q}_{RN}}{\dot{m}_F h_F + \dot{m}_{A2} h_{A2}}$$

Thus,

$$\eta_{I,FB} = \frac{\dot{m}_T (h_T - h_{FW}) + \dot{m}_{RH} (h_{RH} - h_{B7})}{\dot{m}_F h_F + \dot{m}_{A2} C_{p,m} (T_{A2} - T_0)}$$

Assuming that the combustion air temperature entering the furnace is 661°R for an environment temperature of 510.1°R (see previous section) and the heating value of the fuel is 11875 Btu/lb, we have, from Fehring and Gaggioli (1977),

$$0.916 = \frac{1869086\ (1493.8-455.6) + 1640196\ (1520.1-1329.1)}{\dot{m}_F (11875) + 2600000\ (0.24)(661-510.1)}$$

$$\dot{m}_F = 199{,}266 \text{ lb/hr}$$

Additional Fuel Flow Necessary in Case C

The additional fuel flow required in Case C, due to the feedwater temperature entering the economizer being below its design operating level, may be calculated as follows:

Additional energy requirement for Case C = $\dot{m}_{FW}(h_{FW,A}-h_{FW,C})$

$= 1{,}869{,}086\ (455.6-442.9)$

$= 23{,}737{,}392$ Btu/hr

Additional fuel energy requirement for Case C

$= 23{,}737{,}392/0.916$

$= 25{,}914{,}183$ Btu/hr

Thus, the additional fuel flow required is

$$\dot{m}_{F,ADD} = 25{,}914{,}183/11875 = 2182 \text{ lb/hr}$$

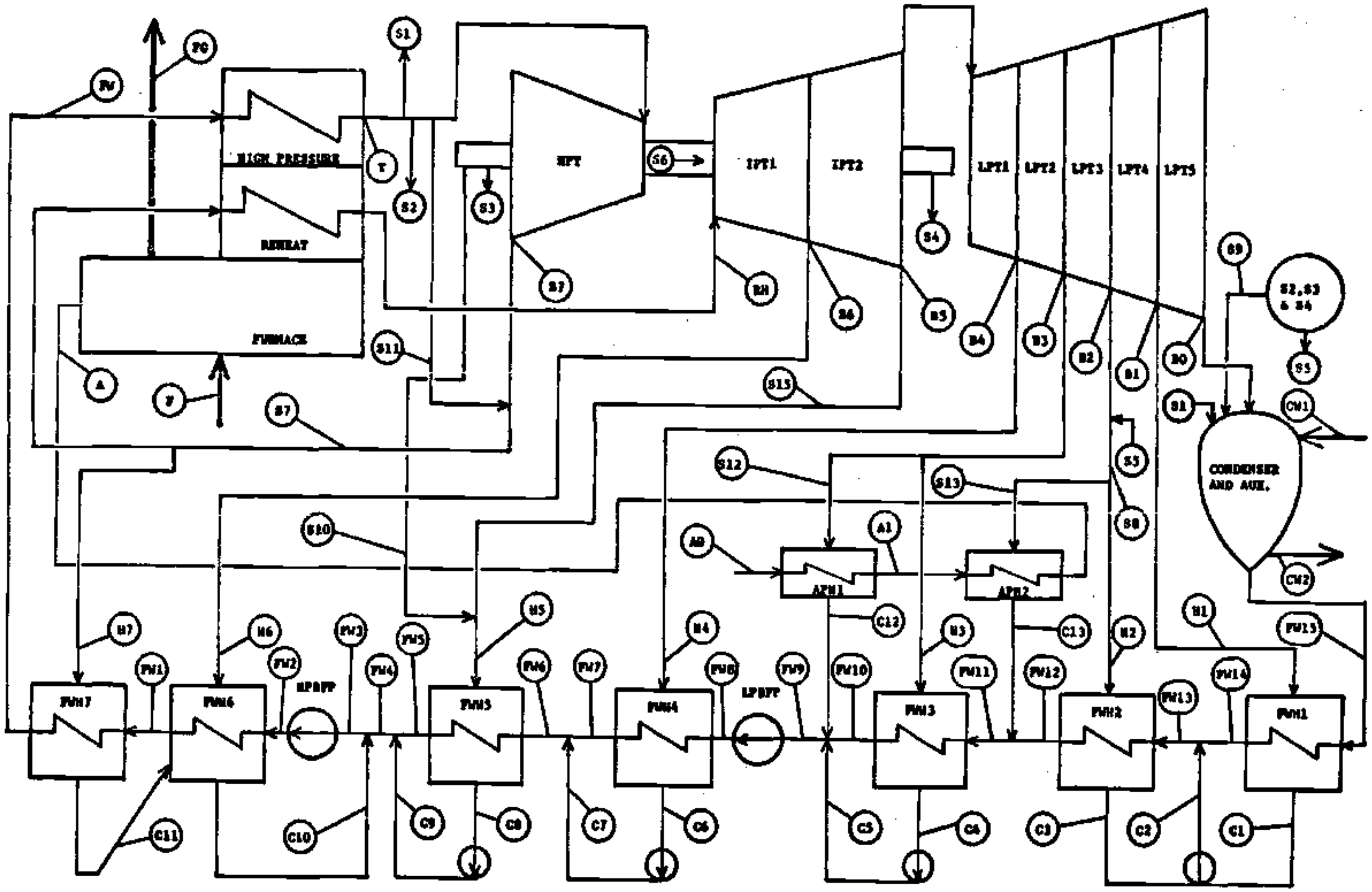

Figure A-1. Schematic of the Power Plant Depicting Reference Points

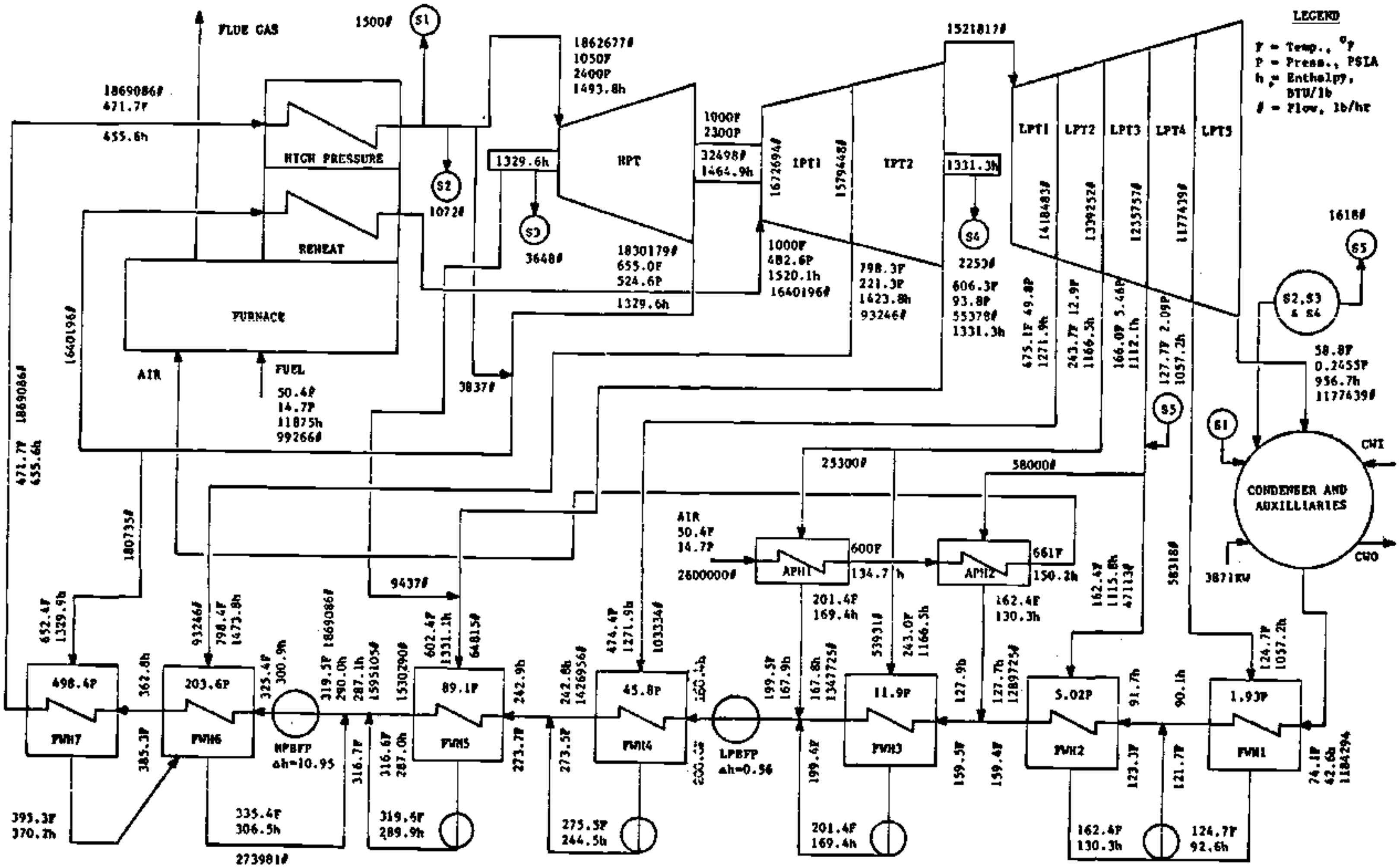

Figure A-2. Schematic of Power Plant Depicting Design Conditions

Table A-1. Properties* and Flow Rates at Various Points in the Power Plant at Design Conditions (Case A)

Point+	Pressure psia	Temperature °F	Enthalpy Btu/lb	Entropy Btu/lb°F	Flow Rate lb/hr
T	2414.7	1050	1493.8	1.555	1869086
B7	524.6	655.0	1329.6	1.5824	1830179
RH	482.6	1000	1520.1	1.742	1640196
S6	2300	1000	1464.9	1.5399	32498
B6	221.3	798.3	1423.8	1.7539	1672694
B5	93.8	603.6	1331.3	1.7671	1579448
B4	49.8	475.1	1271.9	1.7766	1521817
B3	12.9	243.7	1166.5	1.7942	1418483
B2	5.46	166.0	1112.1	1.8041	1339252
B1	2.09	127.7	1057.2	1.8149	1235757
B0	0.2455	58.8	956.7	1.8463	1177439

*Reference states: For H_2O, liquid at 32°F and 1.0 atm. For fuel, components at complete equilibrium in the ambient environment at 510.1°R and 1.0 atm.

+Point refers to the point in the cycle as defined by Figure A-1.

Table A-1 (cont.)

Point	Pressure psia	Temperature °F	Enthalpy Btu/lb	Entropy Btu/lb°F	Flow Rate lb/hr
H7	498.4	652.4	1329.9	1.5878	180735
H6	203.6	798.4	1473.8	1.7628	93246
H5	89.1	602.4	1331.1	1.7724	64815
H4	45.8	474.4	1271.9	1.7855	103334
H3	11.9	243.0	1166.5	1.803	53931
H2	5.02	162.4	1115.8	1.8190	47113
H1	1.93	124.7	1057.2	1.8231	58318
S9	510.0	692.6	1352.2	1.6053	5355
FW15	Saturated Water	74.1	42.6	0.0822	1184294
FW14	Saturated Water	121.7	90.1	0.1673	1184294
FW13	Saturated Water	123.3	91.7	0.1701	1289725
FW12	Saturated Water	159.4	127.7	0.2301	1289725
FW11	Saturated Water	159.5	127.9	0.2303	1347725
FW10	Saturated Water	199.4	167.8	0.2929	1347725
FW9	Saturated Water	199.5	167.9	0.2931	1426956
FW8	215	200.0	168.4	0.2936	1426956
FW7	215	273.5	242.8	0.4005	1426956
FW6	215	273.7	242.9	0.4006	1530290
FW5	215	316.6	287.0	0.4590	1530290
FW4	215	316.7	287.1	0.4591	1595105

Table A-1 (concluded)

Point	Pressure psia	Temperature °F	Enthalpy Btu/lb	Entropy Btu/lb°F	Flow Rate lb/hr
FW3	215	319.5	290.0	0.4628	1869086
FW2	2950	325.4	300.9	0.4654	1869086
FW1	2950	385.3	362.8	0.5414	1869086
FW	2950	471.7	455.6	0.6458	1869086
S6	2300	1000	1464.9	1.5399	32498
S13	11.9	243.0	1166.5	1.803	25300
S14	5.02	162.4	1115.8	1.819	58000
C1	Saturated Water	124.7	92.6	0.1726	58318
C3	Saturated Water	162.4	130.3	0.2350	47113
C5	Saturated Water	201.4	169.4	0.2960	53931
C7	Saturated Water	275.5	244.5	0.4034	10334
C9	Saturated Water	319.6	289.9	0.4632	64815
C10	Saturated Water	335.4	306.5	0.4841	273981
C11	Saturated Water	395.3	370.2	0.5909	180735
C12	Saturated Water	201.4	169.4	0.2960	25300
C13	Saturated Water	162.4	130.3	0.2350	58000
F	14.7	50.4	11875	-1.0445	199266
H_2O Datum State	14.7	50.4	18.5	0.0369	--

Table A-2. Change* in Properties at Various Points in the Power Plant Due to Deterioration of Feedwater Heater Number 5

Point+	Temperature, °F		Enthalpy, Btu/lb	
	Case B	Case C	Case B	Case C
FW		460.0		442.9
FW1		370.0		347.0
FW2	321.0	285.0	296.4	259.9
FW5	311.6	273.7	282.3	242.9

*Data for points whose properties did not change from design conditions are omitted for clarity.

+Point refers to the point in the cycle as defined by Figure A-1.

Table A-3. Change* in Flow Rate at Various Points in the Power Plant Due to Deterioration of Feedwater Heater Number 5

Point**	Flow Rate, lb/hr Case B	Flow Rate, lb/hr Case C
H7		186772
H6	100774	135058
H5	57864	0
H4		104480
Fuel Increase+		2182

*Data for points whose flow rates did not change from design conditions are omitted for clarity.

**Point refers to the point in the cycle as defined by Figure A-1.

+Additional boiler fuel is needed in Case C to take the feedwater from its depressed temperature (460°F) to design operating temperature (471.7°F).

APPENDIX B

STEADY FLOW ESSERGY

For the case of hydrodynamic flow of material across a stationary boundary (i.e., material diffusional flow is excluded), the essergy which is flowing may be obtained by differentiating equation (1) to obtain,

$$d\varepsilon = dE - T_o dS - \sum_c \mu_{co} dN_c \qquad \text{(B-1)}$$

noting that dv = 0 when the only effect upon a system is the flow of material across a stationary boundary. The First Law yields dE = hdN for this case and by the definition of homogeneous flow one has dS = sdN where N is the quantity of matter that flows, $N = \sum_c N_c$, H is the enthalpy H = E+PV and S is the entropy (it being noted that h and s denote the enthalpy H per unit amount of material and entropy S per unit amount of material, respectively). Defining the material fraction x_c by $dN_c = x_c dN$, one may substitute the expressions for dN_c, dE and dS into equation (B-1) to obtain the following expression for a differential amount of essergy $d\varepsilon$ which flows with a differential amount of homogeneous matter across a stationary boundary:

$$d\varepsilon = (h - T_o s - \sum_c \mu_{co} x_c)dN \tag{B-2}$$

For the flow of M amount of material, equation (B-2) may be integrated to give

$$\varepsilon^f = \int_o^{\varepsilon^f} d\varepsilon = \int_o^M (h - T_o s - \sum_c \mu_{co} x_c)dN \tag{B-3}$$

If the flow is steady, then h, s, and $\{x_c\}$ are constant so that equation (B-3) reduces to,

$$\varepsilon^{fs} = M(h - T_o s - \sum_c \mu_{co} x_c) \tag{B-4}$$

where ε^{fs} denotes the value of ε^f which results for steady flow. Summarizing this result in our convenient time derivative form one obtains,

$$\dot{\varepsilon}^{fs} = \dot{M}(h - T_o s - \sum_c \mu_{co} x_c) \tag{B-5}$$

Equation (B-5) represents the essergy flow $\dot{\varepsilon}^{fs}$ associated with steady, homogeneous, hydrodynamic flow of material across a stationary boundary at a rate of M amount of material per unit time.

