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Assessment of Material Requirements for Advanced Steam Cycle Systems (>1100°F)

J. R. DiStefano
J. H. DeVan
L. C. Fuller

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FOR THE UNITED STATES
DEPARTMENT OF ENERGY

Printed in the United States of America. Available from
National Technical Information Service
U.S. Department of Commerce
5285 Port Royal Road, Springfield, Virginia 22161
NTIS price codes—Printed Copy: A04; Microfiche A01

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ORNL/TM-10489

C-95

Metals and Ceramics Division

ASSESSMENT OF MATERIAL REQUIREMENTS FOR ADVANCED STEAM CYCLE SYSTEMS (>1100°F)

J. R. DiStefano, J. H. DeVan, and L. C. Fuller

Date Published: February 1988

Prepared for the
Assistant Secretary for Conservation
and Renewable Energy, Office of
Energy Utilization Research, Energy
Conversion and Utilization Technologies
(ECUT) Program

NOTICE: This document contains information of preliminary nature. It is subject to revision or correction and therefore does not represent a final report.

Prepared by the
OAK RIDGE NATIONAL LABORATORY
Oak Ridge, Tennessee 37831
operated by
MARTIN MARIETTA ENERGY SYSTEMS, INC.
for the
U.S. DEPARTMENT OF ENERGY
under Contract DE-AC05-84OR21400

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EDITOR'S NOTE

Although ORNL has a policy of reporting its work in SI metric units, this report uses English units. The justification is that the conventional power industry at present operates completely with English units, and reporting otherwise would lose meaning to the intended readership. To assist the reader in obtaining the SI equivalents, these are listed below for the units occurring in this report.

Property	Unit used	SI equivalent
Dimension	in.	25.4 mm
Dimension	ft	0.3048 m
Density	lb/ft^3	$16.02\ kg/m^3$
Power	Btu/h	0.2929 W
Thermal conductivity	$Btu/h\ ft^2\ {}^\circ F$	0.1441 W/m•K
Temperature	$^\circ F$	$^\circ C=(5/9)(^\circ F - 32)$
Mass	lb	0.454 kg
Pressure	psi	6898 Pa

FOREWORD AND ACKNOWLEDGMENTS

Although sufficient electric-generating capability is essential to the United States, capital requirements are unusually high and forecasting of future needs has been woefully poor recently. As a result James Schlesinger has suggested we can look forward to sequential periods of complacency followed by panic. Complicating the picture is the technological uncertainty of whether today's fossil plants can successfully meet the needs of 1990 and beyond. Power system planners generally continue to focus on conventional technologies, but they emphasize flexibility, increasing productivity of existing generating capacity, and research and development that will lead to smaller, less expensive plants with satisfactory environmental control and improved flexibility of operation. Since present plants have continued to use materials developed 30 years ago, it is in this latter area that the development of improved materials can prove particularly significant.

Increasing the pressure and temperature of a steam turbine increases cycle efficiency, and thereby decreases costs for coal, coal-handling equipment, feedwater systems, etc. However, when these increases result in the need for more expensive materials and designs, net cost savings may not result. Power plants are extremely complex, and it is quite difficult to acquire highly reliable information for plants that are yet to be designed with materials not yet developed for the intended application. Nevertheless, this is what must be done if we are to evaluate our future needs in time to be able to do something about them. Thus, although our conclusions must necessarily be tempered by the many assumptions that were required, this report examines some of the economic and technological requirements for materials in advanced conventional coal-burning plants.

We are particularly indebted to a number of individuals who provided helpful information, guidance, or critique in carrying out this study: A. P. Fraas, consultant; H. I. Bowers and P. L. Rittenhouse, ORNL; R. I. Jaffee, Electric Power Research Institute; and B. W. Roberts and D. A. Canonico, Combustion Engineering.

EXECUTIVE SUMMARY

Prior to 1973 investment decisions by utilities were generally based on past trends. However, the present climate has become highly uncertain, and decisions are now based on a complicated set of interrelated financial, regulatory, and technological considerations. Although experiences in the 1970s have made the utilities wary of overcommitting to large generating plants, there is also concern over their ability to meet electricity requirements in the next decade and beyond. Requirements for new generating capacity over the next two decades will depend primarily on electricity demand growth, but also on the rate at which aging plants are replaced with new capacity and the extent to which a region can import power from another region with excess capacity. Those anticipating a strong resurgence in electricity demand by the 1990s support continued building of large power plants, citing the economics of scale that would minimize costs over the long run. Those who believe demand growth to be more uncertain favor a strategy of flexibility including conservation and load management. Complicating the picture is the utilities' cautious attitude toward new technology. While traditionally conservative, the industry has grown particularly cautious in light of their recent experiences with nuclear power. If electricity demand growth should accelerate by the early 1990s, conventional coal-fired plants with improved efficiency and flexibility are likely to be the first choice of the utilities.

To a large extent, thermal efficiencies in conventional coal-fired plants are limited by heat losses in the thermodynamic cycle. Due to the reduced availability experienced with some of the early plants that operated with supercritical steam cycles, present plants generally operate with subcritical 2400 psi/1000°F steam conditions. Although the utility industry still seeks to increase the efficiency of its plants, the cost-effective use of advanced cycles to increase operating efficiency will require both improved designs and materials. Recent Electric Power Research Institute (EPRI) studies have recommended a supercritical 4500 psi/1100°F steam cycle based on incremental improvements to present plant designs and materials, and they have initiated a 5-year R&D program with tasks aimed at providing the necessary technology for these plants to meet utility operating requirements. The Japanese have embarked on a similar program that includes verification of the reliability and long-term behavior of various structural materials for both 4500 psi/1100°F and 5000 psi/1200°F steam conditions. The DOE AR&TD Fossil Energy Materials Program has also recently initiated an investigation of candidate alloys for advanced superheaters and reheaters for use with 5000 psi/1200°F steam conditions.

This study was performed to identify areas in which appropriate research and development by the DOE would provide the technological advancements needed (especially in the area of materials development) to facilitate acceptance of advanced steam cycle, coal-fired electric power plants. In general, advanced plant concepts used as a basis for the present study were derived from the previously mentioned EPRI studies because they reflect consideration of the most recent industry requirements including availability, on-off cycling capability, low-load operation, and fuel flexibility.

In selecting advanced steam cycle design parameters, the Tennessee Valley Authority's (TVA) Bull Run Steam Plant was used as a reference and the following generalizations as a guide:

- Increased throttle temperature at a given pressure always increases the heat energy available to do useful work.
- Increased pressure at a given superheater outlet temperature increases the average temperature at which heat is added to the working fluid.
- One or more reheats will increase the heat available to do useful work faster than it increases unavailable heat.
- Increases in throttle pressure result in rapidly increasing feed water pumping power.
- Increased throttle pressure means decreased volume flow tending toward lower internal efficiency, higher packing leakage, and heavier (less efficient) nozzles and blades.
- Increased throttle pressure increases the number of required turbine stages and ultimately may impact turbine section span problems.

Based on these and other considerations, cycle design parameters and features selected were

- 700-MW(e) plant;
- 1100 to 1400°F throttle steam temperature;

- 4500 psia throttle steam pressure;
- two reheats;
- 3-in. Hg abs exhaust pressure;
- 3600-rpm, three-section turbine; and
- nine feedwater heaters.

The PRESTO code, developed by ORNL for conventional high-pressure, superheated steam turbine cycles, was then used to measure the trend in turbine cycle and overall efficiency as a function of increasing temperature. It should be noted, however, that PRESTO was not specifically developed for advanced steam cycle conditions; and because of its limitations, the calculations were limited to pressures of 4500 psia. Calculated turbine cycle efficiencies shown below should be considered as a first approximation.

Effect of higher steam temperature and pressure on net turbine cycle heat rate and efficiency

Throttle pressure/ temperature (psia/°F)	Reheat temperature (°F/°F)	Heat rate (Btu/kWh)	Efficiency (%)
3500/1000	1000/1000	7702	44.3
4500/1100	1100/1100	7293	46.8
4500/1200	1200/1100	7150	47.7
4500/1300	1300/1100	7029	48.5
4500/1400	1400/1100	6924	49.3

Calculations based on overall efficiency and the methods and reference parameters of the Nuclear Energy Cost Data Base were then used to determine the present worth of fuel savings at each temperature level, as shown below.

Fuel savings[a] due to increased overall efficiencies in a 700-MW(e) plant with advanced steam cycles

Throttle pressure/ temperature (psia/°F)	Reheat temperature (°F/°F)	Present worth of fuel savings ($M) (30-year lifetime)
3500/1000	1000/1000	0 (Baseline)
4500/1100	1100/1100	48
4500/1200	1200/1100	65
4500/1300	1300/1100	80
4500/1400	1400/1100	92

[a]Assumptions were (1) coal costs of $1.65/million Btu, (2) plant capacity factor of 0.65, (3) coal cost real escalation rate of 1.2%/year, and (4) real cost of capital 3.8%/year.

Although there is no direct method of estimating the cost of an advanced plant because of the many design uncertainties and uncertainties as to the cost of new materials, we did assume that the present worth of fuel savings would represent the maximum allowable increase in capital cost at which there is economic benefit at the higher temperature and pressure conditions. Then, using information from the TVA Bull Run plant and making a number of assumptions relative to design and material changes as a function of increasing temperature, we calculated the weight of new or advanced materials that would be required. "Old" material, when replaced, was assumed to cost $3.90/lb [based on main steam and reheater piping costs in a 800-MW(e) coal-fired plant]. Although arbitrary, a major assumption was that new material would have the same design allowable stress at each temperature (1100 to 1400°F) as old material has at 1000°F. This latter assumption only partially offsets increases in cost due to larger pipe diameters and thicknesses as a result of increases in specific volume of the steam and increases in length of superheater and reheater tubing at the higher temperatures. The maximum allowable costs of new materials for a 700-MW(e) advanced plant with a 30-year life were thus calculated to be as follows.

Maximum allowable costs of new materials for a 700-MW(e) plant with advanced steam cycles

Throttle pressure/ temperature (psia/°F)	Reheat temperature (°F/°F)	Overall[a] plant efficiency gain percentage points	30-Year fuel savings ($M) (1986 dollars)	Weight of new material (thousands of pounds)	Maximum allowable new material cost ($/lb)
3500/1000	1000/1000				
4500/1100	1100/1100	2.1	48	1544	30
4500/1200	1200/1100	2.9	65	1902	33
4500/1300	1300/1100	3.5	80	2252	34
4500/1400	1400/1100	4.2	92	2595	35

[a]Includes effect of boiler efficiency (89%). Note that previous results in previous table included net turbine efficiency only.

In general, these calculations indicate that, with the assumptions used, increasing steam conditions from 3500 psi/1000°F to 4500 psi/1100°F would likely produce the most significant cost benefit. At a constant steam throttle pressure of 4500 psi, further increases in temperature, with their higher attendant risks, show only marginal increases in allowable material cost per pound. However, it should be remembered that (1) calculations of efficiencies, coal costs, weights of materials, and design allowable stresses required a number of arbitrary assumptions and, therefore, represent only a first approximation; (2) the effect of detailed design on design optimization of advanced plants was not considered; and (3) advanced structural materials now under development potentially have very high design allowable stresses compared with present heat exchanger materials, and this was not fully exploited in making the calculations.

Overall, the study has led to the following conclusions:

- Improved overall plant efficiencies (up to 4.2%) and lower fuel costs are associated with main steam conditions of 4500 psi and increasing temperatures from 1100 to 1400°F.

- Without further increases in steam throttle pressure above 4500 psi, increasing steam temperatures produce less significant increases in efficiency.
- If improved materials have the same strength at 1100 to 1400°F as present materials at 1000°F and cost on the order of $30/lb or less, fuel savings in 30 years will offset increases in capital costs.
- At a constant steam throttle pressure, increasing steam temperatures above 1100°F offer significant fuel savings, but they may offer no significant overall cost savings if increased weights of higher cost improved materials are required. However, there could be an advantage to the higher temperatures if, for example, very high strength materials such as ordered intermetallic alloys or structural ceramics can be used.
- Development of satisfactory boiler materials will potentially be more significant than development of improved turbine materials because capital costs are more sensitive to the cost of superheaters and reheaters, and susceptibility to liquid ash corrosion is likely to remain a major problem.
- Allowable material costs are more sensitive to allowable design stress and fuel costs than steam temperature; therefore, if the efficiency advantages of advanced systems are to be utilized, further development of higher strength materials for these applications is needed.
- Design changes required to operate at temperatures above 1200°F cannot be specified without development of more specific flow charts and further modeling of advanced plant parameters.

ASSESSMENT OF MATERIAL REQUIREMENTS FOR ADVANCED STEAM CYCLE SYSTMS (>1100°F)*

J. R. DiStefano, J. H. DeVan, and L. C. Fuller†

ABSTRACT

Thermal efficiencies of coal-fired electric-generating plants generally peaked in the 1960s and have declined since that period. Because of low costs and high demand in the 1960s, utilities had little incentive to conduct the R&D needed for plants that provided improved performance. However, this situation has changed dramatically since the 1970s, and emphasis is now on improving the efficiency of existing plants and designing smaller, more flexible, less costly plants for the new capacity that will be required. While several options for burning coal show good promise for the future, it remains likely that new capacity will be pulverized-coal plants for the remainder of this century.

