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REDUCED POWER OPERATION OF REACTOR PLANT
TURBINE-GENERATOR COMBINATION
TBR NO. 1

Introduction

In order to reduce the effect of reactor system component failures on plant on-stream efficiencies, it is proposed to study modifications of the TBR layout wherein the main circulating loops are divided into several "part capacity" loops.

The first phase of this study involves consideration of the effects of utilizing four core system heat exchangers in parallel, each of these exchangers being of the same size as the blanket heat exchanger.

Conclusions

From this study it is readily evident that manifolding of reactor system components is economically desirable (even necessary) until such time as the design of reactor equipment approaches the degree of reliability now attainable in modern turbogenerator components. Thus the development of suitable block valves and provisions in station layout for isolation of components without shutting down the station might be a highly paying proposition even where only a few heat exchangers are manifolded.

Discussion

Based on ORNL-CF-55-1-171, "Revision of Reactor Characteristics, TBR Design Memo No. 8," full load heat removal rates of 358 and 82 Mw for the core and blanket circuit respectively were used. The ratio of blanket to core power was held constant for all core power levels.

The heat exchanger and pump system characteristics were computed* on the assumption that the circuit can continue to operate following the loss of one (or more) primary core heat exchangers. Load capability of the turbogenerator system was calculated** at various reduced steam pressures. A cross plot of reactor and turbogenerator system capabilities with varying number of heat exchangers on stream and reduced turbine steam pressures is shown in Fig. 1.

* See Appendix II

** See Appendix I.

Reactor Capability at Various Steam Pressures
Versus
Turbogenerator Output

Figure 1

WFT 10-17-55

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It should be noted that by dropping steam temperature (from 437°F) as a heat exchanger is isolated (on failure) this study shows that the loss of one heat exchanger results in only a 13.5% drop in load, although 20% of the heat transfer surface is lost.

In this figure the electrical load in megawatts is the net station capability, and the turbogenerator curve for the particular 100-Mw machine specified for the TBR was computed for constant mechanical losses. The load decrease with lowered, saturated steam pressures takes into account reduced thermal efficiency. The cross plot showing the power obtainable with different numbers of the four core heat exchangers in service (plus the single blanket exchanger) has also been adjusted for thermal efficiency and the same mechanical losses to utilize the same abscissa as the turbogenerator curve. Thus the true operating point in each case is the intersection of the cross plot with the turbogenerator curve.

From the points of intersection of the curves on Fig. 1 and the data of Fig. 2, we have the following conditions.

3 heat exchangers	86.5 Mw net	at 294 psia	1,170,000 lb/hr steam flow
2 heat exchangers	64.0 Mw net	at 228 psia	900,000 lb/hr steam flow
1 heat exchanger	38.5 Mw net	at 147 psia	590,000 lb/hr steam flow

If we were to consider operation of the unit at these particular electrical outputs, it can be shown that a saving in off-stream penalty charges is realized that could be applied to valve development.

For instance, if it is assumed that for a particular network a net differential penalty of six mills per kilowatt hour is the price paid to meet load commitments

$$\$0.006 \times 100,000 \times 24 = \$14,400/\text{day for the full load of 100 Mw.}$$

Partial load operation for capacities outlined herein require the purchase of 13.5 Mw, 36.0 Mw or 61.5 Mw, i.e., a penalty charge of \$1944, \$5184 and \$8856 for a day's operation with 3, 2 or 1 heat exchanger respectively.

It should also be noted the turbine characteristics shown here are based on the operation of the "installed" turbine below the design point.

Steam Flow Vs Electrical Load For Various Pressures at Turbine

Figure 2

WFF 10-14-55

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More efficient utilization of the steam at reduced pressure could be accomplished by introduction at the optimum stage of a specially designed turbine at the expense of a greater capital investment.

An economic investigation will be required to determine the optimum number and size of heat exchangers to be used in each circuit with respect to the load carrying ability of the station. The economics will be a direct function of an (estimated) varying plant factor.