APPENDIX C

DIFFERENT FORMS OF HYDRODYNAMIC FLOW ESSERGY

The essergy associated with steady, homogeneous, hydrodynamic flow at boundary region b may be expressed by (see Appendix B):

$$\dot{\varepsilon}_b^{fs} = \dot{M}_b \left(h_b - T_o s_b - \sum_c \mu_{co} x_{c,b}\right) \tag{C-1}$$

where:

$\dot{M}_b$ ≡ time rate of material (hydrodynamic flow) through boundary region b in unit amount of material per unit time (e.g., moles/hr, lb/sec, etc.).

h_b ≡ enthalpy* per unit amount of material (e.g., Btu/mole, Btu/lb, etc.) at the conditions of boundary of regions b; T_b, P_b, $\{x_{c,b}\}$.

s_b ≡ entropy per unit amount of material (e.g., Btu/mole °R Btu/lb°R, etc.) at the conditions of boundary region b; T_b, P_b, $\{x_{c,b}\}$.

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*h_b is ordinary enthalpy (i.e. neglecting gravitational, gross kinetic, stress, nuclear, capillarity, electric and magnetic effects, etc.). The results of this discussion may be easily generalized to include all forms of energy using the methods discussed by Evans, et al. (1966).

$\mu_{c,o}$ ≡ Gibbs chemical potential of component c when at equilibrium with the environment.

$x_{c,b}$ ≡ material fraction of component c flowing through boundary region b where the material flow is made up of C different components (i.e., c = 1, 2, 3, ..., C).

Recall that the Gibbs free energy per unit amount of material at boundary region b may be expressed by $h(T_o,P_o,\{x_{c,b}\}) - T_o s(T_o,P_o,\{x_{c,b}\})$.* Adding and subtracting this quantity within the brackets of equation (C-1) yields,

$$\dot{\varepsilon}_b^{fs} = \dot{M}_b[h_b - h(T_o,P_o,\{x_{c,b}\}) - T_o s_b + T_o s(T_o,P_o,\{x_{c,b}\})]$$

$$+ \dot{M}_b[h(T_o,P_o,\{x_{c,b}\}) - T_o s(T_o,P_o,\{x_{c,b}\}) - \sum_c \mu_{c,o} x_{c,b}] \quad \text{(C-2)}$$

The Integrated Gibbs equation will give,

$$\sum_c x_{c,b}\, \mu_c(T_o,P_o,x_{c,b}) = h(T_o,P_o,\{x_{c,b}\}) - T_o s(T_o,P_o,\{x_{c,b}\}) \quad \text{(C-3)}$$

Therefore, in view of the Integrated Gibbs Equation, the following expression may be obtained

---

*The terms $h(T_o,P_o,\{x_{c,b}\})$ and $s(T_o,P_o,\{x_{c,b}\})$ are the enthalpy per unit amount of material and entropy per unit amount of material, respectively, at composition $\{x_{c,b}\}$, environment temperature T_o and environment pressure P_o at boundary region b where the material flow is made up of C different components (i.e., c = 1, 2, 3, ..., C).

$$\dot{\varepsilon}_b^{fs} = \dot{M}_b[h_b - h(T_o, P_o, \{x_{c,b}\}) - T_o s_b + T_o s(T_o, P_o, \{x_{c,b}\})]$$

$$+ \dot{M}_b\{\sum_c x_{c,b}[\mu(T_o, P_o, x_{c,b}) - \mu(T_o, P_o, x_{c,o})]\} \quad \text{(C-4)}$$

The second term in equation (C-4) is a measure of the maximum work that can be obtained from the change in composition from $x_{c,b}$ to $x_{c,o}$ at the environment conditions T_o and P_o and is known as flow cell essergy $\dot{\varepsilon}_b^{fc}$ (since a concentration cell is needed to obtain this type work).

$$\dot{\varepsilon}_b^{fc} = \dot{M}_b\{\sum_c x_{c,b}[\mu(T_o, P_o, x_{c,b}) - \mu(T_o, P_o, x_{c,o})]\} \quad \text{(C-5)}$$

The first term in equation (C-4) is a measure of the maximum work obtainable from combined heat and work effects at fixed composition for the material flow through boundary region b--this type essergy being called flow thermomechanical essergy and denoted $\dot{\varepsilon}^{fTM}$:

$$\dot{\varepsilon}_b^{fTM} = \dot{M}_b[h_b - h(T_o, P_o, \{x_{c,b}\}) - Ts_b + T_o s(T_o, P_o, \{x_{c,b}\})] \quad \text{(C-6)}$$

Therefore, in view of equations (C-4), (C-5), and (C-6) it is seen that flow essergy may be divided into two distinguishable forms:

$$\dot{\varepsilon}_b^{fs} = \dot{\varepsilon}_b^{fTM} + \dot{\varepsilon}_b^{fc} \qquad \text{(C-7)}$$

It is known that,

$$h_b - h(T_o, P_o, \{x_{c,b}\}) = \int_{T_o, P_o, \{x_{c,b}\}}^{T_b, P_b, \{x_{c,b}\}} dh \qquad \text{(C-8)}$$

and

$$s_b - s(T_o, P_o, \{x_{c,b}\}) = \int_{T_o, P_o, \{x_{c,b}\}}^{T_b, P_b, \{x_{c,b}\}} ds \qquad \text{(C-9)}$$

Thus, the flow thermomechanical essergy $\dot{\varepsilon}_b^{fTM}$ may be expressed as follows:

$$\dot{\varepsilon}_b^{fTM} = \dot{M}_b \left[\int_{T_o, P_o, \{x_{c,b}\}}^{T_b, P_b, \{x_{c,b}\}} dh - \int_{T_o, P_o, \{x_{c,b}\}}^{T_b, P_b, \{x_{c,b}\}} ds\right] \qquad \text{(C-10)}$$

Recalling the Maxwell relation for different enthalpy change dh at fixed composition one obtains,

$$dh = Tds + vdp \qquad \text{(C-11)}$$

Substituting equation (C-11) into equation (C-10) and rearranging yields,

$$\dot{\varepsilon}_b^{fTM} = \dot{M}_b[\int_{T_o,P_o,\{x_{c,b}\}}^{T_b,P_b,\{x_{c,b}\}} vdp + \int_{T_o,P_o,\{x_{c,b}\}}^{T_b,P_b,\{x_{c,b}\}} (T-T_o)ds] \quad \text{(C-12)}$$

The first term in equation (C-12) is defined as flow mechanical essergy $\dot{\varepsilon}_b^{fM}$ since it represents the mechanical work that would be produced by the material stream at boundary region b flowing reversibly from T_b,P_b to T_o,P_o at fixed composition $\{x_{c,b}\}$. The second term in equation (C-12) is defined as flow thermal essergy $\dot{\varepsilon}_b^{fT}$ since it represents the essergy associated with the thermal energy flowing with the material stream at boundary area b at fixed composition $\{x_{c,b}\}$. Therefore, it is seen that flow thermomechanical essergy at boundary region b is made up of two distinguishable forms:

$$\dot{\varepsilon}_b^{fM} = \dot{M}_b \int_{T_o,P_o,\{x_{c,b}\}}^{T_b,P_b,\{x_{c,b}\}} vdp \quad \text{(C-13)}$$

$$\dot{\varepsilon}_b^{fT} = \dot{M}_b \int_{T_o,P_o,\{x_{c,b}\}}^{T_b,P_b,\{x_{c,b}\}} (T-T_o)ds \quad \text{(C-14)}$$

In view of equations (C-12), (C-13) and (C-14), equation (C-7) becomes,

$$\dot{\varepsilon}_b^{fs} = \dot{\varepsilon}_b^{f} + \dot{\varepsilon}_b^{fT} + \dot{\varepsilon}_b^{fc} \tag{C-15}$$

Therefore it is observed that the essergy associated with steady, homogeneous hydrodynamic flow may be divided into three different distinguishable forms.

Results identical to the above could be shown for non-steady or non-homogeneous flow. The development would be the same as for the steady, homogeneous conditions except that all operations would have to be performed on the integrand of following integral which is the general expression for hydrodynamic flow essergy:

$$\dot{\varepsilon}^{f} = \int_{o}^{\dot{\varepsilon}^{f}} d\dot{\varepsilon} = \int_{o}^{\dot{M}} (h - T_o s - \sum_c \mu_{co} x_c) d\dot{N} \tag{C-16}$$

APPENDIX D

CALCULATING ESSERGY FLOWS

Work Essergy

In order to obtain an expression for the essergy flow associated with shaft work first differentiate equation (1) to obtain,

$$d\varepsilon = dE + P_o dV - T_o dS - \sum_c \mu_{co} dN_c \tag{D-1}$$

For a given environment dV, dS and $\{dN_c\}$ are all zero when the only effect is reversible shaft work. Therefore, one obtains

$$d\varepsilon = dE \tag{D-2}$$

For this case, the First Law will yield,

$$dE = -dW \tag{D-3}$$

In view of equation (D-3), one obtains for equation (D-2),

$$d\varepsilon = -dW \tag{D-4}$$

If the above differential is considered to be taken with respect to time one gets,

$$\dot{\varepsilon} = -\dot{W} \tag{D-5}$$

Flow Cell Essergy

Equation (C-5) in Appendix C gives the following expression for flow cell essergy:

$$\dot{\varepsilon}_b^{fC} = \dot{M}_b\{\sum_c x_{c,b}(\mu(T_o,P_o,x_{c,b})-\mu(T_o,P_o,x_{c,o})]\} \tag{C-5}$$

Recall that,

$$\sum_c x_{c,b}\ \mu(T_o,P_o,x_{c,b}) = h(T_o,P_o,\{x_{c,b}\}-T_o s(T_o,P_o,\{x_{c,b}\}) \tag{C-3}$$

Therefore,

$$\sum_c x_{c,b}\ \mu(T_o,P_o,x_{c,o}) = h(T_o,P_o,\{x_{c,o}\})-T_o s(T_o,P_o,\{x_{c,o}\}) \tag{D-6}$$

Substituting equations (C-3) and (D-6) into equation (C-5) yields,

$$\dot{\varepsilon}_b^{fC} = \dot{M}_b\{h(T_o,P_o,\{x_{c,b}\})-h(T_o,P_o,\{x_{c,o}\})$$

$$- T_o[s(T_o,P_o,\{x_{c,b}\})-s(T_o,P_o,\{x_{c,o}\})]\} \qquad \text{(D-7)}$$

Equation (D-7) may be recognized as representing the Gibbs-free-energy difference between the material stream at boundary area b at a composition $\{x_{c,b}\}$ and the most stable chemical configuration of all the species at a composition $\{x_{c,o}\}$ that make up the material stream at boundary area b both taken at the temperature T_o and pressure P_o of the environment.

For example, if the material stream at boundary area b is some type of hydrocarbon fuel, equation (D-7) will represent the Gibbs-free-energy change that would occur if all species in the fuel (e.g. various hydrocarbons, sulfur compounds, etc.) are brought to complete, stable, chemical equilibrium with the environment which occurs when each of the species is in its most stable chemical configuration found in the environment (e.g., H_2O, CO_2, $CaSO_4 \cdot 2H_2O$, etc.). That is, when all species are at the Gibbs chemical potential of the environment. Therefore, the terms $h(T_o,P_o,\{x_{c,b}\}-h(T_o,P_o,\{x_{c,o}\})$ and $s(T_o,P_o,\{x_{c,b}\})-s(T_o,P_o,\{x_{c,o}\})$ are the enthalpy and entropy, respectively, of the material stream at b relative to that which would exist if each species making up the material stream were in its most stable chemical configuration found in the environment.

Flow Thermomechanical Essergy

Equation (C-6) in Appendix C gives the following expression for flow thermomechanical essergy:

$$\dot{\varepsilon}_b^{fTM} = \dot{M}_b[h_b - h(T_o, P_o, \{x_{c,b}\}) - T_o s_b + T_o, P_o, \{x_{c,b}\})] \qquad \text{(C-6)}$$

Rearranging equation (C-6) and substituting $h_o = h(T_o, P_o, \{x_{c,b}\})$ and $s_o = s(T_o, P_o, \{x_{c,b}\})$ one gets,

$$\dot{\varepsilon}_b^{fTM} = \dot{M}_b[(h_b - h_o) - T_o(s_b - s_o)] \qquad \text{(D-8)}$$

The flow thermomechanical essergy of a stream at boundary area b may be calculated simply by knowing its flow rate and thermodynamic properties and the thermodynamic properties of the environment which is to be used as the datum state.

It may also be observed that for incompressible flow with constant heat capacity (i.e., v and C_p are constant, $ds = C_p\ dT/T$, where C_p is the heat capacity per unit amount of material at constant pressure), equation (C-12) of Appendix C may be integrated directly to obtain,

$$\dot{\varepsilon}_b^{fTM} = \dot{M}_b[v_b(P_b - P_o) + C_p(T_b - T_o - T_o \ln \frac{T_b}{T_o})] \qquad \text{(D-9)}$$

APPENDIX E

POWER CYCLE ESSERGY FLOWS

The specific essergy and essergy flow associated with steam or water flows at various points in the power cycle were calculated using equation (D-8) from Appendix D.

Recall from Appendix D, for incompressible flow with constant heat capacity, flow thermomechanical essergy may be calculated by the following expression:

$$\dot{\varepsilon}_b^{fTM} = \dot{M}_b[v_p(P_b-P_o) + C_{p,b}(T_b-T_o-T_o \ln \frac{T_b}{T_o})] \qquad \text{(D-9)}$$

For air flows in the plant, the pressure differential P_b-P_o may be neglected so that equation (D-9) becomes,

$$\dot{\varepsilon}_b^{fTM} = \dot{M}_b C_{p,b}(T_b-T_o-T_o \ln \frac{T_b}{T_o}) \qquad \text{(E-1)}$$

This simplification is equivalent to neglecting the flow mechanical part of the essergy associated with air flow and assuming that it is made up completely of flow thermal form. The specific essergy and essergy flowing with combustion air at various points in the power cycle were calculated using equation (E-1).

The value of the specific essergy of the fuel flow to the power plant is close to its heating value. A more precise value for the specific essergy of the fuel may be had by using equation (D-7) from Appendix D. Employing methods such as those illustrated by Obert (1948,1960), values for $h(T_o,P_o,\{x_{c,b}\}-h(T_o,P_o,\{x_{c,o}\})$ and $s(T_o,P_o,\{x_{c,b}\})-s(T_o,P_o,\{x_{c,o}\})$ may be calculated. These calculations have been done by Fehring and Gaggioli (1977) for the fuel used in this power plant. The results are,

$$h(T_o,P_o,\{x_{c,b}\}-h(T_o,P_o,\{x_{c,o}\}) = 11875 \text{ Btu/lb}$$

$$s(T_o,P_o,\{x_{c,b}\})-h(T_o,P_o,\{x_{c,o}\}) = 1.044 \text{ Btu/lb°R}$$

$$\varepsilon_F = 11875 + 510.1(1.044) = 12408 \text{ Btu/lb}$$

$$\dot{\varepsilon}_F^{fC} = \dot{M}_F\varepsilon_F = 2472.4925 \text{ Btu/hr}$$

The environment at P_o = 14.7 and T_o = 50.4°F is taken as the datum state for all of the calculations. The results for power cycle design conditions are presented in Table E-1 while changes in essergy flows due to deterioration of feedwater heater number 5 are presented in Table E-2.