This study was performed to identify areas in which appropriate materials developments would provide the advancements needed to facilitate acceptance of higher temperature and pressure, supercritical steam cycle plants that meet modern day utilities' requirements. For advanced plants that will operate beyond 1100°F/4500 psig steam conditions, new materials technologies will likely be required. The major materials problems in advanced plants are reviewed, and a spectrum of materials that might be considered are discussed. Engineering requirements and economic considerations are presented that consider the effects of higher steam temperatures and pressures on plant efficiency, fuel cost savings, and the design and economics of major plant components.

INTRODUCTION

This study was performed to identify appropriate areas in which research and development by DOE would provide industry with reliable, high-performance materials of construction primarily for advanced steam cycle coal-fired power plants. Although we did not examine emerging alternative

*Research sponsored by the Office of Energy Utilization Research, Energy Conversion and Utilization Technologies (ECUT) Program, and the Exploratory Studies Program of the Oak Ridge National Laboratory under Martin Marietta Energy Systems, Inc., contract DE-AC05-84OR21400 with the U.S. Department of Energy.

†Engineering Technology Division.

power-generating technologies in detail, they were considered in terms of likely materials applications and future U.S. electricity needs. In general, studies sponsored by the Electric Power Research Institute (EPRI) were used as the basis for definition of advanced plant concepts. A point of departure of this study from previous ones is the focus on economic benefits vs system and material requirements at steam temperatures above 1100°F. In addition, classes of materials considered for boiler and turbine system applications include those in a comparatively early developmental stage.

BACKGROUND

From the time of the first electric-generating station in 1880 to the 1960s, advancements in coal-fired power plant technology occurred at a remarkable rate. Efficiency increased eightfold, plant size increased by a factor of 20,000, and electricity costs declined correspondingly. During the past 20 years, the utility industry has continued to seek increases in efficiency and availability of their coal-fired power plants; however, several factors have worked against them. Environmental controls have become more extensive, complex, and expensive, and lower coal quality has caused increased fouling, corrosion, and erosion of materials. Thus, the past decade has seen a reversal of the historic trend of declining real cost of electricity, and this has resulted in under-utilization of generating capacity. Since 1975 cancellation of new capacity has exceeded new orders by 50,000 MW.

Despite its abundance in the United States, coal is not necessarily the most desirable fuel form for electric power generation. It contains less energy per unit mass than natural gas or oil, is not easy to transport, and a number of important environmental issues are associated with its use. These environmental issues and the need for cost reduction have become the primary driving force pushing coal utilization technology in new directions, including coal beneficiation, fluidized bed combustors, coal gasification, and liquefaction. How industry, regulators, and rate payers react to today's changing circumstances will strongly affect the type and amount of U.S. electric-generating capability in the next century. Several factors are already becoming apparent:

- Financial considerations have resulted in emphasis on nonexpansion alternatives such as life extension of existing capacity, conservation, and purchased power.
- Competition among the different methods for electricity production is growing.
- Based on the current nuclear experience, new capacity will likely rely on fossil energy well into the next century.

At a recent coal technology conference, Harry W. Colborn of the North American Electric Reliability Council reported the latest 10-year forecast of electric energy demands. He stated, "United States' utilities are currently projecting peak electric demands which, when aggregated, result in an average annual growth rate of 2.2% over the next ten years. Forecast electric energy requirements over the same period result in an average annual growth rate of 2.4%."[1] Thus, even with life-extension and conservation efforts it is likely that significant new plant capacity will be required over the next 20 years, and it currently appears that this new capacity will primarily come from constructing pulverized coal-fired plants. However, fossil utility and cogeneration plant requirements have changed considerably during the past 10 years, and it is unlikely that existing plant designs will be adequate in the future. A recent EPRI study has indicated these new plants will need operational versatility as evidenced by fuel flexibility and cycling capability, and should have improved heat rate and availability.[2]

APPROACH

To a large extent, plant thermal efficiencies are limited by heat losses in the thermodynamic cycle. However, thermal efficiencies of pulverized coal plants have been decreasing steadily since Eddystone 1, a pioneering 5000 psi, 1200/1050/1050°F supercritical double-reheat plant built in 1962. (Pressure and throttle temperature were reduced somewhat following early material problems.) After Eddystone, 3500 psi/1000°F steam was established as standard for both single- and double-reheat cycles until the mid 1970s. However, due to the reduced availability of some of the early supercritical plants, subsequent plants have retreated to subcritical

2400 psi/1000°F steam conditions. Although it is technically feasible to increase efficiency in utility and cogeneration plants by increasing steam temperature and/or pressure, new plant designs must avoid excessive capital costs and decreases in reliability to be marketable.

The cost-effective use of advanced cycles to improve operating efficiency requires both design and materials developments. In order to define practical limits for these materials requirements, it was necessary to establish a relation between advanced thermodynamic conditions (improved efficiency) and projected capital costs including the cost of materials. Thus the chapter entitled "Engineering Requirements and Economics" considers the effect of higher steam temperature and pressure on plant efficiency, fuel cost savings, and the design and economics of major plant components.

Materials development needs for advanced plants are mostly temperature dependent, and it is generally agreed that demonstrating the adequacy of materials of construction is the most important R&D need before such a plant can be specified and built with confidence. Commercial materials developed prior to the 1970s are generally those considered for near-term (≤10 years) applications. However, if incentives are sufficient to go beyond 1100°F/4500 psig steam conditions, a more advanced materials technology base will be required. With the significant advances in materials science that have occurred in the past decade, confidence exists within the materials community that such a technology can be achieved, although long-term investigations of fabricability, load-carrying properties, and corrosion resistance will be required. Major materials problems in advanced plants are reviewed; and the spectrum of materials, both commercial and developmental, that might be considered are examined. Finally, the results of this study are summarized and recommendations for future directions are provided. Although this study was primarily aimed at pulverized coal-fired power plants, progress in materials R&D is also likely to be useful to other forms of coal combustion technology being developed.

OTHER RELATED WORK

1. EPRI recently funded two studies[3,4] to determine the feasibility of advanced concepts. These studies concluded that a Phase I effort based on incremental improvements could achieve 4500 psi/1100°F steam compared with the 1970s baseline of 3500 psi/1000°F. EPRI has initiated a 5-year program to develop a Phase I system, and contracts have been let to Combustion Engineering for design and materials-related tasks on boilers and to General Electric/Toshiba for design and materials-related tasks on the turbine generator. In both cases the materials tasks consist of testing and evaluation, but do not include development of new or improved materials. The EPRI studies also postulated a more advanced Phase II system which would achieve 5000 psi/1200°F steam conditions, but would require significant materials development.

2. ORNL has initiated an investigation of candidate alloys for advanced superheaters and reheaters for use at 5000 psi/1200°F steam conditions as part of the AR&TD Fossil Energy Materials Program.

3. The DOE Office of Industrial Programs has initiated an assessment of advanced coal-fired steam cogeneration systems using fluidized bed combustors to produce 1500°F steam. This study will identify the limitations of currently available high-temperature materials for these applications.

4. A review of Japanese source material indicates rapid development of coal combustion technology and facilities in recent years. Technical issues being emphasized include verification of the reliability and long-term behavior of various alloys at high temperatures. Their plan, which is similar to that of EPRI, is to evaluate materials and components in an actual plant. Construction of a commercial plant incorporating 4500 psi/1100°F steam conditions was to begin in 1986, and in 1990 construction is to begin on a second plant to operate at 5000 psi/1200°F steam conditions.

ENGINEERING REQUIREMENTS AND ECONOMICS

IMPROVED PLANT EFFICIENCY

Advanced coal-fired plants that achieve higher efficiencies must meet several important requirements. First, no change will be tolerated that reduces the reliability of critical equipment and thus reduces plant availability. This will make introduction of the first plant considerably more difficult since any new design will be by definition relatively untried and untested. Second, coal-fired plants will likely be required to cycle with load while nuclear plants are used for base load. Low-load operation down to 15% of full load may be required as well as on-off cycling. Rapid load changing will be sought. These requirements can lead to undesirable thermal stresses on materials in the boiler and steam turbine. In addition, future plants will have to deal with environmental controls that generally reduce efficiency (and perhaps availability) and increase costs. Similarly, low coal quality and the variability in coals will reduce efficiency and availability, and increase the complexity of boiler design and operation.

In an attempt to improve heat rate and thus reduce fuel cost, steam power plant cycles have become much more complex. Elaborate heat recovery methods are included, each contributing to the overall efficiency of the cycle. In selecting turbine cycle design parameters such as turbine throttle steam pressure and temperature and reheat steam temperature, several generalizations can be made:

- Increased throttle temperature at a given pressure always increases the heat energy available to do useful work.
- Increased pressure at a given superheater outlet temperature increases the average temperature at which heat is added to the working fluid.
- One or more reheats will increase the heat available to do useful work faster than it increases unavailable heat.
- Increases in throttle pressure result in rapidly increasing feed water pumping power.
- Increased throttle pressure means decreased volume flow tending toward lower internal efficiency, higher packing leakage, and heavier (less efficient) nozzles and blades.

- Increased throttle pressure increases the number of required turbine stages and ultimately may impact turbine section span problems.

To estimate thermal efficiency improvements based on higher steam pressures and temperatures, it is necessary to model the thermodynamic performance of the turbine cycle, including high-pressure, intermediate-pressure, and low-pressure turbines; and the regenerative feedwater heaters which utilize extraction steam from the turbines to heat the condensate/feedwater stream flowing from the condenser to the steam generator. The PRESTO code,[5] developed by ORNL for conventional high-pressure, superheated steam turbine cycles, was used for this purpose. The PRESTO computer code is based on algorithms developed by the General Electric Company.[6] Univariate and bivariate polynomials were used extensively in GE's mathematical equations for turbine efficiencies and losses. The GE study was based on 24 heat rate tests on existing power plants, but no attempt was made to forecast performance of advanced steam cycles. Furthermore, the use of polynomial equations outside their relevant ranges would not be justified. Given these limitations, which were built into PRESTO, no cycles were investigated which involved pressures in excess of 4500 psia. Investigation of higher pressures would require new analysis techniques. As was previously shown in ref. 4, increasing throttle and reheat pressures along with temperatures will result in additional heat rate improvements. Other limitations and constraints of PRESTO are:

- Turbine stage group efficiencies in the 1100 to 1400°F range are extrapolated using algorithms developed for steam temperatures lower than 1100°F.
- A four-section turbine, which will likely be required for higher steam pressures and temperatures, cannot be modeled. Current industry practice is limited to three-section turbines.
- Turbine-driven feedwater pumps do not have the option of using extraction steam.
- Topping desuperheaters located in the high-pressure extraction steam lines (for feedwater heating) cannot be modeled per se. They are approximated by desuperheating sections in conventional high-pressure feedwater heaters.

- Condensate booster pumps and feedwater booster pumps in series with the main condensate and feedwater pumps cannot be modeled.

  Figure A.1 in the Appendix shows a simplified flow chart for an advanced cycle employing a four-section turbine and two topping desuperheaters.

Although resources were not available to modify the present PRESTO code, we believe that use of PRESTO is a reasonable approach to develop a first approximation of the trend in efficiency for a steam turbine cycle with the following features:

- 1100 to 1400°F throttle steam temperature;
- 4500 psia throttle steam pressure;
- two reheats (see p. 13 for explanation);
- 3-in. Hg abs exhaust pressure;
- 3600 rpm, three-section turbine; and
- nine feedwater heaters.

Figure A.2 in the Appendix shows a simplified flow chart for a PRESTO cycle used for the 1100°F case. Physical arrangements for the higher temperature cases were identical.

The calculated turbine cycle efficiencies are compared with that of current technology (3500 psia/1000°F) in the Table 1.

Table 1. Effect of higher steam temperature and pressure on heat rate and turbine cycle efficiency

Throttle pressure/ temperature (psia/°F)	Reheat temperature (°F/°F)	Heat rate (Btu/kWh)	Efficiency (%)
3500/1000	1000/1000	7702	44.3
4500/1100	1100/1100	7293	46.8
4500/1200	1200/1100	7150	47.7
4500/1300	1300/1100	7029	48.5
4500/1400	1400/1100	6924	49.3

The results indicate that turbine cycle efficiency increases significantly with increasing throttle and reheat steam temperatures, assuming that extrapolations of the mathematical relationships for turbine stage group efficiencies used in PRESTO are valid. The above efficiencies are "net" in the sense that main boiler feedwater pumping power is included. However, boiler efficiency and other station power requirements, such as that for condensate pumps, feedwater booster pumps and general plant auxiliaries, are not included.

EFFECTS ON PLANT DESIGN

The most obvious components that will be affected by higher steam pressures and temperatures are those that contact main and reheat steam: superheater and reheater heat transfer surfaces in the steam generator, main steam and hot reheat piping, and the high-pressure and intermediate-pressure turbines.

