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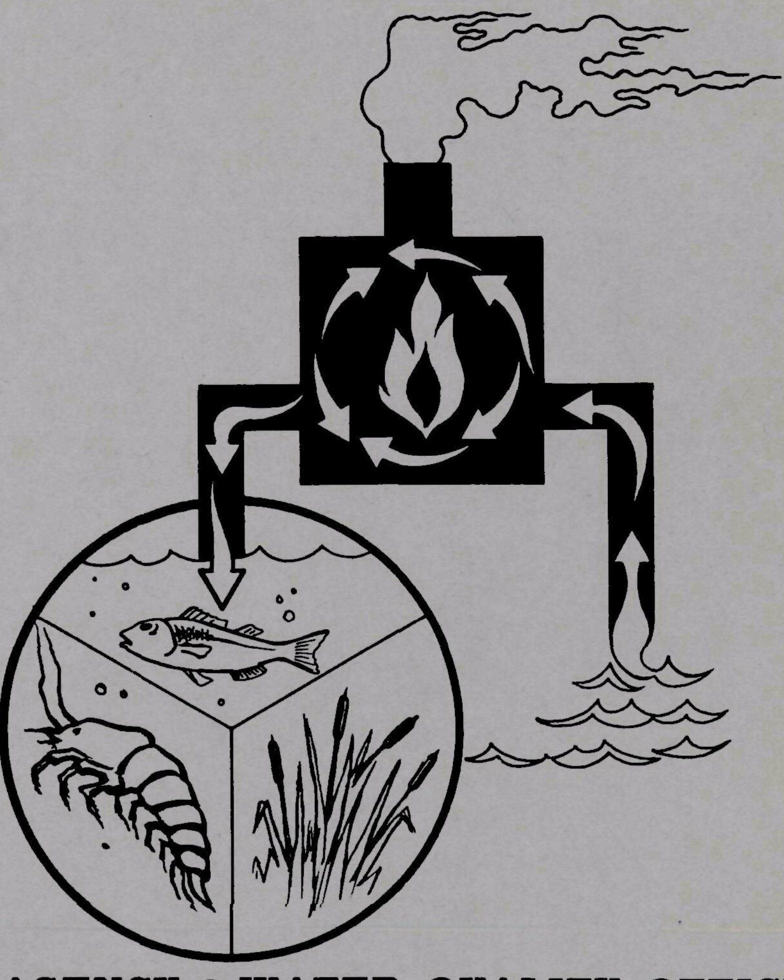
Advanced Nonthermally Polluting Gas Turbines in Utility Applications



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ADVANCED NONTHERMALLY POLLUTING GAS TURBINES IN UTILITY APPLICATIONS

by

United Aircraft Research Laboratories
of the United Aircraft Corporation
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for the

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Water Quality Office

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ABSTRACT

Detailed performance, size, and cost estimates were made for advanced simple-, regenerative-, and compound-cycle gas turbine engines for turbine inlet temperatures of 2000° F and above as anticipated to be commercially available in the next two decades. Conceptual designs for 1000-Mw central *power station utilizing gas turbines and comparisons of complete gas turbine* and steam turbine power station installed costs and total busbar power costs were made for the various regions of the US.

It is shown that the gas turbines in the 1970 decade could produce electric power at lower costs than steam turbines in the South Central region of the US where natural gas is readily available. Elsewhere in the US the gas turbines would be economically competitive if moderately priced clean fuels are available. Advanced gas turbines will become more competitive in the 1980 decade as anticipated increases in turbine inlet temperature, component efficiences and larger engine designs lead to more efficient and lower-cost engines and power stations.

Although the development costs for large, advanced gas turbines would approach from 100 to 200 million dollars, the total amount that utilities are expected to expend for cooling devices to combat thermal pollution over the next two decades will exceed more than ten times this amount. Thus advanced gas turbines should be given serious consideration for increased research and development support.

This report was submitted in fulfillment of Contract 14-12-593 under the sponsorship of the Environmental Protection Agency, Water Quality Office.

CONTENTS

CONTENTS (CONT.)

CONTENTS (CONT.)

CONTENTS (CONT.)

CONTENTS (CONT.)

CONTENTS (CONT.)

LIST OF FIGURES

LIST OF FIGURES (Continued)

LIST OF FIGURES (Continued)

LIST OF FIGURES (Continued)

FIG. NO.

FIG. NO.

LIST OF TABLES

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SECTION I

CONCLUSIONS

1. High-output-capacity open-cycle gas turbines incorporating the design advancements projected to become available during the next two decades could eliminate river and lake thermal pollution while producing power at lower busbar costs than steam power generation systems in those regions of the country where natural gas or other suitable gas turbine fuel is available at a price level comparable to competing fossil fuels.

2. Costs to develop nonthermally polluting gas turbine power systems are estimated to be at least an order of magnitude less than the anticipated investment costs by utilities for supplementary cooling equipment such as cooling towers, ponds, etc., needed for steam plants to reduce thermal pollution over the next two decades.

3. Gas turbines which will become available in the 1970 decade could begin to penetrate the swing-load and base-load electric utility market in the South Central Region of the US where natural gas is available. In other regions of the US the penetration of gas turbines for base-load operation could be delayed until the early 1980's unless low-cost clean fuels become available sooner than anticipated.

4. It is anticipated that domestic natural gas in selected southern regions of the US, and synthetic pipeline gas or imports of LNG in most remaining regions, will be available at price levels within the limits needed to insure competitive busbar electric power costs from gas turbine stations.

5. Improvements in high-temperature turbine materials, turbine cooling techniques, and aerodynamic design derived from current aircraft engine development programs could permit progressively higher maximum operating temperatures and cycle pressure ratios in base-load gas turbines for electric utility applications by the 1980's if pursued vigorously. These advances could result in system efficiencies approaching and exceeding the levels available with modern steam power plants and unit output capabilities of 200 Mw and above for open-cycle fossil-fueled gas turbine plants.

6. The projected growth rate of the utility industry and the emergence of nuclear power will heighten the present cooling water shortages, and together with federal and local regulations, will require a broader evaluation of electric power generation methods and cooling devices. Although cooling towers and ponds will be used with increasing frequency as short-term solutions to avoid thermal pollution, they will add from 1% to 10% to busbar power costs and occupy much-needed space.

7. Independence from cooling water, smaller plant layouts, potential site advantages, quicker delivery schedules, and substantially lower installed power plant costs relative to steam stations will make gas turbines especially attractive for the future mid-range and base-load needs of the electric utility industry.

8. The performance of conventional steam power systems is projected to remain essentially constant over the next two decades. Slight improvements in component efficiency can be expected, but increases in cycle operating conditions to give better performance cannot be economically justified unless very expensive fuels are utilized. The total installed costs are expected to remain relatively constant (in 1970 dollars) since the anticipated economies of scale will be offset somewhat by the need for cooling towers or ponds required to reduce thermal pollution and the continual pressure for higher construction labor rates.

SECTION II

RECOMMENDATIONS

1. Programs aimed at transferring the technology developed for aircraft gas turbine engine applications to base-load electric utility use should be promoted and sponsored by the federal government as a means of eliminating thermal pollution, improving the utilization of our water resources, and providing low-cost electric power.

2. Additional investigations should be undertaken to determine the potential of advanced open-cycle gas turbine power generation systems which can operate independently of a source of cooling water as a solution to the siting of power generation stations and transmission lines while allowing beneficial utilization of our dwindling land resources.

3. Improved processes and techniques leading to the development of adequate supplies of low-cost clean fuels, i.e., LNG, synthetic gas from coal gasification, and domestic natural gas from previously untapped sources, should be encouraged and supported by the federal government as well as the utility industry as a means of producing fuels resulting in lower air pollution and suitable for use in gas turbines so as to eliminate thermal pollution.

4. The encouraging results of this program suggest that a study of the application of gas turbine technology for nuclear-fueled power generation systems is warranted since nuclear power is projected to have an ever-expanding role in the electric power generation industry.

SECTION III

INTRODUCTION

The increasing problem of temperature elevation or thermal pollution of river and lake waters used to cool electric utility power generating plants is becoming a major concern to federal and state governments as well as to electric utilities and conservation groups. Some evidence already exists that the effects of heated water from power stations can be harmful to aquatic life and can adversely change biochemical reaction rates, thus limiting the capability of these waters to assimilate other wastes. Furthermore, the value of water for drinking, recreational, and industrial use usually decreases with higher water temperatures.

If present projections of the electric power industry growth rates are correct, the power generation capacity and output in the United States by 1990 will have expanded to approximately four times the present-day level, and by the year 2000, the generating capacity will be approximately ten times the present-day levels. The cooling of steam condensers in the electric generating plants presently operating in the United States requires over 100,000 million gallons per day of cooling water. If unrestricted use of once-through condenser cooling continues to be permitted, by the year 2000 as much as 600,000 million gallons per day, or the equivalent of one-half of the average daily runoff of all rivers in the US, would be needed for cooling power generating systems. The cooling water shortage will accelerate as nuclear-fueled stations provide a larger portion of electric power demand in the future, since most modern nuclear plants discharge about 50% more waste heat to cooling water than do fossil-fueled plants of the same output.

Technological solutions to waste heat disposal have not kept pace with the increased power production, and concentrated efforts are under way by the electric power industry and government agencies to find solutions through a broad range of approaches. For example, numerous studies (reported in government and trade publications) have been initiated to determine means of minimizing the effects of discharge heat on the aquatic environment, to develop beneficial uses for waste heat, to reduce the waste heat produced from power plants, to utilize cooling schemes that produce no harmful effects, and to devise new and nonpolluting methods of power generation.

Modeling techniques, experiments, and analytical programs are being pursued to minimize the effects of waste heat on the aquatic environment through increased turbulence, greater dilution, and faster dispersion of the cooling water. Unfortunately, these solutions are usually not widely applicable to other locations, and thus power plant site selection where adequate cooling water is available can become a costly procedure. Substantial performance improvements of steam-electric generating systems resulting in reduced waste heat emissions appear unlikely. Careful projections indicate that the maximum operating temperatures of steam power plants will be limited to approximately 1000 F by the excessive costs for materials capable of operating at higher temperatures. Several potentially beneficial uses for the enormous quantities of hot water are being explored, including sea farming and irrigation. The widespread economic utilization of waste heat for such purposes is uncertain at this time. The most promising near-term solutions appear to be increased use of cooling towers, cooling reservoirs, and spray canals. Dry or nonevaporative towers consume almost no water and discharge heat directly into the atmosphere, but these towers, like wet towers, are costly to build, require substantial space, and result in higher fuel costs due to low power plant efficiencies.

Long-term solutions require new methods of generating power which reject their waste heat directly to atmospheric air and hence require no cooling water. An example of such an open-cycle system, i.e., one which utilizes ambient air as the working fluid of the thermodynamic cycle, is the gas turbine engine. Other power-generating methods, such as magnetohydrodynamic generators, thermionic power generators, or other unconventional power generation systems, are being investigated and could reduce thermal pollution, but these methods will require enormous financial support and substantial technological advances before they can be reduced to commercial practice. The gas turbine, however, which requires no cooling water and is already used extensively for peak-power applications as well as in numerous other industrial and military systems, has the potential of eliminating thermal pollution based upon the numerous development programs in progress to date.

Presently, the utilization of gas turbine engines for stationary electric power generation is limited to peaking power applications because of their relatively low thermal efficiency in comparison to fossil-fueled steam plants. However, recent engineering advances achieved during extensive research and development efforts on military and commercial aircraft applications have provided the basis for substantially improved large-capacity base-load gas turbine power systems with significantly higher thermal efficiencies than are attainable with present systems. Because of the higher compressor pressure ratios and higher turbine inlet temperatures which will be attainable within the next two decades, it is possible that base-load gas turbine power plants capable of producing 150 to 350 Mw per unit will become commercially feasible in the foreseeable future.

As a result of these technological advances in gas turbine design and the necessary compromises in steam-electric power plant design (to adhere to recently imposed water temperature standards), it appeared that future gas turbine power systems might be capable of generating base-load electric power at costs competitive with fossil-fueled steam-electric systems. Therefore, the primary objectives of this study were: (1) to identify the design requirements for future fossil-fueled thermally nonpolluting power stations; (2) to define and select advanced fossil-fueled open-cycle base-load gas turbine systems that have the potential for generating lowest-cost electric power while eliminating thermal pollution; and (3) to estimate and compare the costs of producing electric power with advanced open-cycle base-load gas turbine stations and advanced fossil-fueled steam stations designed to reduce or eliminate thermal pollution during the 1970 and 1980 decades.

SECTION IV

SCOPE OF THE STUDY

To achieve the objectives of this study, conceptual design and cost programs developed at UARL were utilized as a means of determining the approximate performance, cost, and size characteristics of advanced gas turbine engines and to incorporate the design advancements, materials, and other features which are already in use in aircraft engines or projected to become available in the next two decades in base-load gas turbines. To cover the wide range of operating conditions and design parameters such as turbine inlet temperature and compressor pressure ratio, the cost and design analyses were heavily dependent on a number of simplifying assumptions. Thus the results are not intended to reflect comprehensive design aspects of advanced gas turbines which would require much more extensive and costly efforts but rather to show general features and levels of performance and cost that might be attained for utility applications.

To provide a realistic appraisal of the potential of gas turbines as a means of eliminating river and lake thermal pollution, extensive review of the available literature and discussions with electric power industry representatives were held to estimate (1) the availability and range of prices for suitable gas turbine and steam turbine system fuels, (2) the extent and severity of cooling water shortages and the implications of thermal pollution restrictions, (3) the operating limitations, performance, and cost characteristics of present-day and projected future steam power stations, and (4) the operating and cost characteristics of cooling towers and reservoirs suitable for use with steam power plants.

SECTION V

SYNOPSIS OF STUDY RESULTS

The national demand for electric power will double every ten years during the next two decades, and nuclear fuel will become a significant source of energy after 1980. The demand for power from thermal sources will increase at the fastest rate in the West, West Central, South Central, and Southeast Regions of the US. Except for the Southeast Region, these same areas will generally lack sufficient natural sources of cooling water to utilize once-through cooling systems in nuclear- and fossil-fueled steam power plants, and the majority of new stations will employ some alternative cooling system such as cooling ponds or wet cooling towers. State and federal restrictions on thermal discharges will further stimulate widespread utilization of cooling ponds and towers except for isolated ocean power plant installations.

Long-term supplies of cheap fossil fuels capable of complying with present and anticipated pollution-control laws are not adequate to meet the demands of the utility industry. Substantially higher prices will be needed to stimulate the development of low-sulfur coal and natural gas supplies. However, natural gas should be available at price levels of 26¢ to 40¢/million Btu near the sources of supply (South Central and Pacific Regions) during the next two decades and at price levels of about 40¢ to 60¢/million Btu from coal gasification or in the form of LNG imports in other coastal and midwestern areas.

The price of coal and residual oil for utility application will increase to the 30¢ to 50¢/million Btu level, except for specific areas, unless efficient low-cost stack-gas cleanup systems are demonstrated.

The performance of conventional steam power systems is projected to remain essentially constant over the next two decades. Slight improvements in component efficiency can be expected, but increases in cycle operating conditions to give better performance cannot be economically justified. The total installed costs of steam power plants, including escalation, interest during construction, and other factors, are expected to remain relatively constant at about $160 and $155 per kw for coal-fired and oil-fired stations, respectively, and at about $140/kw for gas-fired stations. Somewhat lower costs can be expected in the South Central and Pacific Regions of the country. Although the average plant size will approach 1000 Mw by the 1980 decade, the economies of scale will be offset by the need for pollution-abatement equipment. A variety of alternatives to the once-through cooling systems for steam plant condensers is available, and estimates indicate the use of wet cooling towers, cooling ponds or lakes or spray canals would increase the total busbar power costs in fossil-fueled plants by only 1 to 3%. The use of dry rather than wet cooling towers would alleviate some siting difficulties but at a substantial 9 to 10% increase in the busbar power costs. However, the widespread use of dry towers is not anticipated in the next decade and perhaps longer.

It is estimated that gas turbine engines capable of operating at turbine inlet temperatures as high as 2200 to 2400 F could be in operation in utility power generation systems before the end of the 1970's if turbine materials and blade cooling techniques presently under investigation for aircraft and other applications were utilized. By the early 1980's turbine inlet temperatures 200 to 400 F higher can be anticipated, and these advances together with similar improvements in compressor and combustor technology will form the basis for gas turbine engines capable of providing 200 to 250 Mw of electric power in a single unit while achieving overall plant thermal efficiency levels of 36 to 38% in simple- and regenerative-cycle configurations. The higher turbine inlet temperatures will also permit substantial improvements in engine specific power levels which are projected to result in 20 to 30% reductions in future engine and power station selling prices relative to present-day prices for gas turbine systems. The total site area requirements for gas turbine stations would be on the order of 10% of those for conventional steam power stations. Together with elimination of the need for cooling water, the reduction in area requirements could tremendously simplify utility planning.

Precooling the compressor bleed air to approximately 200 F in external heat exchangers prior to its use in the turbine section enhances the performance and cost characteristics of gas turbine systems; hence, precooled air will be used with increasing frequency in the next two decades.

The compound-cycle gas turbine engine offers attractive levels of performance and cost for central power stations, and further study to confirm this preliminary result is recommended.

Advanced open-cycle gas turbines utilizing technology derived from aircraft engine programs offer a means of eliminating thermal pollution while generating electric power at busbar costs substantially below those which will be attainable with future conventional steam systems in the natural-gas-rich South Central region of the US. The estimated busbar costs of the simple-cycle gas turbine station vary from approximately 0.5 mills/kwhr to 1.0 mill/kwhr below those projected for steam stations in the South Central Region during the 1970 and 1980 decades, respectively. The regenerative-cycle gas turbine system would generate power at costs lower than those for the steam system but at a somewhat higher level than the simple-cycle gas turbine system. The conclusions are relatively insensitive to the capital and interest charges, as well as to the fuel cost used in the comparisons.

Simple-cycle gas turbine designs which are projected to be commercially available by the early 1980's could produce power at busbar costs competitive with residual-oil- or coal-burning steam stations in the remaining regional locations for load factors up to approximately 70%, even when burning a fuel costing as much as 20¢/million Btu more than for the steam system.

The reduced transmission-distribution network requirements associated with relatively small gas turbine power generating units, which may be located close to the load centers due to their independence from cooling water supplies, can result in an appreciable savings as compared to networks required with large steam stations. Increased reliability can be achieved for a given power system through a reduction in power generating unit size and a diversification of unit size. The combined effects can result in equal electric power costs with dispersed gas turbines, in comparison with power costs from large steam stations, notwithstanding a cost increment of up to several ¢/million Btu for the gas turbine fuel.

Coal gasification technology is becoming available as the result of various incentives, so that both pipeline-quality high-Btu/ft^3 gas and low-Btu/ft^3 producer-type gas are anticipated to become available in the next decade at 20 to 40¢/million Btu above the price of the coal or residual oil used as feedstock.

Utilities will spend approximately $2 to $4 billion in each of the next two decades for cooling towers, ponds, and other devices in an attempt to reduce or eliminate thermal pollution of the nation's rivers and lakes. Advanced open-cycle gas turbines capable of generating low-cost electric power could be developed for perhaps one-tenth of that earmarked for low-pollution cooling systems for steam power plants.

SECTION VI

DESIGN REQUIREMENTS OF FUTURE FOSSIL-FUELED THERMALLY NONPOLLUTING POWER STATIONS

SUMMARY

An investigation was undertaken to determine the design requirements for future fossil-fueled thermally nonpolluting steam power stations to provide a realistic reference for subsequent comparative evaluation of advanced-design gas turbines. A review of available literature was made to determine those geographical areas of the country which are experiencing or are expected to experience cooling water shortages and/or thermal pollution restrictions. Estimates are presented of the national and regional electric power load growth rates, and the availability and range of prices for suitable fuels which meet pollution regulations. Operating limitations, performance, and cost characteristics of present-day and projected future steam power stations were established from a survey of available literature and from discussions with representatives of the public utility industry. Evaluations of the future need for condenser heat discharge methods other than once-through cooling such as cooling ponds, wet and dry cooling towers were made, and estimates are presented of the present and potential future operating and cost characteristics of towers and cooling reservoirs suitable for use with steam power plants.

The estimates were made for both the 1970 and 1980 decades; reliable predictions further in the future often are not based upon realistic assumptions. To conform with projections of advanced-design gas turbines, steam power plant technology levels have been defined for the 1970 decade, the early 1980's, and the late 1980's.

For the purposes of this study, comparisons among the competing power systems were made on a regional basis rather than on a national, statewide, or even utility level. Comparisons on a statewide or utility level would provide additional insight as to the potential for open-cycle gas turbines as a means of eliminating thermal pollution, but at a substantial increase in the level of effort. However, sufficient similarity exists within a regional area relative to the dominant type of utility fuel and its availability and price, load profiles, supplies of cooling water, and other factors considered by utilities in selecting a power system so that a realistic competitive analysis can be of benefit on this level. Therefore, estimates are provided for the average utility plant size, fuel cost, types of condenser cooling system, and steam plant characteristics in each of the six FPC-(Federal Power Commission) designated power regions of the US.

REVIEW OF NATIONAL ELECTRICAL LOAD GROWTH AND FUEL USAGE PATTERNS

Over the past twenty-five years the installed capacity of the electric utility industry in the United States has doubled every decade to the present level of approximately 343 million kw. Numerous surveys (Refs. 1 through 4) have indicated that the growth rate for this industry will accelerate slightly at least through the 1970 decade and should continue to produce a doubling of the installed capacity every 10 years through the last decade of this century. Thus by the end of 1975, the installed capacity of the utility industry is expected to reach 530 million kw (Ref. 2); as of early 1970, some 200 million kw of new generating capacity were already on order and scheduled for operation. Interestingly, the new capacity to be added exceeds the electric utility capacity in operation as recently as the beginning of 1963.

Projections of the yearly generating additions, based on data compiled in 1969 and 1970 by the National Electrical Manufacturers' Association (NEMA), are shown in Fig. 1a. If the capacity in service by 1978 does reach 625 million kw, as forecast, the average yearly growth rate will have been 8.0% over the decade from 1968 to 1978. The actual yearly additions are not constant but exhibit the historical trend of substantial year-to-year variations. One of the reasons for the cyclical behavior in orders for generating additions is the desire by the utilities to have additional protection in the event of possible delays (which have indeed occurred) in some of the very large advanced-design units which will be coming into operation in that period. An approximate indication of the reserve margins available is shown by the ratio of the year-end capacity to the summer peak load as shown in Fig. 1b. The ratio was extremely high in the early 1960's but is expected to remain at the 1.20 to 1.24 level during the 1970's.

Other forecasts of the growth of the electric utility industry in the US have been made, and the pertinent results of the most recent surveys are summarized in Fig. 2 along with the NEMA data through 1978. The NEMA and EEI (Edison Electric Institute) data shown in Fig. 2 include the hydroelectric capacity in the US as well as the thermal capacity (both fossil- and nuclear-fueled). A similar breakdown according to type of generation is available from the AEC (Atomic Energy Commission) data contained in a 1967 report to the President (Ref. 5) and Ref. 6. The Fig. 2 data illustrate (1) the proportion of the total industry capacity in hydroelectric and thermal plants through 1990 and 2000, respectively, (2) the expanding portion of the thermal capacity which will be provided by nuclear systems in the next 30 years, and (3) the large discrepancy which already exists between the nuclear forecasts presented in the 1967 AEC supplement and the 1970 FPC preliminary data for the forthcoming national power survey. Based on data in Ref. 4, which indicate that 32.5% of the 200 million kw of new additions already on order will be nuclear units, the 1970 FPC data appear to provide a more accurate picture of the extent of the nuclear penetration into the utility industry. Furthermore, since 60.4% of the additions are in other thermal units and only 7.1% in hydroelectric units, which in the past have provided 16 to 20% of the capacity, the diminishing role of hydroelectric plants due to the reduction in the number of desirable sites is also indicated. Reference 4 also provides an indication of the types of generation equipment which are forecast for the 1969-1978 period (see Fig. 1). The data confirm the trends in certain types of capacity forecast, at least for the next decade. In Ref. 6, it is forecast that conventional and pumped-storage hydroplants will account for only 12% of the 575 million kw of generating capacity in 1980, while fossil-fueled and nuclear-fueled steam plants will comprise 62 and 21%, respectively, and gas turbines and diesel plants the remaining 5%. Recent orders for gas turbines, however, have been running considerably ahead of this prediction and many sources now indicate that gas turbines will provide from 15 to 25% of utility installed capacity by the 1980-decade.

Most of the surveys also agree that the electricity generated by those power plants which will be in operation in 1990 will be more than four times the 1970 level. Thus, the number of kilowatt hours generated in thermal power plants is expected to increase from 1300 billion in 1970 to over 5500 billion in 1990. Furthermore, it is stated in Ref. 7 that, "Since the nuclear plants that will be in operation during the next two decades will be base-loaded and will operate at 75 to 80% capacity for most of their life, the generation of power by nuclear plants will grow from a predicted level of 68 billion kwhr in 1970 to 1290 billion kwhr in 1980, and to a level of nearly 4000 billion kwhr by 1990." Thus, by the 1980's, various references predict that nuclear power generation will account for from 30% and 70%, respectively, of the total power generated in thermal plants. Data from Refs. 8 through 10 tend to confirm these estimates, while a Bureau of Mines projection (Ref. 11) indicates that nuclear power will provide only 20% of the total utility energy requirements in 1980 and only 60% by the year 2000. The consumption of natural gas in the utility industry is projected by all five surveys to increase in spite of the diminishing reserves. The role of oil and coal as utility fossil fuels appears to vary among the various surveys. A summary of the role predicted for various energy sources in the utility industry from selected studies is presented in Table I. Additional data from other surveys and a discussion of the methodologies used in various studies is presented in Ref. 12.

ESTIMATES OF REGIONAL FUEL AVAILABILITY AND COST PATTERNS

Although the national picture with respect to installed generating capacity, electrical generation, and raw energy sources is of overall interest, significant changes will be occurring on a regional basis as well. The increase in electrical generation from thermal plants for each of the six regions in the National Power

Survey is shown in Fig. 3. Over the twenty-year period from 1970 to 1990, the West Region, which contains one-third of the contiguous United States, is predicted to experience an annual increase in thermal generation capacity of almost 10%, while the South Central, Southeast, and West Central Regions will experience annual growth rates of 7.2% or higher. Electrical generation growth from thermal plants will be the slowest in the more populous Northeast and East Central Regions.

Fuel Usage

Due to the emergence of nuclear energy, the growing concern and associated legislation for preserving the environment, and the presently predicted shortage of fossil fuels, the utilization of fuels for electric power generation within each region is expected to undergo dramatic changes during the remainder of the twentieth century. Traditionally, natural gas has been the dominant fuel source for power generation in the South Central Region, supplying over 95% of the energy requirements. Natural gas has also been the main fossil fuel in the West Region, supplying almost 75% of the energy for thermal generation, but since hydrogeneration has supplied about 50% of the total power generated, gas accounts for only about 37% of the total electrical generation in this region. In the remaining regions of the US, coal has been the principal fuel, supplying as much as 95% of the raw energy in the East Central Region during 1966. During this same year, coal was used to provide about 60%, 72%, and 74% of the raw energy for electric utility power generation in the Northeast, West Central, and Southeast Regions, respectively. Oil is used predominantly only along the east and west coasts and provides no more than 20% of any regional energy resource.

However, during the next twenty years, nuclear fuel is expected to carve out a substantial portion of the energy market in almost every region of the country. The FPC estimates of the nuclear penetration in each region, summarized in Table II, provide an indication of the shift likely to be experienced in the energy market during the next two decades. For example, in the East Central Region, the FPC estimates (Ref. 1) that nuclear fuel will share the raw energy market with coal by 1990. The estimates of one of the largest architect-engineering firms (Ebasco Services Incorporated) (Ref. 10) appear to provide essentially the same conclusions. Nuclear fuel will dominate in the Northeast, Southeast, West, and West Central Regions as well. Only in the natural gas-rich South Central Region will a single fossil fuel, natural gas, provide the bulk of the regional energy requirements. Of course, the ultimate utilization of each fuel source will depend upon the availability, deliverability, and the final relative prices of the fuels and associated power systems in each region. Thus a brief review of the extent and location of the different fossil fuel resources and the costs which may be incurred to bring them to power generation sites is appropriate.

Natural Gas

Despite the fact that the use of natural gas to generate electric power has been consistently criticized as an inferior use of the best fuel available to mankind, the consumption of gas by electric utilities has risen steadily. At present, gas accounts for about 26% of the total fossil fuel used in steam-electric power generation, and this gas comprises about 16% of the total gas used in the country. In fact, due to the recent growing concern over sulfur dioxide emissions and other types of air pollutants, the consumption of natural gas for utility operations and other manufacturing processes has accelerated even faster than anticipated. For example, the consumption of natural gas in 1968 in the Southeast Region was 60% higher than the 1966 level and had already exceeded the amount forecast in late 1965 and early 1966 by the FPC for use in 1980. The burgeoning utilization of natural gas has served to highlight the decreasing reserves of natural gas and has sparked appeals for additional exploration and production of natural gas (Refs. 13 and 14). Between 1954 and 1968, gas reserves were growing at a rate of 2.1% per year, against a consumption rise of 5.3% per year. As a result, the reserve fell from 29 years of gas supplies in 1954 to 14.6 years in 1968. Today, the recoverable proven gas reserves are down to about 11 years of gas supplies. Reserves are decreasing because their development has been discouraged by low well-head prices set by the FPC for gas that will be used interstate and not because the US or the world is running out of gas. On the contrary, if the estimated potential gas supplies as of 1968 of approximately 1227 trillion cubic feet (Ref. 15) were added to the proven gas reserves of 287 trillion cubic feet, there would be over 62 years of gas at the current annual level of consumption. These estimates may even be pessimistic, since the US Geological Survey (Ref. 16) estimates total proven and unproven gas reserves at 1700 trillion cu ft, while in Ref. 17 the gas reserves are placed at 2300 trillion cu ft. However, the development of these reserves may involve increased costs. Although the annual consumption of natural gas is expected to double in the next twenty years (Ref. 18), new techniques are being investigated to increase natural gas supplies. For example, the current work under the AEC Plowshare program could also result in substantial additions to the US gas reserves. The first Plowshare nuclear shot for gas stimulation was Gas Buggy in New Mexico in December 1967, while the second was Rulison in Colorado. Gas Buggy resulted in the production of 280 million cu ft of gas in 17 months or about three and one-half times the output of the nearest conventional gas well in a 10-year period. These results have prompted the AEC to promote a 3-year, $75 million program to solve the gas shortage. Potential output from stimulated fields is placed at a trillion cu ft within 10 years from now and ultimately 317 trillion cu ft. The quantity and distribution of proven and potential reserves of natural gas in various parts of the US, shown in Fig. 4, highlight the vast reserves that would be available for use in the South Central Region from those areas denoted as D, E, F, G, and J in the Potential Gas Committee Survey. Furthermore, there are proven reserves of 52 trillion cubic feet in Canada, and some west coast utilities are even exploring the possibility of bringing in gas from

South America (Ref. 19). Liquified natural gas will be available for use on the east coast, and studies are under way to determine the costs of bringing Alaskan gas via pipeline to the midwest or as LNG to the west coast.

In the early days of natural gas utilization after World War II, gas was sold to pipeline customers at prices as low as 3 to 5¢ per Mcf. (Gas prices are often quoted in cents per thousand cubic feet (Mcf) and gas from the well will generally average 1075 Btu/Mcf. However, after processing, the heat content of the gas available for sale is usually reduced to about 1000 Btu/Mcf, and this value has been used throughout this report to avoid confusion. Consequently, prices in ¢/Mcf are equal to prices in ¢/million Btu, another pricing quantity often encountered.) Recently, there have been examples where gas has been bought in the field in the South Central Region for more than 20¢ per Mcf, and one pipeline reportedly paid 28¢ per Mcf or 12¢ per Mcf above the in-line price set by FPC (Ref. 20). Although it is generally conceded that an increase in the well-head price of gas would end the reserves decline (Ref. 21), the price increase needed to stimulate the development of additional natural gas production is, at best, unclear at this time; estimates range from 2¢ per Mcf up to 10¢ per Mcf and above (Refs. 21, 22, and 23). An FPC staff study (Ref. 24) indicates that the prices of natural gas which would stimulate the production of supplies adequate to meet demand would range from 22 to 25¢/million Btu in the major gas-producing areas of Texas, Louisiana, and the Rocky Mountains. A large utility serving the South Central Region presently pays 20¢/million Btu for gas in Louisiana and 20½¢ in Texas and has contracts through 1977 which will provide for a slow escalation for Texas gas to 23¢ in 1980 through 1984. The same utility estimates that Louisiana gas, if purchased today, might cost 28¢/million Btu. However, they indicate that the break-even point for fossil fuel as compared with nuclear is in the neighborhood of 37¢ to 40¢/million Btu. Thus, on the basis of these estimates, the price of natural gas in the South Central Region can be expected to increase by about 5 to 10¢/million Btu over the next twenty years, and the price projections made by the FPC of about 30¢/million Btu with extremes of 21 to 38¢/million Btu by 1990 appear reasonable (Ref. 1). If the average figure is accepted, prices in various parts of the US can be estimated using a figure of about 1.1¢/million Btu as the cost of transporting the gas each 100 mi via the established network of pipelines. As a result, the average city-gate price of gas in the Chicago area would be about 40¢/million Btu as compared with estimated prices of about 43¢ for imported Canadian gas (Ref. 21) and about 50¢ in the New York market. This figure appears to fall within the lower range of prices (52 to 58¢/million Btu) for LNG imported from Africa reported in Ref. 25. Thus, most sections of the US would be accessible to some supplies of natural gas although at prices above today's unrealistically low level. Average gas costs reported in 1965 by US utilities on a regional basis are presented in Table III. The California electric utilities, which account for about 80% of the gas used for electric power generation in the eleven western states, are estimating a price increase at an average annual rate of 0.5% compounded through 1990 to about 36¢/million Btu or about 1¢/million Btu every five years (Ref. 1).

Thus, even though there may be other less costly fuels available, utilities in certain areas where stringent air pollution regulations are in force, such as southern California, will be compelled to use higher-priced low-sulfur, low-ash fuels such as natural gas.

Oil

Although petroleum products have never been a dominant energy source in the generation of electricity in this country, supplying only 6% of the energy used, its role in some geographical regions may be changing. Typically, residual oil, that fraction of the crude barrel which remains after the light products are distilled, is the petroleum product used in power generation plants. US refineries attempt to minimize the production of residual oil to meet their market demands for gasoline, jet fuels, and other high-priced products, and thus only about 7% of the crude oil processed in the US ends up as residual oil. In South America and Europe, however, residual oil comprises about 47% and 30%, respectively, of the crude barrel because of the different petroleum product market in these areas. As a result, over 85% of the residual oil burned by utilities in the US originates from Caribbean crude oils (Ref. 25), and 63% of the total residual oil is burned between Maine and Florida. Since the oil is delivered via tanker, the transportation costs are an important factor. Most of the residual oil burned elsewhere is of US origin. The sulfur content of Caribbean residual oil is typically 2.5%. Mid-continent residual oils have sulfur contents between 0.5 and 1%, while West Texas and California residuals will usually contain more than 1% sulfur (Ref. 25). Presently there is some excess supply of oil in the Middle East and South America, and in spite of the high tanker freight rates, this oversupply has no doubt led to the long-term contracts for high-sulfur oil at $1.60 per barrel (bbl)* or 24¢/million Btu delivered to utilities on the US east coast (Ref. 7). Venezuelan residual oil containing 2% sulfur is being offered to midwest utilities at $2.15 per bbl or 32¢/million Btu. However, such fuel oil will not meet most air pollution limits which at present require not more than 1% sulfur and ultimately will require as low as 0.3% sulfur. The cost of processing typical Caribbean residuals to reduce the sulfur content to 1% and 0.3% has been estimated at about $0.30 and $1.00 per bbl, respectively (Ref. 25). Substantial desulfurizing capacity has already been added or is under construction in the Caribbean to produce 1%-sulfur residual oil. However, in the northeast, there is a reported shortage of low-price residual oil, and prices of 50 to 55¢/million Btu have been reported for low-sulfur residual oil in New England.

Since the Caribbean residuals contain roughly 900 ppm of metals (of which 85% is vanadium) and thus require an abnormally high catalyst replacement rate, the costs for treatment are somewhat higher than if typical Middle East residual fuels were used as feedstocks. The US reserves prior to the Alaskan North Slope discovery were approximately 40 billion barrels, and the ratio of reserves to

* bbl will be used as the abbreviation for barrel in this report, based on 42-gallon capacity.

annual production rate was less than ten years. However, recent estimates indicate that the Alaskan North Slope reserves may alone reach 40 billion barrels, and this discovery has dulled the oil industry's interest in tar sands, shale oil, and coal as supplementary sources of oil supplies. Even more important, however, on a worldwide basis there are proven reserves of almost 400 billion barrels which would be sufficient to meet more than thirty years' consumption. Furthermore, the recent discovery of large oil fields in the North Sea is expected to offset some of the traditional imports from the Middle East. This discovery will produce additional surpluses of oil, but as the petroleum product consumption patterns in Europe and the rest of the world approach those of the US, the amount of residual oil will decline.

Imports of residual oil from Europe, the Middle East, Africa, and other sources, for utility generation will depend to a large extent on the tanker freight rates. Due in part to the Suez Canal closing and other factors, a shortage of tanker capacity has driven the present freight rates up to $3.05 per bbl for movements from the Persian Gulf to the US east coast. Such a freight rate is almost an order of magnitude higher than that existing under normal conditions (Ref. 26). As tanker capacity is added, though, the rates should return to near-normal rates, and imports of residual oil should increase. The interrelationship between fuel availability and fuel price can be clearly seen with respect to the present residual oil shortage in the east. The shortage of low-priced residual oil produced by the removal of import quotas and the deeper distillation processes together with the tanker shortage have driven the price of residual oil to levels twice that of last year. These prices are now attractive to the oil companies, and five major producers have indicated that they would make available about 400,000 barrels more per day. In Ref. 27. it is estimated that the price of residual fuel oil in the world market, including delivered cost in the US (based on 1968 dollars), is expected to trend downward moderately for residual fuel oil with no sulfur guarantee. The future cost of low-sulfur residual (containing less than 0.5%) is more uncertain, but assuming a current premium price on the order of 60¢ per barrel, its price is also expected to trend downward, based on 1968 dollars. Reference 7 estimates that the consumption of oil by US electric utilities will continue to grow from the current level of 250 million bbl/year to 644 million bbl/year by 1990. Due to the projected availability of low-sulfur crude from Alaska on the west coast, midwest, and possibly in the future on the east coast, adequate supplies of relatively low-cost residual oil are also predicted in selected locations.

Recent residual oil fuel costs for electric utility generation in selected areas of the country are shown in Table III from Ref. 1 and provide a basis for future projections. A utility in the South Central Region studying the use of residual oil estimates a late-1970-decade price of 38¢ to 40¢/million Btu.

Coal

According to Refs. 17 and 28, about 83% of the known economically recoverable energy reserves in this country are in the form of coal. These sources estimate that about 220 billion tons of coal are recoverable, sufficient to meet the coal needs of the country for more than 400 years at the present rate of consumption. However, many states and localities concerned with the potential harmful effects of certain air pollutants have passed sulfur oxide regulatory laws which currently restrict the sulfur content of coal and oil to be burned in selected industries to less than 1.0% and, in the future, to as low as 0.3%. Since the electric utility industry consumes about 60% of the total coal used in the country each year, the sulfur content of the coal reserves, the production and transportation costs for coal, and the economic feasibility of removing sulfur either prior to combustion or from the combustion products are of importance in assessing the future availability and prices for this energy resource.

Unfortunately, more than one-third of the total coal reserves in the US are high in sulfur content (> 1%). Much of the low-sulfur coal is lignite or sub-bituminous coal, with a heat content lower than that of bituminous coal which now represents over 95% of the present production. A summary of the sulfur content of US coal reserves according to tonnage and heat content is presented in Table IV. Data on reserves, however, can be misleading because much of the readily available coal, especially that located near the major eastern markets, is of high sulfur content. Figure 5 shows the major coal fields in the US, and Table V summarizes the distribution of low-sulfur coal, by type and by state. Data in Ref. 25 indicate that virtually all the low-sulfur coal west of the Mississippi is located in the Rocky Mountain states. Thus its use would require mine-mouth generation* or long-distance rail movements to generate the power near large load centers. East of the Mississippi, the largest reserve of 1%-or-less-sulfur bituminous coal is in West Virginia, but about one-fifth of this coal is contained in narrow seams and/or excessively deep mines which would substantially increase the cost of its recovery. In addition, a large fraction of the coal has chemical characteristics which make its use for steam generation unattractive, without extensive modification, due to different slagging characteristics. Furthermore, the bulk of the low-sulfur coal reserves in the Appalachian states is of metallurgical-grade coking quality and thus commands premium prices from such users as steel companies. For example, it is reported that the Japanese are paying \$12/ton or about 50¢/million Btu for southern Appalachian coking coal. A number of long-term contracts to provide this high-quality coal for export have been signed recently, further

* The availability of low-sulfur, low-cost coal for mine-mouth steam power generation stations in many locations in the arid western states where cooling water shortages exist has been responsible, in part, for the increased interest in the use of large, dry cooling towers by utilities and federal officials.

reducing the available supplies. Although such quality coal has desirable properties, it is suggested in Ref. 25 that a demand for similar-quality power plant coal would result in coal prices about $2 to $3/ton higher than high-sulfur conventional utility bituminous coal. Since the average value of coal at the mine is about $4.65/ton, the use of low-sulfur coal would increase the fuel cost about 40 to 65% or would add about 8 to 12¢/million Btu to the present price of high-sulfur fuel. These figures correspond to recent fuel cost prices, presented in Ref. 7, which indicate that low-sulfur coal, when available, costs about 40¢/million Btu delivered or about 7¢ higher than high-sulfur coal on the east coast. Recent long-term contracts (Ref. 29) signed by a large midwestern utility also indicate substantial premiums for low-sulfur coal with prices ranging from the mid- to high 30¢/million Btu level. Reference 7 also indicates that a large Canadian utility is constructing four 750-Mw power plants to burn oil because sufficient coal is unavailable from US mines.

An additional consideration in the assessment of low-sulfur as well as high-sulfur coal availability is the present shortage of available production capacity. Excluding the economics of transportation which are often a major component in the delivered coal price, there is a three-year lag in bringing new mines into production. Uncertainties concerning the severity of future air pollution regulations have also made coal companies reluctant to commit capital estimated at $10 to $12 per each ton of annual production capacity in new modern mines. Furthermore, the coal mining industry is having difficulty in maintaining adequate levels of production, productivity, and cost. Major technological advances in coal mining since World War II had largely offset the rising costs of labor and supplies and have been the major factor, until recently, in holding the mine price of coal at low levels. However, the present losses in coal mine productivity are quite severe; the reasons include a shortage of miners for underground mines, unstable union-management relationships, and the implementation of mine safety regulations. Increases in capital costs, wages, and supplies have pushed the price of coal upward and are expected to continue to do so. For example, Ref. 30 cites an increase in estimated production costs of 30%, from 18.9¢/million Btu in 1968 to 24.6¢/million Btu in 1975, for a new Pennsylvania-West Virginia mine opened in 1968.

The lack of uncommitted low-cost strip coal in the central and eastern US is also specified as a major reason for the coal suppliers' lack of interest in new business. Philip Sporn claims, in Ref. 30, that if bids were solicited for coal to supply four different new plants of 4 million kw each, only two US coal companies would have the necessary reserves to meet the long-term needs of these plants if these plants were to be located east of the Mississippi or in any one of the five states immediately west of the Mississippi. Because of these factors, the mine price of coal is expected, by many observers, to trend upward over the next decade, especially in the central and eastern US.

Although the number of mine-mouth plants has been increasing in recent years, most of the coal for utilities moves by rail, and the transportation costs can comprise as much as 50% of the delivered cost. Rail transportation costs rose sharply after World War II, and only the development and use of "unit trains" has produced important transportation cost savings. Further development of this concept into "high-speed shuttle trains" and the use of coal slurries via pipelines may result in some transportation efficiencies. Rates for these shipments have been increasing lately, and the shortage of rolling stock is expected to accentuate the trend. In the western states, rail costs to transport coal are about 3.6¢/million Btu/100 mi. In Ref. 1 it is stated that the transportation costs and the overall cost of coal are expected to rise in the Southeast Region, but in Ref. 27 the transportation component of coal costs is expected to decrease slightly.

Sulfur Removal from Coal

Several mechanical and chemical processes are available or have been suggested for the partial removal of some forms of sulfur from coal prior to its use in combustion devices. These methods of cleaning coal will, at best, result in only partial removal of the pyritic sulfur which is only a fraction of the total sulfur in coal, and therefore would not be applicable unless it would provide coal of acceptable quality through a simple means.

Gasification and liquefaction of coal to produce high-quality low-sulfur fuels suitable for use in utilities are being studied intensively for a number of reasons and ultimate market uses. Although a number of studies have been made and pilot plants are under consideration, it appears that gasification processes will produce a fuel whose costs are about 20 to 35¢/million Btu higher than that of the basic feedstock. Its ultimate use as an electric utility fuel will depend upon the economics and availability of alternative fuels, including nuclear power, at the particular locations. Descriptions and preliminary cost projections for processes suitable for the production of high-quality low- and high-Btu gas are described in Section VIII of this report.

Opinions differ widely as to the technical and economic feasibility of removing sulfur oxides from the flue gases of oil- and coal-burning power plants. A large number of processes have been advanced, and some are undergoing tests in power stations or experimental facilities. Preliminary results, however, are encouraging for a number of processes as it is estimated in Ref. 25 that, "The first generation of sulfur dioxide removal plants will operate with additional costs of only $0.75 to $1.00 per ton of coal fired." Furthermore, Ref. 25 states that, "As more becomes known about the technology of the various processes, second- and third-generation systems will incur added costs in the range of 20 to 25¢ per ton of coal fired." Contrary opinions concerning the economic and technical feasibility of stack gas processes are presented in Ref. 31 and elsewhere. It appears that it will be several more years before the final results on stack gas processes are available, but it is clear that these processes probably represent the pivotal factor in determining the future widespread utilization of coal in the utility industry.

If sulfur oxide stack gas cleanup systems or fuel pretreatment schemes become available at the cost levels predicted, then the long-term utilization of coal would be assured. If not, the fuel patterns in the utility industry may be changed drastically, since nuclear power would have almost an unchallenged position in the industry.

In summary, it may safely be concluded that the price of fossil fuels, with only minor exceptions, i.e., coal burned in the Rocky Mountains at mine-mouth plants where dry cooling towers would permit economic utilization of this coal, will be higher by from 5 to 15¢/million Btu in the next two decades in all geographic regions of the US. Substantially higher spot prices for low-sulfur coal will also be experienced until competitive forces tend to stabilize the market. These higher prices have contributed to higher overall busbar energy costs, and prompted most utility companies to apply to regulatory agencies for rate increases (Ref. 32). Recently in some areas, i.e., the Northeast, utilities have been permitted to pass along higher fuel costs to the customer without applying for continual increases. As a basis for comparison and for use in later phases of the study, prices projected for the various fuel sources, when applicable, are presented in Table VI.

REVIEW OF REGIONAL COOLING WATER AVAILABILITY AND THERMAL POLLUTION RESTRICTIONS

During 1965, the cooling of steam condensers in electric generating plants accounted for almost 60% of the total of 110,000 million gallons per day of water used in the US for industrial cooling, and for nearly one-third of the total water used for all purposes (Ref. 33). If present projections of the electric power industry growth rates are correct, and steam power plants remain the dominant type of power generation system, the once-through cooling requirements of this industry alone would reach a point, possibly by the year 2000, where one-half the average daily runoff of all rivers in the US would be needed. The cooling water shortage will accelerate as nuclear-fueled stations provide a larger portion of the electric demand, and with continuing growth in population and industrial productivity. Regional redistribution of population and economic activity toward the west will further aggravate the local water shortages and degradation in quality of water resources in many parts of the country. The availability of cooling water for future power plant sites as well as for additions to existing plants poses a major problem to the electric utilities.

The first national assessment (completed in 1968) of the adequacy of supplies of water necessary to meet all water requirements (domestic, industrial, agricultural, electric power, etc.) in each of 17 water resource regions in the US (see

Fig. 6) showed that shortages of natural runoff and ground water supplies have become serious problems in nearly half of the regions (Ref. 34). The pertinent results of this assessment are summarized in Table VII in which the relative severity of existing and emerging water management problems for each region is identified by an assigned rating from 1 to 4. Further, 11 of the 17 water resource regions surveyed (see Fig. 6) presently lack sufficient natural runoff to satisfy the year-round power plant condenser requirements with once-through cooling; by 1980 only coastal states will possess adequate supplies of cooling water. In Ref. 35, it was estimated that in 1972 about 45% of the total US power generation capacity will require some type of supplemental cooling apparatus, such as cooling towers, to alleviate the demand for condenser cooling water. By the 1980's, the same reference estimates that almost 70% of the installed capacity will use supplementary cooling devices, and only those power stations convenient to the oceans will be able to reject heat in once-through cooling systems.

However, even in those local areas where adequate cooling water exists, unlimited use of rivers, lakes, and estuaries for cooling will not be permitted since effective action was taken through the Water Quality Act of 1965 to control waste heat discharges as well as other types of water pollutants. As a result of this legislation, water quality standards are being set and implemented for all coastal and interstate waters.

All 50 states have submitted water quality standards containing temperature criteria to protect designated water uses, particularly aquatic life propagation. Standards for temperature changes and maximum temperature limits vary from state to state. Table VIII summarizes the temperature criteria proposed by the individual states in the South Central Region, as of the end of 1968, for interstate and coastal waters. As of April 1970, 20 states did not yet have their water temperature standards approved in entirety (Ref. 36). Most states have established 68 F as the maximum allowable temperature and from 0 to 5 F as the maximum allowable change in temperature for streams with cold-water fisheries. For warm-water fisheries, the maximum allowable temperatures are generally in the range of 83 to 93 F, and the maximum allowable rise is in the range of 4 to 5 F (Ref. 33).

Although the importance of the mixing zone in determining the amount of heat that may be discharged to a water body is recognized, allowable limits for this parameter have not been clearly defined in all cases, and as a result a number of utilities have delayed complying with the specified temperature standards (Ref. 36). Furthermore, the heat-accepting capability of a lake, reservoir, or stream is difficult to estimate because of the many variables involved, and some utilities may be anticipating upgrading revisions in the standards. Several studies relating to the ability of water bodies to dissipate heat to the atmosphere have been completed; however, the apparent results of these studies vary considerably among water bodies and geographicl locations. Since the cooling water discharged from power generating plants using once-through cooling is often heated 10 to 25 F above the intake water, both substantial local heating and high temperature levels are sometimes produced in cooling streams, especially during low water conditions. However, in other situations, stratification may occur without substantial mixing and may extend for long distances. In another instance, the temperature of the Monongahela River in August along a 40-mi stretch upriver from its confluence with the Allegheny averages approximately 85 F, with local temperatures as high as 95 F, as shown in Fig. 7 (Ref. 37). Although much of the heat input is from a concentration of industrial plants located near the larger cities along the river, measurements and even model simulations have indicated that 15 to 20 mi can often be required for a river to return to normal temperatures after waste heat discharges from a large power-generation facility (Ref. 38).

Attempts have been made in a number of studies to predict the number of power stations that will be required to meet the electric power demands of the United States to 1990 and the number of these stations that will need auxiliary cooling systems. In one such study (Ref. 33), it is predicted that by 1990 158 stations of a total of 492, of 500-Mw capacity and above, will require cooling towers. In making these projections it was assumed that future stations in the coastal areas and in the vicinity of large lakes and streams would use once-through reservoirs or cooling ponds. Another source (Ref. 39) indicates that these assumptions are optimistic and that the number of stations requiring cooling towers will be even greater than those estimated in Ref. 33. A major manufacturer of steam-electric power plants predicts that by 1990 the typical station in some locations might be required to utilize nonevaporative cooling towers due to the unavailability of cooling water. Another manufacturer concedes that dry towers may be more frequently used in specific locations but they would not be typical even by 1990. It is the opinion of this manufacturer that it would be generally cheaper to transmit power over a longer distance to the load center if availability of cooling water at the first station site is a problem.

A recent survey of the proposed plans by utilities to meet thermal standards indicates substantial increases in the use of supplementary cooling devices (Ref. 36) The 69 companies which replied to the survey reported that 164 stations presently use some form of tower or ponds. All but two of the larger utilities (with capacities greater than 200 Mw) responding to the survey were located in the southwest or arid plains states.

However, in Ref. 40, estimates are made of the potential utilization of supplementary cooling devices under three different assumptions of thermal quality standards. This analysis indicates that less than 8% of the approximately 200 million kw of installed major thermal generation (500-Mw station capacity and above) in the US would use cooling ponds and about 13% would use cooling towers. Under more stringent thermal quality assumptions, almost 90% of the new major capacity added in the US during the 1970's and 1980's would utilize towers or ponds. The Northeast and East Central Regions would utilize cooling towers for from 40 to 60% of the new plant capacity in the next twenty years but there would be relatively minor utilization of cooling ponds for about 20 to 30% of the new capacity. Cooling ponds and cooling towers would be used in at least 50% of the new capacity in the other four regions of the country. Thus, it may be concluded that only those utilities close to ocean water or large rivers such as the Mississippi will be permitted to use once-through cooling systems. Most areas in the South Central Region, as noted in Fig. 6, would require cooling ponds or towers (Ref. 41) and are planning to use these alternatives.

ESTIMATES OF PRESENT-DAY AND FUTURE CONVENTIONAL STEAM POWER PLANT PERFORMANCE AND COST CHARACTERISTICS

For many years, steam power systems have maintained an overwhelming position in the field of electric power generation. Consequently, this type of power system is and will continue to be the standard against which the feasibility of alternative methods of generating electric power must be compared. This section includes a description and discussion of present-day steam power plant characteristics and limitations. Also included are estimates of the performance and cost characteristics of advanced steam power plants which might be built with technology potentially applicable in commercial configurations during both the 1970 and 1980 decades.

Unit Capacities

The unit capacity of a base-load or cycler steam power plant purchased by an electric utility is selected after detailed analyses which include the effects of power generation, transmission, and distribution costs, as well as reliability and availability, on the total cost of providing power. In recent years, the expansion of intertie systems has permitted utilities to take advantage of the economies of scale in purchasing steam power units, and the average-size unit is rapidly increasing. As a result, a rough rule of thumb which has been applied to past base-load capacity additions is that the capacity of new units should not exceed 10% of the total utility system generating capacity. For a large electric utility system, such as TVA or American Electric Power, this 10% rule would dictate the selection of 1000-Mw units today. However, these large systems are not representative of the US electric utility industry as only three utility systems have capacities close to 10,000 Mw. It may be noted that a steam power station may consist of one or more units and the units may be of different sizes.

Data from Ref. 2, on scheduled additions of steam power generating capacity by years, is presented in Table IX. These data are based on scheduled dates of commercial operation as of October 1, 1969. A general trend of increasing unit size with time may be observed for both conventional (fossil) and nuclear steam power plants. Conventional units scheduled for operation this year and for the early 1970's have average capacities of approximately 300 Mw and 500 Mw, respectively. This trend is expected to continue, and by the 1980's average steam power plant units may reach output capacities of approximately 1000 Mw. Data in Ref. 1 appear to substantiate the present average unit size and the trend to larger units. Thus, unit sizes of 500 and 1000 Mw may be considered representative for the 1970 and 1980 decades, respectively. It may be noted that the average size of nuclear units is running ahead of conventional units. Average nuclear unit sizes would be expected to reach the 1000-Mw level by the mid-1970's. The distribution of unit size for 1968-to-1971 fossil- and nuclear-steam power plant installations (Ref. 42) is shown in Fig. 8. However, the availability of large (800 to 1300-Mw fossil-fueled steam plants has been discouragingly low and until there are units which reach the previous levels of availability, there will be a pause in the trend to increasing unit sizes. Although there are some differences between the data in Fig. 8 and the data in Table IX, both sets of data highlight the trend toward larger-capacity units. The scheduled additions of steam power plant generating capacity (based on the same data included in Table IX) are presented in Table X for the six power regions of the US.

The capacity of conventional fossil-fueled units scheduled for operation in late 1969 for almost all regions will average less than 500 Mw, whereas nuclear unit size additions will average approximately 900 Mw or more in all regions.