Table E-1. Essergy Flows at Various Flows in the Power Plant

Point*	Specific Essergy Btu/lb	Essergy Flow Million Btu/hr
T	700.917	1310.0741
RH	631.828	1036.3217
B7	522.741	956.7096
B6	529.458	885.6212
B5	430.225	679.5180
B4	365.979	556.9531
B3	251.601	356.8917
B2	192.151	257.3386
B1	131.742	162.8011
B0	15.225	17.9265
H7	520.235	94.0247
H6	524.919	48.9466
H5	427.322	27.6969
H4	361.439	37.3489
H3	247.112	13.3270
H2	188.251	8.8691
H1	127.559	7.4390
S6	679.720	22.0895
S9	533.659	2.8577
S13	247.112	6.2519
S14	188.251	10.9186
FW15	0.992	1.1748
FW14	5.083	6.0198
FW13	5.255	6.7775
FW12	10.649	13.7343

*Point refers to the point in the cycle as defined by Figure A-1.

Table E-1 (concluded)

Point	Specific Essergy Btu/lb	Essergy Flow Million Btu/hr
FW11	10.747	14.4840
FW10	18.714	25.2213
FW9	18.738	26.7383
FW8	18.957	27.0508
FW7	38.828	55.4058
FW6	38.877	59.4931
FW5	53.187	81.3915
FW4	53.236	84.9170
FW3	54.248	101.3942
FW2	63.822	119.2888
FW1	86.955	162.5264
FW	126.500	236.4394
C1	4.870	0.2845
C3	10.479	0.5064
C5	18.733	1.0133
C7	39.048	4.0350
C9	53.944	3.4964
C10	59.883	16.4068
C11	84.408	15.2555
C12	18.733	0.4739
C13	10.749	0.6234
F,HP	12408	2112.3875
F,RH	12408	360.1049
A1	1.704	3.8510
A,HP	4.490	9.9736
A,RH	4.490	1.7004

Table E-2. Change* in Essergy Flows in the Power Plant Due to Deterioration of Feedwater Heater Number 5

Point**	Essergy Flow Million Btu/hr	
	Case B	Case C
H7		97.1653
H6	52.8982	70.8945
H5	24.7266	0
H4		37.7631
Fuel Increase+		27.0742

*Data for points whose flow rates did not change from design conditions are omitted for clarity.

**Point refers to the point in the cycle as defined by Figure A-1.

+Additional boiler fuel is needed in Case C to take the feedwater from its depressed temperature (460°F) to design operating temperature (471.7°F).

APPENDIX F

STUDY OF INTERNAL ECONOMY USING SIMPLE POWER CYCLES

The simple power cycles studied in order to determine the effect of particular operations such as reheating, regenerative feedwater heating and air preheating on the internal economy of the complex power cycle are illustrated by Figures F-1 through F-6. The symbols used in these figures are defined as follows:

F--temperature, °F

P--pressure, psia

h--specific enthalpy, Btu/lb

e--specific essergy, Btu/lb

s--specific entropy, Btu/lb°R

#--mass flow, lb/hr

W--shaft work flow, Btu/hr

C--cash flow, $/hr

Note that all zone capital costs are assumed to be sunk (i.e., C_R = 0 for all R) and the essergy in the condensate from the feedwater heater(s) and steam air preheater(s) is assumed to have zero economic value.

Table F-1 illustrates the principle of a single arbitrary degree of freedom for setting internal cash flows for the power cycle. The unit essergy cost for the feedwater entering the economizer is set at three different values and the economic balance equations for the cycle solved simultaneously to show that exactly the same value for the feedwater entering the economizer will be obtained.

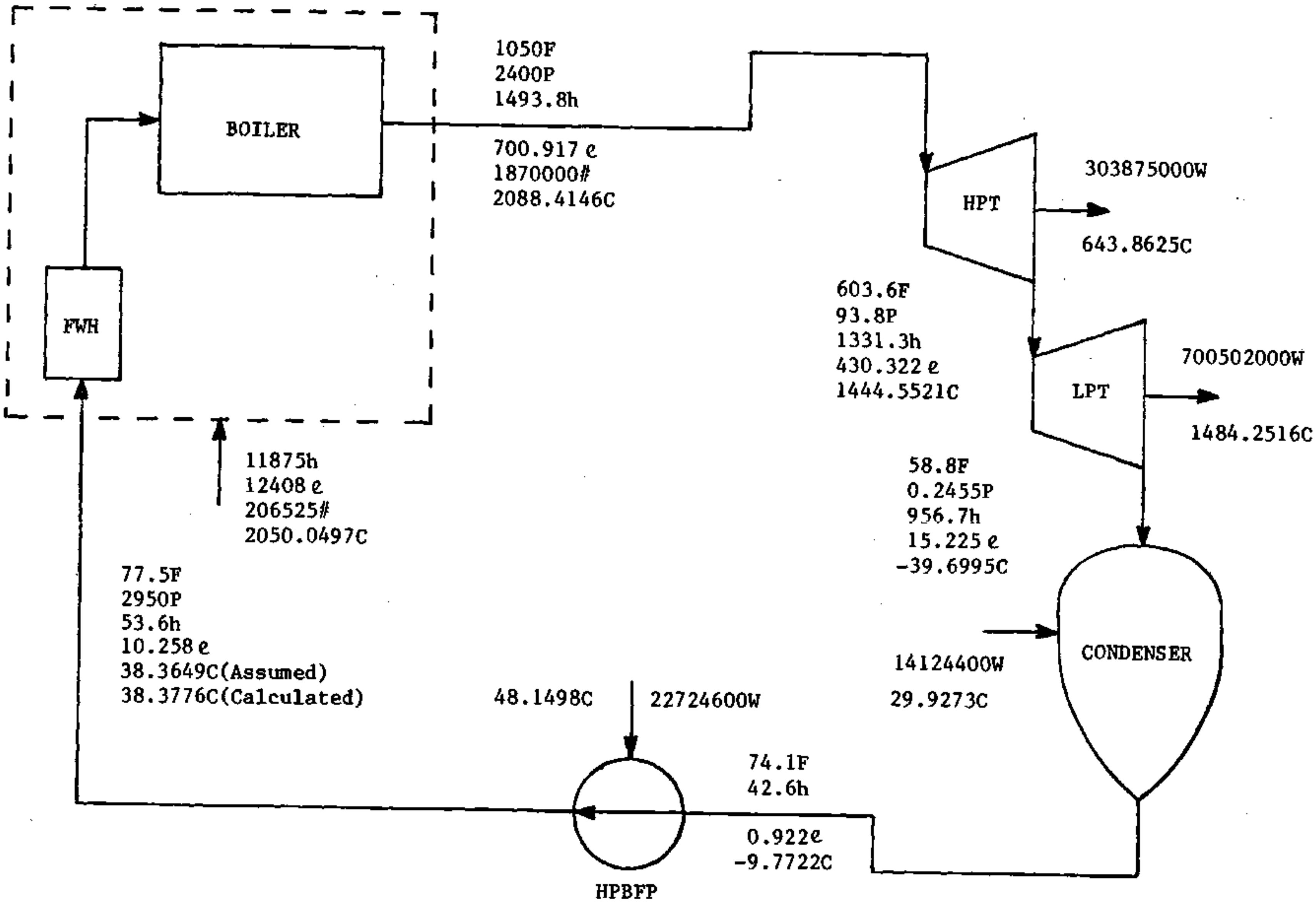

Figure F-1. Simple Power Cycle Depicting the Internal Economy Without Regenerative Feedwater Heating

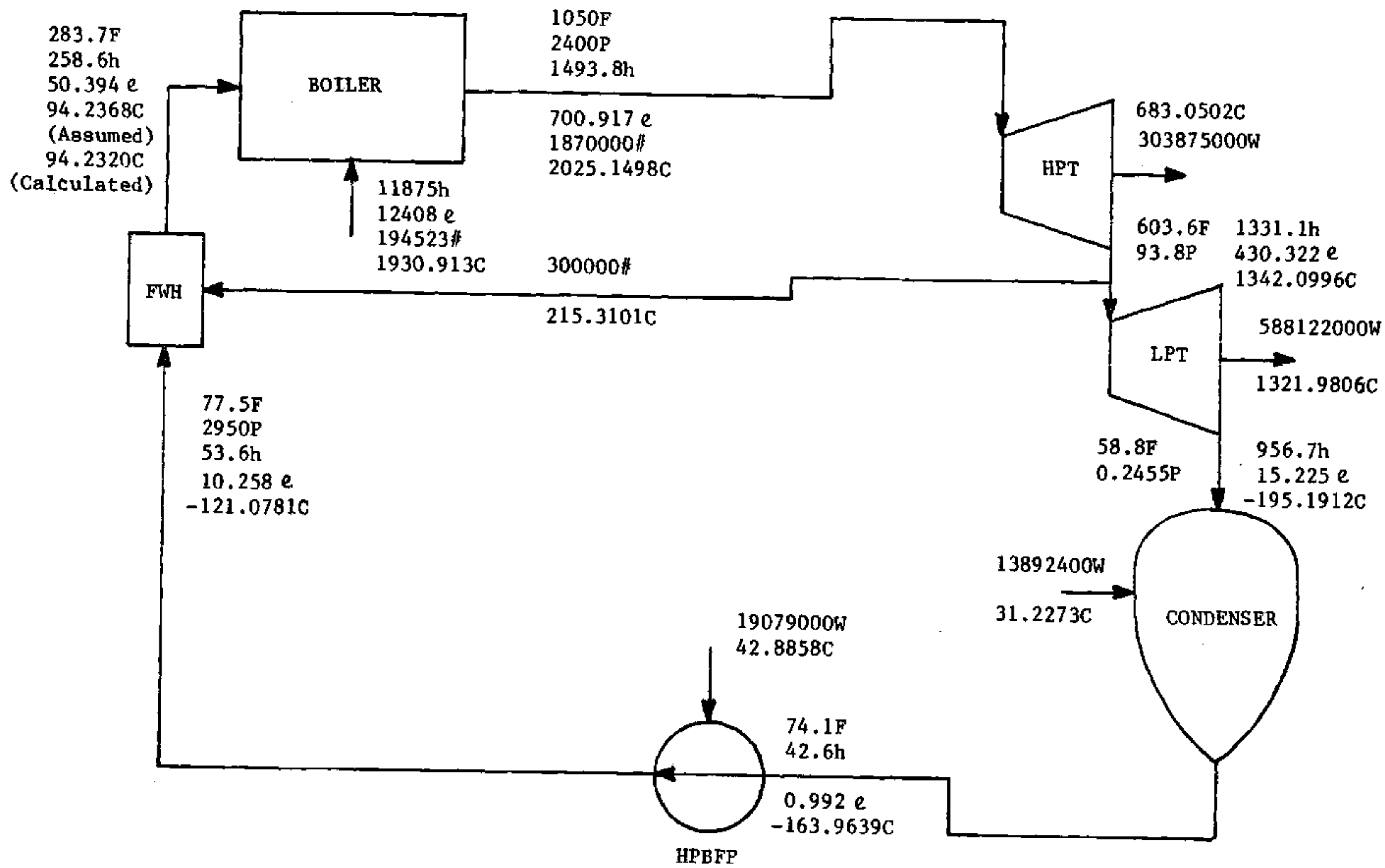

Figure F-2. Simple Power Cycle Depicting the Effect of One Stage of Regenerative Feedwater Heating on Internal Economy

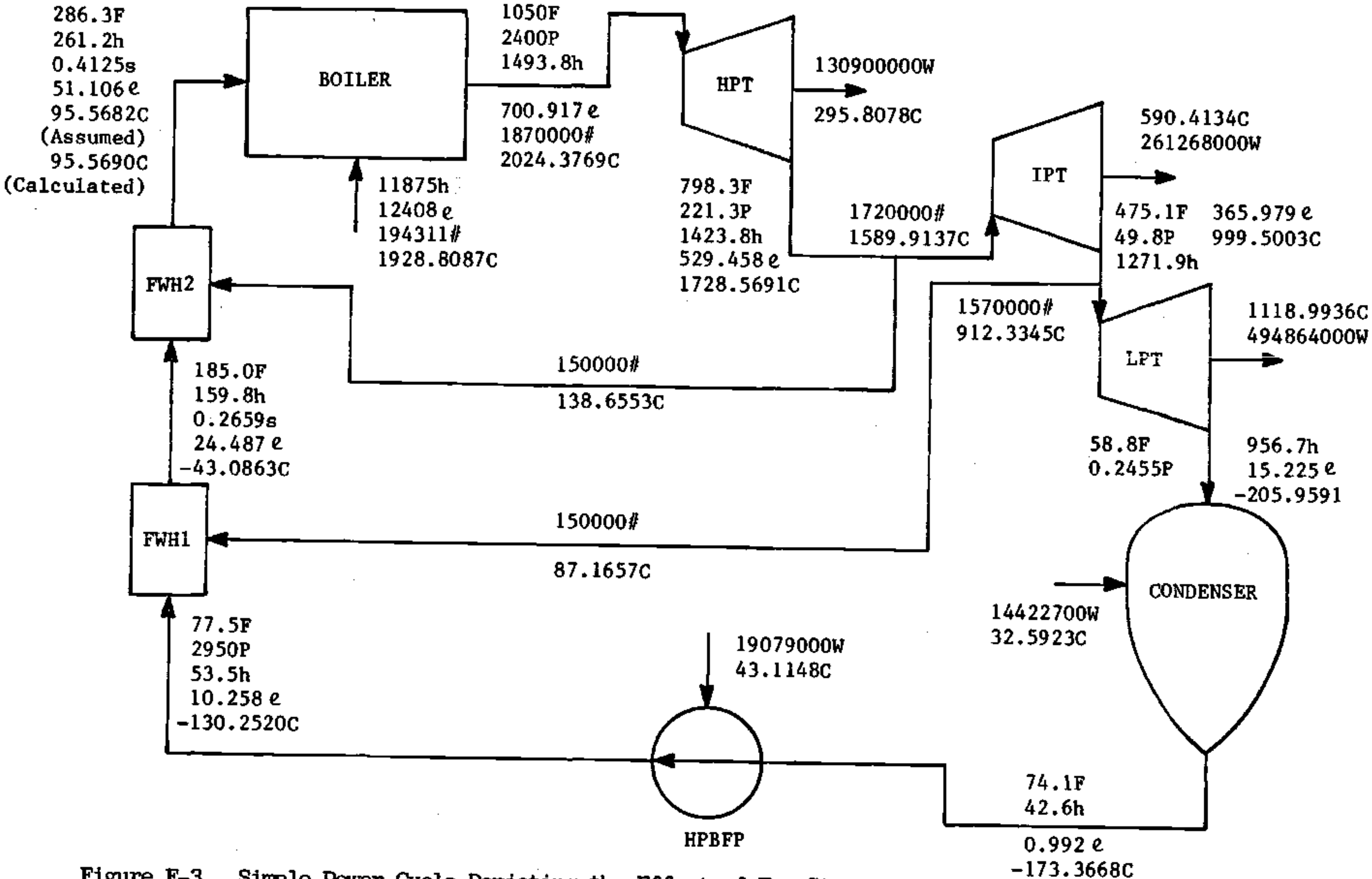

Figure F-3. Simple Power Cycle Depicting the Effect of Two Stages of Regenerative Feedwater Heating on Internal Economy