APPENDIX I

Turbogenerator Characteristics at Reduced Throttle Pressures*

The flow that a condensing steam turbine will pass varies in direct proportion to the initial pressure and in inverse proportion to the square root of the absolute temperature for a given control valve position.

In this study the control valves are assumed to be wide open and the turbine efficiency referred to the Rankine cycle is considered as constant. The assumption of constant efficiency may in some cases be in error but will suffice for a first approximation. True determination of the change in efficiency requires additional investigation to determine the effect of the changed flow on the exhaust efficiency and the effect of redistribution of stage energies within the turbine.

The flow passing ability at the lower steam conditions relative to the design steam conditions is obtained by means of the equation

$$\frac{W}{A} = C_1 \frac{P_1}{\sqrt{T_1}} \quad \text{lb/in}^2\text{-hr}$$

W = weight flow, lb/hr

A = effective flow area, sq in

C = constant for the value of flow per sq in/hr at critical pressure ratio

p = pressure, psia

T = absolute temperature

Thus,

$$\frac{W_2}{W_1} = \frac{p_2/\sqrt{T_2}}{p_1/\sqrt{T_1}} \quad \begin{array}{l} \text{(lower steam condition)} \\ \text{(design steam condition)} \end{array}$$

The ratio of theoretical steam rates is

$$\frac{\text{TSR}_1}{\text{TSR}_2} \quad \begin{array}{l} \text{(design steam condition)} \\ \text{(lower steam condition)} \end{array}$$

The ratio of outputs is then found

$$\frac{kw_2}{kw_1} = \frac{W_2}{W_1} \times \frac{\text{TSR}_1}{\text{TSR}_2}$$

* Salisbury, J. Kenneth, Steam Turbines and Their Cycles, John Wiley and Sons.

These data were plotted per Figure 1. On this curve a cross plot of heat exchanger capabilities revealed the operating point for each condition as outlined in the body of the memo. The turbine throttle steam flows for these points were obtained from Figure 2.

Figure 3 was used to make adjustments in thermal efficiencies while allowing for constant mechanical losses for this particular turbine generator.

APPENDIX II

Reactor System Capabilities at Partial Heat Exchanger Capacity

Assuming that the TBR core circuit can continue to operate following the loss of one or more (of four) primary heat exchangers, this portion of the study takes up the heat output capability of the remaining parts of the circuit. The relationship between reactor core power and steam temperature for varying numbers of heat exchangers on stream is indicated on Figure 6.

This study is limited to the heat output of the core circuit only and takes into account the loss of one or more heat exchangers, connected in parallel, and which can be successfully isolated from the remainder of the system. Failure of the main circulating pump, reactor vessel or common piping would cause the loss of the entire circuit.

A schematic diagram of the circuit is illustrated in Figure 4. Included are four core heat exchangers and one blanket heat exchanger. A single circulating pump is used in each of the two circuits.

Method of Analysis

Two calculations are required, based upon circuit constants previously developed*. One requires that flow conditions be established, and the other takes up the performance of the reactor and heat exchangers at these flow conditions.

To establish the flow rates expected, a pump performance curve and revised circuit are presented. The pump "head vs. flow" curve, Figure 5, is based upon the characteristics of a smaller pump (Westinghouse 400A)

* Unpublished TBR design studies.

TURBINE CYCLE EFFICIENCY
vs.
TEMPERATURE

Figure 3

WFT-11/26/54

10

Core and Blanket Circuits

FIGURE 4

540-10

ASSUMED PUMP CHARACTERISTICS

FIGURE 5

Reactor Power Vs. Steam Temperature

Figure 6

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WRT 10-12-55

540-12

designed for similar service. The revised circuit is taken from incomplete layout studies making use of four, rather than one, heat exchangers. The overall heat exchange surface area, tube size and length remain the same as in the original TBR study.

As heat exchangers are isolated on failure, circuit resistance is increased and the pump volumetric flow (capacity) adjusts in accordance with the pump characteristic curve. The equilibrium conditions in which the circuit resistances equal the available pump heads are indicated on Figure 5 for the cases of one, two, three and four heat exchangers on stream.

