STEAM GENERATORS FOR HIGH-TEMPERATURE GAS-COOLED REACTORS

A. P. Fraas
M. N. Ozisik
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A. P. Fraas    M. N. Ozisik

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ABSTRACT

An analytical approach and an IBM machine code were prepared for the design of once-through steam generators for gas-cooled reactors for both axial-flow and cross-flow tube matrices. The codes were applied to investigate the effects of steam generator configuration, tube diameter, extended surface, type of cooling gas, steam and gas temperature and pressure conditions, and the pumping power-to-heat removal ratio on the size, weight, and cost of steam generators. The results indicate that the least expensive and most promising unit for high-temperature high-pressure gas-cooled reactor plants employs axial-gas flow over 0.5-in.-diam bare U-tubes arranged with their axes parallel to that of the shell. The proposed design is readily adaptable to the installation of a reheater and is suited to conventional fabrication techniques.

Charts are presented to facilitate the design of both axial-flow and cross-flow steam generators for gas-cooled reactor applications.
ACKNOWLEDGMENTS

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INTRODUCTION

The successful development of the Calder Hall reactors in England has led to the construction of over 20 gas-cooled reactors for experimental and central station applications in England and the U. S. A. The total cost of the steam generators for the plants built in the 1957–63 period is over $100,000,000. A new design approach that was evolved in an effort to effect major reductions in the cost of steam generators for advanced gas-cooled reactors is presented here.

The steam generator of a nuclear power plant is important not only because it is one of the most expensive components of the plant but also because of its effects on the over-all plant layout and design. Changes in the reactor coolant or steam conditions or in the steam-generator configuration affect not only the cost of the steam generator but also the cost of the reactor, the shield, the containment vessel, and other components. It appears that this is particularly true of high-temperature gas-cooled reactors employing ceramic fuel elements and that improvements in the steam generator afford one of the most important ways in which to effect reductions in the cost of this type of power plant.

Several approaches to reductions in the cost of the steam generator have been suggested. Some of these, such as increased gas system pressures, should also reduce the cost of the reactor and other plant components, since they should reduce their size for a given power output. An ORNL study\(^1\) carried out in 1958 showed, however, that while increasing the gas system pressure was effective in reducing the steam generator size, the weight and cost of the pressure vessel increased so much that there appeared to be little advantage in gas system pressures above 300 psi for the serpentine-tube type of steam generator used in the low-pressure, low-temperature British plants.
A careful examination of the 1958 high-pressure design disclosed that the serpentine-tube configuration is unsuited to the small vessel diameters required for high-pressure, high-temperature plants. In order to avoid these shortcomings, a new type of axial-tube steam generator was evolved in 1959 to obtain more efficient utilization of the space within the pressure vessel. Other steps taken to obtain a more compact tube matrix included reductions in the tube diameter and the use of bare rather than finned tubes.

The study described here was undertaken to provide a sound basis both for the optimization of either axial-flow or cross-flow steam generators and for the evaluation of the relative merits of major design features.

SUMMARY

The first step in the study was to develop an analytical technique for obtaining an explicit solution for steam generator size for a given set of design conditions for axial-tube configurations. This analytical approach was then extended to cross-flow tube matrices. Concurrently, a series of layout studies was made to aid in evaluating fabrication problems for various configurations, and unit costs for the principal fabrication operations were estimated. A fairly comprehensive parametric study was then carried out to establish the effects of the principal parameters for what appeared to be the more promising combinations of conditions.

This report of the study is divided into nine sections: (1) a discussion of the basis for and the development of the analytical approach for hand-calculation purposes; (2) the procedure used for computing-machine calculations; (3) the bases for cost estimates; (4) typical design charts; (5) the results of the parametric study; (6) a discussion of the fabrication and design considerations, including shell temperature control, hot spots, stress considerations, and boiling-flow stability; (7) a proposed design that includes the more promising proportions and features developed in the study; (8) an analysis and a discussion of the effect of power changes on the temperature in the steam generator and its control;
and (9) a discussion of two-phase pressure drop and the computer calculation for the two-phase pressure drop.

The results of the study have many important implications. The more important appear to be the following:

1. The analysis indicates that, where the gas system pressure and the mean temperature difference are sufficiently high to give average heat fluxes in the steam generator in excess of about 30,000 Btu/hr·ft² of internal tube surface, a smaller and less expensive steam generator can be designed for axial gas flow over bare tubes than can be obtained with axial flow over axially finned tubes or with cross flow over either bare or finned tubes.

2. The analysis indicates that, where the gas system pressure and the mean temperature difference are not high enough to give average heat fluxes above about 30,000 Btu/hr·ft², the least expensive steam generator is obtained with cross flow over finned tubes.

3. If an axial-flow unit can be used, a tube internal diameter of 0.4 to 0.6 in. yields the smallest, lightest, and least expensive steam generator for any given set of design conditions. In the cross-flow units, 0.5-in.-diam tubes do not look attractive because difficulty with tube support would be a serious problem for tube diameters of less than about 1.0 in.

4. The study indicates that it should be possible to design units with Croloy tubes for gas temperatures up to 1150 to 1250°F. If this is done, it appears that there is little incentive to increase the peak gas system temperature beyond this range, since the higher cost of the refractory alloys required largely offsets the savings that can be effected through reductions in steam generator size and weight.

5. For plants designed to produce steam at pressures of the order of 2500 psi, steam generator cost considerations strongly favor the use of a reactor inlet gas temperature of at least 650°F.

6. Important reductions in the size, especially the height, and the cost of axial-flow steam generators can be obtained by increasing the gas system pressure up to about 500 psi. Further increases in pressure yield relatively small further reductions in size and cost.
7. For axial-flow steam generators, the capital cost is reduced sufficiently so that, for minimum costs, the pumping power-to-heat removal ratio in the steam generator should be about 0.5%.

8. The extra cost and complication of a reheater is more than justified for high-temperature gas-cooled reactor plants by the resulting increase in cycle efficiency and the consequent reductions in fuel cycle costs and reactor system capital charges.

9. An examination of the size and cost data from the parametric study, together with the problems that must be solved in developing a complete detailed design for fabrication, indicates that a configuration more promising than straight axial tubes can be obtained with U-tubes arranged so that the legs straddle an annular baffle. The hot gas rising through the central portion of the shell and returning downward through an outer annulus flows countercurrent to the water-steam flow inside the tubes. The U-tube configuration facilitates design for good temperature control of the pressure vessel, incorporation of a reheater, alleviation of hot-spot problems, reduction of the differential expansion between the tubes and the shell, relief of the stresses associated with support of the tubes, and reduction of the over-all height of the steam generator.

10. Control considerations indicate that the hot gas temperature entering the steam generator should not be more than 100 to 150°F above the superheated steam outlet temperature. It appears that the increased cost of the steam generator and the increased gas pumping costs will be roughly offset by reductions in design costs, the cost and complication of the control equipment, increased reliability, and reductions in maintenance costs.

NOMENCLATURE

\begin{align*}
\text{a} & \quad \text{Heat transfer surface area, ft}^2 \\
\text{A} & \quad \text{Flow passage cross-sectional area, ft}^2 \\
\text{A}_{\min} & \quad \text{Minimum free-flow area per tube, ft}^2 \\
\text{b} & \quad \text{Fin thickness, in.} \\
\text{B} & \quad \text{Temperature difference between gas outlet and feedwater inlet at full power, °F}
\end{align*}
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
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<tbody>
<tr>
<td>( c_p )</td>
<td>Specific heat, Btu/lb·°F</td>
</tr>
<tr>
<td>( C_{h'}, C_f )</td>
<td>Constants</td>
</tr>
<tr>
<td>( d )</td>
<td>Tube diameter, ft</td>
</tr>
<tr>
<td>( \bar{d} )</td>
<td>Tube diameter, in.</td>
</tr>
<tr>
<td>( D )</td>
<td>Shell diameter, ft</td>
</tr>
<tr>
<td>( D_e )</td>
<td>Equivalent hydraulic diameter for flow outside tube, ft</td>
</tr>
<tr>
<td>( \bar{D}_e )</td>
<td>Equivalent hydraulic diameter for flow outside tube, in.</td>
</tr>
<tr>
<td>( D_v )</td>
<td>Volumetric hydraulic diameter, ft</td>
</tr>
<tr>
<td>( f )</td>
<td>Friction factor, dimensionless</td>
</tr>
<tr>
<td>( F )</td>
<td>Pumping power-to-heat removal ratio, dimensionless</td>
</tr>
<tr>
<td>( g )</td>
<td>Acceleration of gravity, ( 4.18 \times 10^8 ) ft/hr²</td>
</tr>
<tr>
<td>( G )</td>
<td>Mass flow rate, lb/ft²·hr</td>
</tr>
<tr>
<td>( h )</td>
<td>Heat transfer coefficient, Btu/hr·ft²·°F</td>
</tr>
<tr>
<td>( \Delta h )</td>
<td>Change in enthalpy, Btu/lb</td>
</tr>
<tr>
<td>( k )</td>
<td>Thermal conductivity, Btu/hr·ft·°F</td>
</tr>
<tr>
<td>( L )</td>
<td>Length, ft</td>
</tr>
<tr>
<td>( l )</td>
<td>Width of cross-flow steam-generator matrix, ft</td>
</tr>
<tr>
<td>( m )</td>
<td>Number of tubes per layer in a cross-flow tube matrix</td>
</tr>
<tr>
<td>( M )</td>
<td>Molecular weight of gas</td>
</tr>
<tr>
<td>( N )</td>
<td>Total number of tubes in a tube matrix</td>
</tr>
<tr>
<td>( n )</td>
<td>Number of layers per pass in a cross-flow matrix</td>
</tr>
<tr>
<td>( n )</td>
<td>Ratio of part-load power output to full-power output, dimensionless (chap. 8 only)</td>
</tr>
<tr>
<td>( n_f )</td>
<td>Number of fins per unit length</td>
</tr>
<tr>
<td>( P )</td>
<td>Pressure, psfa</td>
</tr>
<tr>
<td>( q )</td>
<td>Latent heat of vaporization, Btu/lb</td>
</tr>
<tr>
<td>( Q )</td>
<td>Heat flow rate, Btu/hr</td>
</tr>
<tr>
<td>( Pr )</td>
<td>Prandtl number, dimensionless</td>
</tr>
<tr>
<td>( \Delta P )</td>
<td>Pressure drop, psf</td>
</tr>
<tr>
<td>( R )</td>
<td>Resistance to heat flow, hr·ft²·°F/Btu</td>
</tr>
<tr>
<td>( Re )</td>
<td>Reynolds number, dimensionless</td>
</tr>
<tr>
<td>( S )</td>
<td>Tube spacing, ft</td>
</tr>
<tr>
<td>( St )</td>
<td>Stanton number, dimensionless</td>
</tr>
<tr>
<td>( T )</td>
<td>Temperature, °R (i.e., 460 + t in °F)</td>
</tr>
<tr>
<td>( \Delta t_L )</td>
<td>Logarithmic mean temperature difference, °F</td>
</tr>
</tbody>
</table>
t Temperature, °F, or shell thickness, ft
\(\bar{t}\) Shell thickness, in.
\(\delta t\) Temperature drop of gas, °F
\(\Delta t\) Film temperature drop, °F
U Over-all coefficient of heat transfer, Btu/hr*ft\(^2\)°F
V Specific volume, ft\(^3\)/lb
W Total weight flow rate of circulating fluid, lb/hr
X Ratio of mass flow rate at any power to mass flow rate at full power, dimensionless (chap. 8 only)
x Distance along tube, ft
dx Differential distance along tube, ft
y Fin height, in.
Y Ratio of the gas temperature drop at part load to the gas temperature drop at full power, dimensionless (chap. 8 only)
z Ratio of gas-outlet to feedwater-inlet temperature difference at part load to that at full power (chap. 8 only)
\(\alpha\) Fraction by weight of total steam flowing through the reheater, dimensionless
\(\gamma\) Ratio of outside diameter to inside diameter of tube, dimensionless
\(\lambda\) Ratio of pressure drop of gas through duct to pressure drop of gas through matrix
\(\rho\) Density, lb/ft\(^3\)
\(\mu\) Viscosity, lb/hr*ft
\(\epsilon\) Ratio of fin area to the total heat transfer surface area outside finned tubes
\(\eta\) Fin effectiveness
\(\eta\) Ratio of gas-inlet to steam-outlet temperature difference at part load to that at full load (chap. 8 only)
\(\eta'\) Area-weighted fin effectiveness
\(\theta\) Angle with the horizontal, deg
\(\xi\) Quality of steam, lb of steam per lb of (steam + liquid)

Subscripts

B Boiler
d Gas riser duct
E  Economizer
G  Gas
I  Inside of tube
J  Any specified section
L  Spacing in direction of flow
M  Matrix
O  Outside of tube
P  Pressure vessel
R  Reheater
S  Superheater or steam
T  Tube
T  Transverse
W  Water
1  Economizer plus first portion of boiler in a U-tube matrix
2  Second portion of the boiler plus the superheater in a U-tube matrix
3  Reheater matrix in a steam generator with a U-tube matrix

Quantities with primes refer to the condition at a power level other than full power (chap. 8 only).
1. BASIC ANALYSIS

Survey of the Problem

Most approaches to the design of heat exchangers make use of trial-and-error techniques that depend heavily on the judgment of the designer. There are, however, so many possible combinations and permutations of the variables for the case at hand that this time-consuming approach seemed unsatisfactory. The principal parameters are the gas chosen (He, CO₂, N₂, etc.), the gas pressure, the gas inlet and outlet temperatures, the steam system pressure and temperature, the tube diameter, the extent to which an extended surface is employed on the gas side, the pumping power-to-heat removal ratio for the gas system, and the design power output. The principal dependent variables are the number of tubes and the tube diameter, length, and center-to-center spacing. Once these are established, it is easy to calculate the number, weight, and cost of the tubes and the size, weight, and cost of the pressure vessel for a given set of unit costs. The design output affects only the diameter of the pressure vessel and, to a lesser extent, its cost per unit of output but has no effect on the tube length or center-to-center spacing for given values of the other conditions.

In attempting to narrow the range of variables, it was decided that previous design studies clearly showed the advantages of the once-through type of boiler for high-temperature gas-cooled reactor applications. This type of steam generator gives a minimum number of tube-to-header joints, eliminates steam-separating problems, minimizes the number of shell penetrations, and generally gives a simpler, less expensive, and more compact system than any other type. Experience with coal- and oil-fired boilers has shown that the once-through type of boiler is well suited to applications in which the steam is to be delivered at pressures in excess of 1500 psi and temperatures in excess of 900°F, that is, the steam conditions ordinarily used in the more modern coal-fired steam plants. The flow-stability and hot-spot problems that have been the principal objection to the use of once-through boilers in oil- or coal-fired plants should be much less serious in a gas-cooled reactor plant because the
gas-side temperature and heat-flux distribution should be much more uniform and predictable. (This is discussed at some length in later sections.)

It was found that, by limiting the study to once-through boilers, the water velocity through the tubes could be eliminated as a variable. Experience with once-through boilers has shown that the water velocity should be 30 ft/sec or more at the boiler outlet region under full-load conditions if difficulty with water flow stability is to be avoided at low loads during startup or shutdown. Higher water velocities could be used without entailing excessive pressure drops on the water side, but this would be disadvantageous, since it would increase the tube length and the steam generator height, both of which are inclined to be somewhat greater than is desirable. For a given set of gas and steam conditions, the gas flow rate per tube in a once-through boiler is determined by the water flow rate per tube, thus further simplifying the analysis.

Once the water-side velocity is fixed, the heat transfer coefficients on the water side are simply a function of tube diameter and position along the tube. Analysis disclosed that the heat transfer coefficients in the economizer could be taken as independent of position, and a similar situation prevails in the boiler. The large changes in the physical properties of superheated steam in the vicinity of the saturation point lead to substantial changes in the heat transfer coefficient, so it seemed desirable for calculational purposes to break the superheater into a number of regions. Since the gas-side flow rate per tube is fixed for a given water flow rate and given gas and steam conditions, the gas-side heat transfer coefficient is a function primarily of the equivalent passage diameter, that is, the tube spacing.

With the simplifications that this approach made possible, it was found that the balance of the parameters could be related in such a fashion that the problem could be reduced to two simultaneous equations which could be solved either graphically or analytically. The first of the two equations relates the pumping power-to-heat removal ratio to the heat absorbed by the water and the gas pressure drop and flow rate. The second equation relates the heat absorbed by the water to the heat given up by
the gas, the latter being a function of the tube surface area and the
gas-side equivalent passage diameter.

The derivation of the basic equations for the pumping power-to-heat
removal ratio and the tube length are presented in the following sections
for both axial-flow and cross-flow steam generators.

Single-Pass, Axial-Flow, Bare-Tube Steam Generator

The first and simplest case considered is that of Fig. 1, a single-
pass, counterflow, bare-axial-tube steam generator. Steam is generated
inside the tubes while the hot gas flows between the tubes. For analyti-
cal purposes the steam generator can be divided into three distinct regions
on the basis of the heat transfer characteristics on the water side: the
economizer, the boiler, and the superheater.

It is helpful in visualizing the problem to examine Fig. 2, which
shows the temperature and over-all heat transfer coefficient distribution
as a function of the fraction of the heat transferred to the water. It
is apparent from this figure that the water temperature in the economizer
rises continuously to the saturation point; in the boiler it remains al-
most constant until all the water is vaporized; while the temperature of
the superheated steam rises continuously to the steam outlet. The over-
all heat transfer coefficients for the economizer and the boiler section
are almost independent of position, and hence a single mean value was
evaluated for each of these sections. The heat transfer coefficient for
the superheated steam is quite sensitive to temperature changes in the
region near the saturation point, and hence allowance for this effect was
made by dividing the superheater into a series of regions. While the
solid lines in Fig. 2 indicate abrupt changes in the over-all heat trans-
fer coefficient at the transition from one region to another, these simply
reflect the assumptions made for convenience in the calculations. Actually
the transitions would be less abrupt, as indicated by the dashed lines.

For a single tube of the matrix, the pressure drop for the flow out-
side the tube over a section of length $L_t$ is
Fig. 1. Tube Configuration for Straight-Tube Axial-Flow Steam Generators with the Tube Matrix in an Annular Region Surrounding a Central Hot Gas Riser, or Duct.
Fig. 2. Typical Axial Temperature Distribution and Heat Transfer Coefficient Values vs Percentage of Heat Transferred for Axial-Flow Bare-Tube Steam Generators.

\[
\Delta P_0 = f \frac{g^2}{2 \rho_0} \frac{L_t}{D_{eo}},
\]

where the friction factor is approximately\(^2\)

\[
f = 0.21 \times \text{Re}^{-0.2} \quad \text{5000 < Re < 200,000}
\]
and

\[ \text{Re} = \frac{G_o D_e \omega}{\mu_o} . \]

The heat gain and the heat loss of the fluids inside and outside the tube are

\[ G_1 A_1 \Delta h_1 = G_o A_o \Delta h_o . \quad (2) \]

Eliminating \( G_o \) between Eqs. (1) and (2) and solving for \( L_t \) gives

\[ L_t = \left( \frac{2g}{0.21} \right) \Delta P_o \frac{\rho_o}{\mu_o^0.2G_1^1.8} \left( \frac{\Delta h_o}{\Delta h_1} \right)^{1.8} \left( \frac{A_o}{A_1} \right)^{1.8} D_{eo}^{1.2} . \quad (3) \]

The pumping power-to-heat removal ratio, \( F_o \), is the ratio of the pumping power required for gas flow outside the tube to the heat removal from the tube, and it may be expressed as

\[ F_o = \frac{A_o G_o \Delta P_o}{778 \rho_o A_1 \Delta h_o} = \frac{\Delta P_o}{778 \rho_o \Delta h_o} . \quad (4) \]

Substituting \( \Delta P_o \) from Eq. (4) into Eq. (3) gives

\[ L_t = \frac{778 \times 2g}{0.21} \frac{\rho_o^2}{\mu_o^0.2G_1^1.8} \frac{\Delta h_1^2.8}{\Delta h_1^1.8} \left( \frac{A_o}{A_1} \right)^{1.8} D_{eo}^{1.2} , \quad (5) \]

where \( g = 4.18 \times 10^8 \text{ ft/hr}^2 \). The term \( (A_o/A_1) \) can be evaluated by assuming an infinite array of tubes so that

\[ A_o = \frac{T}{4} d_o D_{eo} . \quad (6) \]
If the ratio of the tube outer and inner diameters is defined as $\gamma$, that is,

$$\gamma = \frac{d_0}{d_1},$$

then

$$\frac{A_0}{A_1} = \frac{\pi \gamma d_1 D_{eo}}{\pi d_1^2} = \frac{\gamma D_{eo}}{d_1}.$$  \hspace{1cm} (8)

Substituting Eq. (8) into Eq. (5) and solving for $F_0$ then gives the first of the two simultaneous equations referred to above:

$$F_0 = 0.323 \times 10^{-12} \left[ \frac{\mu_o^0.2 d_1^{1.8} \Delta h_i^{1.8}}{\rho_o^2 \Delta h_o^{2.8} \gamma} \right] \left( \frac{d_1}{d_0} \right)^{1.8} \frac{L_t}{D_{eo}^3}. \hspace{1cm} (9)$$

If it is assumed that the gas on the outside of the tubes is a perfect gas, its density is

$$\rho_o = \frac{PM}{1544T}. \hspace{1cm} (10)$$

The second of the simultaneous equations for the tube length can be derived from heat transfer considerations, that is,

$$\pi d_1 L_t U_i \Delta t_L = \frac{\pi}{4} d_1^2 G_i \Delta h_i$$

and

$$L_t = \frac{1}{4} \frac{U_i G_i \Delta h_i}{\Delta t_L} d_1. \hspace{1cm} (11)$$
In Eq. (11) the over-all coefficient of heat transfer $U_1$ based on the internal tube surface can be evaluated from the tube internal and external heat transfer coefficients; thus

$$\frac{1}{U_1} = \frac{1}{h_1} + \frac{(\gamma - 1) d_i}{2k_t} + \frac{1}{\gamma h_o}.$$  \hspace{1cm} (12)

The heat transfer coefficient for flow inside the tube, $h_1$, differs in form for the economizer, boiler, and superheater sections. For the economizer,$^3$

$$h_1 = 0.023 \frac{k_i}{d_i} \frac{Re^{0.8} Pr^{0.4}}{}$$

$$= 0.023 \frac{c_i^{0.4} \mu_i^{0.6} G_i^{0.8}}{\mu_i^{0.4} d_i^{0.2}}.$$  \hspace{1cm} (13)

For the superheater,$^4$

$$h_1 = 0.0266c \frac{\mu_i^{0.2} G_i^{0.8}}{\mu_i^{0.4} d_i^{0.2}}.$$  \hspace{1cm} (14)

The heat transfer coefficient for boiling is high compared with the gas-side heat transfer coefficient and is relatively insensitive to vapor quality up to about 60%. It then falls off to 1000 to 2000 Btu/hr·ft$^2$·°F but subsequently increases again to about 6000 Btu/hr·ft$^2$·°F. Thus a constant heat transfer coefficient of 5000 Btu/hr·ft$^2$·°F represents a good approximation for the water side in the boiler region and greatly simplifies the analysis.

The heat transfer coefficient for the gas flow outside the tube for the economizer, boiler, and superheater sections is

$$h_o = 0.023 \frac{c_o^{0.4} \mu_o^{0.6} G_o^{0.8}}{\mu_o^{0.4} D_o^{0.2}}.$$  \hspace{1cm} (15)
From Eqs. (2) and (8)

\[ G_o = G_i \frac{\Delta h_i}{\Delta h_o} \frac{d_i}{\gamma D_{eo}} \]  

Substituting Eq. (16) into Eq. (15), gives

\[ h_o = 0.023 \frac{c_p^0,4k_{0,6}}{\mu_o^0,4} \left( \frac{G_i}{\Delta h_o} \right)^{0.8} \left( \frac{d_i}{\gamma} \right)^{0.8} \frac{1}{D_{eo}} \]  

The logarithmic mean temperature difference, \( \Delta t_L \), can be evaluated for each section if the inlet and outlet temperatures at both ends of each section are known. Assuming that the inlet and outlet temperatures of the gas and the water are fixed at the feedwater-inlet and steam-outlet ends of the steam generator, the temperatures at the ends of the boiler section can be evaluated. If the pressure drop on the steam side is small compared with the system steam pressure, the temperature in the boiler region may be taken as equal to the saturation temperature at the steam system pressure. Thus \( \Delta h_{1B} \) is equal to the heat of evaporation of steam at this temperature. Since the inlet temperature of the feedwater, the saturation temperature of steam, and the steam outlet temperatures are all known, \( \Delta h_{1B} \) and \( \Delta h_{1S} \) are defined. Neglecting the variation in the specific heat of the gas with changes in temperature, the temperature drop of the gas along a section \( j \) can be evaluated from the relation:

\[ \delta t_{oj} = \frac{\Delta h_{1j}}{\sum \Delta h_i} \sum \delta t_o \]  

where

\[ \sum \Delta h_i = \Delta h_{1E} + \Delta h_{1B} + \Delta h_{1S} \]  

\[ \sum \delta t_o = \text{total gas temperature drop over the steam generator}, \]

\( j = \text{economizer, boiler, or superheater} \).
The logarithmic mean temperature difference, $\Delta t_L$, for each section can be calculated since the gas and steam temperatures at the inlet and outlet of each section are known. In Eq. (9) the enthalpy drop of the gas, $\Delta h_o$, along a section $j$ can be evaluated from

$$\Delta h_{o,j} = \sum \frac{\Delta h_{i,j}}{\Delta h_i} \sum \Delta h_o,$$

(19)

where $\sum \Delta h_o$ is the total enthalpy drop of the gas over the steam generator.

**Procedure for Hand Calculations**

The principal parameters, such as the gas chosen, the gas and steam pressures, the inlet and outlet temperatures of the gas, the feedwater-inlet and steam-outlet temperatures, the mass flow rate of the water, and the tube diameter and thickness, are conveniently fixed in preparing a tentative design for a steam generator. The dependent variables that determine the pumping power-to-heat removal ratio are then the tube length and the center-to-center spacing (or equivalent diameter for gas flow). For conditions such as this, Eqs. (9) and (11) are most useful in evaluating the pumping power-to-heat removal ratio and the tube length for the different values of the equivalent passage diameter for gas flow.