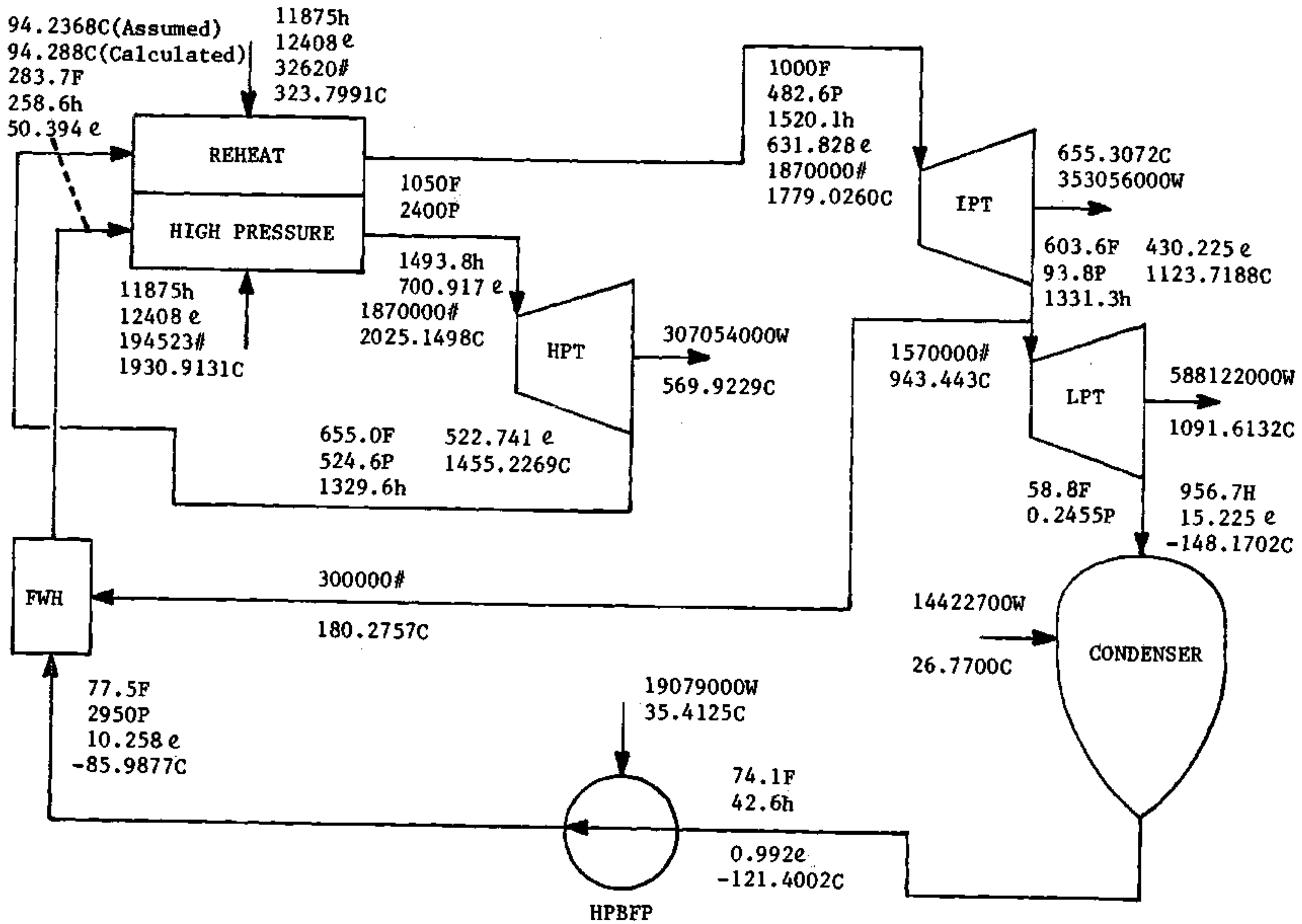

Figure F-4. Simple Power Cycle Depicting the Effect of One Stage of Regenerative Feedwater Heating and Reheating on Internal Economy

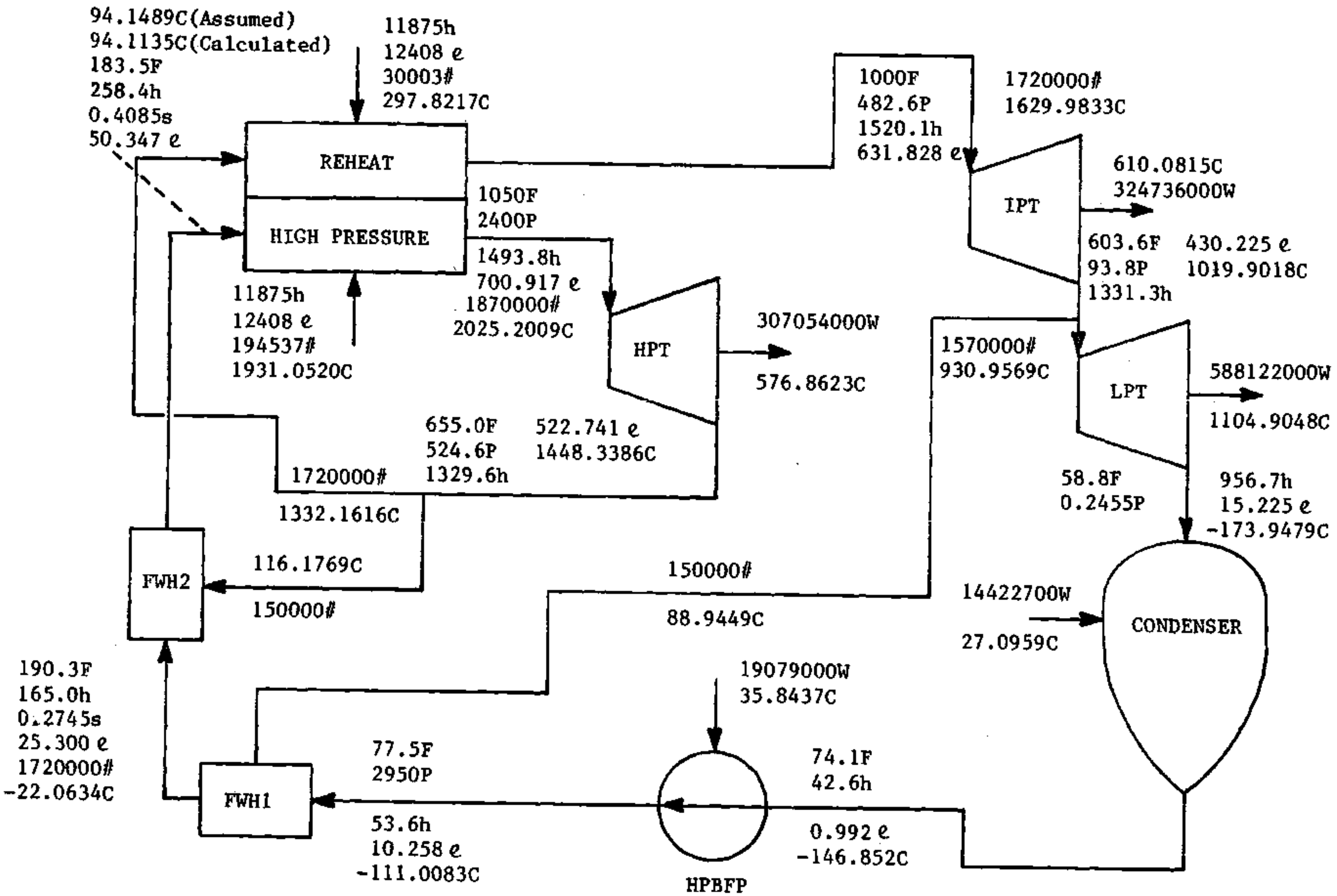

Figure F-5. Simple Power Cycle Depicting the Effect of Two Stages of Regenerative Feedwater Heating and Reheating on Internal Economy

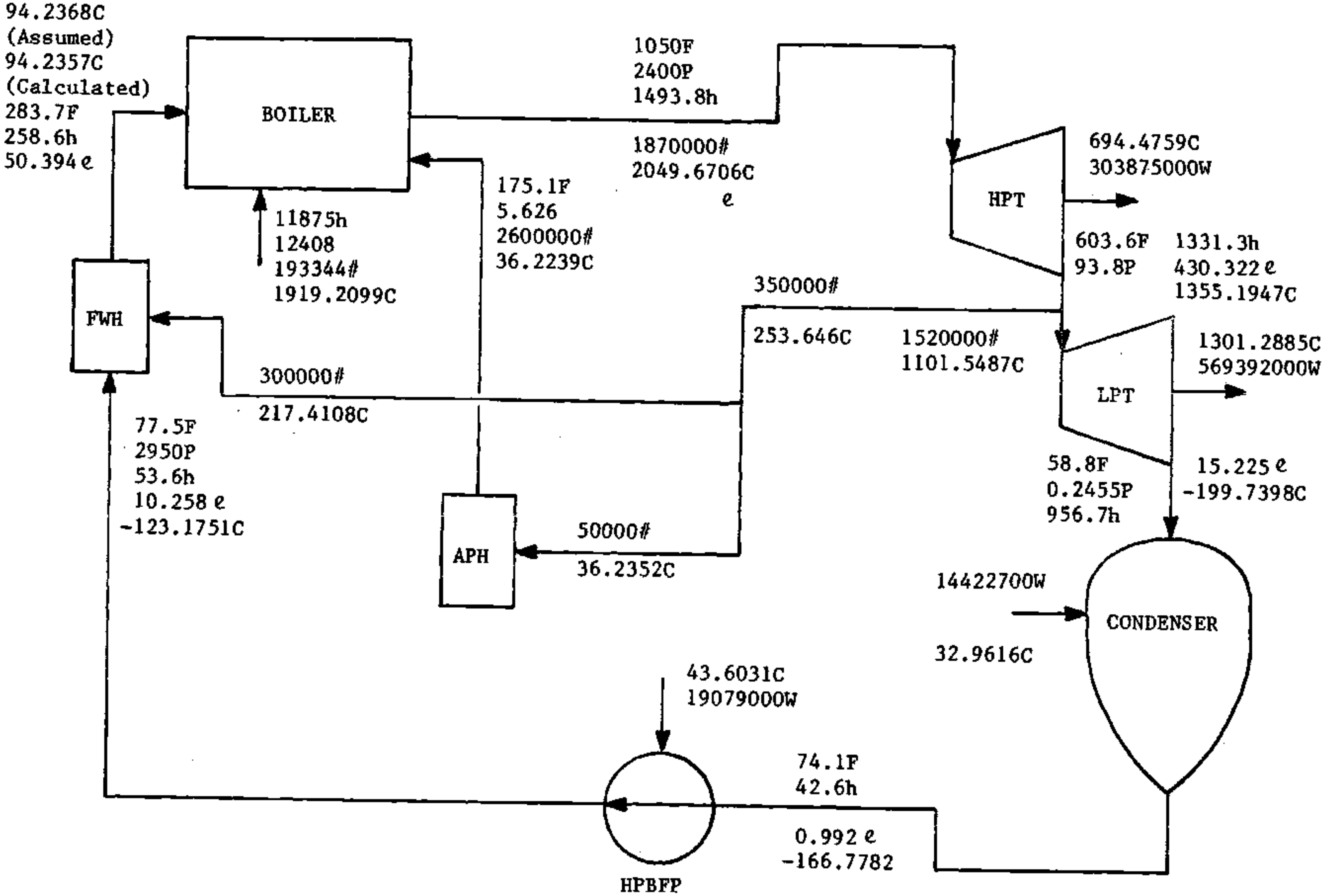

Figure F-6. Simple Power Cycle Depicting the Effect of One Stage of Regenerative Feedwater Heating and Air Preheating on Internal Economy

Table F-1. Economic Value of Various Essergy Flows in the Power Cycle for Different Assumed Values of $c_{e,FW}\dot{\mathcal{E}}_{FW}$

Point*	Economic Value of Essergy Flows, $/Hr		
	Run 1	Run 2	Run 3
FW(Assumed)	472.879 ($c_{e,FW}$ = 2.00 $/MMBTU)	945.757 ($c_{e,FW}$ = 4.00 $/MMBTU)	1418.636 ($c_{e,FW}$ = 6.00 $/MMBTU)
T	2400.301	2892.039	3383.777
RH	1935.140	2369.874	2804.609
B7	1591.351	2072.853	2554.355
B6	1546.849	1990.133	2433.418
B5	1080.777	1499.350	1917.923
B4	853.586	1256.886	1660.186
B3	463.282	839.198	1215.113
B2	277.483	632.401	987.319
B1	105.945	433.435	760.926
B0	-213.336	98.700	410.735
H7	157.308	204.858	252.407
H6	86.231	110.942	135.653
H5	46.099	63.258	80.417
H4	57.960	85.345	112.730
H3	17.614	31.907	46.199
H2	10.245	22.729	35.214

*Point refers to the point in the cycle as defined by Figure A-1.

Table F-1 (continued)

Point*	Economic Value of Essergy Flows,$/Hr		
	Run 1	Run 2	Run 3
H1	10.245	22.729	35.214
FW15	5.000	20.455	35.910
FW14	-20.855	292.988	606.830
FW13	-10.761	318.537	647.834
FW12	4.578	318.537	647.834
FW11	4.578	346.360	647.834
FW10	27.286	346.360	688.142
FW9	27.286	383.360	688.142
FW8	32.210	383.360	739.435
FW7	95.264	388.285	744.359
FW6	95.264	478.724	862.183
FW5	146.458	478.724	862.183
FW4	146.458	547.076	947.693
FW3	146.458	547.076	947.693
FW2	219.152	619.770	947.693

Table F-1 (continued)

Point*	Economic Value of Essergy Flows,$/Hr		
	Run 1	Run 2	Run 3
FW1	310.477	735.806	1020.388
FW (calculated)	472.879	945.758	1161.135
			1418.636

APPENDIX G

POWER CYCLE ESSERGY FLOWS

This appendix presents the hourly essergy costs for various points in the power plant for the three trials considered in this tudy. Table G-1 presents the hourly costs associated with steam water, air and fuel flows while Table G-2 presents the hourly costs associated with shaft work flows.