Superheater and Reheater Heat Transfer Surface

In general, at a constant combustion temperature, as temperature and pressure in the superheater and reheater increase, the amount of heat transfer surface required increases substantially. Heat transfer area is directly proportional to the amount of heat to be transferred and inversely proportional to the mean temperature difference between the combustion gases and the steam being heated (assuming all convective heat transfer). At 4500 psi and 1400°F the heat added in the superheater will be about 44% greater than at 3500 psia and 1000°F, resulting in a 44% greater heat transfer area, everything else (overall heat transfer coefficient and mass flow) being equal. If combustion gas temperatures are held constant, a lower mean temperature difference will also cause the heat transfer area to increase. On the other hand, incentive may exist to redesign the furnace to increase combustion temperatures. However, higher combustion temperatures lead to the formation of nitrogen oxides, a major air pollutant resulting from the combustion of fossil fuels. Hence, the recent trend has been toward reducing combustion temperatures to reduce air pollution.

The change in steam generator design that may be required is a complex problem to analyze. At best, superheater heat transfer area will be increased by approximately 44% at 1400°F compared to 1000°F. The corresponding increases in heat transfer area in the superheater at 1100, 1200, and 1300°F are 11, 23, and 33%. First-stage reheat pressure was raised from the 545-psi condition at the Bull Run plant to 1122 psi for the 1100 to 1400°F cycles. This high reheat pressure reduced reheater heat transfer by about 14% from the Bull Run Steam Plant cycle (Fig. A.3 in the Appendix).

Main Steam and Hot Reheat Piping

For constant mass flow and linear velocity (simplifying assumption), pipe cross-sectional flow area will vary linearly, and pipe diameter will vary as the square root of the specific volume of the fluid. In the area of interest, specific volumes of steam are as follows:

3500 psia/1000°F	0.2066 ft^3/lb
4500 psia/1100°F	0.1736 ft^3/lb
4500 psia/1200°F	0.1937 ft^3/lb
4500 psia/1300°F	0.2122 ft^3/lb
4500 psia/1400°F	0.2296 ft^3/lb

Thus, flow areas and diameters decrease at 1100 and 1200°F and increase at 1300 and 1400°F, assuming equal mass flow rates. Probably more important is the allowable design stress for the high-temperature material, which controls the pipe wall thickness. As a first approximation, if we assume the same allowable design stress and a pipe diameter that varies with the square root of the specific volume, main steam piping wall thickness will increase 18% at 1100°F, 24% at 1200°F, 30% at 1300°F, and 36% at 1400°F. Due to the higher first-stage reheater pressure, the hot reheat piping thickness increases 46% at 1100°F, 52% at 1200°F, 57% at 1300°F, and 62% at 1400°F. However, materials with higher allowable design stresses could lead to significantly thinner-walled pipes even for conventional plants.

High-Pressure and Intermediate-Pressure Turbines

Steam turbine designs today are basically the same as 20 years ago. Turbine manufacturers have attempted to improve the reliability and efficiency of existing designs rather than to design for advanced conditions. Temperatures above 1100°F will require new designs. An increase in pressure to 4500 psi will require at least one additional turbine stage.

Better materials will also be required. The usual turbine consists of four fundamental parts: the rotor which carries the blades or buckets; the stator consisting of cylinder and casing; the nozzles, generally fixed to the inside of the cylinder; and the frame or base for supporting the rotor and stator. Some of these parts will require new materials for cycles at highly elevated temperatures. For a 1400°F cycle, nozzles and blades that operate in the range of 1000 to 1400°F, sections of the cylinders and casings, the rotor, and the steam chest all will probably require improved materials. Many parts, such as the frame, bearings, accessories, lower-temperature nozzles and blades, and the lower-temperature cylinders and casings will not require changes in materials. The extent to which changes in materials will be required is not clear. At a temperature of 1400°F it is likely that at least four stages of the high-pressure turbine section would require materials not now used. The reheat turbine section would require similar changes. It is also not clear how one could by design avoid the use of a rotor made entirely of a new high-temperature material. Further, the differential thermal expansion problems resulting from a rotor of one material and nozzles and blades of another material at the lower-temperature end of the turbine section must be considered. The expansion problems with new materials and a new design of a four-section machine will likely be difficult. Design, development, and testing will require considerable lead time before actual use is contemplated. This redesign process should consider a reduction in power density per stage and an increase in the number of stages toward a more efficient blade design, although this would require longer spans. This lengthening of turbine sections might force a change to a more expensive cross-compound arrangement, although ref. 7 indicated that cross-compound turbines may not be greatly different in cost from tandem compound, and use is frequently a matter of preference rather than economics. At the present time many potential design changes resulting in small increments of performance improvement are becoming available through the development activities of the turbine manufacturers. Although these largely proprietary improvements would not be expected to produce a dramatic reduction in heat rate, efficiency gains can be realized in the present generation 3500-psi cycle plants since these gains result from design advances rather than elevated steam conditions. Many of these improvements will be verified in future turbines.

In addition, secondary effects of higher steam pressures and temperatures must be considered for components in the lower-temperature portions of the steam turbine cycle: condenser, condensate/feedwater pumps, and feedwater heaters.

Condenser

Some of the advanced concepts that have been reported tend to break old conventions in power plant design. Reference 8 points out that "the end point of a steam-turbine expansion process is limited by a combination of theoretical and practical considerations to a relatively small region of the steam chart." All other conditions being equal, increasing superheat at the used energy end point of the low-pressure turbine will decrease turbine cycle efficiency. Reference 9 states, "From an engineering standpoint the pressure of the final reheat involves mechanical and economic considerations which make it desirable to maintain the exhaust-hood temperature close to saturation. This is usually accomplished by designing for a second-stage reheat pressure which gives a turbine used-energy end point in the moisture region under normal operating conditions." Performance calculations were first made using PRESTO at steam conditions of 4500 psi and 1100/1100/1100°F through 1400/1400/1400°F. These calculations showed superheat at the exhaust of the low-pressure turbine from 12 to 132°F. This analysis, using a very low assumed overall heat transfer coefficient (10 Btu/h ft^2 °F), indicated that the "condenser" tube surface area devoted to desuperheating turbine exhaust steam could become as much as 52% of the total heat transfer surface area for the 1400°F case. However, if the heat transfer resistance offered by a combination of condensing and desuperheating is sufficiently low, a liquid film may exist at the condenser inlet which effectively combines the mechanisms of condensing and subcooling. Reference 10 would agree with this conclusion, but ref. 11 adds the caution that "if the temperature of the wall is above the saturation temperature at the prevailing pressure, there will be no condensation and the case should be treated as cooling a gas." One could well question whether this liquid film can be maintained at pressures of approximately 3-in. Hg abs at velocities of the order of 500 ft/s. Reference 7 describes the mechanism of desuperheating followed by condensing as involving a mist in the transition region, suggests that the overall heat transfer coefficient might be 15 to 20 Btu/h sq ft^2 °F rather than 10 as assumed above, and expresses more concern over possible high thermal gradients with exhaust superheat. Further calculations assuming an overall heat transfer coefficient of 20 showed much more modest surface area requirements for desuperheating. Reference 6 suggests the use of crossover feedwater heaters between the IP and LP turbines to reduce turbine exhaust superheat and raise final feedwater temperature. Application of this design feature to the Westinghouse cycle was shown in ref. 4 (page P-2). Therefore, the performance calculations were remade using an 1100°F second-stage reheat temperature for all (1100–1400°F) first-reheat temperatures. All recalculated cases had acceptable turbine exhaust conditions.

Condensate/Feedwater Pumps

Cycles employing 4500 psi throttle steam will require feedwater pressures exceeding 5500 psi. There are no known technical barriers to providing pump designs to meet these conditions; however, we note that feedwater pump operation has been a recurring problem in high-pressure, high-temperature service, particularly in the area of pump seals.

The change in pumping power calculated by PRESTO was not as anticipated. We had expected that a change from 3500 psi to 4500 psi would result in an approximately 29% increase in pumping power [~(4500-3500)/3500]. In fact, the change from a 3500 psi/1000°F cycle to a 4500 psi/1100°F cycle only resulted in an increase in pumping power of approximately 22%. Further increases in main and first-stage reheat temperatures beyond 1100 to 1200, 1300, and 1400°F resulted in increases of only 16, 10, and 6%, respectively, over the base (3500 psi) value. This lower-than-expected increase in pumping power at the higher temperatures resulted from a decrease in feedwater flow with increasing cycle efficiency.

Feedwater Heaters

Desuperheating must also be considered in the feedwater heating arrangement. The cycle described by EPRI (ref. 2, pp. 4-11) for 4500 psig 1100/1100/1100°F employs two topping desuperheaters. These are dry, noncondensing steam-to-water heat exchangers and are located in the feedwater circuit downstream of the high-pressure feedwater heaters (Fig. A.1 in the Appendix). They serve to reduce steam temperatures entering the conventional feedwater heaters and at the same time increase feedwater temperature. Reference 4 indicates that "the design is straightforward and the service is conventional" but cautions that "the high-temperature difference between steam and feedwater will require careful design for thermal shock and thermal expansion." Apparently Alsthom (France) has been using topping desuperheaters for 20 years. It would seem that for a 700-MW(e) cycle this could involve a large, expensive new component, although information received from Westinghouse (ref. 7) indicated that they are relatively small heat exchangers. The Bull Run Plant (ref. 12, p. 397) has a high pressure heater containing a large desuperheating section with a surface area approximately 20% of the condensing section. The EPRI cycle topping desuperheaters remove from 250 to 300°F of superheat. The separation of this desuperheating section from the conventional condensing feedwater heater would limit heat transfer coefficients to those obtainable when cooling a gas. Reference 13 shows a feedwater heater calculation involving desuperheating, condensing, and drain cooling in a single feedwater heater. The desuperheating overall heat transfer coefficient (78 Btu/h ft^2 °F) is about 10% of the condensing overall coefficient.

EFFECTS ON COST

It was shown (Table 1) that turbine cycle net heat rate continues to improve as temperatures are increased in 100°F increments from 1100 to 1400°F. An increasing efficiency implies savings in operating costs due to reduced consumption of coal. Using the methods and reference parameters of the Nuclear Energy Cost Data Base[14] and calculations of overall plant efficiencies, we determined the present worth of fuel savings at each cycle temperature level. Among the major assumptions were: coal costs of $1.65 per million Btu, plant capacity factor of 0.65, coal cost real escalation rate of 1.2%/year, and real cost of capital 3.8%/year. The resulting present worth of fuel cost savings, expressed in 1986 dollars, for a 700-MW(e) plant starting operation in the year 2000 and a 30-year life are shown in Table 2.

Table 2. Fuel savings due to increased overall plant efficiencies with advanced steam cycles

Throttle pressure/ temperature (psia/°F)	Reheat temperature (°F/°F)	Present worth of fuel savings[a] ($M)
3500/1000	1000/1000	0 (Baseline)
4500/1100	1100/1100	48
4500/1200	1200/1100	65
4500/1300	1300/1100	80
4500/1400	1400/1100	92

[a]30-year lifetime.

We have no direct way to estimate the capital costs of advanced plants since detailed designs are unavailable and the cost of new materials is not known. However, we did assume that the present worth of fuel savings would represent the maximum allowable increase in capital cost at which there is economic benefit with higher temperature and pressure conditions. Any positive difference between fuel cost savings and additional capital investment represents an economic benefit. Thus, the maximum allowable costs for new materials can be calculated if the amount of material to be used can be estimated.

As a first order estimate it was assumed that potential problems with the condenser, feedwater pumps, and feedwater heaters are solvable at no increase in plant cost. Lower flow rates resulting from increased turbine cycle efficiency lead to smaller components which tend to offset other cost increases. Therefore, savings in fuel cost can reasonably be allocated to increased capital expenditures for main steam and hot reheat piping, superheater and reheater heat transfer surface, and high-pressure and reheat turbine sections. To estimate the weights of high-temperature materials required for these components, we relied heavily on information in ref. 12 and operating personnel at the Bull Run Steam Plant for the base 3500-psi case. It should be noted that in some cases advanced high-temperature materials would be needed in addition to the existing material, but in other cases they replaced present materials.

Main steam piping weight was based on Bull Run data. Using the known diameter, wall thickness, weight (insulation weight was subtracted), design pressure, and specific volume, a hoop stress was calculated to determine the required wall thickness. As a first approximation we assumed that at 1100 to 1400°F a new material would have strength equal to presently used material at 1000°F. As previously mentioned, the pipe diameter was assumed to vary as the square root of the specific volume. Using 90% of the Bull Run Steam Plant pipe weight [factor to account for smaller 700-MW(e) plant size compared with 914 MW(e) for Bull Run] and the new wall thickness, we then calculated weight of new pipe. Using total costs (adjusted to 1986 dollars) and weights from the United Engineers and Constructors' Energy Economic Data Base - VI [ref. 14 for a 800-MW(e) coal-fired plant], we estimated the cost of present main steam piping to be $3.90/lb. The cost of old material that was not replaced was then subtracted from fuel cost savings to get the allowable additional cost for substituting new material. Although the 1400°F cycle required a 34% increase in main steam piping weight, this was not a significant part of the total added weight

The method used to find hot reheat piping weight was similar to that used for main steam piping. The results were influenced by the selection of 1122 psi as reheater outlet pressure compared with 551 psi for the base case. The lower specific volume decreased pipe size and wall thickness while the higher pressure increased wall thickness. The net effect was that a 1400°F cycle required a 26% increase in hot reheat piping weight but again this was not a significant part of the total added weight for 1100 to 1400°F plants.