Steam Conditions

Historically, improvements in steam power system technology have permitted increases in steam temperature and pressure, thereby resulting in increases in station efficiency and lower net fuel charges. Generally, however, as technology advanced, more expensive equipment was required to contain the steam so that capital cost increased significantly. Until recently, increases in unit size together with higher specific power levels achieved from the higher operating conditions and the resulting reduced fuel charges always outweighed the higher capital costs due to the advanced steam conditions. At the present time, the highest practical steam temperature in new plants appears to have reached a plateau of approximately 1000 F with a single reheat to the same approximate temperature. A second reheat is not justified in present-day units because the capital costs would outweigh the marginal saving in fuel charges at present levels of fuel costs (Ref. 43). Presently, 2400 psig is the most common steam inlet pressure to the high-pressure turbines in medium-size units, although several large units incorporating supercritical boilers operating at 3500 psig are now operating, under construction, or in the planning stages. A tabulation of units under construction and scheduled for operation by 1973 in Ref. 44 further illustrates the diversity of steam conditions in present plants. Data on the characteristics of medium- and large-size units installed by TVA show that 2400-psig steam pressure was used on unit sizes up to 700 Mw, and 3500-psig steam pressure was used on units of 950 Mw and larger

(Ref. 44). Operating problems with supercritical units, however, have not been completely overcome, so that the 2400 psig/1000 F/1000 F steam cycle would be considered representative for present-day and 1970-decade steam power plants (Ref. 45).

Equipment manufacturers as well as architect-engineering firms were questioned specifically regarding the long-term (10 to 20 years) trends in steam conditions (see Ref. 29). Surprisingly, there was unanimous agreement that there would be no increases in steam temperature beyond approximately 1000 F, nor increases in steam pressures beyond approximately 3500 psig. This general conclusion is also stated in Ref. 46. Previous experience with units rated at 1100 F to 1200 F (e.g., the Eddystone Station of the Philadelphia Electric Co. and the Bergen Station of the Public Service Co. of New Jersey) has not been encouraging, and these stations have been downrated to approximately 1000 F to improve their availability. Although these units were installed primarily to determine the reliability and performance obtainable at higher steam conditions, their continued operation at high temperature could not be justified economically. The basic cost problem with high-temperature operation arises because austenitic-type stainless steels must be used above 1000 F. Since austenitic steels are considerably more expensive than ferritic steels, high-temperature boilers would be very expensive and the incremental cost of these boilers over boilers constructed from ferritic steels generally would not be offset by the incremental fuel saving. For example, several boiler manufacturers (Refs. 47 and 48) estimated that a boiler designed to generate 1100 F steam would cost approximately 8% to 10% more than a 1000 F boiler which generally costs about $35/kw. Similarly, steam turbine manufacturers indicated that steam turbines designed for 1100 F and 1200 F would cost approximately 10 to 15% and 20 to 25%, more respectively, than steam turbines designed for 1000 F steam. Several equipment manufacturers, as reported in Refs. 43 and 49, conducted analytical tradeoff studies which indicated that these increased equipment costs would not be justified unless the fuel cost were to exceed approximately 45 to 50¢/million Btu. This result can be substantiated by comparing the fuel economics of a 2400 psig/1000 F/1000 F cycle with a 4000 psig/1200 F/1200 F cycle. The total incremental cost for the high-temperature system was estimated to be approximately $17.3/kw more than the cost of a 1000 F system, whereas the differential efficiency between the high- and low-temperature systems was estimated to be approximately 3.4 percentage points. Using a 14% fixed charge, 70% load factor, and the figures stated above, it was calculated that the 1200 F system could be justified only at fuel charges greater than 53¢/million Btu.

Consequently, 1000 F would be considered representative as the maximum cycle temperature for the 1980-decade as well as the 1970-decade conventional steam stations. Since the 3500-psig supercritical cycles do show a slight increase in efficiency relative to cycles operating at 2400 psig, and since operational problems experienced with these supercritical pressures should be overcome by the 1980 time period, the 3500-psig pressure level would be considered representative for the 1980-decade steam stations.

It may be noted that boiler manufacturers have indicated (see Ref. 49) that the fuel type or its cleanliness has relatively little effect on the maximum allowable steam temperature, although it does have a significant effect on boiler cost.

Thus, during the 1970 decade, representative steam systems would be 500-Mw units operating with 2400 psig/1000 F/1000 F steam conditions. The steam turbines would be 3600-rpm, tandem-compound, 4-flow machines with 30-in. last-stage blades. There would be provisions for 7 stages of extraction for feedwater heating. In addition, steam would be extracted from the low-pressure crossover pipe at approximately 185 psia to supply steam to the two boiler feed pump turbine drives. The boiler and boiler auxiliaries would be completely enclosed in a building in most areas of the country. Each boiler would be pressurized by two half-capacity forced-draft fans, and would be equipped with an air preheater, soot blowers, economizer, and a 100%-capacity, condensate polishing demineralizer to prevent carry-over of dissolved solids. The steam would be condensed in a single-pressure, two-pass, twin-shell condenser wherein the tubes would be perpendicular to the turbine centerline. Wet cooling towers would be used rather than once-through cooling. Steam from the boiler feed pump auxiliary turbine drives would also be condensed in the main condensers.

The design of the 1980-decade units, averaging 1000 Mw and operating with 3500 psig/1000 F/1000 F steam conditions, would be similar to that described above except for the size of the last-stage turbine blades. A detailed arrangement of equipment of a representative large coal-fired station is presented in Fig. 9.

Performance

Although the steam temperature is projected to remain essentially constant throughout the entire 20-year time period under investigation, slight performance improvements are anticipated because of the increase in pressure level between 1970-decade and 1980-decade systems and also because of slight improvements in steam turbine and boiler efficiencies during these time periods. Projected performance characteristics for 1970- and 1980-decade steam power stations are presented in Table XI. Station efficiencies are presented for design-point operation and for operation at 70% load factor, and include allowances for all auxiliaries. Differences in boiler efficiency and power station auxiliary power requirements account for the differences in the net station efficiencies among the coal-, oil-, and natural gas-fueled power systems. The oil-fired power systems exhibit slightly higher efficiencies than coal-fired systems, primarily because of the lower auxiliary power requirements. Power systems fueled with natural gas exhibit lower efficiencies primarily because of the greater moisture losses in the boiler. In actual practice the level of sophistication in the plant design and the ultimate efficiency and cost of the station is related to the cost of fuel, and where low-cost fuel is available plant efficiency would be decreased accordingly.

Station Costs

The cost of building a modern conventional steam power system is dependent upon a great many factors including unit size, type of fuel burned, sophistication of system design, location, type of air pollution controls (if any), type of construction (indoor or outdoor), and the type of heat rejection system. Representative total station costs given in Ref. 50, based on discussions with an architect-engineering firm, for indoor present-day large-capacity base-load coal-fired stations range from about $165 to $175/kw, and for gas-fired stations from $140 to $150/kw. Details describing the cost breakdown for each type of station are presented in Section VIII of this report. Detailed estimates performed for this study based on 8% interest rates and 15% fixed charges indicate that the cost will remain about the same through the 1980's. A method of generalizing these costs for each region will be described in Section VIII. These costs include all indirect items such as engineering, design, and escalation and interest during construction as well as direct component cost. It has been estimated that complete outdoor construction would result in costs $5 to $10/kw lower than those quoted above (Ref. 29).

A summary of costs for the TVA 950-Mw Bull Run 1 coal-fired steam plant taken from Ref. 44 is presented in Table XII and the total of $157.64 essentially verifies the values quoted above. Although the Bull Run cost figures include all direct and indirect costs such as interest and escalation, it should be noted that TVA building costs, in general, are often not representative of the electric utility industry because of their ability to borrow money at low rates and to buy equipment at low prices. Also included in Table XII is a summary of costs from Ref. 45 for a 1000-Mw nominal station containing two 500-Mw coal-fired units. The summation of the direct costs amounts to $124.72/kw, whereas when the indirect costs are included the total cost is $176.95. It should be noted that power plant costs quoted in the literature often are summations of the direct costs only. It is obvious that costs significantly lower than those quoted above, on the order of $80 to $120/kw, do not include the indirect costs and therefore do not reflect the true cost of building a steam power plant.

As previously mentioned, steam power plant specific cost depends upon unit capacity. A typical variation of specific cost ($/kw) with unit capacity (taken from Refs. 27, 51, and 52) is depicted in Fig. 10 for conventional (fossil-fueled) steam power systems together with that for the nuclear plants. It may be noted in Fig. 10 that significant cost savings can still be achieved by building steam power plants in sizes greater than 1000 Mw. Data from a local utility (Ref. 52) for oil-fired base-load, oil-fired cycler, and nuclear units are also shown in Fig. 10, and for the most part verify the level of costs for fossil fuels taken from Ref. 27. It may be seen that cycler-type plants cost approximately $20 to $24/kw less than base-load type plants.

Although the costs quoted above pertain primarily to present-day systems, they are expected to apply to future systems as well. Costs will tend to increase due to the addition of equipment for control or elimination of air pollution and water thermal pollution but will also tend to decrease due to the increased use of larger-capacity units.

ESTIMATE OF PRESENT AND FUTURE PERFORMANCE AND COST CHARACTERISTICS OF ALTERNATIVE METHODS FOR COOLING CONDENSER WATER DISCHARGES

The previous discussion indicates that the availability of naturally occurring waters suitable for large condensing water systems used in steam-powered electric generating plants will be limited in many areas of the country. Where sufficient quantities of cooling water do exist, their use may be restricted because of the temperature effect on these waters from a once-through condensing system. As a result, alternative methods to once-through systems using river or sea water are being given greater consideration in many areas of the country for condenser cooling in steam-powered generating plants. The types of cooling systems which are in use at the present time or considered feasible in the near future include once-through systems using river or sea water, cooling ponds and reservoirs, spray ponds, spray cooling canals, and wet and dry cooling towers. A brief description of the operating principles, performance limitations, area of maintenance requirements, cost characteristics, and effect on the performance of steam systems when using these alternative systems are presented in the following paragraphs.

Description of Alternative Systems

Once-Through Cooling

The most common method used for the removal of heat from steam power plant condensers in water-rich sites in the United States is once-through cooling (see Fig. 11a). Cool water is diverted from a river, natural lake, or estuary and pumped through the power plant condenser tubes, thus condensing the steam working fluid on the outside of the tubes. The river or lake cooling water is generally heated from 10 to 25 F in the condenser, depending upon the number of passes of the cooling water. Single-pass condensers are normally used in once-through systems to minimize their size and surface area. However, to reduce the temperature rise of the water, large flows of cooling water are required. For example, to cool a typical 1000-Mw fossil-fueled plant and limit the temperature rise of the cooling water to 10 F, about 2100 cu-ft/sec of water would be required. Where the natural flow of water available for cooling over the entire year may not be adequate to meet the requirements, two-pass condensers can be used, resulting in higher cooling water temperature rises, higher turbine back pressures, and reduced plant efficiency. Although once-through cooling is usually the most economical to install and operate, the intake and discharge channels must be carefully located to prevent recirculation of the warm discharge water. Sometimes skimmer walls, diffuser systems, or long intake and/or discharge lines are used, and often costly models are needed to predict the hydraulic and thermal flow patterns (Refs. 53 and 54). All these items add to the costs of the once-through cooling system.

Once-through cooling using sea water is perhaps the second most common type of condenser system and much the same factors must be considered in its design. However, higher-quality corrosion-resistant materials are required in sea water systems, as well as long intake and discharge lines to maintain an adequate supply of water during tidal movements. These factors can raise the cost of a sea water installation considerably. According to data in Refs. 55 and 56, the cost of the condenser alone can be 25% more than conventional once-through river units. Discharge conduit costing $500 to $1000 per ft or capital costs from $500,000 to $1 million per 1000 ft are indicated in Ref. 55. Discussions with an architect-engineering firm confirms that once-through sea water systems could add as much as $5 to $10/kw of installed capacity (Ref. 29) in extreme cases.

Cooling Ponds or Reservoirs

A cooling pond or reservoir is a man-made body of water into which the warm condenser discharge is pumped so that it may be cooled and eventually circulated through the condenser. Although it is stated in Ref. 57 that the natural rolling topography of much of the nation is favorable for the formation of man-made lakes to retain the water during high runoff periods, many cooling ponds are found in the hilly Southeast and lower portions of the East Central Regions of the US. A lake may be constructed by placing an earth dam at the junction of one or more small streams and allowing the runoff to fill up the low area. In some instances, it may be necessary to construct the lake and retain the water using earth dikes. With water supply to the lake from runoff, the drainage area necessary is dependent on the natural and forced evaporation rates, rainfall, and expected periods of drought. In general, the drainage area required is about ten times the lake surface area and two or three years may be required for initial filling of the cooling pond. The pond surface area required will depend upon the temperature of the condenser discharge and other factors, but in general 1 to 2 acres of water surface area are used per megawatt of generating capacity. However, the difficulty of installing a cooling pond in the minimum area very often requires that the utility must buy from 2 to 3 times the pond acreage in the surrounding area for an adequate site. Kolflat, in Ref. 57, estimates the cost of a cooling pond at about $2.50/kw for a 1000-Mw plant with the cost of land accounting for about 40% of this total, 40% to clear the land, and the remaining 20% for the dam and spillway. These costs appear reasonable if the land for the entire site costs only several hundred dollars per acre. However, it is suggested in Refs. 56 and

58 that the cooling pond costs may be somewhat higher, especially if a dual-pressure condenser were used to minimize the pond area required. The accessory equipment and a detailed listing of the costs associated with the construction of a cooling pond are presented in Ref. 58. Cooling ponds are often classified according to the circulation pattern and temperature distribution as completely mixed, flow-through, or internally circulating, as described in detail in Ref. 56.

Spray Ponds

The area requirements of a cooling pond can be reduced by at least an order of magnitude if the water used for cooling is sprayed into the air. The evaporation rates are enhanced and cooling occurs more rapidly, since the water droplets remain in intimate contact with the air for longer periods of time. The spray nozzles are usually located up to 10 ft above the surface of the pond and must be carefully placed with respect to each other and according to the prevailing winds if the spray pond is to be effective. Extensive data on spray pond design for utility application is not available although some tests have been under way to verify their performance. The costs of spray ponds, including the piping, nozzles, pumps, and installation are estimated to be about $2.50/kw in Ref. 56.

Spray Cooling Canals

A potentially low-cost variation of the spray pond for condenser cooling is the use of spray cooling canals. In this system the condenser cooling water is directed through a canal where it passes through a series of floating spray nozzles. The water is cooled by evaporation and convection through each pass of the nozzles until the desired approach to the ambient wet bulb temperature is achieved. The basic spray nozzle unit is comprised of a pump and four spray heads with interconnecting straight-line piping. The entire unit floats in the water and is moored in place. It is claimed that flotation eliminates the need for special and expensive basins, foundations, and complex pump and piping distribution systems such as those required for spray ponds (Ref. 59). In addition, a much coarser droplet size of 3/8 to 1/2 in. dia is produced which eliminates clogging of the spray nozzles and reduces carry-over loss. Designs using a multiple of spray units arranged in rows across a channel 160 ft long have been made that will accommodate the waste heat from a typical 1100-Mwe steam plant. Such a system is being installed in a utility in the northeast and pilot tests are in progress at other utilities in the east central, south central, and south atlantic regions. The costs of the powered spray modules only are estimated at approximately $2/kw (Ref. 59). Added costs for channel construction, field assembly, moving materials, etc., are included in Ref. 60 and indicate only another $0.50 to $1.00/kw for these items. The cost of installing a spray cooling canal system is apparently a strong function of the cost of constructing the cooling canal in many installations.

Wet Cooling Towers

In a wet tower, water is cooled largely by evaporation of a portion of the circulated flow. The evaporated water is absorbed by the air flowing through the cooling tower which is in direct contact with the circulated water. During summertime operation, when the air temperature is high, evaporation provides for the larger fraction of the heat transfer to the air. For example, the waste heat from a 1000-Mw fossil-fueled plant would require the evaporation of approximately 6750 gallons of water each minute of operation (Ref. 61). A nuclear-fueled plant, due to the larger quantities of waste heat that must be rejected, would require the evaporation of about 10,125 gallons of water each minute of operation. During the cooler months of the year, the evaporation quantity would be reduced to about 5400 gallons per minute and about 8100 gallons per minute for the fossil- and nuclear-fueled plants, respectively. The remaining waste heat is removed by sensible heat transfer within the water cooling tower due to the temperature difference between the water and air. Of course, the portion of the circulated cooling water that leaves the water cooling tower system as evaporation must be replaced and represents the largest portion of the water demand for siting considerations in a power plant with wet towers.

The most often-used cooling tower arrangement is the total recirculation system. With this system, the tower to be installed must satisfy the total waste heat dissipation requirements of the generating plant regardless of the season of the year or the generating load. Such systems are usually termed "closed-circuit" recirculation systems since the water used for cooling remains within the system and only evaporation, drift, and blowdown losses must be replaced from the water source (see Fig. 11b). Open-circuit, once-through systems are used where sufficient water is available to supply the plant's requirements but water discharge temperatures must be limited (see Fig. 11c) (Ref. 61). Temperature reduction of the discharge flow from the plant condenser is accomplished in the cooling tower and the flow is then returned to the water source. Sometimes the entire flow is pumped through the cooling tower or the tower is used to cool only a portion of the flow which is then mixed with warm water from the condenser discharge to meet the temperature standards. Other configurations are also available, as described in Ref. 61, to match the power plant cooling requirements and remain within regulated temperature limits over the entire year.

Wet towers can be further classified as either mechanical- (forced or induced) draft or natural-draft towers referring to the means of providing the air circulation through the tower. In the mechanical-draft design, large-diameter fans driven by electric motors induce the air through the circulating water which flows over splash surfaces that are provided to interrupt the flow of water and increase the contact period between the air and water (see Fig. 12a). A number of different arrangements of the airflow and cooling water streams are possible (i.e., cross-flow, counterflow) as indicated in Ref. 56. Natural-draft wet cooling towers (Fig. 12b) utilize concrete chimneys to induce air through similarly arranged heat transfer surfaces.

The total water requirements necessary for the installation of a wet cooling tower include that portion needed for evaporation, plus an amount for physical water losses known as "drift" due to droplets entrained in the leaving airstream, and an amount which must be bled from the circulating water system to limit the concentration of dissolved solids in the circulating water. Estimates of the total water demand for 1000-Mw fossil- and nuclear-fueled plants are presented in Ref. 61 and indicate that from 7700 to 20,250 gallons per minute could be required depending on the design temperature range of the tower and allowable chemical concentrations of the makeup water.

Several factors are important in determining the size of a wet cooling tower: (1) heat load, (2) range, (3) approach, and (4) wet bulb temperature. Normal approach temperatures of 10 F are used in mechanical-draft towers with 15 to 20 F typical in natural-draft towers. Due to the higher drafts obtainable with mechanical-draft towers, higher packing water loads are possible relative to those in natural units and the ground area requirements are usually only one-half to one-third of equivalent-capacity natural-draft units. However, mechanical-draft units must be located further away from the plant than natural-draft towers and thus require considerably more connecting piping. Typical dimensions are given in Ref. 56 for a 1000-Mw nuclear plant and indicate base areas of 2.67×10^5 ft^2 and 1.33×10^5 ft^2 for the natural-draft and mechanical-draft units, respectively. Thus, two natural-draft units each 412 ft in diameter could be used, while 10 square cells arranged in a configuration 115 ft wide by 1150 ft long would be used for the mechanical-draft units. Typical heights would be 60 ft for the mechanical-draft tower and over 400 ft for the natural-draft unit. In addition to the area required for the tower itself, land is also required for reservoirs, blowdown ponds, and pumping and storage areas. The towers must also be located with proper spacing between each other to avoid damage and possible destruction during high winds such as that which occurred in England several years ago. The location of some of the larger mechanical- and natural-draft tower installations in the US are shown in Fig. 13.

Dry Cooling Towers

Dry or nonevaporative cooling towers reject the waste heat in the warm condenser discharge flow entirely through convective heat transfer, depending upon the temperature difference between the heated water and ambient air. Dry cooling towers have been used extensively in the chemical processing and petroleum industry but, except for small installations, are not used in the US electric utility industry. However, dry cooling towers have been used in steam generating plants with capacities of 200 Mw in Europe and South Africa (see Ref. 62). There are two basic types of air-cooled condensing systems -- the indirect system and the direct system. With the indirect dry-type cooling tower system (see Fig. 14), a spray condenser is used at the turbine exhaust with circulating water sprayed into the unit to condense the steam. The heated water is then circulated from the spray condenser to a tubed cooling coil in the dry cooling tower over which the cooling air is passed and the warmed air is discharged to atmosphere. For each pound of condensate leaving the condenser hotwell and going to the power plant cycle, about 25 to 40 pounds leaves the hotwell and is pumped through the air-cooled heat exchanger coils. The pressure in the piping and cooling coil tubes is maintained above atmospheric pressure by pumps to prevent air leakage into the system. In some instances, power recovery is proposed by letting the pressure down through a turbine to the pressure level used in the spray condenser. At the present time, dry cooling towers using the indirect condensing cycles (sometimes referred to as the Heller system) are being studied extensively for potential use in power plants (Refs. 64 and 65). The use of a direct air-cooled combining system in which the exhaust steam is piped to tubed condensing coils for air cooling is not considered feasible for large power generating plants beyond 200 Mw because of the large piping and tubing requirements when forced to operate under low absolute pressure conditions.

Dry towers are possible using either mechanical draft or natural draft to provide for air circulation over the cooling coils. Area requirements for each type are estimated in Refs. 62, 63, and 64 and depend upon a number of design factors such as the condenser operating pressure, ambient air temperature, etc. Typical areas are about 300 sq ft and 900 sq ft for each Mw of fossil-fuel plant capacity for the mechanical- and natural-draft types, respectively. Nuclear-fueled plants would require about 60% more area than fossil-fueled plants per Mw of output for the same design conditions. Reference 62 contains an outstanding summary of dry-cooling tower technology as well as the estimated costs and design parameters for cooling towers suitable for conventional nuclear- and fossil-fueled steam plants in 27 US locations.

Performance Penalty with Alternative Cooling Systems

One of the principal parameters which governs the thermal efficiency of a steam power plant is the back pressure at the turbine exhaust. This pressure and thus the temperature at which the steam condenses is a strong function of the temperature of the condenser cooling medium (usually circulating water). Although the condensing temperature of the steam could theoretically approach that of the cooling water in typical systems, the condensing temperature is usually some 25 to 35 F higher than that of entering cooling water to provide the proper economic balance between condenser cost and plant efficiency. For example, with cooling water available at 55 F, a 90 F condensing temperature (equivalent to a 1.42-in. Hg abs back pressure) might be maintained in the system. Thus with higher cooling water temperatures as would be experienced in the summer months, the steam condensing temperature would have to increase accordingly and the plant efficiency would be reduced. Cooling water temperatures for power plant use in the US range from 32 F to as high as 95 F. It appears that average cooling water and condenser temperatures would be about 65 F and 100 F, respectively. The use of wet or dry cooling towers would therefore require somewhat higher condenser design temperatures if the heat is to be rejected to ambient air. In Ref. 56, condenser temperatures 10, 15, and 20 F higher than those of once-through cooling systems are suggested for mechanical-draft wet towers, natural-draft wet towers, and dry towers, respectively. Higher condenser temperatures are indicated in Refs. 62 through 66 for the dry tower systems. Optimum economic condenser design pressures were found to vary from 8.5 to 15.6 in. Hg abs (corresponding to about 155 to 180 F) in a study to determine the applicability of mechanical-draft dry cooling towers in steam power plants suitable for various climatological conditions around the US (Ref. 64). In a study of dry cooling systems for location all over the US, the optimum initial temperature difference was found to be from about 50 to 60 F. Thus the condenser temperature would be about 130 to 140 F for a design dry bulb temperature of 80 F. Generally, large central power station steam turbine generators are limited to operation at back pressures below 5 in. Hg. Present designs of large turbines are based on achieving maximum guaranteed kilowatt output at back pressures of 3.5 in. Hg, with reduced capability for back pressures above 3.5 in. Hg. If prolonged operation were attempted at back pressures above 5 in. Hg, some problems would be anticipated due to bucket heating and vibration, thermal distortion of the exhaust hood and diaphragms, and abnormal stress due to thermal cycling (Refs. 62 and 64).

A turbine which operates satisfactorily at back pressures above 5 in. Hg could possibly be achieved by several alternative methods: (1) eliminating the last row of blades in the low pressure turbines in present turbine generators, (2) designing a large turbine to operate at high back pressure by using somewhat shorter blade lengths in the last stages than present stages but opening up the flow passages to permit higher steam flows, and (3) modifying present turbine designs by using blades only 25 to 30 inches in length, increasing the blade structural strength, and using smaller hood structure and shorter bearing span. The effect of variations in the turbine exhaust pressure on the fuel consumption and power output of typical nuclear- and fossil-fueled steam plants is shown in Fig. 15 based on GE data presented in Refs. 62 and 64 for a present turbine design modified to operate over high backpressures. The data indicate that the steam power plant efficiency would be reduced by 8% to 14% for fossil-fueled plants operating with turbine exhaust pressures of 8 in. Hg abs to 15 in. Hg abs, values which appear typical for dry cooling tower designs. Thus, fossil-fueled plants could achieve a thermal efficiency of 38% (HR = 8980 Btu/kwh) using a once-through cooling system, whereas if a mechanical-draft dry-cooling tower system were used, a thermal efficiency from 32.7 to 35.0% would be achieved. Efficiency penalties for nuclear plants would be almost 60% higher than in fossil-fueled stations but low fuel costs tend to offset this effect when determining the total cost of producing power. The efficiency penalties would be about 3 to 4% with most wet-type cooling tower installations in either fossil- or nuclear-fueled plants. The performance of a new turbine design would be flatter over the range of exhaust pressures with poorer performance at low pressures and improved performance at high pressures (see dashed line in Fig. 15a).

The power output from a steam plant also decreases with higher turbine exhaust pressure as shown in Fig.15b. Thus, during summer months when ambient dry- and wet-bulb temperatures are above the yearly average, the power output from a given plant would be reduced and additional plant capacity would be needed to meet the nameplate rating of the station. This factor could be especially costly and serious in some regions of the country such as the Northeast, East Central, Southeast and South Central Regions where annual peak demands are experienced during the summer months.

Total Cost Penalties

A complete economic comparison of the alternative methods of rejecting condenser waste heat from steam power systems requires specification of a number of design and economic factors including plant geographic location, fuel costs and capital charges, utility system load characteristics, etc. Several studies have been devoted to such comparisons (Refs. 35, 56, 56, and 60 through 62) and provide valuable sources of owning and operating cost data. A selected summary of these data is presented below for each alternative cooling system.

At least four major factors must be evaluated in a cost comparison for alternative condenser cooling systems. They include: (1) the total capital cost to purchase and install equipment such as the condenser, pumps, motors, piping, cooling towers and/or cooling ponds, and accessories, (2) the cost of the auxiliary power needed for circulating pumps and fans to operate this equipment, (3) the maintenance costs for chemical treatment, makeup water, and repairs where applicable, and (4) the increased annual fuel consumption and loss of power output as the result of operating at higher condenser pressures than would be required with a once-through river system.

A comparison of the total capital costs for the alternative cooling systems is presented in Table XIII based on data presented in selected references. Slightly different assumptions were used to obtain the values in the various studies but the data appear fairly consistent for most systems. However, several minor discrepancies are apparent. For example, the Ref. 63 data, unlike those presented in Refs. 56 and 35, indicate that once-through cooling using ocean water is less expensive than using river water. Apparently the Ref. 63 estimates do not include costs for discharge temperature regulation such as skimmer walls or long discharge lines, especially for the ocean installations. The Ref. 56 data also indicate only about a \$1/kw additional cost for a cooling pond system when compared to a once-through river installation. Data in Ref. 57, by Kolflat, as well as those presented in Refs. 63 and 65, would appear to substantiate a cost differential of about \$2.50 to \$3.00/kw. The Ref. 56 cost data for the mechanical-draft dry cooling towers also appear to be somewhat low, especially in light of the extensive optimization analyses performed to arrive at the data presented in Ref. 64. The cost estimates presented for the natural-draft dry cooling towers show a wider range of differences, but detailed estimates presented in Ref. 62 and those in Ref. 66 indicate that \$20 and \$26/kw are realistic.

The costs for the auxiliary power to drive the circulating pumps, fans, and appropriate accessories in alternative cooling systems must also be included in economic analyses. Reference 63 estimates that 0.425% of the generator output in a fossil-fueled plant is required for a once-through river system and cooling lake, whereas a once-through ocean system will need only 0.375% because of the generally lower circulating rates (see Table XIV). The same reference estimates 0.875 and 1.075% of the generator output is needed for auxiliary power in natural-draft and mechanical-draft dry towers. As a rule of thumb, these values can be increased by 60% for nuclear plants. Data in Ref. 67 state that the cooling fans in a mechanical-draft wet tower would require as much as 0.50% of the power plant normal generating capacity while typical values of 0.65% (in Ref. 56) and 0.8% for a nuclear unit are given by Kolflat. The auxiliary power requirements for mechanical-draft dry towers are given in Ref. 64 and range from 1.60 to 3.50% of the generator output, depending upon the condenser pressure level. A value of 3.05% appears only slightly above the average. Although the added fuel cost to operate the auxiliaries produces cost penalties of less than 0.1 mill/kwhr for an 80% load factor, the added capital cost for the auxiliary power equipment can also add a comparable cost penalty. For example, the capital cost to provide an added 3% for the auxiliary power in mechanical-draft dry-tower systems can add 0.065 mill/kwhr to the cost of generating power if the auxiliary power cost is \$100/kw (see Table XIV). Maintenance and water makeup costs for the various systems were assumed to be one-half of the costs for the added fuel to operate the auxiliaries, based in part on the limited data presented in Ref. 56, except for the dry towers which, according to Refs. 62 and 64, should require only minimal maintenance.

The final factor in evaluating the alternative methods is the possible increased annual fuel consumption and loss of power output due to operation at slightly higher turbine back pressure in comparison to most once-through systems For the purposes of this comparison, it can be assumed that the use of cooling ponds, spray ponds, and wet towers will result in the use of a condenser temperature of 115 F (or 15 F above once-through cooling systems) and, for dry cooling towers, 160 F. Thus, the increased fuel consumption as read from Fig. 15 would amount to about 0.4% and 10.0%, and the loss in turbine capability about 0.6% and 9.1% for the wet-tower and dry-tower systems, respectively. Estimates of the added costs for both of these factors, when converted to mills/kwhr, are shown in Table XIV based upon fuel costs of 30¢/million Btu and $100/kw as the value of the incremental capability that would be needed to make up the turbine output deficiency. The added costs for mechanical-draft wet cooling towers in excess of that for a once-through river or ocean system would amount to about 0.07 mills/kwhr which is in general agreement with the extensive cost data for various cooling systems presented in Ref. 60.

In summary, it does not appear that the added installed costs or performance penalties are particularly severe when cooling ponds, spray canals or wet towers are selected as cooling systems for fossil-fueled plants, i.e., they add only 1 to 3% to the overall busbar cost of power production. However, the costs of these alternative systems tend to be substantially higher for nuclear plants, and the use of dry cooling towers in either draft configuration would add at least 10% to the cost of power production and would result in serious consideration of alternative power systems.

Other Considerations

Other factors in addition to the general economic ones previously described are usually considered by a utility before selecting a particular method for cooling condenser water. These factors include the availability and cost of water and environmental effects such as fog potential, consumptive water loss by evaporation, drift, blowdown and aesthetic distractions. Dry cooling towers, unlike wet cooling methods, should have no adverse effects on the environment (Ref. 60) and other than the large units which result with natural-draft types would be completely satisfactory. Their higher operating and capital costs in many instances could be offset in areas where low-cost fuels and makeup water is costly. Cooling ponds are often used by utilities because they can be carefully integrated into the surrounding area and provide beneficial recreation sites open to the entire community.

Although the fog-producing potential of wet cooling towers is often mentioned, visual observations and studies have shown that this is not the case. An excellent review and estimate of the fog potential of wet cooling devices is presented in Ref. 60. Wet cooling towers can sometimes produce localized vapor plumes which together with their physical size can be somewhat objectionable in terms of overall station appearance, proper site location. Careful planning can essentially eliminate all problems. Concern over evaporation losses from wet cooling devices is considered in Ref. 60 and the results indicate that cooling ponds would result in about 25% higher water losses relative to once-through cooling based on average normal conditions in the Lake Michigan area. The water losses for other wet cooling devices would be only about 15% higher than for once-through cooling systems. Drift is usually encountered only in areas in the immediate vicinity of the tower according to Ref. 60 and can be almost completely eliminated by control of air velocity and design of drift eliminators.

Proper maintenance of wet cooling towers to insure extended life requires coating and chemical treatment of the system elements subject to corrosion and deterioration. These inhibitors and the blowdown water rejected to maintain a given water concentration must be carefully controlled to avoid discharging pollutants into the cooling water source. However, in most cases cooling water can be controlled to avoid objectionable waste water discharges. The use of salt water in cooling tower installations has been considered, and manufacturers have apparently stated it would be practical although the tower costs would be about 25% higher than for fresh water use (Ref. 63).

The availability of makeup water to replace evaporation and other sources of water losses from a wet cooling tower and cooling pond is essential for these cooling methods. Sufficient water should be available in all areas generally east of the Mississippi and along the west coast. In the plains states and other western locations, the cost for makeup water would have to be carefully considered before a selection is made.

Finally, climatic conditions such as high winds and high wet- or dry-bulb temperatures will affect the performance of cooling towers and other systems to varying degrees. Wet towers, which depend on low wet-bulb ambient air temperatures for efficient operation, will be favored less in hot, humid areas. Conversely, dry cooling towers would be favored in areas of low dry-bulb temperature. Hurricane-force winds can cause damage to natural draft towers, high winds can cause uneven operation, and structural damage due to freezing can occur if the towers are not properly designed.

Advanced Cooling Systems

With increased emphasis on the use of cooling towers and ponds for large central power stations in the next two decades, larger-capacity designs, improved materials, and greater utilization of remote operation of these systems are anticipated. Most of the advances will occur as new materials such as plastics and fiber glass are used for piping and coatings to reduce costs. Maintenance problems associated with heat exchanger corrosion and fouling are expected to be reduced by the utilization of tougher coatings. A number of advanced cooling tower concepts have been studied in an effort to achieve improved performance or lower installed costs (see Refs. 56, 63, and 65). These programs are in the preliminary stage and specific cost data are generally not available.

There is also a need to explore the use of new exchanger surfaces and tower designs rather than continue with designs which have evolved without extensive analysis over the past half century.

SECTION VII

TECHNICAL AND ECONOMIC CHARACTERISTICS OF ADVANCED GAS TURBINE POWER GENERATING SYSTEMS

SUMMARY

An investigation was undertaken to define and select advanced fossil-fueled open-cycle base-load gas turbine systems that have the potential for generating lowest cost electric power while eliminating thermal pollution. A review of aircraft and industrial gas turbine research and development programs was made to provide the basis for the selection of pertinent base-load engine design parameters. Detailed estimates of the performance, size, and cost characteristics are presented for advanced simple-, regenerative- and compound-cycle gas turbine engines. Conceptual designs of selected engine configurations which are judged to have the greatest technical and economic potential for providing minimum power costs and an engineering layout of a possible future large central power station utilizing open-cycle gas turbines are included.

The performance, cost, and size estimates of advanced gas turbines were made for engines which could be in commercial operation during three time periods -- the 1970-decade and the early and late periods of the 1980 decade. The estimates are based on the assumption that the substantial gas turbine technology developed for aircraft and aerospace application will be transferred unhindered to various industrial applications, including those in the electric utility industry. Natural gas has been selected as the basic gas turbine fuel for convenience in the study; however, a number of other gaseous and liquid distillate fuels are also suitable for use in gas turbines.

The advanced gas turbine power systems considered in this study would not contribute to thermal pollution of rivers and lakes. The total heat load that must be rejected from gas turbine systems includes: (1) the waste heat in the exhaust gases, which accounts for over 90% of the total heat rejection load; (2) heat generated in the bearings, seals, etc.; and (3) heat of compression in the compressor discharge air bled for purposes of turbine cooling; removal of this heat, as shown later in this report, improves power plant performance considerably. The waste heat in the exhaust gases is rapidly dispersed into the atmosphere. The bearing heat load is rejected to a circulating oil cooling system and the compressor bleed air heat load may be rejected to a circulating water cooling system; in both cases final heat rejection to the atmosphere is achieved via coolant-to-air heat exchangers. The elimination of large supplies of cooling water as a siting

criterion and design factor for gas turbine power systems can also provide additional transmission and reserve margin savings, the details of which will be presented in Section VIII of this report.

DESCRIPTION OF BASIC THERMODYNAMIC CYCLES

A brief review of the thermodynamic cycles used in gas turbine power generation systems is presented in this section to provide background concerning the major system components and arrangements as well as the primary operating parameters and to establish a framework for the discussion and presentation of results which follow.

Simple Cycle

The Brayton cycle is the basis of gas turbine engine power generation systems. Ideally it consists of isentropic compression, constant pressure heating, isentropic expansion, and constant pressure cooling. Thus only a compressor, combustion chamber and turbine are needed to achieve the various processes and this arrangement is commonly called the simple cycle. The processes for the ideal simple cycle are depicted on a temperature-entropy (t-s) diagram in Fig. 16a. A flow diagram for a simple-cycle engine is shown in Fig. 17a and representative performance given in Fig. 18a. The difference between the shaded area in Fig. 16a (heat input in combustion chamber) and the cross-hatched area (heat rejected in the turbine exhaust) represents the net work output. Of the total shaft work developed by the turbine, approximately one-half to two-thirds is used to drive the compressor and the remainder to drive the load, i.e., the electric generator. The electrical generator may be mechanically coupled to the same shaft as the compressor turbine (single-shaft version) or it may be driven by a free power turbine on a separate shaft (two-shaft version) aerodynamically coupled to the compressor turbine as shown in Fig. 17a. The combination of the compressor, combustor, and compressor turbine, in a free power turbine configuration, is commonly called the gas generator.

In the basic open-cycle configuration, the working fluid (air and combustion products) passes through the gas generator only once; modifications to this cycle include the closed-cycle and semi-closed-cycle configurations. In the closed-cycle configuration the working fluid, which is usually selected to minimize high-temperature oxidation problems, is continuously recycled. Heat addition is accomplished by heat transfer through the walls of a heat exchanger, to which heat from an external source (such as a furnace or nuclear reactor) is supplied. In the semiclosed-cycle configuration, approximately two-thirds of the working fluid is recirculated. This type of gas turbine requires a precooler for the recirculated gas and a "charging" compressor to provide the necessary air for combustion. The closed-cycle and semiclosed-cycle configurations were not investigated in this study since these configurations are generally not considered competitive for fossil-fuel plants and thus are not being actively pursued in the US at the present time.

To achieve high thermal efficiency and minimum size with the simple open-cycle gas turbine, high values of compressor pressure ratio and turbine inlet gas temperature are required together with efficient components, i.e., compressor, turbine, and combustor. Because of the importance of these parameters to the success of base-load gas turbines for power generation applications, progress made during the last two decades in achieving high compressor ratios and turbine inlet temperatures, together with that projected for the next two decades, is considered in detail in subsequent sections of this report.

Regenerative Cycle

Because materials capable of withstanding high operating temperatures were not available up until the early 1950's, and since the attainment of high component efficiencies required basic research programs beyond the financial capability of private industry, designers of industrial gas turbines began incorporating changes to the simple cycle which would increase thermal efficiency. One of the changes often selected is the use of regeneration in which a portion of the heat in the hot exhaust gases leaving the turbine is transferred through an exchanger to raise the temperature of the compressor exit airflow prior to combustion. The processes in a regenerative cycle are shown on a t-s diagram in Fig. 16b. The heat addition from the turbine exhaust gases (see Fig. 17b) results in an increase in the average temperature of heat addition to the cycle and the net effect is a marked improvement in thermal efficiency as illustrated in Fig. 18b.*

The Fig. 18b results also indicate that for a given turbine inlet temperature, the maximum value of thermal efficiency (η_{th}) is generally reached at a lower compressor pressure ratio in a regenerative-cycle engine when compared to the simple-cycle engine. However, if turbine inlet temperature is increased, the maximum value of η_{th} will occur at higher compressor pressure ratios.

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* The performance data shown in Fig.18 are based upon simplying assumptions and are intended to illustrate broad trends rather than specific levels.

Intercooled Cycle

In the intercooled cycle, the gases in the compression stage are cooled between one or more stages of the compressor and thus the work of compression is reduced. As shown on the t-s diagram in Fig. 16c, the intercooling step tends to lower the average exhaust temperature. An arrangement of components in an intercooled cycle is shown in Fig. 17c, and the effect on the thermal efficiency for engines designed with relatively high compressor pressure ratios is shown in Fig. 18c. However, the main effect is to increase the specific power of the engine and thus reduce the physical size and cost of basic equipment, but these benefits are partly offset by the added cost of the intercooler.

Reheat Cycle

In the reheat cycle, combustion gases which are partially expanded in a turbine to an intermediate pressure, are then reheated in a secondary combustor, and finally expanded in a second turbine to atmospheric pressure. Reheat raises the average temperature of heat addition to the cycle, as shown in Fig. 16d, thereby resulting in a higher output per lb of air. A typical arrangement of components in a reheat cycle is shown in Fig. 17d. Reheat alone does not improve the thermal efficiency of gas turbines; in fact, as shown in Fig. 18d, the use of reheat can result in a slight reduction in η_{th} of low-to-moderate pressure ratio engines. A significant improvement in η_{th} can be achieved, however, when reheat is used in conjunction with regeneration and/or intercooling in a compound cycle (see Fig. 18e).

Compound Cycle

The compound cycle, shown schematically in Fig. 17e, can incorporate all of the foregoing features to achieve maximum efficiency as depicted in Fig. 18e. The t-s diagram for this cycle is shown in Fig. 16e. The shaded area under c-3-d-3 represents fuel-energy input, the hatched area under a-b represents heat rejected in the intercooler, and the hatched area under f-1 represents heat rejected in the regenerator exhaust gas.

The study presented herein has focused on the simple-, regenerative- and compound-cycles; the compound cycle utilizes only reheat and intercooling.

GAS TURBINE DESIGN CONSIDERATIONS

Although the same basic components, namely, a compressor, combustion chamber, and turbine, are found in both the aircraft and industrial gas turbines, their engineering development has been based on distinctly different design philosophies for each of the respective applications.

Basic criteria for jet engines are compact size, light weight and low fuel consumption for a given thrust output. Also required are such features as wide operating range, fast acceleration, low thermal stresses, and above all, high reliability. Consequently, the development of aircraft gas turbines has generally been centered about the simple open-cycle engine, and has involved highly sophisticated and costly programs to develop engines with the (1) highest flow rates and highest work transfer capabilities in each rotating stage, (2) lightest weight materials capable of operating at the highest possible stresses and temperatures without compromising reliability, and (3) highest component efficiencies.

Some effort has been expanded by the aircraft industry, in particular the Allison Division of General Motors, Pratt & Whitney Aircraft, and the Garrett Corp., on the development of regenerative-cycle aircraft engines for long-duration missions where minimum fuel consumption was of particular importance; however, many problems such as combustion-product-deposit accumulation on the heat exchanger surfaces, excessive pressure losses, high costs, and relatively heavy regenerator weights have hindered these programs.

A review of industrial gas turbine designs built over the last 20 years shows that they tend to oscillate between complex reheat, intercooled, and/or regenerative cycles offering relatively high thermal efficiencies, and simple designs which stress reliability. The continued re-emergence of the simple-cycle gas turbine indicates that, despite the effort expended, the industry has failed to provide a completely satisfactory high-efficiency prime mover. Much of the trouble encountered in the development of the more complicated cycles for relatively large gas turbines has been due to miscalculations of pressure losses, particularly in ducting and elbows. The effect of these losses on performance in future high-pressure-ratio (50 to 100:1) designs could be less pronounced due to the high absolute levels of operating pressures.

However, articles such as those appearing in Refs. 68 through 71 continue to promote the use of compound cycles in various applications ranging from moderately large central power stations to land transportation. A 37-Mw compound open-cycle gas turbine power station was constructed by the Fiat Company of Italy; industrial operation of the unit started in September 1962. The system utilizes two compressors with intercooling and two turbines with reheat (Ref. 69). In order to allow for the combustion of cheap residual fuels, both combustion chambers consist of single-can combustors arranged remotely from the compressor turbine assembly. A similar unit, rated at 100-Mw output, was built and rig-tested in Russia in 1967 (Ref. 68). The principal difference between this unit and the Fiat design is the use of multiple-can straight-through combustors instead of the remotely located single-can configuration. The 100-Mw unit was reportedly designed to achieve a pressure ratio of 26:1 even though previous Russian studies has indicated that a pressure ratio over 100:1 would be advantageous.

Most industrial gas turbines follow a set design pattern; a compressor is followed by either a single-shaft or two-shaft turbine, with the simplest bearing arrangement. Single-shaft machines are more often used for constant-speed applications such as electric power generation. Two-shaft machines are more suitable for variable-speed service such as gas or air compressors and pumps. A single large combustor, favored in European designs, or multiple, symmetrical, can combustion chambers, favored in the United States designs, provide heat addition to the engine. Axial-flow turbomachinery is used nearly exclusively in all large gas turbine designs. Centrifugal turbomachinery has been used on a few small units (with ratings up to about 3000 hp), although not extensively.

It should be noted that increased turbine inlet temperatures above the 1500- to 1600-F level presently used in industrial gas turbines represent one of the design advancements available from materials technology in aircraft-type gas turbine programs that will lead to lower fuel consumption per unit of shaft work output.

PROJECTED ADVANCES IN GAS TURBINE COMPONENT TECHNOLOGY

The application of gas turbine engines to provide the mid-range and base-load requirements of the electric utility industry has been limited because of the modest thermal efficiency levels attainable in comparison with conventional steam power systems. The present limited output capability per engine and the requirement for relatively clean-burning and, therefore, moderately expensive fuels (Refs. 68 through 77) have also severely limited nonaircraft gas turbine application. Further significant improvements in specific power (hp per lb/sec of engine airflow) and thermal efficiency for gas turbine power plants can be achieved only by increasing turbine inlet temperature and compressor pressure ratio, since turbomachinery component efficiencies have reached a relatively high level due to over 30 years of extensive research and development on compressors and turbines for aircraft propulsion, and only small further gains may be anticipat

Recent advances in turbomachinery, materials technology, aerodynamic design, heat transfer, and fabrication techniques for gas turbines have resulted from the requirement for new military and commercial aircraft (Refs. 78 through 85) as well as various industrial applications (Refs. 86 and 87). These technological developments provide the basis for substantially improved gas turbines with significantly lower fuel consumption rates than presently attainable. Pertinent aspects of these technological advances, including their effect on gas turbines designed for electric power generation, and projections of further advances anticipated during the 1980's are discussed in the following paragraphs.

Compressors

Performance Parameters

Technological advances in aircraft gas turbine compressor design have resulted in substantial improvements in cycle pressure ratios, increasing from about 4:1 in the early English (National Gas Turbine Establishment) centrifugal compressors of the 1940's to 24:1 in present twin-spool axial-flow compressors (see Fig. 19). A spool is commonly considered to be a compressor and connected turbine. The compressor or turbine may have one or more stages. Thus, twin-spool compressors consist of two tandem compressors with their respective turbines that form two rotor systems which are mechanically independent but related aerodynamically. These configurations facilitate proper compressor-stage matching for relatively high pressure ratio designs as an alternative solution to single-spool compressors that would have to incorporate variable geometry stator vanes. The technology is becoming available that will permit the introduction of machines with pressure ratios exceeding 30:1, a prerequisite for high-efficiency simple-cycle base-load gas turbine engines. Proportionate increases in the number of axial compressor stages to meet these pressure ratio requirements have been avoided through intensive research and development efforts to increase stage pressure ratios. For example, it is presently possible to achieve single-stage pressure ratios of 1.2 to 1.4 while still maintaining stage efficiencies of 90% or more as shown in Fig. 20. These stage performance levels have been extended to multi-stage aircraft compressor designs and permit the attainment of average stage pressure ratios on the order of 1.2 to 1.3 with associated polytropic efficiencies of approximately 90% or greater (Ref. 70). These trends are clearly indicated by the statistical data shown in Fig. 20 for compressors which have been built or are in the study or development stage. Further improvements in polytropic efficiency to peak values of 91 to 93% within the next decade appear feasible (Refs. 70, 88, and 89).

Principal factors which influence stage pressure ratio are rotor tip speed and aerodynamic loading. Rotor tip speeds of 1100 to 1200 ft/sec are attainable with current advanced lightweight compressor designs (Ref. 70) and values as high as 1400 ft/sec are forecast for the early 1980's (see Fig. 20). Although lightweight fan stages are in operation at tip speeds of 1500 ft/sec and above, high stress limits and supersonic Mach number operation will limit compressors to the aforementioned levels. The aerodynamic loading, at a given tip speed, establishes stage pressure ratio. A measure of aerodynamic loading is the airfoil diffusion factor (D_f), a parameter which includes a factor reflecting the overall change in relative velocity across the blade row and a term proportional to the conventional lift coefficient. High values of D_f are desirable since they result in a reduction in the number of compressor stages; values of 0.40 and 0.45 are currently obtainable The present limit in D_f is due primarily to excessive secondary flow or endwall losses and to compressor surge margins that are dangerously low. Further improvements in tip speed and diffusion factor will result from aircraft compressor research and development programs.

The specific flow (lb/sec per unit flow area) of axial aircraft compressors varies from approximately 33 to 40 lb/sec/ft^2 of frontal area. Increases in this parameter result in more compact designs (smaller cross-sectional area); however for a given wheel speed (U_m) and work coefficient, higher specific flows and, alternatively, higher axial velocities (C_x) reflect high flow coefficients (C_x/U_m) and correspondingly lower compressor efficiency.

Adaptation of these developments to industrial units has been slow because of the lack of incentives to undertake parallel costly development programs for long-life stationary gas turbine power plants. Consequently, present industrial gas turbine axial compressors incorporate moderate blade loadings and tip speeds of only 650 to 750 ft/sec, and achieve average stage pressure ratios only on the order of 1.1 to 1.15 (Refs. 74, 86, and 90). As a result, as many as 15 to 17 compressor stages are required to produce cycle pressure ratios above 10:1. Thus, high-thermal-efficiency performance cannot be easily achieved with these designs. European industrial units appear to favor the use of large, massive, multi-unit compressors incorporating stage intercoolers (Refs. 68, 69, 75, and 91) to achieve cycle pressure ratios on the order of 15:1 to 25:1.

Construction Design Features

Future large industrial compressor units will have to incorporate low-cost materials and inexpensive construction techniques without sacrificing reliability, life, and performance. Although typical lightweight aircraft compressors are designed as shown in Fig. 21a, with the rotors formed by rows of blades rooted in structural disks, other configurations may have to be examined in detail (see Figs. 21b and 22c). In aircraft designs, the compressor stages are jointed and positioned axially by cylindrical inner spacers and conical spacers. The inner wall of the flow path is formed partly by the disk, rims, and blade root platform, and partly by the inner shroud of the stators. The outer wall is formed by the outer shroud of the stators. Present industrial compressor designs incorporate disk-drum rotor assembly (Fig. 21b), or a drum rotor assembly (Fig. 21c) construct[illegible] In the disk-drum arrangement, the rotor for each stage consists of a solid disk.

Through-bolts connect the rotor disks to the forward and aft subshafts to form the disk-drum assembly. The bolt circle is placed at the maximum diameter possible within the confines of the rotor design to ensure a strong, stiff rotor assembly. This rotor configuration eliminates the need for labyrinth seals at the blade root section between compressor stages since a close-fitting hub is formed by the adjacent rims of the stacked disk assembly.

The drum rotor configuration affords perhaps the simplest type of structure. Individual blades are attached to the outer surface of a cylindrical cast drum structure thus avoiding the necessity of using individually forged compressor disks. A constant hub diameter design also offers an estimated 15-to-20% reduction in blade cost, relative to a tapered hub configuration, as a result of the simplified blade root pedestal machining procedure. Maximum rotational speeds with the drum rotor construction are limited, however, because high rim speeds must be avoided in order not to exceed maximum allowable stress limits.

<u>Materials</u>

Continuing efforts to eliminate excessive weight in aircraft engines have pushed the limits of materials properties. Until recently, the operating conditions in the compressor section were such that titanium alloys (Ti-6Al-4V) and low-alloy and stainless steels (Types 17-22A, Greek Ascoloy stainless AMS5616, and AISI410) satisfactorily fulfilled all design requirements. However, with the development of high-pressure ratio, high-airflow handling turbofans, exhibiting rim temperatures in excess of 800 F, new superalloys such as A-286, Incoloy 901, and Inconel 718 were developed for disks as well as blades in the high-pressure sections of these advanced designs (Ref. 91), and most of these materials could be easily used in advanced industrial-type engines. A complete listing of materials used for compressor parts in existing and new designs is presented in Ref. 91.

High-pressure ratio advanced-design industrial compressors should, therefore, continue to utilize the martensitic corrosion-resistant AISI Type 410 steel and low-alloy (AISI4340) steel extensively for airfoils and disks, respectively, in the low-temperature regions (rim temperatures below 800 F). However, aircraft-proven nickel-base alloys such as Incoloy 901, Inconel 718, and A-286 will be necessary in the high-temperature sections (800-to-1200 F). The low-alloy steels are usually protected from rusting by a 0.5-mil coating of nickel cadmium (Ref. 92).

The next generation of aircraft compressors will see an increase in the use of composite parts (Refs. 91 and 92); among the most promising are silicon carbide-coated boron fiber and carbon fiber. The composite materials have not yet been proven ready to meet commercial airline reliability requirements as far as erosion resistance and foreign-object damage are concerned. However, these composites may permit the inexpensive fabrication of large parts without the need for new and expensive machine tools; consequently, they show the potential of significant future cost savings, particularly in the manufacture of very large-capacity gas turbines.

Combustors

The primary characteristics of combustors which affect gas turbine performance are combustion efficiency, pressure drop, and uniformity of outlet gas temperature. Whereas combustion efficiency affects only fuel consumption, pressure drop affects both fuel consumption and power output.

Performance Parameters

Combustion efficiencies of 100% are currently achieved in aircraft gas turbine combustors and similar performance can also be achieved in advanced industrial combustors burning clean gaseous fuels. The outstanding achievement in combustion technology has been in the reduction in pressure drop, defined as the difference in total pressure between the compressor exit and turbine inlet. Pressure drops in aircraft combustion designs have been reduced from about 10% in 1945 to under 5% in current advanced designs. Of particular importance is the fact that these reductions in pressure drop have been accompanied by improvements in combustor outlet temperature distribution (Ref. 89). Combustor pressure drop is also related to reference velocity, defined as the theoretical velocity for flow of combustor inlet air thorugh an area equal to the maximum cross section of the combustor casing. Reference velocities of 80 to 150 ft/sec, with corresponding combustor liner pressure losses of 2 to 5%, are common in present aircraft designs; 20 to 80 ft/sec with correspondingly lower pressure losses will be found in the industrial external combustor and multiple-can designs.

The heat release rate within the combustor also affects engine size and is especially important in aircraft gas turbines wherein heat release rates of from 2 to 4 million Btu/hr-ft^3-atm are common and rates as high as 8.5 million Btu/hr-ft^3-atm have been used (Ref. 93). Heat release rates for industrial gas turbines are substantially lower since these units are commonly designed to burn a wide variety of fuel types (Ref. 93).

The combustor must be designed to avoid exit gas temperature peaks since a large temperature gradient reduces the allowable average gas temperature into the turbine and thus limits engine output and efficiency. In addition, turbine vanes are exposed directly to local gas temperatures, and high peak gas temperature will reduce vane life. Continuous efforts to reduce the ratio of the difference between the peak local-to-average gas temperature and the average combustion temperature rise have resulted in typical values for this parameter of between 0.05 and 0.15 for vanes. These techniques can be utilized in gas turbines designed for electric power generation systems and will enhance their performance and operating characteristics.

Construction Design Features

Combustor design configuration and shape depend largely upon the intended applications. Aircraft gas turbines, for instance, specify compactness and lightweight as the primary design criteria. These designs, therefore, will utilize either a single annular combustor, a number of tubular combustors (multiple-cans), or a can-annular arrangement with a number of tubular liners within an annular casing. The combustors may be placed for straight-through flow between the compressor and the turbine, or designed for reverse flow and wrapped around the compressor or turbine. Heavy-duty residual-oil burning industrial turbines are usually designed for ready access and removal of combustor parts, and may use either multiple-can combustors discharging directly into the turbine vanes (Refs. 68, 74, 76, and 90) or one or two external combustors firing into a large turbine nozzle box (Refs. 69, 71, and 77). However, as turbine inlet gas temperatures in advanced-design industrial gas turbines are raised to higher levels to achieve improved thermal efficiency, existing differences between the design philosophies of industrial- and aircraft-type combustors will diminish considerably. For instance, the external combustors and multiple-can combustor construction currently used in industrial designs operating with turbine inlet temperatures of approximately 1300 F to 1700 F will have to be replaced with annular configurations in power plants designed to operate at turbine inlet temperatures much above 2000 F. These changes will be necessary to avoid excessive liner cooling flow rates, high pressure losses, and high local peak temperatures on first-stage turbine vanes.

