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1. ABSTRACT

Costs and heat rates of economically "balanced" steam power plants have been determined as functions of initial steam conditions and incremental heat cost. Immediate determination of optimum power plant investment and of thermodynamic performance is possible for a wide range of initial steam conditions if incremental Btu costs are known.

This study covers several classes of cycles, for each of which (1) the turbine exhaust annulus area and (2) the condenser surface and circulating water system capacity have been varied in addition to the initial steam conditions. Circulating water temperature is assumed constant at 65 F. All cycles considered have gross capabilities of 150 MW.

The significance of the results may be judged by the fact that for the typical initial state of 600 psia saturated, investment costs vary from \$88.60 to \$98.70 per kilowatt of installed capacity for a corresponding change in heat rate from 10,610 to 10,128 Btu/kwh. The incremental costs of heat at which these two investments are justified are \$0.40 and \$0.60 per million Btu, respectively.

Figures F.1 to F.6 summarize the results of this study.

2. INTRODUCTION

A principal problem in the selection of apparatus comprising a power plant is that of economic optimization. This problem relates to the basic choice of the steam cycle and, within that choice, to the design of its components. Each piece of equipment has a range of performance which is technically possible, and within this range improvement in performance is achieved at the expense of added investment. Ideally, the quality of every element in a power plant complex should be balanced with respect to all cooperating elements. The mathematical expression of the necessary condition for the existence of such balance is

$$\begin{aligned} \frac{\partial (\text{annual operating cost})}{\partial (\text{investment})_1} &= \frac{\partial (\text{annual operating cost})}{\partial (\text{investment})_2} \\ &= \frac{\partial (\text{annual operating cost})}{\partial (\text{investment})_n} = \frac{d (\text{annual operating cost})}{d (\text{plant investment})} \quad (\text{F-1}) \end{aligned}$$

where $\partial (\text{annual operating cost})$ is the increment in the yearly expenditure for heat corresponding to an increment, $\partial (\text{investment})$ in the cost of the n_{th} element.

Eq. F-1 says that for all elements in the balanced plant the marginal investment dollars are equally effective in reducing annual operating costs. If a rate of fixed charges is agreed upon, and the yearly output of electrical energy is held constant, the appropriate absolute value for the derivatives is determined and is

given by

$$\frac{\partial(\text{annual operating cost})}{\partial(\text{investment})} \times \frac{1}{\text{rate of fixed charges}} = -1.0 \quad (\text{F-2})$$

Eq. F-2 states that the marginal decrease in annual operating cost is capitalized at the amount of the corresponding increment in plant investment.

In reality, analytical relationships from which the derivatives of Eq. F-1 may be determined cannot often be found, and one must be content with graphical procedures assuming continuous functions. Nevertheless, the concept expressed in this equation has great generality and utility. In particular, it applies to the components of subsections of a power plant and, specifically, to the turbogenerator and its associated equipment.

This appendix examines the "conventional" or turbogenerator portion of a nuclear power plant from this point of view. The results have sufficient generality to permit rapid determination of the cost and performance of apparatus which is to be associated with a reactor producing steam at a given incremental cost and at a given thermodynamic state. Special attention is called to the dependence of this cost and performance on both the state of the steam produced and its incremental cost. The latter includes the cost of additional fuel to produce one extra Btu and the fixed charges on the increment of investment to provide the added reactor thermal capability. The disadvantages of expensive reactors and high fuel costs are reflected in more expensive turbogenerator plants. The more costly the Btu's, the more must be spent to husband them.

3. STATEMENT OF PROBLEM AND METHOD OF ATTACK

3.1 Statement of Problem

The incremental cost of heat from a reactor is defined as the cost required to produce an additional Btu in a suitably scaled-up reactor, including the incremental operating and fixed charges associated with the additional thermal capability. If we designate the value of the incremental cost of heat as $C_q \left[\frac{\$}{\text{Btu}} \right]$ and assume 7000 hr/yr of full load operation (80 per cent load factor), and set 15 per cent as the rate of fixed charges, eq. F-2 may be written as it applies to a PD 150-MW turbogenerator plant:

$$\frac{C_q \left[\frac{\$}{\text{Btu}} \right] \times 150,000 \left[\text{kwh} \right] \times 7000 \left[\frac{\text{hr}}{\text{yr}} \right]}{.15 \left[\frac{\$}{\text{yr}} / \$ \right]} \times \frac{\partial(\text{HR})}{\partial(I)} \left[\frac{\text{Btu}}{\text{kwh} \$} \right] = -1.0 \quad (\text{F-3})$$

where ∂I is the increment in turbogenerator plant cost necessary to produce the increment (algebraic) ∂HR in the plant heat rate. Rearranging,

$$\frac{\partial \text{HR}}{\partial I} = \frac{0.15}{C_q \times 150,000 \times 7000} \quad (\text{F-4})$$

It is understood that eq. F-1 applies to the components of the turbogenerator plant, i.e. $\frac{\partial HR}{\partial I}$ is the derivative of the HR vs I relationship for components of balanced plants of the class being considered for application to the reactor. Clearly such a relationship must exist if increased investment in the turbogenerator plant is reflected in improved Btu economy. In this appendix only those perturbations in turbogenerator plant design which affect the plant heat rate HR , are considered. Concomitantly, changes in turbogenerator operating cost associated with increments of investment are attributable solely to differences in heat consumption.

The specific objective of this appendix is to establish the values of HR and I for which eq. F-4 is satisfied as C_q is varied. This is done for three classes of cycles over a range of initial steam conditions. The results are plotted as HR vs t , and I vs t , curves where t , is the throttle steam temperature. C_q is the parameter for these curves (figs. F.1 to F.6).

3.2 Method of Attack

The classes of turbogenerator plants of most apparent interest for nuclear power plants have been analyzed over the probable range of useful throttle steam conditions.

3.2a Subdivision of the Problem. Just as the turbogenerator plant as a whole is separated from the reactor plant for internal balancing, so may its elements be balanced in pairs or threes. It is readily seen that the assumption of fixed electrical output eliminates a large group of important elements as variables, i.e. the basic building and its services, the electrical distribution system, etc.. Furthermore, for a given initial steam condition commercially available turbines are distinguished from each other almost exclusively by their exhaust annulus areas, $A_{ex} [ft^2]$. This fact is reflected explicitly in the price lists of the major turbine manufacturers. Specification of A_{ex} uniquely defines a turbogenerator for given throttle conditions.

Similarly established engineering practice with respect to the condenser-circulating water system-air removal equivalent complex makes it legitimate to define this combination as a single valued function of condensing surface, $A_c [ft^2]$. A_{ex} and A_c both influence the useful enthalpy drop associated with the exhaust flow, $G_{ex} \left[\frac{lb}{hr} \right]$. If their relative costs are known, a function $A_c = f(A_{ex})$ defining balanced turbine and condenser exists. A first step in our analysis is to find this relationship. Our problem, then, reduces to determining for each class of cycle and each throttle state the costs and performances of turbogenerator plants for each balanced A_c - A_{ex} combination of interest and solving for corresponding values of C_q in eq. F-4.

The question of optimization of regenerative feed heating lies in a category

intermediate between that of basic cycle selection and that of component quality or quantity and is so treated.