Superheater weight calculations were complicated by the number of superheater stages in the Bull Run plant. There are four stages (horizontal, partition panel, platen, and pendant) with different tube dimensions and materials. Based on one Bull Run section, it was assumed that at 3500 psi the entire superheater would be 1.75 in. OD with a 0.375-in. wall thickness. Stress was calculated using known dimensions and design pressure. The required heat transfer surface area was assumed to vary with the change in enthalpy from above the critical temperature to the superheater outlet condition. Wall thickness of superheater tubing at 1100 to 1400°F was calculated using the higher pressure (4500 psi) and the calculated allowable stress value for 3500 psi, and with the ratio of the present plant output to the Bull Run Steam Plant output (~0.8). Weight was then calculated using the new wall thickness and new required lengths of tubing. The added weight of superheater tubing was the decisive contribution.

Bull Run employs a three-section reheater (horizontal, inlet and outlet pendant). The sections all had different dimensions and materials. It was assumed that the entire reheater would be 2.125 in. OD with a 0.148-in. wall. The known dimensions and design pressure were used to calculate the stress. Wall thickness of reheater tubing at 1100 to 1400°F was then calculated using the appropriate pressure and the calculated stress from Bull Run. The reheater heat transfer surface area was assumed to vary with change in enthalpy through the reheater. Weights were calculated using the new wall thickness and new required lengths of tubing. Although reheater weight increased from 30 to 32% for all cases, this resulted from a large increase in wall thickness due to the higher pressure. The added weight for the reheater was significant but somewhat less than the added weight required for the superheater.

New material required for the turbine was somewhat arbitrarily estimated to be 25% of the weight of the Bull Run high-pressure/intermediate-pressure unit. It is based on the assumption that more than zero but less than half of the turbine-generator material would require upgraded material. The resulting estimates of weights of new material and new pipe dimensions for each system are summarized in Table 3.

To determine the cost allowed per pound of new material we used the following relation:

$$(CP)_{nm}\, W_{nm} - (CP)_{pm}\, W_{pm} = PW_{fuel\ savings} ,$$

where

CP_{nm} = allowed $/lb for new materials to be used in advanced plant,

W_{nm} = weight of new materials to be used in advanced plant,

CP_{pm} = $/lb of present materials deleted from advanced plant,

Table 3. Summary of estimates of weights of new materials and pipe dimensions for advanced-coal fired plants (4500 psia)

	Steam temperature (°F)											
	1100			1200			1300			1400		
	Inside diam (in.)	Thickness (in.)	Weight (1000 lb)	Inside diam (in.)	Thickness (in.)	Weight (1000 lb)	Inside diam (in.)	Thickness (in.)	Weight (1000 lb)	Inside diam (in.)	Thickness (in.)	Weight (1000 lb)
Main steam piping	7.79	3.02	321	8.23	3.19	359	8.61	3.34	393	8.96	3.47	425
Hot reheat piping	18.88	2.01	253	19.56	2.09	271	20.22	2.16	290	20.84	2.22	308
Superheater tubing[a]	0.89	0.43	348	0.89	0.43	660	0.89	0.43	955	0.89	0.43	1240
Reheater tubing[a]	1.60	0.26	312	1.60	0.26	302	1.60	0.26	305	1.60	0.26	313
Turbine			311			311			311			311
TOTAL			1545			1903			2254			2597

[a]The inside diameter and wall thickness of superheater and reheater tubing remain constant for the 1100 through 1400°F cases because it was assumed that the design allowable stress was constant at each temperature, and the superheater and reheater pressures are constant.

W_{pm} = weight of present materials deleted from advanced plants,

$PW_{fuel\ savings}$ = present worth ($) of fuel savings for advanced plant from Table 2.

Resulting estimates of $/lb allowed for new material are shown in Table 4.

Table 4. Maximum allowable costs of new materials[a] for advanced plants with 30-year life

Throttle pressure/ temperature (psia/°F)	Reheat temperature (°F/°F)	Overall[b] plant efficiency gain (percentage points)	30-Year fuel savings ($M) (1986 dollars)	New material weight (thousand of pounds)	Allowed new material cost ($/lb)
3500/1000	1000/1000				
4500/1100	1100/1100	2.1	48	1544	30
4500/1200	1200/1100	2.9	65	1902	33
4500/1300	1300/1100	3.6	80	2252	34
4500/1400	1400/1100	4.2	92	2595	35

[a]It is assumed that the new materials for which these calculations apply have the same allowable design stress at 1100, 1200, 1300 or 1400°F that presently used materials have at 1000°F.

[b]Includes effect of boiler efficiency (89%). Results in Table 1 were for turbine cycle efficiency only.

Several variables can affect allowable material costs. Figure 1 shows the sensitivity of allowable cost of new materials to changes in coal cost and allowable design stress. Stress factors of 1.0, 1.5, and 2.0 represent the ratio of allowable design stress for a new material relative to that for presently used materials at 1000°F. A cost of $36/ton for coal was used for the reference case in this study. It can be seen that the allowable cost for new materials is very sensitive to both allowable design stress and the cost of fuel. Most of the increase in turbine cycle efficiency and fuel

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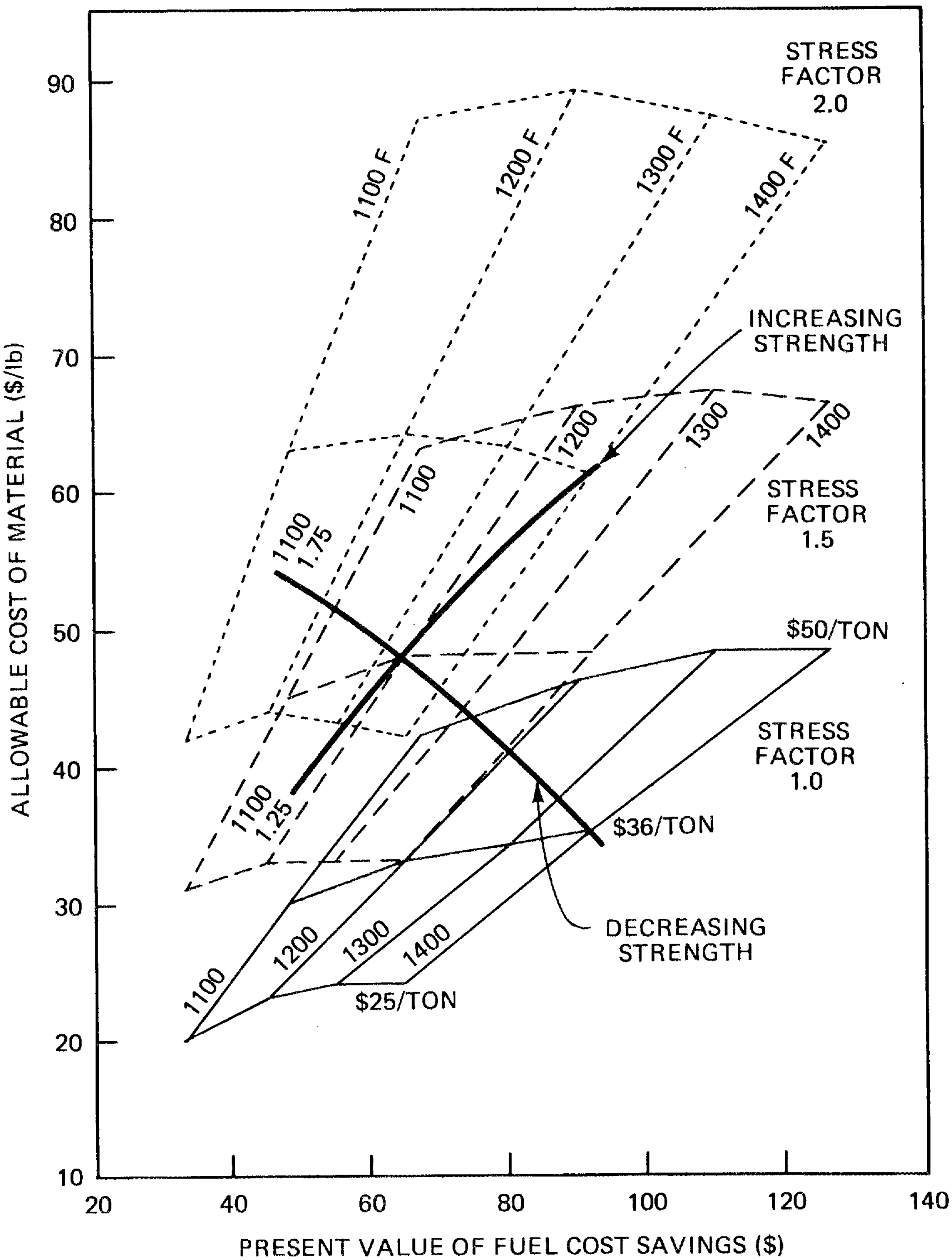

Fig. 1. Sensitivity of allowable material cost to changes in coal cost and allowable design stress.

cost savings occurs when steam conditions are changed from 3500 psi/1000°F to 4500 psi/1100°F. With no further increase in the steam throttle pressure, incremental savings in going to steam temperatures higher than 1100°F are generally offset by the increasing amounts of superheater materials required, and, therefore, at a constant allowable design stress (stress factor = 1) there appears to be no significant economic incentive for going beyond 1100°F.

Figure 1 also indicates the allowable costs for materials that have decreasing allowable design stresses with increasing temperature. For example, the solid dark line labeled "Decreasing Strength" was drawn based on the assumption that a single hypothetical material has a 75% higher allowable design stress at 1100°F, which decreases 25% for every 100°F increase in temperature, relative to the reference material at 1000°F. Table 5 summarizes the results of allowable costs from Fig. 1 and the associated incremental allowable costs as a function of temperature. Note that there is a large positive allowable incremental cost for new material at 1100°F, but it becomes slightly negative with increasing temperature. For this hypothetical material and under the assumed conditions of this study, including that of a reference coal cost of $36/ton and a steam throttle pressure of 4500 psi, there would be no economic incentive for increasing the steam temperature above 1100°F.

Table 5. Estimated allowable costs for a hypothetical material with lower strengths at higher temperatures

Steam temperature (°F)	Assumed design stress ratio	Allowable cost ($/lb)	Allowable incremental cost ($/lb)
1000	2.00	-	-
1100	1.75	54	54
1200	1.50	48	−6
1300	1.25	41	−7
1400	1.00	33	−8

Some new materials under development exhibit much higher strengths at the higher temperatures than do present materials at 1000°F. Figure 1 can also be used to estimate the allowable costs for these types of materials.

For example, if it is assumed that materials are available that have a 25% higher allowable design stress relative to 1000°F for every 100°F increase in temperature, the line labeled "Increasing Strength" can be drawn. Allowable costs based on this assumption and the incremental allowable costs are summarized in Table 6. As with constant allowable design stress, the largest incremental allowable cost for new materials occurs at 1100°F due to the large increase in turbine cycle efficiency in going from 3500 psi/1000°F to 4500 psi/1100°F. There are further, albeit smaller, economic incentives for further increases in steam temperature above 1100°F. However, there are large uncertainties at the higher temperatures relative to the effect of increasing superheater heat transfer surface on steam generator configuration and cost that must also be better evaluated before strong conclusions are warranted.

Table 6. Estimated allowable costs for materials with monotonically higher strengths (increasing stress factors >1) at higher temperatures

Steam temperature (°F)	Assumed design stress ratio	Allowable cost ($/lb)	Allowable incremental cost ($/lb)
1000	1.00	0	0
1100	1.25	38	38
1200	1.50	48	10
1300	1.75	56	8
1400	2.00	62	6

MATERIALS REQUIREMENTS AND CAPABILITIES

REVIEW OF CURRENT MATERIALS TECHNOLOGY

The requirements of boiler and turbine materials for coal-fired steam-electric plants have been of long-standing interest to the Department of Energy Advanced Research and Technology Development Fossil Energy Materials Program. A previous analysis[16] by ORNL reviewed these requirements in terms of the environmental, stress, and temperature conditions used by U.S. utilities.

Corrosion Requirements

A fundamental problem in all central station boilers concerns coal-ash corrosion. Corrosion is a strong function of ash chemistry, ash temperature, metal temperature, and tube material. Figure 2 shows the operating envelope that is considered acceptable for existing boiler steels operating with eastern bituminous coals.[16] With regard to tube wall temperature, corrosion rate increases sharply above approximately 1100°F, reaches a maximum at approximately 1300°F, and then decreases sharply before rising again steeply at approximately 1450°F. Typical corrosion rates for austenitic and ferritic steels[17] are compared in Fig. 3. The cause of rapid corrosion between 1100 and 1400°F is associated with the formation of complex iron-potassium-sodium sulfates which are molten in this temperature range. Although modern-day plants are designed to operate within the envelope shown in Fig. 2 and thus minimize problems of molten salt attack, corrosion nevertheless is the leading cause of outages in certain eastern plants. At least one utility, American Electric Power, has found it economical to retrofit their more failure-prone boilers with a composite alloy 800H tube overlaid with a coextruded cladding of 50% Ni–50% Cr. Such tubes have shown only minimal fireside corrosion damage after 12 years in plants operating at 1050°F steam temperatures (1100–1150°F maximum fireside metal interface temperature). Chromium diffusion coatings are also being used to extend corrosion lifetimes in 1000 and 1050°F plants.