Conventional louvered liners, used in industrial and production aircraft combustors, will be permissible for use at turbine inlet gas temperatures below approximately 2400 F; at 2400 F and above, advanced-design liners with high heat transfer surfaces developed for aircraft combustor designs will also be specified for the industrial counterpart. Length-to-height ratio for combustors has ranged from 2.7 to 5.0, but generally varies between 3 and 3.5. In the future, however, short burners will be favored since recent experiments indicate that in such designs the formation of nitrogen oxides seems to be inhibited. Discussions of the nitrogen oxide emissions from gas turbine power plants and research efforts under way to inhibit the formation of this pollutant are presented in Appendix A. The ability to shorten burners is dependent upon the fuel distribution system as well as the linear hole pattern and liner cooling scheme.

Materials

Alloys used for aircraft combustors and liners and for the transition ducts connecting the combustors to the turbine inlet nozzle must be formable, weldable sheet alloys capable of at least 8000 to 10,000 hr of operation in an oxidizing environment at average metal temperatures of 1650 F and above (Ref. 91). These sheet alloys must be stable for this long-time, high-temperature service and have optimum erosion-corrosion, thermal fatigue, and distortion resistance. Liners are always air-film cooled and are often coated for additional protection. Hastelloy X is by far the most commonly used alloy for aircraft combustion liner applications, whereas AISI Type 321 and 310, oxidation-resistant austenitic stainless steels, have been used in the relatively low-temperature industrial units (Ref. 91). Haynes Development Alloy No. 188, superior to Hastelloy X in high-strength ductility and oxidation resistance, is now widely used by aircraft engine builders for advanced-design combustor applications. Materials presently under development for the next generation of aircraft engine combustor designs include coated TD-Ni, TD-NiCr, and dispersion-strengthened materials (thoria dispersed in nickel). These materials will have substantially higher service temperatures than present materials (see Table XV), and it is foreseeable that many of these new materials would permit long-life operation of 24,000 hr and above.

Turbine

Increases in turbine inlet operating temperatures through the use of improved materials and cooling techniques represent the principal means of improving the performance and increasing the output potential of future gas turbine designs. Although historically most of this effort was directed primarily toward aircraft propulsion systems, some of the advances were eventually used to improve industrial designs. This trend is evident by an inspection of Fig. 22 which shows the progression of maximum operating cycle temperatures with time for gas turbines designed for various aircraft, industrial, and electric power generation applications.

The data indicate that turbine inlet temperatures up to 1800 F are being used in gas turbine electric power plants, and several hundred degrees higher in commercial and military aircraft. Prior to the mid-1960's, advances in materials technology traditionally accounted for a respectable 20-to-40 F increase per year in turbine inlet temperature (Refs. 78 and 89). Recently, however, significant increases in turbine inlet temperature (Refs. 70, 75, 78, 79, 83, and 84) approaching 70 to 80 F per year, have been achieved through substantial improvements in turbine cooling techniques in combination with newer materials. With the expected continuation of current trends, turbine inlet temperatures up to 2400-to-2600 F may be common in military aircraft engines before the end of the 1970's and should be available in industrial engines early in the 1980 decade. Turbine inlet gas temperatures as high as 1900 F for continuous cruise operation, and 2100 F for take-off, are now experienced by the engines utilized in today's 747 and C5A jumbo jets. AiResearch Division of Garrett Corp. has made extensive studies of simple- and regenerative-cycle gas turbines capable of operating at a turbine inlet temperature of 2300 F. Turbine inlet gas temperatures as high as 2400 to 3000 F in base-load industrial gas turbines have been projected for the next decade by manufacturers of industrial units (Refs. 94 and 95).

Performance Parameters

Improvements in turbine performance resulting from technological advances in aircraft turbine designs have paralleled those for the compressor, with turbine isentropic efficiency increasing from about 85% in early-1950 designs to over 90% in current designs; efficiencies of 92 to 93% are feasible in the near future, particularly in high-output designs (Refs. 70 and 89). The design philosophy of turbines for industrial engines will also approach that of aircraft engine turbines. For instance, future industrial turbine designs will incorporate a higher degree of impulse staging than exists in most present designs to achieve a relatively large gas temperature drop in the nozzle, thus avoiding high gas temperatures with corresponding high cooling flows in the rotor (Ref. 75). The stage work for aircraft turbines varies from a low of 20 Btu/lb in the last stage of low-pressure ratio units to a maximum of about 120 Btu/lb in the high-pressure ratio stages of current production units; technology presently exists to increase this value to about 155 Btu/lb without sacrificing turbine efficiency.

High stage work in itself does not cause aerodynamic problems, but when associated with high efficiency it requires high wheel speeds which become limited by structural considerations. Turbine tip speeds in current twin-spool aircraft designs vary from 900 to 1000 ft/sec in the low-pressure ratio stages to as high as 1500 to 1600 ft/sec in the high-pressure ratio stages. These tip speeds can be achieved in future industrial-type designs that utilize the improved turbine materials and cooling techniques described in the following paragraphs.

Materials

The improvements in the temperature capability of turbine materials, with time, for aircraft gas turbines is shown in Fig. 23a. The gains have been most significant for the nickel-base alloys, with an improvement of more than 300 F in temperature capability over the past decade (Ref. 78). Presently, nickel-base cast blade alloys, such as Inconel 718, B1900, and IN-100, have in some aircraft turbine designs replaced forged Udimet 700 (U-700) for high-temperature applications and, at the same time, decreased thermal fatigue failures by a factor of five (Ref. 78).

Figure 23b shows the stress-to-rupture strength of the nickel-base alloy IN-100 for a range of material temperatures. Curves for 1000-, 10,000-, and 100,000-hr lifetimes are shown and illustrate the sharp reduction in allowable stress as the desired operating time is increased. The 1000-hr-life curve is based on experimental data while the 10,000- and 100,000-hr curves are extrapolated data from short-duration test results. Although these materials were developed initially for relatively short-time (1000-hr) high-strength aircraft turbine applications, modifications to the heat treatment cycle have permitted their use for long-time operation (approaching 100,000-hr) as required in base-load power generation. It is anticipated that similar modifications for the aforementioned high-temperature nickel-base alloys will allow the eventual replacement of such alloys as wrought Udimet 500 and M252 presently used in industrial turbines and thus permit high operating temperature.

Turbine blade materials for industrial designs anticipated by the 1980's will be the superalloys currently under development for advanced aircraft gas turbines; these include unidirectionally solidified eutectic alloys such as Ni_3Al-Ni_3Cb (Ref. 85) and particle or fiber dispersion-strengthened metals (Refs. 91 and 96). On the basis of efforts currently underway it is reasonable to expect a continuation of at least a 20 F per year improvement in material temperature. Chromium-base alloys, for instance, with potential firing temperature to 2000 F, are being investigated. Recent breakthroughs in the ability to coat columbium-base alloys also offer a possibility for the use of these alloys, currently being investigated for advanced aircraft propulsion systems. Ceramic materials such as silicon-nitride and silicon-carbide composites are under intensive study for automotive gas turbine applications and offer firing temperatures on the order of 2500 and 2600 F (Refs. 97 and 98).

A brief summary of the projected creep strength properties of present-day and advanced turbine blade materials which could be used for future designs is presented in Fig. 24. The band in Fig. 24 labeled "present materials" represents properties of cobalt-base and nickel-base alloys that are now in use in industrial and aircraft gas turbines. The other bands represent estimated properties of advanced nickel-, chromium-, and columbium-base alloys and combinations currently in advanced stages of development for aircraft engines, which will eventually be adapted to base-load applications. The projected creep strength characteristics of these materials are based on applying Larson-Miller parameter extrapolations from available short-term property data for these materials.

The first-stage nozzle vanes are the hottest parts in the gas turbine and are also subjected to high thermal shock stresses. These vanes are usually coated for improved oxidation resistance. Since they are stationary, they are not highly stressed, and cast nickel- and cobalt-base alloys such as Inconel 718 and 738 and WI-52, which have been used as vane materials in aircraft gas turbines, would be satisfactory for industrial applications.

Coatings

Coatings have been developed which provide adequate oxidation-corrosion resistance for blades and vanes in aircraft turbines. These coatings are aluminide types, applied by pack or slurry techniques. These coatings have eliminated intergranular oxidation attack (which could cause a thermal fatigue failure) and have provided some protection (Ref. 92) against sulfidation attack (corrosion due to the presence of sodium sulfate compounds in the combustion products of distillate fuels). However, coating life decreases rapidly with increases in allowable metal temperatures. For instance, at a metal temperature of approximately 1450 F, a coating life of 50,000 hr may be achieved. A 200 F increase in metal temperature reduces coating life to less than 10,000 hr. Consequently, coating life rather than creep strength properties as determined by peak local temperatures in the first-stage turbine vanes will be an important criterion used to specify turbine inlet gas temperature for different power plant operating modes. Adequate coating life could be especially critical when the fuel used in the engine contains quantities of ash, metals, or sulfur that could cause erosion and corrosion problems. Natural gas, in this respect, is an ideal fuel for high-performance gas turbines. Meanwhile, new coating materials and application techniques including nondiffusion-type coatings are being developed which, when applied in sufficient thicknesses, should provide uninterrupted service for periods exceeding 25,000 hr.

Disks

Turbine disks in new, industrial-type gas turbines operating at 1400 to 1700 F are maintained at metal temperatures of 600 to 750 F (Ref. 86) by the utilization of cooling air extracted from the compressor (see Fig. 25a). As a result of these moderate disk temperatures, fairly inexpensive materials such as austenitic stainless steels (A-286) are used. In production aircraft gas turbines, disks are fabricated from nickel-based alloys such as Inconel 718, Incoloy 901, and Astroloy, and are capable of withstanding metal temperatures of 900 to 1400 F (Refs. 91 and 92). It is anticipated that these materials would be used also in advanced gas turbine power systems.

Turbine Cooling Techniques

Only first-stage vanes and disks of current advanced-design industrial gas turbines, operating at 1600- to 1700-F turbine inlet gas temperatures, are cooled. For long-life, base-load operation at turbine inlet temperatures of 1800 F and above, successive stages of turbine blades will also require cooling (Fig. 25b). Turbine cooling can be accomplished with coolants such as air, water, or liquid metals, but because of the complex cooling system designs and mechanical problems associated with liquid systems, air has been used exclusively as the coolant in all aircraft propulsion systems and for most stationary applications.

Turbine cooling systems in current aircraft engines have progressed from simple convective cooling configurations incorporating cast, round, radial passages, for vanes, and single cavities for blades to advanced convective-heat transfer designs utilizing impingement cooling of the inside surface of the leading edge (Fig. 26). In film-cooled designs, a layer of coolant is injected in hollow blades through radial slots to form an insulating air blanket for the outside blade surface. These designs are in the advanced stages of development for the next generation of aircraft engines. In transpiration-cooled blades, coolant is bled through a porous material which may be formed by a series of small drilled holes along the airfoil surface. These designs are also in the early stages of development and, when applied to 1980-time period aircraft turbines, will offer operation at turbine inlet temperatures approaching 3000 F. Figure 27 summarizes the progress that has been made in the different aforementioned turbine blade cooling system for commercial aircraft engines. The temperature values recorded on the figure for each type of cooling scheme reflect levels that have been demonstrated by actual tests and which have been accepted or appear feasible for commercial aircraft propulsion system applications.

In aircraft engines, the air used to cool the turbine blades and vanes is at an elevated temperature, 800 to 1200 F, even before it is introduced into the turbine, since it is bled from the compressor discharge airstream. Precooling the compressor bleed air to fairly low levels before it is used to cool the turbine has not been a general practice in these aircraft applications because the added cooling system weight detracts from the potential gains in performance that might otherwise be achieved by reducing the turbine cooling flow. However, as turbine inlet temperatures and compressor pressure ratios continue to rise and approach the limits established for proven turbine blade cooling techniques, lightweight low-power, compressor bleed air cooling systems will have to be devised.

For stationary industrial power plants, air-, water-, or even possibly fuel-cooled heat exchangers could be used to reduce the temperature of compressor discharge cooling air to fairly low levels (perhaps to 100 to 200 F) since in these stationary applications weight is not an important criteria. Therefore, with this technique it should be possible to achieve the gas temperature levels indicated in Fig. 27 on a continuous basis in base-load plants with presently available combinations of impingement-conversion cooling techniques, provided the anticipated extended coating lifetimes with the previously mentioned coating improvements are achieved.

Regenerators

Research and development programs on regenerative heat exchangers have concentrated on three different types of regenerator designs, the stationary regenerator (commonly referred to as a recuperator), the rotary regenerator, and the liquid-coupled indirect-transfer regenerator. The liquid-coupled indirect-transfer type has been considered primarily for military aircraft applications. However, several current military programs have been concerned with the recuperator type. Most rotary regenerator programs have been limited to automotive applications. The basic characteristics of the three types of gas turbine regenerators and the reasons for selecting the recuperator type for the intended base-load electric power application are described in the following paragraphs.

In the liquid-coupled indirect-transfer regenerator concept, two separate gas-to-liquid heat exchangers and a circulating liquid loop, coupling the two heat exchangers, are employed. In one heat exchanger, hot engine exhaust gases heat the liquid which is circulated and subsequently cooled in the other exchanger during the process of heating the compressor discharge air. This design has the advantage that the heat exchangers may be conveniently located, thereby requiring the fewest changes in the normal gas turbine flow path. Since the external ducting handles only a high-density liquid, the frontal area is minimized, an important factor in an aircraft application. However, the liquid coupling loop does add complexity and cost. Furthermore, liquid metals which have the best heat transfer characteristics require elaborate handling techniques and special materials because of their corrosiveness and other undesirable characteristics. Some liquid metals are toxic or combustible in air and therefore would create serious problems if a leak occurred. Thus, this type of regenerative gas turbine has not become operational even for military aircraft applications. Even if some of the technical problems of this type of regenerator were solved, it is unlikely that its limited advantages would warrant its use in an industrial gas turbine and thus was not considered further in this study.

Rotary regenerators involve the direct heat transfer to and from a rotating matrix which acts as a heat sink and source, as opposed to heat transfer from one fluid to another through heat exchanger walls, and thus need not withstand large pressure differences across tube walls. Although many matrix designs and materials have been considered in past research and development programs, the porous ceramic core built in the form of a disk such as the Cercor® regenerator, is the only one used to any extent to date. These ceramic rotary regenerators are currently employed in automotive-type gas turbines. The ceramic core is relatively inexpensive and can operate at very high temperatures relative to other materials; furthermore, it can be designed with a high effectiveness (90% or higher) without serious penalties to cost, size, and gas turbine pressure loss. However, the automotive applications of these regenerators have been limited to engines of 300 to 400 hp. In discussions with the leading manufacturer of ceramic regenerator cores (Ref. 99), it was learned that a 400-hp engine requires a 28-in. diameter core and that 36 in. appears to be the largest size that can be built with present technology. This limitation is due to structural considerations which also limits their use to gas turbines with compressore pressure ratios of 6.5 or less. One of the problem areas associated with the rotary regenerator is that of sealing the rotating matrix to prevent the higher-pressure compressor discharge air from leaking directly into the turbine exhaust, thereby resulting in a degradation of performance. Obviously this problem would be worse at the higher compressor pressure ratios and is another reason for the pressure limit previously mentioned. Furthermore, advances in technology in the next twenty years are not expected to significantly affect these limitations. Thus it appears that ceramic rotary regenerators, even in modular form, would not prove practical for the large industrial-type regenerative gas turbines considered in this study. This conclusion is also mentioned in Ref.100, which states that as the power of the gas turbine exceeds 300 hp, large pressure loads inherent with the regenerator introduce structural complexity that tends to offset the low-cost advantages of the ceramic matrix.

In the recuperator (stationary regenerator), compressor discharge air and turbine exhaust gases are ducted to a single heat exchanger in which heat is transferred directly from one fluid to the other through the exchanger wall. Extensive work is currently being done to design and develop lightweight regenerative aircraft gas turbines utilizing the recuperator type of regenerator (Ref. 88) and Harrison Radiator Division of General Motors Corporation has built 75 recuperators since 1957 for use in industrial gas turbines. Since recuperators tend to be large (a recuperator for a current industrial gas turbine of approximately 20,000 hp output capacity weighs approximately 50 tons installed) it is a practical necessity to construct several small modules and then manifold them together into one unit. Most of the industrial gas turbine recuperators in use are constructed with plate-fin type surfaces made of mild steel and operate with a maximum gas temperature less than 1000 F. However, aircraft gas turbine recuperators have been built and operated successfully at significantly higher temperatures. A number of high-temperature recuperator materials are available, and others which are under investigation will provide the desired characteristics for the advanced regenerative-cycle gas turbines which, in general, operate at recuperator inlet temperatures above 1000 F. A brief description of these materials follows.

Recuperator Materials

The primary factors considered in the selection of recuperator base materials are their mechanical properties, hot-corrosion resitance, fabricability, compatibility with brazing alloys, metallurgical stability, and cost.

A recuperator material must have adequate mechanical properties during its design lifetime to withstand the stresses due to thermal transients and fluctuating and steady-state pressure differentials. Therefore, before selecting a material the degradation of its mechanical properties due to environmental attack and by metallurgical changes, such as aging reactions and carbide precipitation, must be considered. Fatigue, ultimate strength, and stress-to-rupture properties are all important in designing a recuperator. Erosion resistance is also important because the gas turbine environment usually contains some solid or liquid particles (dust, sand, or carbon).

Hot corrosion, which is the attack on metal alloy components caused directly or indirectly by contact with the gas turbine combustion products, includes all the effects that contribute to corrosion such as sulfidation, oxidation, erosion, and stress corrosion. This type of attack, even on stainless steels and superalloys, occurs at temperatures likely to be encountered in advanced gas turbine recuperators.

Since brazing is the most economical means of producing recuperator structures, compatibility of the base or structural materials with available brazing filler alloys is extremely important. Base metal-filler metal combinations must have compatible brazing temperatures to prevent metallurgical changes in the base metal as a result of the brazing temperature cycle. Adequate wetting, brazed joint strength, and hot corrosion resistance are also required characteristics. Another requirement is that there should be little or no embrittlement of the base metal by diffusion of the brazing filler metal constituents into the base metal.

Materials selected for formed parts, such as fins and pans, should have adequate formability at room temperature and must be amenable to brazing, welding, and other contemplated manufacturing operations to minimize fabricating costs.

As previously noted, mild (carbon) steel has been used extensively in industrial gas turbine recuperators and possesses adequate properties when operated at temperatures up to approximately 1000 F (maximum gas temperature). This material is relatively inexpensive and is amenable to most low-cost fabrication techniques. Above this temperature level, alloy steels and superalloys must be used.

Type 347 stainless steel, which has been used in many recuperators, provides excellent performance at lower temperatures. It is relatively inexpensive compared to other high-temperature alloys. It has good oxidation resistance at moderate temperatures, relatively good corrosion resistance, and is easily brazed. This alloy has proven successful in recuperator applications (nonindustrial) operating below 1300 F (Ref. 101). Type 347 stainless steel was specified for the recuperator by a recuperator manufacturer (Ref. 102) for a typical 1980-decade engine design (turbine exhaust temperature of 1286 F), whereas a combination of 347 and carbon steel was specified for the 1970-decade engine (turbine exhaust temperature of 1128 F). Because of the relatively low cost of this alloy, another manufacturer even specified 347 steel for the core of its industrial gas turbine recuperator which would operate at turbine exhaust temperatures below 1000 F (Ref. 103).

Type 430 stainless steel is a low-carbon, high-chromium ferritic steel possessing good resistance to oxidation and corrosion at elevated temperatures. It is lower in cost than Type 347 and has good room-temperature strength and ductility. However, it has one-half the yield strength of Type 347 at 1200 F, and its stress-to-rupture properties are also low.

Incoloy 800, an iron-based superalloy, has been used in some high-temperature gas turbine recuperators. Its mechanical properties at 1000 to 1500 F are comparable to Type 347 stainless steel, as is its cost. Incoloy 800 is sometimes specified instead of 347 steel because of its superior resistance to stress corrosion cracking in a chloride ion environment, a situation which might be encountered in a marine gas turbine application. Incoloy 800 is only slightly more expensive than 347 stainless steel. The brazing characteristics of Incoloy 800 are also very good.

Other superalloys which have been considered for high-temperature recuperator applications are Hastelloy X and Inconel 625 (Ref. 101). Although these alloys also have good brazing properties, they are significantly more expensive than 347 stainless steel or Incoloy 800.

The characteristics of the brazing alloy are also important in specifying a recuperator for high-temperature application. The selection of the brazing alloy is made in conjunction with the selection of base material because the brazing temperature must be compatible with the base material properties. In Ref. 101, Palniro 7, a silver alloy, was selected as the first choice for use with both 347 stainless steel and Incoloy 800 because of its excellent brazing characteristics and its hot corrosion resistance;Nicrobraz 135, a nickel-base alloy, was selected as the second choice. Although Palniro 7 is probably more expensive than Nicrobraz 135 on a per-pound basis, the important consideration is how the choice of brazing affects the overall cost of the recuperator which must operate to a specific requirement.

Discussions with the Hamilton Standard Division of United Aircraft have indicated that several new brazing materials are under development which would be applicable to the regenerators of interest. Some of these brazing materials tend to completely coat the base material during the brazing process and, since they are corrosion-resistant, they may even permit the use of low-cost base materials at relatively high operating temperatures.

BASIS FOR SELECTING DESIGN PARAMETERS

The anticipated advances in design technology and materials improvement projections discussed in the preceding sections of this report formed the basis for parametric studies of future natural gas-fueled gas turbine power plant designs

Simple-cycle power plant designs, incorporating single-spool and twin-spool turbomachinery configurations, regenerative-cycle power plant designs with single-spool machinery, and compound-cycle power plant designs incorporating one stage of intercooling and one reheat were investigated.

Ranges of design parameters for base-load gas turbines, reflecting component and materials technology currently available and that projected to become available during the 1970 and 1980 decades, are presented in Table XVI. These design parameters reflect projected advances in turbomachinery flow-path aerodynamic performance and mechanical design features, as well as projected advances in combustor and turbine materials and turbine cooling techniques. The data shown in Table XVI indicate that if compressor bleed air is precooled to temperature levels of 125 to 250 F before being used for turbine vane and blade cooling, increases in turbine inlet operating temperatures of as much as 400 deg can be achieved. The turbomachinery component efficiency values projected in Table XVI for each of the respective time periods indicated could be achieved if anticipated aircraft development programs are pursued. A wide range of compressor pressure ratios was selected, including those levels which are already commonly used, so that the determination of optimum cycle conditions for maximum thermal efficiency simple-cycle, regenerative-cycle, and compound-cycle, gas turbines could be made. In the compound-cycle power plant studies, overall cycle pressure ratios as high as 100:1 were investigated, although the individual unit pressure ratio did not exceed 23:1. Parametric performance studies for the compound-cycle configurations were not as extensive as those for the simple-cycle and regenerative-cycle designs and were restricted primarily to the design technology projected to become available during the early 1980's. The ranges of regenerator effectiveness and pressure drops in Table XVI were selected on the basis of previous experience. A detailed description of the methods and assumptions used to perform these parametric performance and power plant design studies is presented in Appendix B.

PERFORMANCE ESTIMATES

Simple-Cycle Engines

Thermal efficiency and specific output (shp per unit airflow) for simple-cycle gas turbine power plants incorporating the three levels of design technology defined in Table XVI are depicted in Figs. 28 through 30. The performance presented in these figures and in those to follow for the regenerative- and compound-cycle engines are based on the use of methane (HHV = 1000 Btu/ft^3) as the fuel at an ambient state defined in accordance with NEMA standards (80 F and 1000 ft altitude). In addition, these performance values are for the gas turbine only, and although they have been corrected for intake and exhaust stack pressure drops, they would have to be adjusted for auxiliary power loads and generator efficiencies to reflect net station performance. Since these corrections usually represent a constant percentage of the output and thus would not affect the comparions, they have been omitted until a final design point has been selected for each system under investigation.

Present-day gas turbines which operate at turbine inlet temperatures up to 1700 F can achieve thermal efficiencies on the order of 20 to 27% (Refs. 71, 74, 76, to 78, and 86) and specific outputs of 80 to 140 shp per lb/sec airflow. During the 1970 decade it is estimated that simple-cycle gas turbines incorporating the best presently available materials, turbine cooling schemes, and component performance will be capable of substantial improvements in both thermal efficiency and specific output. The specific output is an important parameter since it is directly related to engine size and hence cost; thus the relative increases in this parameter provide an approximation to the potential reduction in power plant size for a given output. The estimated performance of simple-cycle engines (shown schematically in Fig. 31) which could be in commercial operation during the 1970 decade is shown in Fig. 28. Performance data are presented for engines designed to operate at turbine inlet gas temperatures from 1600 to 2200 F without cooling of the compressor bleed air in a separate exchanger, for engines operating at 2200 F turbine inlet temperatures with compressor bleed air precooled to 125 F, and with no compressor bleed air for turbine cooling at all. The Fig. 28 results indicate that if the present practice of using uncooled compressor bleed air is continued, thermal efficiencies and specific outputs as high as 31% and 200 shp per lb/sec airflow, respectively, could be achieved. A comparison of the results obtained at turbine inlet temperatures of 2200 F indicates that precooling the compressor bleed air prior to its use for turbine cooling would account for about a two-percentage point increase in thermal efficiency. The use of precooled compressor bleed air could lower the actual blade operating temperatures by several hundred degrees when compared to the use of uncooled bleed air or alternatively, could reduce the total amount of bleed air necessary to maintain a given blade temperature. The latter approach was used in this study, and for typical conditions the use of precooling reduced the quantity of bleed air required by about 50%. For example, at a turbine inlet temperature of 2200 F and pressure ratio of 16:1, the engine with the uncooled bleed air would require about 11% of the compressor air for turbine cooling while in the precooled design only 6% would be needed. The reduction in bleed flow is largely responsible for the higher efficiency in the precooled designs. The Fig. 28 data also indicate that substantially higher performance could be achieved if no compressor bleed air were required for turbine cooling. This curve implies the use of turbine materials capable of continuous operation at a temperature of 2200 F, and the prospects for achieving such materials in the 1970 decade are not encouraging. However, the results do provide an indication of the performance that might be achieved with future uncooled ceramic composite turbine materials.

Brief studies of the effects of precooled compressor bleed air temperature levels on power plant performance indicated that the cooling air might be provided to the turbine section at temperatures between 125 F and 250 F without serious compromises in performance, thus affording less stringent design conditions on the precooler heat rejection system. Consequently, 200 F was selected as the temperature level for the precooled compressor bleed air, and the performance for simple-cycle gas turbines incorporating 1970-decade design technology is depicted in Fig. 29 as a function of compressor pressure ratio. The Fig. 29 data are based on the use of compressor bleed air precooled to 200 F for turbine cooling and turbine inlet temperatures up to 2400 F. The Fig. 29 results indicate a diminishing rate of improvement in thermal efficiency as turbine inlet temperature is increased above approximately 2200 F. However, significant improvements in specific power continue to be realized above a gas temperature of 2200 F.

The performance of simple-cycle gas turbine systems which could be in commercial operation during the early and late parts of the 1980 decade are summarized in Fig. 30. Thermal efficiency and specific power levels as high as 38% and 270 shp per lb/sec airflow, respectively, should be achievable in the early 1980's with simple-cycle, high-pressure-ratio engines operating with 2400 F turbine inlet gas temperatures and utilizing precooled compressor bleed air turbine cooling techniques. The improved component efficiencies and materials technology projected for the early 1980's (Table XVI) relative to the projections for the 1970 decade result in the superior performance shown in Fig. 30, as compared to the Figs. 28 and 29 performance estimates for a given turbine inlet temperature. The Fig. 30 data also indicate that increases in turbine inlet temperature above 2400 F up to the anticipated level of 2800 F in the early part of the 1980's result in specific power increases of about 20 hp per lb/sec of airflow for each 100 F rise in turbine inlet temperature, with essentially no improvement in thermal efficiency. However, thermal efficiencies of nearly 41% and specific power approaching 400 hp per lb/sec of airflow could be achieved with a 3000-F turbine inlet temperature, a level which could be reached in late-1980 decade engines.

Regenerative-Cycle Engines

If a portion of the waste heat available in the exhaust gases of a gas turbine is used in a regenerator to heat compressor discharge air prior to combustion, a reduction in required fuel flow rates and, hence, significant improvements in thermal efficiency may be realized. The thermal efficiency and specific output performance of regenerative-cycle base-load gas turbines (shown schematically in Fig. 31) operating at a turbine inlet gas temperature of 2000 F and based on 1970-decade technology is shown in Fig. 32. This performance reflects two levels of regenerator total pressure drop, 4% and 8%, and is based upon the assumption of uncooled compressor bleed air being used to cool the turbine section. These engines offer up to seven percentage points improvement in thermal efficiency, depending upon the assumed values of regenerator effectiveness and pressure drop, when compared with a simple-cycle gas turbine designed for 2000 F. Thermal efficiencies approaching 39% are possible with high-effectiveness, low-total pressure drop regenerator designs (Fig. 32). Figure 32 also shows that the peak thermal efficiencies for the regenerative-cycle design occur at relatively low compressor pressure ratios (from about 5:1 up to 8:1); thus, single-spool compressor gas turbine configurations would be possible. The variation of regenerat hot- and cold-side temperatures with compressor pressure ratio for a regenerator air-side effectiveness of 80% is shown in Fig. 33. Relatively high hot-side gas temperatures will exist with compressor pressure ratios of about 4:1, but at a pressure ratio of 10:1 or higher, temperatures below 1000 F would be encountered and mild steel construction could be used. Estimates of the materials required for regenerators at various levels of turbine inlet gas temperature and compressor pressure ratio are shown in Fig. 34. This figure indicates that mild steel could be used for regenerator construction at turbine inlet gas temperatures of about 2200 F if the engine design is based upon a compressor pressure ratio of 10:1 or above. Stainless steels would be required at higher turbine inlet gas temperatures (and the same pressure ratios), and ultimately, nickel-base alloys such as Incoloy 800 would be needed.

The performance of regenerative-cycle, base-load gas turbines utilizing compressor bleed air precooled to 200 F is shown in Fig. 35 for systems which could be in commercial operation during the 1970 decade and early in the 1980's. The utilization of precooled compressor bleed air enables the attainment of 200-F turbine inlet temperatures with 1970-decade design technology and the achievement of thermal efficiencies on the order of 37% with practical regenerator effectivenesses (80 to 85%) as is shown in Fig. 35. This performance reflects about a four-percentage point improvement relative to that of the simple-cycle designs (based on the same design technology) described previously. Furthermore, as previously mentioned, the relatively high thermal efficiency associated with the regenerative-cycle designs is achieved at low compressor pressure ratios (approximately 8:1), by comparison with optimum thermal efficiency simple-cycle designs which require compressor pressure ratios on the order of 16:1 to 20:1. A comparison of Figs. 35 and 30 also shows that the high thermal efficiency levels for the regenerative cycles can be achieved without a compromise in specific output.

Regenerative-cycle designs, based on early-1980's technology, should be capable of achieving thermal efficiencies of about 41% at a turbine inlet temperat of 2400 F, a regenerator effectiveness of 80%, and moderate engine pressure ratios (see Fig. 35). Increases in turbine inlet temperature to 2800 F result in only minimal increases in thermal efficiency, but a 25% increase in specific power. However, with the projected improvements in component efficiencies, and materials anticipated by the late 1980's, additional improvements in both thermal efficiency and specific output can be realized by further increases in turbine inlet gas temperatures to 3000 F. As shown in Fig. 36, thermal efficiencies approaching 45% will be achieved with regenerative-cycle designs, based on regenerator effectiveness values of 80 to 85%.

Compound-Cycle Designs

The addition of reheat and intercooling to the simple-cycle gas turbines (see Fig. 31) offers the potential of achieving further improvements in power plant performance. In the cycle configuration envisioned herein, air enters a low-pressure ratio compressor and is partially compressed. The air is then cooled to a temperature level of 125 F within the intercooler. The cooled air is then further compressed in a high-pressure ratio compressor to the final overall cycle pressure ratios of 50 to 100 atm and heated in the combustor by the combustion of fuel. These high cycle pressure ratios are possible through the use of the intercooling which serves to reduce the total compression work required for a given pressure ratio. Partial expansion of the combustion gases occurs in the gasifier turbine, which drives the high-pressure ratio compressor. Then further heating back to the initial turbine inlet gas temperature occurs in the reheater. Since turbine work is proportional to temperature, the use of reheating increases the total work obtained for a given expansion. Final expansion to atmospheric pressure then occurs in the low-pressure turbine, which drives both the low-pressure ratio compressor and the generator.

A summary of the calculated compound-cycle gas turbine performance is shown in Fig. 37, based on component efficiencies and materials projected for the early 1980's. The results indicate thermal efficiency levels as high as 42% at turbine inlet temperatures of 2200 to 2400 F. Figure 37 also shows that turbine inlet gas temperatures beyond 2400 F would not produce higher thermal efficiencies but would provide practically a linear increase in the specific power capability beyond 400 shp per lb/sec. Specific powers of 400 shp per lb/sec represent a significant 33% increase in specific power beyond that achievable with simple-cycle gas turbines at the same turbine inlet temperatures and technology base (compare Figs. 37 and 30).

Prior to computing the engine performance data depicted in Fig. 37 a preliminary analysis was conducted to determine the effect of varying the amount of compression between the previously defined low-pressure ratio and high-pressure ratio compressors (see Fig. 31). Typical results for total cycle pressure ratios of 50:1 and 100:1, respectively, are shown in Fig. 38. The performance depicted in Fig. 38 indicates that if the low-pressure ratio compressor provides more than 50 percent of the total cycle pressure ratio, the specific power increases while thermal efficiency falls off gradually. Since base-load operation emphasizes the achievement of high thermal efficiencies, the distribution of compressor work was selected to provide maximum efficiency. Thus pressure ratios of 3, 4, and 5 were selected for the low-pressure ratio compressor to correspond with total cycle pressure ratios of 50, 75, and 100, respectively.

Reheat for these designs occurs after expansion to a pressure ratio corresponding to a work split of from 40% of the total turbine work in the relatively low-cycle pressure ratio (50:1) designs to about 65% of the total turbine work in the high-cycle pressure ratio (100:1) designs. If reheat of the combustion gases was provided following complete expansion through the gasifier turbine and prior to the low-pressure turbine, further improvement in specific power would be achieved but at a loss in thermal efficiency. This trend is apparent from an inspection of Fig. 39 which depicts the effects on thermal efficiency and specific output performance of reheat at different levels of expansion through the gasifier turbine.

SELECTION OF GAS TURBINE PARAMETERS FOR MINIMUM-COST POWER

Parametric investigations of the interrelationships among gas turbine performance, design configurations, and engine cost were conducted for the simple-, regenerative-, and compound-cycle gas turbines to determine those system designs which would have the potential for generating lowest-cost electric power within the next two decades. These investigations utilized the gas turbine engine design and costing procedures, developed under Corporate sponsorship, which are described in detail in Appendices B and C, respectively. The selling prices (based on 1970 dollars) were estimated based on a projected market of 4000 Mw per year (see SECTION VIII for further details). Appropriate factors for development, assembly, test costs, general and administrative expenses, and profit are included in the estimated selling price of each unit.

Simple-Cycle Engine Designs

Engine Size

The effect of engine size (output capacity) on the estimated gas turbine specific selling prices is depicted in Fig. 40 for combinations of turbine inlet temperatures and compressor pressure ratios chosen to reflect high-thermal efficiency designs (see Figs. 29 and 30). The results are presented for power turbine output speeds of 3600 and 1800 rpm to match the synchronous rotationals speeds of large electric generators. The trends illustrated by the curves shown in Fig. 40 clearly indicate that specific selling price decreases with unit power capacity up to approximately 100 Mw which is the approximate upper limit for the power which can be developed by a 3600-rpm power turbine. To produce more power than 100 Mw, an 1800-rpm power turbine must be used which initially results in a penalty in specific price relative to 3600-rpm designs. However, as unit power capacities increase, specific price is seen to decrease to apparent minimum attainable levels for capacities in the range of 250 to 300 Mw. Gas turbines of this latter power capacity must be considered to be very large machines compared with those presently in commercial operation. An alternative means of attaining high unit power capacities would be to utilize multiple exhaust ends on the power turbine. This is common practice in steam turbine design as a means of achieving unit capacities of 500 to 1000 Mw. The Fig. 40 data illustrate that the maximum output capacity of gas turbine engines designed with two exhaust ends at 3600 rpm could achieve somewhat lower specific prices and extend the unit capability to about 150 Mw. However, the added complexity of multiple exhaust ends is considered impractical.

The results also illustrate that the selling price of these advanced, large-capacity gas turbines could approach levels as much as 30 to 50% lower than those of present-day gas turbines used in power generation application. Smaller reductions in engine specific selling price ($/kw) are also projected as turbine inlet temperatures are increased toward 2600 to 2800 F, a level representative of early-1980's technology. However, to achieve these cost reductions, blade centrifugal stresses approaching 45,000 psi and above, as indicated in Fig. 40, will be experienced in the last stage of the power turbine, particularly when extending the single-exhaust-end design configurations to output capacities on the order of 75 Mw for 3600-rpm turbines and 250 Mw for 1800-rpm designs.

Beyond output capacities of about 100 Mw, the power turbines must be designed for rotational speeds of 1800 rpm to avoid prohibitive blade stresses. However, for a given turbine inlet temperature and compressor pressure ratio, a step increase in specific selling price occurs, as mentioned previously, when the power turbines are designed for output speeds of 1800 rpm instead of 3600 rpm. This increase is due to the larger power turbine components associated with the 1800-rpm designs. The utilization of an 1800-rpm power turbine enables gas turbines to be designed with a single exhaust end at unit output capacities approaching 250 Mw without exceeding allowable stress limits (see Fig. 40). For 1800-rpm designs, minimum engine selling prices are achieved at about 250 Mw for 2600 F to 2800 F turbine inlet temperature designs and remain relatively constant above the 200 Mw level for the 200 F to 2400 F turbine inlet temperature level representative of 1970 technology designs. The Fig. 40 results provided the basis for concentrating further cost studies on the 200- to 250-Mw size engines for base-load, power generation applications. The high power plant capacities associated with the minimum specific selling price designs also provide further power station cost savings since fewer units and hence less ancilliary equipment and smaller floor space, would be required in a typical large, i.e., 750-Mw to 1000-Mw power station.

Engine Pressure Ratio and Turbine Inlet Temperature

The effects of variations in turbine inlet temperature and compressor pressure ratio on the gas turbine engine selling price are shown in Fig. 41 for engines (based on 1970-, and 1980-decade design technologies) designed to operate at 1800 rpm output speed and to provide a nominal 200-Mw electrical output. All of the engine designs included in Fig. 41 except the 1970-decade, 2000-F turbine inlet temperature engines can employ power turbines with single exhaust ends without exceeding practical blade stress limits of about 34,000 psi in the last stage. Power turbines with two exhaust ends are required in the 2000-F engine designs in order to remain within those specified stress limits. Nearly a 10% reduction in engine selling price could be achieved, as indicated by the circled point in Fig. 41, for the 2000-F engine designs by relaxing the constraint on blade stress, and thereby eliminating an exhaust end, if the blade stress in the last stage of the power turbine is allowed to approach 46,000 psi at the expense of reduced power turbine life. However, even with the relaxation of this design constraint, the specific costs ($/kw) for the 2000-F engine designs remain relatively high.

The estimates in Fig. 41 also indicate that increases in turbine inlet temperature up to 2600 to 2800 F provide substantial reductions in selling price. However, above turbine inlet temperatures of 2800 F, the engine selling prices begin to increase again. A partial explanation of these trends is evident by reviewing the effect of increases in turbine inlet temperatures on the specific power level of the gas turbine (see Fig. 30). As turbine inlet temperature is increased from 2400 F to 2800 F and above, the specific power increases by as much as 50%; thus a lower airflow rate is required to achieve a given power level, and hence smaller and lower-cost components are needed. However, above a turbine inlet temperature of approximately 2800 F, the costs of the hot-section components, i.e., the compressor turbine and the power turbine, begin to increase sharply due to the need for more sophisticated construction materials in each component to withstand high gas temperatures. As a result, the cost reductions for the compressor and burner components, accruing from technology improvements (higher stage loadings, component efficiencies, etc.) as well as higher specific power levels, are offset by the higher costs of the compressor turbine and power turbine sections.

To narrow the range of engine design parameters considered in this study, the influence of compressor pressure ratio on power cost at turbine inlet temperatures corresponding to the various time periods (1970-decade and early and late 1980's) was estimated. The results, based on 80% load factor, 15% capital charges, and 30 and 50¢/million Btu fuel costs, are summarized in Table XVII for selected 200-Mw plants. The effect of variations in the engine specific selling price on the overall cost of the power generating station were also included in the analysis. The results indicate that the cost differentials among the various pressure ratios are generally small. The optimum compressor pressure ratios range from 20:1 at a 2000-F turbine inlet temperature to 36:1 at 2900 F. The improved performance achievable at high pressure ratios is the primary factor in determination of the minimum-power cost system. However, a reduction in the load factor to 70% would tend to favor the lower compression ratio designs which are somewhat less expensive to build. For very low load factors of 20%, comparable to utility peaking applications, compressor pressure ratios of about 15 to 20:1 would be selected for the various time periods. Based on the Table XVII data, attention was focused on simple-cycle engine designs representative of the 1970 decade, operating at compressor pressure ratios of about 20:1. For the early- and late-1980's engine designs somewhat higher compressor pressure ratios, between 24 and 30:1, were selected for the systems which would provide minimum-cost power.

Component Cost Breakdowns

Examination of Fig. 41 also reveals that increases in compressor pressure ratio at any given turbine inlet temperature generally tend to result in higher unit selling prices. As pressure ratio is increased, the number of compressor stages as well as the number of stages in the compressor turbine must increase accordingly. In addition, since an increase in compressor pressure ratio could also mean a decrease in specific output (higher airflow rates for a given output) the sizes of these components increase. As a result, the manufacturing costs of these components are increased. The curves in Fig. 42 provide an indication of the typical distribution of costs among the various major components in an early-1980's gas turbine engine design as pressure ratio is increased. The component costs shown in these figures account for approximately 70% of the total engine cost. Much of the remaining cost is distributed among such components as the casings and the bearings, seals, and shafting. The costs for the assembly and testing of each unit, which account for approximately 10% of the total manufacturing costs, are also not included in the Fig. 42 estimates.

Single- vs Twin-Spool Designs

The estimates presented in Fig. 41 were based upon the use of twin-spool compressor designs and indicate that engine specific selling price tends to minimize at compressor pressure ratios below 15:1. This result suggests that a further reduction in selling price would be possible through the use of a single-spool rather than a twin-spool design. In a single-spool engine design, the desired engine cycle pressure ratio is achieved by a series of compressor stages which are mechanically constrained to operate at the same rotational speed. If the design pressure exceeds about 10:1, an excessive number of compressor stages are required as well as variable-pitch stator blades on the first few stages to avoid compromising the startup and part-load performance characteristics of the engine. In a twin-spool design, the compression process is provided in two separate compressors, each operating at its own optimum rotational speed to provide maximum performance. Thus, the forward, or low-pressure, compressor section is on a common shaft with the low-pressure stages of the compressor turbine. The aft, or high-pressure, compressor section is on a common shaft with the initial (high-pressure) stages of the compressor turbine. The low-pressure spool shaft is concentric with and extends through that of the high-pressure spool. The twin-spool design is a more complicated configuration and requires more bearings, supports, controls, and shafts and thus often more maintenance than a single-spool design.

The variation in specific selling price with compressor pressure ratio for a single-spool engine designed to provide approximately 200 Mw when operating at a turbine inlet turbine of 2200 F is shown in Fig. 43a. The results, shown as a dashed line in Fig. 43a, were obtained using a compressor corrected tip speed of 1000 ft/sec and a diffusion factor of 0.40 (as were the estimates shown in Fig. 41). The specific engine selling prices for the single-spool designs vary between \$32 and \$37/kw, substantially higher than the costs for the twin-spool designs of Fig. 41. If the compressor corrected tip speed were raised to 1150 ft/sec and a diffusion factor of 0.45 were used, a level which is representative of more advanced aircraft engine technology, the selling prices of the single-spool engine designs would be reduced about \$5/kw as shown by the solid line in Fig. 43a. These cost levels are still not lower than those shown in Fig. 41 for the twin-spool designs. Furthermore, compressor tip speeds above 1000 ft/sec could also be utilized on the twin-spool designs to achieve modest reductions in engine selling price. The Fig. 43b data indicate that operation at a compressor tip speed of 1150 ft/sec would eliminate 5 stages from the compressor as well as one stage from the compressor turbine (not shown). The variation in engine selling price with compressors designed for tip speeds above 1000 ft/sec are shown in Fig. 43c. However, unlimited increases in compressor tip speed above about 1200 ft/sec are restricted by a rapid increase in the disk stresses and hence the requirement for more advanced materials.

Power Turbine

Since power turbine component costs are a significant fraction of the total engine costs (Fig. 42), efforts were made to achieve further engine cost reductions from cost optimization studies of the power turbine design configurations. Figure 44a shows the effect on engine specific selling price of variations in the power turbine last-stage hub/tip ratio for a 200-Mw, 1800-rpm gas turbine operating at a 2400-F turbine inlet temperature and a 20:1 compressor pressure ratio. As indicated in Fig. 44a, an increase in hub/tip ratio from 0.50 to 0.70 results in the elimination of two stages from the power turbine (not shown) but the result is less than a 10% reduction in engine selling price. Despite the decrease in blade and vane costs accruing from a reduction in the number of stages as well as from the decrease in blade height, these cost reductions are offset by higher disk costs resulting from the increase in disk diameter. Hence, although two stages have been eliminated by increasing the last-stage hub/tip ratios from 0.50 to 0.70, disk costs have also increased by a factor of almost two. Further cost studies were confined to the range of hub-to-tip ratios between 0.50 and 0.70.

Exit Velocity

A more pronounced effect on engine selling price as well as on engine performance appears to be due to the limits placed on the exit velocity of the gases leaving the last stage in the power turbine. A large annulus area reflects the use of long, highly stressed, high-cost blades and vanes and low exit velocities. The low exit velocities permit more efficient extraction of the energy in the gas (lower leaving losses) and thus more efficient power plant performance. This trend is shown in Fig. 44b which depicts the effects of power turbine exit velocity on engine specific selling price and thermal efficiency performance for a 200-Mw, 1800-rpm, gas turbine designed to operate at a 2400-F turbine inlet temperature and a 20:1 compressor pressure ratio. It can be seen from Fig. 44b that an increase in exit velocity from 500 to 800 ft/sec affords approximately a 20% decrease in specific selling price, but this is accompanied by approximately an 0.8% loss in thermal efficiency. However, a brief analysis indicated that the reduction in engine selling price of 5 to 7 $/kw would offset the higher fuel costs due to the lower power plant efficiencies; hence, the remaining engines were designed for exit velocities of 600 ft/sec or more. The loss in engine performance is minimized by installing a suitable diffuser having a conservative diffuser energy recovery coefficient downstream of the power turbine.

Materials Changes

The effects on the engine selling price of substituting lower-cost materials and of relaxing the first-stage hub-to-tip ratio design constraints in the power turbine were also studied. Attempts to maintain the hub-to-tip ratio at 0.85, a value which generally ensures low turbine losses, had resulted in relatively long last-stage turbine blades for the engines which could be commercially available by the late 1980's. As a result, these engines were found to be somewhat more costly than engines investigated for the earlier time periods (see Fig. 41). However, by relaxing the hub-to-tip ratio constraints to a level of 0.875, a substantial reduction in engine selling price could be achieved, especially as indicated in Fig. 45, at the highest turbine inlet temperature of 3100 F.

The Fig. 45 results also indicate that, because of increased cooling requirements, the substitution of advanced nickel alloy blades and vanes for the more costly columbium alloys tends to increase the engine selling price by about 10 to 13%. At turbine inlet temperatures of 2900 and 3100 F as many as three stages of the power turbine must be cooled if nickel-based alloys are used, and although the nickel-based alloys are less expensive than columbium alloys the costs associated with providing cooling increases the engine selling price.

Coating Life

During the course of the study, additional data were received indicating that the effects of erosion and corrosion on the compressor turbine initial-stage vanes could be the limiting compressor turbine design parameter rather than vane creep strength. The data indicate that at the assumed vane operating temperatures (i.e., 1700 F, 1900 and 2100 F for the 1970-decade and early and late 1980's, respectively) coating life could be relatively short, on the order of approximately 10,000 hr. The coating life would be considerably shorter if ash bearing fuels containing even a small quantity of sulfur were used. Although recoating of the vanes at 10,000 to 15,000-hr intervals would be possible without incurring high maintenance costs, a more practical base-load engine design would utilize lower vane operating temperatures by using a higher percentage of compressor bleed air for vane cooling. The effects of a reduction in allowable vane temperature on engine performance were investigated (the selling prices would remain the same) and typical results are presented in Fig. 46. The Fig. 46 data indicate that an eight-fold increase in coating life could be achieved by cooling the vanes about 200 F lower than the originally assumed temperatures for a representative early-1980's technology engine designed to operate at a turbine inlet temperature of 2600 F and a 28:1 compressor pressure ratio. For example, if the vane temperature were maintained at 1700 F rather than 1900 F, a coating life of about 80,000 hr could be achieved although the thermal efficiency and specific power would be reduced about 1.5% and 3.2%, respectively, due to the added compressor bleed flow required. A 3-to-1 improvement in coating life could be achieved with only a 125-F decrease in allowable vane metal temperature while the performance losses would be reduced. The Fig. 46 results are based on currently available and projected performance of turbine vane coatings; however advanced coatings under investigation are expected to provide protection for time periods on the order of several years. The performance penalties estimated for lower vane temperatures were, nonetheless, included in the final engine efficiency estimates.

Regenerative-Cycle Engine Designs

To define the regenerative-cycle engine designs that have the greatest potential for generating the lowest-cost electric power, the effects of a number of engine and heat exchanger design parameters were investigated, including the type of regenerator surface, gas pressure drop, recuperator* effectiveness and flow arrangements.

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* Recuperator and regenerative are used interchangeably although the type of heat exchanger considered is stationary and heat must be transferred across a surface.

Recuperator Surface Characteristics

The regenerative-cycle gas turbine characteristics presented herein are based on the use of plate-fin type recuperator surfaces although bare- and finned-tube type surfaces were considered as well. Plate-fin type surfaces have many advantages in large industrial applications such as those considered here, probably the most important of which is the ease in which these surfaces can be used in modular construction of the recuperator. Modular construction permits the use of a number of similar small, readily manufactured units to be built up into a large installation such as that advocated in Ref. 103. Compact plate-fin surfaces with high heat transfer surface per unit volume also result in minimal exchanger dimensions so that the station foundations and building size can be minimized. Although tube-type surfaces can be made relatively compact, with surface area densities approaching those of plate-fin exchanges, very small tube diameters (on the order of 0.125 in.) and thin-tube wall thicknesses (of approximately 0.004 in.) are required (see Ref. 88). In the Ref. 88 study, both bare-tube and plate-fin type surfaces were considered, but the application was for a very small military gas turbine engine with an airflow of 5 lb/sec rather than the 1000 lb/sec engines considered in base-load plants. To build a tube-surface recuperator for a 200-Mw base load engine considered in this study, tube header attachment costs would be high for the many miles of small-diameter tubes required. Exploratory calculations performed at the Hamilton Standard Division of United Aircraft (HSD) also indicated that the no-flow direction with tube configurations would be several hundred feet long and introduce difficulty in devising a compact arrangement.

A number of different plate-fin type surface geometries were initially considered, but attention was focused on the compact configurations. Although the Harrison industrial gas turbine recuperators have been constructed with a rather course matrix (1-in. fin height on the gas side and a channel with no fins on the air side), both Ref. 105 and exploratory calculations performed at UARL indicate that significant reductions in core volume could be achieved with a finer matrix (smaller fin height and many fins per inch). Smaller core volumes result in smaller foundations, easier transportation to the site, and reduced overall plant size. The surface geometries employed in the industrial gas turbine recuperators described in Ref. 103 are very fine, with 20 fins per inch and 0.10-in. fin height on the air side and 16 fins per inch and 0.075-in. fin height on the gas side.

Discussions with HSD revealed that the selection of surface fin height and fin spacing will be a function of the desired volume and cost criteria for the application. Furthermore the costs are more sensitive to the number of pieces handled in the fabrication processes than the manufacturing techniques. It was evident that, to perform trade-off analyses in hopes of determining the surfaces which would result in the minimum cost power, a complete breakdown of material and fabrication costs would be required for each candidate surface. This type of generalized cost information was not available from recuperator manufacturers, so a few representative surface geometries were selected from Ref. 104 which contains the basic pressure loss and heat transfer data needed to perform the other trade-off analyses conducted in this study. As a guide, surfaces similar to those recommended in Ref. 103 were selected. Those which appeared to provide the smallest core volumes were plate-fins with 11.1 plain fins per in. on the gas side and 19.8 plain fins per in. on the air side, both with fin heights of 0.25 in. This combination of geometrical characteristics results in a core with heat transfer surface of approximately 464 ft^2/ft^3 of core volume.

Effectiveness

Recuperator air-side effectiveness is one of the most important factors affecting engine performance, recuperator size, and system cost. The data in Figs. 32 and 35 illustrate that a significant increase in gas turbine thermal efficiency can be achieved by increasing recuperator air-side effectiveness. For example, for a turbine inlet temperature of 2000 F and a compressor pressure ratio of 8:1, representative of a 1970-decade engine, the thermal efficiency is increased by approximately 4 points (from approximately 34% to 38%) by increasing effectiveness from 70% to 90%. Therefore, estimates were made of the effects of recuperator effectiveness on recuperator size and cost, and the results are shown in Fig. 47 for a 2000-F turbine inlet temperature engine capable of operating at a pressure ratio of 8:1. The data clearly show the rapid increase in volume requirements, and hence heat transfer area, as the effectiveness approaches 90%. However, the slope of these lines is a strong function of the type of surface selected for each side of the exchanger and the absolute size of the heat exchanger. Furthermore, the UARL data in Fig. 47 are based on a cross-counter-flow type of flow arrangement and the relative increase in recuperator size with effectiveness is influenced by such design characteristics as pressure drop and the coarseness of the heat transfer matrix. Also shown in Fig. 47 is a manufacturer's estimate of the variation in recuperator cost with effectiveness. This curve is for a different plate-fin type of construction and a smaller size than the other regenerators of Fig. 47.

In order to determine the recuperator effectiveness that would result in minimum-cost power, analyses showing the trade-off between engine operating and capital costs were made using both UARL and manufacturers' recuperator cost estimates and assuming various capital charges ranging from 12 to 17% and for fuel costs of 30 and 50¢/million Btu. The results in terms of the added total power costs above a base value of regenerator effectiveness are shown in Fig. 48. For example, using 30¢/million Btu fuel and 12% capital charges, the minimum costs would occur at a recuperator effectiveness of 80% based on manufacturers' estimates of the cost variation between 70 and 90% effectiveness. The total power costs using a 90% recuperator effectiveness would be at least 0.07 mills/kwhr above the minimum, or base, case. In general, the results indicate the minimum power costs occur at approximately 80% effectiveness and therefore this value was used in conducting the remaining trade-off analyses made in this study. Even at very high fuel costs of 50¢/million Btu, a 90% effectiveness provides minimum costs only when capital charges are about 12%. Data from recuperator manufacturers and Ref. 105 also indicate that 80% effectiveness is approximately the design value for many industrial gas turbines regenerators previously built.

Total Pressure Loss

Another relatively important parameter considered in the design of a regenerative-cycle gas turbine is the recuperator total pressure loss. The effects of total pressure loss on the size characteristics of recuperators for gas turbines designed with 1970 technology are illustrated in Fig. 49, and the effects on the performance are shown in Fig. 32. The results presented in Fig. 49a indicate that, as the total pressure loss is increased from 4.5 to 13.5%, the decrease in recuperator core volume is most pronounced at the highest value of effectiveness (90%). In designing the recuperators represented in Fig. 49a it was assumed that $\frac{2}{3}$ of the total pressure loss occurs in the core and the remaining $\frac{1}{3}$ in the manifolds and ducting. The total pressure loss has even a more pronounced effect on the regenerator no-flow length i.e., (see sketch in Fig. 49), an important item when considering the problems of integrating the engine and recuperator, which would affect the overall space requirements of the system.