Calculations

A. Pressure Drop and Flow Relationships

Nomenclature

A - area, ft^2

D - diameter, ft

H - pressure drop, ft (fluid density 56.8 pounds/cu ft)

L - length, ft

V - velocity, ft/sec

W - flow rate,

C - cold fluid, 250°C

g - gravitational constant, 32.2 ft/sec^2

h - hot fluid, 300°C

ρ - density, lbs/ft^3

ψ - temperature function

Constants

W (design flow rate) - 11,448,000 lbs/hr

ρ (fluid density) - 51.4 lbs/cu ft at 300°C

56.8 lbs/cu ft at 250°C

A (flow cross sectional area) - 1.9688 sq ft for 22 in common piping

0.4729 sq ft for 10 in parallel piping

0.8186 sq ft (1950 tubes) in one heat exchanger

L (equivalent piping length) - 110 ft for common return
190 ft for common feed
85 ft for parallel return
85 ft for parallel feed
24.1 ft heat exchanger tube length

$D^{-4.8}$ (diameter function) for 22-in pipe - 0.1102

10-in pipe - 3.48

heat exchanger tube - 8.126×10^7

$$\psi(\text{temperature function}) * \text{for fluid at } 250^{\circ}\text{C} = 8.32 \times 10^{-14}$$
$$300^{\circ}\text{C} - 9.98 \times 10^{-14}$$

275°C - 9×10^{-14} ave temperature
in heat exchanger

The above functions are used to determine pressure drop in the piping according to the following equation,

$$dH/dL = (dF/dL)(\rho/144) = \psi W^{1.8} D^{-4.8**}$$

Reactor Pressure Drop

Two velocity heads are assumed

$$V(\text{cold stream entering}) = W/3600 \times \rho_c \times A = k W/\rho_c$$

$$V(\text{hot stream leaving}) = W/3600 \times \rho_h \times A = k W/\rho_h$$

The resulting loss in feet of cold fluid

$$(k W / \rho_c)^2 / 2g + (\rho_h / \rho_c)(k W / \rho_h)^2 / 2g$$

$$\text{or } (k W / \rho_c)^2 (1 + \rho_c / \rho_h) / 2g = H_1$$

$$V_c = W/3600 \times 56.8 \times 1.9688 = 2.484 \times 10^{-6} \text{ W}$$

$$H = (2.484 \times 10^{-6} \text{ W})^2 (1 + 56.8/51.4)/64.4$$

$$= 20.137 \times 10^{-14} \text{ W}^2$$

Gas Separator Pressure Drop

3.6 velocity heads assumed (hot fluid)

$$H = 3.6 (2.484 \times 10^{-6} \text{ W})^2 (56.8/51.4)/64.4$$

$$= 38.11 \times 10^{-14} \text{ W}^2$$

* Gaines, A. L., "Thermodynamic Properties and Relations for Solving Equation for Heat Transfer Inside Tubes and Equations for Pressure Drop through Tubes for D₂O and Solutions of UO₂SO₄ in D₂O, TBR Memo No. 7", CF-54-11-122. Values plotted on p. 20.

**** Ibid, p. 4.**

Heat Exchanger Losses (assuming four heat exchangers)

- a. Compression loss at header (cold fluid leaving)

$$V_c = 0.25 W / 3600 \times 56.8 \times 0.4729 = 2.585 \times 10^{-6} W$$

$$H = (2.585 \times 10^{-6} W)^2 / 64.4 = 10.376 \times 10^{-14} W^2$$

- b. Expansion loss at header (hot fluid entering)

$$H = 10.376 \times 10^{-14} W^2 \times 56.8 / 51.4 = 11.465 \times 10^{-14} W^2$$