3.2b Selection of Cycles for Study. The three classes of cycle which may reasonably be expected to be associated with nuclear reactors are:

(1) Superheating in combination with regenerative feed heating but no reheating or moisture separation.

(2) Superheating in combination with regenerative feed heating and one stage of moisture separation.

(3) Saturation with regenerative feed heating and as many stages of moisture separation as are necessary to avoid excessive moisture.

Figures F.7, F.8, and F.9 are temperature - entropy diagrams of typical ideal cycles in each of these classes. In addition, they indicate the maximum and minimum throttle pressures and temperatures considered for each class. Extraction for feed heating is not indicated on these diagrams but is understood.

Reheat cycles were dropped from the studies when it became apparent that they would be applicable over only a very narrow range of initial steam conditions, and on the basis of assurances from Westinghouse and General Electric engineers that moisture separation is entirely practicable.

As the study neared completion it became apparent that cycles without regenerative feed heating are likewise of interest. They were investigated for the single inlet state of 500 F saturated.

4. BASIC ASSUMPTIONS AND COST DATA

The following basic assumptions and data are used in the economic optimization of the turbogenerator plant:

4.1 Assumptions with Respect to Performance.

4.1a Electrical Power Output. This study is confined to turbogenerator plants having 150 MW gross output and installed in pairs. Whereas PD has explored the economics of 300-MW nuclear power plants, it appears that circa 1965 relatively few applications for indivisible turbogenerator capacities of 300-MW size will exist. It is quite possible that investment economies may result from installation of 300-MW turbogenerators.

4.1b Condenser Cooling Water. Cooling water is assumed to be fresh and copiously available at 65 F. Whereas this temperature is a representative average value, there is a tendency in central-station practice to design for unfavorable water conditions. The condenser areas computed in this study are therefore somewhat undersize with respect to the turbine exhaust when compared to conventional practice.

There seems to be some question as to the economic justification of this conventional practice, In any case, the discrepancy does not seriously distort the relative plant costs and performances computed in this study.

4.1c Feed-Water Heaters. Initial cycle calculations are predicated on three- or four-stage feed heating depending on the cycle pressure level:

- (1) Four heaters used with throttle pressures exceeding 599 psia.
- (2) Three heaters used with throttle pressures less than 600 psia.
- (3) The low-pressure heater is a drain cooler heater. A closed contact heater is used as a deaerator. The third heater is a drain-cooler type. The fourth, when used, is flashed. Eight per cent pressure drop of the extraction steam is assumed in each feedwater heater. The effects of altering the feed heating arrangements are based on Salisbury's work. (1)

4.1d Turbines. For all cycles, turbines are assumed to have uniform infinitesimal stage efficiencies of 82 per cent with the state line end point (SLEP) on the 12 per cent moisture line at a condensing temperature of 80 F. At higher condensing temperatures the end points lie on the same state line at lower moistures. Leaving losses are assumed to be equal to the kinetic energy associated with the average velocity in the turbine exhaust annulus. This velocity is assumed to be purely axial. Since we are concerned only with design point performance, no penalties are taken for distortions of flow conditions in the low pressure stages as exhaust velocities are varied. An alternate analysis might be predicated on all SLEP's lying on the 12 per cent moisture line, but no significant differences in conclusions are to be expected from such a refinement.

Leakage, windage, and mechanical and electrical losses are neglected. Comparisons with Elston and Knowlton's data⁽²⁾ indicate that gross electrical output heat rates thus calculated are slightly conservative. On the other hand, they yield heat rates more favorable than those predicted by Beldecos et al.⁽³⁾

4.1e Separators. On Westinghouse recommendation, approximately 90 per cent separation effectiveness with 5 per cent pressure drop is assumed. Intermediate moistures are limited to 12 per cent before separation. There are indications from Westinghouse that, so far as the turbine is concerned, this may be excessively conservative, as is the final 12 per cent moisture assumed. General Electric confirms the limits used but apparently would apply them to states existing after internal moisture removal by means of grooved blades. On the basis of probable 1965 technology we are, then, exaggerating the moisture problem. This discrepancy does not seriously distort our over-all results.

4.1f Condensers and Circulating Water System. Since the turbine leaving loss is assumed to be one velocity head, the condensing pressure is equal to that at the state line end point. Condenser surface areas and tube arrangements to achieve

condensate temperatures from 80 to 95 F using the 65 F cooling water are calculated on the basis of Heat Exchange Institute conventions.⁽⁴⁾ Specification of condenser load and temperature thus defines completely the condenser and its auxiliaries — circulating water system and air-removal equipment.

4.2 Costs of Plant Components

4.2a Turbogenerators. Remarkably complete data on turbogenerator installed costs are incorporated in General Electric's June 1953 price lists. A very wide range of initial conditions is covered, and for each condition a number of exhaust areas and shaft arrangements are priced. The interpretation of these data with respect to application of internal moisture separation or to steam conditions less favorable than those contemplated in their preparation has been greatly facilitated by conversations with personnel of both the General Electric and Westinghouse steam divisions. The prices assigned to these major items of power plant equipment are, in consequence, known with considerable precision. In particular, the influence of an increment in exhaust annulus area, A_{ex} , is known very precisely. Although these increments have finite size corresponding to standard frames and bucket lengths, enough variations are standard to justify analysis on the basis of a continuous curve. In Fig. F.10 is plotted price variation vs annulus area for 150,000-kw turbogenerator units.

4.2b Foundations, Buildings, and Allied Costs. Jackson and Moreland, Engineers, have furnished data on foundation, building, and allied costs. The assumption is made that condenser and turbine room space requirements are uniquely related to turbine exhaust annulus area. Accompanying variations in construction cost are treated as part of the incremental investment associated with variations in exhaust area. These costs have been plotted on Figure F.10, both separately and summed with the incremental exhaust area costs. The sums as presented have been multiplied by 1.34 to cover omissions, contractors' overhead and profit, and engineering. A straight-line relation results with the slope, $\partial C_t / \partial A_{ex} = \$12400/\text{ft}^2$.

4.2c Condenser Equipment. Examination of several sources of data led to the conclusion that a flat figure of $\$16.25/\text{ft}^2$ of condenser surface covers costs of condenser, circulating water system, and condenser auxiliaries for the fairly limited range of sizes in which we are interested. Multiplication by 1.34 to cover overheads etc. brings the total value of condenser surface increments to $\partial C_c / \partial A_c = \$21.78/\text{ft}^2$.

4.2d Feedwater Heaters. Basic costs are based on recent Jackson and Moreland records for comparable applications. Since this equipment represents a comparatively small fraction of the total investment, refined designs have not been executed and prices are likewise rough. A figure of \$52/1000 lb/hr of throttle flow has been used for all high pressure heaters. Multiplied by 1.34 to cover overhead, etc. this becomes \$69.70/1000 lb/hr.