Table G-1. Essergy Costs for Steam or Water Flows at Various Points in the Power Plant

Point	Essergy Costs, $/hr		
	Trial 1	Trial 2	Trial 3
T	1689.910	2229.608	2898.5189
RH	1272.937	1750.818	2375.602
B7	1097.754	1626.218	2079.197
B6	1007.096	1494.361	1995.974
B5	680.173	1140.275	1504.864
B4	487.814	931.128	1262.200
B3	177.591	590.803	844.151
B2	32.641	422.772	637.077
B1	-95.623	264.360	437.751
B0	-310.429	32.565	102.811
H7	108.522	160.709	205.484
H6	56.142	83.305	111.268
H5	29.508	48.365	63.484
H4	33.123	63.225	85.706
H3	6.752	22.462	32.095
H2	1.560	15.283	22.894
H1	-4.513	12.476	20.658
F,HP	1689.910	1689.910	1689.910
F,RH	288.084	288.084	288.084
A,HP	0.000	25.077	40.867
A,RH	0.000	4.275	6.967

Table G-1 (concluded)

Point	Essergy Cost, $/hr		
	Trial 1	Trial 2	Trial 3
FW15	--	65.253	297.123
FW14	--	77.729	322.875
FW13	--	77.729	322.875
FW12	--	93.012	350.863
FW11	--	93.012	350.863
FW10	--	115.474	388.052
FW9	--	115.474	388.052
FW8	--	117.111	392.976
FW7	--	180.336	483.776
FW6	--	180.336	483.776
FW5	--	228.701	552.354
FW4	--	228.701	552.354
FW3	--	228.701	552.354
FW2	--	270.607	625.048
FW1	--	353.912	741.410
FW	0.000	514.621	951.988

Table G-2. Essergy and Cash Flows Associated with Shaft Work Inputs and Outputs for Power Plant Equipment

Component	Essergy Flow, Btu/hr	Cost of Essergy Flow $/hr	
		Trials 1 & 2	Trials 3
High Pressure Turbine	300.515	556.978	894.496
Intermediate Pressure Turbine			
Stage 1	159.286	295.223	474.123
Stage 2	146.099	270.781	434.869
Low Pressure Turbine			
Stage 1	90.396	167.541	269.067
Stage 2	149.508	277.100	445.017
Stage 3	72.855	135.031	216.857
Stage 4	67.843	125.741	201.937
Stage 5	118.333	219.319	352.222
Condenser and Auxiliaries	14.124	26.178	42.042
Low Pressure Boiler Feed Pump	0.883	1.636	2.628
High Pressure Boiler Feed Pump	22.610	41.906	67.300
Plant	1067.218	1977.994	3167.617

Note: The cash flow associated with shaft work flows for Trials 1 and 2 are less than Trial 3 because capital cost contributions have been neglected.

APPENDIX H

ECONOMIC CALCULATIONS FOR THE REPLACEMENT OF FEEDWATER HEATER NUMBER 5

Annual Fuel Cost Due to Deterioration of Feedwater Heater Number 5

Data: 8000 hours of operation per year

70% capacity factor

The annual fuel cost due to deterioration of feedwater heater number 5 is determined by multiplying its annual operating time by the difference in the total hourly bleed steam essergy cost for heaters 4 through 7 when heater number 5 is operating in deteriorated condition (Case B) and the total hourly bleed steam essergy cost for heaters 4 through 7 when the plant is operating at design conditions (Case A). For Trial 1, this calculation gives,

$$8000 \text{ hrs/yr} \times 0.70 \times (228.653-227.295) \text{ \$/hr} = \$7605\text{/year}$$

Annual Downtime Fuel Cost for Plugging Tubes in Feedwater Heater Number 5

Data: 3 weeks of downtime per year

The annual downtime fuel cost for plugging tubes in feedwater heater number 5 is determined by multiplying the annual downtime by the difference in the total hourly bleed steam essergy cost for heaters 4 through 7 when heater number 5 is out of service (Case C) and the total hourly bleed steam essergy cost for heaters 4 through 7 when the plant is operating at design conditions (Case A). For Trial 1, this calculation gives

3 wks/yr x 168 hrs/wk x (248.600-227.295) \$/hr = \$10738/year

Maintenance Cost for Repairing Feedwater Heater Number 5

Data: 15 leaks occur per year

28 man-hours of labor are required for repairing each leak

\$10.07 is the charge for each man-hour of labor

The annual maintenance cost for repairing feedwater heater number 5 is given by

15 leaks/yr x 28 man-hrs/leak x \$10.07/man-hr = \$4229/yr

Total Additional Fuel and Maintenance Expenditure Due to Deterioration in Feedwater Heater Number 5

The total additional fuel and maintenance expenditure due to leaks in feedwater heater number 5 is given by:

$$\$7605/yr + \$10738/yr + \$4229/yr = \$22572/year$$

Annual Discounted Cash Flow

For the purpose of calculating annual discounted cash flow the following data will be used.

Data: 6% fuel and maintenance expenditure escalation rate per year

9% cost of capital (interest rate)

20 years service life

50% income tax

Fuel and Maintenance Savings

The fuel and maintenance saving for each year is determined by the following equation:

$$S_a = S_1(1+E)^{a-1}$$

where:

S_a = fuel and maintenance saving for year a

S_1 = fuel and maintenance saving for the first year

E = yearly escalation rate

a = year for which escalated fuel and maintenance saving is desired

For year nine and Trial 1 one obtains,

$$S = \$22572/yr \times (1+.06)^{9-1} = \$35,976$$

Replacement Feedwater Heater Depreciation

The replacement feedwater heater depreciation is determined by the following equation (sum of the years digits method; after Peters and Timmerhaus (1968).

$$d_a = \frac{2(n-a+1)}{n(n+1)} (V-V_s)$$

where:

d_a = depreciation for year a

n = service life

V = initial capital cost of equipment

V_s = salvage value of equipment

a = year for which depreciation is desired

For example, for the feedwater heater at a new capital cost of $235,000 and a salvage value of $18,000, the depreciation for year nine will be:

$$d_a = \frac{2(20-9+1)}{20(20+1)} (\$235{,}000-\$18{,}000) = \$12{,}400$$

Ad Valorem Tax

The ad valorem tax for the replacement feedwater heater is calculated using the following equation derived from the work of Fehring and Gaggioli (1977).

$$\text{Tax} = 5781 - 141y$$

where:

y = year for which the tax is desired.

For example, for year nine one obtains,

$$\text{Tax} = 5781-141(9) = \$4512$$

Taxable Balance

The taxable balance is determined by subtracting the replacement feedwater heater depreciation and ad valorem tax from the annual fuel and maintenance savings that occur if the deteriorated heater is replaced. For year nine and Trial 1 one obtains,

$$\$35,976 - \$12,400 - \$4512 = \$19,064$$

Total Cash Flow

The total cash flow is calculated by subtracting the income taxes at 50% and adding the replacement feedwater heater depreciation back. For year nine and Trial 1 one obtains,

$$\$19064 - 0.5 \times \$19064 + \$12400 = \$21,932$$

Discounted Cash Flow

The discounted cash flow is calculated by multiplying the total cash flow by a present worth factor. The present worth factor is determined by the following equation from Grant (1957).

$$P = S\left[\frac{1}{(1+i)^n}\right]$$

where

i represents an interest rate per interest period

n represents a number of interest periods

P represents a present sum of money

S represents a sum of money n interest periods from the present date that is equivalent to P with interest rate i.

Therefore, the discounted cash flow for year nine and Trial 1 is given by:

$$P = \$21932\left[\frac{1}{(1+.09)^9}\right] = \$21932/\text{yr} \times 0.4604 = \$10097$$

Uniform Annual Cost Savings

The calculations described in the previous sections of this appendix were performed for Trials 1, 2, and 3 for the entire life of a replacement feedwater heater and are presented in Table H-1. The uniform annual cost saving for replacing feedwater heater number 5 is determined by accumulating the discounted cash flows over the service life of the replacement heater and then dividing this sum into uniform annual credits by use of an interest factor. The

Table H-1. Cash Flow Analysis for Feedwater Heater Replacement Evaluation

TRIAL 1

YEAR	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
FUEL AND MAINTENANCE SAVING	$22572	23926	25362	26884	28497	30206	32019	33940	35976	38135	40423	42848	45419	48144	51033	54095	57341	60781	64428	68294
- DEPRECIATION	20667	19633	18600	17567	16533	15500	14467	13433	12400	11367	10333	9300	8267	7233	6200	5167	4133	3100	2067	1033
- AD VALOREM TAX	5640	5499	5358	5217	5067	4935	4794	4653	4512	4371	4230	4089	3948	3807	3666	3525	3384	3243	3102	2961
= TAXABLE BALANCE	-3735	-1206	1404	4100	6897	9771	12758	15854	19064	22397	25860	29459	33204	37104	41167	45403	49824	54438	59259	64300
- INCOME TAX(at 50%)	1868	603	702	2050	3449	4886	6379	7927	9532	11199	12930	14730	16602	18552	20584	22702	24912	27219	29630	32150
= AFTER TAX BALANCE	-1867	-603	702	2050	3448	4885	6379	7927	9532	11198	12930	14729	16602	18552	20583	22701	24912	27219	29629	32150
+ DEPRECIATION	20667	19633	18600	17567	16533	15500	14467	13433	12400	11367	10333	9300	8267	7233	6200	5167	4133	3100	2067	1033
= TOTAL CASH FLOW	18800	19030	19302	19617	19981	20385	20846	21360	21932	22565	23263	24029	24869	25785	26783	27868	29045	30319	31696	33183
X PRESENT WORTH FACTOR	.9174	.8417	.7722	.7084	.6499	.5963	.5470	.5019	.4604	.4224	.3875	.3555	.3262	.2992	.2745	.2519	.2311	.2120	.1945	.1784
= DISCOUNTED CASH FLOW	$17247	16018	14905	13897	12986	12156	11403	10721	10097	9531	9014	8542	8112	7715	7352	7020	6712	6428	6165	5920

TOTAL DISCOUNTED CASH FLOW = $201941

TRIAL 2

YEAR	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
FUEL AND MAINTENANCE SAVING	$21250	22525	23877	25309	26828	28437	30144	31952	33869	35901	38056	40339	42759	45325	48044	50927	53982	57221	60655	64294
- DEPRECIATION	20667	19633	18600	17567	16533	15500	14467	13433	12400	11367	10333	9300	8267	7233	6200	5167	4133	3100	2067	1033
- AD VALOREM TAX	5640	5499	5358	5217	5067	4935	4794	4653	4512	4371	4230	4089	3948	3807	3666	3525	3384	3243	3102	2961
= TAXABLE BALANCE	-5057	-2607	-81	2525	5228	8002	10883	13866	16957	20163	23493	26950	30544	34285	38178	42235	46465	50878	55486	60300
- INCOME TAX(at 50%)	2529	1304	41	1263	2614	4001	5442	6933	8479	10082	11747	13475	15272	17143	19089	21118	23233	25439	27743	30150
= AFTER TAX BALANCE	-2528	-1303	40	1262	2614	4001	5441	6933	8478	10081	11746	13475	15272	17142	19089	21117	23232	25439	27743	30150
+ DEPRECIATION	20667	19633	18600	17567	16533	15500	14467	13433	12400	11367	10333	9300	8267	7233	6200	5167	4133	3100	2067	1033
= TOTAL CASH FLOW	18139	18330	18640	18829	19147	19501	19908	20366	20878	21448	22079	22775	23539	24375	25289	26284	27365	28539	29810	31183
X PRESENT WORTH FACTOR	.9174	.8417	.7722	.7084	.6499	.5963	.5470	.5019	.4604	.4224	.3875	.3555	.3262	.2992	.2745	.2519	.2311	.2120	.1945	.1784
= DISCOUNTED CASH FLOW	$16641	15428	14394	13338	12444	11628	10890	10222	9612	9060	8556	8097	7678	7293	6942	6621	6324	6050	5798	5563

TOTAL DISCOUNTED CASH FLOW = $192579

TRIAL 3

YEAR	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20
FUEL AND MAINTENANCE SAVING	$24298	25756	27301	28939	30676	32516	34467	36535	38727	41051	43514	46125	48892	51826	54935	58232	61725	65429	69355	73516
- DEPRECIATION	20667	19633	18600	17567	16533	15500	14467	13433	12400	11367	10333	9300	8267	7233	6200	5167	4133	3100	2067	1033
- AD VALOREM TAX	5640	5499	5358	5217	5067	4935	4794	4653	4512	4371	4230	4089	3948	3807	3666	3525	3384	3243	3102	2961
= TAXABLE BALANCE	-2009	624	3343	6155	9076	12081	15206	18449	21815	25313	28951	32736	36677	40786	45069	49540	54208	59086	64186	69522
- INCOME TAX(at 50%)	1005	312	1672	3078	4538	6041	7603	9225	10908	12657	14476	16368	18339	20393	22535	24770	27104	29543	32093	34761
= AFTER TAX BALANCE	-1004	312	1671	3077	4538	6040	7603	9224	10907	12656	14475	16368	18338	20393	22534	24770	27104	29543	32093	34761
+ DEPRECIATION	20667	19633	18600	17567	16533	15500	14467	13433	12400	11367	10333	9300	8267	7233	6200	5167	4133	3100	2067	1033
= TOTAL CASH FLOW	19633	19945	20271	20644	21071	21540	22070	22657	23307	24023	24808	25668	26605	27626	28734	29937	31237	32643	34160	35794
X PRESENT WORTH FACTOR	.9174	.8417	.7722	.7084	.6499	.5963	.5470	.5019	.4604	.4224	.3875	.3555	.3262	.2992	.2745	.2519	.2311	.2120	.1945	.1784
= DISCOUNTED CASH FLOW	$18039	16788	15653	14624	13694	12844	12073	11326	10731	10147	9613	9125	8679	7962	7887	7541	7219	6920	6644	6386

TOTAL DISCOUNTED CASH FLOW = $213895

interest factor is determined by the following equation from Grant (1957),

$$R = P\left[\frac{i}{(1+i)^n-1} + i\right]$$

where:

P, i, and n have the same meaning as in the earlier section on discounted cash flow

R represents the end-of-period payment in a uniform series continuing for the coming n periods, the entire series equivalent to P at interest rate i.

Therefore, for Trial 1, one obtains,

Total Discounted Cash Flow (for 20 years service life) =

$201,941

$$\text{Uniform Annual Saving} = \$201941\left[\frac{0.09}{(1.09)^{20}-1} + 0.09\right]$$

$$= \$22{,}122 \text{ yr}$$

Uniform Annual Cost of Replacement Heater

For the purpose of calculating uniform annual cost for replacing a feedwater heater, the following data will be used.

Data: $235,000 heater replacement cost

$23,500 investment tax credit (at 10%)

20 years service life

$18,000 salvage value of replacement heater

9% cost of capital (interest rate)

The uniform annual cost of the replacement heater is determined by the following equation from Thumann (1977):

$$AC = (P-V_s)CR+iV_s$$

where:

AC = uniform annual cost for equipment

P = initial capital expenditure for equipment

V_s = salvage value of equipment

CR = interest factor = $\dfrac{i}{(1+i)^n+1} + i$

i = cost of capital (interest rate)

n = service life for equipment

Therefore, for the replacement feedwater heater, one obtains,

$$AC = (\$23500-\$18000) \times \frac{0.09}{(1.09)^{20}-1} + .09 \times \$18000$$

$$= \$22,817$$

Benefit to Cost Ratio

The benefit to cost ratio for replacing feedwater heater number 5 is calculated by dividing the uniform annual cost saving due to replacement of feedwater heater number 5 by the uniform annual cost of the replacement heater. For

Trial 1 one obtains a benefit to cost ratio of,

$$\frac{\$22{,}122/\text{yr}}{\$22{,}817/\text{yr}} = 0.9695$$

Maximum Allowable Downtime for Retubing Feedwater Heater Number 5

If feedwater heater number 5 is retubed rather than replaced, it will have to be removed from service for an extended period of time. The maximum allowable downtime for retubing may be calculated by dividing the additional feedwater heating essergy cost due to the heater downtime (Case C minus Case A) into the difference between the cost of replacing ($235,000) and retubing ($185,000). For example, for Trial 3 one obtains,

$$\frac{\$235{,}000 - \$185{,}000}{(481.774 - 465.942)\$/\text{hr}} = 3158 \text{ hours}$$

APPENDIX I

ENTROPY CREATION IN A CONDENSER TUBING WALL

An expression for the entropy created when heat is transferred through the finite temperature difference across a condenser tubing wall may be determined by making an entropy balance across a differential section of the tubing wall.