The detailed procedure for the calculations follows directly from the derivations given above. The individual heat transfer coefficients for the typical set of conditions given in Table 1 can be expressed as a function of tube inside diameter and $D_{eo}$, as follows (for convenience the constants have been modified so that $d_i$ and $D_{eo}$ are expressed in inches, and $d_{o}$ is taken as equal to $1.25d_i$):


### Table

<table>
<thead>
<tr>
<th></th>
<th>Economizer</th>
<th>Boiler</th>
<th>Superheater</th>
</tr>
</thead>
<tbody>
<tr>
<td>$h_o$, Btu/hr·ft²·°F</td>
<td>(\frac{d_{1}^{0.8}}{D_{eo}})</td>
<td>(\frac{d_{1}^{0.8}}{D_{eo}})</td>
<td>(\frac{d_{1}^{0.8}}{D_{eo}})</td>
</tr>
<tr>
<td>$h_i$, Btu/hr·ft²·°F</td>
<td>(\frac{2115}{d_{1}^{0.2}})</td>
<td>5000</td>
<td>(\frac{2320}{d_{1}^{0.2}})</td>
</tr>
<tr>
<td>Wall conductance, Btu/hr·ft²·°F</td>
<td>(\frac{1200}{d_{1}})</td>
<td>(\frac{1200}{d_{1}})</td>
<td>(\frac{1200}{d_{1}})</td>
</tr>
</tbody>
</table>

Similarly, the tube length can be expressed as a function of the tube diameter and the over-all heat transfer coefficient based on the internal surface:

\[
L_t = \frac{1}{4} \frac{G_1 \Delta h_1}{\Delta t_L \cdot \delta_1}
\]

and

<table>
<thead>
<tr>
<th></th>
<th>Economizer</th>
<th>Boiler</th>
<th>Superheater</th>
</tr>
</thead>
<tbody>
<tr>
<td>$L_t$, ft</td>
<td>(\frac{2.66 \times 10^4 \delta_1}{U_{1E}})</td>
<td>(\frac{2.39 \times 10^4 \delta_1}{U_{1B}})</td>
<td>(\frac{1.89 \times 10^4 \delta_1}{U_{1S}})</td>
</tr>
</tbody>
</table>

The ratio of the pumping power for gas flow over the tubes in section j to the total heat removal from the entire length of the steam generator is

\[
F_{o,j} = \left[ 0.323 \times 10^{-12} \frac{\mu_0^{0.2} G_1^{1.8}}{\rho_o^2} \frac{\Delta h_1^{1.8}}{\Delta h_o^{2.8}} \left( \frac{d_1}{\Delta h^{1.8}} \right)^{1.8} \left( \frac{L}{D_{eo}} \right) \right] \times \frac{\Delta h_{ij}}{\Delta h_{iE} + \Delta h_{iB} + \Delta h_{iS}}
\]
Table 1. Sample Calculation Showing the Procedure for Determining the Tube Length and Pumping Power-to-Heat Removal Ratio for Axial-Flow Bare-Tube Steam Generators

<table>
<thead>
<tr>
<th>Fluid inside tubes</th>
<th>Economizer</th>
<th>Boiler</th>
<th>Superheater</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>Saturated steam</td>
<td>Superheated steam</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Gas outside tubes</th>
<th>C02</th>
<th>C02</th>
<th>C02</th>
</tr>
</thead>
</table>

| Pressure inside tubes, psia | 2500 | 2490 | 2450 |
| Pressure outside tubes, psia | 1000 | 1000 | 1000 |

| Inside fluid temperature, °F |  |
| Inlet | 520 | 667 | 667 |
| Outlet | 667 | 667 | 1050 |

| Gas temperature, °F |  |
| Inlet | 650 | 813 | 1079 |
| Outlet | 813 | 1079 | 1350 |

| Log mean temperature difference, Δt_L, °F | 138 | 256 | 353 |

| Δh_i, Btu/lb | 218.9 | 365.7 | 398.7 |
| Δh_o, Btu/lb | 43.2 | 72.6 | 78.7 |

| p_o, lb/ft³ | 3.49 | 2.96 | 2.49 |
| c_p, Btu/lb·°F | 0.2642 | 0.2765 | 0.2903 |
| k_o, Btu/hr·ft² (°F/ft) | 0.0258 | 0.0305 | 0.0361 |
| H_o, lb/ft·hr | 0.0711 | 0.0804 | 0.0914 |
| g_i, lb/ft²·hr | 802,440 | 802,440 | 802,440 |

| c_p,i Btu/lb·°F (i.e., Δh_i/Δt_L) | 1.49 |
| k_i, Btu/hr·ft² (°F/ft) | 0.293 |
| h_i, lb/ft·hr | 0.2085 |

| c_p,i, 1.0 |  |
| k_wall, Btu/hr·ft² (°F/ft) | 12 | 12 | 12 |

| Tube outside diameter, in. | 0.5 | 0.5 | 0.5 |
| Tube inside diameter, in. | 0.4 | 0.4 | 0.4 |

| p_o, lb/ft²·hr | 0.0711 | 0.0804 | 0.0914 |
| c_p, Btu/lb·°F | 0.2642 | 0.2765 | 0.2903 |

| wall, Btu/hr·ft²·°F | 12 | 12 | 12 |

| U_i, Btu/hr·°F·ft² of internal tube surface |  |
| For D_e = 1 in. | 543 | 600 | 642 |
| = 2 in. | 272 | 300 | 321 |

| h_i, Btu/hr·ft²·°F | 2570 | 5000 | 2790 |
| Tube wall conductance, Btu/hr·ft²·°F | 2880 | 2880 | 2880 |

| U_i, Btu/hr·°F·ft² of internal tube surface |  |
| For D_e = 1 in. | 452 | 531 | 511 |
| = 2 in. | 272 | 311 | 312 |

| L_i, length, ft |  |
| For D_e = 1 in. | 23.4 | 18.0 | 14.8 | 56.2 |
| = 2 in. | 38.9 | 30.7 | 24.2 | 93.8 |

| F_c, pumping power-to-heat removal ratio |  |
| For D_e = 1 in. | 0.00383 | 0.00406 | 0.00492 | 0.01281 |
| = 2 in. | 0.000794 | 0.000865 | 0.001007 | 0.00267 |
The ratio $F_{O}$ can be expressed as a function of $d_{i}$, $L_{i}$, and $D_{eo}$ as follows, assuming $\gamma = 1.25$:

$$F_{OB} = 0.085 \times 10^{-2} \frac{L_{E_{i}}}{D_{eo}^{3}}$$

$$F_{OB} = 0.117 \times 10^{-2} \frac{L_{D_{i}}}{D_{eo}^{3}}$$

$$F_{OS} = 0.173 \times 10^{-2} \frac{L_{S_{i}}}{D_{eo}^{3}}$$

These relations have been applied to the solution of a typical problem, and the results are presented in Table 1. As may be seen, it is convenient to solve for $L_{t}$ and $F_{o}$ for two values of $D_{eo}$ (i.e., 1.0 and 2.0 in.) in the range of interest. Since $L_{t}$ and $F_{o}$ can be plotted as functions of $D_{eo}$ on logarithmic coordinates to give almost straight lines, values for $D_{eo}$ and $L_{t}$ for any desired pumping power-to-heat removal ratio can be chosen from such a set of curves.

By repeating the set of calculations for a variety of values of $d_{i}$ and $D_{eo}$, charts can be constructed from which the total length of the steam generator tubes and the equivalent passage diameter for the gas flow can be determined for any given value of the pumping power-to-heat removal ratio. This has been done, and a set of such charts is presented in Chapter 4.

The relation between $D_{eo}$, the tube diameter $d_{o}$, and the tube pitch $S$ for an equilateral triangular tube arrangement can be determined as follows:

$$D_{eo} = 4 \frac{\frac{1}{2} S^{2} \cos 30^\circ - \frac{1}{2} \frac{\pi}{4} d_{o}^{2}}{\frac{1}{2} \frac{\pi}{d_{o}}} = d_{o} \left[ 1.1 \left( \frac{S}{d_{o}} \right)^{2} - 1 \right]$$
or, solving for $S$,

$$S = 0.954 \frac{d_o}{D} \left(1 + \frac{D_{eo}}{d_o}\right)^{1/2}$$

**Axial-Flow, Bare-U-Tube Steam Generators with Reheaters**

The second steam generator considered is shown in Fig. 3. The U-tubes in the outer annular regions form the economizer, boiler, and superheater, while the straight tubes in the central region form the reheater. Although other assumptions could be made, a well-proportioned unit is obtained by assuming that the U-bend divides the boiler region into two sections in such a manner that one-half the enthalpy rise in the boiler takes place in one leg of the U and one-half in the other, as indicated in Fig. 4. The hot gas enters the steam generator at the bottom of the superheater and reheater and leaves from the bottom of the economizer. The gas inlet and outlet temperatures and pressures, the steam conditions, the flow rate of the steam, and the tube diameter for the boiler are fixed for each case. The matrix length, the tube spacing, and the diameter of the reheater tubes are then determined for a given value of the pumping power-to-heat removal ratio.

For convenience, subscripts 1, 2, and 3 are used to refer to the economizer plus the first half of the boiler, the second half of the boiler plus the superheater, and the reheater, respectively. For a given set of temperature and flow conditions, the tube length and the pumping power-to-heat removal ratio can be calculated for each of these sections for several values of $D_{eo}$ following the procedure described in the previous section for the axial-flow straight-tube steam generators, and charts can be constructed by plotting $L$ and $F$ against $D_{eo}$ for each of these sections. From such charts the tube spacing and the reheater tube diameter that would give equal matrix heights for the three sections can be determined for any given value of the pumping power-to-heat removal ratio. These calculations can be performed if the enthalpy and the temperature drop of the gas in passing through the economizer and the first
Fig. 3. Vertical Section Through U-Tube Axial-Flow Steam Generator with Reheater.
Fig. 4. Typical Temperature Distribution in a U-Bend Axial-Flow Steam Generator with Reheater.

half and the second half of the boiler, superheater, and reheater sections are known. These quantities can be evaluated by assuming that the gas temperature drops across tube matrix sections 2 and 3 are equal (i.e., \( \Delta h_{o2} = \Delta h_{o3} \)) and using the equation relating the heat absorbed by the water to the heat given up by the gas in tube matrix sections 2 and 3:

\[
W_w \alpha \Delta h_{13} + W_w (1 - \alpha) \Delta h_{12} = W_g \Delta h_{o2},
\]  

(20)

where \( \alpha \) is the weight fraction of the total steam flow that passes through the reheater tube matrix, and \( W_g \) and \( W_w \) are the total weight of the gas.
and water flowing through the matrix, respectively (i.e., steam generator and reheater).

Similarly, for tube matrix 1,

\[ W_w (1 - \alpha) \Delta h_{11} = W_g \Delta h_{01} \]  

Dividing Eq. (20) by Eq. (21) gives

\[ \frac{\Delta h_{02}}{\Delta h_{01}} = \left[ \frac{\alpha}{1 - \alpha} \right] \frac{\Delta h_{13}}{\Delta h_{11}} + \frac{\Delta h_{12}}{\Delta h_{11}} \]  

where

\[ \Delta h_{11} = \Delta h_{1E} + \frac{1}{2} \Delta h_{1B} \]

\[ \Delta h_{12} = \Delta h_{1S} + \frac{1}{2} \Delta h_{1B} \]

\[ \Delta h_{13} = \Delta h_{1R} \]

Hence, the ratio \( \Delta h_{02}/\Delta h_{01} \) is known, because all the quantities on the righthand side of Eq. (22) are known. Another relation between \( \Delta h_{02} \) and \( \Delta h_{01} \) is

\[ \Delta h_{01} + \Delta h_{02} = \bar{c}_p (t_{\text{gas, in}} - t_{\text{gas, out}}) \]  

and the quantities on the right side of this equation are also known. Therefore \( \Delta h_{01} \) and \( \Delta h_{02} \) can be calculated from the simultaneous equations (22) and (23).

The enthalpy drop of the gas in the economizer, the first and second halves of the boiler, the superheater, and the reheater sections can be related to \( \Delta h_{01} \) and \( \Delta h_{02} \), as follows:

\[ \Delta h_{0E} = \Delta h_{01} \frac{\Delta h_{1E}}{\Delta h_{1E} + \frac{1}{2} \Delta h_{1B}} \]
\[ \Delta h_{oB1} = \Delta h_{o1} - \Delta h_{oE} , \]
\[ \Delta h_{o3} = \Delta h_{o2} \frac{\Delta h_{1S}}{\Delta h_{1S} + \frac{1}{2} \Delta h_{1B}} , \]
\[ \Delta h_{oB2} = \Delta h_{o2} - \Delta h_{oS} , \]
\[ \Delta h_{o3} = \Delta h_{o2} . \]

The gas temperature drop over any of these sections, say \( j \), can be evaluated from

\[ \Delta t_{o j} = \frac{\Delta h_{o j}}{c_p} . \]

Knowing the enthalpy and temperature drop of the gas in each section, the matrix height and the pumping power-to-heat removal ratio for these sections can be calculated for the selected values of \( D_{eo} \) following the same procedure as that described previously.

It is also possible to relate the tube size and the tube spacing for tube matrix sections 2 and 3 by using an analytical equation derived from pressure-drop considerations. Such a relation is useful in matching the tube size and spacing. Assuming equal pressure drops and equal heights for tube matrix sections 2 and 3, the pressure drop relation gives

\[ \frac{G_{o3}^{1.8}}{D_{e3}^{1.2}} = \frac{G_{o2}^{1.8}}{D_{e2}^{1.2}} \]

or

\[ \frac{G_{o3}}{G_{o2}} = \left( \frac{D_{e3}}{D_{e2}} \right)^{0.666} \]
The relation between flow rates and enthalpy changes for the water and the gas is

\[ A_i G_i \Delta h_i = A_o G_o \Delta h_o \quad \text{(26)} \]

Substituting \( G_o \) from Eq. (26) into Eq. (25) with appropriate subscripts, then gives

\[ \frac{A_{13} A_{o2}}{A_{12} A_{o3}} \frac{G_{i3} \Delta h_{i3}}{G_{i2} \Delta h_{i2}} \frac{\Delta h_{i3}}{\Delta h_{i2}} = \left( \frac{D_{e3}}{D_{e2}} \right)^{0.666} \quad \text{(27)} \]

Since,

\[ A_i = \frac{\pi}{4} d_i^2 \quad , \]

\[ A_o = \frac{\pi}{4} \gamma d_i D_e \quad , \quad \text{(28)} \]

and

\[ \Delta h_{o2} = \Delta h_{o3} \quad , \]

substituting Eqs. (28) into Eq. (27) gives

\[ \frac{d_{i3} G_{i3} \Delta h_{i3}}{d_{i2} G_{i2} \Delta h_{i2}} = \left( \frac{D_{e3}}{D_{e2}} \right)^{1.666} \]

or

\[ D_{e3} = C \left( \frac{d_{i3}}{d_{i2}} \right)^{0.6} D_{e2} \quad , \quad \text{(29)} \]

where

\[ C = \left( \frac{G_{i3} \Delta h_{i3}}{G_{i2} \Delta h_{i2}} \right)^{0.6} \]
Since the quantities that define C are all known, Eq. (29) fixes the relation between the tube diameter and the equivalent flow passage diameter for tube matrix sections 2 and 3.

Cross-Flow Bare-Tube Steam Generators

The tube arrangement envisioned for a cross-flow steam generator is shown in Fig. 5. The tube matrix has a square cross section with a length \( l \) for the side and height \( L \). The tubes are arranged on an equilateral triangular pitch, with \( m \) tubes per layer and \( n \) layers per pass. The steam is generated inside the tubes by heat transfer from the hot gas flowing outside the tubes. Assuming no reheater, the matrix can be divided into three sections: the economizer, the boiler, and the superheater.

The pumping power-to-heat removal ratio and the matrix height for each of these sections can be determined in the following manner. The pressure drop for the flow outside the tube is

\[
\Delta P_o = f \frac{G^2_o}{2g \rho_o} \frac{L}{D_m} \left( \frac{\mu_w}{\mu_o} \right)^{0.14} \left( \frac{D_v}{S_T} \right)^{0.4} \left( \frac{S_L}{S_T} \right)^{0.6},
\]

where

\[
f = 0.51(10,000/Re)^{0.15},
\]

\[
Re = \frac{G o D_v}{\mu_o}.
\]

Taking \( S_T = S_L = S \) for an equilateral triangular pitch, assuming \( \mu_w = \mu_o \), and substituting Eqs. (31) and (32) into Eq. (30) gives

\[
\Delta P_o = \frac{1.015}{g} \frac{G^{1.85}_o L}{\rho_o D_v^{0.75} S^{0.4}} \mu_o^{0.15}.
\]
Fig. 5. Tube Configuration for Cross-Flow Steam Generator.
The pumping power-to-heat removal ratio is

\[ \frac{\Delta P}{F_0} = \frac{\Delta P}{773 \rho_o \Delta h_o} \]  \hspace{1cm} (34)

Eliminating \( \Delta P \) between Eqs. (33) and (34) and substituting \( g = 4.18 \times 10^4 \) ft/\( \text{hr}^2 \) and solving for \( L_m \) gives

\[ L_m = 32.0 \times 10^{10} \frac{F_0}{0.15} \frac{\rho_o^2}{\mu_o} \Delta h_o S^{0.4} \frac{D_{0.75}^V}{G_{1.85}^O} \]  \hspace{1cm} (35)

Equating the heat loss of the gas to the heat gain for the water gives

\[ G_o A_o \Delta h_o = \frac{\pi}{4} d_1^2 G_i \Delta h_i \]  \hspace{1cm} (36)

Assuming \( d_o = 1.25 d_i \), the outside flow area per tube is

\[ A_o = \frac{(S - d_o)}{n} l = \frac{(S - 1.25 d_i)}{n} l \]  \hspace{1cm} (37)

From Eqs. (36) and (37)

\[ G_o = \frac{\pi}{4} d_1^2 G_i \frac{\Delta h_i}{\Delta h_o} \left( \frac{l}{n} \right)^{-1} (S - 1.25 d_i)^{-1} \]  \hspace{1cm} (38)

The volumetric hydraulic diameter, \( D_v \), for the equilateral triangular tube spacing is

\[ D_v = 4 \frac{\text{net free volume}}{\text{friction surface}} = 4 \frac{\frac{1}{2} \left( 0.866 S^2 - \frac{\pi}{2} d_o^2 \right) L}{\left( \frac{1}{2} \pi d_o \right) L} \]

\[ = d_o \left[ 1.1 \left( \frac{S}{d_o} \right)^2 - 1 \right] = 0.882 d_i \left[ \left( \frac{S}{d_i} \right)^2 - 1.417 \right] \]  \hspace{1cm} (39)
Substituting $C_0$ and $D_v$ from Eqs. (38) and (39) into Eq. (35) and solving for $F_o$ gives the first of two simultaneous equations:

$$F_o = 2.2 \times 10^{-12} \frac{\mu_0^{1.15}g_{i_1}^{1.85}}{\rho_0^2} \frac{\Delta h_i^{1.85}}{\Delta h_o^{2.85}} \frac{d_{i_1}^{1.1}}{S_0^{0.4}} \left( \frac{1}{n} \right)^{-1.85} \times$$

$$\times \left[ \left( \frac{S}{d_i} \right)^2 - 1.417 \right]^{-0.75} \left( \frac{S}{d_i} - 1.25 \right)^{-1.85} L_m . \quad (40)$$

Assuming a perfect gas, the gas density in this equation is

$$\rho_o = \frac{P_M}{1544 \cdot T} . \quad (41)$$

The relation between the matrix height ($L_m$) and the tube length ($L_t$) is

$$L_t = \left( \text{number of bends} \right) \frac{l + L_m}{l} = \frac{L_m}{\left( \frac{\sqrt{3}}{2} \right) S_n} \cdot \quad (42)$$

Solving Eq. (42) for $L_m$,

$$L_m = L_t \left( 1 + \frac{l + \frac{2}{n} \sqrt{3} S}{\sqrt{3} S} \right)^{-1} . \quad (43)$$

The total tube length, $L_t$, can be evaluated from heat transfer considerations, as follows:

$$\pi d_{i_1} L_t U_1 \Delta T_L = \frac{\pi}{4} d_{i_1} g_{i_1} \Delta h_{i_1} . \quad (44)$$

Eliminating $L_t$ between Eqs. (43) and (44), the second of two simultaneous equations for the matrix height, $L_m$, is

$$L_m = \frac{d_{i_1} g_{i_1} \Delta h_{i_1}}{4 U_1 \Delta T_L} \left( 1 + \frac{l + \frac{2}{n} \sqrt{3} S}{\sqrt{3} S} \right)^{-1} . \quad (45)$$
In Eq. (45), the over-all coefficient of heat transfer, $U_1$, based on the internal tube surface area is

$$\frac{1}{U_1} = \frac{l}{h_i} + \frac{d_i}{8k_t} + \frac{l}{1.25h_o},$$

(46)

since $d_o = 1.25d_i$ was assumed. The heat transfer coefficients inside the tube for the economizer and the superheater sections can be taken as the same as those given for the axial-flow bare-tube steam generators.

The heat transfer coefficient for gas flow outside the tube is

$$h_o = 0.33 \frac{k_o}{d_o} \left( \frac{c_p}{k_o} \right)^{1/3} \left( \frac{d_o G_o}{u_o} \right)^{0.6}.$$

(47)

Taking $d_o = 1.25d_i$ and substituting $G_o$ from Eq. (38) into Eq. (47) gives

$$h_o = 0.261 \left( \frac{k_o}{\mu_o} \right)^{0.67} \left( \frac{G_i}{\Delta h_i} \right)^{0.33} \left[ \frac{l}{n \left( \frac{S}{d_i} - 1.25 \right)} \right]^{-0.6} \frac{d_i}{d_i^{0.2}}.$$

(48)

In Eq. (45), $A_{L_i}$ for each section can be evaluated in a manner similar to that employed for the axial-flow bare-tube steam generators. Equations (40) and (45) are coupled. The matrix height, $L_m$, may be calculated first from Eq. (45) for a given set of power plant conditions. Substituting the value of $L_m$ thus calculated into Eq. (40), the corresponding value of $P_o$ may be obtained.

In the case of cross-flow steam generators, there is an additional independent parameter, the ratio $l/n$. Particular values of $l$ and $n$ for a given $l/n$ can be determined from the power output of the steam generator by relating the number of tubes required to handle the steam flow rate to the number of tubes per pass in the cross-flow tube banks. The power output per tube is

$$\frac{\pi}{4} d_i^2 G_i (\Delta h_{iE} + \Delta h_{iB} + \Delta h_{iS}).$$
The total number of tubes, \( N \), can then be calculated for the steam generator output.

The total number of tubes can also be calculated from the geometry of the matrix, that is,

\[
N = (\text{No. of tubes per layer}) \times (\text{No. of layers per pass})
\]

\[= \frac{l}{S} n .\]

Therefore, for a given set of values of \( S \), the \( l/n \) ratio, the product \( ln \), and the total power output of the steam generator are known; hence, \( l \) and \( n \) can be determined.

**Effects of Extended Surfaces**

The heat transfer coefficient is much lower outside than inside the tube for a steam generator with bare tubes. Fins on the outer surface of the tubes increase the external heat transfer surface area and thus increase the heat flux per unit of internal surface area.

The over-all coefficient of heat transfer based on the internal area of a tube having fins on the outer surface is

\[
\frac{1}{U_i} = \frac{1}{h_i} + \frac{(\gamma - 1) d_i}{2k_t} + \frac{1}{h_{\text{eff}}} ,
\]

(49)

where

\[
h_{\text{eff}} = \frac{A_o}{A_i} \eta' h_0 ,
\]

\( a_o, a_i \) = total heat transfer area on the outside and inside of the tube, respectively,

\( \eta' \) = area-weighted fin effectiveness.

The external surface effectiveness is determined by averaging the 100% effectiveness of the prime tube surface with the less than 100% effectiveness \( \eta \) of the fin surface, as follows:
\[ \eta' = (1 - \epsilon) + \eta \epsilon, \quad (50) \]

where

- \( \epsilon \) = fin surface area per total heat transfer surface outside the tube,
- \( \eta \) = fin effectiveness.

The fin effectiveness, \( \eta \), for thin axial fins of constant thickness is

\[ \eta = \frac{\tanh Z}{Z}, \quad (51) \]

where

- \( Z = y \sqrt{2h_0/(kb)} \), for thin-sheet fins,
- \( y \) = fin height, in.,
- \( b \) = fin thickness, in.,
- \( k \) = fin metal conductivity, Btu/hr*ft\(^2\)\(^{\circ}\)F/ft,
- \( h_0 \) = heat transfer coefficient, Btu/hr*ft\(^2\)\(^{\circ}\)F.

A plot of \( \eta \) vs \( Z \) is shown in Fig. 6 for fins with constant cross section.

**Axial-Flow Steam Generators with Longitudinally Finned Tubes**

Figure 7(a) shows the sort of longitudinally finned tube envisioned for an axial-flow finned-tube steam generator. The heat transfer coefficients for flow inside the tube, \( h_i \), can be taken as equal to those given previously for the bare-tube axial-flow steam generators. Assuming that the longitudinal fins affect the heat transfer coefficient through their effect on the equivalent passage diameter, \( h_0 \) should be the same as for bare tubes. Hence, \( h_{\text{eff}} \) for the longitudinally finned tubes is

\[ h_{\text{eff}} = \frac{A_0}{A_i} \eta' h_0 = \frac{A_o}{A_i} \eta' \left( 0.023 \frac{\mu_0^{0.4} k_0^{0.6} G_0^{0.8}}{\rho_0^{0.4} \mu_0^{0.2} D_0^{0.2} \rho_0} \right). \quad (52) \]

In this equation the values of \( A_o/A_i \), \( G_0 \), and \( \eta' \) can be evaluated for the fin geometry shown in Fig. 7(a), as follows:
Fin Effectiveness for Thin-Sheet Fins of Constant Cross Section.

\[
\frac{A_o}{A_i} = \frac{(\pi d_o + \pi d_i n_i 2y) \times 1}{\pi d_i \times 1} = \gamma (1 + 2n_i y), \quad (53)
\]

and

\[
G_o = G_i \frac{\Delta h_i}{\Delta h_o} \frac{A_i}{A_o}, \quad (54)
\]
Fig. 7. Fin Configurations.
since

\[ A_1 = \frac{\pi}{4} d_1^2 , \]

\[ A_o = \frac{D_{eo} \times \text{(wetted perimeter)}}{4} , \quad (55) \]

Wetted perimeter = \( \pi d_1 \gamma (1 + 2n_f y) \).

Substituting Eqs. (55) into Eq. (54) gives

\[ G_o = G_i \frac{\Delta h_i}{\Delta h_o} \frac{d_i}{\gamma (1 + 2n_f y)D_{eo}} . \quad (56) \]

The area-weighted fin effectiveness is

\[ \eta' = (1 - \epsilon) + \eta \epsilon , \quad (57) \]

where

\[ \epsilon = \frac{n_f (b + 2y)}{1 + 2n_f y} . \quad (58) \]

Assuming an equilateral triangular tube arrangement, the tube spacing, \( S \), is related to the equivalent passage diameter \( D_{eo} \), as follows:

\[ S = 0.951 \gamma d_1 \left[ 1 + \frac{D_{eo}}{\gamma d_1} (1 + 2n_f y) + \frac{4n_f y b}{\gamma d_1} \right]^{1/2} . \quad (59) \]

The friction pressure drop for gas flow over longitudinally finned tubes can be calculated from

\[ \Delta P_o = \frac{G_o^2 L_t}{2g \rho_o D_{eo}} = \frac{0.21}{2g} \frac{G_i^{0.8} L_t^1}{\rho_o D_{eo}^{1.2}} . \quad (60) \]
The procedure followed in evaluating the tube length and the pumping power-to-heat removal ratio for steam generators with longitudinally finned surfaces was similar to that used for bare tubes.

**Cross-Flow Steam Generators with Circularly Finned Tubes**

Circular fins of constant cross section of the type similar to that shown in Fig. 7(b) were considered suitable for the cross-flow steam generators. The heat transfer coefficients for flow inside the tube should be the same as those for the bare-tube axial-flow steam generators. The heat transfer coefficient for flow over the cross-flow tube matrix with circular fins may be correlated in the following form:

\[
(St)(Pr)^{2/3} = C_h (Re)^{-n} ,
\]

(61)

or solving for \(h_o\),

\[
h_o = C_h \frac{k_o}{D_{eo}} (Re)^{1-n} (Pr)^{1/3} .
\]

(62)

In this equation the constant \(C_h\) and the exponent \(n\) depend on the tube surface geometry, tube arrangement, and the flow conditions. These constants were obtained from data presented by Kays and London. The equation

\[
h_o = 0.20 \frac{k_o}{D_{eo}} (Re)^{0.60} (Pr)^{1/3}
\]

(63)

was used to determine the heat transfer coefficient for flow over the cross-flow matrix with circular fins in the range of interest and in the Reynolds number range from 400 to 15,000. In Eq. (62), the mass flow, \(G\), in the Reynolds number and the Stanton number, and the equivalent passage diameter are based on the minimum free-flow area through the tube matrix regardless of where the minimum occurs, and the equivalent passage diameter is defined as
Expanding the terms in Eq. (63) gives

\[ h_o = 0.20 \frac{C^0.333_{p_o}^{0.666}}{C_p^{0.266} \mu_o^{0.60}} \frac{l}{D^0.40_{e0}} , \]  

(65)

and \( h_{eff} \) for flow over the cross-flow tube matrix with circular fins is

\[ h_{eff} = \frac{A_0}{A_1} \eta' h_o . \]  

(66)

In Eq. (66) the values of \( A_o/A_1 \), \( G_o \), and \( \eta' \) can be evaluated for the fin geometry shown in Fig. 7(c), as follows:

\[
\frac{A_o}{A_1} = \frac{\frac{\pi}{4} \left[ (d_o + 2y)^2 - d^2 \right] 2 + \pi d_o \left( \frac{1}{n_f} - b \right) + \pi (d_o + 2y) b}{\pi d_i \frac{1}{n_f}},
\]

(67)

\[
= \frac{2y (d_o + y) n_f^2 + d_o + 2y b n_f}{d_i^2},
\]

(68)

where

\[
G_o = G_i \frac{\Delta h_i}{A_1} \frac{A_1}{A_o},
\]

(69)

or

\[
A_o = \frac{\left[ (S - d_o) - 2y b n_f \right] \frac{l}{n}}{n}
\]
and

\[ A_i = \frac{\pi}{4} d_i^2. \]

Substituting Eq. (69) into Eq. (68) gives

\[ G_o = G_i \frac{\frac{\pi}{4} d_i^2}{\Delta h_i \left( \frac{\pi}{4} d_i^2 \right) \left( S - d_o \right) - 2ybn_f} \frac{1}{n_f}. \]  

(70)

The area-weighted fin effectiveness is

\[ \eta' = (1 - \epsilon) + \eta \epsilon, \]  

(71)

where

\[ \epsilon = \frac{\frac{\pi}{4} \left[ \left( d_o + 2y \right)^2 - d_o^2 \right]}{\pi \left[ \left( d_o + 2y \right)^2 - d_o^2 \right] 2 + \pi \left( d_o + 2y \right) b + \pi d_o \left( \frac{1}{n_f} - b \right)} \]

\[ = \frac{2ybn_f \left( d_o + y \right) + bn_f \left( d_o + 2y \right)}{2ybn_f \left( d_o + y \right) + 2ybn_f + d_o}, \]  

(72)

and the fin effectiveness, \( \eta \), can be evaluated as explained previously.

Assuming an equilateral triangular tube arrangement, the tube spacing, \( S \), is related to the equivalent passage diameter, as follows:

\[ D_{eo} = 4 \frac{\left( \frac{\sqrt{3}}{2} S \right) \left[ \left( S - d_o \right) \frac{1}{n_f} - 4y \frac{b}{2} \right]}{\frac{\pi}{4} \left[ \left( d_o + 2y \right)^2 - d_o^2 \right] 2 + \pi d_o \left( \frac{1}{n_f} - b \right) + \pi \left( d_o + 2y \right) b} \]

or

\[ D_{eo} = 1.1 \times S \frac{(S - d_o) - 2ybn_f}{2y (d_o + y) n_f + d_o + 2ybn_f}. \]  

(73)
The gas pressure drop for the cross-flow matrix is given by

\[ \Delta P = 4f \frac{G^2_o L_m}{2g_0 \rho_o D_{eo}} \quad . \]  

(74)

The friction data corresponding to the heat transfer data chosen were correlated by

\[ f = 0.26(Re)^{-0.25} \quad . \]  

(75)

Substituting Eq. (75) into Eq. (74) and expanding the Reynolds term gives

\[ \Delta P = 1.04 \left( \frac{G_o D_{eo}}{\mu_o} \right)^{-0.25} \frac{G^2_o L_m}{2g_0 \rho_o D_{eo}} \]

or

\[ \Delta P = \frac{0.52 \mu_o^{0.25}}{g \rho_o^{1.75}} G^{1.75} \frac{L_m}{D_{eo}^{1.25}} \quad . \]  

(76)

The procedure followed in evaluating the matrix height, \( L_m \), and the pumping power-to-heat removal ratio for the cross-flow steam generators with circular fins was similar to that used for cross flow with bare tubes.
2. PROCEDURES FOR MACHINE CALCULATIONS

In making preparations for an extensive set of calculations on an IBM 7090 computer it became evident that better results could be obtained if the changes in the physical properties of the steam and gas along the tube were included in evaluating the local heat transfer coefficients. This required division of each section into small parts, and hand calculations would have become extremely complex and tedious. The problem could be handled readily on the machine, however, and a code was prepared to solve the simultaneous differential equations for heat transfer along the length of the tube, with the appropriate boundary conditions and the local heat transfer coefficients included.