The effect of varying total recuperator pressure loss between 4 and 8% on gas turbine performance is shown in Fig. 32, and the data indicate only about a one point reduction in thermal efficiency and about a one-third of a percent reduction in specific power at the highest pressure loss. Since the recuperator costs about $15/kw of engine output, the savings in recuperator cost for an 8% total pressure loss level would offset the loss in efficiency relative to a recuperator designed for only 4% total pressure loss. Thus, further efforts were confined to recuperators based on an 8% pressure loss.

Pressure Loss Split

The split of the total pressure loss between the air side and gas side is another independent parameter which must be considered in designing a cross-flow or multipass cross-counterflow type heat exchanger. The effects of varying the pressure loss split are exemplified in Fig. 49b. For the conditions represented in Fig. 49b, the core volume continues to decrease as the percent of total pressure loss on the gas side increases, but the no-flow length appears to be a minimum when the split in pressure drop between the air and gas side is equal. Thus a split of $\frac{2}{3}$ of the total pressure loss on the gas side, the value assumed for the recuperator designs in Fig. 49a, appears to result in a near-optimum recuperator core configuration.

Flow Arrangement

Rough layouts of the engine and recuperators utilizing 2-passes (on the air side) designed for 80% effectiveness indicated that compact configurations using the cross-counterflow arrangement would require no-flow lengths of 100 ft or more (see Fig. 49b) and did not appear practical. Other cross-counterflow arrangements suggested by the various aircraft recuperator designs analyzed in Ref. 88, which involve the use of multipasses on both the gas and air sides, were considered briefly. However, it appeared that these arrangements required relatively complex ducting and manifolding and the designs were tending toward pure counterflow systems. Thus, counterflow recuperator designs were investigated and found to be more desirable for the large industrial engines considered in this study. Counterflow designs usually involve more difficult header configurations but provide far more efficient heat transfer and, as shown in Fig. 50, result in significantly smaller core volumes than the cross-counterflow designs investigated. Furthermore, with the counterflow arrangement there is a certain degree of flexibility which allows for achieving reasonable no-flow lengths for the core. With the counterflow design, the flow lengths for the compressor discharge air and turbine exhaust gases must, by definition, be equal, and for given heat transfer surfaces and heat rejection loads, the split in total pressure is also fixed, as is the core face area. Thus, the two no-flow lengths with a counterflow design can be selected to suit the most convenient arrangement of engine and recuperator, the only stipulation being that the product of the two no-flow lengths is equal to the required core face area.

Compressor Pressure Ratio

Compressor pressure ratio was varied for 1970-decade and 1980-decade engines to assess its effect on the recuperator size. The estimates in Fig. 50 show the reduction of recuperator core volume with increased compressor pressure ratio for engines with 200-Mw output capacity. This trend is due to the higher air densities and lower airflow requirements (for a given output capacity) associated with the higher compressor pressure ratios. The smaller core volumes for the 1980-decade designs relative to the 1970-decade designs are also due to the lower airflows required to yield 200 Mw of output power, since the early 1980's design can achieve substantially higher specific outputs. However, the 1980-decade designs operate at higher temperatures and require the use of better materials in the recuperator and more sophisticated fabrication techniques; these factors are reflected in the recuperator costs.

To determine the compressor pressure ratio that would provide minimum power costs for regenerative-cycle engine power systems, costs were obtained for counterflow recuperator designs and combined with engine costs for a range of selected operating conditions. Counterflow recuperator costs were estimated from the required heat transfer surface area, as determined through the use of an existing UARL heat exchanger computer program, and specific cost factors ($\$/ft^2$) obtained from a correlation of manufacturers' data (Refs. 88, 102, and 106). These data were obtained in response to inquiries and extensive discussions with manufacturers' representatives for recuperators using a variety of different construction materials. For those selected operating conditions where the temperature capabilities of mild steel were exceeded, a correction factor was applied to the specific cost for mild steel recuperators to account for the use of better materials such as Type 347 stainless or Inconel 800. For the recuperator sizes investigated (between approximately 1 to 3 x 10^6 ft^2) the specific cost of mild steel recuperators were estimated to range between \$1.00 and \$0.80/ft^2. Smaller recuperators would have a higher specific cost. The recuperator cost factor, as a function of maximum metal temperature, is shown in Fig. 51a and illustrates the sharp increase in cost as maximum metal temperature increases. The effect of temperature on cost was discussed with recuperator manufacturers on a number of occasions and it appears that temperature not only affects the selection of the base material but also the brazing material and fabrication technique. Furthermore, in a very large recuperator utilizing modular construction, it becomes economically attractive to use more than one material, the choice depending upon the local temperatures encountered. However, the cost factor is depicted in Fig. 51a as a smooth curve rather than a series of step increases which would reflect the use of more expensive material. Mild steel would be used for metal temperatures below approximately 950 F corresponding to the maximum gas temperature of approximately 1000 F at the turbine exhaust. As the temperature to which the recuperator materials will be exposed is increased, Type 430 stainless steel might be used. Type 347 stainless or Incoloy 800 would be used up to metal temperatures of approximately 1300 F. Above this temperature level, more exotic and more expensive base materials would be required, but the recuperator temperatures considered in this study generally did not exceed a temperature level of 1300 F. The maximum metal temperatures and estimated recuperator costs for the counterflow designs represented in Fig. 50 are shown in Fig. 51b with the maximum metal temperature taken to be the average of recuperator gas inlet and air outlet temperatures. Since the maximum metal temperatures depicted in Fig. 51b do not exceed the 1300-F level, base materials no better than Type 347 stainless or Incoloy 800 would be required in the 1970-decade and early-1980 engines. The decrease in recuperator cost with engine pressure ratio (for a given turbine inlet temperature) is also illustrated in Fig. 51b, and it appears that the trend is due primarily to the decrease in maximum metal temperature. Thus, maintaining metal temperature below approximately 1000 F appears to be an important factor in determining recuperator cost.

Representative results showing the effect of compressor pressure ratio on the combined engine and recuperator costs are presented in Fig. 52b.. These costs are based on the regenerative engine costs as shown in Fig. 52a, using conservative levels of compressor tip speeds of 1000 and 1100 ft/sec for the 1970-decade and early 1980's. Also shown in Fig. 52a are the thermal efficiency levels estimated for the regenerative-cycle engines. Combining the hardware and fuel costs in separate calculations indicates that minimum total power costs would be achieved by selecting compressor ratios of about 9:1 or 10:1 for turbine inlet temperatures of 2000 F and about 12:1 for the early-1980 time period when turbine inlet temperatures of 2400 F and above will be utilized. Thus, these designs were utilized in the economic comparisons with steam plants that will be described in SECTION VIII.

Compound-Cycle Designs

The very high specific output capabilities associated with the compound-cycle designs (see Fig. 31b) provided an incentive to estimate representative compound-cycle gas turbine power plant costs despite the fact that large quantities of heat would have to be rejected from this cycle via the system intercooler. Since relatively high cycle pressure levels of 50 to 100 atmosphere are required within some of the cycle components to make this system attractive, moderate compressor design parameters, a maximum cycle gas temperature of 2200 F in both the primary and reheat combustors, and a total cycle pressure ratio of 75:1 were selected for evaluation. These values are representative of projections of early-1980's design technology. Since the low-pressure ratio compressor is driven by the 1800-rpm power turbine (see schematic diagram in Fig. 31b), system output capacity and hence system size was established on the basis of the maximum airflow handling capacity of an 1800-rpm compressor with 1100 ft/sec tip speed design characteristics and a maximum airflow rate per unit flow area of 32 lb/sec per ft^2. These basic assumptions provided the basis for the design of a nominal 470-Mw power plant with approximately 2000 lb/sec airflow handling capability.

The design configurations and pertinent dimensions of the major components for the 470-Mw compound-cycle gas turbine power plant are presented in Table XVIII. The specific outputs associated with this cycle configuration (see Fig. 37) are approximately 25% higher than those associated with simple-cycle configurations (Fig. 30) and offer the use of more compact turbomachinery units and, hence, the realization of attractive selling prices approaching $20/kw. This selling price includes the price of the power plant plus dry cooling tower and the recirculating water cooling systems required to dissipate the heat of compression rejected from the compressor stage intercooler.

The costs of the intercoolers employed in the compound-cycle gas turbines and the air precoolers employed in the simple-cycle or regenerative-cycle power plants are based on the use of a closed cooling water loop. The compressor air is cooled with water which is heated and in turn circulated and cooled in a dry cooling tower. This arrangement permits locating the relatively large dry cooling towers remotely from the gas turbine engines with little cost penalty. The water-cooled air preheater and intercooler sizes were calculated using an existing UARL heat exchanger computer program, and costs were estimated using manufacturers' data in accordance with the heat transfer areas required. The dry cooling tower costs were estimated using the procedure and data contained in

Ref. 64. These costs include allowances for fans, motors, and circulating pumps. The cost of the water-cooled air precooler or intercooler for a given system was found to be small compared to the dry cooling tower. Because even the dry cooling tower costs were estimated to be small compared to the total installed engine costs they were checked against the steam power plant dry cooling tower system costs given in Ref. 64. After adjusting for differences in heat loads and temperature levels, the dry cooling tower costs estimated in this study and those in Ref. 64 were found to be in close agreement.

Although the compound-cycle power plant was estimated to be a most attractive gas turbine system for base-load power generation, with substantial potential economic and performance merits, the use of cycle operating pressures of 50 atmospheres and above would introduce significantly higher blade bending stresses and require more extensive design analysis and cost efforts than that provided in this study. It is recommended that these studies be pursued in additional programs to assess the application of gas turbines for utility power generation systems.

ADVANCED GAS TURBINE STATION CHARACTERISTICS

The results of the performance and cost studies described in the previous sections permitted the selection of simple- and regenerative-cycle engine designs which are judged potentially capable of producing low-cost electric power without river and lake water thermal pollution for the 1970 and 1980 decades. The design characteristics, engine and overall station performance levels, and estimated engine and station selling prices are summarized in Table XIX. Complete details describing the basis for the gas turbine station selling price estimates are presented in SECTION VIII of this report. Station thermal efficiency levels of 30% to almost 39% are projected for the simple-cycle engines, and are as much as three percentage points higher for the regenerative-cycle engines. Included in the station net heat rates are allowances for (1) the power requirements of the station auxiliaries; (2) mechanical losses in the electric generator; and (3) reductions in gas turbine engine performance due to operation at lower vane temperatures and with higher exit velocities than the values assumed in the parametric performance studies. These losses contributed to a 4 to 6% reduction in station heat rate. No performance penalty has been included in the data for operation at off-design conditions. Typically, the performance of steam power stations is reduced by about 5% to reflect off-design operation. It is anticipated that these losses would be negligible in gas turbine stations since the availability of multiple units in a typical large power generating station would provide the flexibility to meet off-design conditions. However, the effects of operating at part-load conditions and at varying ambient conditions are discussed in Appendix D. Total station specific prices, in $/kw, are about 20 to 30% less than present-day gas turbine prices.

The engine selling price estimates are based on the corporate-sponsored computer program developed to estimate the manufacturing and selling prices for selected simple-cycle and regenerative-cycle designs. The devleopment costs for advanced gas turbines have been included in arriving at the estimates of engine selling price. To assist in these efforts, a conceptual design of a simple-cycle base-load gas turbine engine which could be in commercial operation during the early part of the 1980 decade and capable of producing 200-Mw was prepared (see Fig. 53). The engine conceptual design is based on a compressor pressure ratio of 32:1, a turbine inlet temperature of 2400 F, and an airflow of 1176 lb/sec. The overall length of the engine would be slightly less than 60 ft. The conceptual design shown in Fig. 53 differs only slightly with the simple-cycle engine incorporating the early 1980 technology advances selected for use in the comparative studies. The pertinent power plant design characteristics for this engine are summarized in Table XX, and the temperatures, pressures and flow rates are shown in Fig. 54. The primary dimensions of the turbomachinery were obtained from supplementary design procedures and from layouts of the aerodynamic flow path.

The station cost estimates for the 1000-Mw simple-cycle gas turbines utilizing the early 1980's technology were based on the utilization of four 260-Mw units designed to operate at the conditions specified in Table XIX and Fig. 54 for the engine size estimates provided in Table XX. An elevation drawing of such a station, illustrating the placement of one of the four engines, is shown in Fig. 55. A plan view of the station, which would require an area only approximately 165 x 200 ft, is shown in Fig. 56. Arrangement of the gas turbines was made after consideration of the space requirements needed for placement and maintenance requirements of the engine. The plan view shows atmospheric air-cooled heat exchangers which would be used to remove heat from the compressor bleed air, thereby precooling it prior to its use for turbine cooling. The turbine air precoolers, engine oil coolers, and generator coolers are located below the main floor level as indicated in Fig. 55, and would be connected to the atmospheric-air coolers outside the building. Thus, such a power system could be operated independently of a cooling water source and would require a site area about an order of magnitude less than the conventional system. For example, the total site requirements for the 1000-Mw gas turbine power plant shown in Fig. 56 might be only 500 x 500 ft an area about 6 acres square including allowances for a 100-ft exclusion distance, parking area requirements, etc. This size would be about 0.006 acres per Mw of output. Based on data in Ref. 108, gas-fired steam plants require about 0.10 acre per Mw of output.

The pertinent dimensions of the selected early 1980-decade regenerative-cycle engine are given in Table XX. The temperatures, pressures and flow rates at the various locations in the engine design are shown in Fig. 57, and a flow-path diagram is shown in Fig. 58. The gas turbine unit shown in Fig. 58 incorporates an 80% effectiveness recuperator, arranged in a counterflow configuration consisting of multiple modules arranged seven deep by seven in a radial array. The overall dimensions of the regenerative-cycle engine would be 60 ft long with a maximum radial dimension of about 35 ft. The station dimensions for the regenerative-cycle engine would be about 30% greater than for the simple-cycle gas turbine station, and this added area is reflected, in part, in the higher station selling prices.

SECTION VIII

POWER GENERATION COSTS FOR SYSTEMS DESIGNED TO ELIMINATE THERMAL POLLUTION

SUMMARY

An investigation was undertaken to estimate and compare the costs of producing electric power with advanced open-cycle base-load gas turbine stations and advanced fossil-fueled steam stations designed to reduce or eliminate thermal pollution during the 1970 and 1980 decades. A brief review was made to establish the capital charges, interest rates during construction, and maintenance and supervision costs for the competitive systems. Estimates were made for the total installed station capital costs of power generating systems based upon the use of simple and regenerative-cycle gas turbines and conventional steam turbines for the six major FPC regions. Detailed technical characteristics are presented for selected advanced-cycle power stations considered representative of types which could be located in the South Central region, and estimated busbar power costs for these stations are presented and compared. The sensitivity of the results to the basic values used in the study are examined. Potential advantages that would accrue to electric utilities due to wider selection and availability of plant sites, freedom from power plant cooling water requirements with open-cycle gas turbines, and reduced transmission and lower reserve margin were identified and evaluated. Estimates are provided of the time and approximate cost required to develop commercial base-load gas turbine power stations. These development costs are compared with the incremental capital costs which electric utilities will be obliged to spend through 1990 for cooling towers and cooling ponds to eliminate thermal pollution from conventional steam plants.

For the purposes of this study, power stations with nominal 1000-Mw capacity were selected as the basis for the total owning and operating cost comparisons. These comparisons were focused on stations which could be installed in the South Central region during the next two decades. In this region natural gas is projected to be readily available at moderate price levels, the cooling water shortage is already acute, and the growth of demand for electric power generation is expected to continue at or above the present rate.

CAPITAL INVESTMENT AND OPERATING COSTS FOR ADVANCED POWER GENERATING SYSTEMS

Steam System Costs

Capital and operating cost estimates are presented in the following paragraphs for conventional coal-, residual oil-, and natural gas-fired steam power generating stations with performance and size characteristics identified in Section VI to be consistent with stations available for commercial operation in the 1970 and 1980 decades. The data from various references summarized in Section VI of this report provide the basis for the steam station costs. Station costs include all plant equipment up to and including the main station transformers as well as the various factors for escalation, interest on capital during construction, etc. The costs are presented in terms of 1970 dollars for stations capable of providing 1000 Mw of electric power.

Station Investment and Total Installation Costs

Station investment and total installed costs were estimated for coal-, residual oil-, and natural gas-fired steam power generating stations projected to be representative of those in commercial operation during the 1970 and 1980 decades. The steam stations placed in operation during the 1970 decade would consist of two 500-Mw units, each operating at 2400 psig/1000 F/1000 F steam conditions. The basis for these projections and the performance characteristics are presented in Section VI of this report. The actual investment and total installed station costs are based on data presented in Refs. 109, 44, and 45. For example, data are presented in Ref. 45 for a typical coal-fired station located at a mine-mouth site in the East Central region. The same reference also provides extensive station cost data for oil-fired and natural gas-fired plants which reflect typical installations located in the Northeast region. On the basis of these data, plus that available in Refs. 109 and 44, costs were estimated for 1000-Mw fossil-fueled power generating plants that could be in commercial operation during the 1970 and 1980 decades. The installed costs of stations incorporating design advances projected to become available during the 1980 decade were estimated by applying cost scaling factors to the itemized station cost estimates of the comparable present-day designs, described in Section VI, to account for (1) differences in unit size, and (2) for projected improvements in boiler design and operating efficiency. All of the coal-, oil-, and gas-fired stations incorporate design features that provide full protection against the weather. Finally, a period of four years from date of planning to date of operation, as determined from actual construction practice, served as a basis for station cost estimating purposes.

Installed cost summaries for the 1980-decade station designs are presented in Table XXI according to standard FPC account categories which include land and land rights, structures and improvements, boiler plant equipment, turbo-generator units, accessory electric equipment, miscellaneous power plant equipment, and miscellaneous station equipment. The cost of natural draft wet cooling towers (approximately $8/kw) has been included for each of the three stations and is attributed to FPC Account 314 Turbine Generator Units. The costs of 750-ft and 600-ft stacks, with appropriate electrostatic precipitators or mechanical cyclone dust collectors, have also been included for the coal- and oil-fired units, respectively (see Item 312). Also included in the cost summary are other expenses associated with startup, testing, temporary facilities, temporary buildings, removal of temporary facilities, cleanup, and security guards. Indirect construction expenses for engineering, design, construction supervision, and contingency as well as escalation (at 3.5% per annum compounded), and interest during construction (at 8% per annum) are shown in this summary table. It can be seen from Table XXI that the indirect costs account for approximately 33% of the total station cost. The total station costs when all factors have been included amount to approximately $165.6, $153.9, and $137.9/kw of installed capacity for the coal-, oil-, and gas-fired plants, respectively.

Regional Steam-Electric Station Costs

Station cost estimates for coal-, oil-, and gas-fired 1000-Mw steam-electric station designs that might be installed in each of the six FPC power regions were made by applying appropriate correction factors based on Handy-Whitman correlations, furnished by the architect-engineering firm of Burns and Roe, Inc. These regional installed station costs are presented in Tables XXII and XXIII for 1970-decade and 1980-decade designs, respectively.

Station costs based on outdoor construction as well as indoor construction have been included in Tables XXII and XXIII to provide realistic bases of comparison between systems in those regions of the country where indoor construction features may not be needed. As indicated in Tables XXII and XXIII, the outdoor plants provide a net $15/kw saving in total station installed cost over the indoor designs. This saving in total installed cost results from a $10/kw reduction in building cost (FPC Account 311), substantiated in Refs. 29 and 108 to arise from the elimination of much of the enclosure around the powerhouse. The 1980-decade designs installed costs in Table XXIII are also presented for different rates of interest during construction to illustrate the effects of changes in interest rate on construction costs. Whereas 6.25% was the existing interest rate only two years ago, these rates have currently climbed to 8% and higher. A comparison of Tables XXII and XXIII shows that the 1980-decade stations enjoy approximately a 10% decrease in station cost by comparison with the 1970-decade systems. This cost saving is achieved from the improved steam conditions and from the economies of scale.

Annual Owning and Operating Costs

The total annual owning and operating costs for the steam-electric stations as well as for the gas turbine power stations described in the following paragraphs are expressed in terms of mills/kwhr and are equivalent to the busbar power cost. In order to determine busbar power cost, the annual hours of operation were selected on the basis of an annual load factor of 70% (equal to 6132 hr/yr), consistent with present experience in conventional-fueled, base-load, steam-electric systems. Although load factors as high as 80% are common in nuclear-fueled base-load plant operation, fossil-fueled plants tend to achieve a lower utilization per year. The busbar power cost consists of three (see Refs. 108 and 2) items as follows: capital charges, operating and maintenance charges, and fuel charges. The capital charge is the yearly owning cost and includes allowances for interest on borrowed capital, amortization, insurance, taxes, annual maintenance, and depreciation. A total annual fixed charge of 15% (based on the items in Table XXIV), consistent with current cost estimating practice was used as the basis for computing capital charges. These charges are higher than past practice indicates but as noted in Ref. 110 are the result of today's high interest rate levels. Municipalities or federally owned utilities capable of borrowing at lower interest rates would use capital charges several points lower than 15%. Supplies and materials were assessed at a constant cost level of 0.200 mills/kwhr; however, operating and maintenance labor costs were selected as 0.160 and 0.172 mills/kwhr for the 1970-decade oil- and gas-fired stations and the coal-fired stations, respectively. These costs are consistent with similar data published in the available literature and have been substantiated through discussions with engineering-architect firms (Refs. 2, 29, 45, 52, 108, and 109). It was assumed that the utilization of a single 1000-Mw unit in the 1980-decade designs in place of the two 500-Mw 1970-decade units along with anticipated advances in system designs could afford a 0.05 mill/kwhr reduction in the operating and maintenance cost for the 1980-decade designs. Fuel costs were calculated on the basis of projected fuel prices in Section VI of this report and net station heat rates that were modified to compensate for part-load operation and station startup.

Gas Turbine System Costs

Detailed capital and operating cost estimates (based on 1970 dollars) are presented in the following paragraphs for simple-cycle and regenerative-cycle, natural gas-fired 1000-Mw gas turbine power stations. The capital costs for the gas turbine systems include all plant equipment up to and including the main station transformers.

Capital Costs

Detailed cost estimates were made for a number of engine designs reflecting the 1970 and 1980 decades. The results of these estimates were used as the basis for the preparation of Table XVIII. Detailed capital cost breakdowns are presented in Table XXV for early 1980's-decade simple-cycle and regenerative-cycle 1000-Mw gas turbine power systems. As previously mentioned, the gas turbine station designs (see Figs. 55 and 56) comprise four 260-Mw units for the simple-cycle configuration and five 208-Mw units for the regenerative-cycle configuration. The detailed cost breakdowns in Table XXV are presented according to the standard Federal Power Commission (FPC) account categories described in the preceding section of this report. The stations were designed and cost estimates made for full protection against the weather. The costs of the outdoor construction designs were then estimated by applying a 30% correction factor to the cost of structures corresponding to the indoor designs. Since the gas turbine designs are relatively simple by comparison with the steam-electric stations and, in addition, many of the major components (gas turbine, generators, etc.) are delivered to the site in modules, a construction time (as previously defined date of planning to date of operation) of two years was considered to be adequate for cost estimating purposes. Present gas turbine installations using aircraft-type engines are generally completed in slightly more than one year.

Summaries of early 1980-decade station costs are presented in Table XXVI. It can be seen from this table that differences in costs between indoor and outdoor construction amount to only about $1 to $1.5/kw due to the relatively compact construction associated with the gas turbine station designs. In addition, primarily as a result of the shorter construction time for the gas turbine plants, the sum of the costs for indirect expenses (i.e., engineering design, escalation, and interest during construction) accounts for approximately 21% of the total installed gas turbine station costs; this compares with the aforementioned 33% value for the steam stations.

Since on-site construction costs are held to a minimum with the gas turbine plants, due to the fact that most components are assembled before they arrive at the plant site, regional differences in installed station costs are not as significant for these stations as they were for the steam power stations (see Table XXIII). Therefore, gas turbine installed station costs were assumed not to vary among regions. The total installed costs for the early-1980's design simple-cycle and regenerative-cycle designs when all factors have been included amount to approximately $66 and $92/kw, respectively. Similar cost estimates were made for the 1970-decade and late-1980 decade station designs by applying appropriate scaling factors to the FPC account categories for the early 1980-decade designs. Total installed costs of $80 and $100/kw were predicted for 1970-decade simple-cycle and regenerative-cycle gas turbine power station designs, respectively, and $71 and $93/kw for the late-1980's simple-cycle and regenerative-cycle designs, respectively (see Table XIX).

Annual Owning and Operating Costs

Annual owning and operating costs for simple-cycle and regenerative-cycle 1000-Mw gas turbine power systems assumed to be placed on line during each of the three time periods of interest were computed for those regions of the country in which natural gas now is or will in the future become a competitive source of fuel. The annual capital charges (mills/kwhr) were determined on the basis of an 8% interest rate during construction, 15% per annum fixed charges, and a 70% average annual load factor. Fuel costs (mills/kwhr) were calculated on the basis of projected fuel prices for different regions presented in Section VI of this report, and from net station heat rates which include an adjustment of 4% to compensate for generator losses and other house loads, and appropriate correction factors which include the small performance penalties previously described in Section VII. Supplies and materials were estimated to account for 0.200 mills/kwhr. This is the same value estimated for the steam-electric stations. Operating and maintenance costs were estimated at 0.500 mills/kwhr and 0.600 mills/kwhr for the simple- and regenerative-cycle stations, respectively, on the basis of discussions with the Burns and Roe, Inc., architect-engineering firm that had conducted a survey of the operating and maintenance costs of gas turbine plants in operation and data presented in Refs. 52, 72, and 111. The results of this survey indicated that selected actual maintenance costs vary from a minimum of 0.5 mills/kwhr for base-load type operations to a level of about 1.5 mills/kwhr for some peaking plants. It is anticipated that with careful design and attention to maintenance-saving features in the engine, the maintenance costs would equal the 0.5 mills/kwhr level.

COMPARISON OF POWER GENERATION COSTS

Although the advanced open-cycle gas turbine power systems would provide a means of eliminating thermal pollution, there are alternative cooling systems available for use in steam plants which can greatly reduce or eliminate thermal pollution as well. Thus the ultimate acceptance of advanced gas turbine generating systems will depend on the possibility of producing the lowest-cost power in competition with steam plants using the alternative cooling systems. In the following paragraphs the total busbar power generation costs (including all the owning and operating cost elements) are presented for the advanced open-cycle gas turbine systems and steam turbine systems equipped with cooling devices designed to substantially reduce thermal pollution. The primary comparisons and analyses are based on natural gas-fueled stations of 1000-Mw net electrical output located in the South Central region of the country. The South Central region was selected for the comparison since natural gas is abundant and will be available in this area for the next two decades and water for station cooling is often in short supply and will be more difficult to obtain. The comparisons are also generalized to other regions of the country. No particular site in the South Central region was selected but a typical site might be that chosen by the Houston Plant Lighting Power Company for the Cedar Bayou Generator Station (Refs. 112 and 113). This plant complex will ultimately contain 6 units totaling 5000 Mw and will utilize a 2600-acre cooling pond which discharges water at 95.7 F. Based on the performance and cost data presented in Section VI for alternative condenser cooling systems, the cooling ponds may be considered approximately equivalent to the use of cooling towers.

South Central 1970-Decade Stations

A tabulation and comparison of the total busbar power generation costs among 1000-Mw conventional steam turbine stations and advanced simple-cycle and regenerative-cycle gas turbine stations which are projected to be in commercial operation during the 1970 decade in the South Central region are presented in Table XXVII. The results are based upon gas costs of 23¢/million Btu, the gas level projected in Section VI for South Central gas in the 1970 decade, 15% capital charges, and 70% load factor. The Table XXVII results indicate approximately a 0.5 mill/kwhr lower busbar power cost for the simple-cycle gas turbine station in comparison to the steam turbine station. The power costs for the regenerative-cycle gas turbine station would be approximately 6% higher than the costs for simple-cycle gas turbine stations but still less than those for the steam station. The principal advantage of the gas turbine stations is the substantially lower station capital costs, which at the present capital charges of 15% would result in almost 1.25 mills/kwhr lower capital costs for the simple-cycle gas turbine station in comparison to the steam station.

South Central Early 1980-Decade Stations

The comparison of busbar power costs among the various systems projected to be available for the early 1980-decade is shown in Table XXVIII. The data indicate that the busbar power costs would be the lowest for the simple-cycle gas turbine, in comparison to the steam turbine and regenerative-cycle gas turbine. The cost difference would be more than 0.80 mills/kwhr in favor of the gas turbine station. Although the installed costs of the steam station were reduced by almost $12/kw between the 1970-decade and early 1980's designs, the installed costs of the gas turbine were reduced by approximately the same amount. In addition, the net plant efficiency of the gas turbine station increased by more than 5 percentage points as described in Section VII, while the efficiency of the steam plant increased by only 2 percentage points. Thus, even though gas costs are projected to increase to the 30¢/million Btu level, the low installed costs of the gas turbine station would offset the relatively high fuel and maintenance costs and still provide a minimum-power-cost system.

Sensitivity to Economic Factors

Although the values selected herein for capital charges, natural gas costs, interest charges, and escalation rate are based on careful projections, a number of outside influences such as the general economic condition could change these factors considerably. Therefore, the sensitivity of the results presented in Table XXVIII to variations in several important values were considered. The variation in busbar power costs with changes in the capital charges and cost of gas are indicated in Figs. 59a and 59b, respectively. The Fig. 59a results indicate that even with a reduction in capital charges to the 10% level (a level which was considered adequate only 4 or 5 years ago), the simple-cycle gas turbine station would have power costs about 0.400 mills/kwhr lower than either the steam or regenerative-cycle gas turbine station. An increase in the capital charges to 17% from 15% would, of course, only widen the cost difference between the steam station and simple-cycle gas turbine. The effect of a variation in natural gas costs between 20 and 50¢/million Btu on the total busbar power costs are shown in Fig. 59b and indicate only a minor reduction in the cost advantage of the simple-cycle gas turbine stations relative to the other types of power systems at the highest levels of gas costs*.

Only recently has the short-term interest rate climbed to the high 8 to 10% levels presently experienced by the electric utility and other industries. The effect of utilizing a 6% interest rate during construction on the power costs of a steam system are shown by the lower line of the band shown in Fig. 60a. The upper line shows the busbar power costs with 8% interest during construction as used in Table XXVIII. The dashed line in Fig. 60a depicts the increase in busbar power costs as the simple-cycle gas turbine station costs are increased above the $66.5/kw level shown in Table XXVIII. Only at about $93/kw or a level 40% above that estimated in the study would the total busbar costs of the simple-cycle gas turbine station begin to approach that of the steam plant. Figure 60b shows the effect of changes in the heat rate of the steam plant on the total busbar power costs. The results indicate that even at a heat rate of 7500 Btu/kwhr, the total busbar power costs of the steam plant would be about 0.400 mills/kwhr above those of the simple-cycle gas turbine plant presented in Table XXVIII, and about 0.300 mills/kwhr above that of a gas turbine which had a heat rate about 5% poorer than that level projected in Section VII.

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* Fuel costs of 50¢/million Btu would stimulate interest in using nuclear-fueled stations for base-load operation and restrict the use of fossil fuels only for the swing-load sector of the load demand (< 40% load factor).

South Central Late 1980-Decade Stations

The estimated busbar power generation costs for steam and gas turbine systems that could be commercially available by the late 1980's for installation in the South Central region are shown in Table XXIX. The cost of natural gas used in these stations is 40¢/million Btu, the highest level projected for natural gas in the South Central region in Section VI of this report. The Table XXIX results indicate that the simple-cycle gas turbine station costs would produce power at the lowest busbar costs; approximately 0.45 and 0.85 mill/kwhr lower than that of the regenerative-cycle gas turbine and steam turbine stations, respectively.

Other Regions

Although the advanced open-cycle gas turbines are projected to provide lower busbar power generation costs than steam systems in the South Central region where relatively low-cost gas is available, in the other regions of the US the cost of fuels available for use in steam and gas turbine utility systems can be considerably different. For example, on the Pacific Coast in the West Power Region of the US and especially in Southern California, both residual oil at 28¢/million Btu and natural gas at 34¢/million Btu are projected to be available for use in the early 1980's (see Section VI). In some of the Rocky Mountain states both coal and gas would be available, although the cost of gas would be about 20¢/million Btu above that for coal. In the Northeast region coal, residual oil, and gas either in the form of imported LNG or produced as synthetic from coal or via pipeline from the Southwest are projected to be available for utility use. The cost of the gas would be in the 50 to 60¢/million Btu range, while low-sulfur oil might cost 20¢/million Btu less than the gas. Using the fuel cost projections presented in Section VI, and the station installed cost and performance characteristics determined in Section VII and VIII, estimates were made of the total busbar power costs for the competing systems located in different regions of the US. The results are presented in Figs. 61a and 61b for the Northeast and West regions of the country, respectively, for steam and simple-cycle gas turbine systems that would be commercially available in the early 1980's. The Fig. 61a data indicate that in the Northeast region the simple-cycle gas turbine station would generate power at slightly higher costs than the steam systems for load factors greater than about 60%. However, at load factors below 60%, substantially lower power costs would be realized with the simple-cycle gas turbine stations in comparison to the steam stations, in spite of the 20¢/million Btu higher fuel costs for the gas turbine station. For example, at a 40% load factor the cost of producing power could be 8.3 mills/kwhr for the simple-cycle gas turbine and more than 9.5 mills/kwhr for the steam station using residual oil.

Approximately the same results are illustrated in Fig. 61b for competing stations located in the West region. The steam stations would produce power at lower costs than the gas turbine stations only for load factors of 70% and above, in spite of the substantially lower fuel costs that are projected for coal and oil relative to natural gas.

Similar results were obtained for comparisons of power generating costs in the Southeast (not shown), West Central, and East Central regions of the country (see Fig. 62). These areas are heavily dependent on coal as a fuel source and although limited supplies of natural gas are available, the higher prices for it would make gas turbines attractive only for those utility applications where the load factor is projected to be less than about 60%. However, this mid-range or swing-load market will assume further importance as nuclear steam power systems are more widely used for the high-load-factor, base-load applications. But it must be cautioned that not all areas of each region will have access to a sufficient supply of natural gas for power generation applications. Furthermore, the air pollution regulations must be pursued vigorously to prevent reverting to the use of high-sulfur-content, low price coal and oil fuels. If low-cost, high-sulfur fuels were used, their prices would be substantially lower than those for natural gas, and it is likely the gas turbine would not be competitive.

Use of Dry Cooling Towers

Although neither the mechanical- or natural-draft type of dry cooling tower is anticipated to be widely used during the next two decades in any of the US power regions, except possibly the arid portions of the Western states, the effects of their use on the economic comparisons presented herein can be estimated readily from the data presented in Section VI. For example, the installed capital costs of a fossil-fueled station equipped with mechanical-draft dry cooling towers would be $8/kw to $12/kw higher than that for a similar station utilizing natural-draft wet cooling towers (see Table XIII and Ref. 60). For capital charges of 15% and a load factor of 70%, this would add almost 0.20 to 0.30 mills/kwhr to the steam station busbar costs presented in this section. The added operating costs for a steam station when mechanical-draft dry towers are used rather than wet towers are presented in Table XIV of Section VI. Depending upon the region considered, the added operating costs could be from 0.10 to as much as 0.34 mills/kwhr. Thus, the added costs for a dry cooling tower would be from approximately 0.30 to 0.65 mills/kwhr. Even if the lower cost figure were added to the steam station busbar costs shown in Figs. 61 and 62, the competitive position for the gas turbine would be enhanced. The only advantage possible for the steam station would be the utilization of low-cost coal or lignite directly in the boiler, whereas this would be impossible with the gas turbine. However, if a suitable gas turbine fuel were available, the use of dry cooling towers in a steam system would permit the utilization of higher-cost fuel in the gas turbine. Each 0.1 mills/kwhr in added busbar cost for the steam plant would allow the use of fuel costing about 1¢/million Btu more in the gas turbine.

POTENTIAL SITING, TRANSMISSION, AND RESERVE MARGIN ADVANTAGES OF GAS TURBINES

The successful operation of gas turbines without the use of cooling water can not only provide a possible solution to the thermal pollution problem but also produce other advantages in a utility system. For example, since a gas turbine can operate essentially independent of a supply of cooling water, considerably more flexibility is possible in the selection of station sites than is permissible with conventional steam power stations. With this extra flexibility the utility planning engineer would have the freedom to locate the gas turbine plant to take advantage of inexpensive land costs, or alternatively to be closer to a load center, to a system transmission network, or to a fuel supply. Furthermore, the compactness of the gas turbine system together with the flexibility of site selection which it allows could provide the basis for a highly reliable integrated network comprised of moderate-sized units. Since only 50% of the cost of providing power to the consumer is attributable to the cost of power generation and the remaining 10% and 40%, respectively, are attributable to transmission and distribution costs, these costs must also be considered in an analysis of competing power systems.

A brief study was conducted to determine some of the potential advantages that could accrue to an electric utility through the use of open-cycle gas turbines for base-load operation as a result of the smaller unit capacities of gas turbine engines and the wider selection and availability of plant sites made possible through the elimination of power plant cooling water. To assess the full advantages, the investigation included: (1) the effect of power generating unit size, system unit mix, i.e., the variations of unit size within a utility system, and unit forced outage rate on system reliability and reserve requirements; (2) the effect on system reliability and cost of expanding system capacity in small unit sizes; and (3) the effect of unit size and location on station-transmission network tie requirements. In the following discussion cost credits are presented which can be attributed to a base-load gas turbine generating plant that has freedom of location, and whose units are smaller in generating capacity than a conventional steam-powered generating station.

General Transmission and Distribution Considerations

The transmission and distribution (T&D) systems transfer power efficiently and reliably from the generating plant to the consumer. The transmission system performs several functions including (1) the transfer of large blocks of power at high voltage levels (115 kv to 765 kv) from generating plants to areas of high load density; (2) the intraconnection of generating plants and substations; and (3) the interconnection of neighboring utility systems. Alternatively the distribution system distributes the electric power, at a lower voltage level (69 kv or less) to the consumer (Ref. 114).

The system consists largely of circuits and substations. The circuits may be located either overhead or underground, and comprise extra-high-voltage (EHV) transmission-, regular transmission-, subterranean transmission-, primary distribution-, and secondary distribution-lines (Ref. 115). The substations contain the transformers used to step down the voltage, buses for interconnecting transformers and circuits, and circuit breakers for the automatic disconnect of faulty transformers or circuits. In addition to these equipment there are also ancillary equipment including series and shunt capacitors, shunt reactors, current and voltage transformers, relays, air-break switches, communication equipment, panel boards with meters, etc.

Transmission and distribution system failures can be caused by natural hazards, by man-made hazards, by over-voltages generated within the system as a result of switching operations, or by maladjustment or failure of system components. Circuits and equipment are designed individually to provide good reliability. Adequate system reliability and maintainability are achieved largely by the provision of redundant circuits, circuit breakers, buses, and transformers.

Probability methods similar to those outlined in Refs. 116 through 121 are commonly used by the utilities in an effort to determine the degree of reliability and thus the reserve requirements* needed for a system. A loss-of-load value of 0.2 days/yr was used in this study based on Refs. 117 and 122. This value represents the probable number of days per year that the demand exceeds the available generating capacity. For example, the probability of load exceeding capacity on 0.2 days/yr may also be stated as: one day in 5 years or alternatively as a loss-of-load probability of 0.055%.

Present T&D systems have, generally speaking, not been designed in the normal sense of the term, but are the outgrowth of an extensive series of planned additions. As a transmission system grows to meet increasing and expanding loads with the addition of substations and generating plants, the paths for the new transmission circuits (rights-of-ways) and the points at which they terminate are selected so as to provide a total integrated generation-transmission system which is flexible, stable, reliable, and economical. This system which evolves is a complex interlaced network of circuits commonly referred to as a transmission grid. Most of the transmission systems in and around large urban areas have reached the grid stage in their development. However, in many urban areas the space needed for trans-

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* A proper appreciation of the reserve requirements for an electric power system is quite important and discussed in detail in Ref. 123. Two different "rules of thumb" are usually stated as representing reasonable approximation of system installed reserve; they are: (1) the installed capacity should be at least 15% greater than the annual peak load and (2) the reserve, i.e., the excess capacity above that required to meet the annual peak load, should be at least as great as the sum of the two largest units in the system (Ref. 122).

mission lines is scarce and is becoming a significant factor in the selection of station sites. In Ref. 29 and in discussions with utility planners, the need for providing sufficient rights-of-ways for transmission lines has been continually stressed.

Effect of Unit Output Capacity on System Reliability

The loss-of-load probability method was employed to establish the effect of unit output capacity on reliability. The first system considered was assumed to be homogeneous, i.e., consisting of only units that are all of the same level of output capacity and all having the same forced outage rate. The unit output capacities for the homogeneous systems were selected as follows: 20%, 10%, 5%, and 2% of total installed capacity. Therefore, a typical 1000-Mw system could consist of either five units each capable of providing 20% of the system capacity, or ten units of 10% capacity each, etc. The forced outage for all units was assumed equal to 2% of the required operating time (Refs. 116, 117, 120 through 122). Conversely, the unit would be available 98% of the required time. The results of the probability analysis that was performed is presented in Fig. 63. This figure clearly shows the justification for the trend toward unit sizes of about 10% or less than the total system rather than toward the 20% units. A system comprised of units each capable of providing 10% of the total system capacity could meet the loss-of-load criteria and still have an annual peak load to installed capacity ratio of about 0.82. This result is not totally unexpected in that given the same forced outage rate it is quite reasonable that the disabling of one large unit, say a 20% unit, would more likely occur than the disabling of ten 2% units. The same arguments would apply whether the units were steam or gas turbines. However, in actual practice the cost savings inherent with economies of scale (see Fig. 10) and the reliability provided by intertie connections is the major reason for not adopting the use of many small units. However, if it is assumed that the competing systems are initially cost competitive, that is the small units can produce power for the same cost as the large units, the advantages described herein can be considered over and above the costs. The competing systems may be both of the gas turbine or steam turbine type or combinations of the two.

Effect of Degree of Mix and Forced Outage Rate on System Reliability

Having established, from a reliability standpoint, the desirability of having units consisting of 10% of the system capacity, the effects of forced outage rate and unit mix on loss of load were explored for the desired reliability of 0.2 days/yr. The forced outage rate equal to 0.02 (fraction of "required" operating time that system is experiencing a forced outage) was held constant for these 10% units. An additional system was then synthesized having the same total installed capacity. However, the makeup of this second system is slightly different and consists of a mixture of eight 10% units and ten 2% units. This system is called the mixed system since units are included which represent different percentages of total system capacity. The 2% units, which could represent a reasonable ratio of gas turbine power to conventional steam power in a utility system, were assumed to vary in forced outage rate from 0.02 to 0.10 (Ref. 123).

It is important to note that in both systems all the installed capacity, except that capacity which is down for either maintenance or as a result of a forced outage, is available to supply the system load at the time of peak annual load.

The results of the comparison between homogeneous systems consisting of units having capacities of 10% of the system capacity and the mixed system is shown in Fig. 64a. The homogeneous system (i.e., having ten units each comprising 10% of the total system output) with a constant forced outage rate of 0.02 meets the loss-of-load requirements of 0.2 days/yr if the annual peak load is approximately 81.7% of the system installed capacity. Thus the system must have reserves equal to 18.3% of the installed capacity. For example, if the peak load of a utility system were 817 Mw, the installed capacity would have to equal 1000 Mw to provide reserves equal to 183 Mw or 18.3% of installed system capacity. The mixed system with a forced outage rate of 0.02 for the 2% units achieves the same loss of load at a higher annual peak load percentage (82.7%). Therefore, this system would require a reserve margin of only 17.3%. Therefore, the mixed system could meet the peak load requirements for 817 Mw (as presented in the example described above) with only about 990 Mw of installed capacity or 10 Mw less capacity than the homogeneous system. The advantage for the mixed system decreases as the forced outage rate of the 2% units increases and vanishes entirely when the forced outage rate reaches about 0.06. If the forced outage rate of the 2% units exceed 0.06, the mixed system is less reliable than the homogeneous system. At a forced outage rate of 0.10, the annual peak load of the mixed system can be only 80.5% of installed capacity or conversely a reserve of 19.5% is needed to maintain the same loss-of-load probability. Figure 64a thus illustrates one very important fact: that a mixed system can have a greater reliability than a homogeneous system even though the reliability of some of the units in the mixed system are poorer than those in the homogeneous system.

Mixed-System Cost Credits

Although the mixed and homogeneous systems can meet the same reliability criterion, the installed capacities of each system could be different depending on the forced outage rates of the individual units. Figure 64b presents the cost credits (or deficits) that can be assigned to the mixed system as a result of the differences in capacity for such a system relative to a homogeneous system. The results are presented for forced outage rates varying from 0.02 to 0.10 for the 2% units in the mixed system. For example, if the 2% units in the mixed system had a forced outage rate of 0.02, approximately 10 Mw less capacity could be installed relative to the homogeneous system without a loss in overall system reliability.

If the fixed charges are assumed to be 15% and the average system thermal efficiency is equal to 36%, the reduced capacity can be converted into a fuel cost credit as shown in Fig. 64b. Figure 64b illustrates that there exists a positive allowable fuel cost increment for the mixed system if the forced outage rates of the unit are less than 0.06. The magnitude of this positive increment is less than 1¢/million Btu, with values of approximately 0.10 to 0.30¢/million Btu at a load factor of 0.8.

Installed Capacity Expansions

To determine what effect the utilization of the homogeneous system or mixed system would have on future expansion, a situation was synthesized where the systems would have to expand their original installed capacity by 10%. Thus the expansion could be obtained by either of two modes for the homogeneous system. The system could be expanded by adding another unit of 10% of the original installed capacity and having a forced outage rate of 0.02, or it could be expanded by adding five units, each unit having 2% of the original installed capacity and having a slightly higher forced outage rate of 0.03. In the case of the mixed system, the system was expanded by adding five more 2% units. The results of the system expansion are shown in Fig. 65a. Again, the reliability is increased by use of a mixed-unit concept. In fact, a comparison of Fig. 65a with Fig. 64a shows that, for a forced outage rate of 0.03, the mixed system enjoyed a 0.8% reliability advantage over the homogeneous system before expansion; while after expansion the advantage has increased to about 1.0% over that of the homogeneous system. It is also seen that the system that was initially homogeneous and utilized a smaller-unit concept in expansion now enjoys a 0.5% advantage over the system which adhered to the homogeneous unit concept. A cost analysis was performed as described previously and the results are displayed in Fig. 65b. The results indicate that the system which employs a unit mix with large and small units proves less expensive to own and operate than a system of large units with no mix but of comparable forced outage rate.

Effect of Expansion to Meet Future Demands

There is another manner by which system expansion can be evaluated. This method is discussed in Ref. 124 and is utilized here to determine the effect on electric utility system cost of frequent expansion in small, low-capital-cost/kilowatt units such as a gas turbine, as opposed to less frequent expansion in large high-capital-cost units typical of conventional steam systems. Expansion of the utility system in smaller units reduces the amount of capacity in excess of system demand carried at any one time. Thus, the excess cost incurred by producing additional capacity beyond current system requirements is eliminated. It is clearly a more advantageous situation for an electric utility to expand its capacity by frequently adding units which match load demand rather than being forced to add large units simply to gain economies of scale as would be the situation if expansion is by means of conventional steam power.

The savings to the utility system which accrue as the result of expanding in units which much more closely match the demand curve is presented in Fig. 66a. The results were based on the use of small capacity units costing only $70/kw (i.e., gas turbines) and indicate savings of about 0.25 mills/kwhr. The potential mills/kwhr savings shown in Fig. 66a which could result when expanding with small units can be converted into a fuel cost savings by assuming a load factor, fixed charges, and thermal efficiency. These fuel cost savings are presented in Fig. 66b. For a system load factor of 0.8, Fig. 66b shows that the savings amount to an allowable fuel cost increment on the order of 2.2¢/million Btu for a system expanded in small (2% of initial system capacity) units, rather than in large (10% of initial system capacity) units.

Effect of Unit Size and Location on Transmission and Distribution System Costs

The last potential advantage considered involved the determination of the effect of unit size and location on the system-transmission network requirements. To determine this effect, two systems were synthesized; they are presented in Fig. 67. The first system, Case I, consists of a single-unit central generating plant in close proximity to the system grid lines. This single-unit generating plant contains 10% of the system installed capacity. However, this central station is required to transmit its power output over some distance to the load center. To transmit this electrical power over the distance, a step-up transformer is required at the central station and a step-down transformer at the load center. The transmission network tie with the station-load system is at the central generating station on the high voltage side of the transformer. The tie line costs were assumed negligible because of the proximity of the central generating plant and its transmission lines to the transmission network.

The second system, Case II, involves a multiple-unit central generating plant, such as that which would occur if gas turbines were utilized. This multiple-unit central operating plant also contains 10% of the system installed capacity. Again, since the generating plant utilizes gas turbine units, and since they are independent of cooling water, it could be located near the load center. However, this would probably mean that the transmission network tie would have to travel over some distance into the station. For this comparison, the distance over which the energy had to be transmitted in from the transmission network (Case II) was assumed identical to the transmission distance from the central power station to the load center (Case I). The only transformer that need be required in the Case II system is a step-down transformer converting the transmission system grid voltage to the distribution voltage. Finally, the costs of the equipment required for the tie at the transmission network was assumed to be identical for both Cases I and II, as a result they would cancel each other out at the transmission network in an incremental cost approach.

After having synthesized the two cases, it became necessary to determine the required number of circuits from the transmission network to the station-load center system. Two circuits were indicated in Case I; however, since the transmission distance was assumed negligible, circuitry cost considerations did not really have to be analyzed for this case. Reliability considerations were used for Case II to determine the number of circuits required. The results of these considerations are illustrated in Fig. 68. This figure presents a plot of loss of load as a function of the percentage of annual peak load divided by system installed capacity for various unit forced outage rates and single or double circuits. Each circuit carries electric power equal to the output of a single unit. Again, if it is desirable to reduce loss of load to no more than 0.2 days/yr, it would appear that a forced outage rate of at least 0.04 is essential in the units if single circuitry is desired. It should be noted that two electrical circuits will decrease the loss of load by at least two orders of magnitude. It is obvious that either single- or double-circuit transmission lines were acceptable in Case II, depending on the reliability of the units in the central power station. As a result for the remainder of the analysis both options were considered.

The results of considering both options for a transmission distance of 25 mi is shown in Fig. 69a. This figure presents a comparison of system allowable fuel cost increment as a function of load factor for both circuitry options and right-of-way cost. It is clearly shown that there is a greater savings associated with single circuitry as compared to double circuitry. This result was expected along with the decreased saving associated with increased right-of-way cost. The range of land costs gives an indication of the costs that could be expected in locating the generation station near the load center. Obviously, the more densely industrialized or populated the load center is, the greater the premium placed on the land in the area of the load center. Of the three values presented, the figure of $3000/acre most closely approximates the load values within the Northeast Utilities system (Ref. 125). The results presented in Fig. 69b indicate the system allowable fuel cost increment for the same parameters as in Fig. 69a but for the situation where transmission line distances are 50 mi. There is a greater savings in the 50-mi situation as opposed to the 25-mi situation because of the increased savings due to lower transmission line costs. This reduction in transmission line costs results from the decreased transmission line load, i.e., power equal to 1 of the 5 units in Case II vs power equal to the single unit in Case I. As a result, since there is a savings per mile, it follows that as the number of miles increases the savings will also increase.

Concluding Remarks

For a given utility system capacity level both a reduction in power generating unit output capacity and a diversification of unit size can increase reliability. Along with this increase in reliability there also results, at load factors of

70 to 80%, a utility-wide allowable fuel cost savings of 0.3¢/million Btu. There is an appreciable system cost credit for expanding an electric utility system more frequently and in smaller power generating unit output capacity that more closely approximate the demand curve as opposed to expanding the system over longer time periods and in larger unit output capacity. However, the costs involved in borrowing money have not been included in this comparison. The system allowable fuel cost saving which can be credited can be 2.2¢/million Btu for reasonable load factors. The reduced station-transmission network tie requirements associated with relatively small gas turbine power generating units which can be located close to the load center results in an appreciable saving for these units as compared to the larger steam units located at some distance from the load center in order to have access to cooling water. This savings can result in a system allowable fuel cost increment of as great as 1.2¢/million Btu.

GAS TURBINE FUELS

In a previous section of this report it was noted that natural gas, which is an ideal fuel for use in gas turbines, will be in short supply within a few years unless the FPC adopts a more realistic policy for establishing well-head prices for natural gas. However, even if the supply of natural gas continues to dwindle, thus precluding its future use for electric power generation, it appears that alternative fuels will become available and that these fuels will be suitable for use in gas turbine power systems. The most promising alternative fuel appears to be either a high- or low-Btu synthesis gas derived from coal. In the United States, three incentives exist for development of synthetic coal gasification processes: (1) the United States gas industry, desires to obtain a supplementary source of gas to insure its gas supply in the face of worsening reserve-to-production ratio for natural gas in the United States; (2) the federal government wants to stimulate the utilization of this country's vast coal resources to meet the country's energy needs; and (3) the federal government desires to stimulate the use of nonpolluting, low-sulfur fuels in place of the high-sulfur coal and residual fuel oil in common use today.

Gas Turbine Fuel Specifications

A variety of gaseous, liquid, and solid fuels have been used in gas turbine engines. However, with the exception of a few small closed-cycle gas turbine plants which burn coal or use nuclear power, all present-day aircraft-type and heavy-duty industrial gas turbines use liquid or gaseous fuels. Gaseous fuels such as natural gas, butane, and propane are ideal fuels since they contain no harmful alkali or sulfur compounds. Furthermore, they burn readily in small-volume combustors with no smoke or carbon residue and at low flame luminosity.

Fuels with high flame luminosity properties will exhibit high radiation heat transfer characteristics. Consequently, engines burning these fuels would have to protect against higher liner temperatures than engines burning gaseous fuels. Low flame luminosity has also been found to be related to desirable levels of smoke emissions and carbon residue.

Liquid fuels, used by 70% of the gas turbine units in operation today, are available in a variety of petroleum distillates, residual oils, and blends of the two. Liquid fuels for industrial gas turbines traditionally have been petroleum distillates such as American Society for Testing Materials (ASTM) Grade No. 1 or 2 diesel fuel oil. While heavy residual fuels such as ASTM Grade No. 5 or 6 fuel oil have been used in heavy-duty gas turbines which operate at relatively low turbine inlet temperatures, these heavy fuels must be selected with care to minimize maintenance costs, especially in the hot section of the engine. The major problems associated with the utilization of these heavy residual fuels in gas turbines are erosion, ash deposition, vanadium corrosion, and sulfidation of the turbine blades and vanes resulting from the high ash, metal, and sulfur contents of these fuels.

Gas turbines operating with heavy liquid fuels are generally restricted to operating temperatures which are several hundred degrees below those employed when burning clean gaseous or distillate fuels. Ash deposition due to the formation of liquid vanadium and alkali metal compounds during combustion of residual fuel oils can result in severe loss of output power, especially under continuous operating conditions. At turbine inlet temperatures below approximately 1200 F, ash deposits are generally loose and powdery or can be made so by selected fuel additives. These deposits can be readily washed off or spalled off during frequent shutdown intervals. Harder bonded deposits and subsequent vanadium corrosion tend to occur at temperatures above 1200 F with stainless steels, cobalt-base alloys, and to a lesser extent with nickel-base alloys. Satisfactory operation can sometimes be achieved by utilizing magnesium-base additives and water washing of the fuel to reduce the sodium and potassium concentrations. However, sulfidation is a formidable corrosion problem at turbine inlet temperatures of 1500 F and above with fuels containing substantial quantities of sodium and sulfur.

As a result of these operational problems, residual fuels are generally unacceptable for use in advanced, high-temperature gas turbines. The problems associated with utilizing coal as a gas turbine fuel would be far more severe than those encountered with residual oil because of the higher ash and sulfur contents of most coal. Consequently, in order to utilize coal as a gas turbine fuel it must be gasified and purified to a cleanliness comparable to that of natural gas or distillate fuel. No industry-wide specification currently exists for gaseous fuels, because most natural gas made available through gas distribution networks exceeds the cleanliness requirements specified by gas turbine manufacturers. Recently, the ASTM provided tentative specifications (see Table XXX) for four grades of liquid gas turbine fuels. These specifications provide a starting point for discussions between gas turbine manufacturers and fuel suppliers. In addition to normal fuel properties, these specifications provide for specific limits on metals such as vanadium, sodium, potassium, calcium, and lead that could cause corrosion or ash deposition during turbine operation. Pratt & Whitney Aircraft Division of UA in its heavy distillate fuel specifications, further restricts the vanadium content to 0.2 ppm by weight, the sodium plus potassium content to 0.6 ppm by weight, and the sulfur content to 1.3% by weight.