Operating experience with fluidized-bed, limestone-scavenged, coal-fired steam plants is not sufficiently long term to gage fireside corrosion relative to conventionally fired boilers. However, the presence of compact, tightly adhering $CaSO_4$ deposits in these beds suggests that the effects of alkali metal sulfates are being overridden by the limestone addition. Unfortunately, data do not yet exist to predict the extent of oxidation/sulfidation that will occur through the reaction of calcium sulfate and SO_2 with boiler steels operating at steam temperatures above 1000°F. Available data do show negligible oxidation/sulfidation reactions for stainless steels operating at surface temperatures up to 1100°F and unacceptable (though not catastrophic) rates for steels at or above 1470°F.

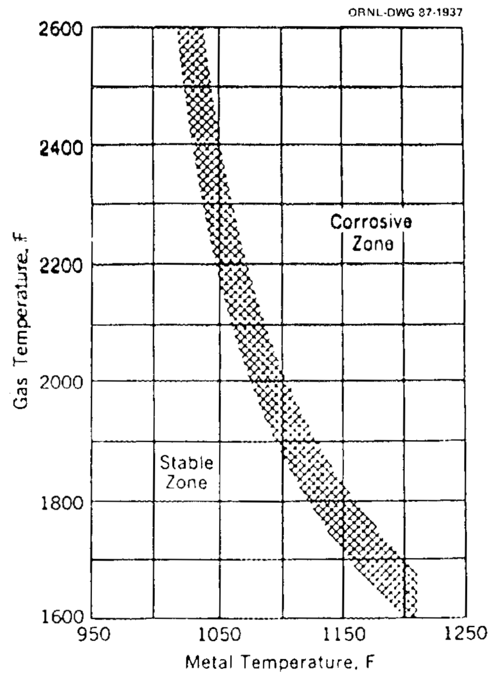

Fig. 2. Fuel-ash corrosion – stable and corrosive zones.

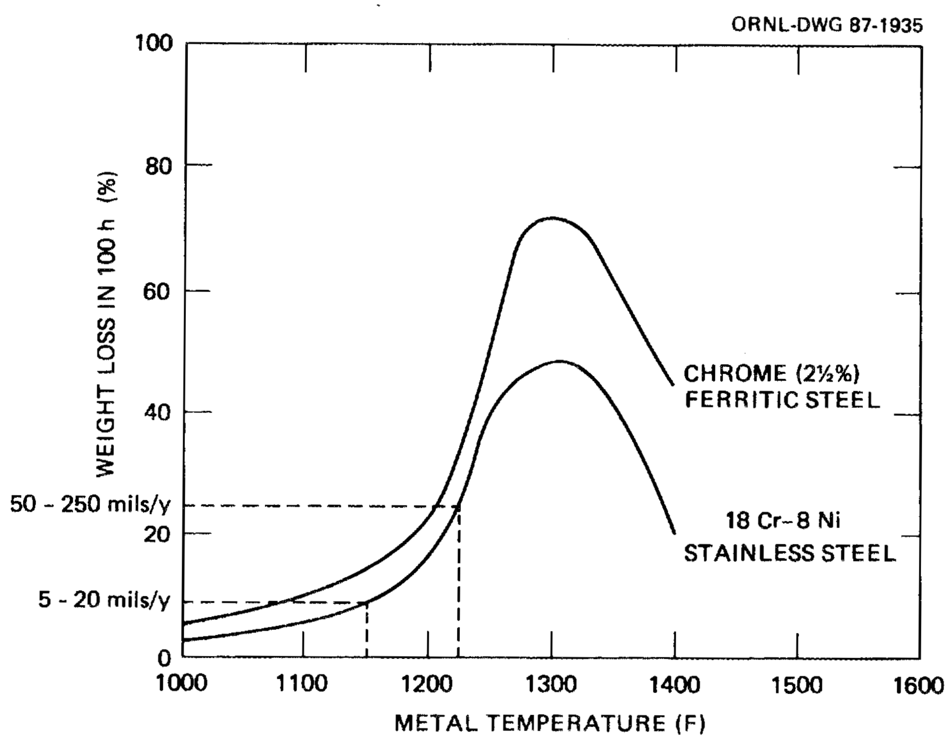

Fig. 3. Effect of temperature on corrosion rate.

Mechanical Property Requirements

The construction of boilers and pressure vessels for power plant applications is governed by the rules of the *ASME Boiler and Pressure Vessel Code* (ASME Code).[18] Material specifications and properties are included as part of these rules. For example, at temperatures below the creep range the maximum allowable stress is the lowest of (1) one-fourth of the minimum tensile strength at room temperature, (2) one-fourth of the tensile strength at temperature, (3) two-thirds of the yield strength at room temperature, or (4) two-thirds of the yield strength at temperature. At temperatures in the creep range, the ASME Code sets the maximum allowable stress values as the lowest of the following:

1. 100% of the average stress for a creep rate of 0.01%/1000 h,
2. 67% of the average stress for rupture at the end of 100,000 h, or
3. 80% of the minimum stress for rupture at the end of 100,000 h.

In addition for certain applications under Section III, Nuclear Applications, special creep-fatigue rules have been developed for austenitic stainless steels. Although these criteria are intended to apply to those materials included in the ASME Code, they can, nevertheless, provide the basis for comparing the limitations of various materials, including some of the advanced materials to be discussed later. Design rules for ceramic materials are less precise and sometimes involve knowing flaw and stress distribution in the material to calculate the probability of failure.

If wall thicknesses and diameters are to be maintained at a manageable size, allowed design stresses should nominally be 8700 psi or higher. Under such a criterion, ferritic alloys such as 2.25Cr-1Mo steel can be considered for steam service up to approximately 1000°F, austenitic stainless steels (e.g., type 316 stainless steel) to approximately 1100°F, and alloy 800H to approximately 1200°F. Unfortunately, fireside corrosion generally limits the practical application of the higher strength alloys to approximately 1100°F.

Although the strength requirements for higher pressure turbine sections are governed by the same general considerations (creep and stress-rupture) as discussed above, additional factors such as thermal and residual stresses must also be considered in setting design allowable stresses. The currently standard U.S. material for the high-pressure rotors of 1000°F turbines is a 1% chromium ferritic steel (Cr-Mo-V). There is also considerable experience, particularly in Western Europe, with martensitic 12% Cr-1% Mo steels at inlet steam temperatures up to 1050°F without rotor cooling, and up to 1100°F with rotor cooling. Above this temperature, austenitic materials must be selected.

MATERIALS REQUIREMENTS FOR ADVANCED PLANTS

Increasing steam temperatures and pressures from current 3500 psi/1000°F conditions to 4500 psi at 1100, 1200, 1300, and 1400°F improves turbine cycle efficiency and can be cost effective if suitable materials are available, as was indicated in "Engineering Requirements and Economics." In reviewing materials requirements for advanced plants that will operate at 1100°F and higher, we will not only consider the materials developments already under way in DOE fossil energy and conservation programs but also the more unconventional materials developments that may be needed to exploit the full potential of the supercritical steam cycle.

Because of economic incentives, the engineering feasibility of steam cycle systems operating up to 1200°F has already drawn some industrial and government funding, as was previously discussed. Current materials developments in the United States and Japan are generally centered around high-strength Fe-Ni-Cr alloys for both turbine and boiler applications. However, a major hurdle confronting the use of these materials in conventionally fired boilers is their susceptibility to coal ash attack between 1200 and 1400°F. Their ability to tolerate high-sulfur coals in this temperature range is very likely contingent on the development of some form of surface protection. Although cladding schemes appear workable from the metallurgical and corrosion standpoints, present concepts are relatively costly. Coal cleaning may ultimately provide a way around the sulfur problem, but such fossil feedstocks may be more profitably exploited through alternative energy concepts than through pulverized coal boilers. Alternative combustion approaches, such as limestone-scavenged fluidized beds, appear to hold significant promise for reducing coal ash corrosion effects, but there are no data at present by which to judge the corrosion characteristics of austenitic stainless steels in fluidized beds at 1200 to 1400°F surface temperatures.

Steam Generator Materials

Programs are under way in the United States, Japan, and Europe to demonstrate a materials capability for operating conventional coal-fired supercritical, double-reheat units with steam conditions of 4500 psi at 1100°F. Such plants will utilize 300 series stainless steels for sections above about 1000°F and will probably require some form of surface treatment to inhibit fireside corrosion (e.g., chromium diffusion coatings). The DOE AR&TD Fossil Energy Program has recently initiated a superheater/reheater materials development program aimed at achieving 5000 psi/1200°F steam conditions using austenitic stainless steels as a base system. The approach is to achieve the desired material properties through microstructural tailoring procedures that have evolved at ORNL over the past 20 years to improve the radiation damage resistance of 300 series steels at higher neutron fluences.[19] Although the program is relatively new and mechanical properties data are relatively short term, the enhancement in creep rupture properties shown by these developmental alloys over the best commercial steels has been remarkable. As Fig. 4 shows, the stress-rupture strengths in ≤3000 h tests are considerably better than Japanese steels being developed for 1200°F steam service (discussed below) and may be approaching the strength of the superalloy Inconel 617 (and Ni_3Al) at 1300°F (see also Table 7). The alloys are being developed to meet a target design stress of about 8700 psi at 1300°F which appears achievable on the basis of the current extrapolations (see Fig. 5).

To achieve adequate fireside corrosion resistance under such conditions in conventional pulverized coal plants will also require the development of improved materials or protective claddings or coatings. Diffusion coatings of chromium have proved effective at approximately 1100°F, but the thermal treatments required for their application work against those used to optimize microstructural properties. As shown in Fig. 6, nickel-based alloys containing ≥30% chromium have performed well in simulated fuel ash tests conducted by the Japanese at 1300°F and can be applied as a coextruded cladding. Such composite tubes are relatively expensive, however, and a less expensive approach remains a major development goal. Rehn at Foster Wheeler Development Corporation has reported some improvement in fireside corrosion resistance of some developmental Fe-Ni-Cr alloys containing Al, Si, and Mn additions.[20]

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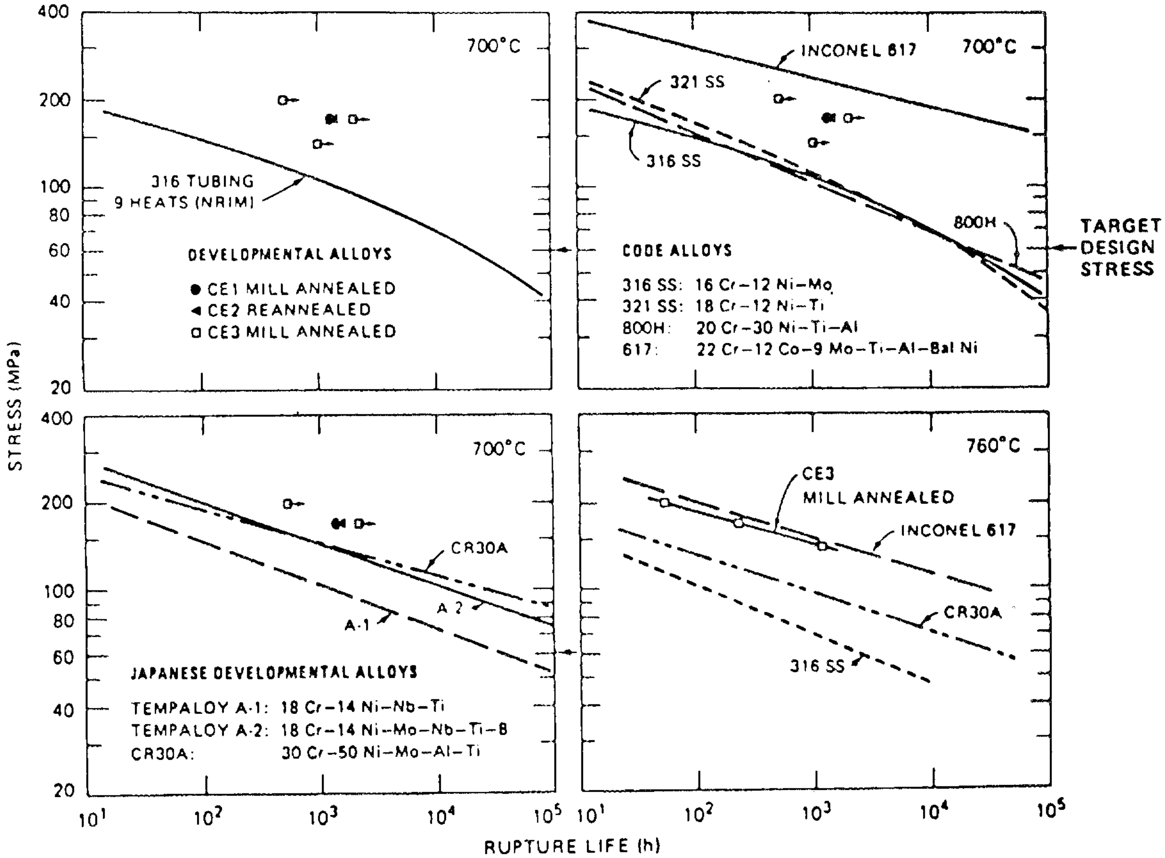

- ADVANCED ALLOYS ARE BETTER THAN 316, 800H AND 321
- ADVANCED ALLOYS ARE BETTER THAN JAPANESE DEVELOPMENTAL ALLOYS – TEMPALOY A-1, A-2 AND CR30A
- ADVANCED ALLOYS ARE APPROACHING THE BEHAVIOR OF INCONEL 617

Fig. 4. Comparative properties of ORNL-modified stainless steels, Japanese-modified steels, and Inconel alloy 617.