4.2e Basic Turbogenerator Plant Costs. Costs for turbogenerator plants are based on a Jackson and Moreland estimate for a modern coal-fired plant of size approximately comparable to those considered by Project Dynamo. This estimate was adjusted to obtain the cost of an economically balanced turbogenerator plant with 2000 psi 1050F initial conditions and 1000 F reheat. These figures were then modified to obtain the costs of seven representative balanced turbogenerator plants as follows:

- (1) Straight superheat cycles
 - (a) 1200 psia, 950 F; 7 heaters
 - (b) 900 psia, 950 F; 5 heaters
 - (c) 600 psia, 900 F; 5 heaters
- (2) Superheat cycles with one moisture separation.
 - (a) 2000 psia, 900 F; 6 heaters
- (3) Saturated cycles with moisture separation.
 - (a) 900 psia; 5 heaters
 - (b) 600 psia; 4 heaters
 - (c) 200 psia; 3 heaters

These estimates are presented in Table F.1.

To these plant costs are applied the differences for varying the numbers of feedwater heaters, and changing the size of the exhaust annulus and condensers as determined on the basis of the cost data discussed in the preceding sections. The following section details the method of determining these differences.

5. DETAILED ANALYSIS

5.1 Balancing Turbine Exhaust Annulus and Condenser Areas

As indicated in Section 3.2a, the turbine exhaust annulus area, A_{ex} , and the condenser surface, A_c , can be balanced with respect to each other in an economic sense without reference to the remainder of the cycle. This is accomplished by determining those combinations of areas for which a transfer of investment from exhaust annulus to condenser would produce no net change in useful enthalpy drop; i.e. the final static enthalpy, h_3 , would be unchanged. Associated with such a transfer the enthalpy, h_2 , at the SLEP and the exhaust loss, EL , are changed in a mutually compensating manner. The accompanying h - s diagram represents the turbine exhaust processes before and after such a transfer of investment. In this diagram h_2 is the enthalpy of the SLEP and h_3 is the stagnation enthalpy at the condenser. $h_1 - h_3 = \text{turbine output } \left[\frac{\dot{Q} \dot{t} u}{16} \right]$. The condition for balanced annulus and condenser areas is $\Delta h_3 = 0$ where $\Delta h_3 = \Delta h_2 + \Delta EL$,

in which

$$\Delta h_2 = \left(\frac{\partial h_2}{\partial A_c} \right)_{G_{ex}, A_{ex}} \Delta A_c$$

Table F.1 — COST OF TURBOGENERATOR PLANTS OF 150 MW GROSS CAPABILITY IN THOUSANDS OF DOLLARS

(Based on two 150 MW units per station)								
Initial steam conditions	2000 lbs 1050 F 1000 F	2000 lbs 900 F	1200 lbs 950 F	900 lbs 950 F	900 lbs Saturated	600 lbs 900 F	600 lbs Saturated	200 lbs Saturated
Class of cycle	Reheat	Separator	Straight expansion	Straight expansion	Separator	Straight expansion	Separator	Separator
1. Turbine plant building	1,500	1,500	1,500	1,500	1,600	1,700	1,700	1,900
2. Feedwater system	615	615	560	525	525	550	485	450
3. Water supply and treatment	150	150	150	150	150	140	140	125
4. Steam generator instrumentation and controls	450	425	425	415	415	400	400	400
5. Piping	1,800	1,550	1,450	1,400	1,300	1,500	1,400	1,700
6. Turbogenerators	3,280	3,500	3,230	3,225	3,650	3,520	3,860	4,210
7. Circulating water system	348	535	535	535	740	850	875	1,086
8. Condensers and auxiliaries	317	488	488	488	675	775	796	990
9. Miscellaneous turbogenerator equipment	360	370	370	370	415	425	425	435
10. Accessory electrical equipment	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000
11. Miscellaneous power plant equipment	120	120	120	120	120	120	120	120
12. Transmission plant	400	400	400	400	400	400	400	400
13. Total material and labor	10,340	10,653	10,228	10,128	10,990	11,380	11,601	12,816
14. Omissions at 10 per cent	1,034	1,065	1,023	1,013	1,099	1,138	1,160	1,282
15. Subtotal	11,374	11,718	11,251	11,141	12,089	12,518	12,761	14,098
16. Contractor's overhead and profit at 11 per cent	1,251	1,289	1,238	1,226	1,330	1,377	1,404	1,551
17. Subtotal	12,625	13,007	12,489	12,367	13,419	13,895	14,165	15,649
18. Engineering at 10 per cent	1,263	1,301	1,249	1,237	1,342	1,390	1,417	1,565
19. Total estimated cost	13,888	14,308	13,738	13,604	14,761	15,285	15,582	17,214
20. Cost per gross kw (Dollars)	92.60	95.50	91.60	90.70	98.30	101.70	103.80	114.70
21. Number of feedwater heaters	7	7	6	5	5	5	4	3
22. Size blades	2 - 26"	3 - 26"	3 - 26"	3 - 26"	2 - 35"	2 - 38"	2 - 38"	2 - 43"
23. Condenser area, ft ²	41,000	63,000	63,000	63,000	87,000	100,000	103,000	128,000
24. Turbine type	TCDF 3600	TCTF 3600	TCTF 3600	TCTF 3600	TCDF 1800	TCDF 1800	TCDF 1800	TCDF 1800
25. Exhaust annulus area, ft ²	82.5	123.3	123.3	123.3	168	190.8	190.8	246
26. Cost of exhaust annulus (Item 25 × \$12,400/ft ²)	1530	1530	1530	1530	2080	2365	2365	3050
27. Cost of condenser and circulating water system (Item 23 × \$21.78/ft ²)	895	1372	1372	1372	1895	2178	2245	2790
28. Item 26 + Item 27	2423	2902	2902	2902	3975	4543	4610	5840
29. Cost of plant less exhaust annulus and condenser, etc. (Item 19 minus Item 28)	11,465	11,406	10,836	10,702	10,786	10,742	10,872	11,374

∞

and

$$\Delta EL = \left(\frac{\partial EL}{\partial A_c} \right)_{G_{ex}, A_{ex}} \Delta A_c + \left(\frac{\partial EL}{\partial A_{ex}} \right)_{G_{ex}, A_c} \Delta A_{ex}$$

where

$$\Delta A_c \frac{\partial C_c}{\partial A_c} = -\Delta A_{ex} \frac{\partial C_{ex}}{\partial A_{ex}} \quad (F-5)$$

is satisfied.

Combining the last four expressions gives the complete condition for balance of annulus and condensing areas:

$$\left(\frac{\partial EL}{\partial A_c} \right)_{G_{ex}, A_{ex}} \Delta A_c + \left(\frac{\partial EL}{\partial A_{ex}} \right)_{G_{ex}, A_c} \Delta A_{ex} + \left(\frac{\partial h_2}{\partial A_c} \right)_{G_{ex}, A_{ex}} \Delta A_c = 0 \quad (F-6)$$

From the assumptions of Section 4.1d,

$$EL = \frac{V_2^2}{2gJ} = \frac{\left(\frac{G_{ex} V_2}{A_{ex}} \right)^2}{2gJ} \quad (F-7)$$

5.1a Influence of Condenser Area. The effect of increasing the condenser area as the mass flow is held constant is to reduce the condensing temperature, t_2 . Figure F.11 plots this relationship for several exhaust losses. Since the EL 's are small compared to the heat rejected in the condenser, the $\frac{A_c}{G_{ex}}$ vs t_2 relation is very insensitive to EL . For this reason the derivative $\frac{\partial A_c}{G_{ex} \partial t_2}$ is taken as a single valued function of t_2 which is also plotted on Figure F.11.