Coal Gasification Technology

The basic coal gasification process consists of reacting coal with steam at elevated temperature to produce a synthesis gas consisting of carbon monoxide, hydrogen, and a variety of impurities. This basic gasification reaction is highly endothermic so that considerable heat must be supplied to sustain the reaction. This heat could be supplied autothermally, wherein a portion of the coal would be combusted with air or oxygen, or by supplying external heat to the process. Simplified schematic diagrams of coal gasification processes utilizing these two methods of heat addition are depicted in Figs. 70a and 70b, respectively.

In addition to providing a source of heat for the reaction, all gasification processes require equipment to scrub coal dust and condensible hydrocarbon tar from the synthesis gas and gas purification equipment to remove undesirable sulfur compounds and, in some cases, carbon dioxide. This equipment is denoted by the solid squares in Fig. 70. If a low-Btu fuel gas were to be the end product, air could be used for partial combustion of the coal in autothermal processes and no further processing would be required after the gas purification step. However, if a high-Btu (900 Btu/ft^3 or higher) pipeline quality gas were the desired end product, then autothermal processes would require the use of pure oxygen for partial combustion to prevent nitrogen in the air from getting into the synthesis gas. In addition, the hydrogen-to-carbon monoxide ratio in the gas must be adjusted to three parts by volume hydrogen to one part by volume carbon monoxide using the water-gas shift conversion so that these gases could undergo subsequent catalytic methanation to produce the desired methane-rich, high-Btu pipeline gas. This optional equipment required for pipeline quality gas applications is designated by the dashed boxes in Fig. 70.

Several alternative oxygen separation, shift-conversion, and purification processes are commercially available so that development of coal gasification processes has concentrated on the basic gasification step and, in the case of pipeline quality gas, the methanation step. A concise description and evaluation of four methanation reactor designs is presented in Ref. 126 and is not repeated herein. The following discussion concerns alternative methods of gasifying coal, including two with in-situ sulfur removal.

Autothermal Gasifiers

Countercurrent, cocurrent, and fluidized-bed autothermal gasifiers have been used commercially (Ref. 127). Countercurrent gasifiers with downflow of coal and upflow of gases have the advantage of high thermal efficiency and high turndown ratio. Their major disadvantages are low gasification rates, relatively small capacities, and the formation of tar which makes gas purification and waste heat recovery difficult. Cocurrent gasifiers, with up or down flow of both coal and gases, have relatively low thermal efficiencies unless expensive waste heat recovery equipment is employed. Gasification rates are higher than those of countercurrent gasifiers because higher temperatures can be employed. A major advantage is that no tar is formed, making gas cleanup and waste heat recovery easy. Fluidized-bed gasifiers have intermediate characteristics. They can be scaled up to relatively large sizes. However, turndown ratios for these units are small due to the necessity of maintaining a minimum fluidizing velocity through the bed.

A summary and comparison of some of the more promising commercial coal gasification processes are given in Table XXX. Of these processes, only the Lurgi dry-ash process is carried out at elevated pressure. Synthesis gas must be at elevated pressure for pipeline transmission and for use in gas turbine power systems. In addition, elevated pressure would be desired in order to reduce the physical size and cost of gas purification equipment. Also, specific gasification rates, i.e., gas produced per cubic foot of reactor volume, are favored by increased pressure. For these reasons, the Lurgi dry-ash gasifier is the only commercial gasifier which appears to be suitable (see Ref. 128). The specific gasification rate for this gasifier compares favorably with other commercial gasifiers, and is amenable to scale up in size such that a single gasifier could provide fuel gas to 80-Mw or larger power stations. For very large stations (1000 Mw), the requirement for a relatively large number of gasifiers could prove to be a disadvantage for the Lurgi gasifier. This limitation in gasification rate results because the Lurgi gasifier is not designed for slagging operation requiring reaction temperature to be kept below the ash fusion temperature of coal (approximately 1700 F).

Advanced autothermal gasifiers could achieve higher gasification rates by operating at higher temperatures, in excess of 2200 F. Under these conditions, the coal ash would melt and become slag. Thus, fixed-bed gasifiers (such as the Lurgi type) could no longer be used and entrained or cocurrent gasifiers would need to be employed. Various high-temperature, cocurrent flow gasifiers have been surveyed (see Table XXX). The Texaco (Ref. 129), US Bureau of Mines (Ref. 130), and the Bituminous Coal Research (Ref. 131) gasifiers are representative of advanced gasifiers. These advanced gasifier configurations vary, but the basic chemistry, gasification rates, and efficiencies appear to be comparable. In all cases, from two- to three-second residence times are required for 90 to 100% carbon conversion, and gasification temperatures range from 2200 F to 2500 F. Oxygen or air requirements must be sufficient to supply heat for preheating feeds, endothermic reactions, and making up heat losses. Steam requirements must be such that the reactant temperature is kept within the 2200 to 2500 F range by the endothermic steam-carbon reaction. Off-gases from these advanced reactors generally reach equilibrium with respect to the water-gas shift conversion, thus obviating the need for separate shift conversion facilities. Unlike the Lurgi dry-ash gasifier, where the synthesis gas exits the gasifier at approximately 950 F and contains less than 8% of the heating value of the coal as sensible heat, synthesis gas would exit from advanced cocurrent gasifiers at 2200 F to 2500 F and contain 15 to 20% of the heating value of the coal as sensible heat. Recovery of this heat by preheating feeds to the process or by generating electricity will be necessary in order to realize satisfactory coal-to-gas energy conversion efficiency.

External Heating Processes

Externally heated gasifiers differ from autothermal gasifiers (see Fig. 70) in that coal need not be oxidized within the gasifier vessel to provide the heat necessary to sustain the endothermic coal-steam reaction. The advantages of external heating are two-fold: first, no oxygen separation equipment would be needed to keep nitrogen out of the synthesis gas, and second, carbon dioxide would not be formed to dilute the synthesis gas. These characteristics make externally heated processes more adaptable to providing high-Btu pipeline gas than low-Btu gas.

Three processes utilizing external heating that have been studied extensively are also listed in Table XXXI. In the HYGAS process, which is being developed by the Institute of Gas Technology (Ref. 132), gasification would occur in two sections. In the first or hydrogasification section, coal would be gasified in a series of contacting stages by a mixture of steam and synthesis gas. The synthesis gas, consisting primarily of hydrogen and carbon monoxide, would be generated in an electrothermal gasification section. In this section, char residue from the hydrogasifier would be reacted with steam to produce the synthesis gas. Heat for the electrothermal gasifier would be provided by electrical resistance heaters. The char residue from the electrothermal gasifier would be used to generate the required electrical power in a combined magnetohydrodynamic-steam power system. A HYGAS pilot plant capable of processing 80 tons/day of coal to produce 1.5 million cu ft of synthetic high-Btu gas was dedicated in Chicago and is expected to begin operation in early 1971. According to an estimate recently made by Stearns-Roger Corporation (Ref. 133), a commercial plant based on the HYGAS process could be designed and constructed by about mid-1974 if a crash program were inaugurated. However, such a program would involve considerable risk by the operator. A more conservative approach would allow another 2 to 3 years for additional pilot plant testing.

The other two processes identified in Table XXXI in which external heating would be used are interesting because a single recirculated material would serve the dual purpose of conveying heat to sustain the endothermic chemical reactions while simultaneously providing in-situ removal of sulfur compounds. In the carbon dioxide acceptor process (Ref. 134) being developed by Consolidation Coal Company, hot calcined limestone or dolomite would be used, whereas in the molten salt process investigated by M. W. Kellogg Company (Ref. 126), hot sodium carbonate would be used. The off-gases from these gasifiers would have to be scrubbed, undergo shift conversion, and be methanated as depicted in Fig. 70. According to Ref. 133, the carbon dioxide acceptor process could be commercially available approximately one year after the HYGAS process.

Coal Gasification Costs

Numerous estimates for the cost of pipeline quality synthetic coal gas (900 Btu/ft^3 or higher) are available in the literature. These estimates range from 30 to 70¢/million Btu, exclusive of transmission cost, depending on the degree of optimism built into the estimates, the cost of coal, interest rates, rate of return on investment, and the time period during which the estimates were prepared. Most estimates prepared during the late 1960's based on then-current coal costs, interest rates, and construction costs and using utility accounting procedures indicate that pipeline gas could be produced for 40 to 50¢/million Btu, exclusive of transmission cost (see Refs. 130, 132, and 134). Other estimates (Ref. 135) indicate that synthetic pipeline quality gas could be generated in West Virginia and piped to Philadelphia for just under 50¢/million Btu range. A recent analysis of the IGT HYGAS and the carbon dioxide acceptor process (which is also slated for large pilot plant evaluation) conducted by Stearns-Roger Corporation engineers (Ref. 133) indicates the price of pipeline quality gas to be about 40¢ higher than the price of coal used. Since processes designed to produce low-Btu gas (150 to 250 Btu/ft^3) would not require equipment for oxygen separation, shift conversion, or methanation, it has been estimated that this type of producer fuel gas could be manufactured for an incremental cost of about 20¢/million Btu higher than the price of coal used (Ref. 45).

DEVELOPMENT TIME AND COST FOR ADVANCED GAS TURBINES

The advanced gas turbine engines described in this report are somewhat larger in physical size and would have output capacities three to four times greater than those presently available or in the design phases. Thus the development time and cost requirements to bring forth these engines would be substantial and must be considered in any realistic appraisal of gas turbines as a future means of eliminating thermal pollution.

The costs associated with the development of new engines for aircraft applications are staggering. For example, the costs to develop the engines for the various jumbo jets, i.e., the Lockheed L-1011, the Boeing 747 and others are projected to reach a level of $200 million (Ref. 136). In Ref. 137, estimates of the cumulative development costs for turbojet engines are presented; and indicate that for engine thrust levels of approximately 100,000 lb (equivalent to a power output of approximately 150,000 hp) the costs would approach $500 million. These costs include the costs spent for continued product improvement of the engine with time. Product improvement is an important part of the engine development process and should not be misconstrued as simply improving reliability or increasing the number of applications. The thrust of the Pratt & Whitney J-57, for example, was increased from approximately 10,000 lb to 21,000 lb over 10 years through the product improvement procedure.

There are numerous reasons for the high development costs for these aircraft gas turbines. Each new engine development usually involves some advancement in the performance and weight characteristics of the engine above the levels available with existing engines. Furthermore, these advances must be achieved without sacrificing engine reliability or the flexibility to operate over a wide range of power settings. The requirements for aircraft engine development also tend to emphasize the attainment of higher and higher component efficiencies. Since each operating part of the engine is so highly loaded, and minor failures in a local area might overload other critical components, the designer often finds his analytical abilities inadequate and each new engine development requires repeated designing, building, and testing of individual components as well as complete engines to achieve the desired results. It is not uncommon for as many as two dozen sets of engine parts to be built and for some engine designs to undergo 10,000 hrs on the test stand before certification and many times that amount during subsequent model changes.

Development costs for advanced gas turbines for electrical utility and other ground-based applications would be substantially less than for aircraft engines of comparable size. Since the attainment of high-power-to-weight ratios is not necessary, this would provide an additional degree of freedom for the gas turbine designer. In addition, the technology of advanced materials and blade cooling would be available from aircraft engine programs. Although reliability would still be a major criterion, it is unlikely that more than three to five sets of engine parts would be required for testing before the design characteristics of a new engine were finalized. Thus, the extensive component and engine testing such as that required before an aircraft engine is certified could be reduced. Furthermore, much of the tooling and manufacturing facilities used for smaller output capacity engines could be utilized for the advanced, higher-capacity designs.

A brief preliminary analysis was made to determine the approximate development costs for the advanced engines designed for industrial applications. The results are shown in Fig. 71a (see line 1) and indicate that from about $100 to $150 million would be required for initial engine development over the first five years for engines ranging in size from 100 to 250 Mw. An additional $64 to $100 million would be needed for product improvement over the next 10 years (see line 2 in Fig. 71a). It is anticipated that the product improvement program could result in advanced and larger versions of the gas turbine engines. These estimates are for a company entering the gas turbine field which would require the construction of new manufacturing facilities and tooling capable of handling engines capable of producing 100 Mw and above. However, a company which had existing facilities and was making large capacity engines, i.e., 50 to 100 Mw, could develop the advanced engines at substantially lower costs since manufacturing facilities would be available. The dashed line (line 3) shows the estimated costs including the 5-year and product improvement program for a company with existing facilities. These levels would be substantially lower and would introduce a lower burden on the selling prices of the engines than for a new manufacturer.

The estimated selling prices of the gas turbine engines discussed in this report were based on a total engine development cost of $125 million, or essentially only the costs associated for the first five years of development, and an anticipated market penetration of 4000 Mw/yr. A review of Figs. 1 and 2 indicates that from 30,000 to 40,000 Mw of thermal generation equipment will be added in the electric utility industry during each of the next twenty years. Therefore, it appears reasonable that any one gas turbine manufacturer after allowances for steam station penetration would be capable of capturing 10% of this total market (say 4000 Mw/yr). However, the effects of variations on market penetration and engine development cost on the estimated engine selling price for a 250-Mw engine are shown in Fig. 71b. The results indicate only a modest increase in engine selling price of about $5/kw as market penetration falls to about half (2000 Mw/yr) of that assumed in the study.

ESTIMATE OF ADDITIONAL CAPITAL COSTS FOR COOLING TOWERS AND COOLING PONDS

It has been projected (Ref. 34) that within the next 50 years approximately $20 billion will have to be invested to provide the cooling water requirements for steam-electric plants and that a large part of the needed investment will be for cooling towers. This projection is substantiated by a recent survey (Ref. 36) which noted that twenty-six steam-electric power companies estimated that they will invest some $170 million of capital in plants now under construction to comply with the new or impending water temperature standards. Specifically, new units under construction for twenty investor-owned companies (representing 41,860 Mw) and six municipal, co-op, or public companies (representing 3240 Mw) will require $156,706,000 and $14,000,000, respectively, to achieve this compliance. It is noted in Ref. 36 that a number of companies contacted in this survey were unable to provide estimates because of uncertainties existing over the final temperature standards or permissible mixing zones in their respective localities. Fourteen of the companies surveyed claim to have already spent nearly $15 million over the past five years to bring their stations into compliance with new temperature standards and 6 companies alone plan to spend $49 million in the near future to bring existing stations into compliance.

Projections of future anticipated investments in cooling facilities in the United States (Ref. 40) indicate the cost for such investments may amount to from about $2 billion to over $4 billion in the 1970 decade; the actual amount will depend upon how stringent thermal quality standards become. It is projected that an additional $3.5 billion to $5 billion will be invested for this purpose in the 1980-decade. These costs are about an order of magnitude greater than the development costs of the advanced open-cycle gas turbines which would provide a solution to the thermal pollution of our river and lake waters. This highlights the need for more intensive development of reliable, high-output-capacity gas turbines operating at temperature levels of 2000 F and above.

SECTION IX

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SECTION X

REFERENCES

1. Federal Power Commission Regional Reports. Prepared by Regional Advisory Committees, used in Preparation of the 1970 National Power Survey, various 1969 dates.

2. 46th Semi-Annual Electric Power Survey. Edison Electric Publication No. 69-58, October 1969.

3. 47th Semi-Annual Electric Power Survey. Edison Electric Publication No. 70-26, April 1970.

4. Second Biennial Survey of Power Equipment Requirements of the US Electric Utility Industry 1969-78. Survey sponsored by Power Equipment Division, National Electric Manufacturers Association, New York, New York, February 1970.

5. Civilian Nuclear Power 1967, Supplement to the 1962 Report to the President. USAEC, February 1967.

6. McKennitt, D. B.: The US Electric Power Industry. Stanford Research Institute Report No. 321, May 1967.

7. Gambs, G.: The Electric Utility Industry: Future Fuel Requirements 1970-1990. Mechanical Engineering, April 1970.

8. Energy in the United States 1960-1985. Sartorius & Co., September 1967.

9. Outlook for Energy in the United States. Energy Division, The Chase Manhattan Bank, October 1968.

10. Ritchings, F. A.: Raw Energy Resources for Electric Energy Generation. Paper presented at the 1968 American Power Conference, April 1968, Chicago, Illinois.

11. Morrison, W. E.: Simulated Models of Future Energy Demand - Probability and Contingencies for 1980 and 2000 A.D. ASME Paper 68-PWR-4 presented at the IEEE-ASME Joint Power Conference in San Francisco, California, September 16-19, 1968.

12. A Review and Comparison of Selected United States Energy Forecasts. Pacific Northwest Laboratories of Battelle Memorial Institute, December 1969.

13. More Natural Gas is Sought for Use in Eastern States. New York Times, August 16, 1970, pp. 1 and 36.

14. Congressional Record - Extension of Remarks, pp. E6015-E6016, June 26, 1970.

15. Potential Supply of Natural Gas in the United States (as of December 31, 1968). Prepared by Potential Gas Committee, Colorado School of Mines Foundation, Inc., Golden, Colorado.

16. US Geological Survey Circular No. 522.

17. Cambel, A. B., et al.: Energy R&D and National Progress. US Government Printing Office, Washington, 1965.

18. Future Natural Gas Requirements of the United States. Volume No. 3, Denver Research Institute, University of Denver, Denver, Colorado, September 1969.

19. "Big West Coast Utility Eyes Faraway Gas." The Oil and Gas Journal, August 10, 1970, pp. 90-91.

20. Pipeline Pays 28¢ for Texas Gas to Ease Supply Bind. The Oil and Gas Journal, July 13, 1970, p. 37.

21. Gas Supply Would Rise with Price Hike. The Oil and Gas Journal, May 4, 1970, pp. 94-95.

22. Higher Gas Prices Coming, Question is How Much. The Oil and Gas Journal, June 15, 1970, pp. 33-36.

23. Virtually No Uncommitted Gulf Gas Left. The Oil and Gas Journal, May 18, 1970, pp. 42-44.

24. FPC Study Points Up Big Gas-Price Gap. The Oil and Gas Journal, September 14, 1970, pp. 62-63.

25. Air Pollution and the Regulated Natural Gas and Electric Utility Industries. FPC Report, September 1968.

26. Soaring Tanker Rates Felt in US. The Oil and Gas Journal, July 13, 1970, p. 36

27. Bennett, R. R.: Energy for the Future. Combustion, April 1970.

REFERENCES (Continued)

28. DeCarlo, J. A., E. T. Sheridan, and Z. E. Murphy: Sulfur Content and United States Coal. Bureau of Mines Information Circular 8312.

29. Davison, W. R.: Visit to Burns and Roe, Inc. to Discuss Electric Utility Industry Projects. United Aircraft Research Laboratories Report UAR-J175, July 7, 1970.

30. Sporn, P.: Developments in Nuclear Power Economics, January 1968-December 1969. Report prepared for the Joint Committee on Atomic Energy, Congress of the US.

31. Congressional Statement - Extension of Remarks by Hon. J. Randolph (W. Va.), pp. E7963-E7975, September 2, 1970.

32. To Keep the Lights Burning. Forbes Magazine, July 15, 1970, pp. 22-29.

33. Problems in Disposal of Waste Heat from Steam-Electric Plants. FPC Staff Report, 1969.

34. The Nation's Water Resources. The US Water Resources Council, Washington, D.C., November 1968.

35. Hauser, L. G.: Cooling Water Requirements for the Growing Thermal Generation Additions of the Electric Utility Industry, Paper Presented at the American Power Conference, Chicago, Illinois, April 22-24, 1969.

36. Olds, F. C.: Thermal Effects: A Report on Utility Action. Power Engineering, April 1970.

37. Clark, J. L.: Thermal Pollution and Aquatic Life. Scientific American, Vol. 220, No. 3, March 1969.

38. Jaske, R. T.: The Need for Advance Planning of Thermal Discharges. Nuclear News, September 1969.

39. Considerations Affecting Steam Powerplant Site Selection. A report sponsored by the Energy Policy Staff, Office of Science and Technology, December 1968.

40. Warren, F. H.: Electric Power and Thermal Output in the Next Two Decades. Stanford Research Institute Report No. 321, May 1967.

41. Presentation Before the New York Society of Security Analysts by Gulf States Utilities Company, September 1970.

REFERENCES (Continued)

42. Lokay, H. E., H. L. Smith, and G. D. Broome: Changing Patterns in Generation Planning Results. Paper presented at American Power Conference, Chicago, Illinois, April 23-25, 1968.

43. Lewis, G. P.: Feasibility Studies of Advanced Power Cycles, Progress Report No. 10, Burns and Roe, Inc., Oradell, New Jersey, May 6, 1970.

44. Palo, G. P., et al.: Units 500 Mw and Larger Found to Yield Savings. Electrical World, March 31, 1969, pp. 30-34.

45. Robson, F. L., et al.: Technological and Economic Feasibility of Advanced Power Cycles and Methods of Producing Nonpollution Fuels for Utility Power Stations. United Aircraft Research Laboratories Report J-970855-13, November 1970.

46. Woodson, H. H.: Short Term Prospects for Improving Efficiency of Power Plants. Paper presented at Thermal Considerations in the Production of Electric Power (a Joint Meeting of the Atomic Industrial Forum and Electric Power Council on Environment), Washington, D. C., June 28-30, 1970.

47. Giramonti, A. J.: Discussion of COGAS Systems with Riley Stoker Corporation. United Aircraft Research Laboratories Report UAR-H241, September 30, 1969.

48. Giramonti, A. J.: Discussion of COGAS Systems with Foster Wheeler Corporation. United Aircraft Research Laboratories Report UAR-H210, September 5, 1969.

49. Giramonti, A. J.: Discussions of Steam and COGAS Systems with Babcock and Wilcox Company. United Aircraft Research Laboratories Report UAR-H246, September 30, 1969.

50. Biancardi, F. R.: Feasibility Study of Nonthermal Pollution Power Generating Systems. United Aircraft Research Laboratories Report J-970978-4, July 10, 197

51. Nuclear Power for the Under-Developed? Electrical World, January 12, 1970, pp. 24-26.

52. Biancardi, F. R.: Memorandum of communication with Northeast Utilities, "Calculation of Power Generation Costs," May 10, 1970.

53. "TVA Contrasts Cooling Water Designs," Electrical World, December 22, 1969, pp. 23-26.

54. Neale, L. C.: The Use of River Models in Power Plant Heat Effect Studies. Paper presented at Meeting on Thermal Considerations in the Production of Electric Power, Washington, D. C., June 28-30, 1970 (A Joint Meeting of Atomic Industrial Forum and Electric Power Council on Environment).

55. Christianson, A. G., and B. A. Tichenor: Economic Aspects of Thermal Pollution Control in the Electric Power Industry, No. 67, September 1969. Federal Water Pollution Control Administration Northwest Region, Pacific Northwest Water Laboratory, Corvallis, Oregon.

56. Carey, J. H., J. T. Ganley, and J. S. Maulbetsch: Task I Report; Survey of Large-Scale Heat Rejection Equipment Prepared for Federal Water Pollution Control Administration. US Dept. of the Interior, Corvallis, Oregon. Under Contract No. 14-12-2177 by Dynatech R/D Company, July 21, 1969.

57. Kolflat, T.: Natural Bodies of Water for Cooling. Paper presented at Thermal Considerations in the Production of Electric Power, Washington, D. C., June 28-30, 1970 (Joint Meeting of the Atomic Industrial Forum and Electric Power Council on Environment).

58. A Cooling Pond Proves Cheaper. Electrical World, November 30, 1953, pp. 84-85.

59. Letter to F. Biancardi from Mr. G. Crossland of Ceramic Cooling Tower Company, Fort Worth, Texas, February 8, 1971.

60. Feasibility of Alternative Means of Cooling for Thermal Power Plants Near Lake Michigan. US Department of the Interior, Federal Water Quality Administration Report, August 1970.

61. Kadel, J. O.: Cooling Towers - A Technological Tool to Increase Plant Site Potentials. Paper presented at American Power Conference, Chicago, Illinois, April 23, 1970.

62. Rossie, J. P., and E. A. Cecil: Research on Dry-Type Cooling Towers for Thermal Electric Generation. Prepared for US Department of the Interior, Federal Water Quality Administration, under Contract No. 14-12-823, November 1970.

63. Woodson, R. D.: Cooling Towers for Large Steam-Electric Generating Units. Paper presented at Symposium on Thermal Considerations in the Production of Electric Power, Washington, D. C., June 1970 (Joint Meeting of the Atomic Industrial Forum and the Electric Power Council on the Environment).

64. Smith, E. C., and M. W. Larinoff: Power Plant Siting. Performance and Economics with Dry Cooling Tower Systems. Paper presented at American Power Conference, Chicago, Illinois, April 1970.

65. Heeren, H., and L. Holly: Air Cooling for Condensation and Exhaust Heat Rejection in Large Generating Stations. Paper presented at American Power Conference, April 1970.

66. Kolflat, T.: How to Beat the Heat in Cooling Water. Electrical World, October 14, 1968, pp. 31-33.

67. Cooling Tower Fundamentals and Application Principles. Marley Co. publication, 1967.

68. Chervishev, P. S., et al.: Experience with Development Work and Manufacture of 100-Mw Gas Turbine Plant at LMZ. ASME Paper No. 70-GT-30, 1970.

69. Congiu, A.: A 37/42 Mw Gas Turbine for Power Generation. ASME Paper No. 64-GTP-4, 1964.

70. Stewart, W. L., et al.: Brayton Cycle Systems. Selected Technology for the Electric Power Industry. NASA SP-5057, September 1968.

71. STAL-LAVAL Turbine Company: Technical Information Letter - Gas Turbines for Peak Load Generation, 1964.

72. Baldwin, C. J., et al.: Future Role of Gas Turbines in Power Generation. Proceedings of the American Power Conference 27th Annual Meeting, Vol. XXVII, 1965, pp. 484-500.

73. Bailey, W. D.: Operating Experience with a Multijet Gas Turbine-Generator. ASME Paper No. 68-GT-57, 1968.

74. Gatzemeyer, J. B., et al.: Characteristics of a New 46,000-kw Packaged Gas Turbine Power Plant Presented at American Power Conference 30th Annual Meeting, 1968.

75. Starkey, N. E.: Long-Life Base-Load Service at 1600 F Turbine Inlet Temperature. ASME Paper No. 66-GT-98, 1966.

76. Gaskins, R. C., and J. M. Stevens: World's Largest Single-Shaft Gas Turbine Installation. ASME Paper No. 70-GT-124, 1970.

77. Bankoul, V., and J. H. B. Kean: A New 30-Mw Packaged Gas Turbine Power Plant. ASME Paper No. 70-GT-11, 1970.

78. Martens, W. R., and W. A. Raabe: The Materials Challenge of High-Temperature Turbine Vanes and Blades. ASME Paper No. 67-GT-17, 1967.

79. Hare, A., and H. H. Malley: Cooling Modern Aero Engine Turbine Blades and Vanes. SAE Paper No. 660053, January 1966.

80. Sharp, W. H.: High Temperature Alloys for the Gas Turbine - The State of the Art. SAE Paper No. 650708, October 1965.

81. Oberg, A.: Design Features for Maintainability in the Pratt and Whitney Aircraft JT9D Gas Turbine Engine. SAE Paper No. 680337, May 1968.

82. Keen, J. M. S.: Design Features of Rolls-Royce Advanced Technology Engines. SAE Paper No. 680338, 1968.

83. Freche, C., and R. W. Hall: NASA Programs for Development of High-Temperature Alloys for Advanced Engines. AIAA Journal of Aircraft, September-October 1969, pp. 424-431.

84. Halls, G. A., and S. G. Baker: Turbine Blade Cooling - The Global Picture. AIAA Technical Information Service. Presented to the 9th International Aeronautical Congress, June 1969.

85. Thompson, E. R., et al.: Investigation to Develop a High Strength Eutectic Alloy with Controlled Microstructure. UA Research Laboratories Report J-910868-4, July 31, 1970.

86. Schloesser, V. V.: A Large Peaking Gas Turbine. Proceedings of the American Power Conference, Vol. 31, 1969.

87. Allen, R. P., and R. C. Petitt: New Gas Turbine Design for Large Power Systems. General Electric Company. Paper presented at American Power Conference 32nd Annual Meeting, April 22, 1970.

88. McDonald, C. F.: Study of a Lightweight Integral Regenerative Gas Turbine for High Performance. AiResearch Report 70-6179.

89. Weir, R. H.: Advances in Gas Turbine Technology. The Chartered Mechanical Engineer, March 1962.

90. Petitt, R. C.: Design and Development of a New 11,000 hp Industrial Gas Turbine. ASME Paper No. 69-GT-111, March 1969.

91. Battelle Memorial Institute: Current and Future Usage of Materials in Aircraft Gas Turbine Engines, February 1, 1970.

92. Bradley, E., et al.: The Pratt & Whitney Gas Turbine Story. Metal Progress, March 1970.

93. Hazard, H. R.: Combustors, Gas Turbine Engineering Handbook, First Edition, Gas Turbine Publications, 1966.

94. Burns and Roe Correspondence with Westinghouse under Department of Health, Education and Welfare Contract CPA-22-69-114 to UA Research Laboratories, July 1, 1969.

95. Burns and Roe Correspondence with General Electric under Department of Health, Education, and Welfare Contract CPA-22-69-114 to UA Research Laboratories, July 1, 1969.

96. Ault, G. M.: Engineering Mechanics and Materials. Selected Technology for the Electric Power Industry. NASA SP-5057, September 1968.

97. Peters, D., and J. Mortuner: Ceramic Turbines: Why Britain is Leading the Race. The Engineer, February 26, 1970, pp. 29-33.

98. Kraft, E. H.: An Analysis of W/SI_3N_4 Composite as a Possible High Strength, High Temperature Material. UA Research Laboratories Report J-110603-1, September 9, 1970 (Controlled).

99. Landerman, A.: Discussions with Corning Glass Works Personnel Concerning Cercor® Gas Turbine Regenerators. UA Research Laboratories File Memorandum, June 1, 1970.

100. Recuperators vs Regenerators. Discussion by Paul A. Pitt. Gas Turbine Magazine, September-October 1966.

101. Curbishley, G., et al.: Hot Corrosion Resistance of Materials for Small Gas Turbine Recuperators. USAAVLABS Technical Report 69-92, December 1969.

102. Letter from M. H. McClew, Harrison Radiator Division of General Motors Corporation to A. M. Landerman, UA Research Laboratories, June 10, 1970.

103. Jakubowski, S. T.: Plate-Fin Recuperator 9000 hp to 2000 hp Industrial Gas Turbine Engines. AiResearch Manufacturing Division, Garrett Corporation, Report No. 69-5471, August 27, 1969.

104. Kays, W. M., and A. C. London: Compact Heat Exchangers. McGraw-Hill Book Company.

105. Wolfe, P., and H. F. May: Design Experience with Regenerators for Industrial Gas Turbines. ASME Paper 69-GT-106, March 1969.

106. Letter from M. H. McClew, Harrison Radiator Division, General Motors Corporation, to W. R. Davison, UA Research Laboratories, January 5, 1970.

107. Smith, E. C.: Technical Data Relevant to Direct Use of Air for Process Cooling, Hudson Engineering Corporation.

108. 16th Steam Station Cost Survey. Electrical World, November 3, 1969, pp. 41-56.

109. Choosing Your Next Plant? Interview with Ken Hamming, Sargent & Lundy Engineers, pp. 32-34.

110. Swengel, F. M.: A New Era of Power Supply Economics. Power Engineering, March 1970.

111. Pfersdorff, D. H.: Electric Utility Gas Turbines - A Maintenance Report. Paper No. 66-GT-100, presented at Gas Turbine Conference and Products Show, Switzerland, March 1966.

112. Cooling Pond Planned for Cedar Bayou Plant. Electrical World, June 8, 1970. p. 28.

113. Galveston Bay: Test Case of an Estuary in Crisis. Science, February 20, 1970, pp. 1102-1107.

114. Brown, W. D.: Twentieth Annual Electric Industry Forecast. Electrical World, September 15, 1969, pp. 93-98.

115. Bauer, H. E., et al.: Electric Utility Equipment Requirements, II - Equipment Trends. United Aircraft Research Laboratories Report E-110303-2, August 1966.

116. Baldwin, C. J.: Probability Calculation of Generation Reserves. Westinghouse Engineer, March 1969, pp. 34-40.

117. Kirchmayer, L. K.: Application of Probability Methods to Generating Capacity Problems. AIEE Transactions, February 1961.

118. Calabrese, G.: Generating Reserve Capacity Determined by the Probability Method. AIEE Transactions, Vol. 66, 1947.

119. Miller, A. L.: Details of Outage Probability Calculations. AIEE Transactions August 1958.

120. Kist, C., and G. J. Thomas: Probability Calculations for System Generation Reserves. AIEE Transactions, August 1958.

121. Limmer, H. D.: Determination of Reserve and Interconnection Requirements. AIEE Transactions, August 1958.

122. Garver, L. L.: Reserve Planning for Interconnected Systems. Power Engineering, May 1970, pp. 40-43.

123. Parisian, R. W.: How Reliable are Today's Prime Movers? Power, January 1970, pp. 45-47.

124. Carstens, J. P.: Economic Advantages of Power System Expansion with Dispersed Gas Turbines. United Aircraft Research Laboratories Report B-110052-6, August 1963.

125. Lessard, R. D.: Telephone conversation with Mr. P. Ashton of HELCO. Memorandum to Mr. F. R. Biancardi, November 4, 1970.

126. Skaperdas, G. T.: Commercial Potential for the Kellogg Coal Gasification Process. M. W. Kellogg Co. Research and Development Report No. 38, Office of Coal Research Contract No. 14-01-0001-380, 1967.

127. Bituminous Coal Research: Gas Generator Research and Development, Survey and Evaluations, Phase 1, Vols. I and II. Office of Coal Research Contract No. 14-01-0001-324, 1965.

128. Rudolph, P. F. H.: New Fossil-Fueled Power Plant Process Based on Lurgi Pressure Gasification of Coal. Combined Meeting of the American Chemical Society and Chemical Institute of Canada, Toronto, Canada, May 1970.

129. Eastman, duB.: Gasification and Liquefaction of Coal. Proceedings of the American Institute of Mining Engineers, 1953.

130. Forney, A. J., et al.: A Process to Make High-Btu Gas from Coal. US Department of the Interior, Bureau of Mines Technical Progress Report 24, April 1970.

131. Bituminous Coal Research: Coal Gasification for Combined-Cycle Power Generation without Atmospheric Pollution. BCR Report RPP-127 R2, June 23, 1967.

132. Institute of Gas Technology: Cost Estimate of a 500 Billion Btu/Day Pipeline Gas Plant Via Hydrogasification and Electrothermal Gasification of Lignite. US Department of the Interior, Office of Coal Research, Research and Development Report No. 22, 1968.

133. Anon.: Coal-to-Gas Plant "Possible" by 1974. The Oil and Gas Journal, October 26, 1970.

134. Theodore, F. W.: Low Sulfur Boiler Fuel Using the CONSOL CO_2 Acceptor Process. US Department of the Interior, Office of Coal Research Contract No. 14-01-0001-415, Report No. 2, PB 176 910, November 1967.

135. Linden, H. R.: Sources of Gas Supply for the USA to the Year 2000. Paper presented at the International Gas Union Conference, June 1970.

136. Airbus is Ready, But the Airlines are Not. Business Week, July 18, 1970, pp. 80-81.

137. Watts, F. A.: Aircraft Turbine Engines. Development and Procurement Cost. A Rand Corporation Report RM-4670-PR, November 1965.

138. Allen, R. P., and R. C. Petitt: New Gas Turbine Design for Large Power Systems. Presented at the American Power Conference, April 22, 1970.

139. Cuffe, S. T., and R. W. Gerstle: Emissions from Coal-Fired Power Plants; A Comprehensive Summary. Presented at the American Industrial Hygiene Association Meeting, May 1965.

140. Bagwell, F. A., et al.: Oxides of Nitrogen Emission Reduction Program for Oil and Gas Fired Utility Boilers. Presented at the American Power Conference, April 21-23, 1970.

141. Bell, A. W., N. B. deVolo, and B. P. Breen: Nitric Oxide Reduction by Controlled Combustion Process. Presented at the Spring Meeting of Western States Section, The Combustion Institute, April 20-21, 1970.

142. Peters, G. T.; Editor: Reference Handbook of Prime Mover Characteristics. United Aircraft Research Laboratories Report D-110287-1, 1966.

TABLE I

UNITED STATES CONSUMPTION OF ENERGY RESOURCES BY ELECTRIC UTILITIES

Trillions of Btu's

	Actual Consumption	Projected						
Type of Fuel	1965	1970	1975	1980	1985	1990	2000	Reference
Coal	-	8050	-	7,450	-	5910	-	(7)
	-	8493	9,050	9,640	9,780	-	-	(8)
	6391[2]	8035	11,134	12,516	-	-	18,720	(11)
	6400	-	-	11,000	-	-	-	(9)
	5880	-	8,520	-	11,300	-	-	(10)
Oil	-	1570	-	3,230	-	3740	-	(7)
	-	837	720	659	655	-	-	(8)
	890[2]	856	863	861	-	-	861	(11)
	700	-	-	625	-	-	-	(9)
	743	-	940	-	1,070	-	-	(10)
Gas	-	3240	-	4,600	-	6100	-	(7)
	-	3336	3,963	5,156	6,619	-	-	(8)
	2691[2]	2589	2,789	2,976	-	-	4,128	(11)
	2440	-	-	4,400	-	-	-	(9)
	2399	-	3,770	-	4,850	-	-	(10)
Hydro	-	-	-	-	-	-	-	(7)
	-	804	893	1,098	1,286	-	-	(8)
	2039[2]	2193	2,422	3,027	-	-	5,056	(11)
	2090	-	-	4,060	-	-	-	(9)
	2050	-	2,580	-	3,200	-	-	(10)
Nuclear	-	1000	-	12,000	-	41,500	-	(7)
	-	737	5,964	13,300	25,913	-	-	(8)
	52[2]	874	1,803	4,878	-	-	43,526	(11)
	-	-	-	11,300	-	-	-	(9)
	38	-	4,260	-	15,500	-	-	(10)

(1) Converted from data in Ref. 7 using 26.2 x 10^6 Btu/ton for coal, 5.8 x 10^6 Btu/barrel for oil, 10^3 Btu/cu-ft for gas.

(2) Data for 1966 inserted, natural gas liquids included in oil figures

TABLE II

REGIONAL ELECTRIC GENERATION BY FUEL TYPE AND HYDROELECTRIC POWER

Billion kwhr
Data from Ref. 1

Region	Fuel	Actual 1966	Projected 1970	1975	1980	1985	1990
South Central	Coal	5.3	7.35	24.30	43.2	70.30	94.0
	Oil	.03	.12	.85	.82	.67	.65
	Gas	110.5	164.0	225.50	289.0	339.0	431.0
	Nuclear	-	-	13.90	73.0	198.0	404.0
	Hydro(1)	-	-	-	-	-	-
Southeast	Coal	161.1	218.1	245.8	259.4	282.3	316.4
	Oil	18.0	14.7	12.4	12.4	13.0	12.3
	Gas	14.8	23.2	27.8	24.6	36.5	45.3
	Nuclear	-	7.2	114.5	286.7	487.2	761.3
	Hydro	25.0	28.7	35.4	36.6	38.5	41.3
West	Coal	12.5(2)	29.8	-	125.0	-	211.5
	Oil	11.6	17.9	-	35.8	-	30.2
	Gas	64.1	94.0	-	76.0	-	80.6
	Nuclear	0.89	7.45	-	210.0	-	685.0
	Hydro	120.0	159.0	-	184.0	-	198.0
East Central	Coal	199.7	244.2	279.9	348.9	371.3	428.1
	Oil	0.2	0.3	0.1	0.1	-	0.2
	Gas	0.2	0.4	0.2	0.3	0.1	0.5
	Nuclear	0.9	4.3	57.5	110.0	23.6	420.1
	Hydro	2.2	2.6	5.9	6.6	7.0	6.6
West Central	Coal	99.9	124.8	146.3	142.4	154.2	155.3
	Oil	.9	1.1	1.3	1.5	1.7	2.0
	Gas	23.9	30.1	35.0	34.5	37.8	39.5
	Nuclear	1.4	17.5	69.5	178.4	300.1	489.1
	Hydro	12.2	12.2	12.9	13.7	14.4	15.6
Northeast	Coal	130.5	157.5	156.1	141.9	122.0	101.5
	Oil	43.4	43.8	36.5	30.1	26.0	23.6
	Gas	9.3	15.3	16.9	15.1	13.3	12.1
	Nuclear	2.2	30.7	148.5	309.3	492.7	747.1
	Hydro	10.1	13.8	15.5	18.4	23.5	27.3

(1) Data not provided.
(2) Estimates included for 1965.

TABLE III

REGIONAL FOSSIL FUEL COSTS FOR ELECTRIC ENERGY GENERATION
1965

From Ref. 10

Per Cent Total Fossil Btu			Region	Average Fuel Cost ¢/million Btu			Weighted Average Fossil Fuel Cost ¢/million Btu
Coal	Oil	Gas		Coal	Oil	Gas	
61	36	3	New England	32.4	34.4	34.2	33.8
77	16	7	Middle Atlantic	25.4	32.3	33.8	27.7
97	0	3	East North Central	23.7	66.2	25.9	23.3
51	1	48	West North Central	25.6	50.8	24.2	25.5
80	12	8	South Atlantic	24.8	33.7	32.3	26.7
92	0	8	East South Central	18.4	62.8	23.8	19.3
0	0	100	West South Central	-	-	19.8	19.8
49	4	47	Mountain	19.0	26.2	27.1	23.3
0	19	81	Pacific	-	32.0	31.4	31.5

Note: Regions are not identical with those designated by the FPC.

TABLE IV

SULFUR CONTENT AND DISTRIBUTION OF COAL RESERVES

From Ref. 25

(a) Sulfur Content of all US Coal Reserves

	Low Sulfur (1% or less)		Medium Sulfur (1.1 to 3.0%)		High Sulfur (over 3.0%)		Total	
	Billion Tons	% of Total	Billion Tons	% of Total	Billion Tons	% of Total	Billion Tons	% of Total
Bituminous	215.7	13.7	194.9	12.4	314.2	19.9	724.7	46.0
Sub-bituminous	387.2	24.6	1.5	0.1	------	----	388.7	24.7
Lignite	406.0	25.8	41.6	2.6	------	----	447.6	28.4
Anthracite	14.7	0.9	0.4	----	------	----	15.2	0.9
Total	1023.6	65.0	238.4	15.1	314.2	19.9	1576.2	100.0

(b) Distribution of US Coal Reserves According to Heat Content*

	Low Sulfur	Medium Sulfur	High Sulfur	Total % of Total
Bituminous	17.3	15.6	25.1	58.0
Sub-bituminous	22.4	0.1	----	22.5
Lignite	16.6	1.7	----	18.3
Anthracite	1.2	----	----	1.2
Total	57.5	17.4	25.1	100.0

*Based on heating values as follows

Bituminous	- 26,200,000 Btu/ton
Sub-bituminous	- 20,000,000 Btu/ton
Anthracite	- 25,400,000 Btu/ton
Lignite	- 13,400,000 Btu/ton

TABLE V

DISTRIBUTION OF COAL WITH SULFUR CONTENT OF ONE PERCENT OR LESS

From Ref. 25

West of the Mississippi:	Billion Tons	Low-Sulfur Coal % of Total
Bituminous	134	13.1
Sub-bituminous	387	37.8
Lignite	406	39.6
Anthracite	2	0.2
	929	90.7
East of the Mississippi:		
Bituminous	82	8.0
Sub-bituminous	--	---
Lignite	--	---
Anthracite	13	1.3
Total	95	9.3

Distribution by States	Geological Reserves in Place* Billion Tons	Low-Sulfur Bituminous Output Million Tons	% of Total State Output	% of Total National Output
West Virginia	47.5	89.0	63	18.4
Kentucky (eastern portion)	22.1	40.4	90	8.3
Virginia	8.1	26.0	82	5.4
Alabama	2.1	8.9	62	1.8
Pennsylvania	1.2	----	--	---
Other	1.0	1.7	--	0.4
Total	82.0	166.0		34.3

* Only 50% of the geological reserves are recoverable.

TABLE VI

SUMMARY OF PROJECTED FUEL COSTS
IN SELECTED REGIONS OF THE US

From Various Sources

Fuel Price Estimates (¢/million Btu) excluding transportation	1968	1970	1980	1990
		Regions		
		West		
Coal (in Rocky Mountain Region)	16	15	16	17
Natural Gas	30	31	34	36
Oil, No. 6	32	32	28	32
Oil, Low Sulfur	-	41	44	44
Uranium	26	20	15	13
Thorium	-	20	15	13
		South Central		
Coal	-	17 to 29	-	25 to 31.5
Natural Gas	-	23	-	21 to 38
Oil, Low Sulfur	-	-	37 to 40	-
		East Central		
Coal, Low Sulfur	-	35 to 40	40	40
Coal, High Sulfur	-	27.5	25	25.0
Oil, Low Sulfur	-	32	-	30.0
		Southeast		
Coal	-	25	30	32
Natural Gas (Plus Gasified Fuel)	-	25	30	40 +
		West Central		
Coal	-	32	-	30
Oil, High Sulfur	-	35	-	35
Oil, Low Sulfur	-	-	-	-
		Northeast		
Coal, High Sulfur	-	25-33	30	30
Coal, Low Sulfur	-	45-55	55	45
Residual-Oil, Low Sulfur	-	50	45	45
Residual-Oil, High Sulfur	-	25	30	35

TABLE VII

SUMMARY OF EXISTING AND EMERGING REGIONAL WATER MANAGEMENT PROBLEMS

From Ref. 34

Water Resource Regions	Adequacy of(1) Annual Natural Runoff	Ground Water(2) Storage Depletion	Water Quality			
			Wastes(3)	Heat(4)	Salinity(5)	Sediment(6)
North Atlantic	3	3	1	1	4	3
South Atlantic-Gulf	4	3	2	3	4	2
Great Lakes	3	4	1	1	4	3
Ohio	3	4	2	2	4	3
Tennessee	4	4	3	3	4	3
Upper Mississippi	3	4	2	2	4	3
Lower Mississippi	4	4	3	4	4	1
Souris-Red-Rainy	2	4	3	4	3	4
Missouri	2	2	3	3	3	2
Arkansas-White-Red	2	1	3	4	1	2
Texas Gulf	2	1	2	3	2	2
Rio Grande	1	1	2	4	1	1
Upper Colorado	1	4	3	4	3	2
Lower Colorado	1	1	2	4	1	1
Great Basin	1	3	2	4	3	3
Columbia-North Pacific	3	3	3	2	4	4
California	2	2	2	3	2	3
Alaska	4	4	3	4	4	4
Hawaii	4	3	3	4	4	4
Puerto Rico	4					

(1) Comparison of projected consumptive use with natural runoff which includes perennial yields of ground water aquifers.

(2) An indication of the extent that use of ground water would exceed recharge.

(3) An indication of pollution loading and of investment required for alleviation.

(4) An indication of waste heat discharges from industrial and steam-electric cooling requirements and of investment required for alleviation.

(5) An indication of the relative severity of the salinity problem from both natural sources and man-caused sources

(6) An indication of the relative severity of sediment from land and stream bank erosion both natural and man-caused

Order of Severity:	Rating
Severe problem in some areas or major problem in many areas	1
Major problem in some areas or moderate problem in many areas	2
Moderate problem in some areas or minor problem in many areas	3
Minor problem in some areas	4

TABLE VIII

LIMITING TEMPERATURE CRITERIA IN WATER QUALITY STANDARDS FOR SOUTH CENTRAL POWER REGION

From Ref. 33

(Temperatures in Deg F)

States and Other Jurisdictions	Limiting Uses: Cold Water Fish		Limiting Uses: Small-mouth Bass		Limiting Uses: Warm Water Fish		All Waters		Exceptions and Remarks
	Temp.	Rise	Temp.	Rise	Temp.	Rise	Temp.	Rise	
Ark.	68	5	86	5	95	5	95	5	
Kan.							90	5	Not approved.
La.							97	5	Except some rivers with 95° max. and 4° rise
Miss.							93	10	10° rise not approved.
Mo.							90	5	Except Des Moines where max. temp. is 93° and except North Fork White, Current, and Eleven Point Rivers where max. rise is 2°
Okla.	70	5	75	5	93	5	93	5	
Tex.							96 93	5 5 4 1.5	Except Canadian River and tidal waters. Canadian River Fall, winter, spring - tidal waters Summer tidal waters

TABLE IX

SCHEDULED ADDITIONS OF STEAM POWER ELECTRIC GENERATING CAPACITY BY YEARS

From Ref. 2

Year	Conventional			Nuclear		
	Number of Units	Total Capacity, Mw	Average Unit Capacity, Mw	Number of Units	Total Capacity, Mw	Average Unit Capacity, Mw
1969 (last 3 months)	21	6,134	292	4	1,817	455
1970	53	16,689	314	7	4,865	
1971	51	18,876	370	11	8,643	785
1972	41	19,888	485	16	13,531	845
1973	34	17,371	511	16	15,396	960
1974	14	6,115	436	12	11,009	920
1975 and later	5	3,721	734	9	8,492	935
Total	219	88,793		75	63,752	

(1) Based on scheduled dates of commercial operation as of October 1, 1969, in terms of manufacturers' ratings of the units.

TABLE X

SCHEDULED ADDITIONS OF STEAM POWER ELECTRIC GENERATING CAPACITY BY REGIONS(1)

From Ref. 2

Region	Conventional			Nuclear		
	Number of Units	Total Capacity, Mw	Average Unit Capacity, Mw	Number of Units	Total Capacity, Mw	Average Unit Capacity, Mw
Northeast	24	10,771	450	25	22,137	885
East Central	42	19,645	467	8	7,343	915
Southeast	37	17,999	486	20	18,102	909
West Central	33	9,025	274	14	10,994	545
South Central	61	20,303	330	1	903	903
West(2)	22	9,050	411	2	4,373	1,093
Total	219	88,793		75	63,752	

(1) Based on scheduled dates of commercial operation as of October 1, 1969 in terms of manufacturers' ratings of units. Regions defined by map shown in Fig. 3.

(2) For purposes of simplification, estimates of the power additions from original 8 FPC power regions have been incorporated into present 6 regions.

TABLE XI

ESTIMATED PERFORMANCE OF TYPICAL STEAM POWER STATIONS

1970 Decade and Early 1980's	Net Station Efficiency, % @ 70% Load Factor	Net Station Efficiency, % @ Design Point
Coal	36.6	38.4
Residual Oil	37.0	38.9
Natural Gas	36.0	37.8
Late 1980's Decade		
Coal	38.6	40.5
Residual Oil	39.0	41.0
Natural Gas	38.1	40.0

TABLE XII

CAPITAL COST SUMMARY FOR COAL-FIRED STEAM STATIONS

$/kw

	TVA Bull Run 1[1]	Design Study[2]
Land and Land Rights	$/kw 1.88	$/kw 0
Structures and Improvements	21.36	13.07
Boiler Plant Equipment	59.47	61.00
Turbogenerator Units	22.07	36.08
Accessory Electrical Equipment	6.34	11.05
Misc. Power Plant Equipment	2.25	0.52
Transmission Plant	4.82	(3)
General Expense and Overhead	21.84	(3)
Station Equipment	(3)	1.72
Direct Design Cost and Final Drawings	7.64	(3)
Other Expenses	(3)	1.25
Subtotal	$/kw 147.67	$/kw 124.72
Interest During Construction	9.97	19.80
Engineering, Design, Construction Supervision, and Contingency	(3)	12.24
Escalation	(3)	20.20
Total	$/kw 157.64	$/kw 176.96

(1) Represents costs for an actual plant which has been completed (Ref. 44).
(2) Represents estimated costs for a plant which could be built (Ref. 45).
(3) Not applicable due to the differences in the method of reporting costs.

TABLE XIII

INVESTMENT COSTS FOR ALTERNATE METHODS OF COOLING CONDENSER WATER DISCHARGES

$/kw

	Fossil-Fueled Plants				Nuclear-Fueled Plants			
	Data Source				Data Source			
Cooling Water System	Ref. 63	Ref. 56*	Ref. 60	Ref. 62	Ref. 63	Ref. 56*	Ref. 35***	Ref. 62**
Once-Through River	6.25	5.30-5.00	Base	--	9.25	5.24-5.88	8.00	--
Once-Through Ocean	6.11	6.00-6.30	--	--	9.00	6.24-6.88	9.68	--
Cooling Pond/Reservoir	8.50	6.50	1.65	--	12.00	7.50	9.65	--
Spray Pond	--	7.60	--	--	--	8.10	--	--
Spray Cooling Canals	--	--	3.41	--	--	--	--	--
Wet Cooling Tower - Mech. Draft	8.00	7.20	3.75	--	11.75	9.40	11.94	--
Wet Cooling Tower - Nat. Draft	11.25	7.50-8.50	6.92	--	17.5	11.50-12.50	14.17	--
Dry Cooling Tower - Mech. Draft	19.25	13.0	19.07	17	30.5	15.00	29.90	23
Dry Cooling Tower - Nat. Draft	39.00	20.0	20.82	20	62.5	22.00	--	27

* Costs vary with temperature rise in condenser from 10 to 20 F.

** Costs vary with condenser design pressure from 5.5 to 16.0 in. Hg abs.

*** Values presented in Ref. 35 are relative to a base for once-through river cooling. This base value was selected as $8.00/kw for comparison purposes only.

TABLE XIV

ADDITIONAL COST FACTORS FOR ALTERNATIVE COOLING SYSTEMS

Cooling Water System	Auxiliary Power Requirements[1] (% Generator Output)	Added Fuel Cost[2] for Auxiliaries (mills/kwhr)	Added Cost[3] for Auxiliary Power (mills/kwhr)	Maintenance, Water Treatment mills/kwhr	Loss in[4] Capability (mills/kwhr)	[5] Added Fuel Costs (mills/kwhr)
Once through River	0.425	0.0116	0.0085	0.0058	0	0
Once through Ocean	0.375	0.0102	0.0075	0.0050	0	0
Cooling Pond/Reservoir	0.425	0.0116	0.0085	0.0058	0.012	0
Spray Pond or Canal	0.875	0.0240	0.0175	0.0120	0.012	0.0108
Wet Cooling Tower-Mech. Draft	1.075	0.0294	0.0215	0.0147	0.012	0.0108
Wet Cooling Tower-Nat. Draft	0.875	0.0240	0.0175	0.0120	0.012	0.0108
Dry Cooling Tower-Mech. Draft	3.04	0.0930	0.0605	0.0147	0.18	0.2705
Dry Cooling Tower-Nat. Draft	0.91	0.0249	0.0182	0.0120	0.18	0.2705

(1) Based on data from Ref. 63.
(2) Based on heat rate of 10,200 Btu/kwhr for dry tower-mechanical draft, 10,000 Btu/kwhr for dry tower-natural draft, 9,110 Btu/kwhr for all others, and fuel cost at 30¢/10^6 Btu.
(3) Based on \$100/kw, 80% load factor, 14% capital charges.
(4) Based on \$100/kw for incremental capacity.
(5) Cost factors for loss in capability and added fuel cost will depend on climatic conditions (see Ref. 60).

TABLE XV

GAS TURBINE COMBUSTOR MATERIALS

Metal Surface Temperature - F	Material	Condition	Status	Primary Properties
1200*	AISI Type 310	Uncoated	Industrial	Strength
1600	Hastelloy X	Uncoated	Production	Strength; oxidation resistant
1800	Haynes 188	Coated	New Engine	Strength; creep; fatigue; and oxidation resistant
2000	TD-Ni	Coated	New Engine	Strength; creep; fatigue; and oxidation resistant
2200	TD-NiCr	Uncoated	Experi-mental	Strength; creep; fatigue; and oxidation resistant
2300	Dispersed thoria in nickel	--	Experi-mental	Strength; creep; fatigue

* Turbine inlet temperatures are normally some 600 F above these values in present engines.

TABLE XVI

PROJECTED TECHNOLOGY FOR BASE-LOAD GAS TURBINE ENGINES

Simple-Cycle and Regenerative-Cycle Power Systems

Parameter	Time Period		
	1970 Decade	Early 1980's	Late 1980's
Turbine Inlet Gas Temperature - F	1600 to 2200[1] 2200 to 2400[2]	2000 to 2400[1] 2400 to 2800[2]	2400 to 2800[1] 2800 to 3100[2]
Compressor Pressure Ratio	4 to 28	4 to 26	4 to 36
Compressor Polytropic Efficiency - %	89	92	93
Turbine Nominal Adiabatic Efficiency - %	90	92	93
Regenerator Airside Effectiveness - %	70 to 90	70 to 90	70 to 90
Regenerator Total Pressure Drop - %	4 and 8	4 and 8	4 and 8
Turbine Cooling Technique	Advanced Impingement-Convection	Advanced Impingement-Convection	Advanced Impingement-Convection

(1) Compressor bleed air uncooled

(2) Compressor bleed air precooled to levels of 125 to 250 F.