- c. Expansion loss at tube exit (cold fluid)

$$V_c = 0.25 W / 3600 \times 56.8 \times 0.8186 = 1.493 \times 10^{-6} W$$

$$H = (1.493 \times 10^{-6} W)^2 / 64.4 = 3.461 \times 10^{-14} W^2$$

- d. Compression loss at tube entrance

$$H = 3.461 \times 10^{-14} W^2 \times 56.8 / 51.4 = 3.824 \times 10^{-14} W^2$$

- e. Tube friction losses*

$$\Delta H = \psi L W^{1.8} D^{-4.8}$$

$$H = (9 \times 10^{-14})(24.1)(W/4 \times 1950)^{1.8}(7.126 \times 10^7)(54.1/56.8) \\ = 14.52 \times 10^{-12} W^{1.8}$$

Piping Losses

- a. Common line (hot fluid)

$$H = (9.98 \times 10^{-14})(190)(W)^{1.8}(0.1102)(51.4/56.8) \\ = 1.89 \times 10^{-12} W^{1.8}$$

- b. Parallel line (hot fluid)

$$H = (9.98 \times 10^{-14})(85)(0.25 W)^{1.8}(3.48)(51.4/56.8) \\ = 2.2 \times 10^{-12} W^{1.8}$$

- c. Parallel line (cold fluid)

$$H = (8.32 \times 10^{-14})(85)(0.25 W)^{1.8}(3.48) \\ = 2.02 \times 10^{-12} W^{1.8}$$

- d. Common line (cold fluid)

$$H = (8.32 \times 10^{-14})(110)(W)^{1.8}(0.1102) \\ = 1.01 \times 10^{-12} W^{1.8}$$

* Ibid., p. 4.

Total Pressure Drop

$$\sum H = 87.4 \times 10^{-14} W^2 + 22.6 \times 10^{-12} W^{1.8} \text{ (4 heat exchangers)}$$

For a flow of 11,448,000 pounds an hour, a pressure head of about 225 feet is developed. Flow and head relationships are indicated on Figure 5 for the system using 4, 3, 2 or 1 heat exchangers.

B. Heat Transfer Relationships

Nomenclature

A - area, ft²

C_p - specific heat, Btu/lb x °F

D - diameter, ft

H, h - heat transfer coefficient, Btu/ft² x hr x °F

K - conductivity, Btu/ft² x hr x °F/in

Q - heat output, Btu's

R - resistance to heat flow, 1/U

T - temperature

U - overall heat transfer coefficient, Btu/ft² x hr x °F

W - flow rate, lbs/hr

Subscripts

i - inside surface of heat exchanger tube

o - outside surface of heat exchanger tube

r - reactor

s - steam

w - wall at heat exchanger tube

x - wall thickness of heat exchanger tube, inches

∅ - temperature function

ΔT_m - log mean temperature across heat exchanger

ΔT_r - temperature (inlet to outlet) gradient across reactor

Δt - ΔT_r/2

Development of the heat exchanger coefficient, U_o

$$Q = U_o A_o \Delta T_m$$

$$1/U_o = D_o/D_i h_i + D_o/D_w d_w + 1/h_s + 1/h_o$$

$$h_i = (\phi_i \times W_1^{0.8})/D_i^{1.8*}$$

$$\phi_i = 25.55 \times 10^{-3} ** \text{ (an average temperature of } 275^\circ\text{C was assumed to determine temperature function } \phi_i \text{)}$$

$$D_o = 0.375 \text{ in} \quad W_1 = 11,448,000/7800 = 1468 \text{ lbs/hr x tube (4 Ht-X's)}$$

$$D_i = 0.277 \text{ in} \quad = 10,100,000/5850 = 1726 \text{ lbs/hr x tube (3 Ht-X's)}$$

$$D_w = 0.326 \text{ in} \quad = 8,250,000/3900 = 2115 \text{ lbs/hr x tube (2 Ht-X's)}$$