**Code for Axial-Flow Steam Generators**

The differential equations used for the economizer and the superheater sections were

\[
\frac{d}{dx} \left( A_0 G_0 c_p \frac{dt_0}{dx} \right) = \pi d_1 U_1 \left( t_0 - t_1 \right) \quad (78)
\]

and

\[
\frac{d}{dx} \left( A_1 G_1 c_p \frac{dt_1}{dx} \right) = \pi d_1 U_1 \left( t_0 - t_1 \right) . \quad (79)
\]

In the boiler section the temperature of the water-steam mixture remains constant, but the steam quality changes along the tube. The differential equations used for the boiler section were

\[
\frac{d}{dx} \left( A_0 G_0 c_p \frac{dt_0}{dx} \right) = \pi d_1 U_1 \left( t_0 - t_1 \right) \quad (80)
\]
and

\[ A_i G_i \left( \frac{d\xi}{dx} \right) = \pi d_i U_i \left( t_o - t_i \right), \tag{81} \]

where

\[ \xi = \text{quality of steam}, \]
\[ q = \text{latent heat of vaporization}. \]

A gross heat balance for the entire steam generator (economizer, boiler, and superheater sections) gives

\[ A_o G_o \Delta h_o = A_i G_i \Delta h_i, \tag{82} \]

where

\[ \Delta h_i \equiv \text{(enthalpy of superheated steam leaving)} - \text{(enthalpy of feedwater entering)}, \]
\[ \Delta h_o \equiv \int_{t_{\text{gas in}}}^{t_{\text{gas out}}} c p_o \, dt_o, \]
\[ A_i = \frac{\pi}{4} d_i^2, \]
\[ U_i = \text{local over-all heat transfer coefficient based on internal tube surface}. \]

Substituting the above value of \( A_i \) in Eq. (82) gives

\[ A_o G_o = \frac{\pi}{4} d_i^2 \frac{\Delta h_i}{\Delta h_o}. \tag{83} \]

Substituting Eq. (83) into Eqs. (78) and (79), the differential equations for the economizer and superheater sections become

\[ \frac{dt_o}{dx} = \frac{4 \Delta h_o U_i}{d_i G_i \Delta h_i c p_o} \left( t_o - t_i \right). \tag{84} \]
and

\[ \frac{dt_i}{dx} = \frac{4}{d_i G_i c_{pi}} U_i (t_o - t_i) \quad (85) \]

Similarly, Eqs. (80) and (81) for the boiler become

\[ \frac{dt_0}{dx} = \frac{4 \Delta h_o U_i}{d_i G_i \Delta h_i c_{po}} (t_o - t_i) \quad (86) \]

and

\[ \frac{dt_i}{dx} = \frac{4U_i}{d_i G_i q} (t_o - t_i) \quad (87) \]

The physical properties of the gas and the water were fed into the machine as functions of temperature so that their effects were included in solving the above equations at each point along the tube. Since the pressure drop inside the tube should be small compared with the steam pressure, the properties of the steam were evaluated at the average steam system pressure.

Once the pressures, the fluid inlet and outlet temperatures, the tube diameter, the tube pitch, and the mass flow rate of the steam (i.e., \(G_i\)) were fixed, the machine could evaluate the tube lengths in the economizer, boiler, and superheater sections by solving the above coupled differential equations. For the economizer section, these equations were solved until the temperature of the feedwater reached the saturation temperature. The integration was continued for the boiler section until the steam quality reached 100%. Similarly, the process was continued through the superheater until the desired superheater outlet temperature was reached. The machine then calculated the pressure drop and the pumping power-to-heat removal ratio for the gas side. The water-side pressure drop in the economizer was not computed because it would be small, and two-phase flow in the boiler makes it difficult to compute the pressure drop there. Thus this item was not included in the original code, but
later work led to the machine code presented in Chapter 9. The steam pressure drop in the superheater can be computed easily, and this was done to give an indication of the total pressure drop across the boiler tubes.

Much previous work with gas-cooled reactor systems has shown that such important relations as those between capital charges and operating costs are heavily dependent on the pumping power-to-heat removal ratio for the gas system. For meaningful comparisons of the type contemplated in this study, it has been found best to carry out parametric calculations for each of several fixed values of the pumping power-to-heat removal ratio. Because of this, although the equations used permit an explicit evaluation of the tube length and the corresponding pumping power-to-heat removal ratio for a given tube spacing in perhaps 10 sec of machine time, it was decided to fix the pumping power-to-heat removal ratio and code the machine to make a series of iterations to establish the tube spacing. This iterative operation required about 2 min of machine time.

It was assumed that the hot gas from the reactor entered the steam generator at the bottom as shown in Fig. 1 and rose vertically to the top through a central duct before entering the tube matrix. For a given power output of the steam generator the diameter of this duct was determined by fixing the allowable pressure drop through the duct. Hence, the factor \( X \), defined as the ratio of the pressure drop of the gas through the duct to the pressure drop of the gas across the tube matrix was used for determining the duct diameter. The friction pressure drop for the duct was calculated from:

\[
\Delta P_d = \frac{0.21}{2g} \frac{L_t}{\frac{d^1.8}{D_d \frac{d^D}{d}}}.
\]

The total number of tubes, \( N \), was calculated from the power level of the steam generator; that is, \( N \) was taken to be the ratio of the total thermal power of the steam generator to the thermal power per tube. The pressure vessel diameter, \( D_p \), was then calculated for an equilateral triangular tube spacing with a central duct of diameter \( D_d \) from the following
relation:

\[ \frac{D_p}{d} = (D_d^2 + 1.1NS^2)^{1/2} \]

It is to be noted that this relation does not include any allowances for internal thermal insulation or for the coolant annulus shown in the horizontal cross section of Fig. 8.

In preparing the code, it was assumed that the tubes would have an outside diameter equal to 1.25 times the inside diameter and would be

Fig. 8. Horizontal Cross Section Used as the Basis for the Axial-Flow Steam Generator Parametric Calculations (No Reheater).
arranged in an equilateral triangular array. In the case of finned tubes, longitudinal rectangular steel fins were assumed for the axial-flow matrices, while transverse circular steel fins were assumed for the cross-flow units. Helium and CO₂ were the two gases considered in the code. The program could be used for any other gas if the physical properties of the gas as a function of the temperature were introduced into the code. The heat transfer coefficients could be altered as a subroutine.

The cost estimates (see chap. 3) were based on the use of tube bundles similar to that of Fig. 9. The unit costs could be altered as a subroutine. It should be mentioned that once the dimensions of the tube matrix were determined, the cost calculations could be performed in a matter of seconds.

The initial program was for a steam generator with economizer, boiler, and superheater sections. This was readily extended to perform calculations for reheaters, since, from the machine code standpoint, a reheater is essentially equivalent to a superheater. For a reheater calculation the economizer and boiler portions of the code are taken out of the program, and the machine performs only the superheater calculations.

Code for Cross-Flow Steam Generators

The code for the cross-flow steam generators was essentially similar to that for the axial-flow units. The differential equations for
determining the tube length and the heat transfer coefficients for the flow of fluid inside the tubes were the same, for example. Since the cross-flow arrangement entails a large number of passes, no attempt was made to include a correction for crossflow; as in the axial-flow case, the temperature difference calculated was simply that for pure counterflow. The heat transfer coefficients used for the gas and steam sides were those described in the previous sections for the cross-flow arrangement with bare or finned tubes. The relation between the tube length, $L_t$, and the matrix height, $L_m$, assuming an equilateral triangular tube arrangement, was taken as

$$L_m = L_t \left(1 + \frac{l}{n} \frac{2}{\sqrt{3} S}\right)^{-1}$$

Once the pressures, the fluid inlet and outlet temperatures, the mass flow rate of steam (i.e., $G_i$), the tube diameter, the tube pitch, the thermal power output of the steam generator, and the $l/n$ ratio were fixed, the machine could calculate the tube length, the matrix height and width, and the number of tube layers per pass. The pressure drop and the pumping power-to-heat removal ratio were then calculated for the gas flow over the tube matrix, and the steam pressure drop was calculated for the superheater section.

As in the calculations for the axial-flow units, the pumping power-to-heat removal ratio for the gas flow over the tube matrix was fixed, and the appropriate tube pitch was determined. By varying the $l/n$ ratio for a given pumping power-to-heat removal ratio and a given power output, the cross-flow matrix could be made tall and slender or short and large in diameter.

A horizontal cross section of the cross-flow matrix is shown in Fig. 10. The hot gas from the reactor enters the steam generator at the bottom and flows upward through the four ducts arranged around the perimeter between the tube matrix and the pressure vessel. In establishing the outside diameter of the pressure vessel, a 2-in. annular space was allowed between the duct and the vessel wall for shell cooling purposes,
and an additional radial gap of 3 in. was provided between the inner edge of this cooling duct and the corner of the tube matrix. It was assumed that after allowing for the support structure, the net cross-sectional area for the ducts would be 75% of the area between the tube matrix and the pressure vessel cooling annulus. Calculations showed that the resulting duct area was large enough to give a small pressure drop in the gas flowing through these ducts for the range of parameters covered in this study. The pressure drop in the gas stream flowing through the ducts was related to the pressure drop through the cross-flow matrix by
the factor, \( \lambda \), defined as the ratio of the gas pressure drop through the duct to the gas pressure drop through the cross-flow matrix.

The pressure drop through the duct was calculated from the relation

\[
\Delta P_d = \frac{0.21}{2g} \frac{q_1^{1.8}}{\rho} \frac{\mu^{0.2}}{D_1^{1.2}} \frac{L_m}{D_e^{1.2}},
\]

and the equivalent diameter was given, approximately, by

\[
D_e \approx \frac{4A_d}{2(x + y)},
\]

where

\[
y = 0.207l - 0.375 \quad \text{(see Fig. 10)},
\]

\[
A_d = 0.25 \times 0.75 \left[ \pi (0.707l + 0.25)^2 - (l + 1.25)^2 \right],
\]

\[
x = A_d/y.
\]

Calculations showed that, for all the cross-flow configurations studied, \( \lambda < 0.1 \). Since a parametric investigation of axial-flow units showed that the major figures of merit are insensitive to \( \lambda \) for values of \( \lambda \) less than 0.1, no iterations were made to adjust \( \lambda \) to give any specified value.

If the width, \( 2l \), of the cross-flow matrix is determined for a given set of conditions, it is apparent from Fig. 10 that the diameter of the pressure vessel becomes

\[
D_p = 1.412l + 0.832.
\]

In the analysis a square horizontal cross section was assumed for the cross-flow matrices. It is interesting to note that the program could also be used for cross-flow matrices with rectangular cross sections. In that case the actual power of the steam generator should be multiplied in the input sheet by a factor, say \( r \), and the resulting steam generator should be divided vertically into \( r \) equal parts. Any one of these parts would have the same matrix height and pumping power-to-heat removal ratio.
as the original large unit, but its power would be $1/r$ times that of the larger unit.

The program was subsequently extended to include recirculating boilers. For a recirculating type of boiler it was assumed that only a certain fraction of the circulating water would be evaporated in the boiler section. The remaining hot water would be separated from the steam in the steam separator drum and mixed with the incoming feedwater entering the economizer section for recirculation. The superheater was assumed to be a separate unit, the dimensions of which could be calculated in the usual manner.

**Detailed Data on the Computer Codes**

The codes used have been identified by the word GCSG (i.e., gas-cooled steam generators) and include routines for the following types of steam generator:
1. axial-flow steam generators,
2. axial-flow reheaters,
3. cross-flow steam generators,
4. cross-flow reheaters, and
5. cross-flow recirculating boilers.

Calculations for any one of these groups can be carried out for bare or finned tubes with either a fixed tube pitch or a fixed pumping power-to-heat removal ratio on the gas side.

The computer input data for the axial- and cross-flow steam generators and reheaters included the following information with the units specified:

- **Water temperature**
  - Inlet $^\circ F$
  - Outlet $^\circ F$

- **Gas temperature**
  - Inlet $^\circ F$
  - Outlet $^\circ F$

- **System pressure**
  - Inside tubes psia
  - Outside tubes psia
Mass flow rate of water inside tubes
Molecular weight of gas
He
CO₂
Tube inside diameter
Thermal conductivity of tube wall
Electric output of power plant*
Cycle efficiency*
λ, for axial flow only
l/n, for cross flow only
Tube pitch
Pumping power-to-heat removal ratio for gas flow over the tube matrix

In the case of finned tubes, the following additional information was needed:
Fin height
Fin thickness
Thermal conductivity of fin material
Number of fins per inch of perimeter or of tube length

In the case of recirculating boilers, the recirculating ratio, defined as the fraction of the total water flowing in the tube that is recirculated, was required:

Recirculating ratio

Sample output and input data sheets, showing typical calculations performed for the axial- and cross-flow steam generators are presented in Tables 2 through 7.

*Power output of steam generator equal to electrical output of power plant divided by cycle efficiency. The absolute values of power plant output and the cycle efficiency are unimportant for the input sheet, so long as the resulting power output for the steam generator gives the required value; that is, a 1250-Mw(e) steam generator could be written on the input sheet as a 500-Mw(e) power plant with a cycle efficiency of 0.40.
Table 2. Sample Computer Input Data for Axial-Flow Bare-Tube Steam Generators

<table>
<thead>
<tr>
<th>AXIAL-FLOW STEAM GENERATOR</th>
<th>AXIAL-FLOW REHEATER</th>
<th>CROSS-FLOW STEAM GENERATOR</th>
<th>CROSS-FLOW REHEATER</th>
<th>CROSS-FLOW RECIRCULATING BOILER</th>
<th>AXIAL-FLOW RECIRCULATING BOILER</th>
</tr>
</thead>
<tbody>
<tr>
<td>INSIDE DIAMETER (in.)</td>
<td>NUMBER of W/Q</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>0.40</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

*This card must appear only once no matter how many cases are run, since it controls whether there will be iteration for the outside hydraulic diameter. If iteration is desired, any non-zero integer can be placed in column two. The second field will then contain the fractional percentage of accuracy desired in W/Q. If no iteration is desired, (if W/Q is to be calculated), this card is left blank.

<table>
<thead>
<tr>
<th>PRESSURE INSIDE TUBES (psia)</th>
<th>PRESSURE OUTSIDE TUBES (psia)</th>
<th>TEMPERATURE OF WATER INTO TUBES (°F)</th>
<th>TEMPERATURE OF GAS INTO TUBES (°F)</th>
<th>TEMPERATURE OF STEAM FROM TUBES (°F)</th>
<th>TEMPERATURE OF GAS OUT OF TUBES (°F)</th>
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</thead>
<tbody>
<tr>
<td>1</td>
<td>2.500</td>
<td>1.000</td>
<td>5.200</td>
<td>13.500</td>
<td>1.050</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>FIN THICKNESS (in.)</th>
<th>FIN HEIGHT (in.)</th>
<th>NUMBER OF FINS PER INCH</th>
<th>THERMAL CONDUCTIVITY OF FIN (Btu/hr·ft·°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>10</td>
<td>20</td>
<td>30</td>
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</table>

<table>
<thead>
<tr>
<th>RATIO OF OUTSIDE TO INSIDE DIAMETER OF TUBE</th>
<th>MOLECULAR WEIGHT OF GAS</th>
<th>THERMAL EFFICIENCY OF THE CYCLE</th>
<th>NET ELECTRIC OUTPUT OF POWER PLANT (MW)</th>
<th>MASS FLOW RATE INSIDE TUBES (lb/ft²·sec)</th>
<th>THERMAL CONDUCTIVITY OF TUBE METAL (Btu/hr·ft·°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2.5</td>
<td>4.4</td>
<td>0.4</td>
<td>5.0</td>
<td>2.2</td>
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</table>

<table>
<thead>
<tr>
<th>OUTSIDE HYDRAULIC DIAMETER (in)</th>
<th>W/Q (PUMPING POWER-TO-HEAT REMOVAL RATIO, %)</th>
<th>TUBE PITCH, in. (FOR CROSS FLOW)</th>
<th>OR HYDRAULIC DIAMETER, in. (FOR AXIAL FLOW)</th>
<th>NUMBER OF LAMBDAS OR (1/n)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.9</td>
<td>24</td>
<td>32</td>
<td>34</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>L/n, ft (FOR CROSS FLOW)</th>
<th>OR Lambdas, DIMENSIONLESS (FOR AXIAL FLOW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>8</td>
</tr>
</tbody>
</table>
Table 3. Sample Computer Output Data for Axial-Flow Bare-Tube Steam Generators

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tube Inside Diameter</td>
<td>0.4011 in</td>
</tr>
<tr>
<td>Hydraulic Diameter Outside the Tubes</td>
<td>2.111 in</td>
</tr>
<tr>
<td>Gas Outlet Temperature from Superheater</td>
<td>1072.1°F</td>
</tr>
<tr>
<td>Gas Outlet Temperature from Boiler</td>
<td>812.86°F</td>
</tr>
<tr>
<td>Steam Temperature in the Boiler</td>
<td>668.13°F</td>
</tr>
<tr>
<td>Length of Economizer</td>
<td>38.00 ft</td>
</tr>
<tr>
<td>Length of Boiler</td>
<td>28.14 ft</td>
</tr>
<tr>
<td>Length of Superheater</td>
<td>20.82 ft</td>
</tr>
<tr>
<td>Total Length</td>
<td>86.97 ft</td>
</tr>
<tr>
<td>Pumping Power Outside Economizer/Total Heat Removed</td>
<td>0.00063</td>
</tr>
<tr>
<td>Pumping Power Outside Boiler/Total Heat Removed</td>
<td>0.00066</td>
</tr>
<tr>
<td>Pumping Power Outside Superheater/Total Heat Removed</td>
<td>0.00071</td>
</tr>
<tr>
<td>Pumping Power Inside Superheater/Total Heat Removed</td>
<td>0.00066</td>
</tr>
<tr>
<td>Total Pumping Power/Total Heat Removed</td>
<td>0.00200</td>
</tr>
<tr>
<td>Pressure Drop Outside Economizer</td>
<td>2.31127 PSIA</td>
</tr>
<tr>
<td>Pressure Drop Outside Boiler</td>
<td>2.06598 PSIA</td>
</tr>
<tr>
<td>Pressure Drop Outside Superheater</td>
<td>1.87091 PSIA</td>
</tr>
<tr>
<td>Pressure Drop Inside Superheater</td>
<td>13.67915 PSIA</td>
</tr>
<tr>
<td>Total Pressure Drop Outside Tubes</td>
<td>6.24816 PSIA</td>
</tr>
<tr>
<td>Mass Flow Rate of Gas Outside the Tubes</td>
<td>6.0362729E+06 LBS/HR.FT²</td>
</tr>
<tr>
<td>Mean Heat Transfer Per Unit External Surface Economizer</td>
<td>0.38651941E-05 BTU/HR.FT²</td>
</tr>
<tr>
<td>Mean Heat Transfer Per Unit External Surface Boiler</td>
<td>0.8566079E-05 BTU/HR.FT²</td>
</tr>
<tr>
<td>Mean Temperature Difference in Economizer</td>
<td>131.60°F</td>
</tr>
<tr>
<td>Mean Temperature Difference in Boiler</td>
<td>265.37°F</td>
</tr>
<tr>
<td>Mean Temperature Difference in Superheater</td>
<td>408.30°F</td>
</tr>
<tr>
<td>Heat Transfer Coefficient Outside Economizer</td>
<td>0.36835951E-03 BTU/HR.FT².F</td>
</tr>
<tr>
<td>Heat Transfer Coefficient Outside Boiler</td>
<td>0.40709216E-03 BTU/HR.FT².F</td>
</tr>
<tr>
<td>Heat Transfer Coefficient Outside Superheater</td>
<td>0.45324896E-03 BTU/HR.FT².F</td>
</tr>
<tr>
<td>Heat Transfer Coefficient Inside Economizer</td>
<td>0.25211901E-04 BTU/HR.FT².F</td>
</tr>
<tr>
<td>Heat Transfer Coefficient Inside Boiler</td>
<td>0.50000000E-04 BTU/HR.FT².F</td>
</tr>
<tr>
<td>Heat Transfer Coefficient Inside Superheater</td>
<td>0.13306741E-04 BTU/HR.FT².F</td>
</tr>
<tr>
<td>Tube Metal Heat Transfer Coefficient</td>
<td>0.30000000E-04 BTU/HR.FT².F</td>
</tr>
<tr>
<td>Overall Coefficient of Heat Transfer for Economizer</td>
<td>0.29602930E-04 BTU/HR.FT².F</td>
</tr>
<tr>
<td>Overall Coefficient of Heat Transfer for Boiler</td>
<td>0.34210104E-04 BTU/HR.FT².F</td>
</tr>
<tr>
<td>Overall Coefficient of Heat Transfer for Superheater</td>
<td>0.31012960E-04 BTU/HR.FT².F</td>
</tr>
<tr>
<td>Total Number of Tubes</td>
<td>6212.75</td>
</tr>
<tr>
<td>Pitch for Tubes</td>
<td>1.0860 in</td>
</tr>
<tr>
<td>Lambda (Pressure Drop in Duct/Total Pressure Drop)</td>
<td>0.010</td>
</tr>
<tr>
<td>Diameter of Duct</td>
<td>7.552 FT</td>
</tr>
<tr>
<td>Diameter of Shell</td>
<td>10.630 FT</td>
</tr>
<tr>
<td>Total Weight of Bare Tubes (Excluding Fins)</td>
<td>132613.654 LBS</td>
</tr>
<tr>
<td>Total Weight of Shell</td>
<td>541402.797 LBS</td>
</tr>
<tr>
<td>Pressure Vessel Thickness</td>
<td>3.986 IN</td>
</tr>
<tr>
<td>Total Height of Shell</td>
<td>97.599 FT</td>
</tr>
<tr>
<td>Cost of Tubes</td>
<td>1245945</td>
</tr>
<tr>
<td>Cost of Shell</td>
<td>1047461</td>
</tr>
<tr>
<td>Cost of Insulation</td>
<td>47540</td>
</tr>
<tr>
<td>Total Cost of Tube and Shell</td>
<td>2340947</td>
</tr>
<tr>
<td>STRAIGHT BARE TUBE STEAM GENERATOR COUNTERFLOW</td>
<td>CASE 1.</td>
</tr>
<tr>
<td>-----------------------------------------------</td>
<td>--------</td>
</tr>
<tr>
<td>GAS PRESSURE OUTSIDE</td>
<td>1000.00PSIA</td>
</tr>
<tr>
<td>STEAM PRESSURE INSIDE</td>
<td>2500.00PSIA</td>
</tr>
<tr>
<td>GAS INLET TO SUPERHEATER TEMPERATURE</td>
<td>1350.00F</td>
</tr>
<tr>
<td>GAS OUTLET TEMPERATURE FROM ECONOMIZER</td>
<td>650.00F</td>
</tr>
<tr>
<td>FEED WATER INLET TEMPERATURE</td>
<td>520.00F</td>
</tr>
<tr>
<td>GAS OUTLET TEMPERATURE FROM SUPERHEATER</td>
<td>1050.00F</td>
</tr>
<tr>
<td>GAS OUTSIDE THE TUBES</td>
<td>CO2</td>
</tr>
<tr>
<td>MASS FLOW RATE OF FLUID INSIDE THE TUBES</td>
<td>802439.99LBS/FT2.HR</td>
</tr>
<tr>
<td>MASS FLOW RATE OF FLUID INSIDE THE TUBES</td>
<td>222.9OLBS/FT2.SEC</td>
</tr>
<tr>
<td>TUBE METAL CONDUCTIVITY</td>
<td>12.5CBTU/HR.FT.F</td>
</tr>
<tr>
<td>RATIO OF TUBE OUTSIDE DIAMETER TO INSIDE DIAMETER</td>
<td>1.25</td>
</tr>
<tr>
<td>POWER PLANT OUTPUT</td>
<td>500000.00KWE</td>
</tr>
<tr>
<td>THERMAL EFFICIENCY</td>
<td>0.4C</td>
</tr>
<tr>
<td>REACTOR GAS TEMPERATURE RISE</td>
<td>700.00F</td>
</tr>
<tr>
<td>REACTOR GAS OUTLET TEMPERATURE</td>
<td>1350.00F</td>
</tr>
<tr>
<td>ENTHALPY CHANGE OF GAS OUTSIDE ECONOMIZER</td>
<td>44.09BTU/LB</td>
</tr>
<tr>
<td>ENTHALPY CHANGE OF GAS OUTSIDE BOILER</td>
<td>72.75BTU/LB</td>
</tr>
<tr>
<td>ENTHALPY CHANGE OF GAS OUTSIDE SUPERHEATER</td>
<td>80.52BTU/LB</td>
</tr>
<tr>
<td>ENTHALPY CHANGE OF WATER INSIDE ECONOMIZER</td>
<td>219.89BTU/LB</td>
</tr>
<tr>
<td>ENTHALPY CHANGE OF FLUID INSIDE BOILER</td>
<td>360.53BTU/LB</td>
</tr>
<tr>
<td>ENTHALPY CHANGE OF STEAM INSIDE SUPERHEATER</td>
<td>400.21BTU/LB</td>
</tr>
<tr>
<td>TOTAL HEAT TRANSFERED PER TUBE</td>
<td>0.68669274E 06BTU/HR.PER TUBE</td>
</tr>
<tr>
<td>TOTAL HEAT TRANSFER PER INSIDE SURFACE AREA</td>
<td>0.75399650E 05BTU/HR.FT2</td>
</tr>
</tbody>
</table>
Table 4. Sample Computer Input Data for Cross-Flow Bare-Tube Steam Generators

<table>
<thead>
<tr>
<th>AXIAL-FLOW STEAM GENERATOR</th>
<th>AXIAL-FLOW REHEATER</th>
<th>CROSS-FLOW STEAM GENERATOR</th>
<th>CROSS-FLOW REHEATER</th>
<th>CROSS-FLOW RECIRCULATING BOILER</th>
<th>AXIAL-FLOW RECIRCULATING BOILER</th>
</tr>
</thead>
<tbody>
<tr>
<td>INSIDE DIAMETER (in.)</td>
<td>NUMBER of W/Q</td>
<td>INSIDE DIAMETER (in.)</td>
<td>NUMBER of W/Q</td>
<td>INSIDE DIAMETER (in.)</td>
<td>NUMBER of W/Q</td>
</tr>
<tr>
<td>1.</td>
<td>1</td>
<td>2</td>
<td>10</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

*This card must appear only once no matter how many cases are run, since it controls whether there will be iteration for the outside hydraulic diameter. If iteration is desired, any non-zero integer can be placed in column two. The second field will then contain the fractional percentage of accuracy desired in W/Q. If no iteration is desired, (if W/Q is to be calculated), this card is left blank.

<table>
<thead>
<tr>
<th>PRESSURE INSIDE TUBES (psia)</th>
<th>PRESSURE OUTSIDE TUBES (psia)</th>
<th>TEMPERATURE OF WATER INTO TUBES (°F)</th>
<th>TEMPERATURE OF GAS INTO TUBES (°F)</th>
<th>TEMPERATURE OF STEAM FROM TUBES (°F)</th>
<th>TEMPERATURE OF GAS OUT OF TUBES (°F)</th>
<th>RECIRCULATING RATIO (RECIRCULATING BOILER ONLY) OF WATER RECIRCULATED (lb) TO STEAM GENERATED (lb)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2, 5, 0, 0</td>
<td>5, 0, 0</td>
<td>5, 2, 0</td>
<td>1, 3, 5, 0</td>
<td>1, 0, 5, 0</td>
<td>6, 5, 0</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>FIN THICKNESS (in.)</th>
<th>FIN HEIGHT (in.)</th>
<th>NUMBER OF FINS PER INCH</th>
<th>THERMAL CONDUCTIVITY OF FIN (Btu/hr-ft-°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

*If the tube is bare this card must be left blank.