Along the assumed condition line V_2 is a unique function of t_2 , as is the derivative $\frac{\partial V_2}{\partial t_2}$. Both of these functions are plotted on Figure F.12. On Figure F.13 are plotted the comparable enthalpy-temperature relations, h_2 vs t_2 and $\frac{\partial h_2}{\partial t_2}$ vs t_2 .

The data of Figures F.11 and F.13 can be combined to give

$$\frac{\frac{\partial h_2}{\partial t_2}}{\frac{\partial \frac{A_c}{G_{ex}}}{\partial t_2}} = \frac{\partial h_2}{\partial \frac{A_c}{G_{ex}}} = f(t_2) \quad (F-8)$$

Differentiating eq. F-7 and multiplying by G_{ex} gives

$$\left(\frac{\partial EL}{\partial \frac{A_c}{G_{ex}}} \right)_{G_{ex}, A_{ex}} = \frac{\left(\frac{G_{ex}}{A_{ex}} \right)^2 v_2}{3600^2 g J} \frac{\partial v_2}{\partial \frac{A_c}{G_{ex}}} \quad (F-9)$$

Taken together with the data of Figures F.11 and F.12, eq. F-9, gives $\left(\frac{\partial EL}{\partial \frac{A_c}{G_{ex}}} \right)_{G_{ex}, A_{ex}}$ as a function of t_2 and $\frac{G_{ex}}{A_{ex}}$.

5.1b Influence of Annulus Area The effect of increasing the turbine exhaust annulus is somewhat simpler. As can be seen on Figure F.11, a small change in EL with $\frac{A_c}{G_{ex}}$ held constant scarcely affects t_2 , and consequently the SLEP remains essentially fixed when A_{ex} is varied. This accounts for the single appearance of ΔA_{ex} in eq. F-6. This term in eq. F-6 is evaluated by differentiating eq. F-8 holding v_2 and G_{ex} constant:

$$\left(\frac{\partial EL}{\partial \frac{A_{ex}}{G_{ex}}} \right)_{G_{ex}, A_c} = - \frac{\dot{v}_2}{g J \left(\frac{A_{ex}}{G_{ex}} \right)} \quad (F-10)$$

Taken with the data of Figure F.12 this gives $\left(\frac{\partial EL}{\partial \frac{A_{ex}}{G_{ex}}} \right)_{G_{ex}, A_{ex}}$ as a function of t_2 and $\left(\frac{A_{ex}}{G_{ex}} \right)$.

5.1c Consequences of an Exhaust Annulus-Condenser Balance. Since the incremental costs of exhaust annulus and condenser areas are known (Secs. 4.2a and 4.2c), ΔA_c can be written from eq. F-5 in terms of ΔA_{ex} :

$$\Delta A_c = -\Delta A_{ex} \times \frac{\partial C_{ex}}{\partial A_{ex}} \times \frac{\partial A_c}{\partial C_c} \quad (F-11)$$

$$\Delta A_c = -\Delta A_{ex} \frac{12,400}{21.78} = -568 \Delta A_{ex} \quad (F-11a)$$

Substituting in the condition for balance, eq. F-6, gives

$$-\left(\frac{\partial EL}{\partial \frac{A_c}{G_{ex}}} \right)_{G_{ex}, A_{ex}} 568 \Delta A_{ex} + \left(\frac{\partial EL}{\partial \frac{A_{ex}}{G_{ex}}} \right)_{G_{ex}, A_c} \Delta A_{ex} - \left(\frac{\partial h_2}{\partial \frac{A_c}{G_{ex}}} \right)_{G_{ex}, A_{ex}} 568 \Delta A_{ex} = 0$$

Dividing by $\frac{\Delta A_{ex}}{G_{ex}}$ yields

$$\left(\frac{\partial EL}{\partial \frac{A_{ex}}{G_{ex}}} \right)_{G_{ex}, A_c} = \left(\frac{\partial EL}{\partial \frac{A_c}{G_{ex}}} \right)_{G_{ex}, A_{ex}} 568 + \left(\frac{\partial h_2}{\partial \frac{A_c}{G_{ex}}} \right)_{G_{ex}, A_{ex}} 568 \quad (F-12)$$

The derivative on the left-hand side of this equation is known as a function of t_2 and $\frac{A_{ex}}{G_{ex}}$ (see preceding section), and the derivatives on the right-hand side are

functions of t_2 and $\frac{A_{ex}}{G_{ex}}$ and of t_2 alone (see sec. 5.1a). The simultaneous solution^(a) of all these functions yields $\frac{G_{ex}}{A_{ex}}$ as a single valued function of the condensing temperature, t_2 . This important relationship is plotted on Figure F.16. This relation and the v_2 vs t_2 relation of Figure F.12 together with eq. F-7 determine EL as a function of t_2 as plotted on Figure F.17.

5.2 Regenerative Feed Heater Balance.

By means of entirely conventional cycle calculations based on the premises of Section 4.1 and $EL-t_2$ combinations determined from Figure F.17 corresponding to balanced turbine and condenser, heat rates, $(HR)_0$ and throttle and condenser mass flows, $(G_{th})_0$ and $(G_{ex})_0$, to develop 150 MW are determined for arbitrary numbers of feed heaters. The first twelve columns of tables F.2 and F.3 indicate the development of these results and the sources of data. Columns 13 and 14 indicate the annulus and condenser areas required to secure the required mass flow at the exhaust conditions postulated.

From Figure F.14 or F.15 the effect of a dollar investment in either turbine or condenser on the useful enthalpy drop in the exhaust flow is found. From this, $(G_{th})_1$, $(G_{ex})_1$ and the heat added in the boiler, Q_{blr} , the derivative of heat rate with dollar of turbine-condenser investment can be found. Using Salisbury's data and the cost data of Section 4.2d,^(b) the number of heaters justified in view of the turbine investment is determined. This number, which is substantially constant for each initial state over the range of turbine investments considered, is entered as $(N_H)_2$ in column 15. In column 16 the ratio of heat rate for the cycle with $(N_H)_2$ heaters to that with originally assumed number is entered. Columns 18 through 22 develop the final exhaust flow, $(G_{ex})_2$, for a fully balanced 150-MW turbogenerator plant.

5.3 Costs of Fully Balanced Turbogenerator Plants.

In columns 23 and 24 are recorded the cost of exhaust annulus area (total turbogenerator cost with overheads, etc. less a hypothetical cost for a unit with zero annulus, assuming that the linear relationship of Figure F.10 is valid to this zero area) and the cost of condenser with overheads, etc. Column 25 sums these costs and consequently represents the difference between the cost of a balanced plant for the assumed condensing temperature, t_2 and a hypothetical turbogenerator plant with no condenser or circulating water system annulus and a zero area turbine.

(a) The actual process of solution was largely graphical. Figures F.14 and F.15 are examples of working curves used in effecting the solutions leading to Figures F.16 and F.17.

(b) Figures F.18 and F.19 are samples of the working curves synthesized from Salisbury's work and our cycle calculations and cost data.