$$= 4,850,000/1950 = 2487 \text{ lbs/hr x tube (1 Ht-X's)}$$

$$h_i = 7684 \text{ Btu/hr x sq ft x deg F (4 Ht-X)}$$

$$= 8746 \text{ Btu/hr x sq ft x deg F (3 Ht-X)}$$

$$= 10,290 \text{ Btu/hr x sq ft x deg F (2 Ht-X)}$$

$$= 11,715 \text{ Btu/hr x sq ft x deg F (1 Ht-X)}$$

$$h_w = K_w/x_w = 130/0.049 = 2653 \text{ Btu/hr x sq ft x deg F}$$

$$K(SS) = 130 \text{ Btu/hr x deg F}$$

$$x = 0.049 \text{ in}$$

$$h_s(\text{scale}) = 2000 \text{ Btu/hr x sq ft x deg F (assumed)}$$

$$h_o = \alpha(\Delta T_o)^\beta = 7673 \text{ Btu/hr x sq ft x deg F}$$

For the sake of simplicity the magnitude of this coefficient was taken directly from TBR design studies (not published). The resistance offered by the steam side coefficient is assumed to decrease by the 0.9 power for the conditions of partial loading. As the reliability of the above formula is doubtful, for the conditions of the study, and as this resistance change only represents a minor part of the overall resistance, refinement was not considered necessary.

* Ibid., p. 5.

** Ibid., Figure Hd1, p. 25.

Tabulation of Film Coefficients and Resistances

Ht-X's	h_i	h_w	h_s	h_o	U_o
4	7684	2653	2000	7673	806.5
3	8746	2653	2000		822.9
2	10,290	2653	2000		841.0
1	11,715	2653	2000		854.4

	R_i	R_w	R_s	R_o	$R_t \times 10^{-4}$
4	1.7618	4.335	5	1.3033	12.4
3	1.548	4.335	5	1.269	12.152
2	1.3156	4.335	5	1.239	11.89
1	1.1556	4.335	5	1.213	11.704

Heat Balance

$$Q = U_o A_o \Delta T_m \text{ (heat passing from soup to steam across Ht-X)}$$

$$= W C_p \Delta T_r \text{ (heat generated in reactor)}$$

For the condition in which 3 heat exchangers are used, U_o , A , W and C_p are known, therefore

$$\frac{U_o A}{W C_p} = \frac{\Delta T_r}{\Delta T_m}$$

$$U_o A = (822.9)(13840.5) = 11.3893 \times 10^6 \text{ Btu/hr} \times ^\circ\text{F}$$

$$W C_p = (10,100,000)(1.1805) = 11.923 \times 10^6 \text{ Btu/hr} \times ^\circ\text{F}$$

$$A = 4613.5 \text{ sq ft per exchanger}$$

$$C_p \text{ for soup at } 275^\circ\text{C is } 1.1805 \text{ Btu/pound}$$

$$\Delta T_r / \Delta T_m = 11,3893 / 11.923 = 0.9952$$

$$\Delta T_r = (275 + \Delta t) - (275 - \Delta t) = 2\Delta t$$

Δt is the difference between the average temperature and the temperature of the fluid entering or leaving the reactor. These are assumed equal.

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$$\Delta T_m = \frac{(275 + \Delta t - t_s) - (275 - \Delta t - t_s)}{\ln (275 + \Delta t - t_s) / (275 - \Delta t - t_s)}$$

$$0.9952 = \Delta T_r / \Delta T_m = \ln (275 + \Delta t - t_s / 275 - \Delta t + t_s)$$

$$t_s (\text{steam temperature } ^\circ\text{C}) = 275 - 2.1658 \Delta t$$

The above formula relates the reactor temperature gradient and corresponding reactor heat output, with the necessary steam temperature to remove the heat from the reactor system. In the above example, 3 heat exchangers were used. Below is a tabulation for all conditions considered in the study. The results are plotted in Figure 6, as reactor power output vs. steam temperature.

4 Ht-X's	$t_s = 275 - 1.996 \Delta t$
3 Ht-X's	$t_s = 275 - 2.1648 \Delta t$
2 Ht-X's	$t_s = 275 - 2.7236 \Delta t$
1 Ht-X	$t_s = 275 - 3.01877 \Delta t$

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