<table>
<thead>
<tr>
<th>RATIO OF OUTSIDE TO INSIDE DIAMETER OF TUBE</th>
<th>MOLECULAR WEIGHT OF GAS</th>
<th>THERMAL EFFICIENCY OF THE CYCLE</th>
<th>NET ELECTRIC OUTPUT OF THE POWER PLANT (MW)</th>
<th>MASS FLOW RATE INSIDE TUBES (lb/ft²-sec)</th>
<th>THERMAL CONDUCTIVITY OF TUBE METAL (Btu/hr-ft-°F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2</td>
<td>4</td>
<td>0</td>
<td>4</td>
<td>0</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>OUTSIDE HYDRAULIC DIAMETER (in)</th>
<th>W/Q (PUMPING POWER-TO-HEAT REMOVAL RATIO, %)</th>
<th>TUBE PITCH, in. (FOR CROSS FLOW)</th>
<th>HYDRAULIC DIAMETER, in. (FOR AXIAL FLOW)</th>
<th>NUMBER OF LAMBDAS OR (1/n)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0. 5</td>
<td>3</td>
<td>1</td>
<td>3</td>
</tr>
</tbody>
</table>

1/n, ft (FOR CROSS FLOW)

<table>
<thead>
<tr>
<th>LAMBDAS, DIMENSIONLESS (FOR AXIAL FLOW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
</tr>
</tbody>
</table>
### Table 5. Sample Computer Output Data for Cross-Flow Bare-Tube Steam Generators

<table>
<thead>
<tr>
<th>Case #</th>
<th>CROSSFLCW BARE TUBE STEAM GENERATOR</th>
<th>CAE</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>12CC Tube Inside Diameter (in)</td>
<td></td>
</tr>
<tr>
<td>2.</td>
<td>H2 Gas Outside the Tubes</td>
<td></td>
</tr>
<tr>
<td>3.</td>
<td>222.0C Water Flow Rate (lbs/sec-ft)</td>
<td></td>
</tr>
<tr>
<td>4.</td>
<td>134.4 Gas Flow Rate (lbs/sec)</td>
<td></td>
</tr>
<tr>
<td>5.</td>
<td>500000. Total Power (kwe)</td>
<td></td>
</tr>
<tr>
<td>6.</td>
<td>0.40% Thermal Efficiency</td>
<td></td>
</tr>
<tr>
<td>7.</td>
<td>Temperatures at each block (F)</td>
<td></td>
</tr>
<tr>
<td>8.</td>
<td>Economizer Boiler Superheater</td>
<td></td>
</tr>
<tr>
<td>9.</td>
<td>Pumping Power to Total Heat Removal (Percent)</td>
<td></td>
</tr>
<tr>
<td>10.</td>
<td>Pressure Drop and Absolute Pressures (psia)</td>
<td></td>
</tr>
<tr>
<td>11.</td>
<td>Mean Heat Transfer per Unit Area (BTU/hr-ft² of Inside Tube Surface)</td>
<td></td>
</tr>
<tr>
<td>12.</td>
<td>Mean Temperature Difference (F)</td>
<td></td>
</tr>
<tr>
<td>13.</td>
<td>Mean Heat Transfer Coefficients (BTU/hr-ft² of Inside Tube Surface)</td>
<td></td>
</tr>
<tr>
<td>14.</td>
<td>Superheater Heat Transfer Coefficients</td>
<td></td>
</tr>
<tr>
<td>15.</td>
<td>Enthalpies at each block (BTU/lb)</td>
<td></td>
</tr>
<tr>
<td>16.</td>
<td>Width of Generator (Ft)</td>
<td></td>
</tr>
<tr>
<td>17.</td>
<td>Number of Layers per Pass</td>
<td></td>
</tr>
<tr>
<td>18.</td>
<td>Diameter of Lumber</td>
<td></td>
</tr>
<tr>
<td>19.</td>
<td>Tube Pitch (in)</td>
<td></td>
</tr>
<tr>
<td>20.</td>
<td>69% Total Number of Layers</td>
<td></td>
</tr>
<tr>
<td>21.</td>
<td>26146.1 Total Tube Length (Ft)</td>
<td></td>
</tr>
<tr>
<td>22.</td>
<td>617315. Total Tube Weight (lbs)</td>
<td></td>
</tr>
<tr>
<td>23.</td>
<td>24.520 Shell Diameter (Ft)</td>
<td></td>
</tr>
<tr>
<td>24.</td>
<td>4.560 Shell Wall Thickness (in)</td>
<td></td>
</tr>
<tr>
<td>25.</td>
<td>79.4 Shell Height (Ft)</td>
<td></td>
</tr>
<tr>
<td>26.</td>
<td>114.76 lbs. Shell Weight (lbs)</td>
<td></td>
</tr>
<tr>
<td>27.</td>
<td>1762C. Tube Cost</td>
<td></td>
</tr>
<tr>
<td></td>
<td>247R283. Shell Cost</td>
<td></td>
</tr>
<tr>
<td></td>
<td>96NCC Insulation Cost</td>
<td></td>
</tr>
</tbody>
</table>

#### Table 5. Sample Computer Output Data for Cross-Flow Bare-Tube Steam Generators

<table>
<thead>
<tr>
<th>Case #</th>
<th>CROSSFLCW BARE TUBE STEAM GENERATOR</th>
<th>CAE</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>12CC Tube Inside Diameter (in)</td>
<td></td>
</tr>
<tr>
<td>2.</td>
<td>H2 Gas Outside the Tubes</td>
<td></td>
</tr>
<tr>
<td>3.</td>
<td>222.0C Water Flow Rate (lbs/sec-ft)</td>
<td></td>
</tr>
<tr>
<td>4.</td>
<td>134.4 Gas Flow Rate (lbs/sec)</td>
<td></td>
</tr>
<tr>
<td>5.</td>
<td>500000. Total Power (kwe)</td>
<td></td>
</tr>
<tr>
<td>6.</td>
<td>0.40% Thermal Efficiency</td>
<td></td>
</tr>
<tr>
<td>7.</td>
<td>Temperatures at each block (F)</td>
<td></td>
</tr>
<tr>
<td>8.</td>
<td>Economizer Boiler Superheater</td>
<td></td>
</tr>
<tr>
<td>9.</td>
<td>Pumping Power to Total Heat Removal (Percent)</td>
<td></td>
</tr>
<tr>
<td>10.</td>
<td>Pressure Drop and Absolute Pressures (psia)</td>
<td></td>
</tr>
<tr>
<td>11.</td>
<td>Mean Heat Transfer per Unit Area (BTU/hr-ft² of Inside Tube Surface)</td>
<td></td>
</tr>
<tr>
<td>12.</td>
<td>Mean Temperature Difference (F)</td>
<td></td>
</tr>
<tr>
<td>13.</td>
<td>Mean Heat Transfer Coefficients (BTU/hr-ft² of Inside Tube Surface)</td>
<td></td>
</tr>
<tr>
<td>14.</td>
<td>Superheater Heat Transfer Coefficients</td>
<td></td>
</tr>
<tr>
<td>15.</td>
<td>Enthalpies at each block (BTU/lb)</td>
<td></td>
</tr>
<tr>
<td>16.</td>
<td>Width of Generator (Ft)</td>
<td></td>
</tr>
<tr>
<td>17.</td>
<td>Number of Layers per Pass</td>
<td></td>
</tr>
<tr>
<td>18.</td>
<td>Diameter of Lumber</td>
<td></td>
</tr>
<tr>
<td>19.</td>
<td>Tube Pitch (in)</td>
<td></td>
</tr>
<tr>
<td>20.</td>
<td>69% Total Number of Layers</td>
<td></td>
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<td>23.</td>
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<tr>
<td>24.</td>
<td>4.560 Shell Wall Thickness (in)</td>
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</tr>
<tr>
<td>25.</td>
<td>79.4 Shell Height (Ft)</td>
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<td></td>
<td>247R283. Shell Cost</td>
<td></td>
</tr>
<tr>
<td></td>
<td>96NCC Insulation Cost</td>
<td></td>
</tr>
<tr>
<td></td>
<td>4384477. Tube, Shell, and Insulation Cost</td>
<td></td>
</tr>
</tbody>
</table>
### Table 6. Sample Computer Input Data for the Cross-Flow Circular-Pinned Tube Steam Generators

<table>
<thead>
<tr>
<th>AXIAL-FLOW STEAM GENERATOR</th>
<th>AXIAL-FLOW REHEATER</th>
<th>CROSS-FLOW STEAM GENERATOR</th>
<th>CROSS-FLOW REHEATER</th>
<th>CROSS-FLOW RECIRCULATING BOILER</th>
<th>AXIAL-FLOW RECIRCULATING BOILER</th>
</tr>
</thead>
</table>

\*This card must appear only once no matter how many cases are run, since it controls whether there will be iteration for the outside hydraulic diameter. If iteration is desired, any non-zero integer can be placed in column two. The second field will then contain the fractional percentage of accuracy desired in \( W/Q \). If no iteration is desired, (if \( W/Q \) is to be calculated), this card is left blank.

1. **Inside Diameter (in.)**
   - **Number of \( W/Q \)**
   - \( W/Q = 1 \)

2. **Pressure inside tubes (psia)**
   - \( 2500 \)
   - **Pressure outside tubes (psia)**
   - \( 500 \)
   - **Temperature of water into tubes (\(^\circ\)F)**
   - \( 520 \)
   - **Temperature of gas into tubes (\(^\circ\)F)**
   - \( 1250 \)
   - **Temperature of steam from tubes (\(^\circ\)F)**
   - \( 1050 \)
   - **Temperature of gas out of tubes (\(^\circ\)F)**
   - \( 650 \)
   - **Recirculating ratio (recirculating boiler only)**
   - **Of water recirculated (%) to steam generated (lb)**
   - \( 70 \)

3. **Fin thickness (in.)**
   - \( 0.5 \)
   - **Fin height (in.)**
   - \( 2.5 \)
   - **Number of fins per inch**
   - \( 6.0 \)
   - **Thermal conductivity of fin (Btu/hr-ft.-F)**
   - \( 1.6 \)
   - \( 5 \)

   \*If tube is bare this card must be left blank.

4. **Ratio of outside to inside diameter of tube**
   - \( 2.5 \)
   - **Molecular weight of gas**
   - \( 40 \)
   - **Thermal efficiency of cycle**
   - \( 40 \)
   - **Net electric output of the power plant (Mw)**
   - \( 50 \)
   - **Mass flow rate inside tubes (lb/hr-\(^\circ\)F)**
   - \( 12.5 \)
   - **Thermal conductivity of tube metal (Btu/hr-ft.-F)**
   - \( 60 \)

5. **Outside hydraulic diameter (in)**
   - **W/Q (pumping power-to-heat removal ratio, %)**
   - \( 16 \)
   - **Tube pitch, in. (for cross flow)**
   - \( 24 \)
   - **Minimum lambda, dimensionless (for axial flow)**
   - \( 5.0 \)
   - **Number of lamda or \((l/n)\)**
   - \( 8 \)
   - **Maximum lambda, dimensionless (for axial flow)**
   - \( 34 \)

6. **L/h, ft (for cross flow)**
   - \( 5.0 \)
   - \( 8 \)
Table 7. Sample Computer Output Data for Cross-Flow Circular-Finned-Tube Steam Generators

<table>
<thead>
<tr>
<th>Description</th>
<th>Value 1</th>
<th>Value 2</th>
<th>Value 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Insulation cost</td>
<td>3220365</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Tube cost</td>
<td>1590255</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Shell cost</td>
<td>1589225</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Total cost</td>
<td>1590255</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Case</td>
<td>#1</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**CIRCULAR FINNED TUBE CROSSFLOW STEAM GENERATOR**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value 1</th>
<th>Value 2</th>
<th>Value 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tube inside diameter (in)</td>
<td>1.2000</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Gas outside the tubes</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Water flow rate (lbs/sec-ft²)</td>
<td>222.90</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Gas flow rate (lbs/sec)</td>
<td>1592.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Total power (kw)</td>
<td>500000.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Thermal efficiency</td>
<td>0.400</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Temperatures at each block (°F)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Inside economizer boiler</td>
<td>520.00</td>
<td>668.73</td>
<td>1050.00</td>
</tr>
<tr>
<td>Outside economizer boiler</td>
<td>650.00</td>
<td>704.00</td>
<td>1008.23</td>
</tr>
<tr>
<td>Inside boiler superheater</td>
<td>500.00</td>
<td>784.00</td>
<td>1250.00</td>
</tr>
<tr>
<td>Outside boiler superheater</td>
<td>650.00</td>
<td>874.00</td>
<td>1511.54</td>
</tr>
<tr>
<td>Inside economizer boiler superheater total</td>
<td>0.093</td>
<td>0.871</td>
<td>0.941</td>
</tr>
<tr>
<td>Outside economizer boiler superheater total</td>
<td>2.752</td>
<td>7.210</td>
<td>7.500</td>
</tr>
<tr>
<td>Mean heat transfer per unit area (BTU/hr-ft² of inside tube surface)</td>
<td>510.67</td>
<td>1091.08</td>
<td>1249.26</td>
</tr>
<tr>
<td>Mean temperature difference (°F)</td>
<td>122.4k</td>
<td>227.59</td>
<td>274.79</td>
</tr>
<tr>
<td>Mean heat transfer coefficients (BTU/hr-ft² of inside tube surface)</td>
<td>346.8k</td>
<td>497.5k</td>
<td>612.0k</td>
</tr>
<tr>
<td>Inside economizer boiler superheater</td>
<td>346.80</td>
<td>497.50</td>
<td>612.00</td>
</tr>
<tr>
<td>Outside economizer boiler superheater</td>
<td>128.4k</td>
<td>227.59</td>
<td>274.79</td>
</tr>
<tr>
<td>Inside economizer boiler superheater total</td>
<td>0.322</td>
<td>0.496</td>
<td>0.492</td>
</tr>
<tr>
<td>Overall inside economizer boiler superheater</td>
<td>498.6k</td>
<td>649.2k</td>
<td>874.5k</td>
</tr>
<tr>
<td>Overall outside economizer boiler superheater</td>
<td>128.4k</td>
<td>227.59</td>
<td>274.79</td>
</tr>
<tr>
<td>Overall economizer boiler superheater total</td>
<td>347.6k</td>
<td>498.6k</td>
<td>649.2k</td>
</tr>
<tr>
<td>Superheater heat transfer coefficients (BTU/lb of inside tube surface)</td>
<td>346.8k</td>
<td>497.5k</td>
<td>612.0k</td>
</tr>
<tr>
<td>Location</td>
<td>159.5k</td>
<td>175.1k</td>
<td>208.2k</td>
</tr>
<tr>
<td>Inside economizer boiler superheater</td>
<td>346.8k</td>
<td>497.5k</td>
<td>612.0k</td>
</tr>
<tr>
<td>Outside economizer boiler superheater</td>
<td>128.4k</td>
<td>227.59</td>
<td>274.79</td>
</tr>
<tr>
<td>Inside economizer boiler superheater total</td>
<td>0.322</td>
<td>0.496</td>
<td>0.492</td>
</tr>
<tr>
<td>Overall inside economizer boiler superheater</td>
<td>498.6k</td>
<td>649.2k</td>
<td>874.5k</td>
</tr>
<tr>
<td>Overall outside economizer boiler superheater</td>
<td>128.4k</td>
<td>227.59</td>
<td>274.79</td>
</tr>
<tr>
<td>Overall economizer boiler superheater total</td>
<td>347.6k</td>
<td>498.6k</td>
<td>649.2k</td>
</tr>
<tr>
<td>Enthalpies at each block (BTU/lb)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Inside economizer boiler superheater</td>
<td>510.67</td>
<td>1091.08</td>
<td>1249.26</td>
</tr>
<tr>
<td>Outside economizer boiler superheater</td>
<td>766.9k</td>
<td>1933.23</td>
<td>2121.50</td>
</tr>
<tr>
<td>Inside economizer boiler superheater total</td>
<td>153.6k</td>
<td>374.1k</td>
<td>437.4k</td>
</tr>
<tr>
<td>Overall inside economizer boiler superheater</td>
<td>510.67</td>
<td>1091.08</td>
<td>1249.26</td>
</tr>
<tr>
<td>Overall outside economizer boiler superheater</td>
<td>766.9k</td>
<td>1933.23</td>
<td>2121.50</td>
</tr>
<tr>
<td>Overall economizer boiler superheater total</td>
<td>153.6k</td>
<td>374.1k</td>
<td>437.4k</td>
</tr>
<tr>
<td>lambdas</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>16. λ25 width of generator (ft)</td>
<td>692.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>17. number of layers per pass</td>
<td>3440.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>18. 0.35184 λ25</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>19. 2.2112 tube pitch (in)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>20. total number of tubes</td>
<td>154428.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>21. total tube length (ft)</td>
<td>692.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>22. total bare tube weight (lbs)</td>
<td>38918.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>23. shell diameter (ft)</td>
<td>18.122</td>
<td></td>
<td></td>
</tr>
<tr>
<td>24. shell wall thickness (in)</td>
<td>5.3999</td>
<td></td>
<td></td>
</tr>
<tr>
<td>25. shell height (ft)</td>
<td>1.143</td>
<td></td>
<td></td>
</tr>
<tr>
<td>26. shell weight (lbs)</td>
<td>9213.1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>27. insulation cost</td>
<td>692.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>28. total cost</td>
<td>1590255.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>29. insulation cost</td>
<td>1590255.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>30. total cost</td>
<td>1590255.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>31. insulation cost</td>
<td>1590255.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>32. total cost</td>
<td>1590255.0</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
3. BASES FOR COST ESTIMATES

An important portion of the study covered by this report was the estimation of the costs for labor, material, and shop overhead, that is, the shop costs for fabrication. The costs of design, general plant overhead, sales, transportation, and allowances for contingencies were not included because these depend so much on the particular installation involved. The following sections present in detail the bases for the estimates of the shop costs.

**Axial-Flow Bare-Tube Steam Generators**

The internal arrangement used as a basis for the axial-flow steam generator cost calculations is shown in the cross section of Fig. 8. It was assumed that the tubes would be shop fabricated into bundles of 16 tubes, as shown in Fig. 9, and transported to the location for final assembly. Inlet and outlet pipes for each bundle would penetrate the steam generator shell and be connected to header drums located in the biological shield about 10 ft from the shell.

**Shell**

It was assumed that the steam generator shell would be made of type SA-212, grade B, carbon steel and that an allowable tensile strength of 16,000 psia could be used for calculating the shell thickness. The unit cost of shell fabrication for shell thicknesses from 0.5 to 10 in. was represented with the following relation:

\[
\text{Cost in } \$/\text{lb} = 4.52 t^{0.77} ,
\]

where \( t \) is the shell thickness in feet.

Assuming a cylindrical shell with hemispherical heads, the weight of the shell is given by

\[
\text{shell weight} \approx \pi D t (L + D) \times \text{density} \quad \text{if } t \ll D ,
\]

(89)
where \( L \) is the straight portion of the steam generator tube and \( D \) is the shell diameter. The shell thickness can be evaluated from

\[
t = \frac{PD}{2\sigma} . \tag{90}
\]

Then the cost of the shell is

\[
\text{shell cost, } \$ = (\text{weight, lb})(\$/\text{lb}) = \pi D t (L + D) (\text{density})(4.52 \times t^{0.77})
\]

\[
= 4.52 \pi D (L + D) (\text{density}) t^{1.77}
\]

\[
= 4.52 \pi D (L + D) (\text{density}) \left( \frac{PD}{2\sigma} \right)^{1.77}.
\]

Assuming a density of 500 lb/ft\(^3\) and a value for \( \sigma \) of 144 \times 16,000 lb/ft\(^2\),

\[
\text{shell cost, } \$ = 1.15 \times 10^{-8} (L + D) D^{2.77} P^{1.77} , \tag{91}
\]

where \( P \) is the pressure inside the shell in psf.

The reflective insulation between the steam generator shell and its cooling annulus was assumed to consist of two layers of ordinary 20-gage steel. Taking a unit fabrication cost of \$1/lb, the cost of the thermal insulation was found to be about \$4.50/ft\(^2\) of shell wall. The installed cost of a 4-in.-thick layer of magnesia insulation over the outer surface of the shell was taken as \$4/ft\(^2\) of shell wall and the cost of the inner ductwork as equivalent to \$7.50/ft\(^2\) of shell wall. Hence the installed cost of the external thermal insulation, the internal reflective insulation, and the ducts was about \$16/ft\(^2\) of shell wall.

**Tubing**

The unit cost of 1.5\% Cr–0.5\% Mo alloy steel tubes with an outside-to-inside diameter ratio of 1.25 was assumed to be 1.5 \( \overline{d} \) \$/ft for tube internal diameters from 0.4 in. to 1.2 in., and 1.4 \( \overline{d} \) \( \overline{d} \) \$/ft for tube internal diameters greater than 1.2 in. The unit welding cost in dollars per joint was related to the tube inside diameter by
welding cost = $2.5 \overline{d}_1^{0.9}$.

The cost in dollars for each bend in a tube was evaluated from

\[
\text{cost of bend} = 18 \overline{d}_1^{0.9} \$/\text{bend}.
\]

The joints were assumed to be x-ray inspected and dye-penetrant tested, and the unit cost of these inspections was taken as $30 per joint, irrespective of the tube diameter. The cost of tube testing, hangers, and installation was evaluated as a percentage of the basic cost of the tubing, with the following assumptions:

<table>
<thead>
<tr>
<th>Inspection Type</th>
<th>Percentage</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ultrasonic testing of tubes</td>
<td>30%</td>
</tr>
<tr>
<td>Dye-penetrant testing of tubes</td>
<td>20%</td>
</tr>
<tr>
<td>Water-leak testing of tubes</td>
<td>10%</td>
</tr>
<tr>
<td>Installation of tubes</td>
<td>15%</td>
</tr>
<tr>
<td>Baffles and hangers for tubes</td>
<td>15%</td>
</tr>
<tr>
<td><strong>Sum</strong></td>
<td><strong>90%</strong></td>
</tr>
</tbody>
</table>

In selecting a tube header arrangement for cost estimating purposes, the configuration shown in Fig. 9 was chosen. The other arrangements considered entail fewer welds and hence should be less expensive. Based on assumptions listed above, the total cost of one 16-tube bundle of the type shown in Fig. 9 is

<table>
<thead>
<tr>
<th>No. of Joints or Bends</th>
<th>Diameter of Joint or Bend</th>
<th>Cost, $</th>
</tr>
</thead>
<tbody>
<tr>
<td>48</td>
<td>$\overline{d}_1$</td>
<td>$48 \times 2.5 (\overline{d}_1)^{0.9}$</td>
</tr>
<tr>
<td>16</td>
<td>$\sqrt{2} \overline{d}_1$</td>
<td>$16 \times 2.5 (\sqrt{2} \overline{d}_1)^{0.9}$</td>
</tr>
<tr>
<td>8</td>
<td>$\sqrt{4} \overline{d}_1$</td>
<td>$8 \times 2.5 (\sqrt{4} \overline{d}_1)^{0.9}$</td>
</tr>
<tr>
<td>4</td>
<td>$\sqrt{8} \overline{d}_1$</td>
<td>$4 \times 2.5 (\sqrt{8} \overline{d}_1)^{0.9}$</td>
</tr>
<tr>
<td>10</td>
<td>$\sqrt{16} \overline{d}_1$</td>
<td>$10 \times 2.5 (\sqrt{16} \overline{d}_1)^{0.9}$</td>
</tr>
<tr>
<td>16</td>
<td>$\overline{d}_1$</td>
<td>$16 \times 18 (\overline{d}_1)^{0.9}$</td>
</tr>
<tr>
<td>8</td>
<td>$\sqrt{2} \overline{d}_1$</td>
<td>$8 \times 18 (\sqrt{2} \overline{d}_1)^{0.9}$</td>
</tr>
<tr>
<td>4</td>
<td>$\sqrt{4} \overline{d}_1$</td>
<td>$4 \times 18 (\sqrt{4} \overline{d}_1)^{0.9}$</td>
</tr>
<tr>
<td>2</td>
<td>$\sqrt{8} \overline{d}_1$</td>
<td>$2 \times 18 (\sqrt{8} \overline{d}_1)^{0.9}$</td>
</tr>
</tbody>
</table>
X-ray and dye-penetrant testing of:

- 10 header joints (10 inspections) 10 x 30
- 28 bend joints (12 inspections) 12 x 30
- 48 tube joints (12 inspections) 12 x 30

The tube cost, including material cost, ultrasonic testing, dye-penetrant testing, leaktesting, installation, hangers, and baffles is

- 16 tubes per bundle, each L ft long $1.9 \times 16 \times L \times 1.5 \bar{d}_i$
- 2 header tubes, each 15 ft long and $\sqrt[4]{16 \bar{d}_i}$ in. in diameter (i.e., diameter is larger than 1.2 in.)

The cost per tube can be obtained by adding the above cost items for the tube bundle and dividing by 16.

Summarizing all the main cost items gives

\[
\text{Cost per tube in bundle, } \$, = 65 \bar{d}_i^{0.9} + 28 \bar{d}_i^{1.25} + 2.85 L \bar{d}_i + 64
\]

\[
\text{Cost of shell, } \$, = 1.15 \times 10^{-8} \times (L + D) D^{2.77} P^{1.77}
\]

\[
\text{Cost of insulation, } \$, = (\pi DL) 16
\]

where

- $\bar{d}_i$ = inside diameter of the heat exchanger tube, in.
- $D$ = diameter of steam generator shell (i.e., the pressure vessel), ft
- $L$ = tube matrix height, ft
- $P$ = pressure inside the shell, psf

The total cost of the steam generators, based on the above cost units, was defined as follows:

\[
(\text{total cost of steam generator}) = (\text{cost/tube})(\text{total No. of tubes}) + (\text{pressure vessel cost}) + (\text{insulation cost})
\]

The weights of the pressure vessel and the tubes are also important quantities for a steam generator design. The pressure vessel weight was calculated in the manner described above. The tube weights were computed using the following unit weights for the various sizes employed:
Axial-Flow Longitudinally Finned Tube Steam Generators

The unit costs for the axial-flow steam generators with longitudinally finned tubes were taken to be the same as those for the bare-tube steam generators, except that the material cost was different for the finned tubes. Assuming 1.5% Cr-0.5% Mo alloy steel finned tubes, the unit material cost was taken as

\[ \text{Cost} = (5 \overline{d}_1) \$/\text{ft of finned tube}, \]

where \( \overline{d}_1 \) is the inside diameter of the tube in inches. The cost per tube, including the cost of welding, the bends, the x-ray and dye-penetrant testing of the joints, water-leak testing, installation, ultrasonic testing of the tubes, baffles and hanger, and the tube material, was taken as

\[ \text{Cost} = (65 \overline{d}_1^{0.9} + 28 \overline{d}_1^{1.25} + 9.5 L \overline{d}_1 + 64) \$/\text{ft} \]

for the longitudinally finned tubes.

Cross-Flow Bare-Tube Steam Generators

The internal arrangement used as a basis for the cross-flow steam generator cost calculations is shown in Figs. 5 and 10. The materials of construction, inspection processes, and unit costs were taken to be the same as those used in the axial-flow steam generators where the above
relations were applicable. The fabrication process for the serpentine bends was assumed to entail welding 180-deg bends to straight lengths of tubing. The cost for each serpentine bend, including the cost of the two welds joining the bend to the tubes, was taken as

\[
\text{Cost} = 18 \left( \overline{d} \right)^{0.9} \$/\text{bend},
\]

where \( \overline{d} \) is in inches.

It was assumed that the two welds in each bend could be x-rayed in one shot and that each tube-to-header joint needed a separate x-ray. The cost of each x-ray was taken as $30. The total number of x-rays needed per tube was taken as \( N_{\text{bend}} + 2 \), where \( N_{\text{bend}} \) is the number of bends per tube.

It was assumed that the tubes would be welded to header drums in a manner similar to that shown in Chapter 6, Fig. 47. The cost of a header drum for 100 tubes having an outside diameter of 1.0 in. was estimated to be about $5000, giving a unit cost of $50/tube. Variations in this cost with the tube diameter were neglected.

Summarizing the costs listed above, the tube cost was taken as

\[
\text{Cost} = [1.9 C_M L_t + 18 (\overline{d})^{0.9} N_{\text{bend}} + 30 (N_{\text{bend}} + 2) + 50] \$/\text{tube},
\]

where

\[
L_t = \text{length of tube} = L_{tE} + L_{tB} + L_{tS}, \text{ ft},
\]

\[
N_{\text{bend}} = \text{number of bends per tube} = L_t / [((\sqrt{3}/2) S_n],
\]

\[
C_M = \text{tube material cost} = 1.5 \overline{d} \$/\text{ft}
\]

for \( 0.4 \text{ in.} < \overline{d} < 1.2 \text{ in.} \),

\[
= 1.4 (\overline{d})^{1.25} \$/\text{ft}
\]

for \( \overline{d} > 1.2 \text{ in.} \).

Hence, the total cost of the tubes was

\[
\text{Cost} = ($/\text{tube})(\text{total No. of tubes}).
\]
The height of the pressure vessel was taken as equal to the tube matrix height plus the pressure vessel diameter. The pressure vessel cost was calculated in the same manner as that described for the axial-flow steam generators. The shop cost of the cross-flow steam generator given in this report includes the cost of the tubes, pressure vessel, insulation, and header drum.

The unit costs for circular finned tube units were essentially the same as those for the bare-tube cross-flow steam generator except that the unit material cost of the tube was different. It was assumed that circular fins would be brazed to 1.5% Cr-0.5% Mo alloy tubes. The cost per tube, including the cost of welding, ultrasonic and dye-penetrant testing, water-leak testing, x-ray testing, bends, baffles and hangers, tube material, and installation, was calculated from the following relation:

\[
\text{Cost} = [1.9 \, C_M \, L_t + 18 \, (\bar{d}_t)^{0.9} \, N_{\text{bend}} + \\
+ 30 \, (N_{\text{bend}} + 2) + 50] \, \$/\text{tube},
\]

where the units are the same as those given above, except that

\[
C_M = 3.6 \, (\bar{d}_t)^{0.12} \, \$/\text{ft} \quad \text{for } 0.5 < \bar{d}_t < 1.2 \, \text{in.} \quad \text{and} \quad 1/8 \, \text{in.} \leq \text{fin height} \leq 3/8 \, \text{in.}
\]
4. DESIGN CHARTS

If facilities for carrying out machine calculations are not readily available, or if only a few cases are to be calculated, charts are a valuable aid. They not only make it possible to eliminate much tedious work, but they provide an excellent insight into the effects of the major variables.

It should be mentioned that in the machine code the temperature dependence of the heat transfer coefficient along the tube was considered, so the results of a machine calculation represent a better approximation than that obtainable from hand calculations, especially for the superheater section. A comparison of the tube lengths obtained from hand calculations with those from the computer indicated that the tube lengths for the economizer and the boiler sections agreed closely, but the tube length for the superheater section obtained with the computer was shorter than that given by hand calculations made with the steam physical properties at the mean superheater temperature. The charts presented in Figs. 11, 12, 13, and 19 were therefore prepared from machine calculations. The charts in Figs. 14 through 18 are for U-tube configurations for which no machine code was prepared, and hence these were constructed from hand calculations.