TABLE XVII

INFLUENCE OF COMPRESSOR PRESSURE RATIO ON POWER COST

200-Mw Unit Size

Turbine Inlet Temperature	Technology Time Period	Compressor Pressure Ratio	Power Cost Differential* mills/kwhr, Fuel Costs 30¢/10^6 Btu	Power Cost Differential* mills/kwhr, Fuel Costs 50¢/10^6 Btu
2000 F	1970 Decade	13:1	+ 0.14	+ 0.25
		20:1	base	base
		28:1	+ 0.22	+ 0.31
2400 F	1970 Decade	13:1	+ 0.24	+ 0.38
		20:1	base	base
		28:1	+ 0.03	+ 0.007
2600 F	Early-1980's Decade	12:1	+ 0.380	+ 0.67
		20:1	+ 0.091	+ 0.160
		28:1	base	base
2900 F	Late-1980's Decade	20:1	+ 0.23	+ 0.381
		28:1	+ 0.04	+ 0.09
		36:1	base	base

* Based on 0.8 load factor, 15% capital charges.

TABLE XVIII

APPROXIMATE CHARACTERISTICS OF 470-MW
COMPOUND-CYCLE GAS TURBINE DESIGN

Early-1980's Technology

Turbine Inlet Gas Temperature - F	2200
Reheat Gas Temperature - F	2200
Cycle Pressure Ratio	75:1
Low-Pressure Compressor Ratio	4:1
Engine Airflow lb/sec	2005
Net Thermal Efficiency - %	40%
Compressor Inlet Diameter - ft	11.7
Low-Compressor Stages	7
High-Compressor Stages	13
Gas Generator Turbine Stages	3
Power Turbine	
Stages	3
Rotational Speed - rpm	1800
Last-Stage Tip Diameter - ft	14.2
Last-Stage Blade Height - in.	26.9
Last-Stage Blade Root Stress - psi	47,000
Engine Selling Price - $/kw	20

TABLE XIX

CHARACTERISTICS OF ADVANCED GAS TURBINE POWER STATIONS

Time Period	Turbine Inlet Turbine F	Engine Pressure Ratio	Engine Size Mw	Engine Thermal Efficiency %	Station Thermal Efficiency %	Net Heat Rate Btu/kwhr	Engine Selling Price $/kw	Station* Prices $/kw
SIMPLE-CYCLE ENGINES								
1970-decade	2200 to 2400	20:1	200	31.9	30.6	11,160	32.0	80.0
Early 1980's	2400 to 2600	28:1	250	37.3	35.8	9,530	24.5	66.5
Late 1980's	2600 to 2900	30:1 to 36:1	250	40.4	38.7	8,820	28.0	71.0
REGENERATIVE-CYCLE ENGINES								
1970-decade	2000 to 2200	10:1	200	35.4	33.9	10,070	48.0	100
Early 1980's	2200 to 2400	12:1	200	39.2	37.6	9,080	43.7	92.7
Late 1980's	2600 to 2800	14:1	200	43.1	40.6	8,410	44.0	93.0

* Indoor construction assumed, outdoor construction will be only about $1 to 2/kw lower than figures shown

TABLE XX

POWER PLANT CHARACTERISTICS
EARLY-1980 DESIGN TECHNOLOGY

Cycle	Simple	Regenerative
Nominal Output-Mw	250	200
Turbine Inlet Temperature - F	2600	2400
Compressor Pressure Ratio	28:1	12:1
Engine Airflow - lb/sec	1295	1175
Compressor Stages Low/High*	13/5	14
Compressor Inlet Tip Diameter - ft	9.5	8.9
Compressor First-Stage Blade Height - in.	20.	20.
Compressor Exit Tip Diameter	5.5	7.5
Compressor Last-Stage Blade Height - in.	2.3	3.6
Compressor Turbine Stages High/Low*	1/3	2
Compressor Turbine First-Stage Blade Height - in.	7.7	9.0
Compressor Turbine Last-Stage Blade Height in.	12.2	13.7
Power Turbine Stages	3	2
Power Turbine Rotational Speed - rpm	1800	1800
Power Turbine Last-Stage Tip Diameter - ft	12.5	13.0
Power Turbine Last-Stage Blade Height - in.	22.8	24
Power Turbine Last-Stage Blade Centrifugal Stress - psi	36,200	40,000
Compressor Turbine First-Stage Vane Temperature - F	1925	1925
Compressor Turbine First-Stage Blade Temperature - F	1540	1686
Compressor Bleed Flow for Turbine Cooling ** - %	9.5	5.5
Power Turbine Exhaust Temperature - F	991	866

* Twin-spool design used for simple-cycle power plants.

** Compressor bleed air precooled to 200 F.

TABLE XXI

1000-MW STEAM-ELECTRIC STATION COSTS - 1980-DECADE DESIGNS

COSTS IN 1970 DOLLARS PER KILOWATT

Steam Conditions	3500 psig/1000 F/1000 F		3500 psig/1000 F/1000 F		3500 psig/1000 F/1000 F	
Number of Units	One		One		One	
Fuel	Coal		Oil		Gas	
Type Construction	Indoor		Indoor		Indoor	
Location	East Central		Northeast		Northeast	
Construction Time, Years	4		4		4	
FPC Account No. Description						
310 Land and Land Rights	$/kw	0.03	$/kw	0.22	$/kw	0.22
311 Structures and Improvements		9.00		7.64		7.03
312 Boiler Plant Equipment*		54.83		48.98		38.36
314 Turbine Generator Units**		34.20		33.69		33.69
315 Accessory Electrical Equipment		10.02		9.52		9.52
316 Miscellaneous Power Plant Equipment		0.46		0.50		0.50
353 Station Equipment		1.55		1.59		1.59
Total		110.09		102.14		90.91
Other Expenses		1.24		1.22		1.22
Subtotal		111.33		103.36		92.13
Engineering Design, Construction Supervision, and Contingency		13.08		12.28		11.54
Subtotal		124.41		115.64		103.67
Escalation		18.35		17.06		15.26
Subtotal		142.76		132.70		118.93
Interest During Construction @ 8%		22.83		21.23		18.98
Total	$/kw	165.59	$/kw	153.93	$/kw	137.91

* Includes cost of stacks, and dust collectors

** Includes cost of cooling tower

TABLE XXII

1000-MW STEAM-ELECTRIC STATION CAPITAL COSTS - 1970-DECADE DESIGNS

(COSTS IN 1970 DOLLARS PER KILOWATT)

Two 500-Mw Units
Steam Conditions: 2400 psig/1000 F/1000 F

Construction Time 4 Years
Interest Rate During Construction - 8%

Fuel	Type Construction	Region					
		Northeast	East Central	South-east	West Central	South Central	West
Coal	Indoor	$/kw 188	$/kw 180	$/kw 200	$/kw 178	$/kw 183	$/kw 183
	Outdoor	172	165	184	164	168	168
Oil	Indoor	172	166	184	164	168	168
	Outdoor	158	152	168	150	154	154
Gas	Indoor	152	146	161	144	148	148
	Outdoor	$/kw 136	$/kw 131	$/kw 145	$/kw 129	$/kw 132	$/kw 132

TABLE XXIII

1000-MW STEAM-ELECTRIC STATION CAPITAL COSTS - 1980-DECADE DESIGNS

(COSTS IN 1970 DOLLARS PER KILOWATT)

One 1000-Mw Unit
Steam Conditions: 3500 psig/1000 F/1000 F

Construction Time 4 Years

Fuel	Type Construction	Interest Rate During Construction	Region					
			Northeast	East Central	South-east	West Central	South Central	West
Coal	Indoor	6%	$/kw 166	$/kw160	$/kw177	$/kw158	$/kw162	$/kw 162
		8%	172	166	184	164	168	168
		10%	178	171	190	170	174	174
	Outdoor	6%	151	146	162	144	148	148
		8%	157	151	168	149	153	153
		10%	161	155	172	153	158	157
Oil	Indoor	6%	149	143	159	141	145	145
		8%	154	148	164	146	150	150
		10%	159	154	170	152	156	155
	Outdoor	6%	134	129	144	128	131	131
		8%	139	134	149	132	136	136
		10%	144	138	154	137	141	141
Gas	Indoor	6%	133	128	142	127	130	130
		8%	138	132	147	131	134	134
		10%	143	137	153	136	139	139
	Outdoor	6%	119	114	127	113	116	116
		8%	123	119	132	117	120	120
		10%	128	123	136	121	124	124

TABLE XXIV

BREAKDOWN OF FIXED CHARGES

Interest on Borrowed Capital:	Fixed Charges, %
Internal Capital	1.40
Debt Capital	4.80
Equity Capital	3.00
Subtotal	9.20%
Insurance	0.60
Taxes	1.45
Annual Maintenance	0.50
Depreciation	3.25
Total	15.00%

TABLE XXV

DETAILED COST BREAKDOWN FOR 1000-Mw SIMPLE-CYCLE AND REGENERATIVE-CYCLE GAS TURBINE STATIONS

(COSTS IN 1970 DOLLARS)

Early 1980-Decade Designs

FPC Account No. 341 - Structures and Improvements	Four 250-Mw Units Simple Cycle	Five 200-Mw Units Regenerative Cycle
Site Improvements		
Site Grading	$ 25,000	$ 25,000
Building Excavation	10,000	12,500
Borings	6,000	6,500
Landscaping	23,000	25,000
Fresh Water Supply	12,000	12,000
Fire Protection	100,000	100,000
Sewage Disposal and Drainage	19,000	19,000
Flagpole	5,000	5,000
Guard House	7,600	7,600
Railroad	50,600	50,600
Roads and Parking Lot	20,900	20,900
Fencing	15,000	15,000
Switchyard	10,700	10,700
Structures		
Administration Building	407,500	407,500
Turbine-Generator Building*	3,230,000	3,916,000
Gas Meter Area	2,700	2,700
Subtotal	$3,945,000	$4,636,000

* Indoor construction. Outdoor construction cost would equal 70% of this value.

TABLE XXV (Cont'd.)

FPC Account No. 343 - Prime Movers	Four 250-Mw Units Simple Cycle	Five 200-Mw Units Regenerative Cycle
Gas Turbines	$ 24,550,000	$ 27,600,000
Start-Up Motors	30,000	37,500
Torque Converters	300,000	375,000
Lube Oil Purification and Storage	74,000	88,000
Lube Oil Fire Protection	80,000	100,000
Turbine Air Precooler System	1,000,000	437,500
Air Compressor Serv. & Inst.	50,000	50,000
Breeching Incl. Liners, Silencers, and Insulation	2,700,000	3,160,000
Expansion Joints	120,000	150,000
Inlet Filter Screen	135,000	150,000
Turbine Enclosure Air Coolers	80,000	100,000
Emergency Cooling Water, Tank, Pumping and Piping	10,000	10,000
Misc. Pump and Tanks	20,000	25,000
Control Boards, Inst. and Controls	200,000	250,000
Computer	200,000	200,000
Piping	980,000	1,000,000
Insulation	152,000	150,000
Regenerators	---	16,100,000
Subtotal	$ 30,681,000	$ 49,983,000
FPC Account No. 344 - Generators		
Generators	$ 9,875,000	$ 9,875,000
Hydrogen Seal Oil Coolers	40,000	50,000
Subtotal	$ 9,915,000	$ 9,925,000
FPC Account No. 345 - Accessory Electrical Equipment		
Auxiliary Transformers	29,100	48,500
Start-Up Transformers	86,200	35,000
8000A Insul. Phase Bus Duct	---	1,122,800
1200 A Insul. Phase Bus Duct	62,100	103,500
Potential Transformers	39,000	65,000
Surge Protection	19,855	32,000

TABLE XXV (Cont'd.)

FPC Account No. 345 - Accessory Electrical Equipment	Four 250-Mw Units Simple Cycle	Five 200-Mw Units Regenerative Cycle
480 Volt Power Switchgear	$ 77,325	$ 114,275
480 Volt Motor Control Centers	33,235	46,615
Remote Motor Controls	2,625	3,500
Duplex Relay Switchboard	68,000	68,000
Annunciator Panel	16,500	16,500
Control Console	34,500	34,500
Turbine Control Panel	6,000	6,000
Temperature Detection Panel	15,000	15,000
Equipment Connect	1,800,000	1,800,000
Testing	378,300	378,300
250 Volt DC Switchboard	27,500	27,500
250 Volt DC Panelboard	3,600	3,600
Station Battery and Rack	53,000	53,000
Battery Chargers	56,500	56,500
Cable Tray	82,000	82,000
600 Volt Instrument Cable	60,400	80,000
600 Volt Control Cable	122,000	160,000
Grounding Systems	370,500	370,500
480 Volt Valve Control Center	25,600	25,600
Conduit - Fittings	48,200	61,205
600 Volt Power Cable	23,935	31,330
1000 Volt Power Cable	35,125	43,545
12,000A Insol. Phase Bus Duct	1,073,900	---
Subtotal	$ 4,650,000	$ 4,884,270
FPC Account No. 346 - Misc. Power Plant Equipment		
Laboratory and Sampling Equipment	10,000	10,000
Tools, Shop, Stores, and Work Equip.	75,000	75,000
Lockers	3,000	3,000
Emergency Equipment	10,000	10,000
Misc. Cranes and Hoists	15,000	15,000
Portable Fire Extinguishers	20,000	20,000
Communication Equipment	50,000	50,000
Lunch Room Equipment	20,000	20,000
Office Furniture and Machines	15,000	15,000
Subtotal	$ 218,000	$ 218,000
FPC Account No. 353 - Station Equipment		
Station Transformer	$ 1,719,000	$ 1,719,000

TABLE XXVI

1000-MW GAS TURBINE STATION COSTS

(COSTS IN 1970 DOLLARS)
Early 1980 Technology

Cycle Turbine Inlet Temperature / Compressor Pressure Ratio / Regenerator Effectiveness		Simple-Cycle 2600 F/28:1	Simple-Cycle 2600 F/28:1	Regenerative 2400 F/12:1/80%	Regenerative 2400 F/12:1/80%
Number of Units		4	4	5	5
Fuel		Gas	Gas	Gas	Gas
Type Construction		Indoor	Outdoor	Indoor	Outdoor
Net Station Efficiency, %		35.8	35.8	37.6	37.6
Construction Time, yr		2	2	2	2
Net/Gross Output, Mw		1000/1040	1000/1040	1000/1040	1000/1040
FPC Account Number	Description				
340	Land & Land Rights	$ 100,000	$ 30,000	$ 100,000	$ 30,000
341	Structures & Improvements	3,945,000	2,975,000	4,636,000	3,460,000
343	Prime Movers	30,681,000	30,681,000	49,983,000	49,983,000
344	Generators	9,915,000	9,915,000	9,925,000	9,925,000
345	Acces. Elect. Equipment	4,650,000	4,650,000	4,884,270	4,884,270
346	Misc. Power Plant Equip.	218,000	218,000	218,000	218,000
353	Station Equip.	1,719,000	1,719,000	1,719,000	1,719,000
	Subtotal	$51,228,000	$50,188,000	$71,465,270	$70,219,270
Other Expenses		1,250,000	1,250,000	1,250,000	1,250,000
	Subtotal	$52,478,000	$51,438,000	$72,715,270	$71,469,270
Engineering, Design, Construction, Supervision and Contingency		5,640,000	5,529,585	8,000,000	7,850,000
	Subtotal	$58,118,000	$56,967,585	$80,715,270	$79,319,270
Escalation		3,500,000	3,420,000	5,110,000	5,000,000
	Subtotal	$61,618,000	$60,387,585	$85,825,270	$84,319,270
Interest During Construction		4,925,000	4,820,000	6,860,000	6,740,000
	TOTAL	$66,543,000	$65,207,585	$92,685,270	$91,059,270

TABLE XXVII

POWER GENERATION COSTS FOR ADVANCED POWER SYSTEMS LOCATED IN SOUTH CENTRAL REGION

1970-Decade Designs
70% Load Factor
Outdoor Construction
Gas Cost 23¢/million Btu[3]
1000-Mw Stations

	Steam Turbine	Simple-Cycle Gas Turbine	Regenerative-Cycle Gas Turbine
Operating Conditions	2400 psig/1000 F/1000 F	P_r = 20:1, 2400 F	P_r = 10:1, 2200 F
Number of Units	Two 500-Mw	Five 200-Mw	Five 200-Mw
Station Thermal Efficiency, %	36	30.6	33.9
Net Station Heat Rate, Btu/kwhr	9490	11,160	10,070
Station Installed Cost[1], $/kw	132.3	80	100
Capital Charges[2], mills/kwhr	3.250	1.980	2.450
Operation and Maintenance, mills/kwhr	0.365	0.700	0.800
Fuel, mills/kwhr	2.180	2.561	2.318
Busbar Power Cost, mills/kwhr	5.792	5.241	5.568

(1) 8% interest rate during construction
(2) 15% fixed charges
(3) Gas cost estimates from Table VI

TABLE XXVIII

POWER GENERATION COSTS FOR ADVANCED POWER SYSTEMS LOCATED IN SOUTH CENTRAL REGION

Early 1980's Designs
70% Load Factor
Gas Costs 30¢/million Btu[3]
Outdoor Construction

	Steam Turbine	Simple-Cycle Gas Turbine	Regenerative-Cycle Gas Turbine
Operating Conditions	3500 psig/1000 F/1000 F	P_r = 28:1, 2600 F	P_r = 12:1, 2400 F
Number of Units	One 1000-Mw	Four 250-Mw	Five 200-Mw
Station Thermal Efficiency, %	38	35.8	37.6
Net Station Heat Rate, Btu/kwhr	8955	9530	9080
Station Installed Cost[1], $/kw	120.4	66.5	92.7
Capital Charges[2], mills/kwhr	3.009	1.630	2.270
Operation and Maintenance, mills/kwhr	0.312	0.700	0.800
Fuel, mills/kwhr	2.690	2.860	2.722
Busbar Power Cost, mills/kwhr	6.011	5.190	5.792

(1) 8% interest rate during construction
(2) 15% fixed charges
(3) Gas cost estimates from Table VI

TABLE XXIX

POWER GENERATION COSTS FOR ADVANCED POWER SYSTEMS LOCATED IN SOUTH CENTRAL REGION

Late 1980's Designs
70% Load Factor
Gas Costs 40¢/million Btu[3]
Outdoor Construction

	Steam Turbine	Simple-Cycle Gas Turbine	Regenerative-Cycle Gas Turbine
Operating Conditions	3500 psig/1000 F/1000 F	P_r = 30:1, 2800 F	P_r = 14:1, 2600 F
Number of Units	One 1000-Mw	Four 250-Mw	Five 200-Mw
Station Thermal Efficiency, %	39.1	38.7	40.6
Net Station Heat Rate, Btu/kwhr	8740	8820	8410
Station Installed Cost[1], $/kw	120.4	71.0	93.0
Capital Charges[2]	3.009	1.742	2.228
Operation and Maintenance, mills/kwhr	0.312	0.700	0.800
Fuel, mills/kwhr	3.490	3.525	3.365
Busbar Power Cost, mills/kwhr	6.811	5.967	6.393

(1) 8% interest rate during construction
(2) 15% fixed charges
(3) Gas cost estimates from Table VI

TABLE XXX

PROPOSED ASTM SPECIFICATIONS FOR GAS TURBINE FUELS

Designation (a)	Grade of Gas Turbine Fuel Oil	Flash Point, deg F (deg C)	Pour Point deg F (deg C)	Water and Sediment, Percent by Volume	Carbon Residue on 10% Residuum, Percent	Ash, Percent by Weight	Distillation Temperature 90% Point, deg F (deg C)		Saybolt Viscosity, sec(b) Universal at 100 F (38 C)		Saybolt Viscosity, sec(b) Furol at 122 F (50 C)		Kinematic Viscosity, Centistokes(b) at 100 F (38 C)		Kinematic Viscosity, Centistokes(b) at 122 F (50C)	Gravity deg API	Vanadium (V), ppm by Weight	Sodium plus Potassium (N_a + K), ppm by Weight	Calcium (Ca), ppm by Weight	Lead (Pb), ppm by Weight	Magnesium-to Vanadium Weight Ratio (Mg/V)	
		Min	Max	Max	Max	Max	Min	Max	Min	Max	Min	Max	Min	Max	Max	Min	Max	Max	Max	Max	Min	Max
No. 1-GT(c)	A volatile distillate for gas turbines requiring a more clean burning fuel than No. 2-GT.	100(38) or legal	0 (-18) (d)	0.05	0.15	0.01	--	550 (200)	--	(34.4)	--	--	1.4	2.5	--	35	2	5	5	5	--	--
No. 2-GT	A distillate fuel of low ash and medium volatility suitable for gas turbines not requiring No. 1-GT.	100(38) or legal	20 (-7) (d)	0.10	0.35	0.01	540 (252)	675 (357)	(>2.6)	(45)	--	--	2.0	5.8	--	30	2	5	10	5	--	--
No. 3-GT(e)	A low volatility, low ash fuel that may contain residual components.	130(54) or legal	--	1.0	--	0.03	--	--	--	--	45	300	(5.8)	--	(638)	--	2 (e)	5 (e)	10 (e)	5	--	--
No. 4-GT	A low volatility fuel containing residual components and having higher vanadium content than No. 3-GT.	150(66) or legal	--	1.0	--	--	--	--	--	--	45	300	(5.8)	--	638)	--	500	10 (f)	10 (f)	5	3.0 (g)	3.5 (g)

Notes:

(a) No. 1-GT: Corresponds in general to ASTM D396, Grade No. 1 fuel and ASTM D975 Grade No. 1-D diesel fuel in physical properties.
No. 2-GT: Corresponds in general to ASTM D396, Grade No. 2 fuel and ASTM D975 Grade No. 2-D diesel fuel in physical properties.
No. 3-GT and
No. 4-GT: Viscosity range brackets ASTM D396, Grades No. 4, No. 5 (light), No. 5 (heavy) and No. 6 and ASTM D975, Grade No. 4-D, diesel fuel, which may be supplied provided metals composition requirements are met.

(b) Viscosity values in parentheses are for information only and are not limiting.

(c) Recognizing the necessity of additional requirements for certain types of gas turbines, the following may be specified for No. 1-GT fuel:
Luminometer number, min = 40
Sulfur, percent by weight, max = 1.0
Thermal stability test for 5 hours at 250 F (121 C) preheater temperature, 350 F (177 C) filter temperature and at a flow rate of 6 lb (2.7 kg) per hour:
Filter pressure drop, max = 12 in.of Hg (30 cm of Hg)
Preheater deposit code, max = 2.

(d) Lower or higher pour points may be specified whenever required by conditions for storage or use. When a pour point less than 10 F (-18 C) is specified for Grade No. 2-GT, the minimum viscosity shall be 1.5 cs (32.0 sec, Saybolt Universal) and the minimum 90 percent a point shall be waived.

(e) For gas turbines operating below 1200 F (640 C) maximum gas temperature, the limitations on vanadium, sodium plus potassium, and calcium may be waived, provided that a silicon-base additive, or equivalent, is employed. The special requirements covering the addition of and the type of additive shall be specified only by mutual agreement between purchase and seller.

(f) Where water washing facilities are available at the point of use, these requirements may be waived by mutual agreement between the purchaser and seller.

(g) Special requirements covering the addition of and the type of magnesium-base additive,or equivalent, to be used shall be specified only by mutual agreement between purchaser and seller.

TABLE XXXI

REPRESENTATIVE COAL GASIFICATION PROCESSES SURVEYED

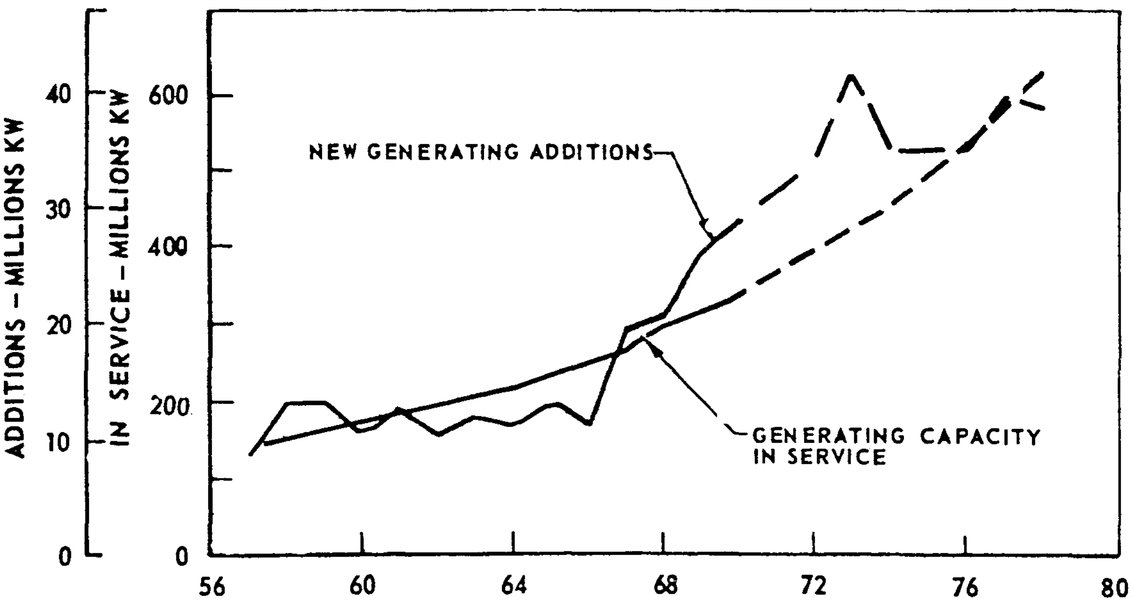
Process Name	Description	Status	Press. Atm.	Gasifier Outlet Temp., F	Btu of Gas (HHV) / Btu of Coal (HHV)	Sensible Heat of Gas / Btu of Coal (HHV)	Gasification Rate lb/hr-ft^2	Gasification Rate lb/hr-ft^3
Wellman - Galusha	Autothermal, fixed bed, counter-current flow	Commercial	1	1250	0.82	0.13	112.4	—
Power Gas	Autothermal, fixed bed, counter-current flow	Commercial	1	1250	0.85	0.13	63	—
Bamag - Winkler	Autothermal, fluidized bed	Commercial	1	1600	0.66	0.10	170	14
Ruhrgas Vortex	Autothermal, cocurrent, up flow, slagging	Commercial	1	1900	0.66	0.27	—	1.3
Lurgi Dry Ash	Autothermal, fixed bed, counter-current flow	Commercial	20	950	0.82	0.09	300	—
Koppers Totzek	Autothermal, cocurrent, tangential flow, slagging	Commercial	1	2000	0.67	0.19	--	16
B&W - DuPont	Autothermal, cocurrent, up flow, slagging	Commercial	1	2200	0.68	0.20	--	24
Rummel Single Shaft	Autothermal, cocurrent, up flow, slagging	Commercial	1	1800	0.79	0.15	--	21
Texaco Partial Oxidation	Autothermal, cocurrent, down flow, slagging	Pilot	14	2200	0.72	0.19	—	57
USBM Pressure Gasifier	Autothermal, cocurrent, down flow, slagging	Pilot	40	1740	0.64	0.11	—	430
BCR 2-Stage Gasifier	Autothermal, cocurrent, up flow, slagging	Pilot	68	1700	0.74	0.07	—	58
IGT Pressure Gasifier	External Heating, cocurrent, down flow, slagging	Pilot	68	1500	0.83	--	—	40
Kellogg Molten Salt	External Heating, salt bath	Pilot	28	1800	0.82	0.08	—	40
Consol CO_2 Acceptor	Fluidized bed, cocurrent flow	Pilot	1	1600	0.60	0.03	—	—

FROM REFS. 2,4

ESTIMATED NEW ADDITIONS BY TYPE OVER 1969–78 PERIOD

FOSSIL STEAM	58.0%
NUCLEAR	28.2
GAS TURBINE	5.3
CONVENTIONAL HYDRO	3.1
PUMPED STORAGE HYDRO	5.4
TOTAL	100%

(a) NEW GENERATING ADDITIONS AND YEAR–END CAPACITY IN SERVICE



(b) RATIO OF YEAR–END CAPACITY TO SUMMER PEAK LOAD

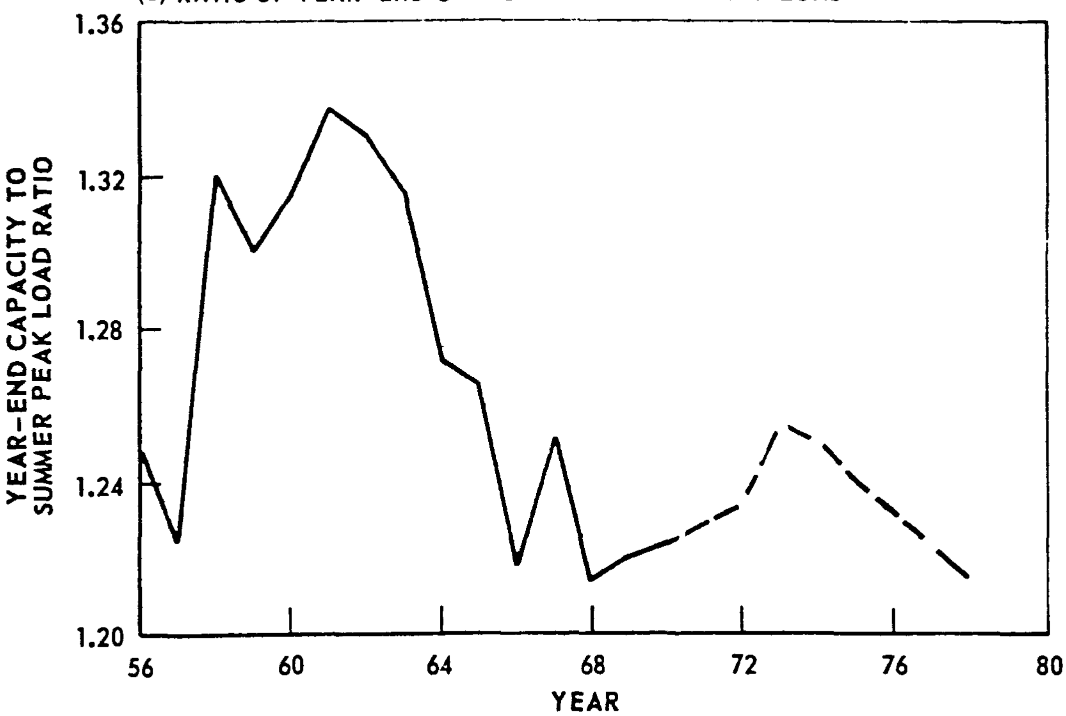


FIG. 1. YEARLY ADDITIONS TO GENERATING CAPACITY AND YEAR–END MARGINS IN ELECTRIC UTILITY INDUSTRY

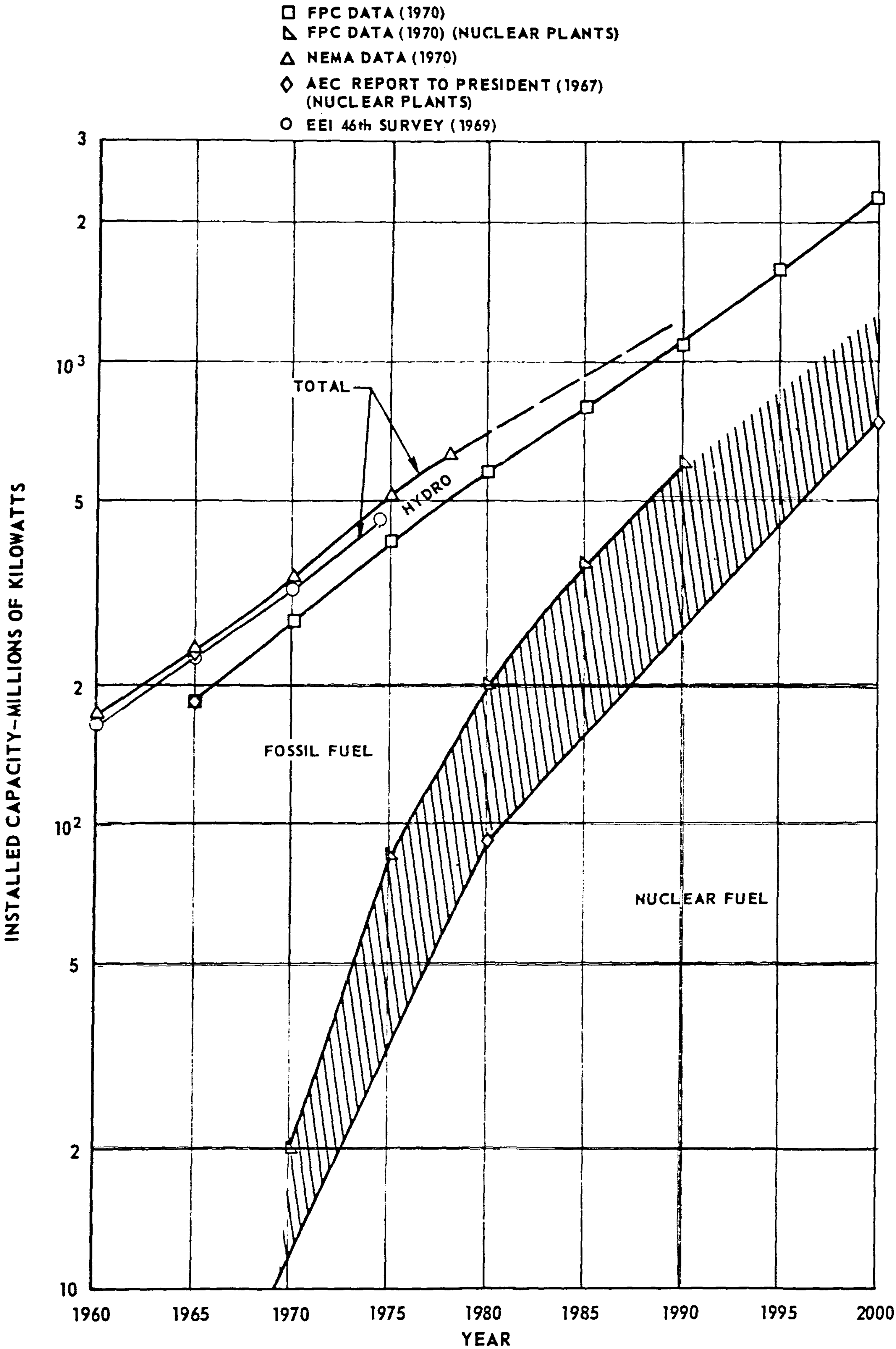


FIG. 2. PREDICTED GROWTH OF ELECTRIC UTILITY GENERATION CAPACITY

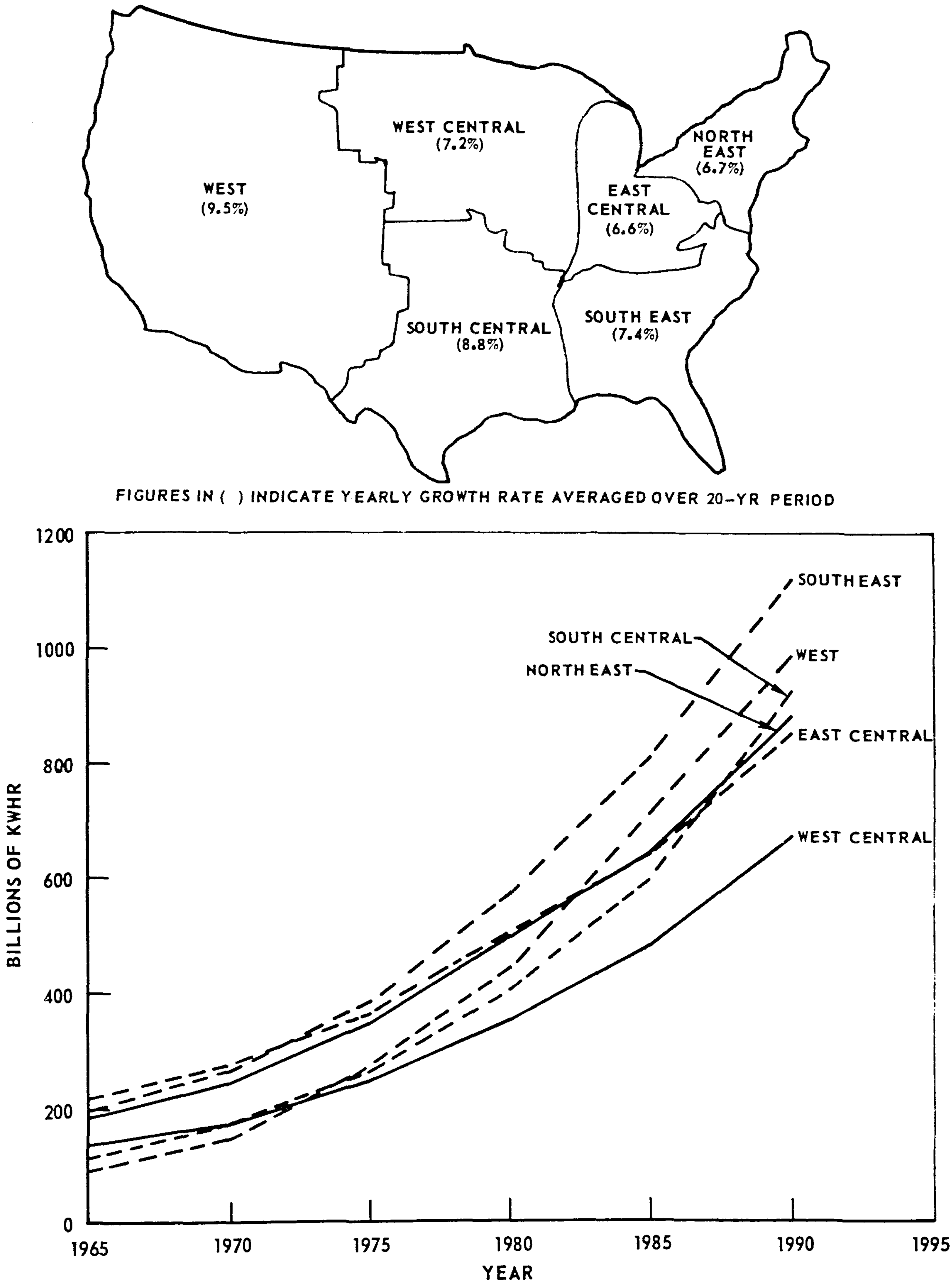
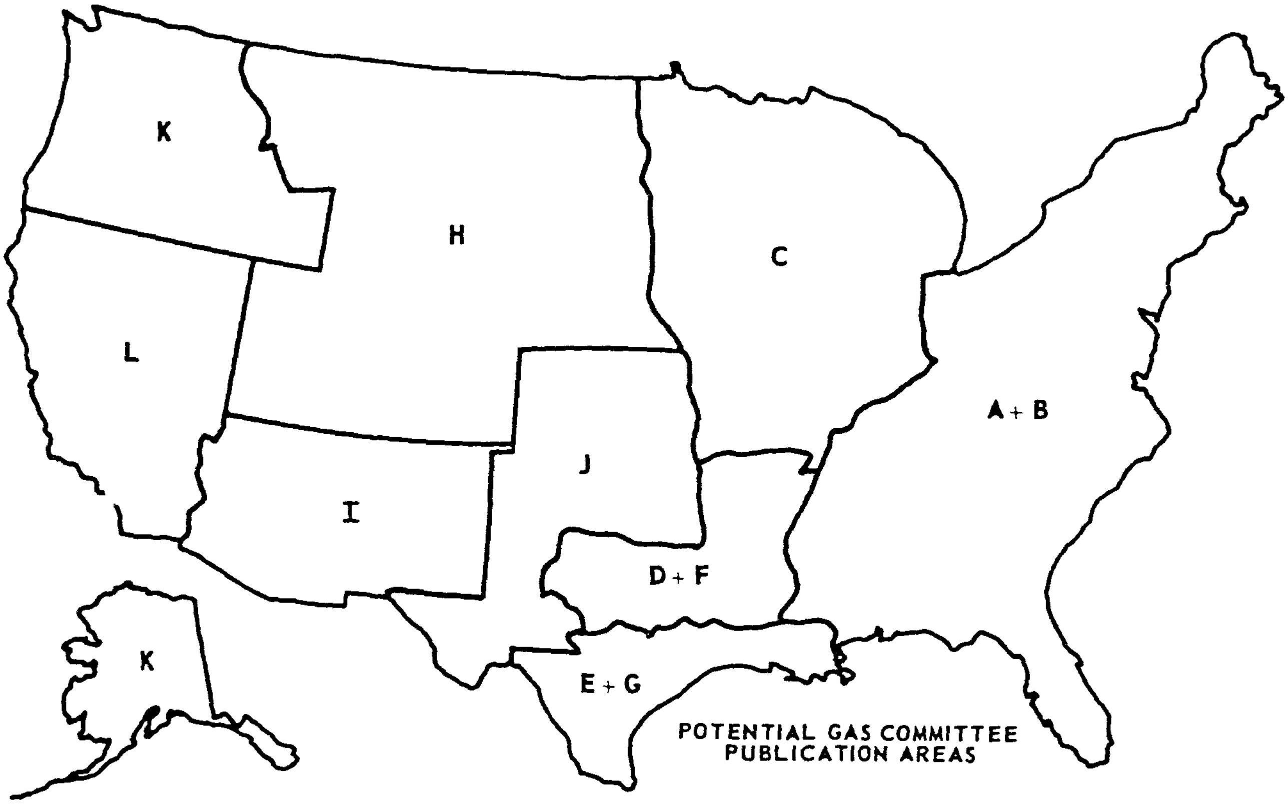


FIG. 3. REGIONAL FORECAST OF ELECTRICAL GENERATION IN THERMAL PLANTS



ESTIMATED PROVEN AND POTENTIAL SUPPLY OF NATURAL GAS IN
UNITED STATES AS OF DECEMBER 31, 1968

Trillion Cubic Feet @ 14.73 psia and 60°F

Area		Proven	Potential			Total
			Probable	Possible	Speculative	
A+B		8	44	20	57	129
C		1	2	1	2	6
D+F		18	18	32	70	138
E+G	Onshore	154	80	55	9	440
	Offshore		33	90	19	
H		9	14	26	21	70
I		15	12	3	13	43
J		70	29	75	21	195
K		5	22	18	392	437
L		7	6	15	28	56
Total		287	260	335	632	1514

FIG. 4. LOCATION OF NATURAL GAS RESERVES

FROM REF. 1

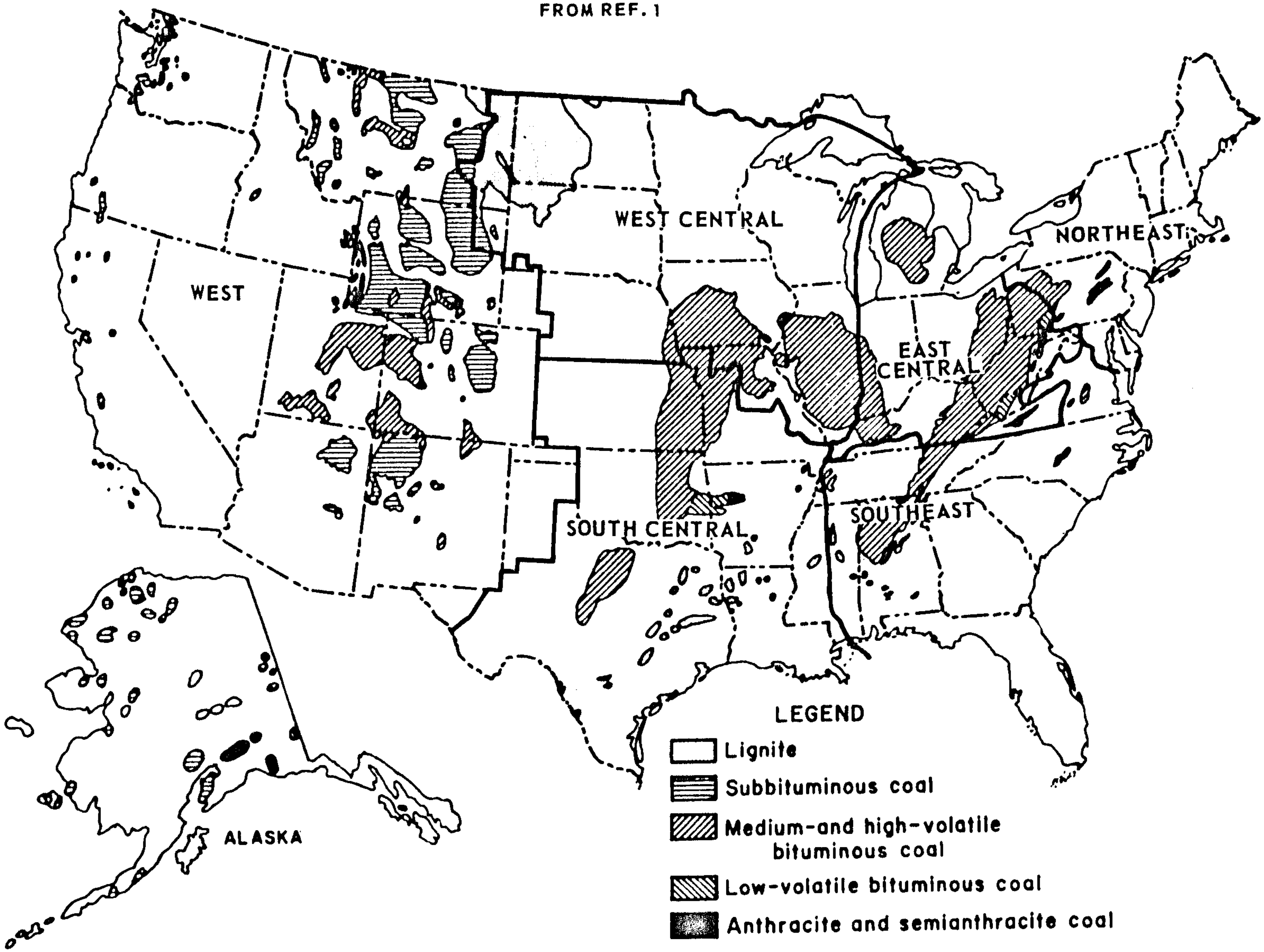
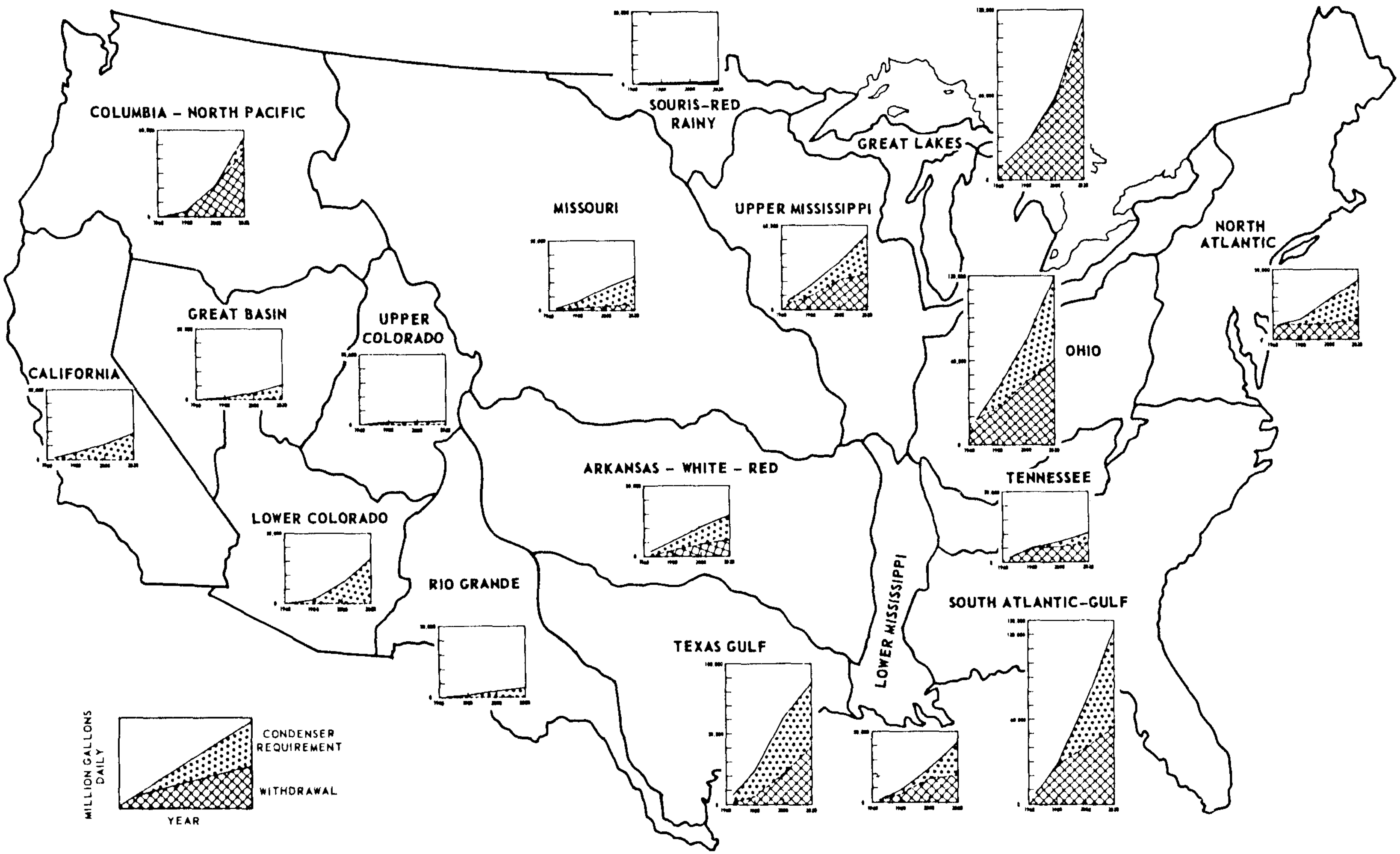