The latter figure is recorded for each of the seven reference plants in Table F.1. Costs for other initial states are interpolated values. Combined with the costs of column 25 they give final turbogenerator plant investment. This cost is plotted against the corresponding heat rate for each cycle analyzed. Figure F.20 is a sample of these plots.^(a)

6. FINAL RESULTS

The investment vs heat rate curves like those in Figure F.20, are differentiated to give values of $\frac{\partial HR}{\partial I}$. Recalling eq. F-4, $\frac{\partial HR}{\partial I} = \frac{0.15}{C_a \times 150,000 \times 7000}$, it is seen that the appropriate incremental cost of heat corresponding to each turbogenerator plant investment and heat rate is immediately calculable. The results of such calculations as plotted on Figures F.1 - F.6 constitute the conclusions of this study.

7. DISCUSSION

There are a number of limitations on this study. Some of these limitations stem from necessary approximations in the performance and cost data; others like the restriction to 65 F circulating water and 15 per cent fixed charges, are made deliberately in order to narrow the problem to manageable proportions. Other cycles may also be of interest, in particular, more extensive investigation of non-regenerative plants or plants with restricted amounts of feed-heating is indicated.

It is unfortunate that PD's interest in cheap plants was not anticipated so that the extensive extrapolations employed might have been avoided. It is probably a fact that condensers which would operate at the high heat transfer loadings implied in the large temperature differences between the circulating water and the condensate would have to be developed and might prove more expensive than anticipated. On the other hand, the assumption of a single condition line going through the 12 per cent moisture, 80 F point has penalized those plants with higher exhaust temperatures. In almost every instance it is advantageous to accept the highest permissible moisture in the exhaust. Similarly turbines with small annulus areas - where this smallness is associated with short low-pressure blading - could probably have better than 82 per cent stage efficiency.

It was pointed out in Section 4.1b that conventional condenser installations are designed for unfavorable water conditions. This is attributable in large part to the desire to maintain performance in periods of high water temperatures. Condensers intended for operation with large temperature differences will be much less

(a) The dotted portion of this curve indicated extrapolation into regions of low investment which were not originally thought to be of interest.