The simplest case covered by a chart is that of Fig. 11 for 0.5-in.-diam straight bare-tube steam generators operating with 1000-psia helium entering at 1350°F and leaving at 650°F, feedwater entering at 520°F, and steam leaving at 1050°F and 2500 psia. The effects of the equivalent diameter of the gas-side passages on the tube length and the pumping power-to-heat removal ratio are shown for the economizer, boiler, and superheater sections. As may be seen, the curves on this figure are almost straight lines, so such charts can easily be constructed by calculating only a few points.

The chart presented in Fig. 11 may be employed to obtain the solution of many problems. For example, if the pumping power-to-heat removal ratio for the entire steam generator is to be 1%, it can be seen by inspection of the upper set of curves that the equivalent passage diameter on the gas side should be about 1.32 in., so the sum of the pumping
Fig. 11. Design Chart for Axial-Flow Bare-Tube Steam Generator Superheater, Boiler, and Economizer Sections.
power-to-heat removal ratios for the three sections will be

\[ 0.295 + 0.315 + 0.39 = 1.0\% \]

For this case the lower set of curves gives tube lengths of 27.8, 21.4, and 17.4 ft for the economizer, boiler, and superheater sections, respectively, and thus an overall tube length of 66.6 ft. Note how much simpler it is to use this chart than to go through a set of calculations such as that in Table 1.

Figure 11 may be applied to many other problems. For example, since the gas-side heat transfer coefficient depends on the gas mass flow rate rather than the gas pressures, the data presented can be used for other

---

**Fig. 12.** Design Chart for Axial-Flow Bare-Tube Steam Generator Operating with CO₂ as a Function of Tube Diameter.
gas pressures simply by changing the scale of the pumping power by the inverse ratio of the square of the pressure (i.e., at a given gas mass flow rate the pumping power-to-heat removal ratio would be increased four times if the gas pressure were halved). The chart can also be used if a tube matrix configuration is to be considered in which the gas-side equivalent passage diameters differ for the economizer, boiler, and superheater.

Figure 12 presents a chart that differs from that of Fig. 11 in several respects. The gas is CO₂, and both the total tube length, L (i.e., sum of the economizer, boiler, and superheater sections), and the equivalent diameter on the gas side were plotted against the pumping power-to-heat removal ratio for several different tube diameters. Again the curves are almost straight lines, and only a few points were required to construct each curve.

![Design Chart for Axial-Flow Bare-Tube Steam Generator Operating with Helium as a Function of Gas Inlet Temperature.](chart.png)
Fig. 14. Design Chart for U-Bend Axial-Flow Bare-Tube Steam Generator (No Reheater) as a Function of Gas Passage Equivalent Diameter for Inner Leg.
GAS: CO\textsubscript{2} GAS AT 1000 psia GAS IN AT 1350 °F GAS OUT AT 650 °F

STEAM GENERATOR: WATER IN AT 520 °F STEAM OUT AT 1050 °F STEAM AT 2500 psia

MASS FLOW RATE OF STEAM = 222.9 lb/ft\textsuperscript{2}.sec TUBES = 0.4 in. ID, 0.5 in. OD, BARE

REHEATER: STEAM IN AT 650 °F STEAM OUT AT 1000 °F STEAM AT 450 psia

MASS FLOW RATE OF STEAM = 222.9 lb/ft\textsuperscript{2}.sec

EVAPORATION IN THE BOILER EQUALLY DIVIDED BETWEEN LEGS 1 AND 2

---

Fig. 15. Design Chart for U-Bend Bare-Tube Axial-Flow Steam Generator with Reheater Operating with CO\textsubscript{2}. 
Figure 13 is a still different variation of Fig. 11 that describes the effects of changing the gas inlet temperature from 1350 to 1250°F. Figure 14 presents a chart to aid in the design of axial-flow U-tube steam generators (without reheaters) for the operating conditions stated on the figure. The matrix length, pumping power-to-heat removal ratio,  

![Design Chart for U-Bend Bare-Tube Axial-Flow Steam Generator with Reheater as a Function of Gas Passage Equivalent Diameters Operating with CO\textsubscript{2}](image)

Fig. 16. Design Chart for U-Bend Bare-Tube Axial-Flow Steam Generator with Reheater as a Function of Gas Passage Equivalent Diameters Operating with CO\textsubscript{2}.
and the fraction of the boiling taking place in the inner (steam downflow) leg were plotted against the gas-passage equivalent diameter for the outer leg for each of several values for the gas-passage equivalent diameter for the inner leg. Figure 14 was prepared from a chart for an axial-flow straight-tube steam generator with no reheater, similar to that shown in Fig. 11. The tube length was obtained from the following relation by assuming a uniform heat flux along the boiler section in the U-tube matrix as a fair approximation and by noting that the fraction of fluid evaporated was proportional to the length of the section for

![Diagram](https://example.com/diagram.png)

Fig. 17. Design Chart for U-Bend Bare-Tube Axial-Flow Steam Generator with Reheater Operating with Helium.
boiling:

\[ L_{1S} + xL_{1B} = L_{oE} + (1 - x) L_{oB} \]

or

\[ x = \frac{L_{oE} + L_{oB} - L_{1S}}{L_{1B} + L_{oB}} \]

---

**Fig. 18.** Design Chart for U-Bend Bare-Tube Axial-Flow Steam Generator with Reheater as a Function of Gas Passage Equivalent Diameter; Operating with Helium.
where

\[ L_{IS}, L_{IB} = \text{lengths of the superheater and boiler sections obtained from a chart similar to Fig. 11 for the equivalent passage diameter, } D_{ei}, \text{ for the inner leg,} \]

\[ L_{OE}, L_{OB} = \text{lengths of the economizer and boiler sections obtained from a chart similar to Fig. 11 for the equivalent passage diameter, } D_{eo}, \text{ for the outer leg,} \]

\[ x = \text{fraction of boiler tube length in the inner leg or fraction of steam evaporating inside the inner leg.} \]

---

**Fig. 19.** Design Chart for Cross-Flow Steam Generator (No Reheater).
The values of \( L_{OB} \) and \( L_{OB} \) were obtained from a chart similar to that shown in Fig. 11 for a given value of \( D_{eo} \) and \( L_{15} \) and \( L_{1B} \) for a given value of \( D_{ei} \). Hence, the fraction of boiler tube in the inner leg, \( x \), can be calculated for the specified values of \( D_{ei} \) and \( D_{eo} \). Knowing \( x \), the matrix length, \( L \), and the pumping power-to-heat removal ratio were then calculated.

Figure 15 is a design chart for a U-bend axial-flow bare-tube steam generator with a reheater. The procedure for calculating the tube length and the pumping power-to-heat removal ratio for steam generators with a reheater was described in Chapter 1 and was followed in constructing this chart. As may be seen the data in this figure are for 0.5-in.-o.d. U-bend (i.e., steam-generator) tubes. The relation between the diameter and the spacing of the reheater tubes for a given matrix length is presented as a separate chart in the lower right of Fig. 15. It may be seen that, for a given tube-matrix length, the diameter and the spacing of the reheater tubes should be selected so that they satisfy the equation shown on the figure. This equation was derived by assuming that the pressure drop across the reheater leg was equal to the pressure drop across the steam generator leg on the reheater side. A trial-and-error solution is required for each case.

For many types of calculation, the data of Fig. 15 can be employed more conveniently when replotted in the form of Fig. 16. Figures 17 and 18 are similar to Figs. 15 and 16 except that they are for helium instead of \( CO_2 \). Figure 19 is a design chart for a 1.5-in.-o.d. bare-tube cross-flow steam generator with equilateral tube spacing. The pumping power-to-heat removal ratio and the tube matrix height were plotted against the \((l/n)\) ratio for two values for the tube spacing, \( S \). The curve in the lower right corner gives the total thermal output of the steam generator matrix as a function of the total number of tubes, which is equal to \( ln/S \). Therefore, for any given pumping power-to-heat removal ratio and either of the two given values for the tube spacing, the tube matrix height and the \((l/n)\) ratio can be determined from the upper left chart. From the thermal power output of the matrix, the quantity \( ln/S \) can be determined from the chart at the lower right. Since the value of \( S \) was
chosen, \( l/n \) and \( \ln \) are known. Hence, the matrix width \( l \) and the number of layers per pass \( n \) can be determined.

Design charts to facilitate the determination of gas passage equivalent diameter and flow passage area as functions of tube diameter and spacing are presented in Figs. 20 and 21.

Fig. 20. Ratio of Equivalent Passage Diameter on Shell Side to Tube Outside Diameter as a Function of Ratio of Tube Spacing to Tube Outside Diameter for Equilateral Pitch.
Fig. 21. Ratio of Gas-Side Flow Passage Area to Total Tube Bundle Cross-Sectional Area as a Function of Tube Diameter and Spacing.
The principal parameters affecting the size, cost, and weight of the steam generators for gas-cooled reactors are the tube matrix configuration, the tube diameter, the amount of extended surface, the pumping power-to-heat removal ratio, the gas employed as a heat transfer medium, and the gas and steam temperature and pressure conditions. The combinations of all these factors of possible interest are too numerous to examine in detail. In the present study the axial-flow and cross-flow tube arrangements were compared for the range of temperature and pressure conditions likely to be of interest for gas-cooled reactors employing either stainless steel-encapsulated UO$_2$ or all-ceramic fuel elements. While a broad range of steam pressure and temperature conditions was investigated for some of the more important cases, in most instances the steam cycle considered employed a feedwater inlet temperature of 520°F and a steam pressure and temperature at the superheater outlet of 2500 psi and 1050°F for a 500-Mw electrical output power plant.

A summary of the cases that were calculated on the IBM 7090 computer for the axial- and cross-flow steam generators is presented in Table 8. The parametric studies were first performed to investigate the effects of tube size, the pumping power-to-heat removal ratio, and the allowable pressure drop for the gas flow in the central gas duct on the cost and size of bare-tube axial-flow steam generators for 500-Mw net electrical output plants. The effects of other parameters on size and cost were then investigated for the more promising combinations of conditions from the standpoint of cost, size, or detail design considerations.

Perhaps the most important consideration in evaluating the effects of the various parameters is some measure of over-all costs. The method of calculation made it easy to estimate the cost of the tubing, the tube-to-header joints, the shell, and the thermal insulation, that is, those items which can be considered as constituting the shop cost. Plant overhead, engineering, sales, transportation, erection, and testing costs constitute additional items that are difficult to estimate but which would probably make the over-all cost two to four times the shop costs,
Table 8. Summary of the Cases Investigated on the Computer for Axial- and Cross-Flow Steam Generators

<table>
<thead>
<tr>
<th>No.</th>
<th>Purpose of Study and Number of Parameters Investigated</th>
<th>Bare or Finned Tubes</th>
<th>Axial-Flow Steam Generators</th>
<th>Bare</th>
<th>F</th>
<th>λ</th>
<th>Gas</th>
<th>Steam Pressure (psia)</th>
<th>Gas Inlet Temperature (°F)</th>
<th>Gas Outlet Temperature (°F)</th>
<th>Feedwater Inlet Temperature (°F)</th>
<th>Steam Outlet Temperature (°F)</th>
<th>Thermal Efficiency of Cycle</th>
<th>Power Plant Size, Net Electrical Output (MW)</th>
<th>Mass Flow Rate of Steam (lb/ft²·sec)</th>
</tr>
</thead>
<tbody>
<tr>
<td>I</td>
<td>Effect of $\delta$, $F_0$, and $\lambda$ with $CO_2$ at 135°F; $4 \times 4 \times 4 = 64$ cases</td>
<td>Bare 0.5 0.002 0.01</td>
<td>$CO_2$</td>
<td>1000</td>
<td>2500</td>
<td>1350</td>
<td>650</td>
<td>520</td>
<td>1050</td>
<td>0.40</td>
<td>500</td>
<td>222.9</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>II</td>
<td>Effect of $F_0$ and $\lambda$ with helium at 1250°F; $4 \times 5 = 20$ cases</td>
<td>Bare 0.5 0.002 0.01</td>
<td>Helium</td>
<td>1000</td>
<td>2500</td>
<td>1250</td>
<td>650</td>
<td>520</td>
<td>1050</td>
<td>0.40</td>
<td>500</td>
<td>222.9</td>
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<td></td>
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</tr>
<tr>
<td>III</td>
<td>Effect of $F_0$ and $\lambda$ with helium at 1250°F; $4 \times 5 = 20$ cases</td>
<td>Bare 0.5 0.002 0.01</td>
<td>Helium</td>
<td>1500</td>
<td>2500</td>
<td>1250</td>
<td>650</td>
<td>520</td>
<td>1050</td>
<td>0.40</td>
<td>500</td>
<td>222.9</td>
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<tr>
<td>IV</td>
<td>Effect of gas pressure with $CO_2$ and helium; $2 \times 4 = 8$ cases</td>
<td>Bare 0.5 0.005 0.10</td>
<td>$CO_2$</td>
<td>1500</td>
<td>2500</td>
<td>1250</td>
<td>650</td>
<td>520</td>
<td>1050</td>
<td>0.40</td>
<td>500</td>
<td>222.9</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>V</td>
<td>Effect of gas outlet temperature; 3 cases</td>
<td>Bare 0.5 0.005 0.10</td>
<td>Helium</td>
<td>500</td>
<td>2500</td>
<td>1250</td>
<td>650</td>
<td>520</td>
<td>1050</td>
<td>0.40</td>
<td>500</td>
<td>222.9</td>
<td></td>
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<tr>
<td>VI</td>
<td>Effect of gas inlet temperature; 4 cases</td>
<td>Bare 0.5 0.005 0.10</td>
<td>Helium</td>
<td>500</td>
<td>2500</td>
<td>1250</td>
<td>650</td>
<td>520</td>
<td>1050</td>
<td>0.40</td>
<td>500</td>
<td>222.9</td>
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<tr>
<td>VII</td>
<td>Effect of feedwater inlet temperature; 3 cases</td>
<td>Bare 0.5 0.005 0.10</td>
<td>Helium</td>
<td>500</td>
<td>2500</td>
<td>1250</td>
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<td>VIII</td>
<td>Effect of steam outlet temperature with 1200-psia steam; 4 cases</td>
<td>Bare 0.5 0.005 0.10</td>
<td>Helium</td>
<td>500</td>
<td>1200</td>
<td>1250</td>
<td>550</td>
<td>420</td>
<td>800</td>
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<tr>
<td>IX</td>
<td>Effect of $\delta$, and steam outlet temperature with 2500-psia steam; $4 \times 5 = 20$ cases</td>
<td>Bare 0.5 0.005 0.10</td>
<td>Helium</td>
<td>500</td>
<td>2500</td>
<td>1250</td>
<td>650</td>
<td>520</td>
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<tr>
<td>X</td>
<td>Effect of steam outlet temperature with 2500-psia steam; 4 cases</td>
<td>Bare 0.5 0.005 0.10</td>
<td>Helium</td>
<td>500</td>
<td>3000</td>
<td>1250</td>
<td>687</td>
<td>557</td>
<td>900</td>
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<tr>
<td>XI</td>
<td>Effect of steam outlet temperature; 4 cases</td>
<td>Bare 0.5 0.005 0.10</td>
<td>Helium</td>
<td>500</td>
<td>2500</td>
<td>1250</td>
<td>650</td>
<td>520</td>
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<tr>
<td>XII</td>
<td>Effect of power plant size and gas pressure; 3 cases</td>
<td>Bare 0.5 0.005 0.10</td>
<td>Helium</td>
<td>300</td>
<td>2500</td>
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<td>520</td>
<td>1050</td>
<td>0.40</td>
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### Table 8. (continued)

<table>
<thead>
<tr>
<th>No.</th>
<th>Purpose of Study and Number of Parameters Investigated</th>
<th>Bare or Finned Tubes</th>
<th>Lc (in.)</th>
<th>Po</th>
<th>λ</th>
<th>Gas</th>
<th>Gas Pressure (psia)</th>
<th>Steam Pressure (psia)</th>
<th>Gas Inlet Temperature (°F)</th>
<th>Gas Outlet Temperature (°F)</th>
<th>Feedwater Inlet Temperature (°F)</th>
<th>Steam Outlet Temperature (°F)</th>
<th>Thermal Efficiency of Cycle</th>
<th>Power Plant Size, Net Electrical Output (Mw)</th>
<th>Mass Flow Rate of Steam (lb/ft²·sec)</th>
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<tr>
<td>XIII</td>
<td>Effect of fin height; 3 cases</td>
<td>Axial fins&lt;sup&gt;a&lt;/sup&gt;</td>
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<td>Cross-Flow Steam Generators</td>
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<tr>
<td>I</td>
<td>Effect of gas pressure, Bare</td>
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<td>&lt;0.1</td>
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<td>tube diameter, gas temperature</td>
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<td>Effect of fins</td>
<td>Circular fins&lt;sup&gt;b&lt;/sup&gt;</td>
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<td>Helium</td>
<td>150</td>
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<sup>a</sup>Axial fins 1/8, 1/4, or 3/8 in. high, 0.050 in. thick, 6 fins/in.

<sup>b</sup>Measured at roots of fins.

<sup>c</sup>Circular fins 0.25 in. high, 0.050 in. thick, 6 fins/in., k = 16.5 Btu/hr·ft·°F.
depending on the fabricator's background of experience with this type of steam generator and on the number of similar units to be built. In plotting the curves, the calculated shop costs were divided by $10^6$ and plotted as "relative costs." If these relative cost are multiplied by from $2,000,000$ to $4,000,000$, they will probably give a good indication of the actual delivered price.

**Straight-Tube Axial-Flow Steam Generators**

The first set of parametric studies was carried out for straight bare-tube axial-flow steam generators because these presented the simplest case from the computing standpoint. Since design studies for high-temperature gas-cooled reactors had indicated important advantages for concentric-duct configurations in which the hot gas duct entering the steam generator would be surrounded by the cooler gas leaving the steam generator, it was assumed in all cases that the hot gas would enter the bottom of the steam generator and flow upward to the top through a central circular duct. It would then enter the tube matrix at the top of the steam generator and flow downward through the tube matrix between the central duct and the pressure vessel.

**Effects of Tube Diameter and Ratio of Pumping Power to Heat Removal**

The effects of tube diameter, pumping power-to-heat removal ratio, and lambda, the ratio of the pressure loss in the central duct to the pressure loss in the tube matrix, for CO$_2$ as the reactor coolant entering the steam generator at 1350°F and leaving at 650°F are shown in Figs. 22 through 25. At first glance it appears that the higher the value of lambda the lower the cost and weight of a unit because the abscissa in these curves is the ratio of pressure drop to heat removal for the tube matrix alone. It can be seen from Fig. 26, however, that, if the data are re-plotted against the over-all steam generator pressure drop, that is, the duct pressure loss plus the tube matrix pressure loss, the size and cost are near minimum for $\lambda = 0.10$; this value of $\lambda$ was therefore used in most of the subsequent calculations.
NET ELECTRICAL OUTPUT: 500 Mw
CO₂ PRESSURE: 1000 psia
STEAM PRESSURE: 2500 psia
WATER INLET TEMPERATURE: 520 °F
STEAM OUTLET TEMPERATURE: 1050 °F
STEAM FLOW RATE: 222.9 lb/ft²·sec
λ = RATIO OF PRESSURE DROP IN DUCT TO PRESSURE DROP OVER TUBE BUNDLE

Fig. 22. Effects of Pumping Power-to-Heat Removal Ratio and Duct Pressure Drop on the Cost and Size of Axial-Flow Bare-Tube Steam Generators with 0.50-in.-OD Tubes Operating with CO₂.

In examining the curves of Figs. 22 through 25 for the effects of tube diameter, it is apparent that the larger the tube diameter the greater the size of the steam generator and the higher the cost, especially at the lower values of pumping power-to-heat removal ratios. In view of this, all subsequent work with the axial-flow steam generators was carried out for 1/2-in.-OD tubes.

The cost of the steam generator is not the only factor that influences the choice of pumping power-to-heat removal ratio; the capital and
Fig. 23. Effects of Pumping Power-to-Heat Removal Ratio and Duct Pressure Drop on the Cost and Size of Axial-Flow Bare-Tube Steam Generators with 0.75-in.-OD Tubes Operating with CO₂.

Operating costs of the blowers are also important. Figure 27 shows the sum of the capital charges for both the steam generator and that portion of the blower chargeable to the steam generator as a function of the pumping power-to-heat removal ratio in the steam generator. The cost of the blowers was assumed to be $75/kw of blower power input. No allowance was made for power costs; these would tend to shift the minimum cost point to the left. Thus it appears that the costs are close to a minimum for a
Fig. 24. Effects of Pumping Power-to-Heat Removal Ratio and Duct Pressure Drop on the Cost and Size of Axial-Flow Bare-Tube Steam Generators with 1.00-in.-OD Tubes Operating with CO₂.

The principal gases being considered for use as coolants in gas-cooled reactors are CO₂ and helium. To investigate the influence of the
Fig. 25. Effects of Pumping Power-to-Heat Removal Ratio and Duct Pressure Drop on the Cost and Size of Axial-Flow Bare-Tube Steam Generators with 1.25-in.-OD Tubes Operating with CO₂.
Fig. 26. Effects of Pumping Power-to-Heat Removal Ratio for Tube Bundle Plus That for the Duct on the Cost and Size of Steam Generators Operating with CO₂.

Fig. 27. Effects of Pumping Power-to-Heat Removal Ratio on the Sum of the Steam Generator Cost and the Blower Cost Chargeable to the Steam Generator Operating with Helium.
choice of coolant gas on the size and cost of axial-flow steam generators, Fig. 28 was prepared for the same conditions as Fig. 22, except that helium was used in place of CO₂. A comparison of Figs. 22 and 28 indicates that the choice of gas has relatively little effect, except to increase the tube spacing for helium, and this increases the pressure vessel cost to some extent.

Fig. 28. Effects of Pumping Power-to-Heat Removal Ratio and Duct Pressure Drop on the Cost and Size of Axial-Flow Bare-Tube Steam Generators with 0.50-in.-OD Tubes Operating with Helium at an Inlet Temperature of 1350°F.
Effects of Gas Temperature Entering the Steam Generator

The choice of reactor outlet gas temperature is influenced by steam generator cost considerations. As a first step in investigating the effects of steam generator gas inlet temperature, Fig. 29 was prepared for the same conditions as Fig. 28 except that the gas temperature entering the steam generator was dropped from 1350 to 1250°F. In comparing the two sets of curves it is apparent that changing the gas temperature has essentially no effect on the general shape and character of the curves.

Fig. 29. Effects of Pumping Power-to-Heat Removal Ratio and Duct Pressure Drop on the Cost and Size of Axial-Flow Bare-Tube Steam Generators with 0.50-in.-OD Tubes Operating with Helium at an Inlet Temperature of 1250°F.
The principal effect was to increase the diameter by about 10%, the height by about 15%, the pressure vessel weight by nearly 50%, and the cost by about 25% for \( \lambda = 0.10 \). These effects are shown more clearly in Fig. 30, where the principal parameters are plotted as a function of the gas temperature entering the steam generator for a range from 1100 to 1500°F. Perhaps the most important curve is that for the sum of the costs for the

Fig. 30. Effects of Gas Inlet Temperature on the Size and Cost of Axial-Flow Bare-Tube Steam Generators Operating with Helium.
tubes, the shell, and the thermal insulation. This shows that the manufacturing costs drop rapidly to about 1250°F and then less rapidly to 1500°F. The size and weight parameters vary in a similar fashion.

While it would appear from Fig. 30 that the higher the gas temperature entering the steam generator the lower the cost, reliability considerations indicate that it is best not to use gas temperatures entering the steam generator greater than about 1250°F. As shown in Chapter 8, the difference in the cost of the control system, not to mention the difference in reliability, might more than counterbalance the lower cost at higher gas temperatures. Further, reactor fuel element fission-gas retention considerations also favor a lower reactor coolant gas temperature. For these reasons the bulk of the subsequent analysis was carried out relative to a basic reference condition of 1250°F for the gas temperature entering the steam generator.

**Effects of Gas Outlet Temperature**

The choice of the reactor inlet gas temperature, and hence the steam generator outlet gas temperature, depends on several major considerations. Perhaps the most important is that the pressure vessel operating temperature should not exceed about 700°F because of strength considerations. In this respect there is little advantage to reducing the pressure vessel operating temperature, especially since radiation damage from fast neutrons becomes progressively more important as the operating temperature is reduced below about 600°F. At the same time, the over-all cycle efficiency is improved by increasing the amount of regenerative feedwater heating, and hence it is desirable to employ a feedwater temperature of at least 500°F. On the other hand, the higher the gas temperature entering the reactor the greater the pumping power requirement. While it was not appropriate to attempt an over-all optimization of the complete nuclear power plant in this study, the effects of varying the reactor outlet gas temperature over the range from 600 to 700°F were investigated, and the results are presented in Fig. 31. The curves show that there is a definite incentive to increase the outlet gas temperature to 650°F, but that the incentive to increase it further to 700°F is not large.
Fig. 31. Effects of Gas Outlet Temperature on the Size and Cost of Axial-Flow Bare-Tube Steam Generators Operating with Helium.

These curves indicate that small changes in operating conditions have little effect on the relative position of curves such as those in Fig. 28, that is, on the particular values of pumping power-to-heat removal ratio and \( \lambda \) chosen for purposes of comparison. Because of this, the bulk of the subsequent test work was carried out using a pumping power-to-heat removal ratio of 0.5% for the tube matrix, and the pressure losses in the hot duct (where such a duct was used) were kept to 10% of the pressure losses in the tube matrix.

Effects of Gas Pressure

The selection of the gas pressure for a gas-cooled reactor system is difficult and depends in substantial measure on the effect of gas
pressure on the cost and size of the steam generator. These effects were investigated and the results plotted in Fig. 32 for both helium and $\text{CO}_2$. The cost of the tubing, shell, and thermal insulation drops rapidly as the system pressure is increased to about 500 psi and then less rapidly as the pressure is increased to 1000 psi. The choice of coolant gas appears to have little effect, except that, as in the previous comparison,

Fig. 32. Effects of Gas Pressure on the Size and Cost of Axial-Flow Bare-Tube Steam Generators Operating with Helium or $\text{CO}_2$. 
the CO₂ requires less flow passage area and hence permits the use of a smaller diameter pressure vessel and thus gives a lower unit cost. In choosing a gas pressure for a reference condition for subsequent analysis, it seemed best to choose 500 psi, in part because this represents less of an extrapolation from past experience and in part because economic studies of over-all reactor systems favor a lower pressure than that giving minimum steam generator costs.

Effects of Steam Conditions

The effects of feedwater temperature on the major parameters were investigated for the basic reference design conditions, and the results are plotted in Fig. 33. Again it is apparent that the 520°F feedwater
inlet temperature chosen for reference purposes seems to be a good compromise in that the cost of the tubing, shell, and thermal insulation drops fairly rapidly as the feedwater inlet temperature is reduced to 520°F and less rapidly for further reductions. The tube length and cost increase with increasing feedwater temperature, but the cycle thermal efficiency was about 38% with 470°F feedwater temperature and 43% with 570°F feedwater temperature. Improving the cycle efficiency reduces the fuel cycle cost and the capital charges. These opposing effects tend to give an optimum feedwater temperature between 500 and 550°F. On the whole, the over-all costs are relatively insensitive to the feedwater inlet temperature.

The effects of the steam temperature leaving the superheater were investigated for three pressure conditions, namely, 1200, 2500, and 3000 psi, and the results are plotted in Figs. 34 through 36. It is apparent

![Graphs showing effects of steam outlet temperature on cost and size of bare-tube steam generators.](image)

**Fig. 34.** Effects of Steam Outlet Temperature on the Cost and Size of Axial-Flow Bare-Tube Steam Generators.
Fig. 35. Effects of Steam Outlet Temperature and Tube Diameter on the Cost and Size of Axial-Flow Bare-Tube Steam Generators Operating with Helium and a Steam Pressure of 2500 psia.

CONDITIONS SAME AS FOR FIG. 34, EXCEPT FOR STEAM PRESSURE AND TUBE DIAMETER (AS INDICATED)
from Fig. 35 that the tube length and cost increase with increasing steam temperature and tube diameter. On the other hand, the cycle efficiency was higher with higher steam outlet temperatures (see Table 8, Nos. VIII, IX, and X). The improved cycle efficiency reduces the fuel cycle cost and the capital charges. In general, it appears that the costs are relatively insensitive to either steam temperature or pressure, except that they seem to rise fairly rapidly for superheater outlet temperatures in
excess of 1100°F. This increase in cost actually would be more pronounced if an allowance had been made for the need to use a higher alloy tubing for the higher pressure and temperature conditions. Thus it seemed best to use a superheater outlet temperature of 1050°F and a pressure of 2500 psi as the reference condition, since this should give both near-minimum costs and a unit whose requirements are consistent with modern steam generator construction and design practice.

The effects of steam mass flow rate were investigated and the results plotted in Fig. 37. It is apparent from these curves that the cost of the tubing, shell, and thermal insulation is nearly minimum at the steam mass flow rate chosen for the reference design conditions. This is fortuitous. It should be recalled that a lower steam flow rate seemed inadvisable because of flow stability considerations at lower power outputs, and higher steam flow rates were felt to be unattractive because they would increase the tube length and hence the height of the steam generator. It appears from Fig. 37 that a steam velocity at the outlet of the boiler of 30 ft/sec represents a good over-all compromise.