FIG. 5. COAL FIELDS OF THE UNITED STATES

FROM REF. 34



WHEN CONDENSER REQUIREMENT EXCEEDS WITHDRAWAL, FRESH-WATER SUPPLIES FOR ONCE-THROUGH COOLING ARE INADEQUATE

FIG. 6. PROJECTIONS OF REGIONAL FRESH WATER SUPPLIES FOR ONCE-THROUGH CONDENSER COOLING

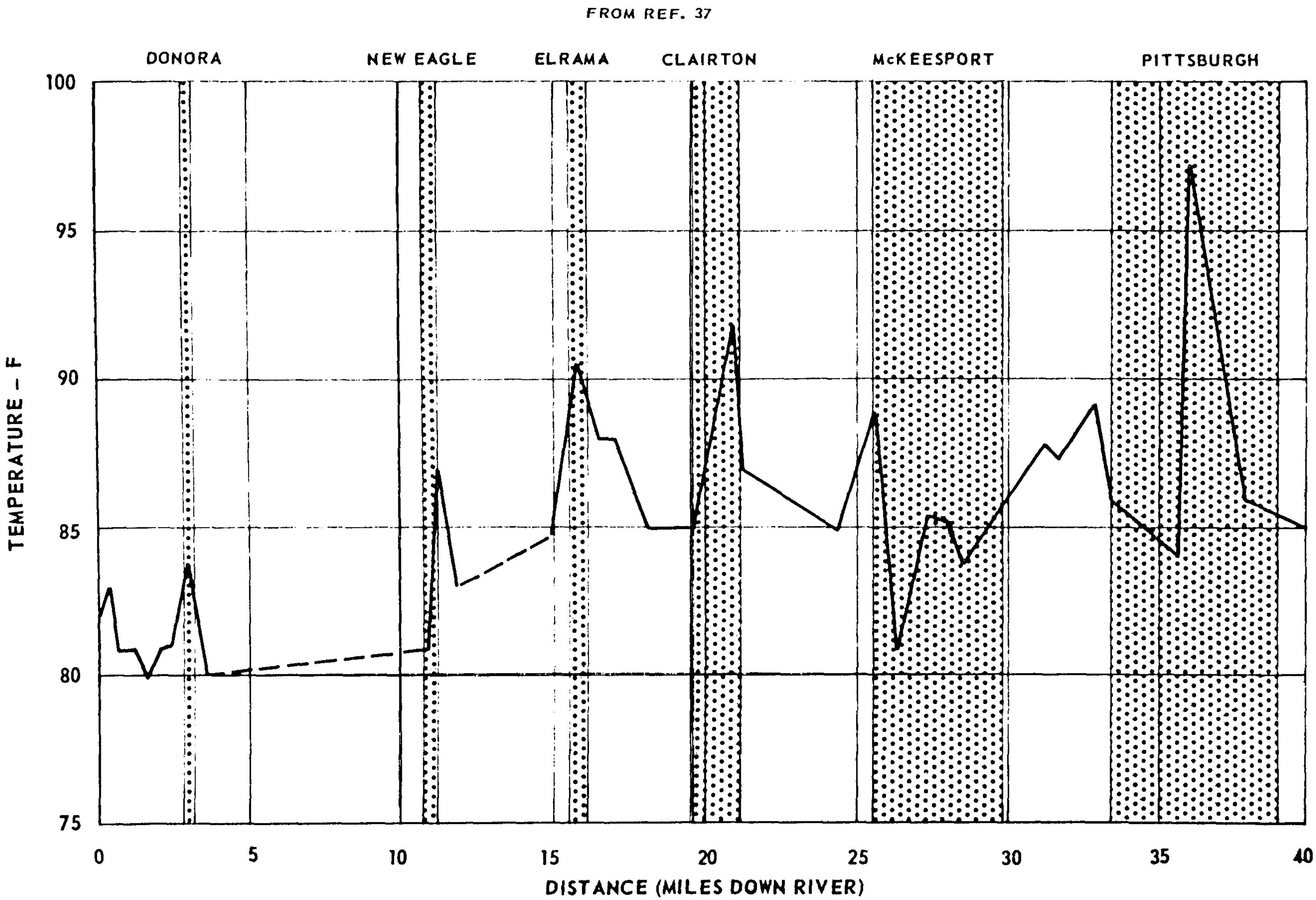


FIG. 7. TYPICAL TEMPERATURE VARIATIONS ALONG MONONGAHELA RIVER DUE TO HEAT REJECTION FROM VARIOUS SOURCES

FROM REF. 42

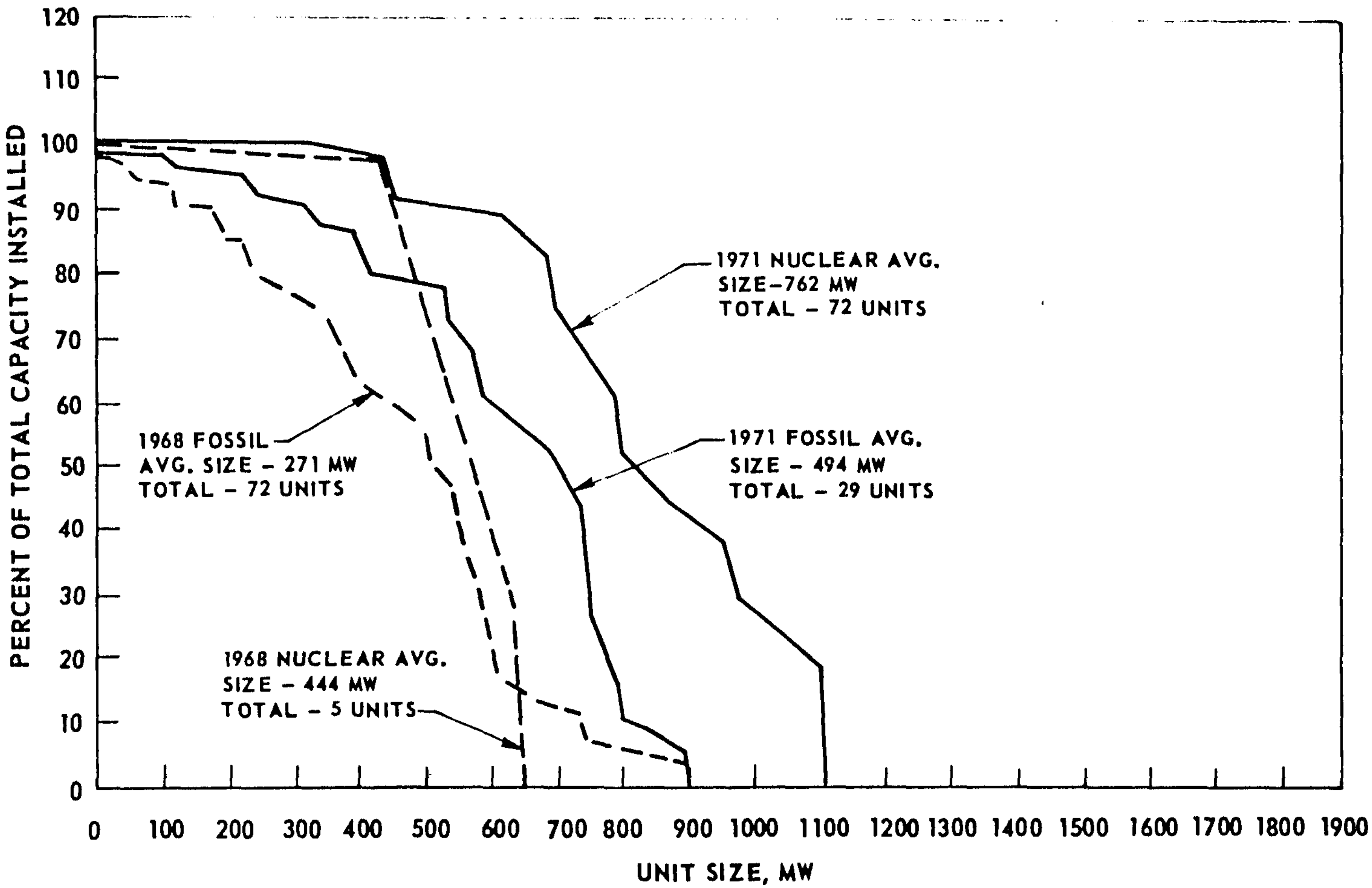


FIG. 8. DISTRIBUTION OF UNIT SIZE FOR 1968–1971 NUCLEAR AND FOSSIL STEAM INSTALLATIONS

169

NOMINAL 500–MW CAPACITY

FROM REF. 45

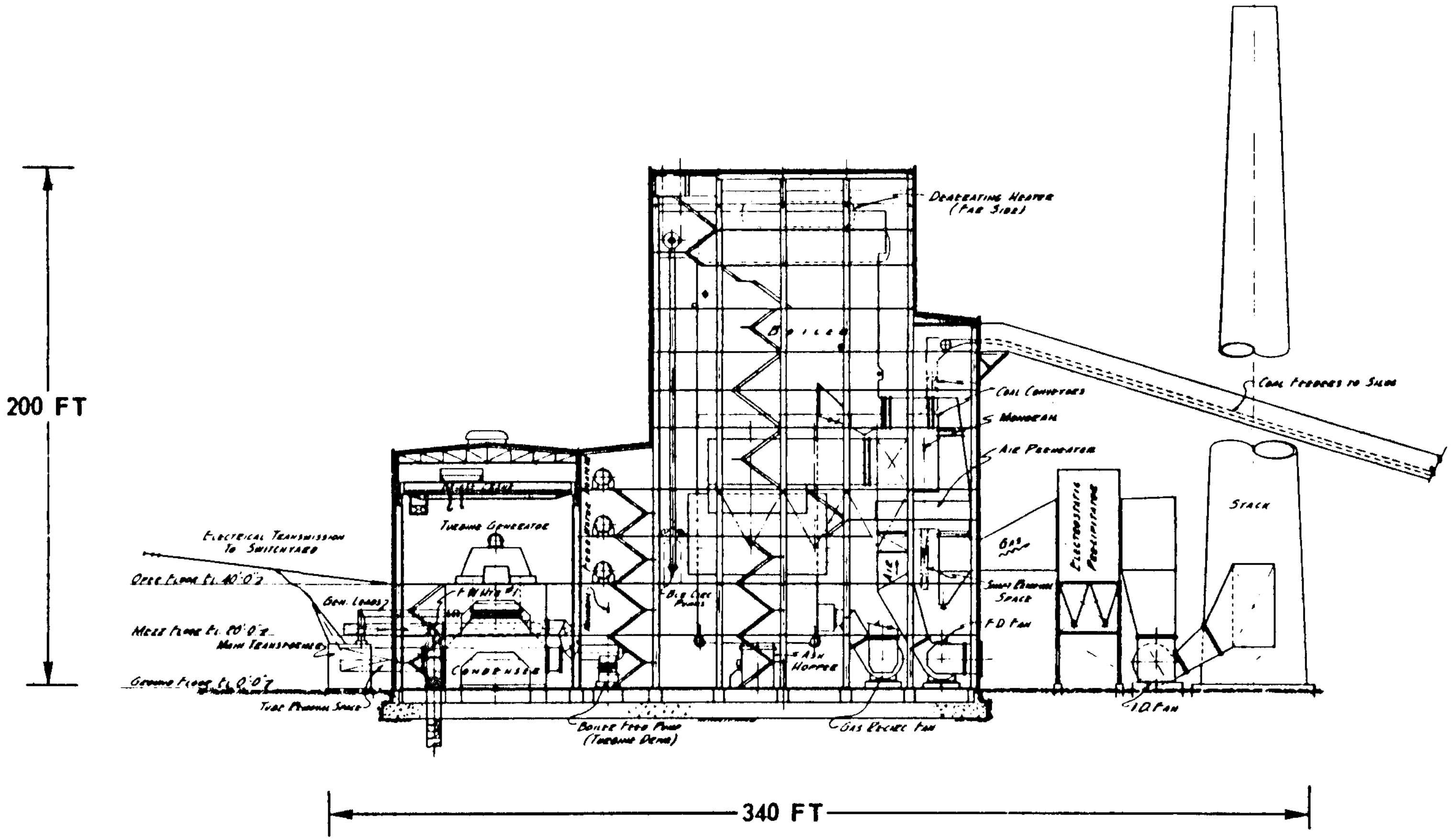


FIG. 9. ELEVATION VIEW OF TYPICAL STEAM POWER STATION

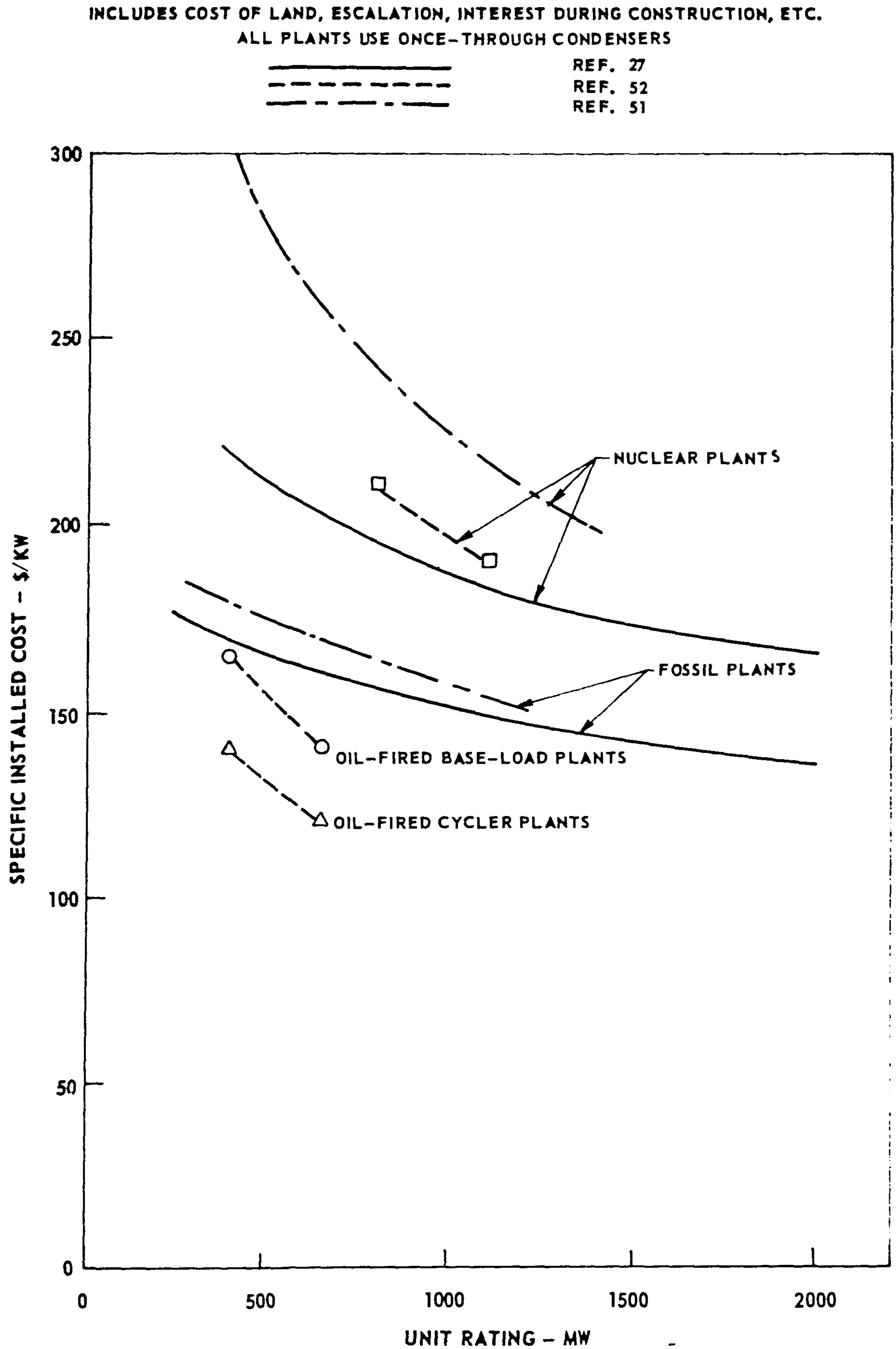


FIG. 10. TYPICAL INSTALLED COSTS OF STEAM POWER PLANTS

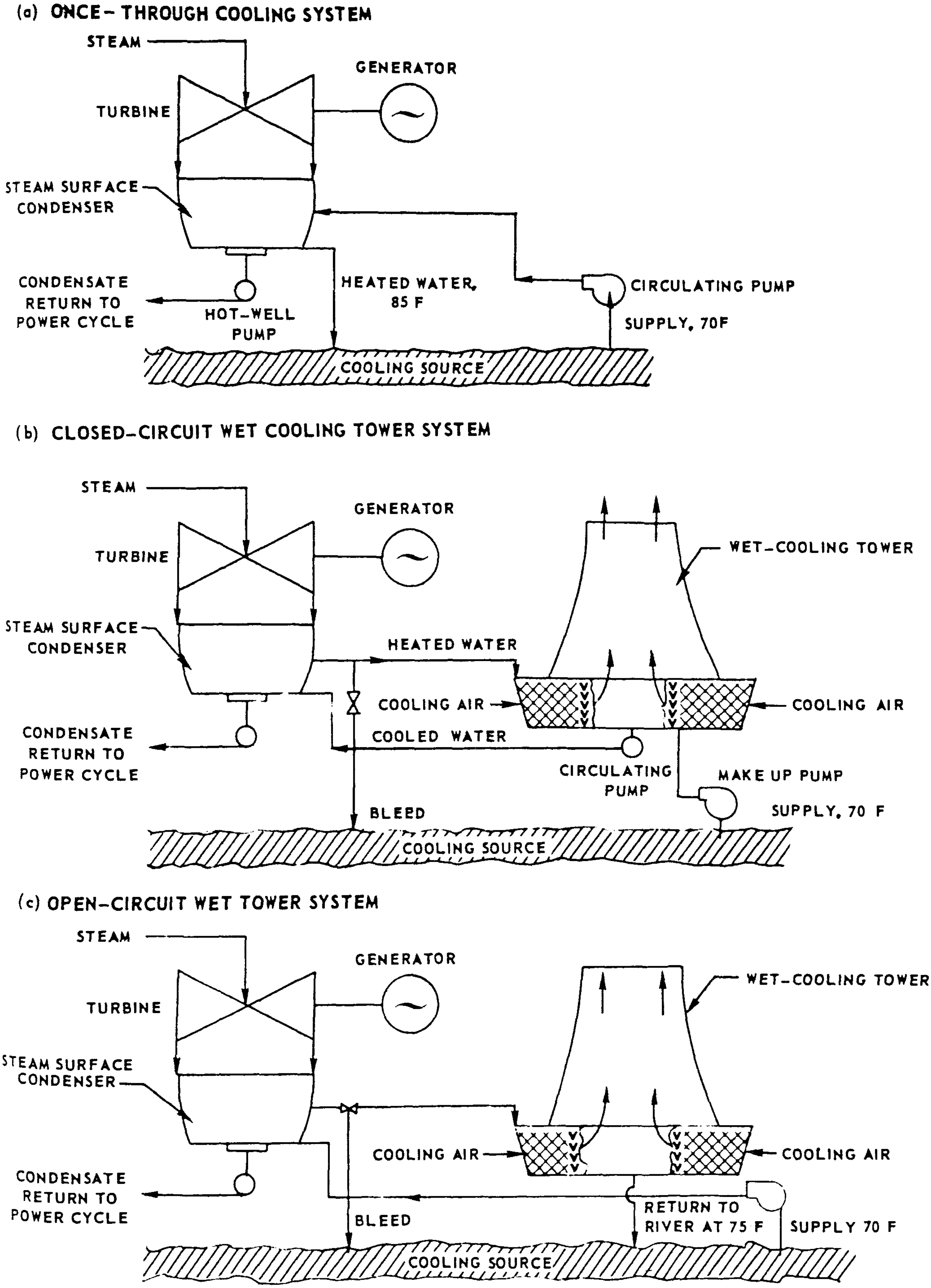
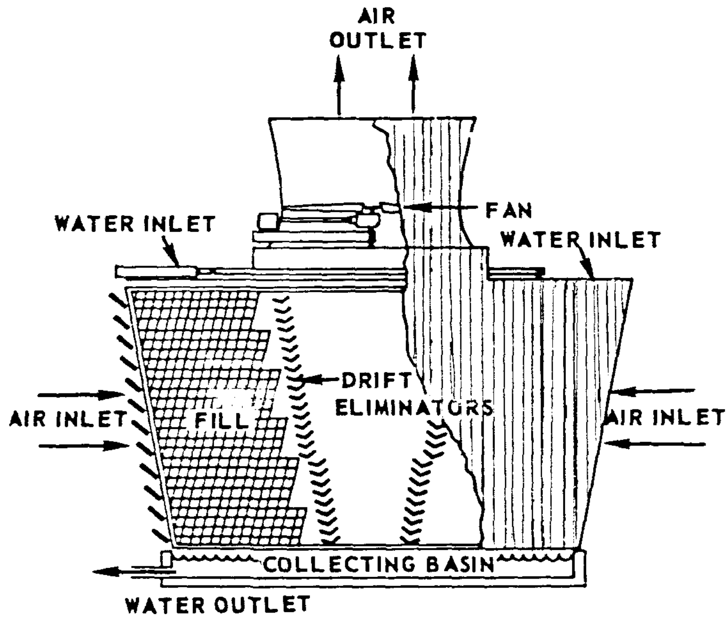


FIG. 11. SCHEMATIC DIAGRAMS OF ALTERNATIVE CONDENSER COOLING METHODS

(a) CROSS-FLOW MECHANICAL-DRAFT TOWER



(b) HYPERBOLIC NATURAL-DRAFT TOWER

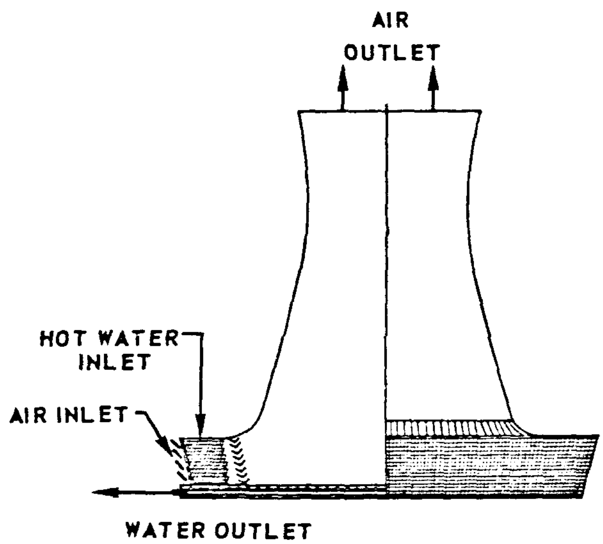


FIG. 12. TYPES OF WET COOLING TOWERS

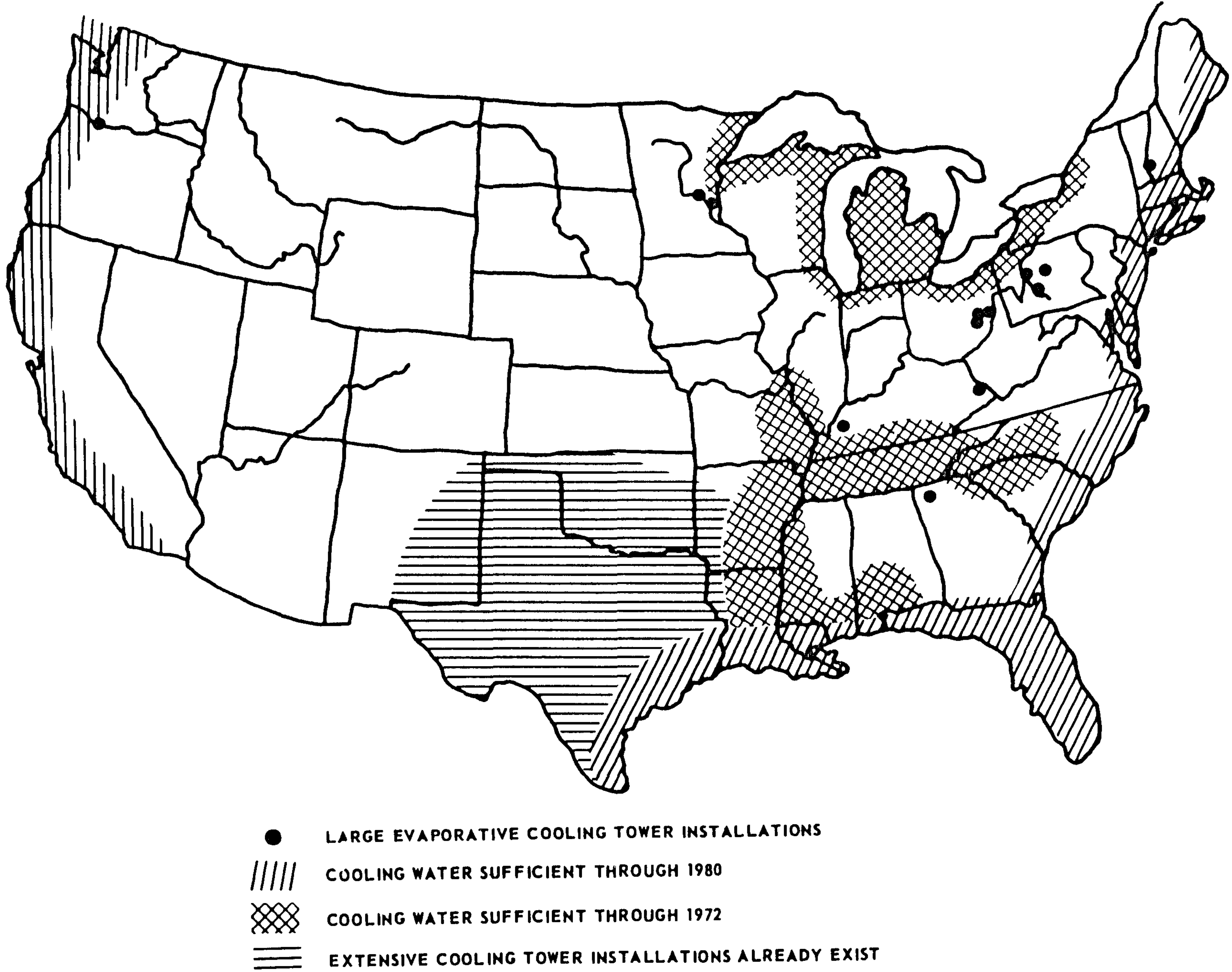


FIG. 13. GEOGRAPHICAL AREAS OF COOLING WATER SUFFICIENCY

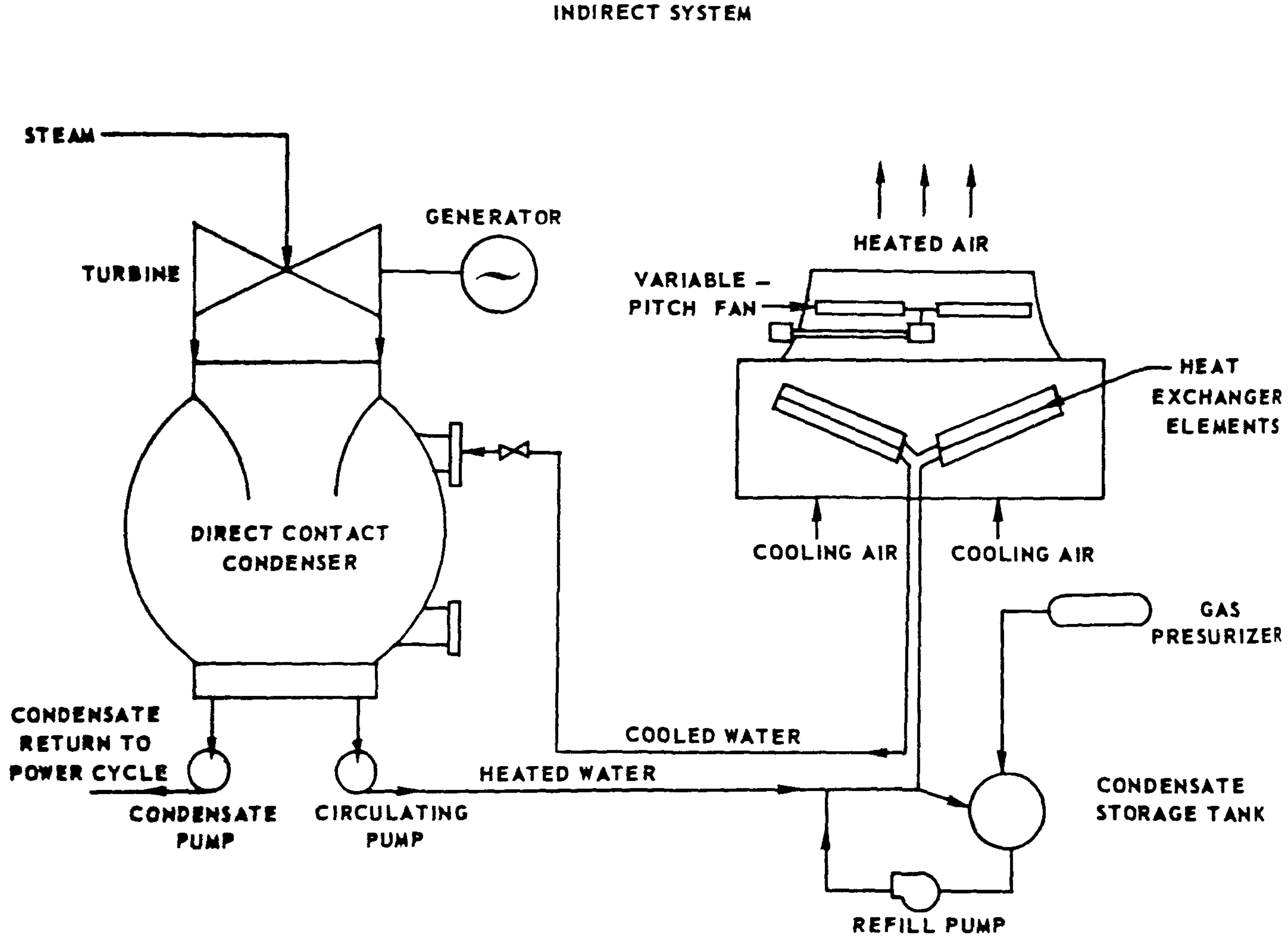


FIG. 14. TYPICAL MECHANICAL-DRAFT DRY COOLING TOWER SYSTEM

DATA FROM REF. 64

BASED ON ESTIMATE OF MANUFACTURER FOR CONVENTIONAL TURBINE MODIFIED FOR OPERATION AT HIGH EXHAUST PRESSURE

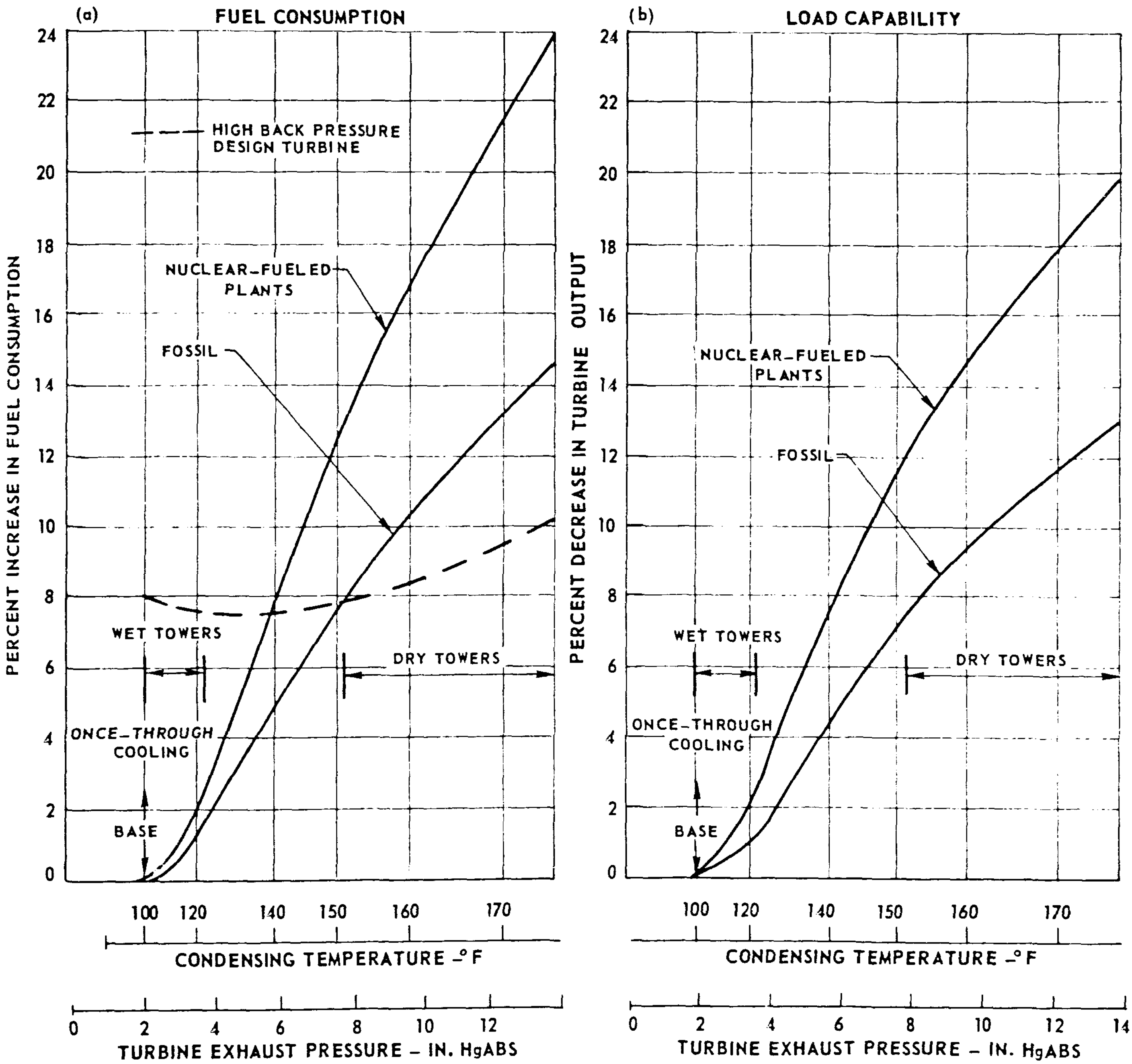


FIG 15. ESTIMATED EFFECT OF CONDENSER BACK PRESSURE ON STEAM PLANT PERFORMANCE

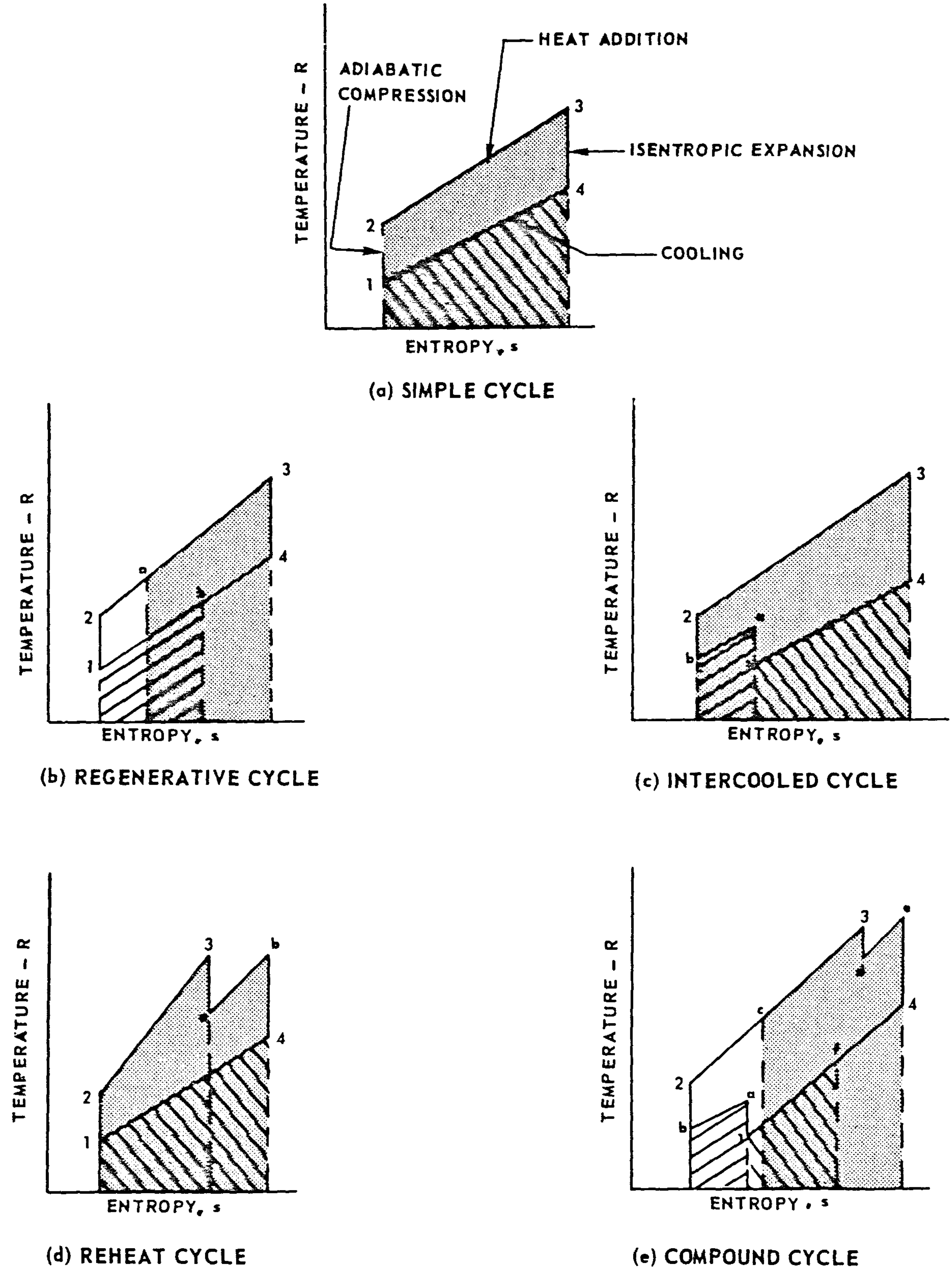


FIG. 16. DIAGRAMS FOR SELECTED GAS TURBINE CYCLES

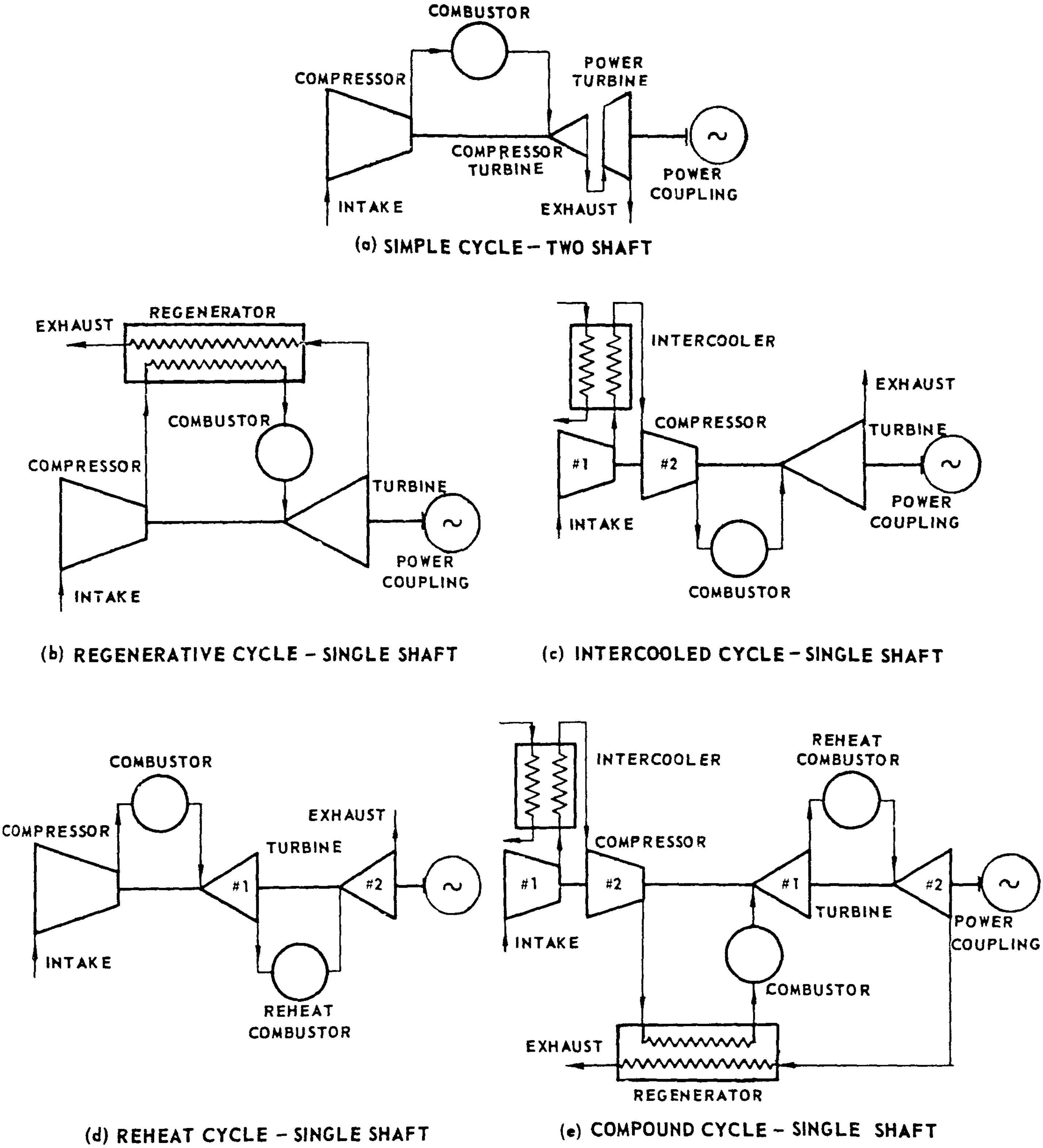


FIG. 17 SCHEMATIC DRAWINGS OF TYPICAL GAS TURBINE ENGINES

AMBIENT TEMPERATURE – 60 F

TURBINE INLET TEMPERATURE – 1500 F

CYCLE PRESSURE LOSSES – ZERO

COMPRESSOR EFFICIENCY–85%

TURBINE EFFICIENCY–85%

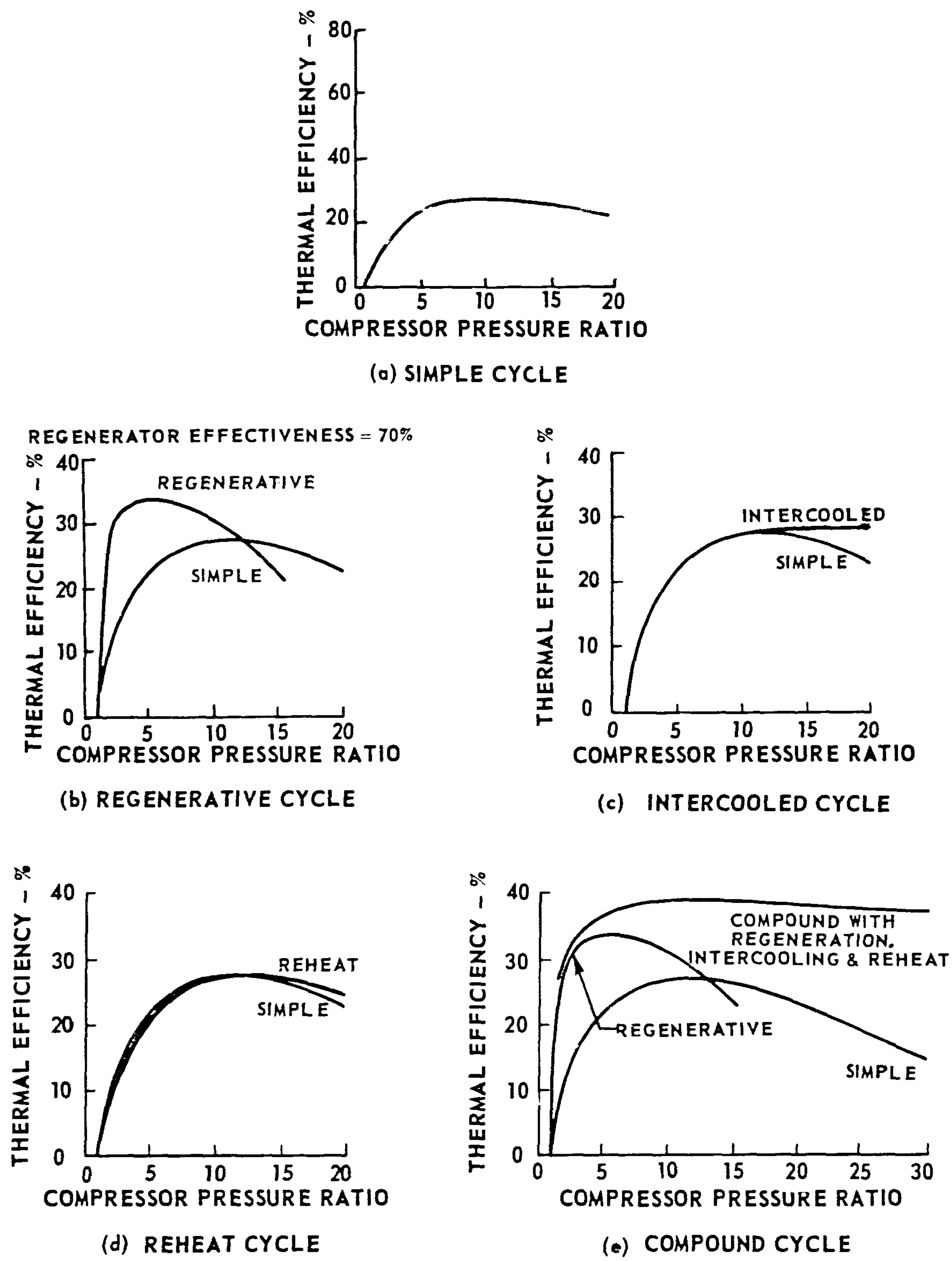


FIG. 18. THEORETICAL PERFORMANCE FOR MODIFIED GAS TURBINE CYCLES

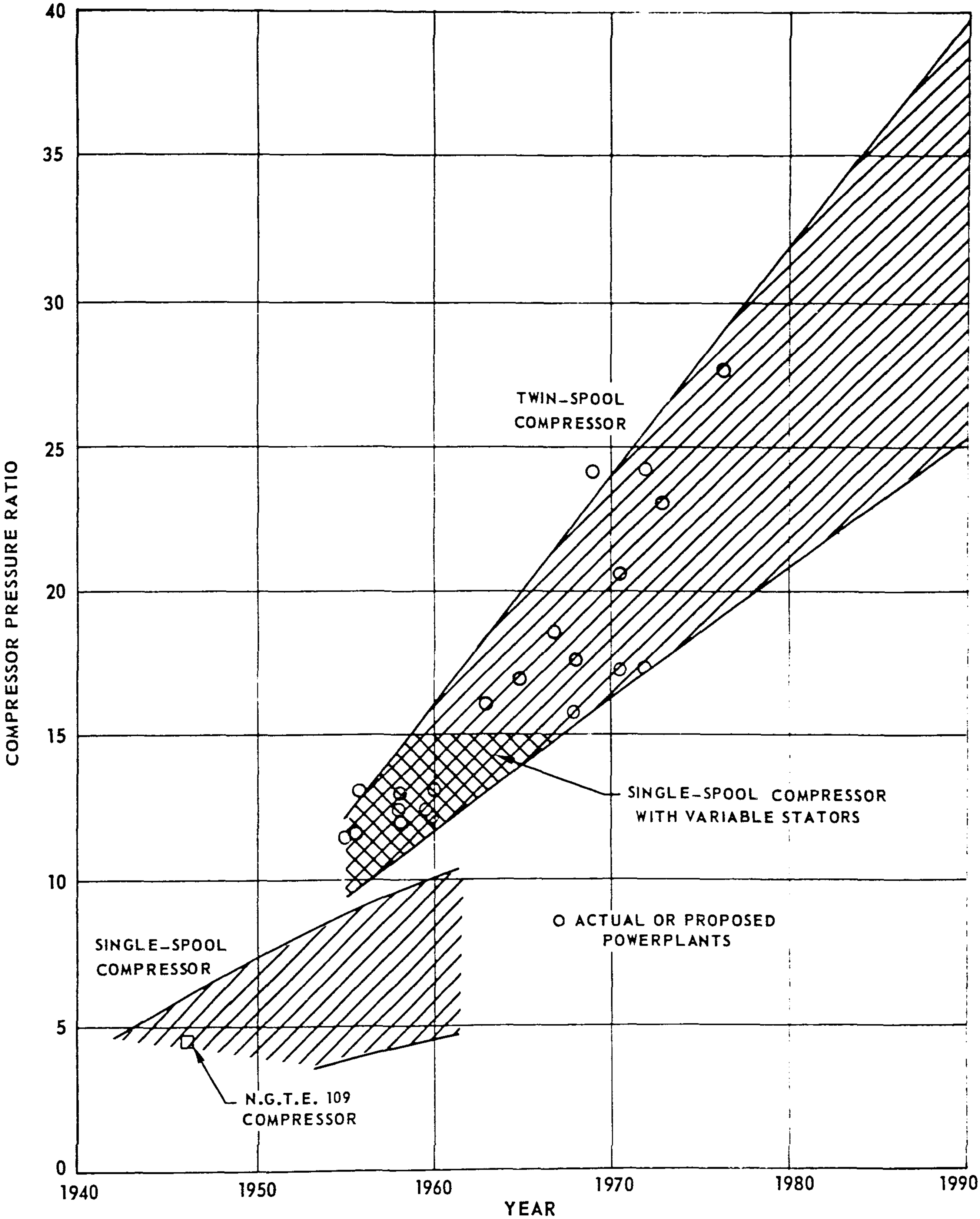


FIG. 19. PROGRESSION OF AIRCRAFT COMPRESSOR TECHNOLOGY

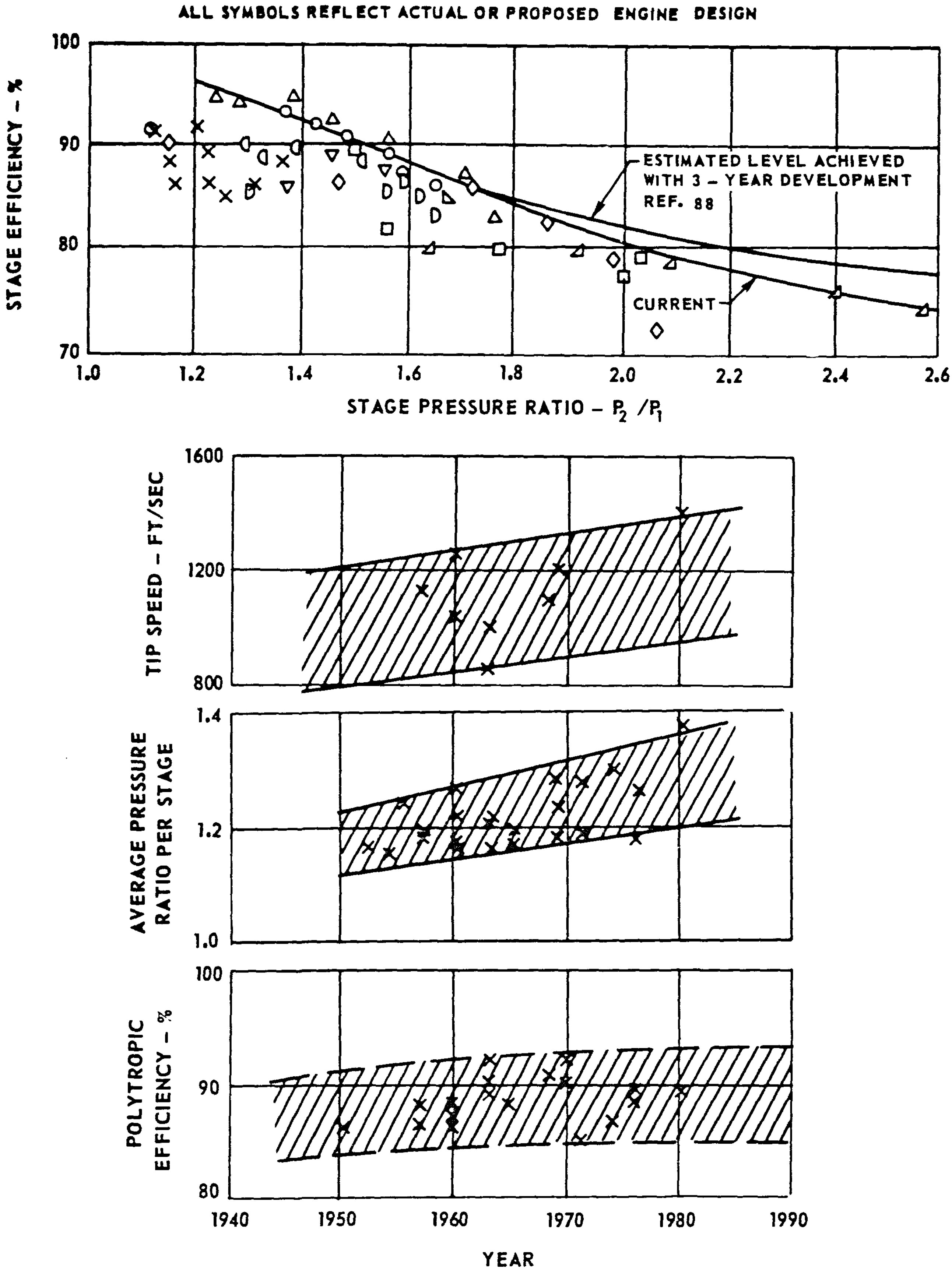


FIG. 20. ADVANCES IN COMPRESSOR PERFORMANCE PARAMETERS

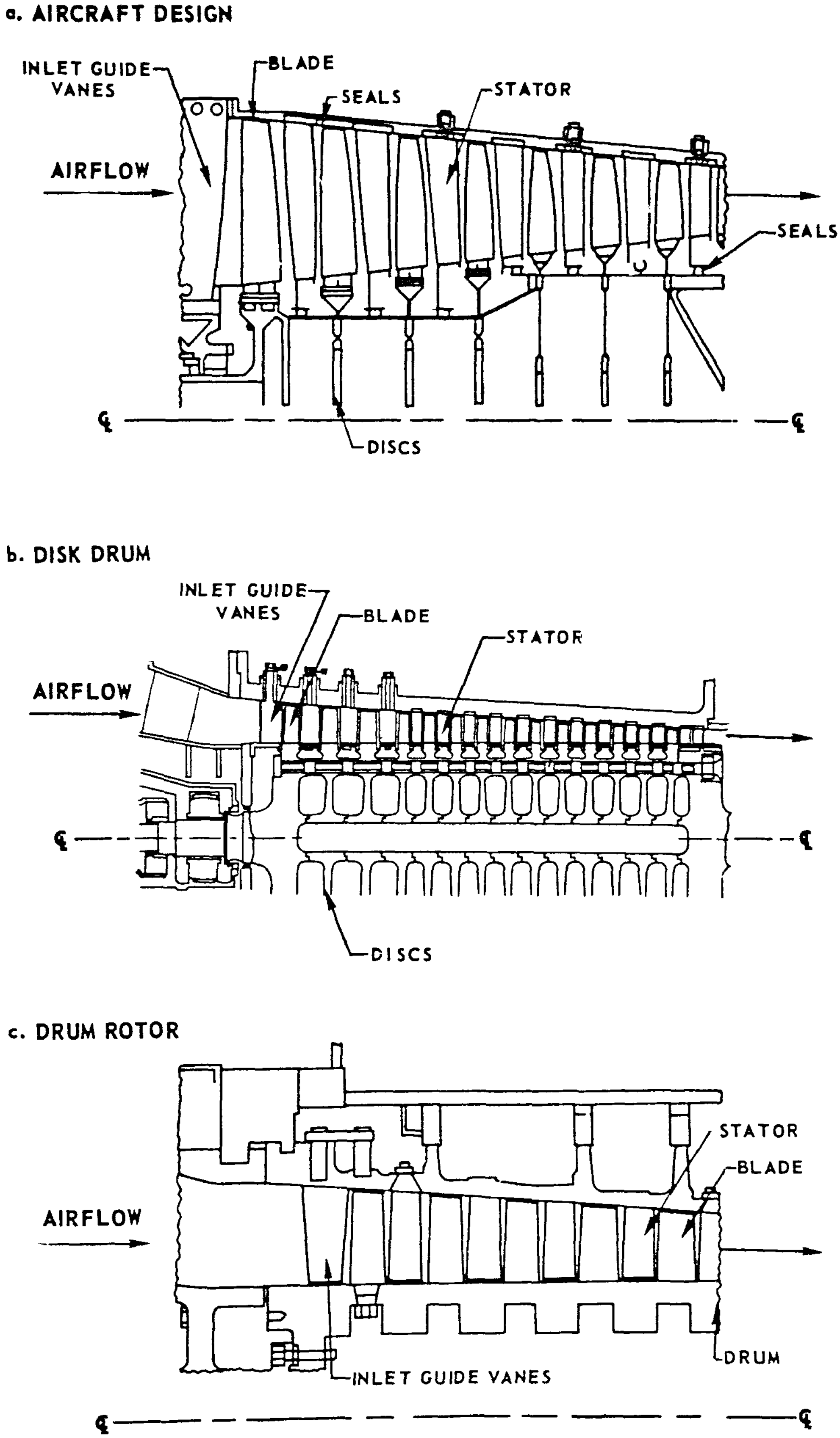


FIG. 21. COMPRESSOR CONSTRUCTION TECHNIQUES

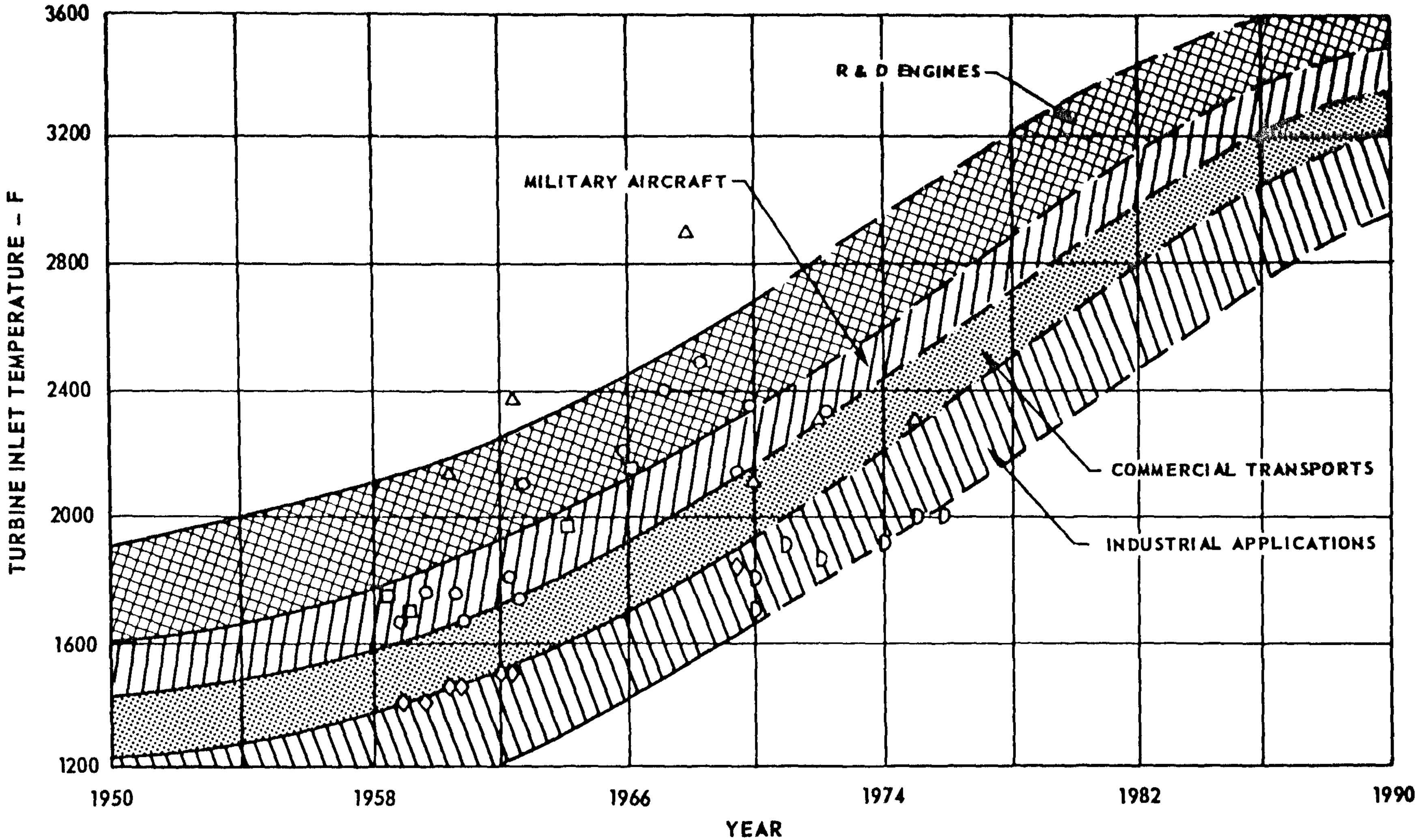


FIG. 22. ESTIMATED TURBINE INLET TEMPERATURE PROGRESSION

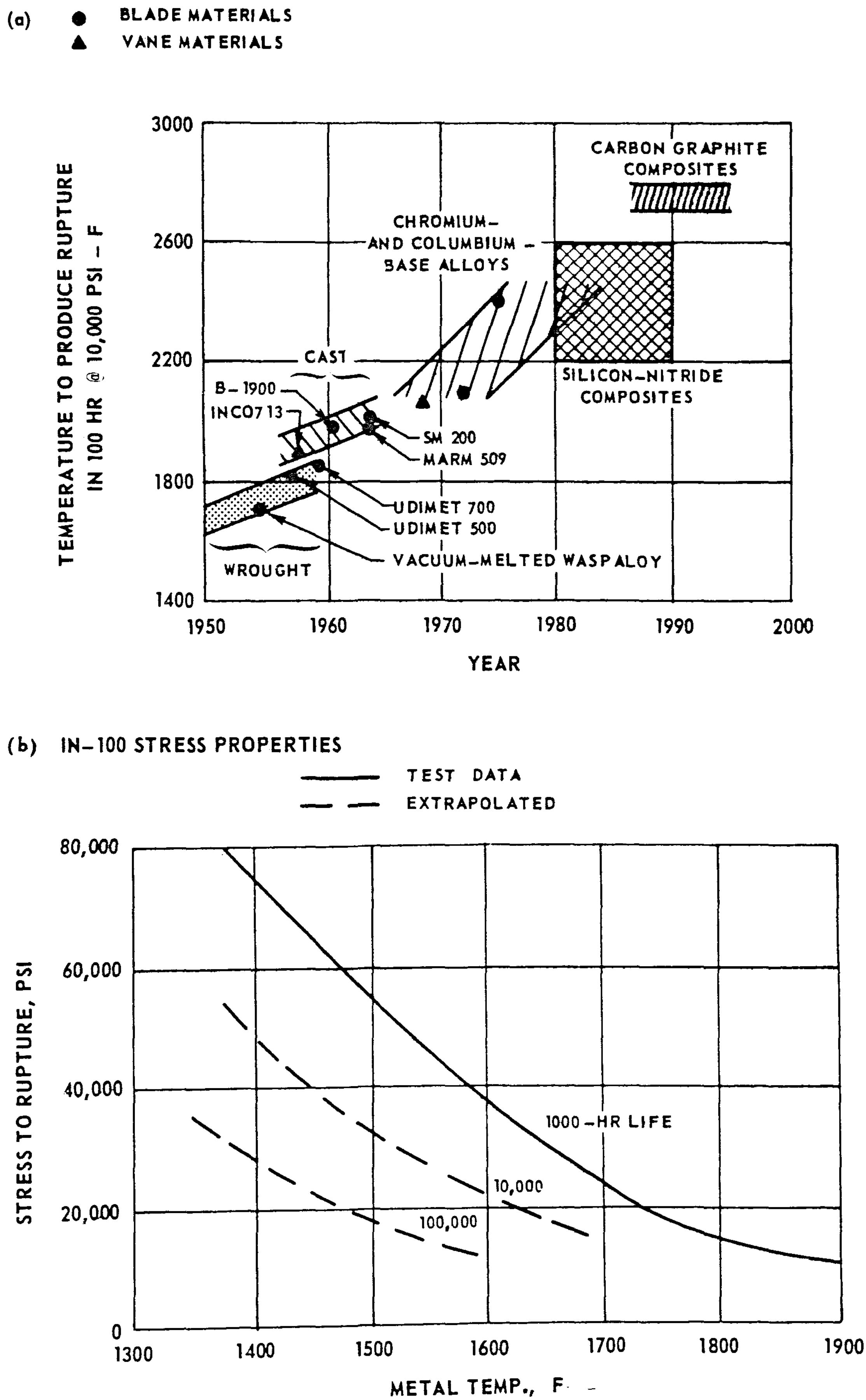


FIG. 23. ADVANCES IN TURBINE BLADE MATERIALS

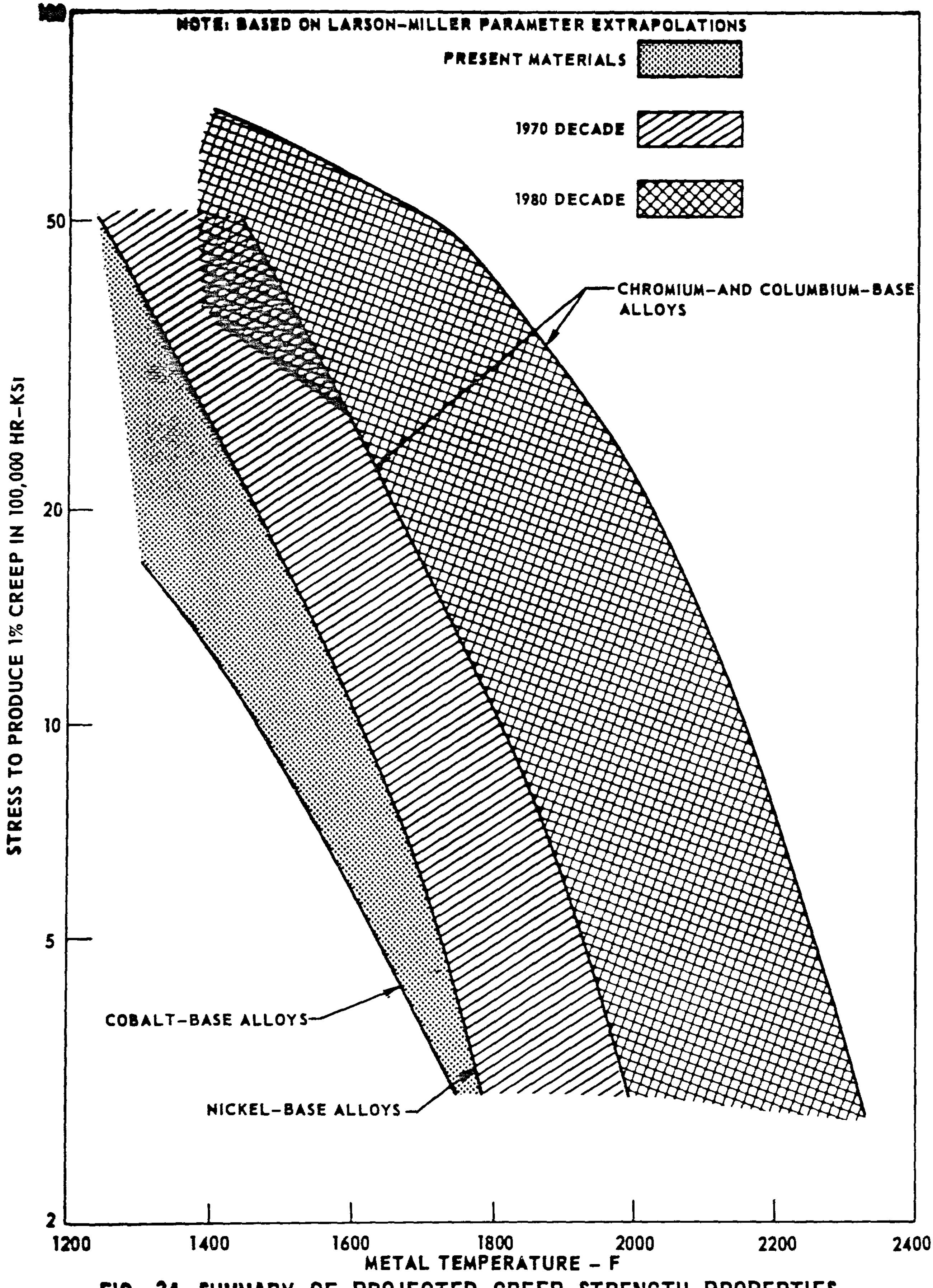
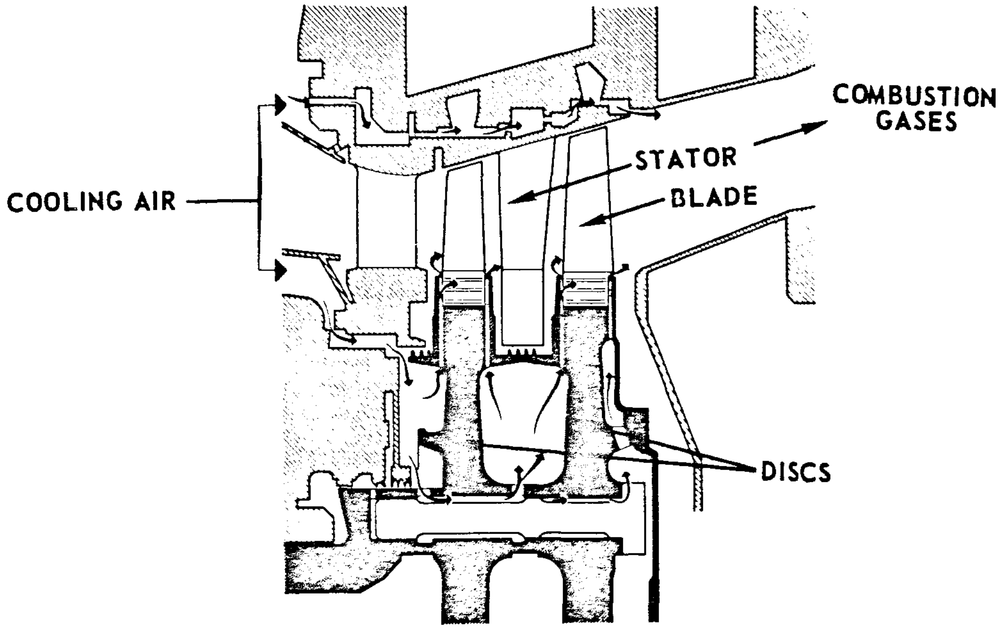