Table 7. Comparison of strengths of materials for advanced steam cycle applications

Material	Allowable stress (ksi)			
	1100°F	1200°F	1300°F	1400°F
9Cr-1Mo steel	9.7[a]	4.1[a]	1.5[a]	-
Type 316 stainless steel	12.4[a]	7.4[a]	4.1[a]	2.3[a]
Alloy 800H	13.5[a]	8.4[a]	5.4[a]	3.6[a]
Controlled microstructure austenitics	14[b]	14[b]	9[a]	4.3[a]
Inconel 617	22[b]	18[a]	12.6[a]	7.2[a]
Ni_3Al - wrought	30[c]	27[a]	12[a]	4.4[a]
Ni_3Al - cast	38[c]	36[a]	22[a]	14[a]
Si_3N_4	25–30[d]	25–30[d]	25–30[d]	25–30[d]
SiC	30–35[d]	30–35[d]	30–35[d]	30–35[d]

[a]Allowable stress was taken to be 0.67 times the average or typical rupture stress in 100,000 h;

[b]Allowable stress was taken to be 2/3 YS min;

[c]Allowable stress was taken to be 1/4 UTS min;

[d]Allowable stress for the structural ceramics was somewhat arbitrarily taken to be 0.5 times the fast fracture stress.

The Japanese approach to a 1200°F steam plant follows both a composite- and single-wall tube development path. Their base alloy development includes TEMPALOY A-1,* a modified 304H with small Ti and Nb additions, TEMPALOY A-2,* a modified 316H, and TEMPALOY CR 30A,* containing 0.06% C, 30% Cr, 50% Ni, 2% Mo, 0.2% Ti, 0.02% Zr, bal Fe. The former alloys are being developed for use with low sulfur coals and the latter with high sulfur coals. A comparison of the strength properties of TEMPALOY A-1 and A-2 with other boiler tubing alloys is given in Fig. 7. Composite tube development incorporates TEMPALOY A-1 and A-2 clad with NAC CR 35A* 0.07% C, 35% Cr, 46% Ni, 0.5% Nb, bal Fe) or with TEMPALOY CR 30A. As in the DOE program, the major development problem appears to be the attainment of acceptable fireside corrosion resistance for a total cost that does not usurp the dollars gained from the higher operating temperature The Japanese have completed a number of 100 h tests in synthetic coal ash

*Trademark of Nippon Kokan K.K., Fukuyama, Japan.

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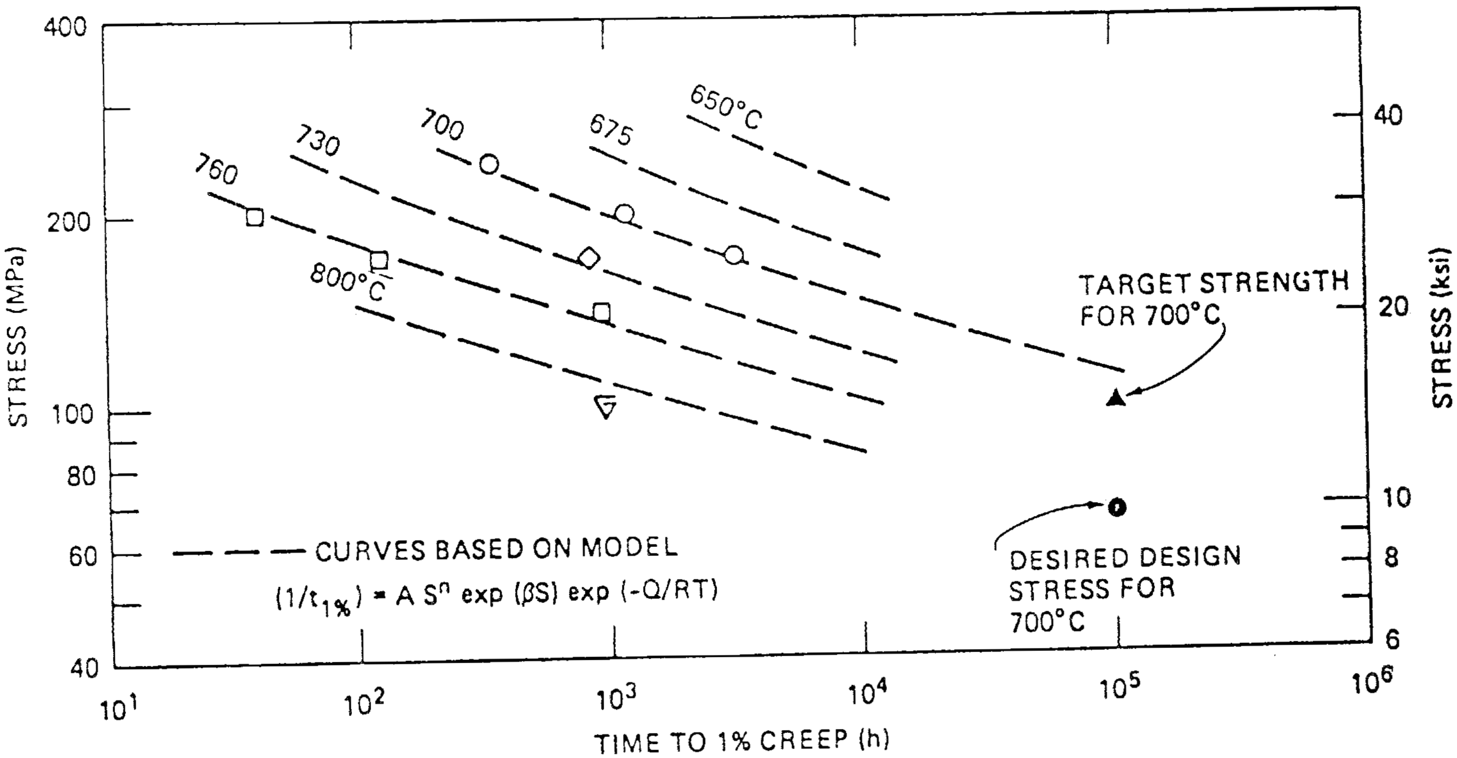

Fig. 5. Stress vs time to 1% creep for a developmental Cr-Ni-Mo stainless steel.

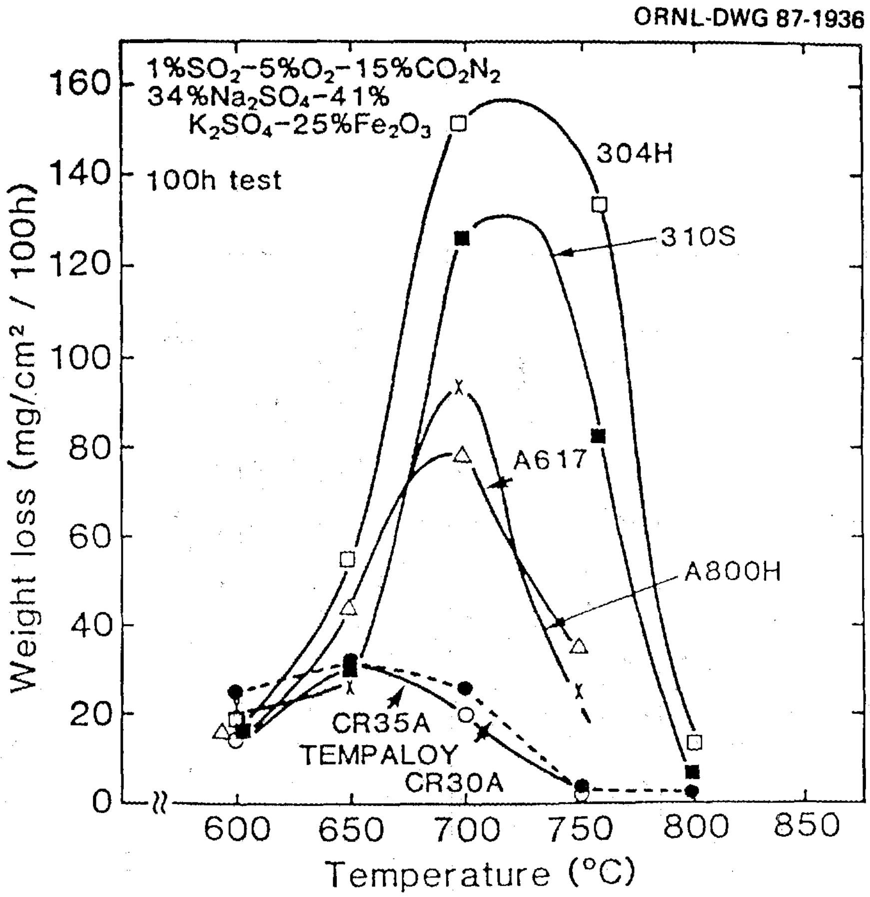

Fig. 6. Comparison of corrosion losses of high-temperature alloys tested with synthetic coal ash (composition shown at top of figure). Alloys CR35A and CR30A contain 35 and 30% chromium, respectively.

at 1100 to 1470°F and have demonstrated the effectiveness of chromium (at levels >25%) for suppressing molten sulfate attack (see Fig. 6). They currently believe that a monolithic tube based on the TEMPALOY CR 30A composition will be the most cost-effective approach for the higher temperature steam generator sections in 1200°F steam plants using high-sulfur coals. Based on current fabrication practices, the cost of making a composite tube with chromium-lean (18%), chromium-rich (30–35%) segments is higher than the cost of a full-section TEMPALOY CR 30A tube. Again, a breakthrough in the manufacture of more economical composite tubes appears essential if the full potential of austenitic stainless steels is to be exploited in advanced steam plants. Otherwise, the viability of stainless steels in higher temperature steam plants must depend on the ultimate development of coal cleaning and alternative combustion techniques.

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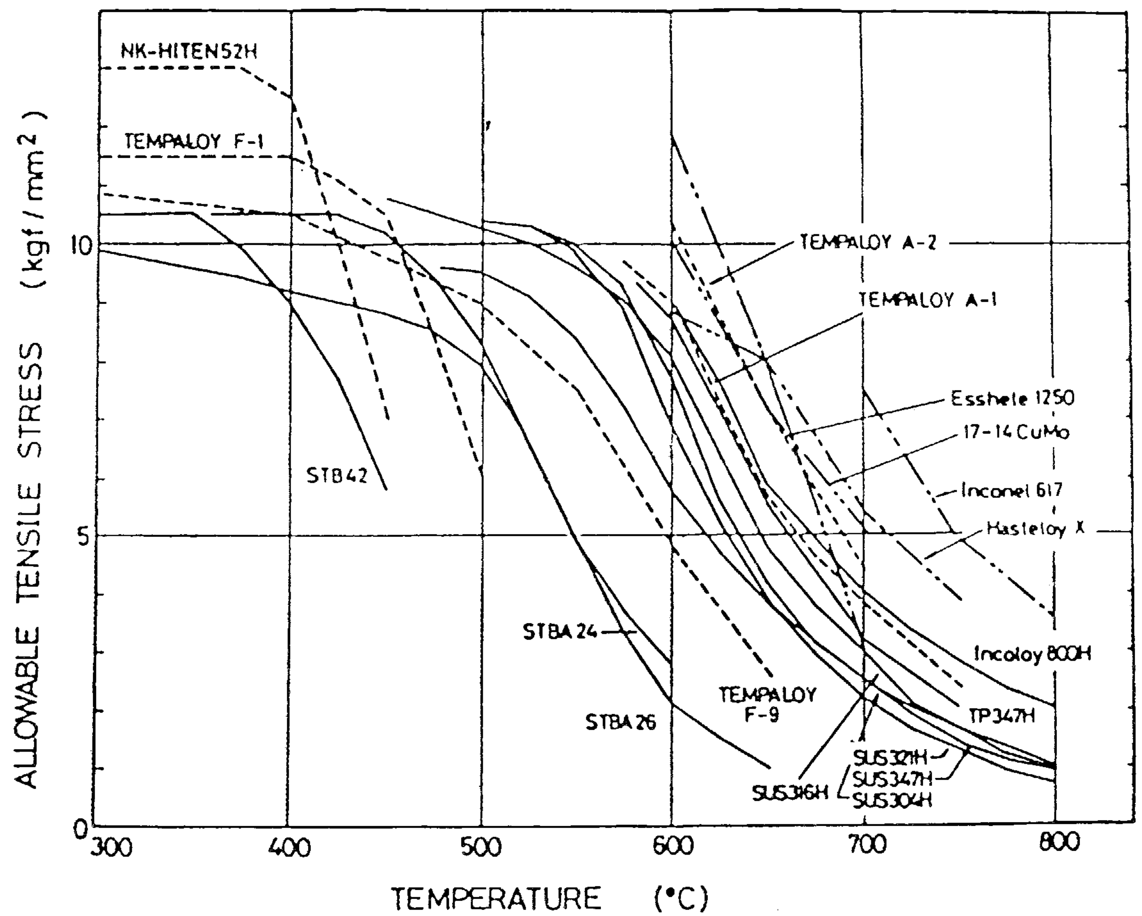

Fig. 7. Strength comparison of boiler tubing alloys. *Source*: *Materials of Ultra Super Critical Boiler (Nkk Tempaloy Series: Application of USC Boiler)*, *Nippon Kokan K. K.*, Fukuyama, Japan, February 1984.