By the Second Law, the entropy created by a differential amount of heat $d\dot{Q}$ being transferred from a temperature of $T_{C,B}$ to a temperature of T_{FW} through a differential section of tubing wall dL is given by:

$$d\dot{S}_{ht} = \frac{d\dot{Q}}{T_{FW}} - \frac{d\dot{Q}}{T_{C,B}} = d\dot{Q}\ [\frac{1}{T_{FW}} - \frac{1}{T_{C,B}}] \qquad (I-1)$$

Since $d\dot{Q} = \dot{m}_{FW}\ C_{p,FW} dT_{FW}$, one may write,

$$dS_{ht} = m_{FW}\ C_{p,FW}\ [\frac{1}{T_{FW}} - \frac{1}{T_{C,B}}] dT_{FW} \qquad (I-2)$$

If $\dot{M}_{FW}$ and $C_{p,FW}$ are considered constant, equation (I-2) may be integrated directly from the entering feedwater temperature T_{FW} to the exiting feedwater temperature T_{Fwe} to obtain,

$$\dot{S}_{ht} = \int_i^e d\dot{S}_{ht} = \dot{m}_{FW}C_{P,FW} \int_{T_{FWi}}^{T_{FWe}} [\frac{1}{T_{FW}} - \frac{1}{T_{C,B}}]dT_{FW}$$

$$\dot{S}_{ht} = \dot{m}_{FW}C_{P,FW} [\ln \frac{T_{FWe}}{T_{FWi}} - \frac{T_{FWe}-T_{FWi}}{T_{C,B}}] \qquad \text{(I-3)}$$

Since the total heat transferred in the feedwater heater is given by,

$$\dot{Q}_{FH} = \dot{m}_{FW}C_{P,FW}(T_{FWe}-T_{FWi})$$

one obtains for the entropy creation $\dot{S}_{ht}$ due to heat transfer in the feedwater heater,

$$\dot{S}_{ht} = \dot{Q}_{FH} \left[\frac{\ln T_{Fwe}/T_{FWi}}{T_{FWe}-T_{FWi}} - \frac{1}{T_{C,B}}\right] \qquad \text{(I-4)}$$

APPENDIX J

The expression for feedwater heater cost given in equation (100) is of the following form:

$$y = a\chi + b\,\frac{1-e^{-\chi}}{C-e^{-\chi}} + d \qquad (J\text{-}1)$$

where:

$$y = \dot{C}$$

$$a = \dot{c}_z'$$

$$b = -\frac{c_{e,B} T_o \dot{Q}_{FH}}{T_{FWi}}$$

$$c = \frac{T_{FWe}}{T_{FWi}}$$

$$d = \frac{c_{e,B} T_o \ln \frac{T_{FWe}}{T_{FWi}}}{T_{FWi}\left(\frac{T_{FWe}}{T_{FWi}}-1\right)}$$

To determine the extreme of y in {χ}, differentiate equation (J-1) with respect to χ and equate to zero to obtain,

$$a(c-e^{-\chi})^2 + b(c-1)e^{-\chi} = 0 \qquad (J\text{-}2)$$

which may be rearranged to yield the following expression:

$$a(c-e^{-\chi})^2 - b(c-1)(c-e^{-\chi}) + bc(c-1) = 0 \qquad \text{(J-3)}$$

Equation (I-3) is of the form,

$$aW^2 + BW + C = 0 \qquad \text{(J-4)}$$

where:

$W = c-e^{-\chi}$

$B = -b(c-1)$

$C = bc(c-1)$

Equation (J-4) may be recognized as a quadratic which has the solutions,

$$W = \frac{-B \pm (B^2-4aC)^{1/2}}{2a} \qquad \text{(J-5)}$$

Substituting for the dummy variables B, C, and W in equation (J-5) yields,

$$c-e^{-\chi} = \frac{b(c-1) \pm \{b^2(c-1) - 4abc(c-1)\}^{1/2}}{2a} \qquad \text{(J-6)}$$

Substituting for the dummy variables a, b and c in equation (J-6) and solving for χ one obtains,

$$\chi = -\ln\left\{\left[\frac{c_{e,B}T_o\dot{Q}_{FH}}{T_{FWi}}\left(\frac{T_{FWe}}{T_{FWi}}-1\right) \pm \frac{T_{FWe}}{T_{FWi}}\left\{\left(\frac{c_{e,B}T_o\dot{Q}_{FH}}{T_{FWe}}\left(\frac{T_{FWe}}{T_{FWi}}-1\right)\right)^2 + 4\dot{c}_z'\left(\frac{c_{e,B}T_o\dot{Q}_{FH}}{T_{FWe}}\left(\frac{T_{FWe}}{T_{FWi}}-1\right)\right)\right\}^{1/2}\right]\frac{1}{2\dot{c}_z'} + \frac{T_{FWe}}{T_{FWi}}\right\} \tag{J-7}$$

To determine which solution is the valid expression for the optimum number of transfer units, first rearrange equation (J-7) to obtain,

$$\chi = -\ln\left\{\frac{T_{FWe}}{T_{FWi}}\left[\frac{c_{e,B}T_o\dot{Q}_{FH}}{T_{FWe}}\left(\frac{T_{FWe}}{T_{FWi}}-1\right) \pm \left\{\left(\frac{c_{e,B}T_o\dot{Q}_{FH}}{T_{FWe}}\left(\frac{T_{FWe}}{T_{FWi}}-1\right)\right)^2 + 4\dot{c}_z'\left(\frac{c_{e,B}T_o\dot{Q}_{FH}}{T_{FWe}}\left(\frac{T_{FWe}}{T_{FWi}}-1\right)\right)\right\}^{1/2}\right]\frac{1}{2\dot{c}_z'} + 1\right\} \tag{J-8}$$

Let $K = \frac{c_{e,B}T_o\dot{Q}_{FH}}{T_{FWe}}\left(\frac{T_{FWe}}{T_{FWi}} - 1\right)$, so that equation (J-8) becomes,

$$\chi = -\ln\left\{\frac{T_{FWe}}{T_{FWi}}\left[\frac{K \pm \{K^2 + 4\dot{c}_z'K\}^{1/2}}{2\dot{c}_z'} + 1\right]\right\} \tag{J-9}$$

Rearranging equation (J-9) one obtains,

$$\chi = -\ln\left[\frac{T_{FWe}}{T_{FWi}}\left(1 + \frac{K}{2\dot{c}_z'} \pm \left\{\left(\frac{K}{2\dot{c}_z'}\right)^2 + \frac{K}{\dot{c}_z'}\right\}^{1/2}\right)\right] \qquad \text{(J-10)}$$

Let $R = K/\dot{c}_z'$ so that equation (J-10) becomes,

$$\chi = -\ln\left[\frac{T_{FWe}}{T_{FWi}}\left(1+R/2\pm\{(R/2)^2+R\}\right)\right] \qquad \text{(J-11)}$$

In order for the number of transfer units χ to make sense it must be positive which implies that the following inequality must hold.

$$\ln\left[\frac{T_{FWe}}{T_{FWi}}\left(1+R/2\pm\{(R/2)^2+R\}^{1/2}\right)\right]<0 \qquad \text{(J-12)}$$

Since $\ln a<0$ only for $0\leq a<1$, equation (J-12) implies that,

$$\frac{T_{FWe}}{T_{FWi}}\left(1+R/2\pm\{(R/2)^2+R\}^{1/2}\right)<1 \qquad \text{(J-13)}$$

must hold. Since $T_{FWe}>T_{FWi}$ always, it is seen that $T_{FWe}/T_{FWi}>1$ always and since $c_{\ell,B}, T_o, \dot{Q}_{FH}$, T_{FWe} and T_{FWi} are always positive, one obtains $R>0$ always.

Now check the two solutions:

<u>Case I</u>

Consider,

$$\chi = \frac{T_{FWe}}{T_{FWi}}\left(1 + \frac{R}{2} + \{(\tfrac{R}{2})^2+R\}^{1/2}\right)$$

By observation,

$$1 + R/2 + \{(R/2)^2 + R\}^{1/2} > 1 \quad \text{for all } R>0$$

In view of the fact that $T_{FWe}/T_{FWi}>1$ always, one obtains,

$$\frac{T_{FWe}}{T_{FWi}} (1 + \frac{R}{2} + \{(\frac{R}{2})^2 + R\}^{1/2} > 1 \quad \text{always}$$

Therefore

$$X = - \ln[\frac{T_{FWe}}{T_{FWi}} (1 + \frac{R}{2} + \{(R/2)^2 + R\}^{1/2})] < 0$$

always for $R>0$ and $T_{FWe}/T_{FWi}>1$ is not a valid expression for the optimum number of transfer units.

Case II

Consider,

$$X = \frac{T_{FWe}}{T_{FWi}} (1 + R/2 - \{(R/2)^2 + R\}^{1/2})$$

By observation,

$$1 + \frac{R}{2} - \{(\frac{R}{2})^2 + R\}^{1/2} < 1 \quad \text{for all } R>0$$

In view of the fact that $T_{FWe}/T_{FWi}>1$ always, one has,

$$\frac{T_{FWe}}{T_{FWi}} \left(1 + \frac{R}{2} - \{(R/2)^2+R\}^{1/2}\right) < 1$$

for certain values of $R>0$ and $T_{FWe}/T_{FWi}>1$. Therefore,

$$\chi = -\ln\left[\frac{T_{FWe}}{T_{FWi}} \left(1+R/2-\{(R/2)^2+R\}^{1/2}\right)\right] > 0$$

for certain values of $R>0$ and $T_{FWe}/T_{FWi}>1$ and the optimum number of transfer units is given by,

$$\chi_{opt} = -\ln\left[\frac{T_{FWe}}{T_{FWi}} \left(1 + \frac{R}{2} - \{(\tfrac{R}{2})^2+R\}^{1/2}\right)\right] \qquad \text{(I-14)}$$

where:

$$R = K/\dot{c}_z'$$

$$K = \frac{c_{e,B} T_o \dot{Q}_{FH}}{T_{FWe}} \left(\frac{T_{FWe}}{T_{FWi}} - 1\right)$$

To determine the lower limit of validity for our expression of χ_{opt} given by equation (J-14), one must first recognize that the minimum value for χ_{opt} occurs as,

$$\frac{T_{FWe}}{T_{FWi}} \left[1 + \frac{R}{2} - \{(\tfrac{R}{2})^2+R\}^{1/2}\right] \to 1$$

Therefore, let the following expression hold:

$$\frac{T_{FWe}}{T_{FWi}} [1 + \frac{R}{2} - \{(\frac{R}{2})^2 + R\}^{1/2}] = 1$$

or
$$1 + \frac{R}{2} - \{(\frac{R}{2})^2 + R\}^{1/2} = \frac{T_{FWi}}{T_{FWe}}$$

Rearranging one gets,

$$\{(\frac{R}{2})^2 + R\}^{1/2} = 1 - \frac{T_{FWi}}{T_{FWe}} + \frac{R}{2}$$

Let $\alpha = 1 - T_{FWe}/T_{FWi}$ so that,

$$\alpha + R/2 = \{(R/2)^2 + R\}^{1/2}$$

Squaring both sides of this equation yields

$$(\alpha + R/2)^2 = (R/2)^2 + R$$

Expanding, cancelling like terms of opposite sign and rearranging will give,

$$R(1-\alpha) = \alpha^2$$

or
$$R = \frac{\alpha^2}{1-\alpha}$$

Substituting $\alpha = 1-T_{FWi}/T_{FWe}$ into the above expression and rearranging will yield,

$$R = \frac{(T_{FWe}-T_{FWi})^2}{T_{FWi}T_{FWe}}$$

Therefore, R approaches $(T_{FWe}-T_{FWi})^2/T_{FWe}T_{FWi}$ as χ_{opt} approaches its lower limit of validity.

From the earlier definition of R one obtains,

$$R = \frac{c_{e,B}T_o\dot{Q}_{FH}}{\dot{c}'_{z,opt}T_{FWe}}\left(\frac{T_{FWe}}{T_{FWi}} - 1\right)$$

Substituting $\dot{c}'_{z,opt} = \dot{m}_{FW}C_{p,FW}\dot{c}'_A/U_{opt}$ and $\dot{Q}_{FH} = \dot{m}_{FW}C_{p,FW}(T_{FWe}-T_{FWi})$ into this expression for R and rearranging will give,

$$R = \frac{c_{e,B}T_o}{\dot{c}'_A/U_{opt}}\,\frac{(T_{FWe}-T_{FWi})^2}{T_{FWi}T_{FWe}}$$

Equating this value for R to our earlier determined value that R approaches as the expression for χ_{opt} approaches its lower limit of validity will yield,

$$\frac{c_{e,B}}{\dot{c}_A/T_oU_{opt}}\,\frac{(T_{FWe}-T_{FWi})^2}{T_{FWi}T_{FWe}} = \frac{(T_{FWe}-T_{FWi})^2}{T_{FWi}T_{FWe}}$$

Cancelling like terms one obtains,

$$\frac{c_{e,B}}{\dot{c}_A'/T_o U_{opt}} = 1$$

or

$$c_{e,B} = \dot{c}_A'/T_o U_{opt}$$

Recall from Chapter V it was determined that, $\dot{c}_A' = 1.4\ \dot{c}_A$ so that,

$$c_{e,B} = \frac{1.4\ \dot{c}_A}{T_o U_{opt}}$$

Therefore, the lower limit of validity for the expression for χ_{opt} is approached for

$$R \rightarrow \frac{(T_{FWe} - T_{FWi})^2}{T_{FWe}\ T_{FWi}}$$

or

$$c_{e,B} \rightarrow \frac{1.4\ \dot{c}_A}{T_o U_{opt}}$$

In order to determine the upper limit of validity of the expression for χ_{opt}, one must investigate its behavior as R approaches infinity. That is, one must investigate,

$$\lim_{R\to\infty} \chi_{opt} = \lim_{R\to\infty} \left\{ -\ln\left(\frac{T_{FWe}}{T_{FWi}} \left\{ (1 + \frac{R}{2} - \{ (\frac{R}{2})^2 + R \}^{1/2} \right\} \right) \right\}$$

The operand of the logarithm in above expression may be rearranged as follows:

$$\frac{T_{FWe}}{T_{FWi}} (1+\frac{R}{2} - \{ (\frac{R}{2})^2 + R \}^{1/2}) = \frac{T_{FWe}}{T_{FWi}} (1 + \frac{R}{2} - \frac{R}{2} \{1 + \frac{4}{R}\}^{1/2})$$

Using series expansion one obtains the following expression for the term $\{1 + 4/R\}^{1/2}$

$$\{1+4/R\}^{1/2} = 1 + \frac{2}{R} - \frac{2}{R^2} + \ldots$$

Neglecting all terms with a power of three or greater will give,

$$\{1+4/R\}^{1/2} \cong 1+2/R-2/R^2$$

Substituting this expression into the operand of the logarithm will yield,

$$\frac{T_{FWe}}{T_{FWi}} (1 + \frac{R}{2} - \frac{R}{2} \{1 + \frac{4}{R}\}^{1/2}) = \frac{T_{FWe}}{T_{FWi}} [1 + \frac{R}{2} - \frac{R}{2} (1 + \frac{2}{R} - \frac{2}{R^2})]$$

If like terms with opposite sign in the above equation are cancelled one obtains,

$$\frac{T_{FWe}}{T_{FWi}} \left(1 + \frac{R}{2} - \frac{R}{2} \{1 + \frac{4}{R}\}^{1/2}\right) = \frac{T_{FWe}}{T_{FWi}} \frac{1}{R}$$

Now, for the limit of χ_{opt} as R approaches infinity one gets,

$$\lim_{R\to\infty} \chi_{opt} = \lim_{R\to\infty} \left(-\ln\left[\frac{T_{FWe}}{T_{FWi}} \frac{1}{R}\right]\right)$$

Evaluating this limit will give,

$$\lim_{R\to\infty} \chi_{opt} = -\ln(0) = -(-\infty) = +\infty$$

Therefore, it is easily seen that the upper limit of validity for the expression for χ_{opt} is positive infinity. To summarize all of the above development, the expression for the optimum number of transfer units is given by,

$$\chi_{opt} = -\ln \left[\frac{T_{FWe}}{T_{FWi}} \left(1 + \frac{R}{2} - \{(\frac{R}{2})^2 + R\}^{1/2}\right)\right]$$

where:

$$R = \frac{c_{e,B} T_o \dot{Q}_{FH}}{\dot{c}'_{z,opt} T_{FWe}} \left(\frac{T_{FWe}}{T_{FWi}} - 1\right)$$

$$\dot{c}'_z = \dot{m}_{FW} C_{p,FW} \dot{c}''_A / U_{opt}$$

for $$(T_{FWe}-T_{FWi})^2/T_{FWi}T_{FWe}<R<\infty$$

or $$c_{e,B} > \frac{1.4\ \dot{c}_A}{T_o U_{opt}}$$

APPENDIX K

SPECIAL CASES OF ESSERGY

Figure K-1 is used by permission from Evans (1969) and presents the special cases of essergy developed by other workers in the field.