Table F.2 -- CALCULATION OF COSTS OF CONDENSER SYSTEMS AND EXHAUST ANNULI FOR BALANCED TURBOGENERATOR PLANTS

①	②	③	④	⑤	⑥	⑦	⑧	⑨	⑩	⑪	⑫	⑬	⑭	⑮	⑯	⑰	⑱	⑲	⑳	㉑	㉒	㉓	㉔	㉕
Cycle Class t, p	t_2 [°F]	EL [$\frac{Btu}{lb}$]	$\frac{G_{EX}}{A_{EX}}$ [$\frac{lb}{hr ft^2}$]	$\frac{A_c}{G_{EX}}$ [$\frac{hr ft^2}{lb}$]	$(G_{EX})_o$ [$\frac{lb}{hr}$]	$(G_{TH})_o$ [$\frac{lb}{hr}$]	$(HR)_o$ [$\frac{Btu}{kwh}$]	$\frac{(Q_{Blr})_o}{(G_{TH})_o}$ [$\frac{Btu}{lb}$]	(G_{Gross}) [MW]	$(G_{EX})_i$ [$\frac{lb}{hr}$]	$(G_{TH})_i$ [$\frac{lb}{hr}$]	$(A_{EX})_i$ [ft^2]	$(A_c)_i$ [ft^2]	$(NH)_2$	$(HR)_2$ [$\frac{Btu}{kwh}$]	$(HR)_2$ [$\frac{Btu}{kwh}$]	$\frac{(Q_{Blr})_2}{(G_{TH})_2}$ [$\frac{Btu}{lb}$]	$(G_{TH})_2$ [$\frac{lb}{hr}$]	$\frac{(G_{TH}-G_{EX})_2}{(G_{TH}-G_{EX})_1}$	$(G_{TH}-G_{EX})_2$ [$\frac{lb}{hr}$]	$(G_{EX})_2$ [$\frac{lb}{hr}$]	C_{EX} [\$]	C_c [\$]	ΣC [\$]
Source	Assumed	Fig F.17	Fig F.16	Fig F.11	Assumed	Cycle analysis	Cycle analysis	Cycle analysis	$\frac{⑨ \times ⑦}{⑧}$	$\frac{150 \times ⑥}{⑩}$	$\frac{150 \times ⑦}{⑫}$	$\frac{⑪}{⑬}$	$\frac{⑭ \times ⑤}{⑮}$		Salisbury ⁽¹⁾ $\frac{⑧ \times ⑯}{⑰}$		$\frac{⑱ \times ⑰ \times ⑨}{⑲ + ⑱}$	Salisbury ⁽¹⁾ $\frac{⑲ \times (⑫ - ⑪)}{⑲ - ⑫}$	$\frac{⑳ \times (⑫ - ⑪)}{⑲ - ⑫}$	$\frac{(G_{EX})^{(a)} \times ㉒ \times ㉓}{(A_{EX})^{(b)} \times ㉑}$	$\frac{(C_c)^{(b)} \times ㉔ \times ㉕}{(A_c)^{(b)} \times ㉑}$	$\frac{㉓ + ㉔}{㉑}$		
Regenerative Saturated Cycle with Moisture Separation	80	7.62	3.985×10^3	0.1390	1.499×10^6	2.009×10^6	12,141	959.0	158.7	1.417×10^6	1.899×10^6	355.6	197.0×10^3		.9906	12,027	-12.3	1.905×10^6	1.142	$.552 \times 10^6$	1.353×10^6	4.217×10^6	4.097×10^6	8.313×10^6
	84	9.00	4.890	0.1081	1.499	2.006	12,390	957.5	155.0	1.450	1.941	296.5	156.7		.9906	12,274	-12.1	1.946	1.142	.562	1.384	3.516	3.258	6.774
	87	9.87	5.580	0.0930	1.499	2.005	12,575	956.5	152.5	1.474	1.972	264.1	137.1	3	.9906	12,457	-12.0	1.976	1.142	.568	1.408	3.132	2.851	5.983
382°F, 200 psia	91	10.65	6.503	0.0793	1.499	2.002	12,809	955.0	149.3	1.506	2.012	231.5	119.4		.9906	12,689	-11.9	2.016	1.142	.578	1.438	2.746	2.483	5.229
2 feed heaters assumed	95	11.11	7.436	0.0687	1.499	2.000	13,033	953.5	146.3	1.537	2.051	206.7	105.6		.9906	12,910	-11.7	2.057	1.142	.589	1.468	2.453	2.196	4.651
Regenerative Saturated Cycle with Moisture Separation	84	9.00	4.890×10^3	0.1081	1.380×10^6	1.956×10^6	11,768	923.9	153.6	1.348×10^6	1.910×10^6	275.7	145.7×10^3		1.000	11,768	0.0	1.910×10^6	1.000	$.562 \times 10^6$	1.348×10^6	3.424×10^6	3.173×10^6	6.597×10^6
	87	9.87	5.580	0.0930	1.380	1.952	11,926	923.1	151.1	1.370	1.938	245.5	127.4		1.000	11,926	0.0	1.938	1.000	.568	1.370	3.049	2.775	5.824
	89	10.32	6.042	0.0847	1.380	1.949	12,030	922.5	149.5	1.385	1.956	229.2	117.3	3	1.000	12,030	0.0	1.956	1.000	.571	1.385	2.847	2.555	5.401
397°F, 240 psia	91	10.65	6.503	0.0793	1.380	1.947	12,129	922.1	148.0	1.399	1.973	215.1	110.9		1.000	12,129	0.0	1.973	1.000	.574	1.399	2.672	2.415	5.087
3 feed heaters assumed	93	10.89	6.970	0.0736	1.380	1.944	12,225	921.5	146.5	1.413	1.991	202.7	104.0		1.000	12,225	0.0	1.991	1.000	.578	1.413	2.518	2.265	4.783
	95	11.11	7.436	0.0687	1.380	1.941	12,320	921.0	145.1	1.427	2.007	191.9	98.0		1.000	12,320	0.0	2.007	1.000	.580	1.427	2.383	2.134	4.518
Regenerative Saturated Cycle with Moisture Separation	80	7.62	3.985×10^3	0.1390	1.083×10^6	1.778×10^6	9,820	858	155.3	1.046×10^6	1.718×10^6	262.5	145.4×10^3		.9926	9,747	-17.0	1.740×10^6	1.081	$.736 \times 10^6$	1.004×10^6	3.129×10^6	3.040×10^6	6.169×10^6
	84	9.00	4.890	0.1081	1.083	1.773	9,961	857	152.5	1.065	1.744	217.8	115.1		.9926	9,887	-16.7	1.765	1.081	.743	1.022	2.597	2.407	5.004
	87	9.87	5.580	0.0930	1.083	1.770	10,065	856	150.5	1.079	1.763	193.4	100.3	4	.9926	9,991	-16.5	1.784	1.081	.748	1.036	2.306	2.099	4.405
486°F, 600 psia	91	10.65	6.503	0.0793	1.083	1.765	10,197	855	148.0	1.098	1.789	168.8	87.1		.9926	10,122	-16.3	1.809	1.081	.754	1.055	2.013	1.822	3.835
3 feed heaters assumed	95	11.11	7.436	0.0687	1.083	1.760	10,321	854	145.6	1.116	1.813	150.1	76.7		.9926	10,245	-16.0	1.834	1.081	.762	1.072	1.791	1.605	3.396
Regenerative Saturated Cycle with Moisture Separation	80	7.62	3.985×10^3	0.1390	9.830×10^5	1.838×10^6	9,246	778	154.6	9.538×10^5	1.783×10^6	239.3	132.6×10^3		.9940	9,191	-20.0	1.819×10^6	1.040	8.80×10^5	9.390×10^5	2.972×10^6	2.888×10^6	5.860×10^6
	84	9.00	4.890	0.1081	9.830	1.833	9,365	777	152.1	9.694	1.808	198.2	104.8		.9940	9,309	-19.7	1.844	1.040	8.90	9.543	2.462	2.283	4.745
	87	9.87	5.580	0.0930	9.830	1.830	9,453	777	150.3	9.810	1.826	175.8	91.2	5	.9940	9,396	-19.5	1.862	1.040	8.96	9.656	2.183	1.986	4.169
540°F, 962 psia	91	10.65	6.503	0.0793	9.830	1.825	9,563	776	148.1	9.956	1.849	153.1	79.0		.9940	9,506	-19.3	1.884	1.040	9.04	9.800	1.902	1.721	3.622
4 feed heaters assumed	95	11.11	7.436	0.0687	9.830	1.821	9,669	775	146.0	1.010×10^6	1.871	135.8	69.4		.9940	9,611	-19.0	1.907	1.040	9.13	9.940	1.687	1.512	3.198
Regenerative Superheat Cycle without Moisture Separation	80	7.62	3.985×10^3	0.1390	1.393×10^6	1.716×10^6	11,695	1065.0	156.3	1.337×10^6	1.647×10^6	341.3	189.0×10^3		.9950	11,637	-12.0	1.657×10^6	1.081	$.337 \times 10^6$	1.320×10^6	4.186×10^6	4.064×10^6	8.250×10^6
	84	9.00	4.890	0.1081	1.393	1.711	11,917	1063.7	152.7	1.368	1.681	279.8	147.9		.9950	11,857	-11.7	1.691	1.081	.340	1.351	3.432	3.181	6.613
	87	9.87	5.580	0.0930	1.393	1.708	12,083	1062.7	150.2	1.391	1.706	249.3	129.4	4	.9950	12,023	-11.5	1.716	1.081	.342	1.374	3.058	2.783	5.841
600°F, 184 psia	91	10.65	6.503	0.0793	1.393	1.703	12,295	1061.3	147.0	1.421	1.738	218.5	112.7		.9950	12,234	-11.3	1.748	1.081	.345	1.403	2.680	2.424	5.104
3 feed heaters assumed	95	11.11	7.436	0.0687	1.393	1.699	12,497	1060.0	144.1	1.450	1.769	195.0	99.62		.9950	12,435	-11.0	1.779	1.081	.347	1.432	2.392	2.143	4.535
Regenerative Superheat Cycle without Moisture Separation	80	7.62	3.985×10^3	0.1390	1.067×10^6	1.397×10^6	9,767	1089	155.8	1.027×10^6	1.344×10^6	257.7	142.8×10^3		.9880	9,650	-14	1.357×10^6	1.130	$.361 \times 10^6$	0.9956×10^6	3.103×10^6	3.014×10^6	6.117×10^6
	84	9.00	4.890	0.1081	1.067	1.393	9,909	1088	152.9	1.047	1.367	214.1	113.2		.9880	9,790	-14	1.380	1.130	.365	1.015	2.580	2.391	4.971
	87	9.87	5.580	0.0930	1.067	1.390	10,015	1087	150.9	1.061	1.382	190.1	98.7	5										

(a) App. F, Sec. 4.2

(b) App. F, Sec. 4.2

Note: Subscripts ()_o and ()_i apply to quantities assumed or determined for cycle with arbitrary number of heaters; subscripts ()₂ apply to quantities for cycle with optimum number of heaters.