Effects of Output per Unit

An important factor in the design of gas-cooled reactor power plants is the choice of the design power output and the number of steam generators to be operated in parallel. Figure 38 was prepared to show the effects of the unit output on the size and cost of steam generators for three different gas system pressures. Again the most important curves are those for the total cost of tubing, shell, and thermal insulation and for the over-all weight. Both factors increase with unit output at a rate slightly greater than linear. On the other hand, this effect would probably be at least counterbalanced by the fact that design costs and overhead would probably increase at somewhat less than a linear rate with an increase in output per unit. Thus it appears that, once a background of experience has been built up with steam generators of this type, the choice of a given output should be made on the basis of considerations other than those of steam generator costs because the steam generator cost per kilowatt of output is insensitive to unit design output.
Fig. 37. Effects of Steam Flow Rate on the Cost and Size of Axial-Flow Bare-Tube Steam Generators Operating with Helium and a Steam Pressure of 2500 psia.
Fig. 38. Effects of Power Plant Size on the Cost and Size of Axial-Flow Bare-Tube Steam Generators Operating with Helium at Reference Conditions. Steam generator output is the power plant electrical output (Mw) divided by the thermal efficiency (0.40).
U-Bend vs Straight-Tube Steam Generators

All the analysis up to this point has been concerned with straight-tube steam generators because these are easier to analyze than U-tube units. Fortunately, it is not difficult to take the results from the straight-tube calculations, apply perturbations, and establish the characteristics of U-tube steam generators. This was carried out for the reference design conditions and the results are plotted in Fig. 39 as a function of the pumping power-to-heat removal ratio. The data indicate that the U-tube unit is lighter and less expensive than the straight-tube unit for pumping power-to-heat removal ratios of up to about 1%. For any given pumping power-to-heat removal ratio the U-tube matrix height is cut to nearly half that of the corresponding straight-tube unit, a very important advantage in most installations.

Fig. 39. Comparison of Cost and Size of U-Bend and Straight-Tube Steam Generators Operating with Helium at Reference Conditions.
Effect of Fins

All the parametric study curves presented up to this point have been for bare-tube axial-flow steam generators. It was apparent from the beginning that axial fins could be applied to reduce the tube length, and this might be effective in reducing the steam generator size and cost, particularly for the larger diameter tubes. A series of calculations were carried out for fin heights of 1/8, 1/4, and 3/8 in. for both 0.5 and 1-in.-OD tubes, assuming six steel fins per inch of perimeter, a fin thermal conductivity of 16.5 Btu/hr-ft-°F, and a fin thickness of 0.050 in. The results are presented in Fig. 40. It is apparent that the fins

![Effect of Fins Diagram]

Fig. 40. Effects of Fin Height on the Cost and Size of Axial-Flow Steam Generators.
are effective in reducing the length of the tubes, but the costs actually increase for 0.5-in.-diam tubes. Increasing the fin height much beyond 1/8 in. is not worthwhile because the fin efficiency falls off rapidly with fin height. While the tube length can be reduced by using fins, the shop costs are actually higher with fins for 0.5-in.-diam tubes, and there is no saving in costs for the 1-in.-diam tubes. In view of the reduced integrity of axially finned tubes compared with bare tubes, it appears that it is probably best to use bare tubes rather than finned tubes for axial-flow steam generators.

Cross-Flow Steam Generators

While the prime reason for undertaking this study was to provide a basis for choosing steam generators for high-temperature high-pressure all-ceramic reactor systems for which the axial-flow units seemed eminently well suited, it seemed desirable to apply the same basic techniques to cross-flow units to determine their performance relative to that of the axial-flow units. In view of the complexities introduced by the many additional parameters, the investigation was limited to those whose effects were considered likely to be of major importance.

Effects of Tube Spacing

The tube matrix height for a cross-flow steam generator cannot be fixed in the same manner as for the axial-flow units. A given value of the pumping power-to-heat removal ratio can be obtained from either a short, fat or a tall, slender cross-flow steam generator of a given power output, depending on the tube length and pitch selected. In the present study, different tube spacings were therefore tried for each case to find that for minimum cost. Figure 41 shows the effects of the tube diameter on the cost and size of a 500-Mw electrical (1250 Mw thermal) unit. As in the work on axial-tube units, the reference design conditions were taken to be a pumping power-to-heat removal ratio of 0.5%, a steam pressure of 2500 psia with 520°F feedwater and 1050°F steam outlet temperatures, and a gas pressure of 500 psia with 1250°F gas inlet and 650°F gas outlet temperatures.
Fig. 41. Effects of Tube Diameter on the Cost and Size of Cross-Flow Bare-Tube Steam Generators Operating with Helium at Reference Conditions.

The data of Fig. 41 indicate that the smaller tubes give a lighter and less expensive steam generator. On the other hand, the problem of tube support becomes very difficult with 0.5- or 0.8-in. tubes, especially in large units, since the number of tubes is large and the tubes are too flimsy to support themselves. The tube support structure will certainly be more costly, if not prohibitively expensive, for the smaller tubes, and no allowance was made for this in the calculations. Thus the calculations really are applicable only to 1.25- or 1.5-in.-diam tubes, and, hence, only those larger tube sizes were included in the subsequent calculations.
Effects of Gas and Steam Conditions for the Cross-Flow Steam Generators

The effects of the gas inlet and outlet temperatures and the gas pressure on the cost and size of the reference cross-flow steam generator with 1.5-in. tubes for a pumping power-to-heat removal ratio of 0.5% are shown in Fig. 42. As in most of the previous calculations, the steam-side pressure was taken as 2500 psia with 520°F feedwater inlet and 1050°F feedwater outlet.

![Fig. 42. Effects of Gas Pressure and Temperature on the Cost and Size of Bare-Tube Cross-Flow Steam Generators Operating with Helium at Reference Conditions.](image-url)
steam outlet temperatures. These curves indicate that relatively small reductions in the cost and weight of the steam generator can be obtained by increasing the gas outlet temperature from 650 to 750°F.

The cost and weight of the steam generator decrease rapidly with increasing gas inlet temperature from 1100 to 1300°F and considerably less rapidly in the range from 1300 to 1500°F.

The cost of tubing in the present study was based on Croloy tubes, which are suitable for gas temperatures up to about 1250°F; more costly stainless steel tubes are required for gas temperatures above 1250°F. The higher cost of the stainless steel could more than offset the saving in tubing and vessel weight that can be effected at gas temperatures above 1250°F.

The effects of gas pressure were investigated for the range from 150 to 500 psia. It appears that gas pressures between 150 and 300 psia give the lowest cost cross-flow steam generators.

Effects of Power Output

The effects of power output on the shop costs of cross-flow steam generators were investigated for units giving 100-, 250-, and 500-Mw net electrical (or 250-, 625-, and 1250-Mw thermal) output. A pumping power-to-heat removal ratio of 0.5%, a gas pressure of 300 psia, a steam pressure of 2500 psia, and a tube diameter of 1.5 in. were assumed. Helium inlet and outlet temperatures were 1250 and 650°F, and the steam outlet and feedwater inlet were 1050 and 520°F, respectively. The cost data may be summarized as follows:

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This comparison indicates that the cost per 100 Mw is higher for the 500-Mw single unit than for the smaller units. An inspection of Table 9,
Table 9. Summary of the Data for Cross-Flow Bare-Tube Steam Generators Operated with Helium at Reference Conditions

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Note: All data is in units of Btu/hr-ft² of inside tube surface.
which summarizes the machine calculations for the cross-flow bare-tube steam generators, discloses that this stems from the rapid increase in the pressure vessel cost with the increase in the wall thickness required for a given gas pressure. It should be remembered that the relative cost is for shop costs only. Design costs would increase much less rapidly, and thus over-all costs per kilowatt of output would probably decrease up to unit sizes that would give serious pressure vessel fabrication and shipping problems.

**Effects of Extended Surface for Cross-Flow Steam Generators**

The effects of gas inlet and outlet temperatures and the gas pressure on the cost and size of cross-flow steam generators with circular steel fins 0.050 in. thick, 0.25 in. high, spaced six fins per inch on 1.2-in.-ID, 1.5-in.-OD Croloy tubes are presented in Fig. 43. The feedwater inlet temperature was taken as 520°F, the steam outlet temperature as 1050°F, and the steam pressure as 2500 psia. The pumping power-to-heat removal ratio was 0.5%. To facilitate comparison, the temperature and pressure conditions selected for the finned-tube cross-flow steam generators were similar to those for the bare-tube cross-flow steam generators, so that the curves in Figs. 42 and 43 are directly comparable. It is apparent from these figures that the finned tubes reduce the cost of the cross-flow steam generators almost 40%. Relatively little reduction in the cost and weight of the finned-tube cross-flow steam generators can be obtained by increasing the gas outlet temperature from 650 and 750°F. However, the cost and weight decrease rapidly if the hot gas inlet temperature is increased from 1100 to 1300°F and less rapidly above 1300°F. These effects are similar to those for the bare-tube cross-flow steam generators.

The curves in Fig. 43 indicate that a gas pressure of about 150 psia gives the lowest cost for the finned-tube cross-flow steam generators. A summary of the data for the cross-flow finned-tube steam generator is presented in Table 10. There was no significant change in the cost for tube sizes from 1.5 in. in outside diameter (at the fin roots) to 0.625 in. in outside diameter. No allowance was made in these calculations for
Table 10. Summary of the Data for Cross-Flow Circular-Finned-Tube Steam Generators Operating with Helium at Reference Conditions

<table>
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<tr>
<th>Case No.</th>
<th>Tube ID (in.)</th>
<th>Plant Power (MW)</th>
<th>Gas Pressure (psia)</th>
<th>Gas Temperature (°F)</th>
<th>Plant Gas Inlet (°F)</th>
<th>Plant Gas Outlet (°F)</th>
<th>Tube Shell Weight (lb)</th>
<th>Shell Weight (lb)</th>
<th>Relative Cost</th>
<th>Minimum Total Cost ($10^3)</th>
<th>Shell Total Weight (lb)</th>
<th>Shell Total Weight (lb)</th>
<th>Relative Cost</th>
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Fig. 43. Effects of Gas Pressure and Temperature on the Cost and Size of Finned-Tube Cross-Flow Steam Generators Operating with Helium at Reference Conditions.

The higher cost of the more complex support structures that would probably be required for the smaller tubes.

Comparison of Axial-Flow and Cross-Flow Steam Generators

The lowest cost for a cross-flow bare-tube unit for each set of conditions is listed in Table 9, along with similar data for cross-flow finned-tube and axial-flow bare-tube steam generators. The heat transfer rates per square foot of the internal surface are also included for
the axial-flow steam generator as an indication of the power density in the tube matrix.

It is apparent from the comparison that axial-tube generators can be designed with 0.5-in.-OD bare tubes to give lower cost units than can be obtained with cross-flow over either bare or finned tubes if the pressure and temperature conditions are high enough to give tube internal surface heat fluxes in excess of about 30,000 Btu/hr*ft². Where the gas pressure and the available temperature differences are not high enough to give tube internal surface heat fluxes in excess of about 30,000 Btu/hr*ft², the cross-flow over finned tubes gives the least expensive steam generator.

Effects of Reheat

The cycle efficiency is expected to be improved by 2 to 3% if a reheater is used. Thus for a given reactor there would be an increase of from 5 to 7% in the output of a reheate system as compared with a nonreheat system. As a result there would be reductions of from 5 to 7% in both the fuel-cycle costs and the capital charges for the reactor and the auxiliary equipment. Since the steam generator cost is about 10% of the cost of the complete plant, and since the incremental cost of the reactor is from 15 to 20% of the cost of the steam generator, the reductions in fuel cycle costs and reactor capital charges would more than justify the additional cost of the reheater.
6. GENERAL DESIGN CONSIDERATIONS

Many factors affecting costs other than those considered in the parametric survey should be considered in preparing a steam generator design. Geometry and space considerations are paramount in evolving a well-proportioned unit that can be fabricated readily. Static and thermal stresses, steam system flow stability and pressure drop, hot spots, pressure vessel temperature distribution, and installation problems must all be considered if each pound of steel is to be used effectively. This section presents some of the more important considerations that are difficult to express quantitatively but which are likely to be determining in the choice of a steam generator configuration.

Header Layouts

In the higher power units, the number of tubes is so great that there is an incentive to group them into bundles of 10 to 30 tubes with the headers for the bundles inside the pressure vessel. This drastically reduces the number of pressure vessel wall penetrations and should facilitate assembly. This approach depends on the thesis that leaks in tubes or tube-to-header joints will be rare, and, if a leak develops, the entire tube bundle could be blocked off at convenient points outside the pressure vessel, probably where the inlet and outlet pipes for that bundle enter the main header drums. Since up to the time of writing, no leak has been reported in some 100 steam-generator-unit operating years at the Calder Hall and Chapel Cross plants, there is reason to believe that the incidence of leaks should be sufficiently low that, at most, only a few per cent of a unit's capacity would be lost during the life of the plant.

The most important single consideration in the design of these units appears to be that they should have exceptionally high reliability. This in turn means that the design should minimize the effects of differential thermal expansion both between the tube bundles and the pressure vessel and between the individual tubes in any given tube bundle. The design must be such that, if a tube bundle is blocked off, there will not be a serious temperature irregularity in adjacent tube bundles. An examination of the
problem indicated that two steps should be taken to minimize the adverse effects of blocking off the feedwater flow to one tube bundle on the temperature distribution in adjacent bundles. First, the tube bundles should be flat rather than triangular or hexagonal so that there will be a maximum opportunity for crossflow and mixing through the dead tube bundle in order to minimize temperature irregularities. Second, the tube spacers should be designed to promote a modest amount of cross flow in the tube bundle so that the hotter stream filaments in the dead bundle will be mixed with the cooler stream filaments in the adjacent bundles. A general large-scale swirl within the pressure vessel apparently would be a desirable condition.

An ideal header layout would make use of only one basic type of tube bundle, which could be arranged inside the pressure vessel in a fashion that would use all available space to advantage. If this is not possible, the next best approach is to employ a relatively small number of basic types of tube bundle in an array that makes good use of the available space.

Several tube bundle arrangements were considered. The configuration of Fig. 44 has the disadvantage that the tube bundle cross section is a triangle rather than thin and flat and hence is less desirable from the standpoint of the temperature distribution for conditions with one tube bundle blocked out. The configuration of Fig. 45 is well suited to the use of thin, flat, tube bundles, but has the disadvantage that the insulated baffle between the boiler and superheater is hexagonal in cross section, and hence the weight of the steel plate required to resist bending stresses from the radial pressure differential might be large. The arrangement of Fig. 46 was derived from that of Fig. 45 by sliding the tube bundle layers relative to each other and by employing a somewhat more complex region in the corner zone between sextants. It is believed that the heat transfer and fluid flow would not be appreciably affected by such deviations from a perfect equilateral triangular pitch. The extra complication in geometry will appear only in the design of the hangers, and the extra design and fabrication costs may be more than compensated by the
Fig. 44. Hexagonal Arrangement of Triangular Tube Bundle for Axial-Flow U-Tube Steam Generators.
Fig. 45. Hexagonal Arrangement of Flat Tube Bundles for Axial-Flow U-Tube Steam Generators.
Fig. 46. Circular Arrangement of Flat Tube Bundles for Axial-Flow U-Tube Steam Generators.
lighter and simpler insulated baffle and by the more nearly circular envelope for the tube matrix, which should give some reduction in the diameter, weight, and cost of the pressure vessel.

In any of these arrangements the radial tube spacing on the economizer side of the baffle could be made considerably less than the circumferential spacing, while that on the superheater side could be made greater than the circumferential spacing. This would change the equivalent diameter of the gas flow passages, a desirable feature that will be discussed later.

The simplest tube header arrangement from the fabrication standpoint appears to be one using header drums 2 or 3 in. in diameter and about 15 in. long, similar to those shown in Fig. 47. The tubes could extend into these in such a way that they would pass through the drum wall at an angle, as in Fig. 47(a), or they could be bent to enter the drum radially, as in Fig. 47(b), to facilitate welding and to minimize the pressure stresses associated with the perforation of the drum. The latter arrangement might be preferable, even though it would cost more to fabricate. The tubes could not, however, be bent very much if the header drum is used to support a tube bundle from the top, because bending stresses in the tube wall will become excessive, particularly for the layout of Fig. 1.

The header drum arrangement of Fig. 47 constitutes a serious obstruction to the gas flow entering and leaving the tube matrix, even if the drums are staggered in elevation, as indicated in Fig. 1. This difficulty could be largely eliminated by employing bifurcated tubes, as shown in Fig. 9, instead of header drums. Another approach would be to bend the tubes in such a way as to spread them out in the direction of the gas flow and joggle them to open up lanes between the tubes, as indicated in Fig. 48. With this arrangement the header drums can be mounted with their axes vertical, as in Fig. 3. Either the bent-and-joggled tube arrangement of Fig. 48 or the bifurcated tube arrangement of Fig. 9 should give relatively small inlet and exit losses, even for tube matrices with close center-to-center tube spacing.

It might be suggested that it would be better to install plugs in the form of dummy tube bundles in the regions requiring special or irregular tube bundles. Aside from the desire to make the most effective possible
Fig. 47. Typical Header Drum and Tube Bundle Configurations.
Fig. 48. The Bent-and-Joggled Tube Arrangement.
use of the space available, the construction and installation of spacer plugs would not be a trivial problem, and their cost would probably be about the same as that of specially shaped tube bundles. For this reason no dummy bundles were used in the layouts of Figs. 44 and 45, and only 12 small triangular dummy bundles were used in the layout of Fig. 46.

Pressure stresses in the header drums are not too serious in the feedwater headers where the temperature is low, or in the reheater headers where the pressure differential is small, but they constitute a major design problem for the superheater outlet headers, where both the temperature and the pressure differential are high. While the problems are too complex to treat here, it is clear that the greater the ratio of tube spacing to tube diameter the less serious is the header stress problem.

**Tube Spacers**

In the long spans between headers, tube spacing transverse to the flow direction can probably be best accomplished through the use of plates about 1/16 in. thick inserted obliquely in the tube bundles, as indicated in Fig. 49(a). This arrangement should induce flow transverse to the
tubes and promote the mixing desired to minimize distortion of the temperature distribution in the vicinity of a blocked-off tube bundle. Bending the plate edges as shown would increase their stiffness under axial compression loads and at the same time provide spacing between the tube bundles if they were staggered vertically.

Another arrangement designed to avoid bending loads in the tubes is shown in Fig. 49(b). In the event that a tube bundle was blocked off, the lower ends of the tubes in that bundle might undergo almost 1 in. of differential thermal expansion relative to the adjacent tubes in the functioning bundles, and the spacers must provide for the consequent relative movement. The oblique spacer plates might be attached to the tubes by spot welding the two halves together to give a tight fit around the tubes, although difficulties with chafing and fretting at contact points might be a problem. Tack welding the spacers to the tubes might be satisfactory; this is common practice in boiler construction. Differential thermal expansion between adjacent tubes in the spans between welded spacers could be accommodated by axial column buckling of the tubes, especially since they would almost certainly have a slight initial bow. The spacer plates should be located at intervals of about 5 ft axially along the tube bundle, judging from experience derived from ART heat exchanger tests. It should be noted that the tubes would be stable relative to bowing from differential thermal expansion and that, since they would be hung in tension, they would also be stable relative to gravity loads.

**Differential Thermal Expansion**

Differential thermal expansion between the tube bundle and the pressure vessel could be accommodated through bending of the pipe extending from the tube bundle header through the pressure vessel wall. In the axial-flow U-tube configuration, this pipe would have a diameter of only about 1 in. at the feedwater inlet end and a diameter of about 2 in. at the superheater outlet. To provide sufficient flexibility, the length of this pipe would have to be about 5 ft for the feedwater connection and about 12 ft for the superheater pipe to accommodate a differential
thermal expansion of as much as 1.0 in. in the superheater region. The chart of Fig. 50 is helpful in evaluating proposed arrangements. It may prove best at the superheater outlet to bend and joggle the tubes in the fashion indicated in Fig. 48 and to run them horizontally into header drums as indicated in Fig. 3 to take advantage of the flexibility in the tubes rather than to provide a sufficiently long 2-in.-diam pipe between the bundle header and the pressure vessel wall.

Fig. 50. Effects of Tube Length and Diameter on the Deflection of Steel Tubes Loaded as Simple Cantilever Beams with a Concentrated Force on the End Sufficient to Induce a Maximum Stress of 16,000 psi. The force required is also given for tubes having a wall thickness equal to 10% of the tube outside diameter.
Reheater

Tube bundles similar to those of the boiler can be employed in the reheater. Manifolds would be placed at both the upper and lower ends, and the tubes would be straight rather than formed into the U-bends contemplated above. The tube bundle could be supported by the supply pipe at the upper end where the metal temperature would be about 750°F. Differential thermal expansion between the reheater tube bundles and the pressure vessel would be accommodated by bending and/or torsion in the horizontal pipe leading from the outlet header to the pressure vessel wall.

Baffle

Thermal insulation should be placed on the baffle between the hot gas rising upward through the superheater and reheater region and the cooler gas descending through the boiler-economizer region. This baffle should prevent appreciable gas leakage from one stream to the other, since the leakage flow would bypass the bulk of the tube matrix and hence would be ineffective in heating the steam. To avoid an awkward sliding joint and seal problem, it would be best to support this insulated baffle at its base. Differential thermal expansion between the top of the baffle and the pressure vessel wall would not be a problem, since the upper end would be free to grow vertically. In view of the fact that there would be a radial pressure differential of approximately 5 psi at the base of the baffle, it would be desirable to make it a simple cylinder. If this is done, and if the reflective insulation is designed to keep the outer wall of the baffle close to the temperature of the gas leaving the economizer, the thickness of this stressed steel wall need be only approximately 1/4 in. to accommodate the radial pressure stresses. If the barrier were made in hexagonal form, the stressed wall thickness would have to be as much as 1 in. in a CO₂ system or 0.5 in. in a helium system to resist the bending stresses from the radial pressure differential, even if the baffle shell were cooled to the gas temperature at the outlet. The radial pressure differential at the top of the stack would be
quite small and hence would not pose a problem in that region. There would be some radial differential expansion between this baffle and the pressure vessel at the top of the steam generator, but the hanger rods supporting the tube bundles could be made sufficiently flexible to accommodate the radial movement, or, if desired, adequate clearance could be provided to take care of it, since it would be less than 1/4 in.

**Pressure Vessel Cooling**

The strength of type SA212, grade B, steel falls off rapidly as the temperature is increased above about 700°F. On the other hand, there is an incentive to keep the temperature of the gas leaving the steam generator up to 650°F (see Fig. 31).

Analysis shows that the heat losses on the outside of the pressure vessel are controlling. To keep the heat losses down and ease the shield cooling load, the pressure vessel should be covered with approximately 4 in. of thermal insulation having a thermal conductivity of about 0.1 Btu/hr·ft·°F. For a 150°F air temperature in the region between the pressure vessel and the shield and a thermal-convection heat transfer coefficient of 3 Btu/hr·ft²·°F, the outer surface of the thermal insulation around the pressure vessel would be about 200°F and the heat flux from the vessel wall would be about 150 Btu/hr·ft². The temperature drop through the pressure vessel wall would be only about 5°F.

In turning to the vessel interior, two typical cases are of particular interest, that is, the straight axial tube configuration of Fig. 1 and the U-tube unit of Fig. 3. The temperature of the hot gas stream at the top of the vessel will be about 1250°F for the former and 950°F for the latter. The internal temperature differentials between the vessel wall and the main gas stream thus become about 600 and 300°F, respectively, for the two cases. The thermal conductivity of helium is about 0.15 Btu/hr·ft·°F in this temperature range. Thus a static helium layer 1.0 in. thick between two 0.060-in.-thick plates with one or two intermediate layers of stainless steel foil to inhibit thermal convection and form a simple layer of reflective insulation will give a conductance of about 3 Btu/hr·ft²·°F. In regions in which the gas temperature exceeds
the vessel temperature by 300°F, this gives a heat flux that is about six times the heat loss from the outer surface of the vessel.

While it might be possible to increase the thickness of the internal reflective insulation or to use an insulating material of low thermal conductivity, such as zirconia, it seems better to provide a separate channel for a separate gas stream between the pressure vessel and the internal thermal insulation. This arrangement can be made to give excellent temperature control so that the entire pressure envelope can be held within less than 50°F of a mean temperature during startup, shutdown, and emergency conditions. With such an arrangement, the heat flow to the channel through the thermal insulation will run about 7500 Btu/hr-ft of vessel circumference for a 50-ft-tall vessel for a U-tube steam generator. The helium velocity up through a 2.0-in.-thick cooling channel between the pressure vessel and the internal thermal insulation will have to be only about 2.5 ft/sec for a temperature rise of 50°F to carry off the heat transmitted through the reflective insulation. The corresponding velocity for a straight-tube steam generator would be about 10 ft/sec, partly because of the higher gas temperature at the top and partly because of the greater vessel height.

In the event of a blower failure, thermal convection in the cooling annulus for the U-tube unit should give a helium coolant flow of about 2 ft/sec with a gas temperature rise of 100°F. Since the steam generator would tend to overcool the gas for this "blower-out" condition, the gas temperature leaving the economizer would drop by at least 50°F, and increasing the vessel cooling gas temperature rise by 50°F should not increase the vessel temperature. Thus the temperature of both the gas in the annulus and the vessel would change very little under thermal-convection conditions for the U-tube units.

It would be awkward to provide adequate thermal-convection cooling to prevent excessive pressure vessel wall temperatures in the straight-tube steam generators, since the heat load would be four times as great and the cooling passage twice as long. The amount of gas that would bypass through the pressure vessel cooling annulus and dilute the hot gas at the top of the U-tube unit would be only 3 lb/sec, or only about 0.2% of the total gas flow through a unit for a 500-Mw plant. It is difficult to provide
adequate flow passage area for the vessel cooling gas stream under the "blower-out" condition and yet avoid an excessive gas flow rate at full power. One way to accomplish this would be to design the passage inlet as a jet pump so that orificed jets would deliver a controlled gas flow rate from a plenum at the blower outlet pressure. Jets designed to give the proper cooling annulus flow for full power operation would be more than adequate to induce a positive flow rate during startup conditions so that the pressure envelope would be maintained at a uniform temperature, with the gas system essentially isothermal. Calculations indicate that the system could be designed so that once the normal system operating temperature differentials were reached, natural thermal convection would serve to sustain an adequate flow in the event of complete failure of the blowers.

**Fabrication and Assembly**

Either shop or on-site fabrication could be employed for the construction of this type of steam generator. For power plant electrical outputs up to about 300 Mw, the diameter and weight of the pressure vessel would not be so large that it could be shipped to most locations by rail or water. Pressure vessels weighing over 300 tons have been fabricated and shipped. This is probably not far from the upper weight limit that can be handled in erection. Shop fabrication would have the advantage that the welds could be made less expensively and should be superior in quality to those made in the field. While tube damage during shipment might be a problem, it would seem that this could occur only if the unit were subjected to high accelerations. In view of the large mass of the pressure vessel, high accelerations seem most unlikely.

The obvious way to assemble the unit would be to install the tubes with the pressure vessel in a vertical position, although much headroom would be required. Special fixtures could be prepared to insert the tube bundles horizontally, but these would tend to be large and cumbersome. The tube bundles would be long and flimsy, that is, about 10 in. by 2 in. by 50 ft, although their weight would be only about 600 lb per bundle of 16 tubes.
Irrespective of whether the unit were fabricated in the shop or field, an important question would be the stage at which the tubes should be installed. The vessel could be completed, except for the hemispherical dome, before installing the tube bundles. Temporary bridge work within the vessel could be used to support the bundles until the hemispherical head had been installed. Another approach that might be used would be to place an access hatch about 30 in. in diameter in the center of the upper hemispherical head so that the pressure vessel could be completed before beginning the installation of the steam generator tubes. This would avoid having the tube bundles within the vessel during the stress-relieving heat treatment for the pressure vessel welds, although it is not clear that this heat treatment would have an adverse affect on the tube bundles, especially since the operation could be carried out with an inert gas atmosphere within the vessel.

If the tubes were installed through a central access opening at the top of the vessel, the cylindrical baffle separating the boiler and superheater could be built with one or more gaps in its perimeter. The tube bundles could then be lowered through the center hatch and mounted on a small crane installed inside the pressure vessel. Using this small overhead crane, the economizer portion of the tube bundle could be moved radially outward through the gap in the baffle and then the U-tubes could move circumferentially to their proper position straddling the baffle. This approach would permit direct attachment of the tube bundles to the hemispherical head of the vessel, and no temporary bridge work would be required. The tube bundles would be quite flexible and could be twisted during the installation to provide additional clearance for getting the headers into place. The gap in the cylindrical baffle could be closed by any one of a number of different techniques. For example, a panel containing a rugged partially prefabricated interlocking box seam could be used and the seam crimped after installation to prevent appreciable gas leakage from the superheater to the boiler. It should be remembered that in a U-tube unit with a reheater this closure could be effected before the installation of the tubes for the reheater. Thus there would be a large central hole in the tube matrix when this operation was carried out.
The stress-relieving heat treatment following the welding operations on the pressure vessel would ordinarily entail the use of a large horizontal furnace. Another approach to consider would be to apply thermal insulation to be used for the life of the reactor, choosing a material suitable for operation at the temperatures required for the stress-relieving heat treatment. The heat treatment could then be carried out using one of the main blowers operated at low speed to circulate hot gas through the pressure vessel assembly. Heat could be supplied from electrically heated banks of grids requiring about 1000 kw. These could be installed temporarily in the space provided for the reactor core. Unfortunately, after the pressure tests (following the stress relief) it would probably be necessary to strip off the thermal insulation to permit a detailed inspection for surface defects. If this approach were followed there would be an incentive to design the thermal insulation to facilitate its installation and removal.

**Flow Stability**

**Effects of Static Head**

In heat transfer matrices in which the direction of flow is either vertically upward or downward, the static head of the liquid column influences the flow stability characteristics. If the heat input to a given channel is constant, the height of the relatively high-density fluid column and the consequent static head imposed at the inlet will be directly proportional to the flow rate. The effects of this factor are shown graphically in Fig. 51 for a typical case of a high-pressure system in which the flow is upward through the channel. The bottom curve represents the relative pressure drop versus relative flow curve for an amount of preheating equal to 60% of the heat of vaporization, and includes no allowances for a static head. The effects of static head were included by applying perturbations about the 100% vapor quality point for the basic curve.