In assessing the needed properties of other candidate materials for advanced steam cycle concepts, the following performance standards would be desirable:

1. for applications at 1300 and 1400°F, at least twice the creep-rupture strength of type 316 stainless steel at 1300°F and strength equivalent to Inconel 617 at 1400°F;
2. fireside corrosion resistance at 1100 to 1450°F;
3. steamside corrosion resistance at temperatures up to 1300°F;
4. fabricability comparable to current superheater alloys (unless proven alternative boiler configurations are forthcoming); and
5. ability to be joined to conventional boiler materials.

Over the longer range, alloys based on the ordered system Ni_3Al appear promising with respect to all but the second of these standards. However, based on the known resistance of the NiAl intermetallic to sulfur attack, simple surface diffusion treatments with aluminum could prove effective in preventing coal ash attack. This solution may be more workable in the case of Ni_3Al, where Al is a major constituent, than in the case of stainless steels, although more obviously needs to be known concerning fireside attack of both NiAl and Ni_3Al. The economics of Ni_3Al is another concern, although if certain technical problems can be overcome, the cost of the alloy in fabricated form could be below that of today's composite co-extruded stainless steel tubes, even if the former must be aluminized.

A ceramic tube boiler could give a major boost to the temperature and pressure conditions achievable with steam, and development of structural ceramics such as silicon nitride and/or silicon carbide have already reached the point that longer tubes and leaktight joints are commercial realities for some applications.

Work on structural ceramics for heat engines and heat exchangers show that both SiC and siliconized SiC ceramics have long-term potential for advanced plant applications. Other structural ceramics such as Si_3N_4, the sialons (e.g., Si-Al-O-N), or fiber-reinforced SiC are also excellent candidate materials. However, in addition to the well-known problems of fabricability and ductility, corrosion also remains a major concern, both in the context of fireside corrosion (i.e., reaction with complex alkali-iron sulfates) and with steamside corrosion (i.e., conversion to SiO_2 and dissolution in supercritical steam), and stress corrosion cracking. Also, it needs to be remembered that the same developments in ceramics technology could be exploited effectively for high-temperature Brayton and magnetohydrodynamic cycles.

Turbine Materials

The operation of large steam turbines (≥700 MW) has generally been limited to steam conditions of up to 3500 psi at 1000°F. However, smaller steam topping turbines capable of operating up to 5000 psi and 1200°F have been available since ~1950. Turbine manufacturers generally are optimistic that the latter capabilities can be extended to larger coal-fired power plants with only modest materials and design developments. Up to 1050°F steam temperatures, 12% chromium steel rotors can be operated without rotor cooling. Uncooled rotors above this temperature would require the use of austenitic stainless steels. Statistics accumulated for steam topping cycle turbines in the 100 MW to 199 MW size range operating from 1020 to 1184°F show no more forced outages (with an average of 125,000 h per unit) than for similarly sized units operating at lower temperatures.[21] One problem with the higher-temperature turbines has been manifested in disassembly times. Corrosion and creep have caused jamming of tight-fitting components, but the problems have gradually been overcome through improved materials and larger clearances.

If use of austenitic steels is required to achieve 1100°F and higher steam conditions with a 700 to 1000 MW class of turbine, the capability to forge rotors with dimensions up to 3 ft in diameter and weights up to 17 tons must be determined. Through advanced steel refining treatments such as electroslag remelting and argon-oxygen decarburization, it should be possible to achieve ingots of the quality needed to make defect-free forgings in the size range needed.

The lower thermal conductivity and higher thermal expansion of the austenitic steels (compared to ferritic or martensitic) also pose greater problems in the areas of residual stresses and low cycle (thermal) fatigue. Given the higher strengths of the austenitics, however, these problems can be addressed by proper attention to the design allowable stresses and attemperation schemes that bypass steam around the turbine during startups and shutdowns. One other problem confronting the use of austenitic stainless steel rotors relates to stress-corrosion cracking, which will require stringent limits on water impurities.

The recent improvement in the ductility of ordered intermetallic alloys such as Ni_3Al has introduced a new class of materials which appears to hold considerable potential as a turbine material. However, it is still too early in the development of Ni_3Al to judge its forging characteristics or steam corrosion resistance. Nevertheless the material may afford significant advantages over conventional austenitic stainless steels given its lower thermal coefficient of expansion and potentially higher creep strengths. Figure 8 compares the time to 1% creep at 1400°F/40 ksi for various alloys, and it can be seen that an Ni_3Al alloy containing chromium has been strengthened to the point that it surpasses one of the better gas turbine superalloys (WASPALLOY).

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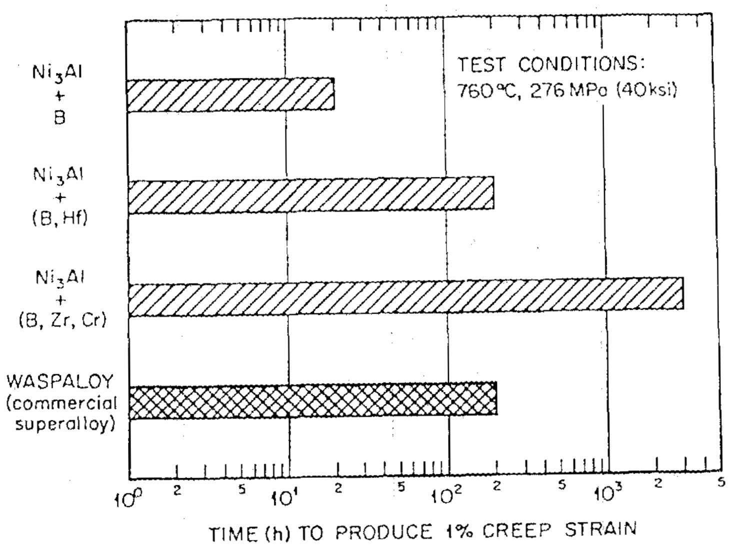

Fig. 8. Comparative creep behavior of Ni_3Al alloys and Waspalloy at 760°C.

Other materials with long-term potential for advanced turbine applications are silicon carbide and/or silicon nitride. Current developments in association with blades and rotors for automotive gas turbines would translate to an operating temperature envelope as high as 1400 to 1800°F. Just where these capabilities fit with respect to steam service, however, cannot be judged from currently available data and must await design studies to establish size requirements, pressure boundary approaches, mass transport rates of SiO_2 in steam, etc.

The mechanical properties of several of the materials that might be used for advanced system applications at 1100, 1200, 1300, and 1400°F are compared in Table 7. Even if the strength potential of some of the advanced materials can be realized, the feasibility of materials development for steam temperatures above 1200°F cannot be fully answered until the engineering characteristics and materials costs of such systems are better defined. Because of their complexity, conventional pulverized-coal systems may have difficulty competing for large utility applications against the combined-cycle (gas turbine/steam turbine) approaches now under development, and design studies comparing the two approaches would have to be undertaken. More specialized applications, such as process heat combined with electricity production for industrial and residential complexes, may provide a better context for ultrahigh-temperature (>1200°F) steam systems. Here next-generation materials such as "ductile" ceramics and ordered alloys could find application as boiler and turbine materials.

An estimation of the relative cost of several of these advanced materials are compared with current materials in Table 8. The fabricated product cost figures are for large-diameter pipe for which comparison data were available. Assuming that current fabricability problems can be solved, we estimate the cost of Ni_3Al pipe to be <\$30/lb, which is required for break-even in advanced plants. Since the structural ceramics are less than one-half as dense as the metal alloys listed, cost per foot is a more meaningful comparison; and, as shown in Table 8, although SiC costs \$50/lb as fabricated pipe, it costs about the same as Inconel 617 and, perhaps, less than Ni_3Al on a per foot basis.

Table 8. Comparison of currently estimated costs of materials for advanced steam cycle applications

Material	Estimated cost: Base ($/lb)	Fabricated product ($/lb)	Fabricated product ($/ft)[a]
Type 316 stainless steel	0.80	2.90	30
9Cr-1Mo steel	0.60	2.50	25
Controlled microstructure austenitics	~1.25	~5.00	~50[b]
Alloy 800H	1.35	4.90	50
Inconel 617	5.40	20	200
Ni_3Al	~4	<30	<300[b]
Si_3N_4	36	-	-
SiC	10	~50	~200[b]

[a]Based on 3.5-in.-OD × 0.3-in. wall pipe; other sizes and more restrictive specifications could change the cost substantially, e.g., the cost of 2.375-in.-OD × 0.343-in. wall type 316 stainless steel purchased to a nuclear specification was $38/ft or $5.10/lb.

[b]Estimated.

CONCLUSIONS AND RECOMMENDATIONS

Requirements for new generating capacity over the next two decades will depend primarily on electricity demand growth, but also on the rate at which aging plants are replaced with new capacity and the extent to which a region can import power from another region with excess capacity. If electricity demand growth should accelerate by the early 1990s, conventional coal-fired plants with improved efficiency and flexibility are likely to be the first choice of the utilities. However, alternative generating technologies such as integrated gasification/combined cycle and atmospheric fluidized-bed combustion offer significant potential for sizable deployment beyond the turn of the century.

Even though future regulatory and economic variables will determine the size and type of future generating plants, within the framework of the assumptions used several important conclusions relative to conventional coal-fired plants have emerged from this study:

- Improved overall plant efficiencies (up to 4.2%) and lower fuel costs are associated with main steam conditions of 4500 psi and increasing temperatures from 1100 to 1400°F.
- Without further increases in steam throttle pressure, increases in steam temperature produce a less significant increase in efficiency. Further studies should be conducted to determine effects of increased pressure on efficiency and allowable material costs.
- If improved materials have the same strength at 1100 to 1400°F as present materials at 1000°F and cost on the order of $30/lb or less, fuel savings in 30 years will offset increases in capital costs.
- Based on the design and strength assumptions that were used, increasing steam temperatures above 1100°F offer significant fuel savings, but they may offer no significant overall cost savings if increased weights of higher cost improved materials are required. However, there could be a more significant advantage to higher temperatures and pressures if advanced materials like nickel aluminide and structural ceramics can be used because of their higher strengths.
- Development of satisfactory boiler materials will potentially be more significant than development of improved turbine materials

because capital costs are more sensitive to the cost of superheaters and reheaters, and susceptibility to liquid ash corrosion is likely to remain a major problem.

- Allowable material costs are more sensitive to allowable design stress and fuel costs than steam temperature; therefore, if the efficiency advantages of advanced systems are to be utilized, development of higher strength materials for these applications is needed.
- Design changes required to operate at temperatures above 1200°F cannot be specified without development of more specific flow charts and further modeling of advanced plant parameters.

Although improvements in efficiency are attainable with increases in steam temperature, a question that is not easily answered is how rapidly advanced materials would be incorporated into an advanced energy system. EPRI has indicated a 4500 psi/1100°F conventional plant is feasible, and both the United States and Japan are considering 5000 psi/1200°F plants. The former may be built before the end of the century, but it is unlikely a 1200°F plant will be built until after the year 2000. Among electric generating alternatives it may be that with continued government support there could be significant deployment of small cogeneration systems within 15 to 20 years. Since these systems will probably operate at steam boiler temperatures of 1400 to 1500°F, advanced materials will be required that go beyond the strength and corrosion resistance being sought by the present EPRI and DOE (AR&TD Fossil) materials programs. Because of their high-temperature strength potential, metallic alloy systems based on intermetallic compounds (either as ductile, ordered alloys, or precipitation strengtheners) and ductile structural ceramics could be long-term candidates for advanced conventional or cogeneration systems. Therefore, a materials development program for applications at 1400 to 1500°F would be a logical extension of present materials development programs, and should result in higher strength alternative materials to those now being investigated for 1100 and 1200°F applications. Without such a materials development program it seems likely that advanced electric generating systems based on fossil fuels will continue to be materials limited.

REFERENCES

1. Harry W. Colborn, "The Demand for Electricity – Ten Years Hence," *Coal and Power Management Issues*, Vol. 1, proceedings of 8th International Coal & Solid Fuels Utilization Exhibition and Conference, Pittsburgh, Pa., Nov. 12–14, 1985, p. 48.

2. Gilbert Associates, Inc., Electric Power Research Institute, *Development Plan for Advanced Fossil Fuel Power Plants*, EPRI CS-4029, Project 1403-4, Electric Power Research Institute, Palo Alto, Calif., May 1985.

3. G. M. Yasenchak et al., *Engineering Assessment of an Advanced Pulverized-Coal Power Plant*, EPRI CS-2555, Electric Power Research Institute, Palo Alto, Calif., Project 1403-2, August 1982.

4. S. B. Bennett, *Engineering Assessment of an Advanced Pulverized-Coal Power Plant*, EPRI CS-2223, Project 1403-1, Electric Power Reasearch Institute, Palo Alto, Calif., January 1982.

5. L. C. Fuller and T. K. Stovall, *User's Manual for PRESTO, A Computer Code for the Performance of Regenerative Superheated Steam-Turbine Cycles*, ORNL-5547, June 1979.