Table K-1. Connections Among Essergy, Availability, Exergy and Free Energy

NAME	FUNCTION	COMMENTS
ESSERGY	$E + P_oV - T_oS - \sum_c \mu_{co} N_c$	This function was formulated for the special case of an existing medium in 1878 (by Gibbs) and in general in 1962 (Ref. 12). Its name was changed from "available energy" to "exergy" in 1963, and from "exergy" to "essergy" (i.e., "essence of energy") in 1968.
AVAILABILITY	$E + P_oV - T_oS - (E_o + P_oV_o - T_oS_o)$	Formulated by Keenan in 1941, this function is shown on page 32 to be a special case of the essergy function.
EXERGY	$E + PV - T_oS - (E_o + P_oV_o - T_oS_o)$	Introduced by Darrieus (1930) and Keenan (1932), this function (which Keenan has called the "availability in steady flow") was given the name "exergy" by Rant in 1956. As shown on page 39, this function is a special case of essergy.
FREE ENERGY	HELMHOLTZ: $E - TS$ GIBBS: $E + PV - TS$	The functions E-TS and E+PV-TS were introduced by von Helmholtz and Gibbs (1873). These two functions are Legendre transforms of energy which were shown by Gibbs to yield useful alternate criteria of equilibrium. As measures of the potential work of systems, these two functions are shown on page 44 to represent special cases of the essergy function.