Two additional conditions were considered, that is, static heads at the 100% quality points of 40 and 100% of the relative pressure drop. The relative pressure drop at each of several values of the relative flow was then calculated by taking the relative pressure drop for the base curve.
STEAM PRESSURE, 2500 psia
PREHEATING EQUAL TO 60% OF THE LATENT
HEAT OF EVAPORIZATION
VAPOR SPECIFIC VOLUME EQUAL TO 5 TIMES
THE LIQUID SPECIFIC VOLUME

Fig. 51. Effects of Static Head on the Stability Characteristics
of Boiling Flow.
and adding to it the static head, which was taken as directly proportional to the relative flow rate. It is apparent from Fig. 51 that the allowance for the static head can be an important factor in stabilizing the flow. By the same token, if the flow is downward, the static head may have a destabilizing influence.

**Effects of Orifices at the Inlet**

The effects of orificing the inlet to the economizer are very similar and can be computed in much the same way. Figure 52 shows the results of a set of perturbations to the same base curve as that used for Fig. 51. In this instance it was assumed that orifices at the inlet to the economizer would be sized to yield pressure drops across the orifice equal to, respectively, 5 and 10 times the system pressure drop with liquid flow only. The curves were constructed simply by multiplying the relative pressure drop given by the 100% liquid curve at a given relative flow by factors of 5 and 10, respectively, and adding the resulting values to the relative pressure drop at the corresponding flow for the base curve. As in the construction of Fig. 51, the calculations were so simple that no table of calculations was necessary; the results were plotted directly as they were obtained. It is evident from Fig. 52 that orificing has a pronounced stabilizing influence and that the effect increases with the ratio of the orifice pressure drop to the pressure drop for the rest of the passage with no boiling in the liquid.

**Differences in Heat Addition to Parallel Passages**

All the previous discussion was concerned with flow in single channels. In applying these relations to heat transfer matrices with a multiplicity of channels in parallel, it is necessary to consider the effects of differences in heat addition rates between parallel channels coupled with common headers. The effects of such differences in heat addition rates can be envisioned by examining Fig. 53, which was prepared for a case representative of conditions in a modern 2400-psi once-through boiler. The base curve used was that for a gas-liquid specific volume ratio of 11, for which the preheating is equivalent to 10% of the heat of vaporization.
Fig. 52. Effects of Orificing the Inlet to the Preheater on the Stability Characteristics of Boiling Flow.

The chart was constructed simply by taking the curve for 100% heat input and sliding the 1.0 relative flow rate point along the line for 100% liquid so that in each instance the flow rate was changed by a factor equal to the change in the heat addition rate relative to the base curve.

In examining these curves it is apparent that, while there would be no difficulty with flow stability, some channels would deliver excessively superheated steam, whereas others would deliver a mixture of vapor and
Fig. 53. Effects of Heat Input Per Channel on the Pressure Drop Versus Flow Relationship for Boiling with an Amount of Preheating Equal to 10% of the Heat of Vaporization and a Vapor Specific Volume Equal to 11 Times the Liquid Specific Volume.
water. Although the flow would be stable, overheating of some of the tube walls would result in part from the higher steam temperature and in part from the increased temperature difference between the steam and the tube wall. Since the excessively heated vapor would be in the tubes with the highest heat flux, the difference in temperature between the tube wall and the steam would be greatest in the hot channels. In some designs these two effects combined could cause certain tubes to overheat 200 to 300°F.

The principal assumption made in constructing Fig. 53 was that the heat input distribution along the channels would be similar for each case and that the total amount of heat input to each channel would differ by the factor noted.

**Hot-Spot Problems**

It is highly desirable to avoid metal temperatures more than 50°F above the superheater steam outlet temperature. If this can be done, the tubes could be made of 2.5% Cr–1% Mo alloy steel. This would cut the cost of the tubing and would avoid possible troubles with chloride corrosion or dissimilar metal welds, which would be inherent in the use of stainless steel. These considerations are particularly important in view of the exceptionally high degree of reliability desired.

The average internal tube wall temperature at the superheater outlet will be only about 20°F above the steam temperature for the design shown. The heat transfer coefficient on the steam side is about 1000 Btu/hr·ft²·°F, whereas that on the gas side is only about 200 to 250 Btu/hr·ft²·°F. The temperature drop through a 0.10-in.-thick 2.5% Cr–1% Mo alloy tube wall will be only about 15°F. This gives a mean tube wall temperature of 1100°F where the hot gas from the reactor enters the superheater. Since the allowable stress for 2.5% Cr–1% Mo alloy steel at 1100°F is 4200 psi, and since the pressure differential across the tube wall is about 2000 psi, the 2.5% Cr–1% Mo alloy steel should be satisfactory if irregularities in the steam, gas flow, and temperature distributions are sufficiently small so that the peak metal temperature will be within about 20°F of the average value where the gas enters the steam generator.
In examining the system for sources of irregularity, the first factor considered was the tube diameter. Commercial tolerances on the internal diameter for 1/2-in.-OD tubes of good quality will be approximately ±0.003 in., or about 1.2% of the diameter. Since the flow will vary as the 2.6 power of the diameter for a given pressure drop, this will lead to irregularities in the flow distribution of approximately 3%. If the heat input to the tubes is uniform, this will give variations in the superheater outlet temperature of about 46°F. The variations in steam flow will, however, change the heat transfer coefficient on the water side, and the increase in the steam outlet temperature will lead to a reduction in the temperature difference between the steam and the gas of about 20% at the superheater outlet. Since the water-side heat transfer coefficient is from 5 to 10 times that for the gas side, the former effect would reduce the irregularity only about 5°F, but the latter effect should give a reduction of about 15°F. Thus the increase in the steam outlet temperature for the low-flow tube would probably be only about 25°F.

The variations in flow from one tube to another arising from differences in tube internal diameter and irregularities in flow in the outlet headers could be reduced by flow testing each assembly after installation of the outlet headers but prior to the installation of the inlet headers. Since the water velocity through the latter is low, the pressure drop in that region will be trivial compared with that in the balance of the tube. Since the dynamic head in the steam leaving the superheater tube is about 1.8 psi, differences from one tube to another because of header geometry may have an appreciable effect on the flow distribution in the bundle. The combined effects of the variations in tube inside diameter and outlet header geometry can be determined by water or air flow tests during assembly of a tube bundle, and orifices could be inserted at the inlet to the economizer portion of the tube prior to installation of the inlet header drum. If this were done, the principal factors that would introduce differences in temperature of the steam leaving the superheater would be differences in the external surface area of various tubes from one tube to another and irregularities in the gas flow distribution. The tube spacers can be designed as baffles to induce a small swirl component in the gas flow through the steam generator so that any given tube lies
in the path of a wide variety of gas stream filaments. This should help greatly to reduce the effects of the small irregularities in the gas temperature leaving the core, as well as irregularities in tube spacing in the steam generator tube matrix.

While it is difficult to estimate the combined influences of all these effects, preliminary estimates indicate that variations in superheater steam outlet temperature can be held to less than ±30°F for the design contemplated. If further study showed that excessive local tube wall temperatures were likely to be a problem, consideration could be given to reducing the reactor gas outlet temperature to 1200°F, or even 1150°F, and the effects of such a reduction on the cost and reliability of the steam generator should be examined. Thorough testing would be essential to obtain sufficient statistical data to check such analytical estimates, no matter how elegant the analysis.
7. TYPICAL DESIGN WITH BARE U-TUBES AND REHEATER

In reviewing the work on steam generators to determine how best to design a unit for an all-ceramic reactor, a number of features were seen to be of major importance. First, for the high heat fluxes obtainable, 0.50-in.-OD axial tubes give a smaller, less expensive unit than serpentine tubes in cross flow. Secondly, the straight axial tubes give an excessively tall steam generator, particularly if it is to be mounted in a pressure vessel common to the reactor; a better layout can be obtained if the tubes are bent into the form of a U, since this cuts the over-all height almost in half. This configuration is also less expensive, in part because it utilizes the volume otherwise wasted in the central stack. Further, it eases the pressure-vessel-cooling problem and has the advantage that the tubes are suspended from a zone in which the metal temperature should be under 900°F, thus greatly increasing the allowable stresses at the top of the tubes for supporting the weight loads. It can be readily adapted for the installation of a reheater, and it gives an extra degree of freedom in adjusting the heat flux distribution through changes in the tube spacing. The tubes in the economizer and the first portion of the boiler can be closely spaced to give a high gas-side heat transfer coefficient where the temperature differential available is low, while the tubes in the latter portion of the boiler and the superheater can be on an open pitch to reduce the heat transfer coefficient on the gas side where the temperature differential is large.

With the above considerations in mind, the basic U-tube configuration of Fig. 3 was chosen as the most promising of the layouts considered. A reheater operating in parallel with the latter half of the boiler and the superheater was placed in the central region, while the economizer in series with the first half of the boiler was placed in the outer annulus. Figure 46 shows a typical cross section. Hot gas from the reactor rises through the superheater and reheater central section and returns downward through the economizer and initial portion of the boiler. This arrangement appears acceptable from the water flow stability standpoint since, at the top of the U-tubes, 50% of the water by weight will be evaporated and the steam-water mixture will be 83% steam by volume; hence
the steam-water mixture will have a density nearly as low as that of saturated steam. Thus most of the stabilizing effects of vertical upward flow in the boiler will be retained. There would be no abrupt discontinuity in the water-side heat transfer coefficient between the boiling zone and the superheater, since in the initial portion of the superheater the high effective specific heat of saturated steam yields heat transfer coefficients on the water side that are almost as high as those which prevail in the boiling zone (see Fig. 2).

In choosing the design conditions, a plant electrical power output of 500 Mw was chosen as representative. Helium was chosen as the cooling gas because its use is contemplated for most of the advanced gas-cooled reactors currently projected. A helium system pressure of 500 psig was chosen because parametric studies indicated that this would minimize over-all reactor system costs. The steam conditions were chosen to be the same as those for a modern steam plant, that is, the Tennessee Valley Authority's Colbert Plant Unit No. 5. A 1250°F peak gas temperature was chosen as the highest gas temperature practical for use with 2.25% Cr-1% Mo alloy steel tubes. The gas temperature leaving the steam generator was chosen as 650°F, since Fig. 31 indicates that this is close to the value for minimum costs. A pumping power-to-heat removal ratio of 0.5% was selected because it gave both near-minimum costs and good thermal convection. To relieve the hot-spot problem where the hot gas enters the tube matrix, only 25% of the total pressure drop was taken in the reheater-superheater region.

The principal performance and dimensional data are presented in Tables 11, 12, and 13. The quantities of material required are summarized in Table 14.
Table 11. Thermodynamic Data for the Steam Cycle

\[
\text{Heat input} = 983.3 + 191.6 \times \frac{2521}{3429} = 1124.3 \text{ Btu/lb of feedwater}
\]

Gross turbine output = 509 Btu/lb
Gross thermal efficiency = 45.15%.

<table>
<thead>
<tr>
<th>Point in System</th>
<th>Temperature (^{\circ}F)</th>
<th>Pressure (\text{psia})</th>
<th>(h) Btu/lb</th>
<th>Specific Volume (\text{ft}^3/\text{lb})</th>
<th>(\Delta h) Btu/lb</th>
<th>Steam Flow (\text{lb/hr})</th>
</tr>
</thead>
<tbody>
<tr>
<td>Economizer inlet</td>
<td>520</td>
<td>2500</td>
<td>510.5</td>
<td>0.0209</td>
<td>218.9</td>
<td>3,429,600</td>
</tr>
<tr>
<td>Boiler inlet</td>
<td>667</td>
<td>2490</td>
<td>729.4</td>
<td>0.0286</td>
<td>365.7</td>
<td>3,429,600</td>
</tr>
<tr>
<td>Superheater inlet</td>
<td>666</td>
<td>2460</td>
<td>1095.1</td>
<td>0.1347</td>
<td>398.7</td>
<td>3,429,600</td>
</tr>
<tr>
<td>Superheater outlet</td>
<td>1050</td>
<td>2415</td>
<td>1493.8</td>
<td>0.3352</td>
<td>191.6</td>
<td>2,421,000</td>
</tr>
<tr>
<td>Reheater inlet</td>
<td>650</td>
<td>475</td>
<td>1330</td>
<td>1.3033</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Reheater outlet</td>
<td>1000</td>
<td>428</td>
<td>1521.6</td>
<td>1.9827</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Turbine outlet</td>
<td>91.7</td>
<td>1.5 in. Hg</td>
<td>1026</td>
<td>444.9</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 12. General Dimensional and Performance Data for a Once-Through U-Tube Axial-Flow Steam Generator with a Reheater

<table>
<thead>
<tr>
<th>Power plant electrical output, Mw</th>
<th>500</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steam generator thermal output, Btu/hr</td>
<td>(4.2 \times 10^8)</td>
</tr>
<tr>
<td>Circulating gas</td>
<td>Helium</td>
</tr>
<tr>
<td>Circulating gas pressure, psia</td>
<td>515</td>
</tr>
<tr>
<td>Circulating gas inlet temperature, (^{\circ}F)</td>
<td>1250</td>
</tr>
<tr>
<td>Circulating gas outlet temperature, (^{\circ}F)</td>
<td>650</td>
</tr>
<tr>
<td>Circulating gas flow rate, lb/sec</td>
<td>1530</td>
</tr>
<tr>
<td>Circulating gas mean density, lb/ft(^3)</td>
<td>0.137</td>
</tr>
<tr>
<td>Tube matrix height, ft</td>
<td>44</td>
</tr>
<tr>
<td>Pressure vessel height, ft</td>
<td>61</td>
</tr>
<tr>
<td>Pressure vessel outside diameter, ft</td>
<td>15</td>
</tr>
<tr>
<td>Pressure vessel wall thickness, in.</td>
<td>3.6</td>
</tr>
<tr>
<td>Pressure vessel operating temperature, (^{\circ}F)</td>
<td>650</td>
</tr>
<tr>
<td>Pressure vessel material</td>
<td>SA212, grade B, steel</td>
</tr>
<tr>
<td>Tube material</td>
<td>2.25% Cr-1% Mo alloy steel</td>
</tr>
</tbody>
</table>
Table 13. Dimensional Data on the Tube Matrix

<table>
<thead>
<tr>
<th></th>
<th>Economizer</th>
<th>Boiler</th>
<th>Superheater</th>
<th>Reheater</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of tubes</td>
<td>5190</td>
<td>5190</td>
<td>5190</td>
<td>1920</td>
</tr>
<tr>
<td>Tube outside diameter, in.</td>
<td>0.50</td>
<td>0.50</td>
<td>0.60(^a)</td>
<td>1.0</td>
</tr>
<tr>
<td>Tube inside diameter, in.</td>
<td>0.40</td>
<td>0.40</td>
<td>0.40</td>
<td>0.80</td>
</tr>
<tr>
<td>Tube length, ft</td>
<td>33</td>
<td>29(^b)</td>
<td>26</td>
<td>44</td>
</tr>
<tr>
<td>Tube spacing, in.</td>
<td>1.0</td>
<td>1.25</td>
<td>2.0</td>
<td></td>
</tr>
<tr>
<td>Total footage of tubing, ft</td>
<td>157,000</td>
<td>148,000</td>
<td>130,000</td>
<td>81,000</td>
</tr>
<tr>
<td>Total outside surface, ft(^2)</td>
<td>20,600</td>
<td>19,400</td>
<td>20,000</td>
<td>21,300</td>
</tr>
<tr>
<td>Total inside surface, ft(^2)</td>
<td>16,400</td>
<td>15,600</td>
<td>13,300</td>
<td>17,100</td>
</tr>
<tr>
<td>Total cross-sectional area of region, ft(^2)</td>
<td>32</td>
<td>50</td>
<td>47</td>
<td></td>
</tr>
<tr>
<td>Inner diameter of region, ft</td>
<td>11.7</td>
<td>8.0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Outer diameter of region, ft</td>
<td>13.5</td>
<td>11.3</td>
<td>8.0</td>
<td></td>
</tr>
</tbody>
</table>

\(^a\)The tube outside diameter near the superheater outlet would have to be 0.80 in. to satisfy stress requirements.

\(^b\)The heat transfer coefficient for boiling was taken as 5000 Btu/hr·ft\(^2\)·°F for the entire boiler length. If it were taken as 1500 for the portion of the boiler section with a steam quality above 50%, the total length of the steam generator would be less than 4% longer.

Table 14. Summary of Estimated Quantities of Material

<table>
<thead>
<tr>
<th>Item</th>
<th>Quantity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tubing</td>
<td>221,000 lb</td>
</tr>
<tr>
<td>Shell</td>
<td>430,000 lb</td>
</tr>
<tr>
<td>Piping and headers</td>
<td>100,000 lb</td>
</tr>
<tr>
<td>Number of tube joints</td>
<td>13,000</td>
</tr>
<tr>
<td>Thermal insulation</td>
<td>2,000 ft(^2)</td>
</tr>
</tbody>
</table>
8. REACTOR AND STEAM GENERATOR SYSTEM CONTROL

The previous chapters showed that a wide range of steam generator geometries and conditions would yield economical steam generators for gas-cooled reactor systems. As indicated in Chapter 6, however, there are factors other than cost that must be considered in the selection of a steam generator and its operating conditions. One of these factors is system control. The control system is a major expense item, the cost of which may, in some instances, approach the fabrication cost of the steam generator; further the control problems are so important that not only may the minimum cost steam generator not give the lowest cost, but it may be practically inacceptable from the control standpoint. The major problems and relations are outlined here, and analyses of a number of typical cases are examined to establish the design characteristics of the steam generator that appear to be desirable from the standpoint of over-all system control.

Control Requirements

Ideally, the control system for a gas-cooled reactor coupled to a steam generator should maintain both the reactor gas outlet temperature and the steam outlet temperature within rather narrow limits for a wide range of reactor power outputs. The steam temperature in particular should be kept within ±10°F to avoid distortion in the steam-turbine casing. The variations in the gas temperature leaving the reactor should be kept within ±50°F to avoid thermal stresses in structural elements. Similarly, the gas temperature leaving the steam generator to return to the reactor should be kept within a narrow temperature range. This requirement, it turns out, is not difficult to meet, since a variety of over-all system design considerations lead to a small temperature difference between the feedwater and the gas leaving the steam generator, so a relatively large proportion of the heat transfer surface area in the steam generator is in the economizer or the first portion of the boiler. Further, the large thermal inertia in the feedwater system leads to an essentially constant feedwater inlet temperature to the steam generator. Thus the
feedwater inlet and gas outlet temperatures are closely coupled and both tend to stay constant as the load is varied.

If the feedwater, steam, and gas temperatures are to remain essentially independent of power output, it is apparent that the gas and steam flow rates should be directly proportional to the power output. These characteristics are both desirable and readily obtainable with conventional control equipment. The gas flow rate can be made to vary linearly with the load by making the blower speed directly proportional to the load. Similarly, by using a venturi flow meter in the steam line to the turbine throttle, the feedwater flow rate can be controlled automatically so that it is directly proportional to the steam flow rate to the turbine, with a modulating control to maintain a constant pressure ahead of the turbine throttle. Suitably fast response times can be obtained for these basic control units.

The steam turbine output is normally controlled by throttling the turbine inlet pressure so that the temperature distribution through the turbine casing is kept constant. Thus the turbine output is directly proportional to the steam pressure at the inlet to the turbine casing, and this in turn can be regulated with a throttle valve by a governor to hold the turbine speed constant.

System Characteristics

The degree to which the characteristics of the ideal system described above can be approximated in an actual system depends heavily on the full-power operating conditions chosen. A major difficulty arises because the heat transfer coefficient varies as the 0.8 power of the gas flow rate, so the heat transfer effectiveness of the steam generator is much higher at low loads than at full loads. That is, the temperature difference between the steam leaving the superheater and the hot gas entering the steam generator will be reduced markedly on a percentage basis in going from full load to a 10% load condition. Thus it is not possible to maintain the feedwater and superheater outlet temperatures constant while at the same time keeping the gas inlet and outlet temperatures constant. At first thought it might appear that the gas flow rate could be modulated
to compensate for the changes in heat transfer effectiveness. While this can be done, elementary heat balance considerations show that the gas (or steam) inlet or outlet temperature must change as well. As will be shown later in the detailed analysis, the resulting changes may make this approach unacceptable.

The object of the following analysis is to explore several possible methods of effecting control at part load and the consequent variations in system temperature distribution so that the features of each method can be examined and its suitability appraised.

**Basic Approach Employed in the Analysis**

In setting up the relations between the major parameters, it was convenient to begin by considering a gas-cooled reactor coupled to a steam generator operating at the full design power output. A representative steady-state temperature distribution plotted against the percentage of the heat transferred from the gas to the steam at full power is shown in Fig. 54.

It is interesting to consider the effects of suddenly altering the power output of the reactor, assuming that:

1. the feedwater inlet temperature stays constant for all reactor power outputs;
2. the steam saturation temperature stays constant, that is, steam pressure stays constant, for all reactor power outputs;
3. the steam outlet temperature stays constant for all reactor power outputs, that is, the control system alters the feedwater flow rate so that it is proportional to the reactor power output; and
4. the control system changes the mass flow rates of the gas by regulating the blower speed.

After the transients have passed, the steam-side temperature distribution will stay the same as before, but a new temperature distribution similar to that shown by the dotted line in Fig. 54 will result for the gas side. The mass flow rate of the gas selected and the heat transfer characteristics of the system will affect the temperature distribution
on the gas side. In this analysis the equations have been derived to give the gas-side temperature as a function of the reactor power output, the gas flow rate, and the heat transfer characteristics of the system.

Fig. 54. Effects of Load on the Temperature Distribution in a Once-Through Steam Generator if the Feedwater Inlet and Steam Outlet Temperatures Are Fixed.
Analysis of a Once-Through Steam Generator with Only a Boiler Section

Before analyzing the complete system with the economizer, boiler, and superheater sections, as shown in Fig. 54, a simpler case with only the boiler section, as shown in Fig. 55, was considered. The analysis was then extended to steam generators with economizer, boiler, and superheater sections.

Fig. 55. Temperature Distribution for a Simple Boiler.
Assuming that the system is operating at the design power output of the reactor with the temperature distribution of Fig. 53, the steady-state heat transfer relations for the steam generator are

\[ Q = A_c G_c g \beta t_g \]  
\[ Q = aU \Delta t_L \]  

(92)

(93)

Assuming that the power output of the reactor is altered and the steam and gas flow rates are adjusted in the manner described previously, the new steady-state condition can be related to the initial steady-state condition as follows:

\[ Q' = nQ \]
\[ G'_g = X G_g \]
\[ \beta t'_g = Y \beta t_g \]
\[ B' = zB \]  

(94)

where the primes refer to the new steady-state condition.

The steady-state heat transfer relations for the steam generator at the new reactor power output become

\[ nQ = A_c G_c g (X G_g) (Y \beta t_g) \]  
\[ nQ = aU' \Delta t'_{L} \]  

(95)

(96)

From Eqs. (92) and (95)

\[ n = X Y \]  

(97)

From Eqs. (93) and (96)

\[ \frac{l}{n} = \frac{U \Delta t_{L}}{U' \Delta t'_{L}} \]  

(98)
In Eq. (98) the quantities $\frac{U}{U'}$ and $\Delta t_L' / \Delta t_L$ can be evaluated. The relation

\[
\frac{U}{U'} = \frac{R'_w + R_t + R'_g}{R_w + R_t + R_g}
\]  

(99)

can be evaluated assuming that the boiling heat transfer coefficient remains constant for all power outputs. The resistance to heat transfer on the gas side depends only on the mass flow rate of the gas if the effects of small variations in gas properties with temperature are neglected. Hence,

\[
R'_w = R_w
\]  

and

\[
R'_g = \frac{R_g}{x^{0.8}}
\]  

(100)

Substituting Eq. (100) into Eq. (99) gives

\[
\frac{U}{U'} = \frac{R_w + R_t + \frac{R_g}{x^{0.8}}}{R_w + R_t + R_g} = \frac{(R_w + R_t + R_g) + R_g \left( \frac{1}{x^{0.8}} - 1 \right)}{R_w + R_t + R_g}
\]

or

\[
\frac{U}{U'} = 1 + \frac{U}{h_g} \left( \frac{1}{x^{0.8}} - 1 \right)
\]  

(101)

Substituting the value of $X_g$ from Eq. (97) into Eq. (101) gives

\[
\frac{U}{U'} = 1 + \frac{U}{h_g} \left[ \left( \frac{Y^{0.8}}{n} \right) - 1 \right]
\]  

(102)
The logarithmic mean temperature difference for the temperature distribution shown in Fig. 55 is

\[ \Delta t_L = \frac{(\delta t + B) - B}{\ln \left( \frac{\delta t + B}{B} \right)} = \frac{\delta t}{\ln \left( 1 + \frac{\delta t}{B} \right)} . \tag{103} \]

The steady-state gas temperature distribution for the new power output is shown by the dotted line in Fig. 55. The logarithmic mean temperature difference for this new distribution is

\[ \Delta t'_L = \frac{\delta t'}{\ln \left( 1 + \frac{\delta t'}{B'} \right)} , \tag{104} \]

since, by definition,

\[ \delta t' = Y \delta t \]

and

\[ B' = zB . \]

Equation (104) becomes

\[ \Delta t'_L = \frac{Y \delta t}{\ln \left( 1 + \frac{Y \delta t}{zB} \right)} . \tag{105} \]

From Eqs. (103) and (105)

\[ \frac{\Delta t'_L}{\Delta t_L} = \frac{\ln \left( 1 + \frac{Y \delta t}{zB} \right)}{Y \ln \left( 1 + \frac{\delta t}{B} \right)} . \tag{106} \]
Substituting Eqs. (102) and (105) into Eq. (98) then gives

\[
\frac{1}{n} = \left(1 + \frac{U}{h_g} \left[\left(\frac{Y}{n}\right)^{0.8} - 1\right]\right) \frac{\ln \left(1 + \frac{Y \delta t_g}{zB}\right)}{Y \ln \left(1 + \frac{\delta t_g}{B}\right)}
\]

or

\[
1 - \left(\frac{n}{Y}\right) \left(1 + \frac{U}{h_g} \left[\left(\frac{Y}{n}\right)^{0.8} - 1\right]\right) \frac{\ln \left(1 + \frac{Y \delta t_g}{zB}\right)}{\ln \left(1 + \frac{\delta t_g}{B}\right)} = 0 \quad . (107)
\]

In Eq. (107), the quantities \( \delta t_g, B, U, \) and \( h_g \) are for the full-power condition and hence are all known for a given design. The values of \( z, Y, \) and \( n \) are unity at full power. When the power output is altered to a new power level, \( n \), the unknown quantities in Eq. (107) are \( z \) and \( Y \). If the value of \( z \) is fixed, the only unknown, \( Y \), can be evaluated from Eq. (107). The resulting change in the gas flow rate, \( X_g \), from Eq. (97), is \( X_g = n/Y \).

**Analysis of a Once-Through Steam Generator with Economizer, Boiler, and Superheater Sections**

In setting up relations between the major control parameters, it was considered that the steam generator would be operating with the temperature distribution shown in Fig. 54. The steady-state heat transfer relations for a section, \( j \), of the steam generator are

\[
Q_j = a_{j} c_{j} G_{j} \delta t_{j}
\]

\[
Q_j = a_{j} U_{j} \Delta t_{Lj}
\]

where \( j = E, B, \) or \( S \) (i.e., economizer, boiler, or superheater).
If the power output were altered, the steam and gas flow rates would be adjusted as described previously. The steady-state value of the quantities for the new power output can be related to those for the design power output as follows:

\[
Q'_j = nQ_j ,
\]

\[
G'_g = X G_g ,
\]

\[
G'_w = nG'_w ,
\]

\[
\delta t'_g = Y \delta t_g ,
\]

\[
\delta t'_w = \delta t'_w ,
\]

\[
B' = zB .
\]

The steady-state heat transfer relations for a section, \( j \), of the steam generator for the new power output of the reactor are

\[
nQ'_j = A_g c_p (X G_g) (Y \delta t_g ,)
\]

\[
nQ'_j = a'_j U'_j \Delta t'_L_j .
\]

From Eqs. (103) and (111),

\[
n = X_g Y
\]

From Eqs. (109) and (112),

\[
\frac{1}{n} = \frac{a'_j U'_j \Delta t'_L_j}{a'_j U'_j \Delta t'_L_j} .
\]
Since the total heat transfer area of the steam generator remains the same for all power outputs,

\[ a_E + a_B + a_S = a'_E + a'_B + a'_S \]

or

\[ \sum a_j = \sum a'_j \quad \text{(115)} \]

Substituting \( a'_j \) from Eq. (114) into Eq. (115),

\[ \sum a_j = \sum n a_j \frac{U_j \Delta t_{Lj}}{U'_j \Delta t'_{Lj}} \]

or

\[ \sum a_j \left( 1 - n \frac{U_j \Delta t_{Lj}}{U'_j \Delta t'_{Lj}} \right) = 0 \quad \text{(116)} \]

In Eq. (116) the term \( \frac{U_j}{U'_j} \) can be evaluated in the same manner as described previously. Hence for the boiler section,

\[ \frac{U_B}{U'_B} = 1 + \frac{U_B}{h_{gB}} \left[ \left( \frac{Y}{n} \right)^{0.8} - 1 \right] \quad \text{(117)} \]

For the economizer and superheater sections, both the gas-side and water-side resistances to heat flow vary inversely with the 0.8 power of the mass flow rate. The water flow rate was varied in the same proportion as the reactor power output during the power changes. The corresponding change in the gas flow rate was fixed by Eq. (113) as \( X_g = n/Y \). Hence the water-side and gas-side resistances to heat transfer for the economizer and superheater sections are
Making use of the relations in Eq. (118), it can be shown that the \( \frac{U_j}{U_j'} \) for the economizer and superheater sections is:

\[
\frac{U_j}{U_j'} = 1 + \frac{U_j}{h_{w_j} n^{0.8}} + \frac{U_j}{h_{g_j} \left( \frac{1}{n} \right)^{0.8}}
\]

In Fig. 54 the temperature distribution on the gas side for full-power is shown by the solid line. The logarithmic mean temperature difference in the economizer section for the full-power condition is

\[
\Delta t_{LE} = \frac{(\delta t - \delta t_{wE}) - B}{\ln \left( \frac{\delta t + B - \delta t_{wE}}{B} \right)} = \frac{\delta t_{gE} - \delta t_{wE}}{\ln \left( 1 + \frac{\delta t_{gE} - \delta t_{wE}}{B} \right)}.
\]

For the boiler section,

\[
\Delta t_{LB} = \frac{(\delta t + \delta t_{gE} + B - \delta t_{wE}) - (\delta t_{gE} + B - \delta t_{wE})}{\ln \left( \frac{\delta t_{gB} + \delta t_{gE} + B - \delta t_{wE}}{\delta t_{gE} + B - \delta t_{wE}} \right)} = \frac{\delta t_{gB}}{\ln \left( 1 + \frac{\delta t_{gB}}{\delta t_{gE} + B - \delta t_{wE}} \right)}.
\]
For the superheater section,

\[
\Delta t_{LS} = \frac{(\delta t_{gS} + \delta t_{gB} + \delta t_{gE} + B - \delta t_{wE} - \delta t_{wS}) - (\delta t_{gB} + \delta t_{gE} + B - \delta t_{wE})}{\ln\left(\frac{\delta t_{gS} + \delta t_{gB} + \delta t_{gE} + B - \delta t_{wE} - \delta t_{wS}}{\delta t_{gB} + \delta t_{gE} + B - \delta t_{wE}}\right)}
\]

\[
= \frac{\delta t_{gS} - \delta t_{wS}}{\ln\left(1 + \frac{\delta t_{gS} - \delta t_{wS}}{\delta t_{gB} + \delta t_{gE} + B - \delta t_{wE}}\right)}.
\]  

(122)

The new gas temperature distribution may be related to the original gas temperature distribution as follows:

\[
\delta t'_{gJ} = Y \delta t_{gJ}
\]  

(110d)

\[
\delta t'_{wj} = \delta t_{wj}
\]  

(110e)

\[
B' = zB
\]  

(110f)

Therefore, the expression for the logarithmic mean temperature difference for the new gas temperature distribution is similar to the expressions given by Eqs. (119), (120), and (121), except that \(\delta t'_{gJ}, \delta t'_{wj},\) and \(B'\) should be replaced by relations (110d, e, and f).