Table F.3 -- CALCULATION OF COSTS OF CONDENSER SYSTEMS AND EXHAUST ANNULI FOR BALANCED TURBOGENERATOR PLANTS

①	②	③	④	⑤	⑥	⑦	⑧	⑨	⑩	⑪	⑫	⑬	⑭	⑮	⑯	⑰	⑱	⑲	⑳	㉑	㉒	㉓	㉔	㉕
Cycle Class t_1, p_1	t_2 [°F]	EL [Btu/lb]	$\frac{G_{EX}}{A_{EX}}$ [$\frac{lb}{hr ft^2}$]	$\frac{A_c}{G_{EX}}$ [$\frac{hr ft^2}{lb}$]	(G _{EX}) _o [$\frac{lb}{hr}$]	(G _{TH}) _o [$\frac{lb}{hr}$]	(HR) _o [$\frac{Btu}{kwh}$]	$\frac{(Q_{BIR})_o}{(G_{TH})_o}$ [$\frac{Btu}{lb}$]	Gross Output [MW]	(G _{EX}) _i [$\frac{lb}{hr}$]	(G _{TH}) _i [$\frac{lb}{hr}$]	(A _{EX}) _i [ft ²]	(A _C) _i [ft ²]	(N _H) ₂ (HR) _o	(HR) ₂ (HR) _o	(HR) ₂ [$\frac{Btu}{kwh}$]	$\frac{(Q_{BIR})_2}{(G_{TH})_2}$ [$\frac{Btu}{lb}$]	(G _{TH}) ₂ [$\frac{lb}{hr}$]	$\frac{(G_{TH}-G_{EX})_2}{(G_{TH}-G_{EX})_1}$	(G _{TH} -G _{EX}) ₂ [$\frac{lb}{hr}$]	(G _{EX}) ₂ [$\frac{lb}{hr}$]	C _{EX} [\$]	C _C [\$]	ε C [\$]
Source	Assumed	Fig F.17	Fig F.16	Fig F.11	Assumed	cycle analysis	cycle analysis	cycle analysis	$\frac{9 \times 7}{8}$	$\frac{150 \times 6}{10}$	$\frac{150 \times 7}{10}$	$\frac{11}{4}$	⑪ × ⑤	Salisbury ^(a)	⑧ × ⑯		$\frac{12 \times ⑯ \times 9}{9 + ⑯}$	Salisbury ^(a)	⑳ × (⑫ - ⑪)	⑲ - ㉑	$\frac{(C_{EX})^{(a)} \times ㉒ \times ㉓}{A_{EX} \times ⑪}$	$\frac{(C_C)^{(b)} \times ㉔ \times ㉕}{A_C \times ⑪}$	㉓ + ㉔	
Regenerative Superheat Cycle with Moisture Separation	80	7.62	3.985x10 ³	.1390	1.190x10 ⁶	1.770x10 ⁶	10,488	925	156.1	1.143x10 ⁶	1.701x10 ⁶	286.8	158.9x10 ³	.9930	10,416	-15	1.717x10 ⁶	1.081	0.609x10 ⁶	1.108x10 ⁶	3.453x10 ⁶	3.354x10 ⁶	6.807x10 ⁶	
	84	9.00	4.890	.1081	1.190	1.766	10,659	924	153.1	1.166	1.730	238.4	126.0	.9930	10,584	-15	1.746	1.081	0.615	1.131	2.872	2.662	5.534	
	87	9.87	5.580	.0930	1.190	1.762	10,780	923	150.9	1.183	1.752	212.0	110.7	.9930	10,705	-15	1.768	1.081	0.620	1.148	2.554	2.339	4.893	
500°F, 430 psia	91	10.65	6.503	.0793	1.190	1.758	10,945	922	148.1	1.205	1.780	185.3	95.6	.9930	10,868	-15	1.796	1.081	0.627	1.169	2.233	2.020	4.253	
3 feed heaters assumed	95	11.11	7.436	.0687	1.190	1.753	11,080	921	145.7	1.225	1.805	164.7	84.2	.9930	11,002	-15	1.821	1.081	0.632	1.189	1.986	1.780	3.766	
Regenerative Superheat Cycle with Moisture Separation	80	7.62	3.985x10 ³	.1390	0.9109x10 ⁶	1.532x10 ⁶	8,888	890	153.4	0.8907x10 ⁶	1.498x10 ⁶	223.5	123.8x10 ³	.9895	8,795	-15	1.508x10 ⁶	1.075	0.657x10 ⁶	0.8506x10 ⁶	2.650x10 ⁶	2.574x10 ⁶	5.224x10 ⁶	
	84	9.00	4.890	.1081	0.9109	1.528	8,999	889.2	151.0	0.9049	1.517	185.1	97.8	.9895	8,905	-15	1.528	1.075	0.664	0.8645	2.196	2.036	4.232	
	87	9.87	5.580	.0930	0.9109	1.524	9,075	888.5	149.2	0.9158	1.532	164.1	85.2	.9895	8,980	-15	1.542	1.075	0.667	0.8752	1.947	1.773	3.720	
700°F, 1100 psia	91	10.65	6.503	.0793	0.9109	1.520	9,175	887.8	147.1	0.9289	1.549	142.8	73.66	.9895	9,079	-15	1.560	1.075	0.672	0.8881	1.695	1.534	3.229	
4 feed heaters assumed	95	11.11	7.436	.0687	0.9109	1.515	9,270	887	145.0	0.9423	1.567	126.7	64.7	.9895	9,173	-15	1.577	1.075	0.676	0.9013	1.505	1.349	2.854	
Regenerative Superheat Cycle with Moisture Separation	80	7.62	3.985x10 ³	.1390	0.8260x10 ⁶	1.471x10 ⁶	8,323	873	154.3	0.8030x10 ⁶	1.430x10 ⁶	201.5	111.60x10 ³	.9880	8,223	-17	1.443x10 ⁶	1.075	0.681x10 ⁶	0.7620x10 ⁶	2.375x10 ⁶	2.307x10 ⁶	4.682x10 ⁶	
	84	9.00	4.890	.1081	0.8260	1.467	8,414	872	152.0	0.8151	1.447	166.7	88.11	.9880	8,313	-17	1.459	1.075	0.685	0.7739	1.966	1.821	3.787	
	87	9.87	5.580	.0930	0.8260	1.463	8,480	872	150.5	0.8233	1.459	147.5	76.57	.9880	8,378	-17	1.471	1.075	0.689	0.7820	1.740	1.584	3.324	
800°F, 1640 psia	91	10.65	6.503	.0793	0.8260	1.459	8,564	871	148.4	0.8349	1.475	128.4	66.21	.9880	8,461	-17	1.487	1.075	0.694	0.7932	1.515	1.370	2.885	
4 feed heaters assumed	95	11.11	7.436	.0687	0.8260	1.455	8,644	870	146.4	0.8463	1.491	113.8	58.10	.9880	8,540	-17	1.503	1.075	0.699	0.8044	1.344	1.202	2.546	
Regenerative Superheat Cycle with Moisture Separation	80	7.62	3.985x10 ³	.1390	0.7440x10 ⁶	1.421x10 ⁶	7,812	845	153.7	0.7261x10 ⁶	1.386x10 ⁶	182.2	100.9x10 ³	.9830	7,679	-23	1.402x10 ⁶	1.08	0.721x10 ⁶	0.6806x10 ⁶	2.121x10 ⁶	2.060x10 ⁶	4.181x10 ⁶	
	84	9.00	4.890	.1081	0.7440	1.418	7,886	844	151.8	0.7352	1.401	150.3	79.5	.9830	7,752	-23	1.417	1.08	0.727	0.6895	1.750	1.623	3.373	
	87	9.87	5.580	.0930	0.7440	1.416	7,940	843	150.3	0.7425	1.413	133.1	69.1	.9830	7,805	-23	1.428	1.08	0.732	0.6964	1.550	1.411	2.961	
905°F, 2400 psia	91	10.65	6.503	.0793	0.7440	1.413	8,009	841	148.4	0.7520	1.428	115.6	59.6	.9830	7,873	-23	1.444	1.08	0.738	0.7055	1.348	1.218	2.566	
4 feed heaters assumed	95	11.11	7.436	.0687	0.7440	1.410	8,074	840	146.7	0.7607	1.441	102.3	52.3	.9830	7,937	-23	1.458	1.08	0.744	0.7140	1.192	1.069	2.261	

Note: Subscripts ()_o and ()_i apply to quantities assumed or determined for cycle with arbitrary number of heaters; subscripts ()₂ apply to quantities for cycle with optimum number of heaters.