APPENDIX L

COMPUTER CODE

This appendix presents the code for the digital computation used in this study. Program BH1 calculates turbine stage capital costs and makes the plot in Figure 9, the dimensionless function $f(T_{in})$ which represents the effect on capital cost of maximum turbine operating temperature. Program WAH2 performs simultaneous solution of the power cycle economic balance equations. Program BH2 calculates optimum cost per transfer unit, velocity and number of transfer units for a feedwater heater with the same operating conditions as feedwater heater number 6 and makes the plots shown in Figures 14 through 18.

```
      PROGRAM BH1 (INPUT,OUTPUT,CALCOM,TAPE5=INPUT,TAPE6=OUTPUT,
     1TAPE9=CALCOM)
      DIMENSION TX(152),FTIN(152),IBUF(512),EFFT(8),HIN(8),HOUT(8),
     1FM(8),C(8),TEN(8),FTEN(8),TURB(8),WS(8),ES(8),EIN(8),EOUT(8)
      READ *,TR,CKT,A,TO,B
      DO 50 I = 1,8
   50 READ *,TEN(I),EFFT(I),HIN(I),HOUT(I),FM(I),EIN(I),EOUT(I)
      DO 100 I = 10,120
      TIN = FLOAT(I)
      TX(I) = TIN*10.0
      FTIN(I) = 1.0 + ((TX(I)-TO)/(TR-TO))**B
  100 CONTINUE
      GO TO 201
      WRITE (6,150)
  150 FORMAT (8X,'T',8X,'F(T)',/,5X,'------',5X,'-------',/)
      DO 200 I = 10,120
      WRITE (6,160) TX(I),FTIN(I)
  160 FORMAT (5X,F6.1,5X,F6.4)
  200 CONTINUE
  201 DO 400 I = 1,8
      FTEN(I) = 1.0 + ((TEN(I)-TO)/(TR-TO))**B
      WS(I) = FM(I)*(HIN(I)-HOUT(I))
      ES(I) = FM(I)*(EIN(I)-EOUT(I))
      IF (I.NE.2) GO TO 350
      WS(I) = WS(I) + 0.032498*(1464.9 - 1423.8)
      ES(I) = ES(I) + (.032498*(679.2-EOUT(I)))
  350 C(I) = CKT*FTEN(I)*((1.0 - EFFT(I))**(-A))*ES(I)
      WRITE (6,351)FTEN(I),EFFT(I),ES(I)
  351 FORMAT (F8.6,3X,F5.3,3X,F8.3,/)
  400 CONTINUE
      DO 425 I = 1,8
      READ (5,420) TURB(I)
  420 FORMAT (A4)
  425 CONTINUE
      WRITE (6,450)
  450 FORMAT (//,5X,'TURBINE STAGE',5X,'CAPITAL COST,$/HR',
     1/,5X,'-------------',5X,'-----------------',/)
      DO 500 I = 1,8
      WRITE (6,475) TURB(I),C(I)
  475 FORMAT (10X,A4,14X,F8.4,/)
  500 CONTINUE
      CALL PLOTS (IBUF,512,9,50)
      CALL PLOT (1.0,1.0,-3)
      CALL FACTOR (0.75)
      CALL SCALE (TX(10),8.0,111,1)
      CALL SCALE (FTIN(10),9.0,111,1)
      CALL AXIS (0.0,0.0,'INLET TEMP(DEGREES F)',-21,8.0,0.0,
     1TX(121),TX(122))
      CALL AXIS (0.0,0.0,'FUNC OF INLET TEMP',18,9.0,90.0,
     10.0,FTIN(122))
      CALL PLOT (0.0,1.0,-3)
      CALL LINE (TX(10),FTIN(10),111,1,10,1)
      CALL PLOT (0,0,999)
      END
```

```
      PROGRAM BH2 (INPUT,OUTPUT,CALCOM,TAPE5=INPUT,TAPE6=OUTPUT,
     1TAPE9=CALCOM)
      DIMENSION V(202),CAU(202),C(202),X(202),CV(202)
      DIMENSION CAPT(202),ESST(202),CAPV(202),ESSV(202),CAUTR(202)
      DIMENSION TMTO(10),CACS(10,202),TUBN(10,202),CAD(10)
      DIMENSION CBD(10,102),AOPT(10,102),IBUF(512)
      READ *,CAY,TIN,TOUT,EPI,EPO,FP,CS,WSP,CAPP
      READ *,TO,RHO,VIS,DIA,THK,CP,HIN,HOUT,CB
      TM = (TIN + TOUT)/2.0
      CON = 365.0*24.0*3600.0
      CA = CAY/CON
      VISS = VIS/3600.0
      THKS = THK/3600.0
      ESO = FP*(EPO-EPI)
      CE = (CAPP+(CS*WSP))/(ESO*778.16)
      CKV = 0.023*(TO/TM)*CE*((VISS/DIA)**0.2)*((RHO**0.8)/32.174)
      PR = (VIS*CP)/THK
      CKH = 0.023*(PR**0.333)*((RHO/VISS)**0.8)*(THKS/(DIA**0.2))
      DO 10 I = 10,200
      V(I) = FLOAT(I)
      V(I) = V(I)/10.0
      CAU(I) = ((CA*(V(I)**(-0.8)))/CKH)+((CKV*(V(I)**2.0))/CKH)
   10 CONTINUE
      VOPT = ((0.4*CA)/CKV)**(1.0/2.8)
      CAUOPT = ((CA*(VOPT**(-0.8)))/CKH)+((CKV*(VOPT**2.0))/CKH)
      CAUOTR = CAUOPT*100000000.0
      DO 15 I = 10,200
   15 CAUTR(I) = CAU(I)*100000000.0
      WRITE (6,16)
   16 FORMAT (10X,'VELOCITY',5X,/,10X,'--------',5X,'---------',//)
      DO 18 I = 10,200
      WRITE (6,17) V(I),CAUTR(I)
   17 FORMAT (11X,F5.2,7X,F8.5)
   18 CONTINUE
      CZOPT = CAUOPT*FP*CP
      WRITE (6,19) VOPT,CAUOTR,CZOPT
   19 FORMAT (//,5X,'OPTIMUM VELOCITY = ',F8.5,/,22X,'= ',F8.5,/,22X
     1,'= ',F8.5,)
      Q = FP*(HOUT-HIN)
      TRATIO = TOUT/TIN
      CETQ = (CP*TO*Q)/1000000.0
      CETQIN = CETQ/TIN
      CETQOUT = CETQ/TOUT
      CETQO = CETQIN*(TRATIO-1.0)
      CETQT = CETQOUT*(TRATIO-1.0)
      DISCR = SQRT((CETQT**2.0) + (4.0*CZOPT*CETQT))
      XOPTT = -ALOG(((CETQO-(TRATIO*DISCR))/(2.0*CZOPT))+TRATIO)
      WRITE (6,27) XOPTT
   27 FORMAT (5X,'OPTIMUM NTU       = ',F8.5,////)
      EK = (ALOG(TRATIO))/(TRATIO-1.0)
      DO 20 I = 1,200
      X(I) = FLOAT(I)
      X(I) = X(I)/10.0
      ECR = (1.0-EXP(-X(I)))/(TRATIO-EXP(-X(I)))
      CAPT(I) = CZOPT*X(I)
      ESST(I) = CETQIN*(EK-ECR)
      C(I) = CAPT(I)+ESST(I)
   20 CONTINUE
      WRITE (6,21)
   21 FORMAT (7X,'NTU',6X,'CAPITAL COST TERM',3X
     1,'ESSERGY COST TERM',5X,'   COST   ',/,4X,'----------',
     23X,'-----------------',3X,'-----------------',5X,'-----------',//)
      DO 23 I = 1,200
```

```
      WRITE (6,22) X(I),CAPT(I),ESST(I),C(I)
   22 FORMAT (5X,F8.5,8X,F8.5,13X,F8.5,10X,F8.5)
   23 CONTINUE
      ECROPT = (1.0-EXP(-XOPTT))/(TRATIO-EXP(-XOPTT))
      COPT = (CZOPT*XOPTT)+(CETQIN*(EK-ECROPT))
      WRITE (6,24) COPT
   24 FORMAT (//,5X,'OPTIMUM COST      = ',F8.5,////)
      DO 30 I = 10,200
      CZ = CAU(I)*FP*CP
      DISCRV = SQRT((CETQT**2.0)+(4.0*CZ*CETQT))
      XV = -ALOG(((CETQO-(TRATIO*DISCRV))/(2.0*CZ))+TRATIO)
      ECRV = (1.0-EXP(-XV))/(TRATIO-EXP(-XV))
      CAPV(I) = CZ*XV
      ESSV(I) = CETQIN*(EK-ECRV)
      CV(I) = CAPV(I)+ESSV(I)
   30 CONTINUE
      WRITE (6,31)
   31 FORMAT (////,5X,'VELOCITY',14X,'CAPITAL COST TERM',3X
     1,'ESSERGY COST TERM',5X,'    COST    ',/,5X,'--------',3X
     2,'--------'
     3,3X,'-------------------',3X,'-----------------'
     4,5X,'----------',//)
      DO 33 I = 10,200
      WRITE (6,32) V(I),CAUTR(I),CAPV(I),ESSV(I),CV(I)
   32 FORMAT (6X,F5.2,5X,F8.5,8X,F8.5,12X,F8.5,10X,F8.5)
   33 CONTINUE
      AT = (3.14159*(DIA**2.0))/4.0
      DO 200 J = 1,10
      READ *,TMTO(J),RHON,VISN
      PRINT *,TMTO(J)
      WRITE (6,75) TMTO(J)
   75 FORMAT (//,5X,'TM = ',F6.1,//)
      DO 100 I = 10,200
      CACS(J,I) = FLOAT(I)
      CACS(J,I) = CACS(J,I)/100.0
      VO = (0.4*32.174*(DIA**0.2))/(0.023*(RHON**0.8)*(VISN**0.2))
      VOP = (VO*TMTO(J)*CACS(J,I)*24.675)**(1.0/2.8)
      TUBN(J,I) = FP/(VOP*RHON*AT*3600.0)
      WRITE (6,80) CACS(J,I),TUBN(J,I)
   80 FORMAT (5X,F5.2,5X,F7.2)
  100 CONTINUE
  200 CONTINUE
      DO 300 I = 1,10
      CAD(I) = FLOAT(I)
      WRITE (6,225) CAD(I)
  225 FORMAT (//,10X,F5.2,///)
      CAD(I) = CAD(I)/(365.0*24.0*3600.0)
      VOD = ((0.4*CAD(I))/CKV)**(1.0/2.8)
      CAUD = (1.4*CAD(I)*(VOD**(-0.8)))/CKH
      CZD = FP*CP*CAUD
      U = CKH*(VOD**0.8)
      DO 250 J =10,100
      CBD(I,J) = FLOAT(J)
      CBD(I,J) = CBD(I,J)/10.0
      CBTQO = ((CBD(I,J)*TO*Q)/TIN)*(TRATIO-1.0)
      CBTQT = ((CBD(I,J)*TO*Q)/TOUT)*(TRATIO-1.0)
      CBTQO = CBTQO/1000000.0
      CBTQT = CBTQT/1000000.0
      DISCRD = SQRT((CBTQT**2.0)+(4.0*CZD*CBTQT))
      ATERM = -ALOG(((CBTQO-(TRATIO*DISCRD))/(2.0*CZD))+TRATIO)
      AOPT(I,J) = (FP*CP*ATERM)/(U*3600.0)
      WRITE (6,226) CBD(I,J),AOPT(I,J)
  226 FORMAT (5X,F5.2,5X,F10.2)
  250 CONTINUE
  300 CONTINUE
```

```
 CALL PLOTS (IBUF,512,9,00)
 CALL PLOT (1.5,1.5,-3)
 CALL FACTOR (0.5)
 CALL PLOTMX(20.0)
 CALL AXIS (0.0,0.0,'NO. OF TRANSFER UNITS',-21,
110.0,0.0,0.0,2.0)
 CALL AXIS (0.0,0.0,'COST($/HR)',10,13.0,90.0,
10.0,1.0)
 X(201) = 0.0
 X(202) = 2.0
 CAPT(201) = 0.0
 CAPT(202) = 1.0
 ESST(201) = 0.0
 ESST(202) = 1.0
 C(201) = 0.0
 C(202) = 1.0
 CALL LINE (X(10),CAPT(10),191,1,10,1)
 CALL LINE (X(10),ESST(10),191,1,10,1)
 CALL LINE (X(10),C(10),191,1,10,1)
 CALL PLOT (20.0,0.0,-3)
 CALL AXIS (0.0,0.0,'VELOCITY(FT/SEC)',-16,
110.0,0.0,0.0,2.0)
 CALL AXIS (0.0,0.0,'COST($/HR)',10,12.0,
190.0,6.0,0.25)
 V(201) = 0.0
 V(202) = 2.0
 CV(201) = 6.0
 CV(202) = 0.25
 CALL LINE (V(10),CV(10),191,1,10,1)
 CALL PLOT (20.0,0.0,-3)
 CALL AXIS (0.0,0.0,'VELOCITY(FT/SEC)',-16,
110.0,0.0,0.0,2.0)
 CALL AXIS (0.0,0.0,'C/U     ($F/BTU)',17,10.0,
190.0,5.0,0.5)
 CAUTR(201) = 5.0
 CAUTR(202) = 0.5
 CALL LINE (V(60),CAUTR(60),141,1,10,1)
 CALL PLOT (20.0,0.0,-3)
 CALL FACTOR (0.8)
 CALL AXIS (0.0,0.0,'     ',-5,10.0,0.0,0.0,0.2)
 CALL AXIS (0.0,0.0,'     ',5,10.0,90.0,0.0,50.0)
 CACS(1,201) = 0.0
 CACS(1,202) = 0.2
 CACS(2,201) = 0.0
 CACS(2,202) = 0.2
 CACS(3,201) = 0.0
 CACS(3,202) = 0.2
 CACS(4,201) = 0.0
 CACS(4,202) = 0.2
 CACS(5,201) = 0.0
 CACS(5,202) = 0.2
 CACS(6,201) = 0.0
 CACS(6,202) = 0.2
 CACS(7,201) = 0.0
 CACS(7,202) = 0.2
 CACS(8,201) = 0.0
 CACS(8,202) = 0.2
 CACS(9,201) = 0.0
 CACS(9,202) = 0.2
 CACS(10,201) = 0.0
 CACS(10,202) = 0.2
 TUBN(1,201) = 0.0
 TUBN(1,202) = 50.0
 TUBN(2,201) = 0.0
```

```
TUBN(2,202) = 50.0
TUBN(3,201) = 0.0
TUBN(3,202) = 50.0
TUBN(4,201) = 0.0
TUBN(4,202) = 50.0
TUBN(5,201) = 0.0
TUBN(5,202) = 50.0
TUBN(6,201) = 0.0
TUBN(6,202) = 50.0
TUBN(7,201) = 0.0
TUBN(7,202) = 50.0
TUBN(8,201) = 0.0
TUBN(8,202) = 50.0
TUBN(9,201) = 0.0
TUBN(9,202) = 50.0
TUBN(10,201) = 0.0
TUBN(10,202) = 50.0
CALL LINE (CACS(1,10),TUBN(1,10),191,10,10,1)
CALL LINE (CACS(2,10),TUBN(2,10),191,10,10,1)
CALL LINE (CACS(3,10),TUBN(3,10),191,10,10,1)
CALL LINE (CACS(4,10),TUBN(4,10),191,10,10,1)
CALL LINE (CACS(5,10),TUBN(5,10),191,10,10,1)
CALL LINE (CACS(6,10),TUBN(6,10),191,10,10,1)
CALL LINE (CACS(7,10),TUBN(7,10),191,10,10,1)
CALL LINE (CACS(8,10),TUBN(8,10),191,10,10,1)
CALL LINE (CACS(9,10),TUBN(9,10),191,10,10,1)
CALL LINE (CACS(10,10),TUBN(10,10),191,10,10,1)
CALL PLOT (20.0,0.0,-3)
CALL FACTOR (0.5)
CALL AXIS (0.0,0.0,'          ',-5,10.0,0.0,0.0,1.0)
CALL AXIS (0.0,0.0,'          ',5,14.0,90.0,0.0,250.0)
CBD(1,101) = 0.0
CBD(1,102) = 1.0
CBD(2,101) = 0.0
CBD(2,102) = 1.0
CBD(3,101) = 0.0
CBD(3,102) = 1.0
CBD(4,101) = 0.0
CBD(4,102) = 1.0
CBD(5,101) = 0.0
CBD(5,102) = 1.0
CBD(6,101) = 0.0
CBD(6,102) = 1.0
CBD(7,101) = 0.0
CBD(7,102) = 1.0
CBD(8,101) = 0.0
CBD(8,102) = 1.0
CBD(9,101) = 0.0
CBD(9,102) = 1.0
CBD(10,101) = 0.0
CBD(10,102) = 1.0
AOPT(1,101) = 0.0
AOPT(1,102) = 250.0
AOPT(2,101) = 0.0
AOPT(2,102) = 250.0
AOPT(3,101) = 0.0
AOPT(3,102) = 250.0
AOPT(4,101) = 0.0
AOPT(4,102) = 250.0
AOPT(5,101) = 0.0
AOPT(5,102) = 250.0
AOPT(6,101) = 0.0
AOPT(6,102) = 250.0
AOPT(7,101) = 0.0
```

```
ADPT(7,102) = 250.0
ADPT(8,101) = 0.0
ADPT(8,102) = 250.0
ADPT(9,101) = 0.0
ADPT(9,102) = 250.0
ADPT(10,101) = 0.0
ADPT(10,102) = 250.0
CALL LINE (CBD(1,10),ADPT(1,10),91,10,10,1)
CALL LINE (CBD(2,10),ADPT(2,10),91,10,10,1)
CALL LINE (CBD(3,10),ADPT(3,10),91,10,10,1)
CALL LINE (CBD(4,10),ADPT(4,10),91,10,10,1)
CALL LINE (CBD(5,10),ADPT(5,10),91,10,10,1)
CALL LINE (CBD(6,10),ADPT(6,10),91,10,10,1)
CALL LINE (CBD(7,10),ADPT(7,10),91,10,10,1)
CALL LINE (CBD(8,10),ADPT(8,10),91,10,10,1)
CALL LINE (CBD(9,10),ADPT(9,10),91,10,10,1)
CALL LINE (CBD(10,10),ADPT(10,10),91,10,10,1)
CALL PLOT (0,0,999)
END
```

```
      PROGRAM WAH2 (INPUT,OUTPUT,TAPE5=INPUT,TAPE6=OUTPUT)
      DIMENSION HC(30),CAP(30),WS(30),RAT(30),RATC(30),HCBL(30),COMP(30),
      DIMENSION HCWS(30),H(30),FM(30)
      READ *,FMF,FMFW,FMFHP,FMFRH,CF,CFW,EFW,EF
      READ *,FMCA,FMCAHP,FMCARH,CCA,ECA
      HCF = FMF*EF*CF
      HCFW = CFW*FMFW*EFW
      HCFHP = FMFHP*EF*CF
      HCFRH = FMFRH*EF*CF
      DO 4 I = 1,29
    4 READ *,CAP(I),WS(I)
      DO 5 I = 2,11
      READ *,H(I),FM(I)
      IF (I.EQ.2) GO TO 5
      IF (I.EQ.4) GO TO 5
      IF (I.EQ.5) GO TO 5
      WS(I) = FM(I)*(H(I-1)-H(I))
    5 CONTINUE
      WS(5) = FM(4)*(H(4)-H(5))+0.032498*(1464.9-H(5))
      DO 6 I = 2,11
    6 WS(1) = WS(1) + WS(I)
      DO 7 I = 12,29
    7 WS(1) = WS(1) - WS(I)
      DO 10 I = 2,29
   10 CAP(1) = CAP(1) + CAP(I)
      HC(1) = CAP(1) + HCF
      CS = HC(1)/WS(1)
      WRITE (6,300) CS
  300 FORMAT (/,5X,'CS = ',F13.10,/)
      HCWS(1) = CS*WS(1)
      DO 15 I = 3,11
      IF (I.EQ.5) GO TO 15
      READ *,FLOW,FLOW1,FLOW2
      RAT(I) = FLOW1/FLOW
      RATC(I) = FLOW2/FLOW
   15 CONTINUE
      HCCA = FMCA*ECA*CCA
      WRITE (6,600) HCCA
  600 FORMAT (F13.8,///)
   16 HCCAHP = FMCAHP*ECA*CCA
      HCCARH = FMCARH*ECA*CCA
      WRITE (6,650) HCCAHP,HCCARH,HCFW
  650 FORMAT (/,5X,F13.8,5X,F13.8,5X,F13.8,/)
      DO 30 I = 2,11
      IF (I.EQ.2) GO TO 22
      IF (I.EQ.3) GO TO 25
      IF (I.EQ.4) GO TO 24
      IF (I.EQ.5) GO TO 23
      IF (I.GE.6) GO TO 25
   22 HC(I) = CAP(I) + HCFHP + HCFW + HCCAHP
      GO TO 30
   23 CTERM = (32498.0/1869086.0)*HC(I-3)
      HC(I) = CAP(I) + HC(I-1) + CTERM - CS*WS(I)
      HCWS(I) = CS*WS(I)
      GO TO 30
   24 TERMO = HC(I-1)*(1.0 - (13085.0/1830179.0))
      TERMI = (3837.0/1869086.0)*HC(I-2)
      HCBRH = TERMO + TERMI
      HC(I) = CAP(I) + RAT(I)*HCBRH + HCFRH + HCCARH
      GO TO 30
   25 HC(I) = CAP(I) + RAT(I)*HC(I-1) - CS*WS(I)
      HCWS(I) = CS*WS(I)
   30 CONTINUE
```

```
      XMIX = (1072.0/1869086.0)*HC(2)
      YMIX = (3648.0/1830179.0)*HC(3)
      ZMIX = (2253.0/1579448.0)*HC(6)
      SMIX = XMIX + YMIX + ZMIX
      HCBL(13) = RATC(11)*HC(10)
      BLEE = RATC(10)*HC(9) + (1618.0/6973.0)*SMIX
      HCBL(15) = (47113.0/105113.0)*BLEE
      HCBL(17) = RATC(9)*HC(8)
      HCBL(20) = RATC(8)*HC(7)
      HCBL(22) = RATC(7)*HC(6) + (9437.0/1830179.0)*HC(3)
      HCBL(26) = RATC(6)*HC(5)
      HCBL(27) = RATC(4)*HCBRH
      HCBL(2) = HCBL(27)
      HCBL(4) = HCBL(26)
      HCBL(5) = HCBL(22)
      HCBL(6) = HCBL(20)
      HCBL(7) = HCBL(17)
      HCBL(8) = HCBL(15)
      HCBL(9) = HCBL(13)
      DO 40 I = 12,27
      IF (I.EQ.14) GO TO 35
      IF (I.EQ.16) GO TO 35
      IF (I.EQ.18) GO TO 35
      IF (I.EQ.21) GO TO 35
      IF (I.EQ.23) GO TO 35
      IF (I.EQ.24) GO TO 35
      SUM = CAP(I) + HC(I-1)
      IF (I.EQ.12) GO TO 32
      IF (I.EQ.19) GO TO 31
      IF (I.EQ.25) GO TO 31
      HC(I) = SUM + HCBL(I)
      GO TO 40
   31 HC(I) = SUM + CS*WS(I)
      HCWS(I) = CS*WS(I)
      GO TO 40
   32 TERMX = (1500.0/1869096.0)*HC(2)
      TERMY = (5355.0/6973.0)*SMIX
      SUM = SUM + TERMX + TERMY
      HC(I) = SUM + CS*WS(I)
      HCWS(I) = CS*WS(I)
      GO TO 40
   35 HC(I) = HC(I-1)
   40 CONTINUE
      HC(28) = CAP(28) + (0.0253/1.418483)*HC(8)
      HC(29) = CAP(29) + HC(28) + (0.058/0.105113)*BLEE
      IF (ABS(HC(29)-HCCA).LE.0.0001) GO TO 55
      HCCA = HC(29)
      CCA = HC(29)/(ECA*FMCA)
      GO TO 16
   55 HCWS(16) = HCWS(19)
      HCWS(19) = HCWS(25)
      DO 56 I = 2,29
   56 HC(I-1) = HC(I)
      J = 13
      DO 60 I = 14,29
      IF (I.EQ.14) GO TO 60
      IF (I.EQ.16) GO TO 60
      IF (I.EQ.18) GO TO 60
      IF (I.EQ.21) GO TO 60
      IF (I.EQ.23) GO TO 60
      IF (I.EQ.24) GO TO 60
      J = J + 1
```

```
      CAP(J) = CAP(I)
   60 CONTINUE
      DO 360 I = 1,11
      WRITE (6,355) WS(I),HCWS(I),HC(I)
  355 FORMAT (5X,F15.8,5X,F15.8,5X,F15.8,/)
  360 CONTINUE
      DO 65 I = 1,23
      READ (5,64) COMP(I)
   64 FORMAT (A5)
   65 CONTINUE
      READ (5,66) ALP
   66 FORMAT (A3)
      WRITE (6,67) ALP
   67 FORMAT (3X,'CASE ',A3,/)
      WRITE (6,75)
   75 FORMAT (15X,'***CAPITAL AND POWER COST OF PLANT EQUIPMENT***',//)
      WRITE (6,80)
   80 FORMAT (9X,'COMPONENT',10X,'CAPITAL COST,$/HR',10X,
     1 'POWER COST,$/HR')
      WRITE (6,85)
   85 FORMAT (9X,'---------',10X,'-----------------',10X,
     1'---------------',/)
      DO 90 I = 1,23
      WRITE (6,86) COMP(I),CAP(I),HCWS(I)
   86 FORMAT (11X,A5,17X,F8.3,17X,F8.3,/)
   90 CONTINUE
      DO 92 I = 1,26
      READ (5,91) COMP(I)
   91 FORMAT (A4)
   92 CONTINUE
      WRITE (6,94) HCF,HC(29)
   94 FORMAT (///,9X,'TOTAL HOURLY COST OF FUEL              =   ',
     1F11.6,/,9X,'TOTAL HOURLY COST OF COMBUSTION AIR  =   ',
     2F11.6,////////)
      WRITE (6,67) ALP
      WRITE (6,95)
   95 FORMAT (20X,'***HOURLY COSTS FOR FEEDWATER HEATING***',//)
      WRITE (6,100)
  100 FORMAT (23X,'POINT',10X,'HOURLY COST,$/HR')
      WRITE (6,105)
  105 FORMAT (23X,'-----',10X,'----------------',/)
      DO 110 I = 1,26
      IF (I.EQ.2) GO TO 107
      IF (I.EQ.4) GO TO 107
      IF (I.EQ.5) GO TO 107
      IF (I.EQ.6) GO TO 107
      IF (I.EQ.7) GO TO 107
      IF (I.EQ.8) GO TO 107
      IF (I.EQ.9) GO TO 107
      WRITE (6,106) COMP(I),HC(I)
  106 FORMAT (24X,A4,12X,F9.4,/)
      GO TO 110
  107 WRITE (6,108) COMP(I),HCBL(I)
  108 FORMAT (24X,A4,12X,F9.4,/)
  110 CONTINUE
      WRITE (6,120)
  120 FORMAT (////)
      END
```

APPENDIX M

POST-DISSIPATION CONCEPT FOR TERMINAL ZONES*

Treatment of essergy outputs from terminal zones which because of physical or economic necessity are thrown away (exhausted and allowed to dissipate in the environment for no useful purpose) are difficult to handle conceptually. For this study, their essergy and economic values were viewed as being worth nothing (equal to zero) when calculating effectiveness and economic balance. This viewpoint assures that the values calculated for effectiveness and economic balance will reflect the essergy and economic values that are by necessity thrown away. This concept is misleading since the outputs do have essergy and economic values as is evidenced when the situation warrants their application to some useful purpose such as space heating or absorption cooling in a total energy system.

Perhaps a better conceptual way to view this type of essergy output is to extend the system boundary for the terminal zone so that the dissipation of the essergy output in the environment is included inside the zone. For this case, there will no longer be an essergy output but simply

---

*Terminal zones are those which have one or more outputs which leave the power plant.

additional dissipation in the zone known as "post-dissipation." This viewpoint represents a more sound rationale since this added dissipation is recognized as zone essergy or economic "cost" which should be reduced if possible. If an economical means is found for reducing the "post-dissipation," the system boundary is simply moved to show the reduction in "post-dissipation" as an essergy output to a newly formed terminal zone which includes the remainder of the "post-dissipation" within its system boundary.

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