Hence, the value of \(\Delta t_{LJ}/\Delta t'_{LJ}\) for the economizer section is

\[
\frac{\Delta t_{LE}}{\Delta t'_{LE}} = \frac{\delta t_{gE} - \delta t_{wE}}{Y \delta t_{gE} - \delta t_{wE}} \frac{\ln\left(1 + \frac{Y \delta t_{gE} - \delta t_{wE}}{zB}\right)}{\ln\left(1 + \frac{\delta t_{gE} - \delta t_{wE}}{B}\right)}.
\]  

(123)

For the boiler section,

\[
\frac{\Delta t'_{LB}}{\Delta t_{LB}} = \frac{1}{Y} \frac{\ln\left(1 + \frac{Y \delta t_{gB}}{zB - \delta t_{wE}}\right)}{\ln\left(1 + \frac{\delta t_{gB}}{B - \delta t_{wE}}\right)}.
\]  

(124)
For the superheater section,

\[
\frac{\Delta t_{LS}}{\Delta t_{LS}'} = \frac{\delta t_{gS} - \delta t_{wS}}{Y \delta t_{gE} - \delta t_{wE}} \ln \left(1 + \frac{Y \delta t_{gS} - \delta t_{wS}}{Y \delta t_{gB} + Y \delta t_{gE} + zB - \delta t_{wE}}\right). 
\]  

(125)

To summarize the analysis, the fractional change, \(Y\), in the gas temperature drop for a given fractional change, \(n\), in the power output and \(z\) in the gas outlet temperature can be calculated from the equation

\[
\sum a_j \left(1 - n \frac{U_j \Delta t_{Lj}}{U_j' \Delta t_{Lj}'}\right) = 0. 
\]  

(126)

In this equation, \(a_j\) is the heat transfer area in section \(j\) of the steam generator for full-power operation, and its value is known for a given design. The term \(U_j/U_j'\) is given by Eqs. (117) and (119), and \(\Delta t_{Lj}/\Delta t_{Lj}'\) is given by Eqs. (120), (121), and (122). For convenience, these terms are summarized below for the economizer, boiler, and superheater sections.

For the economizer,

\[
\frac{U_E}{U_E'} = 1 + \frac{U_E}{h_{wE}} \left(\frac{1}{n^{0.8}} - 1\right) + \frac{U_E}{h_{gE}} \left[\left(\frac{Y}{n}\right)^{0.8} - 1\right]
\]

and

\[
\frac{\Delta t_{LE}}{\Delta t_{LE}'} = \frac{\delta t_{gE} - \delta t_{wE}}{Y \delta t_{gE} - \delta t_{wE}} \ln \left(1 + \frac{Y \delta t_{gE} - \delta t_{wE}}{zB}\right). 
\]  

(125)
For the boiler,

\[
\frac{U_B}{U_B'} = 1 + \frac{U_B}{h_gB} \left[ \left( \frac{Y}{n} \right)^{0.8} - 1 \right]
\]

and

\[
\frac{\Delta t_{LB}'}{\Delta t_{LB}} = \frac{1}{Y} \ln \left( 1 + \frac{Y \delta t_{gB}}{Y \delta t_{gE} + zB - \delta t_{wE}} \right)
\]

For the superheater,

\[
\frac{U_S}{U_S'} = 1 + \frac{U_S}{h_wS} \left( \frac{1}{n^{0.8}} - 1 \right) + \frac{U_S}{h_gS} \left[ \left( \frac{Y}{n} \right)^{0.8} - 1 \right]
\]

and

\[
\frac{\Delta t_{LS}'}{\Delta t_{LS}} = \frac{\delta t_{gS} - \delta t_{wS}}{Y \delta t_{gS} - \delta t_{wS}} \ln \left( 1 + \frac{Y \delta t_{gS} - \delta t_{wS}}{Y \delta t_{gB} + Y \delta t_{gE} + zB - \delta t_{wE}} \right)
\]

In these relations, the quantities \( B, \delta t_{gj}, \delta t_{wj}, U_j, h_{wj}, \) and \( h_{gj} \) are for the full-power output and hence are all known for a given design. The values of \( n, Y, \) and \( z \) are all unity for full-power output. When the power output is altered to a new power level \( n \), the value of \( Y \) can be evaluated from Eq. (126) for a given value of \( z \). The resulting change in the gas flow rate from Eq. (113) is \( X_g = n/Y \). Equation (126) is too complicated for hand calculation, and programming of this equation is necessary. Equation (126) reduces to Eq. (107) for a steam generator.
having only a boiling region, if the quantities $a_E$, $a_S$, $\delta t_{sE}$, $\delta t_{wE}$, $\delta t_{gs'}$, and $\delta t_{WS}$ are taken as zero.

**Calculated Data for Part-Load Conditions**

The principal objective of the study was to investigate the effects on steam generator control of the temperature differential between the hot gas entering and the steam leaving the steam generator. This matter could be investigated either by fixing the steam conditions and considering a number of different design-point hot gas temperatures or by fixing the gas conditions and considering a number of different sets of design-point steam conditions. The latter approach was chosen because it had been suggested that the steam system for test reactors might be simplified by designing for only a small amount of superheat or possibly eliminating the superheater altogether. The five design conditions chosen for study are summarized in Table 15 and Fig. 56. The feedwater inlet, the steam saturation, and the gas inlet and outlet temperatures are the same for all of these cases, but the five steam outlet temperatures considered were 1195, 1050, 1000, 900, and 800°F. The resulting temperature differences between the entering hot gas and leaving steam were 55, 200, 250, 350, and 450°F, respectively.

Figure 57 shows the effect of power output on the steam generator gas temperature drop and the gas flow rate for these five cases. It was assumed that the feedwater inlet, gas and steam outlet, and steam saturation temperatures remained at the full-power output design values, and hence the water flow rate would be directly proportional to the power output. Equation (126) was programmed for machine calculation to obtain the steady-state gas inlet temperatures as a function of the power output. The temperature data in Fig. 57 show that the larger the temperature difference between the gas inlet and steam outlet, the larger the variation in the gas inlet temperature at reduced loads. For a typical gas temperature drop through the steam generator of 600°F, these variations become quite large in going from case 1 to case 5. Much more serious is the fact that the gas flow curve for case 5 shows a reversal in direction as the power output is reduced.
Table 15. Five Typical Design Conditions for Investigating the Part-Load Characteristics of Axial-Flow Steam Generators

Gas: Helium
Gas pressure: 500 psia
Steam pressure: 2500 psia
Gas inlet temperature: 1250°F
Gas outlet temperature: 650°F
Water inlet temperature: 520°F
Ratio of pumping power to heat removal: 0.5%
Mass flow rate of steam: 222.9 lb/ft²⋅sec
Bare tubes: 0.4 in. ID, 0.5 in. OD

<table>
<thead>
<tr>
<th>Case No.</th>
<th>Steam Outlet Temperature (°F)</th>
<th>Section</th>
<th>Heat Transfer Coefficients Based on External Tube Surface (Btu/hr⋅ft²⋅°F)</th>
<th>Δt_g (°F)</th>
<th>Δt_w (°F)</th>
<th>Tube Length (ft)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>h_g h_w U</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>1195</td>
<td>Economizer</td>
<td>200 2017 171</td>
<td>122.3</td>
<td>148.0</td>
<td>62.6</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Boiler</td>
<td>204 4000 182</td>
<td>201.9</td>
<td>0</td>
<td>56.6</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Superheater</td>
<td>209 948 162</td>
<td>275.8</td>
<td>527.0</td>
<td>80.5</td>
</tr>
<tr>
<td>2</td>
<td>1050</td>
<td>Economizer</td>
<td>237 2017 191</td>
<td>134.0</td>
<td>148.0</td>
<td>53.4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Boiler</td>
<td>232 4000 204</td>
<td>220.5</td>
<td>0</td>
<td>46.0</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Superheater</td>
<td>226 1065 182</td>
<td>245.5</td>
<td>382</td>
<td>37.2</td>
</tr>
<tr>
<td>3</td>
<td>1000</td>
<td>Economizer</td>
<td>233 2017 195</td>
<td>138.2</td>
<td>148.0</td>
<td>51.0</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Boiler</td>
<td>238 4000 209</td>
<td>228.3</td>
<td>0</td>
<td>43.1</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Superheater</td>
<td>244 1128 188</td>
<td>233.5</td>
<td>332.0</td>
<td>30.1</td>
</tr>
<tr>
<td>4</td>
<td>900</td>
<td>Economizer</td>
<td>244 2017 203</td>
<td>150.4</td>
<td>148.0</td>
<td>46.7</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Boiler</td>
<td>250 4000 218</td>
<td>246.6</td>
<td>0</td>
<td>37.9</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Superheater</td>
<td>256 1324 201</td>
<td>204.0</td>
<td>232</td>
<td>19.6</td>
</tr>
<tr>
<td>5</td>
<td>800</td>
<td>Economizer</td>
<td>255 2017 211</td>
<td>161.9</td>
<td>148.0</td>
<td>42.4</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Boiler</td>
<td>262 4000 227</td>
<td>272.8</td>
<td>0</td>
<td>32.7</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Superheater</td>
<td>268 1711 215</td>
<td>165.3</td>
<td>132.0</td>
<td>11.5</td>
</tr>
</tbody>
</table>

Case 5 points up a major difficulty that is quite subtle. If the power output is to be reduced while holding the steam and gas conditions constant, except for the temperature of the hot gas entering the steam generator, in order to satisfy both heat transfer and heat balance relations it is necessary to increase the gas flow rate to drop the power from 100% down to about 65% of full load. For further reductions in power, the gas flow must be decreased. Such a reversal in control action is probably unacceptable from the control standpoint.

This conclusion is so unpleasant that a more detailed rationalization of the problem seems to be in order. If the gas flow were reduced a small amount to compensate for a small reduction in load in the region of full power, the steam outlet temperature would increase because the
Fig. 56. Steam Temperature Distribution in Axial-Flow Steam Generators for Five Typical Full-Power Output Conditions.

heat transfer coefficient would decrease less rapidly than the load. If the hot gas temperature were reduced in an effort to compensate for this, the rate at which heat was transported to the steam generator by the gas would not be sufficient to maintain the desired power output. In other words, this control approach leads to a reduction in heat transport capacity at a greater rate than that at which the heat transfer capacity is reduced, and, while it is at first hard to believe, equilibrium conditions could be attained only if the gas flow rate were increased.
TEMPERATURES FIXED:
STEAM OUTLET 1195, 1050, 1000, 900, 800 °F FOR CASES 1 TO 5, RESPECTIVELY
STEAM SATURATION, 668 °F
FEEDWATER INLET, 520 °F
GAS OUTLET, 650 °F

Fig. 57. Effects of Power Output on the Gas Inlet Temperature and Gas Flow Rate for the Five Cases of Fig. 56.

The situation would be even worse than for case 5 if the superheater were eliminated.

The large variation in the gas inlet temperature at the reduced loads could be improved if the gas outlet temperature were also allowed to vary with the reactor load. Figures 58 through 61 show the gas inlet temperature as a function of the load for each of several different values of the gas outlet temperature for cases 2, 3, 4, and 5. It is apparent from these curves that a condition can be found for each power output that will give a constant gas inlet temperature. The locus of these points was superimposed as a dotted line on both the temperature and gas flow curves. It is to be noted that in each of these curves, the gas flow condition giving a constant steam generator gas inlet temperature is such that the gas flow is directly proportional to the power output. In Figs. 58 through 61 the locus of the points on the temperature curve for the constant gas...
Temperatures Fixed:
Steam Outlet, 800 °F
Steam Saturation, 668 °F
Feedwater Inlet, 520 °F
Water Flow Rate Directly Proportional to Power Output
See Nomenclature for \(X, Y,\) and \(z\)

Fig. 58. Effects of Power Output and Gas Outlet Temperature on the Gas Temperature Drop and Flow Rate for Case No. 5.

Inlet temperature is almost the same for the five cases considered. This is shown more explicitly in Fig. 62, in which \(\eta,\) the ratio of the temperature difference between the entering hot gas and the leaving steam at part load to that at full power, was plotted against \(n\) for each different value of \(z.\) This shows that curves for the five different conditions were almost coincident for any given value of \(z.\) The curves in each group are close together for \(\eta = 1.\) Figure 63 was then prepared by cross-plotting the data of Fig. 62 for \(\eta = 1.\) This figure shows \(z,\) the fractional variation in the gas outlet to feedwater inlet temperature difference, as a function of \(n\) for constant gas inlet and steam outlet temperatures. It is apparent from this that at 10% load the variation in the gas outlet temperature will be about 70°F of the 130°F difference between the gas outlet and feedwater inlet temperatures at full-power output. The magnitude of the variation in the gas outlet temperature would be less
Fig. 59. Effects of Power Output and Gas Outlet Temperature on the Gas Temperature Drop and Flow Rate for Case No. 4.

if the initial temperature difference between the gas outlet and the feedwater inlet were smaller than 130°F.

Figure 64 shows the variations in the gas temperature and the gas flow rate with power level for a simple boiler with no economizer or superheater sections. The dotted lines on this figure show the conditions for a constant gas inlet temperature. It is apparent from these curves that the control problem would be complex if this approach were used. Some additional insight is given by Fig. 65, which was obtained from the curves in Fig. 64 to show the variation in the steam generator gas outlet temperature with the load for a constant steam generator gas inlet temperature. A comparison of Fig. 65 and Fig. 63 shows that variations in the gas outlet temperature are greater for a simple boiler than for a steam generator with economizer, boiler, and superheater sections. For instance, at 10% load the decrease in the steam generator gas exit
TEMPERATURES FIXED:
STEAM OUTLET, 1000 °F
STEAM SATURATION 668 °F
FEEDWATER INLET, 520 °F
WATER FLOW RATE DIRECTLY PROPORTIONAL TO POWER OUTPUT
SEE NOMENCLATURE FOR $X$, $Y$, AND $Z$

$z = 0.6$

$z = 0.7$

$z = 0.8$

$z = 0.9$

$z = 1.0$

Fig. 60. Effects of Power Output and Gas Outlet Temperature on the Gas Temperature Drop and Flow Rate for Case No. 3.

TEMPERATURES FIXED:
STEAM OUTLET, 1050 °F
STEAM SATURATION, 668 °F
FEEDWATER INLET, 520 °F
WATER FLOW RATE DIRECTLY PROPORTIONAL TO POWER OUTPUT
SEE NOMENCLATURE FOR $X$, $Y$, AND $Z$

$z = 0.6$

$z = 0.7$

$z = 0.8$

$z = 0.9$

$z = 1.0$

Fig. 61. Effects of Power Output and Gas Inlet Temperature on the Gas Temperature Drop and Flow Rate for Case No. 2.
Fig. 62. Effects of Power Output and Gas Outlet Temperature on the Gas Temperature Drop.

Fig. 63. Effects of Power Output on the Difference Between the Feedwater Inlet and Gas Exit Temperatures for a Constant Difference Between the Inlet Gas and Outlet Steam Temperatures for a Once-Through Boiler with Economizer and Superheater.
Fig. 64. Effects of Power Output and Gas Outlet Temperature on the Gas Temperature Drop and Flow Rate for a Simple Boiler (No Economizer and No Superheater).

Fig. 65. Effects of Power Output on the Difference Between the Feedwater Inlet and Gas Exit Temperatures for a Constant Difference Between the Inlet Gas and Outlet Steam Temperatures for a Once-Through Boiler (No Economizer and No Superheater).
temperature would be about 100°F for a simple boiler if the initial temperature difference between the gas outlet and feedwater inlet were 130°F.

Another case of interest was the effect of power output on the steam outlet temperature, assuming that the gas and feedwater inlet temperatures and the boiler pressure remain constant. Figure 66 shows the fractional variations in the gas flow rate and the gas inlet-to-steam outlet temperature difference with the power output for case 2. It is apparent from this figure that the variation in the steam outlet temperature would be large, especially at the reduced load, if the temperature difference between the hot gas inlet and the steam outlet were large at the full-power output condition.

Other Approaches to the Control Problem

Other means of control than those considered above could be employed, but they would entail substantial increases in the complexity and cost.
of the steam generator. Desuperheating the steam about half way through the superheater is commonly employed as a means of controlling coal-fired plants. If applied to steam generators of the type considered here, this would double the number of tube-to-header joints and shell penetrations, and thus would increase the cost and complexity of the system.

A number of different gas bypass arrangements can be employed to aid in the control of steam temperatures. For example, gas can be allowed to bypass the reactor so that it dilutes and cools the gas entering the steam generator. All these systems require moving parts in the hot gas stream and complicated control equipment, and they are likely to give trouble with stratification of the hot and cold streams because of inadequate mixing in the region where the cold bypass stream rejoins the hot main stream.

Fast Transients

No attempt was made to investigate fast transients in the preceding analysis. This seemed to be a secondary consideration, since steam plant operation is normally restricted to rather low rates of change of load, the limit usually being about 2% per minute. The principal exception is that the system must be designed to accept an abrupt loss in the electrical load. If this occurs in conventional plants, the load on the steam generator is reduced slowly and the steam is blown down, either to the atmosphere or through a desuperheater to the condensers. The same procedure could be followed in a nuclear plant.

While no attempt to deal with rapid changes in load can be included here, it may be noted that the transit time through a conventional steam boiler is ordinarily several minutes, whereas the transit time through the steam generator of Chapter 7 is only about 15 sec, of which about 65% is in the economizer. This implies that there may be problems in obtaining control equipment with sufficiently short response times. The system would be more closely coupled, so it is possible that the control characteristics could be improved over those of conventional plants. This might entail a control system design philosophy somewhat different from that ordinarily followed in coal-fired plants.
Appraisal of Control System Analyses

In appraising the results of this study, a number of points appear to be significant:

1. From the standpoint of the steam system design, it is desirable that the feedwater inlet temperature and the steam outlet temperature and pressure be kept constant (within ±10°F) over a wide range of loads (from 10 to 20% power to full power). This means that the steam flow will be directly proportional to the power.

2. To minimize thermal stresses in the gas system pressure envelope occasioned by load transients, the gas temperature leaving the steam generator should be kept nearly constant (within ±15°F) for load variations from 10 to 20% power to full power.

3. To minimize thermal stresses in internal parts of the reactor, steam generator, and connecting piping associated with load transients, the temperature of the hot gas entering the steam generator should be kept within a small temperature range (±40°F) for load variations from 20% power to full power.

4. To satisfy the above conditions in the types of steam generator covered in this study, the temperature of the hot gas entering the steam generator should not be more than about 150°F above the steam temperature leaving the superheater. If this is done, the inherent control characteristics of the system should be excellent, and the design of the instrumentation and control equipment required for the steam system should be straightforward.

5. The above requirement will increase the steam generator fabrication cost by about $1/kw of plant electrical output. This increase in cost should be offset in large measure by reduced costs for design, instrumentation, control equipment, and shakedown testing.

6. The requirement proposed above in item (4) has other advantages in that reducing the hot gas temperature for a given steam outlet temperature should lead to greater plant reliability, increased availability, and reduced maintenance costs.
9. TWO-PHASE PRESSURE DROP

A brief discussion of flow stability was given in Chapter 6, General Design Considerations. It has been pointed out that the shape of the relative pressure vs relative flow curves, presented in Figs. 51, 52, and 53, could be used for flow stability comparisons. A simplified model was used in deriving the two-phase pressure drop equation from which the above curves were calculated. That model included the friction and elevation components of the pressure drop, but it did not allow for the momentum or acceleration components. In the economizer and superheater sections, acceleration of the fluid is so small under normal heat input rates that its effect on pressure drop can be neglected as compared with the friction pressure drop. The momentum pressure drop could be important in the boiler section, since the velocity increases rapidly as evaporation takes place. In this section a more detailed study of the two-phase pressure drop will be presented because of its importance relative to flow stability.

Two-phase pressure drop has been studied by many investigators, and various correlations have been proposed. A general equation for predicting the two phase pressure drop gradient, as given by Owens, is:

\[ - \left( \frac{dP}{dx} \right)_B = \phi \left[ 1 + \xi \left( \frac{V_S}{V_W} - 1 \right) \right] + \frac{G^2 V_W}{g} \left( \frac{V_S}{V_W} - 1 \right) \frac{d\xi}{dx} + \frac{\sin \theta}{V_W \left[ 1 + \xi \left( \frac{V_S}{V_W} - 1 \right) \right]} \frac{dV_S}{dP} \frac{G^2}{g} \]

where

- \( V_S, V_W \) = specific volume of steam and water, respectively,
- \( \xi \) = quality of steam, lb of steam per lb of steam + water,
- \( dx \) = differential distance along the tube,
- \( \theta \) = angle with the horizontal, degrees,
- \( \phi = (1/2g)[(f_B G^2 V_W)/d_i] \), to be evaluated at boiler inlet.
\[ f_E \approx f_E = 0.184 \left( \text{Re}_E \right)^{-0.2}, \]
\[ \text{Re}_E = \frac{d \cdot v}{\mu}, \] to be evaluated at economizer outlet.

The three terms on the righthand side of Eq. (127) refer to the friction, momentum, and elevation components of the pressure drop, respectively.

The pressure gradient relations for the economizer and superheater sections are much simpler, since no momentum term is involved. That for the economizer section is

\[ \frac{dP}{dx} = - \left( \frac{dP}{dx} \right)_E = \frac{1}{2g} \frac{f_E v_G^2}{d_1} + \frac{\sin \theta}{v_w}, \tag{128} \]

where
\[ f_E = 0.184 \left( \text{Re}_E \right)^{-0.2}, \]
\[ \text{Re}_E = \frac{d \cdot v}{\mu}. \]

For the superheater section,

\[ \frac{dP}{dx} = - \left( \frac{dP}{dx} \right)_S = \frac{1}{2g} \frac{f_S v_S G^2}{d_1} + \frac{\sin \theta}{v_S}, \tag{129} \]

where
\[ f_S = 0.184 \left( \text{Re}_S \right)^{-0.2}, \]
\[ \text{Re}_S = \frac{d \cdot v}{\mu}. \]

Equations (127), (128), and (129) were programmed because hand computation of Eq. (127) would be very time-consuming. The parameters fixed for a given problem were the total tube length, the fluid inlet temperature and pressure, the tube internal diameter, the tube slope, and the heat input rate along the boiler. The machine would first calculate the length of the economizer, boiler, and superheater sections for a given flow rate from heat transfer relations and then determine the pressure drops for each section. By repeating this calculation with different flow rates over the range of interest, the pressure drop as a function of flow rate could be determined. In order to allow for a variable heat input along the tube length, the following equation for the heat input
gradient (in Btu/hr·ft) was used in the program:

\[
\frac{dQ}{dx} = \frac{a_1 + a_2x + a_3x^2}{1 + a_4x + a_5x^2}
\]

where \(a_1, a_2, \text{ etc.}\), are constants.

The friction, momentum, and elevation components of the total pressure drop were recorded separately on the output data sheet for different locations along the boiler section. Sample input and output computer data sheets are shown in Tables 16 and 17. A typical set of data from the computer calculations is plotted in Fig. 67, which shows the pressure drop as a function of flow rate for flow through a horizontal and a vertical (upflow) tube for the conditions stated. The top two curves give the total pressure drop for the tube (i.e., economizer, boiler, and superheater sections). The slopes of these two curves indicate improved stability for the upflow tube. The lower two curves are for the components given by the friction and momentum pressure drops in the boiler section. These curves are included in order to show their relative magnitudes with respect to the total pressure drop. It is apparent that, for this particular case, the momentum pressure drop in the boiler section constitutes only a small fraction of the total pressure drop, especially when compared with the upflow case. The reason for this is that a long tube has been employed; a smaller diameter tube would further increase the frictional pressure drop. The tube lengths and diameters used in the present steam generator study were such that the contributions of the static and friction pressure drops tend to be much higher than the momentum pressure drop. Therefore, neglecting the contribution from the momentum pressure drop should not seriously affect the slope of pressure drop versus flow curves obtained from the simplified analysis.
Table 16. Output Data from the Computer for Two-Phase Pressure Drop

<table>
<thead>
<tr>
<th>PROBLEM NO</th>
<th>2</th>
<th>CASE NO</th>
<th>4</th>
<th>FLUID INSIDE WATER</th>
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<td>FLOW</td>
<td>1000,0E03 LBS/HR/FT**2</td>
<td></td>
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<tr>
<td>INLET TEMPERATURE</td>
<td>520,000 DEG, FAHR.</td>
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<td></td>
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<tr>
<td>INLET PRESSURE</td>
<td>2500,000 P.S.I.</td>
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<td></td>
<td></td>
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<tr>
<td>TUBE LENGTH</td>
<td>150,000 FEET</td>
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<td></td>
</tr>
<tr>
<td>INSIDE DIAMETER</td>
<td>1,000 INCHES</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>INCLINATION</td>
<td>90,000 DEGREES</td>
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</table>

| PRESSURE DROP | 36,7734 P.S.I. |
| ECONOMIZER    | 14,7319 P.S.I. |
| BOILER        | 17,8621 P.S.I. |
| SUPERHEATER   | 4,1793 P.S.I. |

DISTANCES ARE IN FEET, PRESSURE DROPS ARE IN P.S.I.

| ECONOMIZER LENGTH | 50,3627 |
| ECONOMIZER AT DISTANCE | 12,7499 25,2499 37,7498 50,3627 |
| PRESSURE DROP | 4,0536 7,8608 11,3399 14,7319 |
| REYNOLDS NUMBER | INLET 3,4513708E05 OUTLET 4,6304110E05 |

| BOILER LENGTH | 79,8062 |
| BOILER AT DISTANCE | 13,4999 26,7498 39,9997 53,2496 66,4995 79,8062 |
| PRESSURE DROP | 3,3181 6,1148 8,8458 11,6652 14,6507 17,8621 |
| FRICTION DROP | 0,6610 1,6278 2,9111 4,5123 6,4331 8,7208 |
| MOMENTUM DROP | 0,2968 0,5885 0,8807 1,1733 1,4663 1,7654 |
| ELEVATION DROP | 2,3603 3,8984 5,0541 5,9797 6,7513 7,4244 |
| QUALITY | 0,1699 0,3364 0,5025 0,6684 0,8340 1,0000 |

| SUPERHEATER LENGTH | 19,831 |
| SUPERHEATER AT DISTANCE | 5,2500 10,2499 15,2499 19,8311 |
| PRESSURE DROP | 1,0338 2,0580 3,1244 4,1793 |

| DISTANCE | 12,7499 25,2499 37,7498 50,3627 |
| PRESSURE DROP | 4,0536 7,8608 11,3399 14,7319 |
| REYNOLDS NUMBER | INLET 3,4513708E05 OUTLET 4,6304110E05 |

<p>| BOILER AT DISTANCE | 13,4999 26,7498 39,9997 53,2496 66,4995 79,8062 |
| PRESSURE DROP | 3,3181 6,1148 8,8458 11,6652 14,6507 17,8621 |
| FRICTION DROP | 0,6610 1,6278 2,9111 4,5123 6,4331 8,7208 |
| MOMENTUM DROP | 0,2968 0,5885 0,8807 1,1733 1,4663 1,7654 |
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| QUALITY | 0,1699 0,3364 0,5025 0,6684 0,8340 1,0000 |</p>
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<th>25</th>
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Fig. 67. Two-Phase Pressure Drop as a Function of Flow.
REFERENCES

1. A. M. Perry et al., unpublished study, 1958.
3. Ibid., p. 168.
10. Ibid., Part 4, Fig. 11.1.
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