(a) App. F, Sec. 4.2

(b) App. F, Sec. 4.2

sensitive to variations in circulating water temperature, and much more modest allowances will be in order.

8. CONVENTIONAL POWER PLANT COST

Table F.4 presents a cost estimate for a modern 300 MW reheat plant with coal-fired boilers. Based on the estimated investment costs, an in-the-bunker coal cost of \$0.30 per million Btu and a boiler efficiency of 87 per cent, this plant would produce power for 6.78 mills/kwh.

The turbogenerator cost for this plant is based on the estimate included in Table F.1.

REFERENCES FOR APPENDIX F

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2. Elston, C. W. and P. H. Knowlton, Comparative Efficiencies of Central Station Reheat and Non-Reheat Steam Turbine-Generator Units. ASME Trans. Nov. 1952, vol 74, No. 8, p 1389.
3. Beldecos, N. A., A. R. Jones, and A. K. Smith. Comparative Performance of Modern Steam Turbine Generator Units for Atomic Power Plants. Westinghouse Electric Corp.; Atomic Equipment Dept., Report No. WIAP-2, March 1953.
4. Heat Exchange Institute Standards for Steam Surface Condensers. New York, 1952.

Table F.4 — ESTIMATED COST OF MODERN 300 MW COAL FIRED POWER PLANT

PLANT SPECIFICATIONS

Steam conditions	2000 lbs, 1050 F, reheat to 1000 F
Station gross heat rate	9200 Btu per kwh
Boiler efficiency	87 per cent
Heat added to steam	2400×10^6 Btu per hr
Fuel cost	\$0.30 per 10^6 Btu
Net generation	2.00×10^9 kwh per year

INVESTMENT COST

Boiler Plant

Site	\$ 300,000
Boiler plant building	3,000,000
Boiler equipment	6,200,000
Boiler plant piping	230,000
Combustion control and instruments	152,000
Fuel handling equipment	1,120,000
Draft system	1,280,000
Miscellaneous boiler plant equipment	326,000
Electrical equipment related to fuel burning	<u>500,000</u>
	13,108,000
Omissions at 10 per cent	<u>1,311,000</u>
Subtotal	14,419,000
Overheads and profit at 11 per cent	<u>1,586,000</u>
Subtotal	16,005,000
Engineering at 10 per cent	<u>1,601,000</u>
Total estimated investment	17,606,000

Turbogenerator Plant (from Table F.1)

\$13,888,000 $\times$ 2	<u>27,776,000</u>
Total investment	\$45,382,000

COST OF POWER

	<u>Annual Cost</u>	<u>Mills/kwh</u>
Fixed charges at 15 per cent	\$ 6,800,000	3.40
Fuel cost		
$\frac{2400 \times 10^9}{.87} \times \frac{\$0.30}{10^6} \times 7000$ hrs	5,780,000	2.89
Operating labor	300,000	.15
Annual maintenance	<u>681,000</u>	<u>.34</u>
Total cost	\$13,561,000	6.78

OPTIMUM TURBOGENERATOR PLANT INVESTMENT vs THROTTLE STEAM CONDITIONS FOR REGENERATIVE SUPERHEAT CYCLE

FIG. F.1

OPTIMUM HEAT RATE vs THROTTLE STEAM CONDITIONS FOR REGENERATIVE SUPERHEAT CYCLE

FIG. F.2

OPTIMUM TURBOGENERATOR PLANT INVESTMENT vs THROTTLE STEAM CONDITIONS FOR REGENERATIVE SUPERHEAT CYCLES WITH SEPARATORS

FIG. F.3

OPTIMUM HEAT RATE vs THROTTLE STEAM CONDITIONS FOR REGENERATIVE SUPERHEAT CYCLES WITH SEPARATORS

FIG. F.4

OPTIMUM TURBOGENERATOR PLANT INVESTMENT vs THROTTLE STEAM CONDITIONS FOR REGENERATIVE SATURATED STEAM CYCLES WITH SEPARATORS

FIG. F.5

OPTIMUM HEAT RATE vs THROTTLE STEAM CONDITIONS FOR REGENERATIVE SATURATED STEAM CYCLES WITH SEPARATORS

FIG. F.6

SUPERHEATED STEAM CYCLES

FIG. F.7

SUPERHEATED STEAM CYCLES WITH MOISTURE SEPARATION

FIG. F.8

ENTHALPY VARIATION OF STATE LINE END POINT
WITH CONDENSING TEMPERATURE FOR A STATE LINE
OF 82% STAGE EFFICIENCY AND 12% MOISTURE AT 80°F.

FIG. F. 13

FIG. F. 14

Partial Derivative of Enthalpy Drop with Respect to Condenser Cost per Unit of Flow

VARIAION OF G_{ex}/A_{ex} WITH $\frac{\partial(\Delta h)}{\partial\left(\frac{C_c}{G_{ex}}\right)}$

FIG. F. 15

OPTIMUM CONDENSER FLOW PER UNIT TURBINE EXHAUST AREA vs CONDENSING TEMPERATURE

FIG. F. 16

EXHAUST LOSS AS A FUNCTION OF CONDENSING TEMPERATURE FOR OPTIMUM TURBINE EXHAUST ANNULUS AREA AND CONDENSER SURFACE

FIG. F. 17

HEAT RATE vs INCREMENTAL COST OF FEED HEATERS
800°F REGENERATIVE SUPERHEAT CYCLE

$P_{THROTTLE} = 470 \text{ psia}$

FIG. F. 18

Incremental Cost of Condenser Above Base of
80°F Ex. Temperature, Millions of Dollars
HEAT RATE vs COST OF CONDENSER AND TURBINE
800°F REGENERATIVE SUPERHEAT CYCLE WITHOUT
SEPARATION, 4 FEEDHEATERS, $P_{\text{THROTTLE}} = 470 \text{ psia}$

FIG. F. 19

REGENERATIVE SUPERHEAT CYCLE
WITHOUT MOISTURE SEPARATION
 $t_1 = 800^\circ\text{F}$
3 FEED HEATERS
VARIATION OF PLANT INVESTMENT WITH HEAT RATE
FOR REGENERATIVE SUPERHEAT CYCLE WITHOUT
MOISTURE SEPARATION

FIG. F. 20

STEAM POWER PLANTS

Nomenclature

A	area, ft^2
C	cost, \$
C_a	incremental cost of heat, \$/Btu
EL	exhaust loss, Btu/lb
g	standard acceleration of gravity, 32.17 ft/sec^2
G	mass rate of flow, lb/hr
h	specific enthalpy, Btu/lb
HR	heat rate, Btu/kwh
I	investment, \$
J	mechanical equivalent of heat, 778 ft lb/Btu
$(N_H)_2$	final number of feedwater heaters
s	specific entropy, Btu/lb F
$SLEP$	state line end point
t	temperature, F
v	specific volume, ft^3/lb
∂	partial infinitesimal increment
Δ	finite increment

Subscripts

$()_0, ()_1, ()_2$	stages of approximation
$()_1$	throttle conditions
$()_2$	SLEP conditions
$()_3$	condensing conditions
$()_{blr}$	boiler
$()_c$	condenser
$()_{ex}$	exhaust
$()_t$	turbine
$()_{th}$	throttle