6. R. C. Spencer, K. C. Cotton, and C. N. Cannon, "A Method for Predicting the Performance of Steam Turbine-Generators. . . 16,500 KW and Larger," ASME paper No. 62-WA-209, presented at the Winter Annual Meeting, New York, November 25–30, 1962.

7. G. J. Silvestri, Westinghouse Electric, Orlando, Fla., telephone communication, March 21, 1986, to L. C. Fuller, Oak Ridge National Laboratory.

8. J. G. Yevick, *Fast Reactor Technology: Plant Design Report*, p. 253, The M.I.T. Press, Massachussetts Institute of Technology, Cambridge, Mass., 1966.

9. R. L. Bartlett, *Steam Turbine Performance and Economics*, McGraw-Hill, New York, 1958.

10. C. O. Bennett and J. E. Myers, *Momentum, Heat, and Mass Transfer*, 1st ed., McGraw-Hill, New York, p. 367.

11. J. H. Perry, *Chemical Engineers' Handbook*, 4th ed., McGraw-Hill, New York, 1963, p. 10-21.

12. *The Bull Run Steam Plant*, Tennessee Valley Authority, Knoxville, Tenn., TVA Technical Report No. 38, 1967.

13. F. S. Aschner, *Planning Fundamentals of Thermal Power Plants*, Halsted Press, New York, 1978, pp. 135—37.

14. *Nuclear Energy Cost Data Base - A Reference Data Base for Nuclear and Coal Fired Power Plant Power Generation Cost Analysis*, DOE/NE-0078, U.S. Department of Energy, Office of Nuclear Energy, December 1986.

15. *Phase VII Update (1984) Report For The Energy Economic Data Base Program EEDB - VII*, DOE/NE-0051/2, prepared for U.S. Department of Energy by United Engineers & Constructors Inc., August 1985.

16. P. L. Rittenhouse et al., *Assessment of Materials Needs for Advanced Steam Cycle Coal-Fired Plants*, ORNL/TM-9735, August 1985.

17. *Steam, Its Generation and Use*, 39th ed., Babcock & Wilcox Company, New York, 1978.

18. 1983 *ASME Boiler and Pressure Vessel Code*, Section I, "Power Boilers," The American Society of Mechanical Engineers, United Engineering Center, New York.

19. R. W. Swindeman et al., *Alloy Design Criteria and Evaluation for Advanced Austenitic Alloys in Steam Service*, ORNL-6274, May 1986.

20. Foster Wheeler Development Corporation, *Laboratory Fireside Corrosion of Improved Superheater Tube Alloys and Coatings*, EPRI-CS-3134, Electric Power Research Institute, Palo Alto, Calif., June 1983.

21. H. Haas, W. Engelke, J. Ewald, and H. Termuehlen, "Turbines for Advanced Steam Conditions," pp. 330—38 in *Proc. American Power Conference*, Vol. 44, 1982.

APPENDIX

STEAM POWER PLANT FLOW CHARTS

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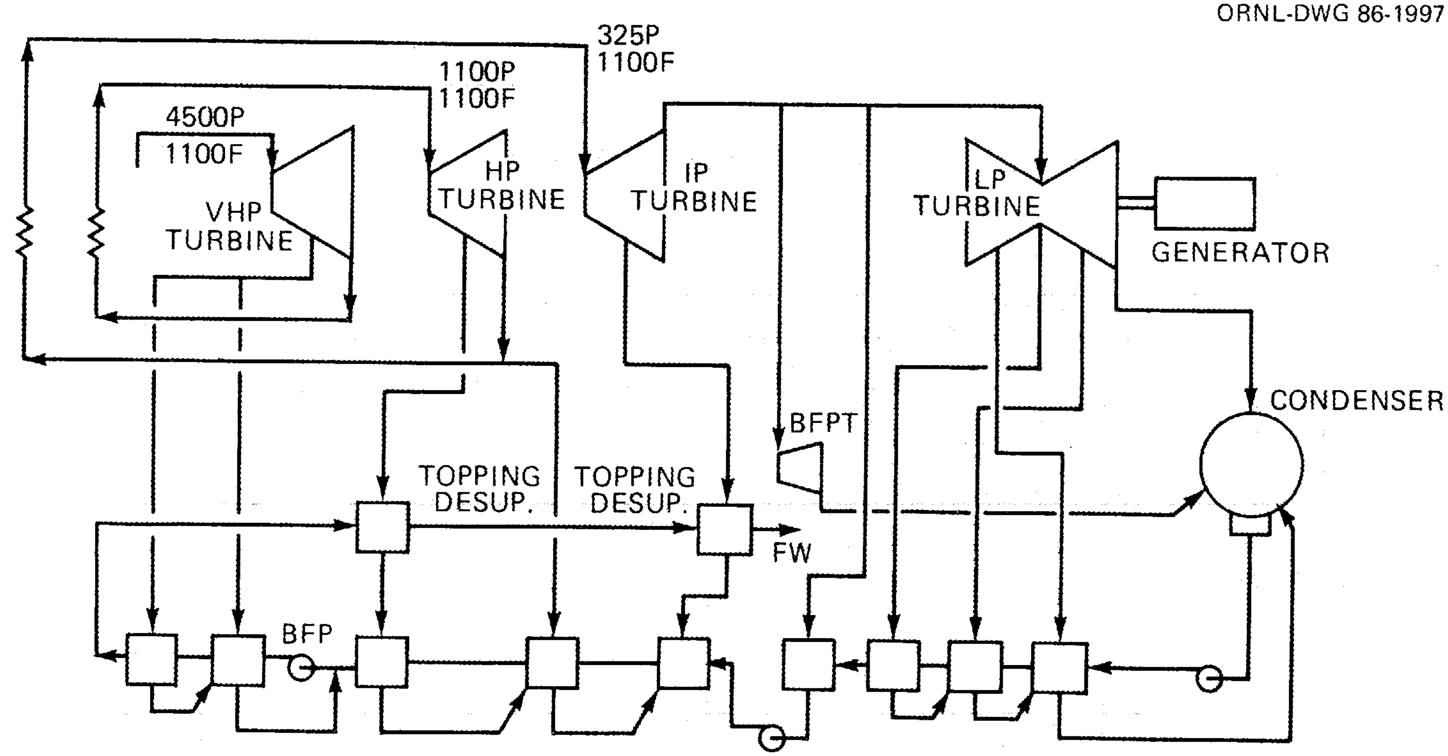

Fig. A.1. Advanced plant flow chart (4500 psi/1100°F steam conditions).

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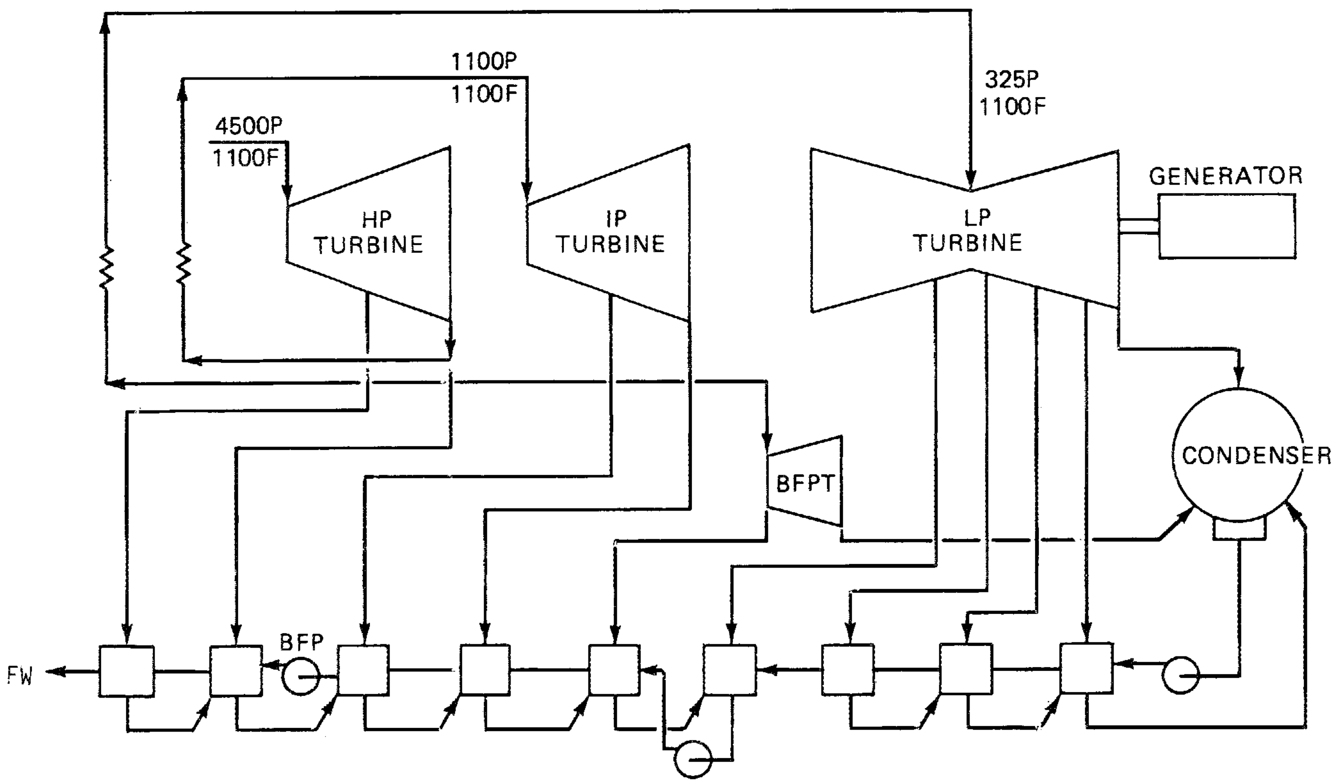

Fig. A.2. Advanced plant (PRESTO code) chart (4500 psi/1100°F steam conditions).

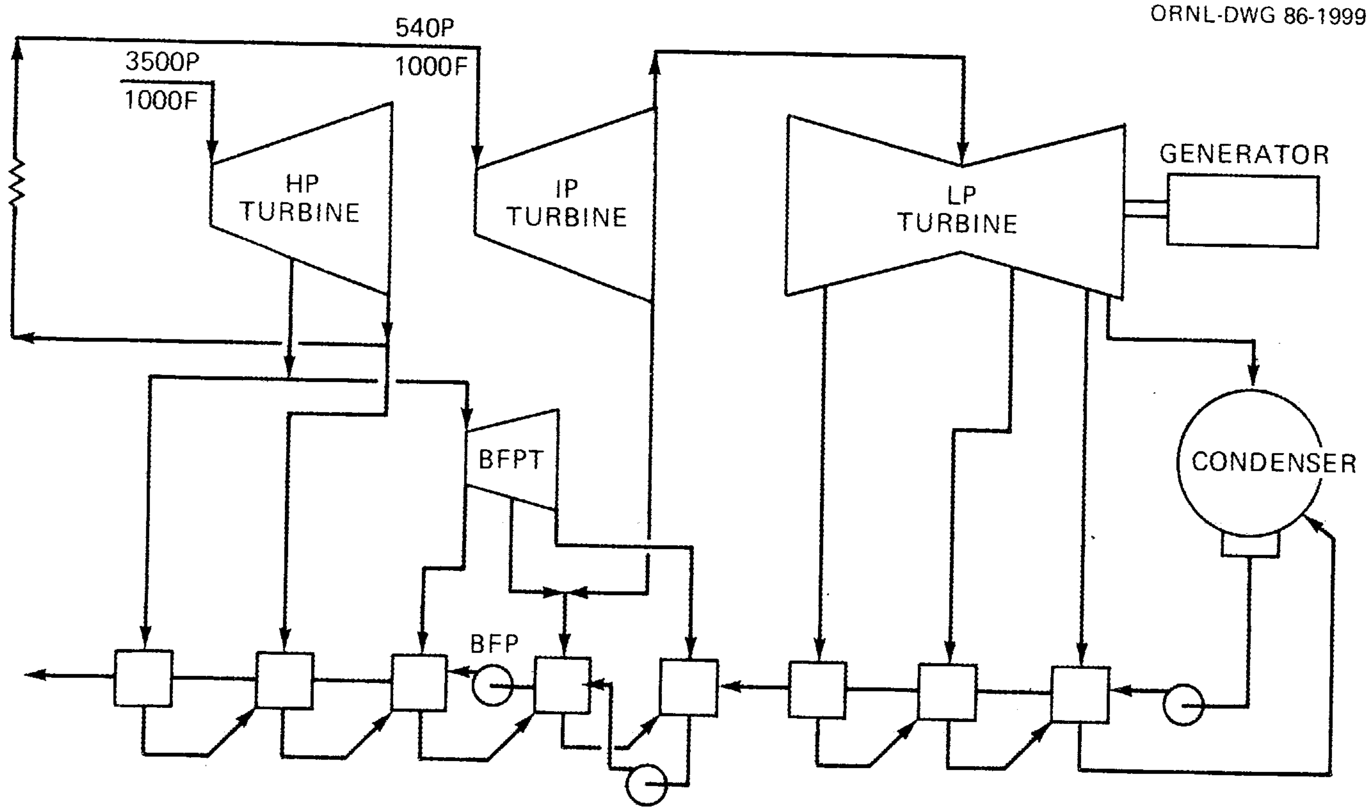

Fig. A.3. Bull Run Steam Plant flow chart (3500 psi/1000°F steam conditions).

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