FIG. 24. SUMMARY OF PROJECTED CREEP STRENGTH PROPERTIES FOR ADVANCED TURBINE BLADE MATERIALS

(a) DISC COOLING CONFIGURATION



(b) BLADE AND VANE COOLING

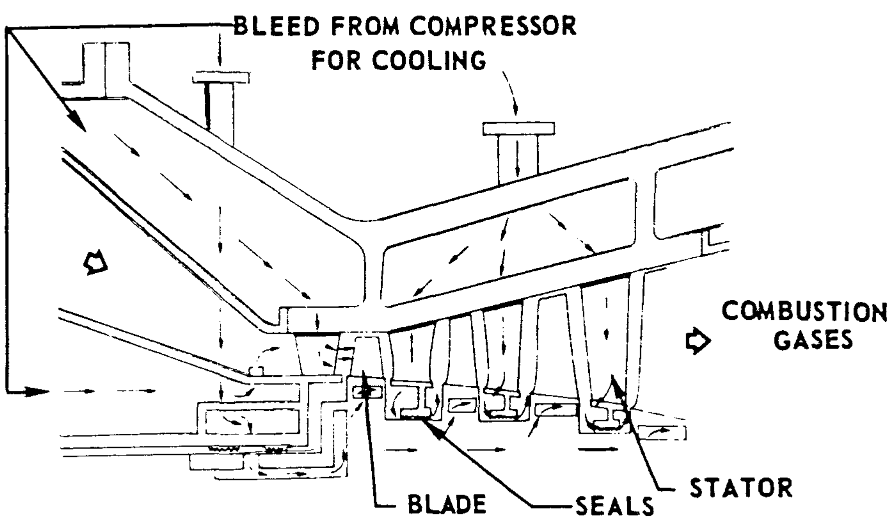
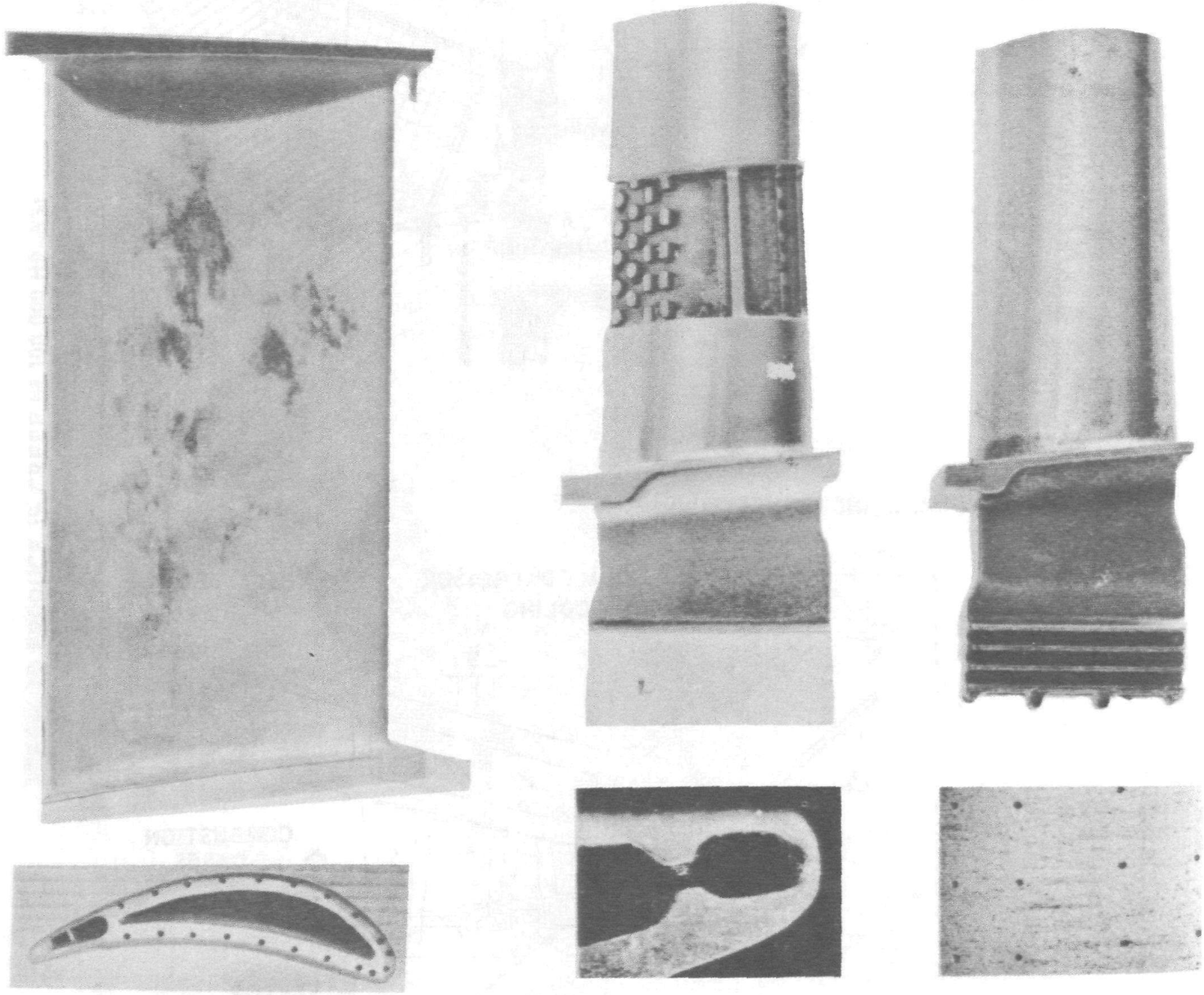


FIG. 25. TURBINE COOLING SCHEMES



FILM-COOLED IMPINGEMENT–CONVECTION TRANSPIRATION COOLED

FIG. 26. ADVANCED BLADE COOLING CONFIGURATIONS FOR AIRCRAFT POWERPLANTS

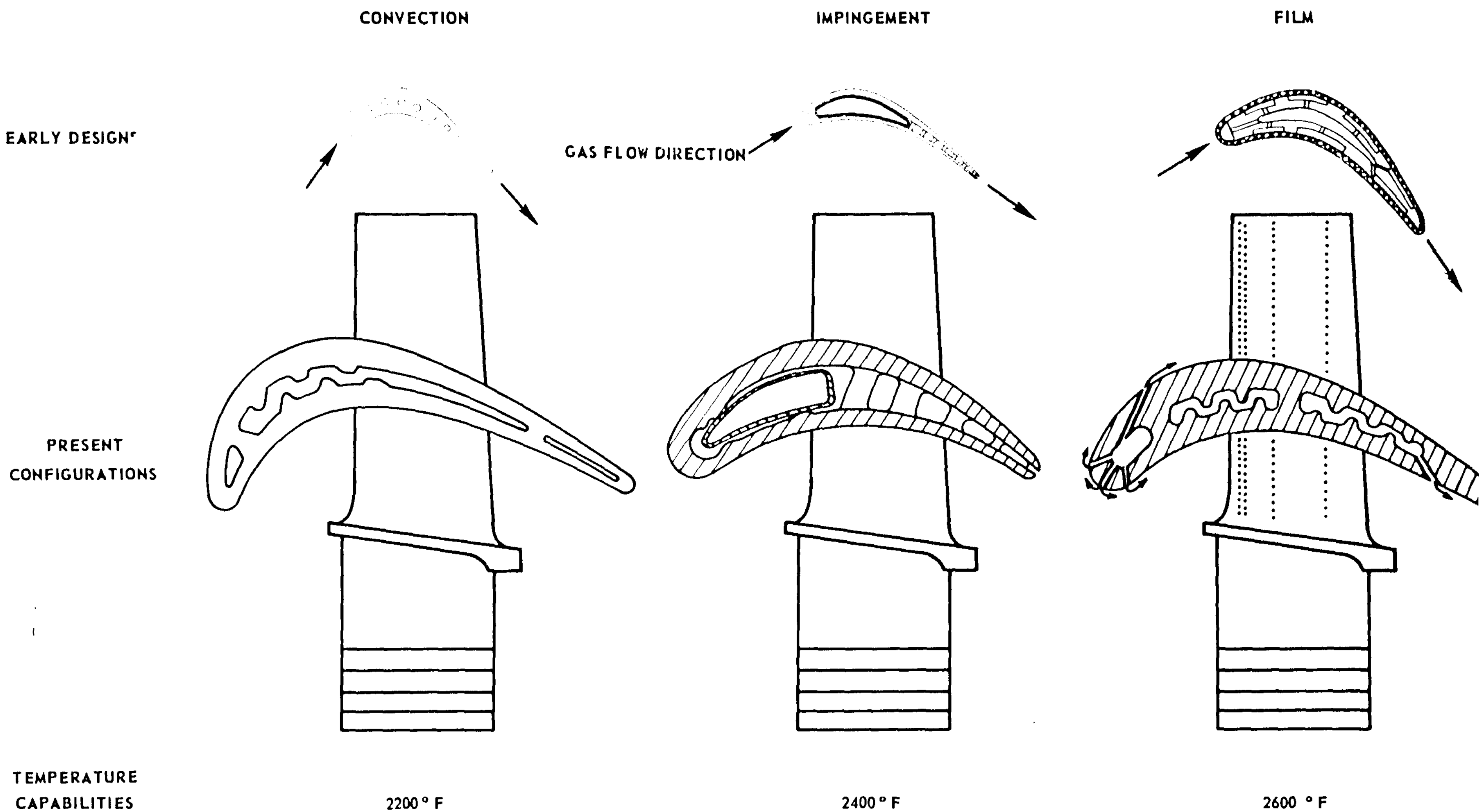


FIG. 27. TURBINE BLADE COOLING BLADE IMPROVEMENTS

FUEL–METHANE (HHV ≈ 1000 BTU/FT^3)

AMBIENT – 80 F AND 1000 FT

TURBINE COOLING CONFIGURATION:ADVANCED IMPINGEMENT–CONVECTION

ELECTRIC GENERATOR EFFICIENCY AND AUXILIARY POWER REQUIREMENTS NOT INCLUDED

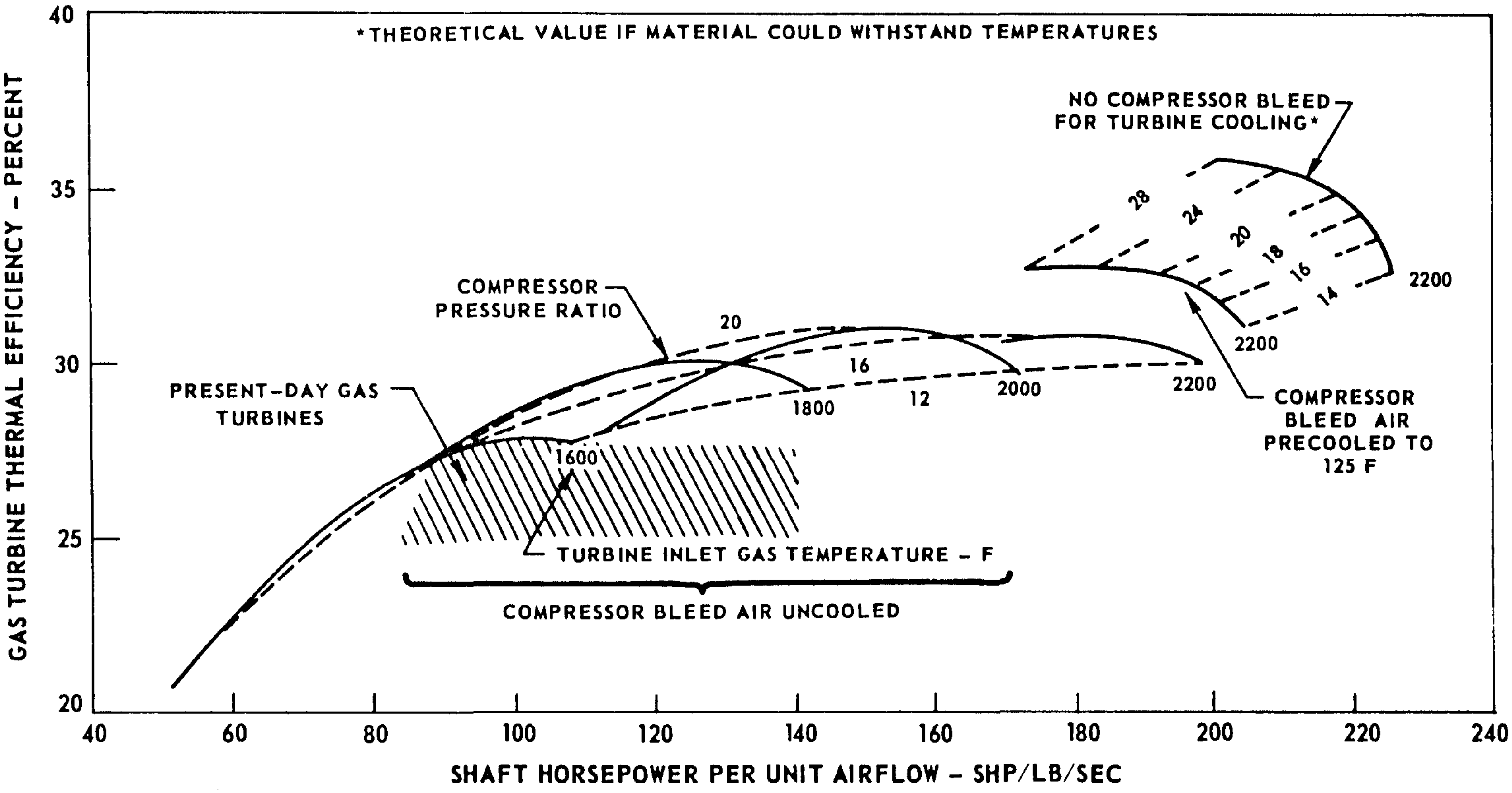


FIG. 28. ESTIMATED 1970–DECADE SIMPLE–CYCLE BASE–LOAD GAS TURBINE PERFORMANCE

COMPRESSOR BLEED AIR PRECOOLED TO 200 F
FUEL-METHANE (HHV = 1000 BTU/FT^3)
AMBIENT-80 F AND 1000 FT
TURBINE COOLING CONFIGURATION: ADVANCED IMPINGEMENT-CONVECTION
ELECTRIC GENERATOR EFFICIENCY AND AUXILIARY POWER REQUIREMENTS NOT INCLUDED

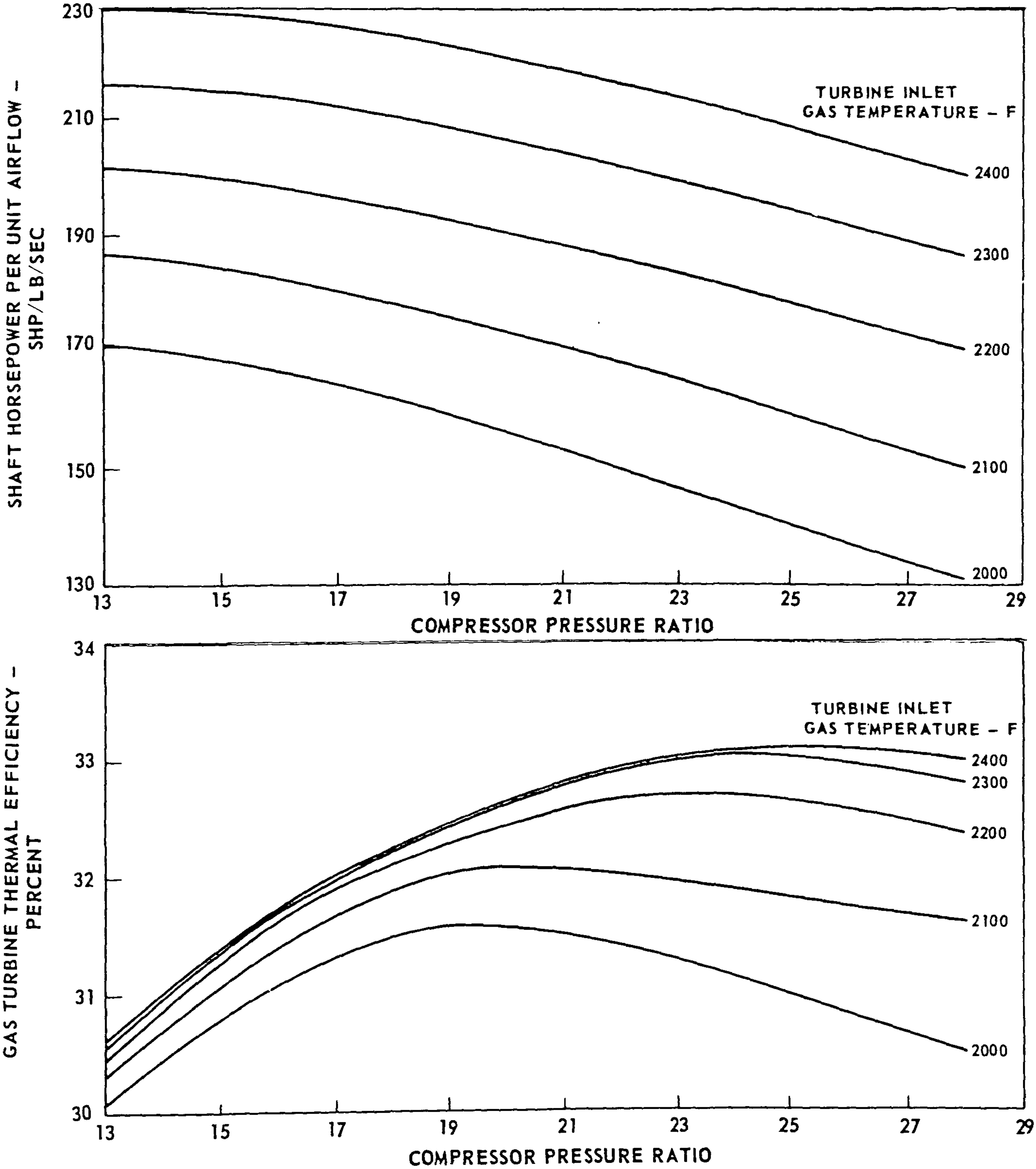


FIG. 29. ESTIMATED 1970-DECADE SIMPLE-CYCLE BASE-LOAD GAS TURBINE PERFORMANCE WITH SUPPLEMENTARY COOLING

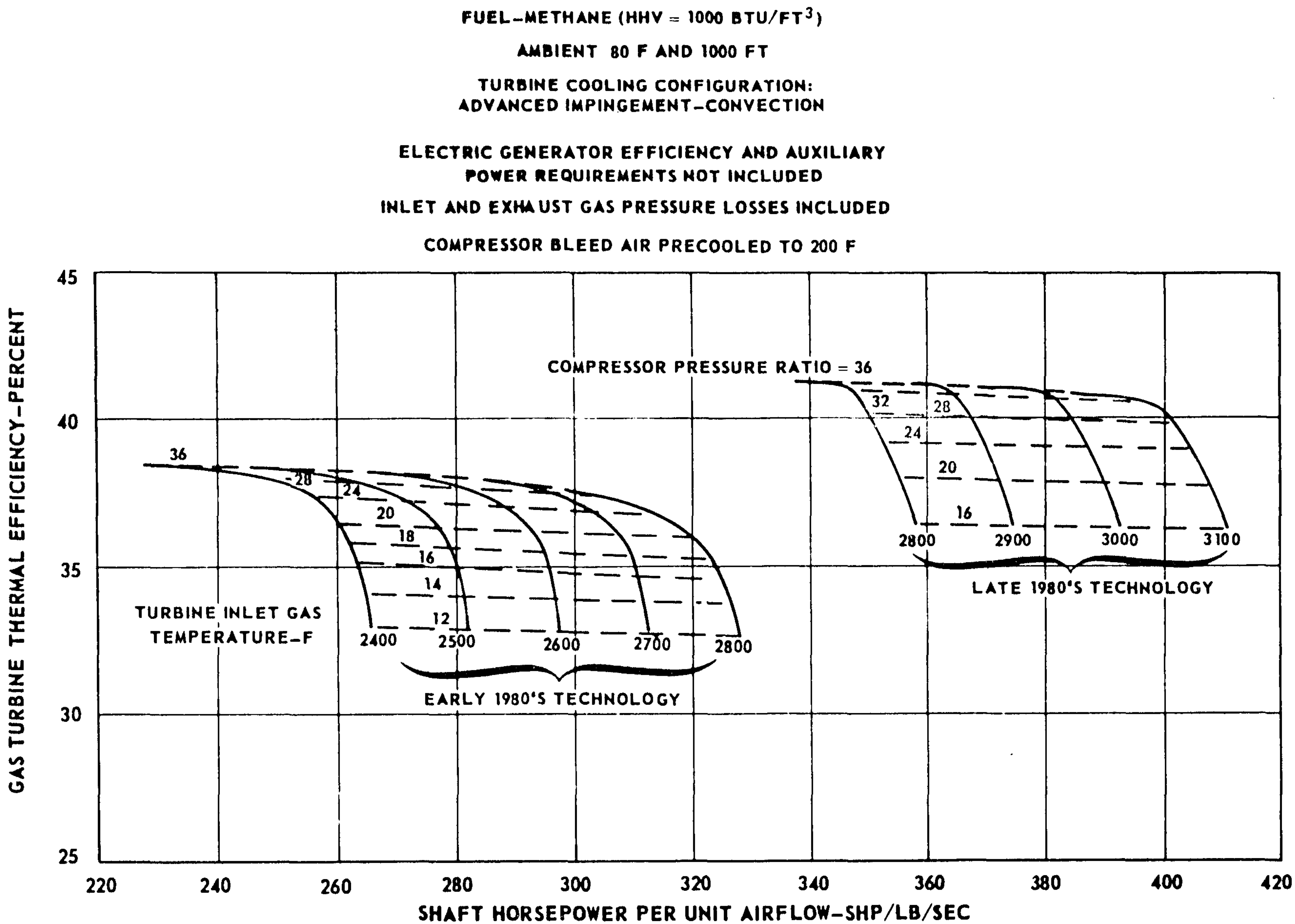
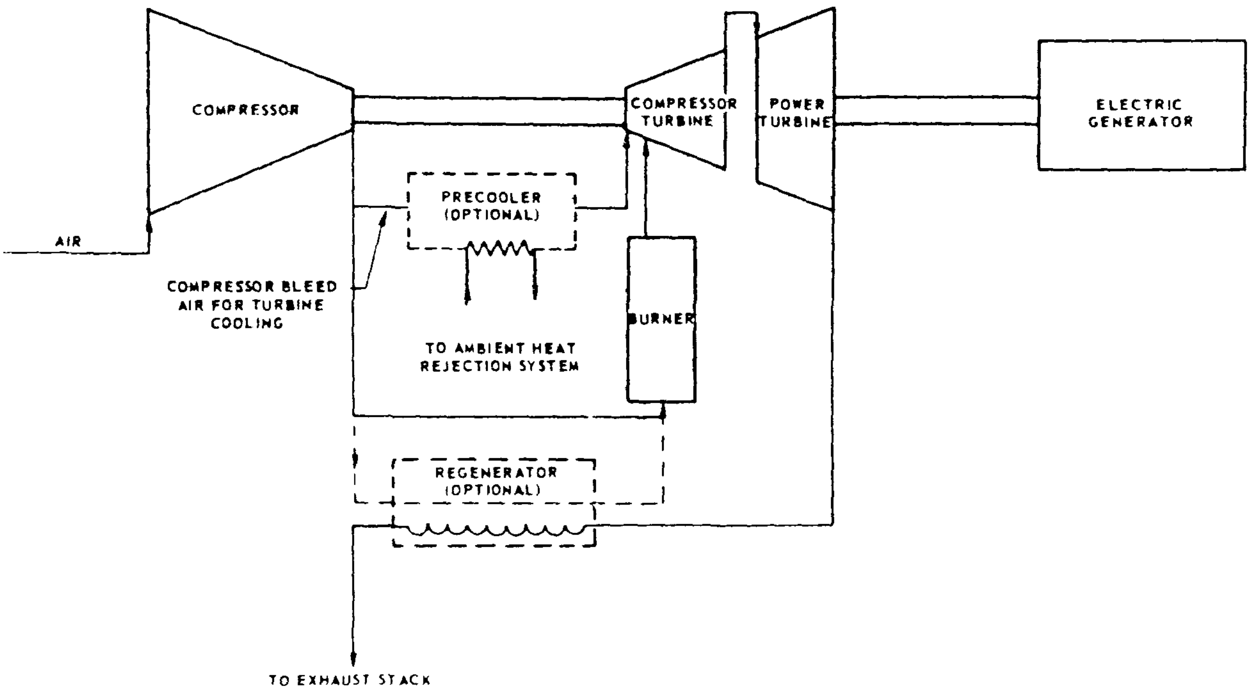


FIG. 30. ESTIMATED 1980 – DECADE SIMPLE-CYCLE BASE-LOAD GAS TURBINE PERFORMANCE

(a) SIMPLE AND REGENERATIVE CYCLE



(b) COMPOUND CYCLE

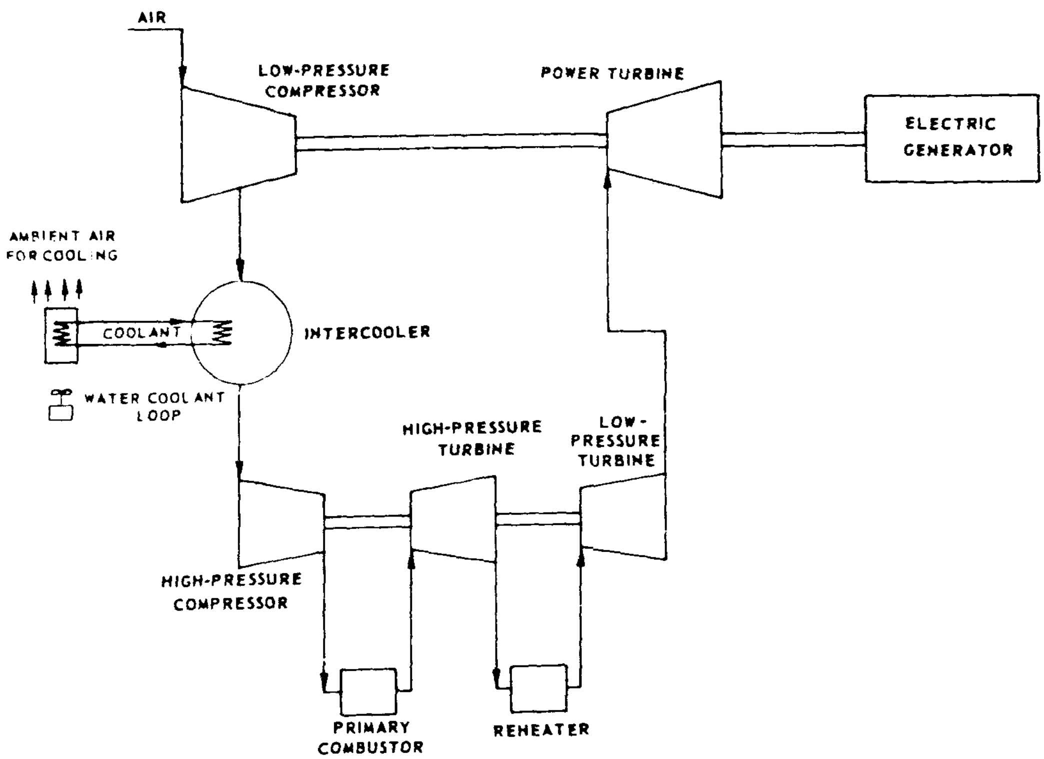


FIG. 31. GAS TURBINE FLOW DIAGRAMS

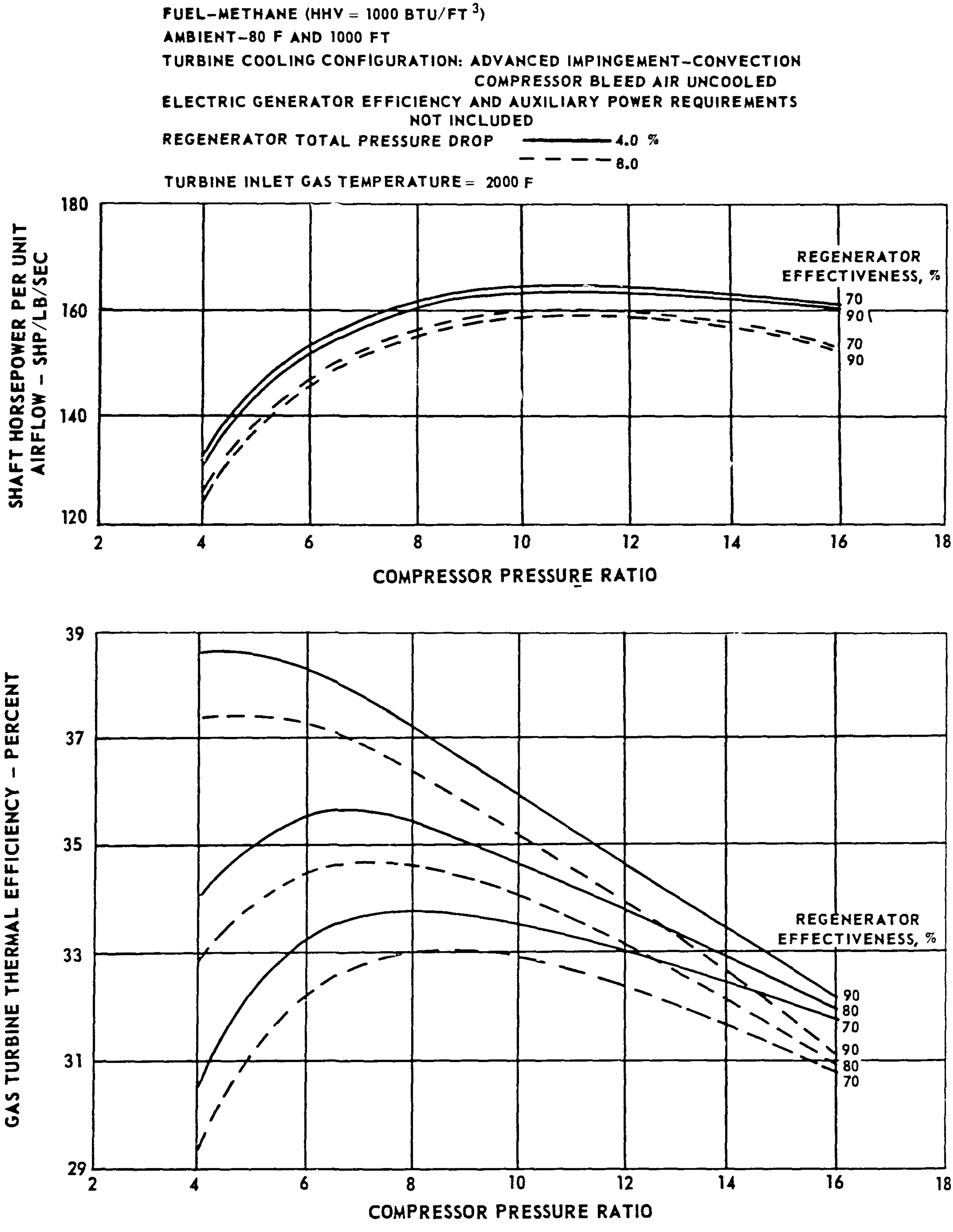
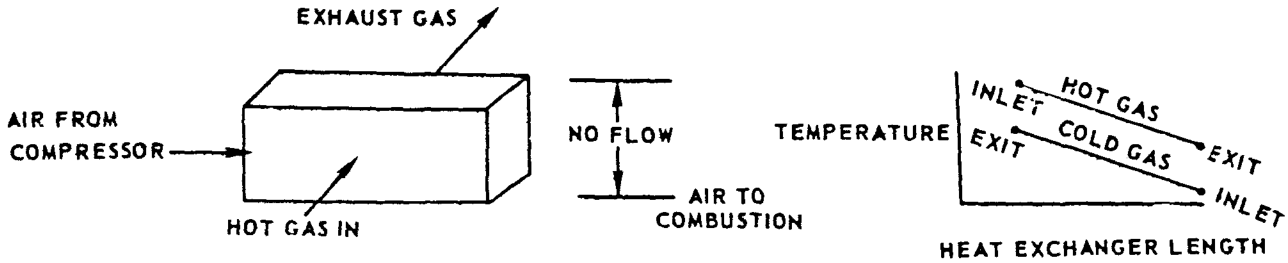


FIG. 32. ESTIMATED 1970-DECADE REGENERATIVE-CYCLE BASE-LOAD GAS TURBINE PERFORMANCE



FUEL-METHANE (HHV = 1000 BTU/FT^3)

AMBIENT-80 F AND 1000 FT

TURBINE COOLING CONFIGURATION: ADVANCED IMPINGEMENT-CONVECTION

COMPRESSOR BLEED AIR UNCOOLED

REGENERATOR PRESSURE LOSS = 4.0%

REGENERATOR AIRSIDE EFFECTIVENESS = 80 %

TURBINE INLET GAS TEMPERATURE = 2000 F

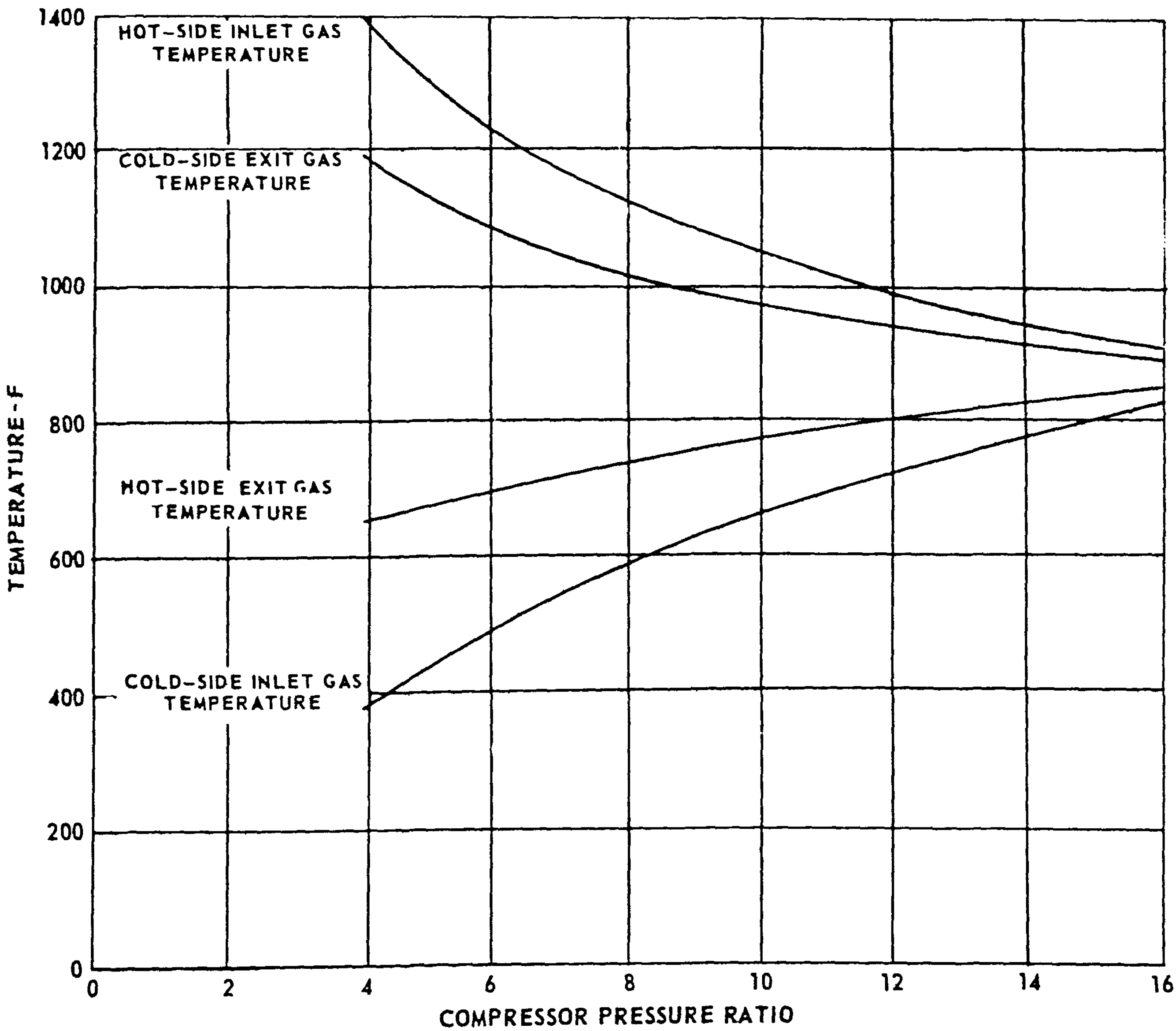


FIG. 33. REGENERATOR GAS TEMPERATURES FOR 1970-DECADE DISIGNS

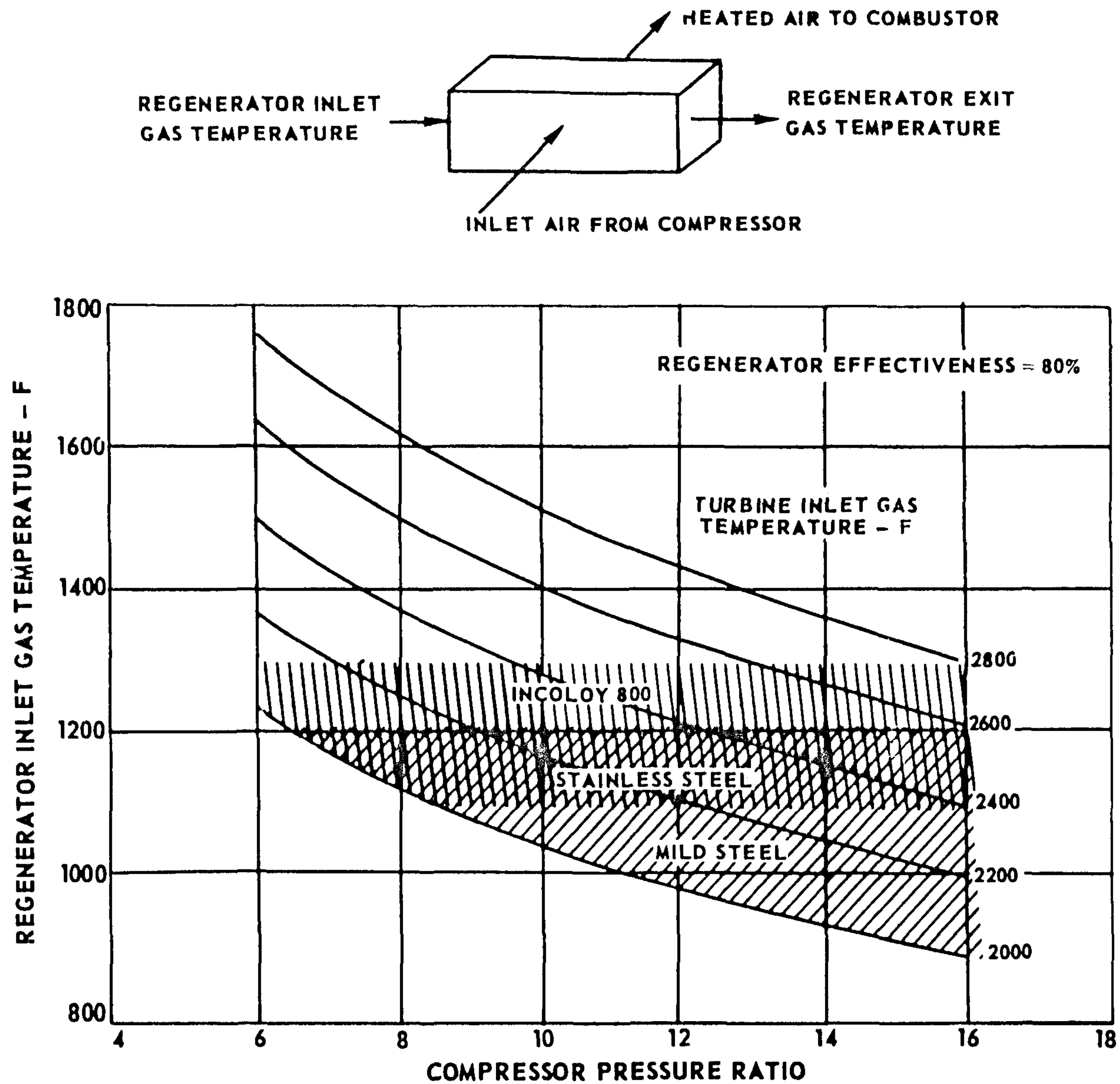


FIG. 34. ESTIMATES OF MATERIAL TYPES REQUIRED IN REGENERATIVE-CYCLE ENGINE

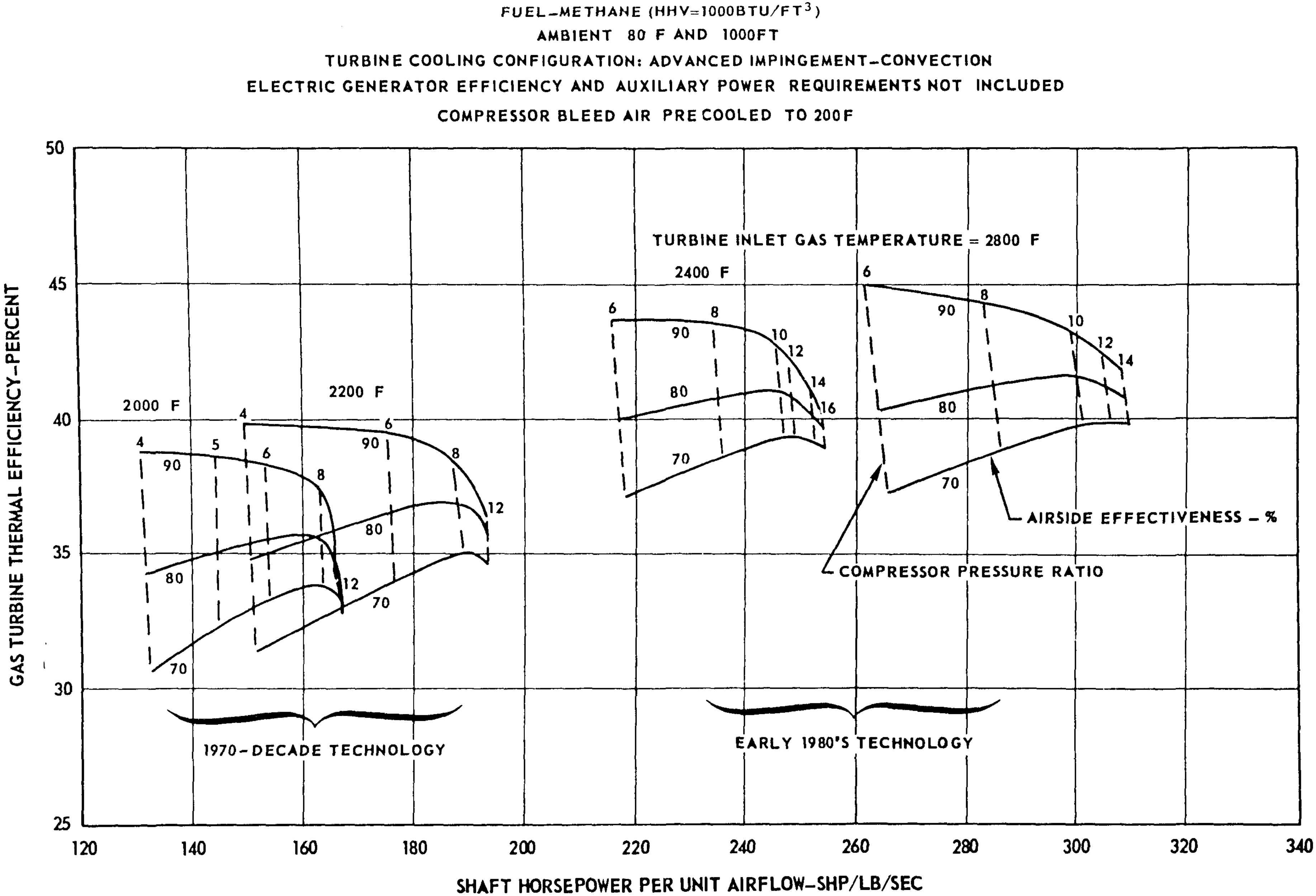


FIG. 35. ESTIMATED 1970 - AND EARLY 1980 - DECADE REGENERATIVE-CYCLE BASE-LOAD GAS TURBINE PERFORMANCE

INLET AND EXHAUST GAS PRESSURE LOSSES INCLUDED

FUEL: METHANE (HHV = 1000 BTU/FT3)

AMBIENT: 80 F AND 1000 FT

TURBINE COOLING CONFIGURATION: ADVANCED IMPINGEMENT-CONVECTION

ELECTRIC GENERATOR EFFICIENCY AND AUXILIARY POWER REQUIREMENTS NOT INCLUDED

COMPRESSOR BLEED AIR PRECOOLED TO 200 F

REGENERATOR TOTAL PRESSURE DROP = 4%

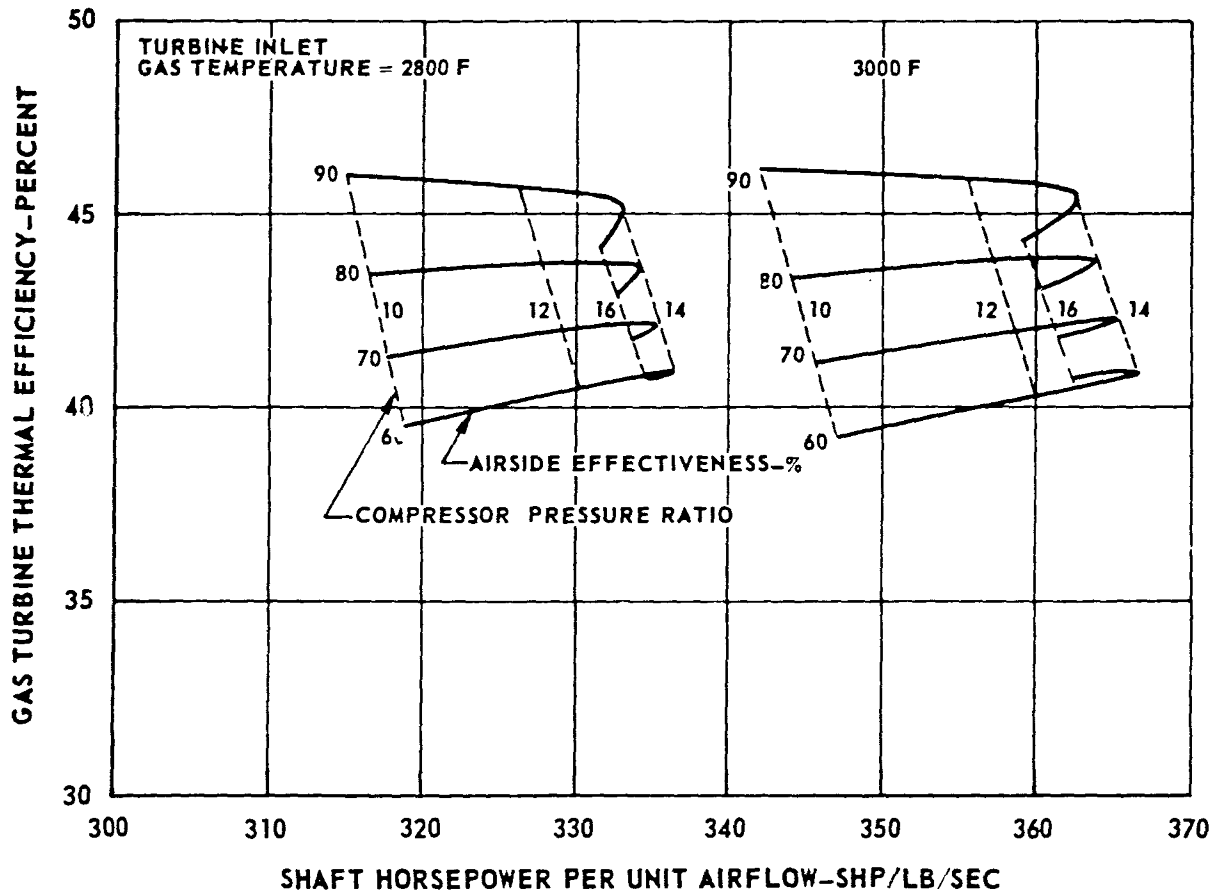


FIG. 36. ESTIMATED LATE 1980'S - DECADE REGENERATIVE-CYCLE GAS TURBINE PERFORMANCE

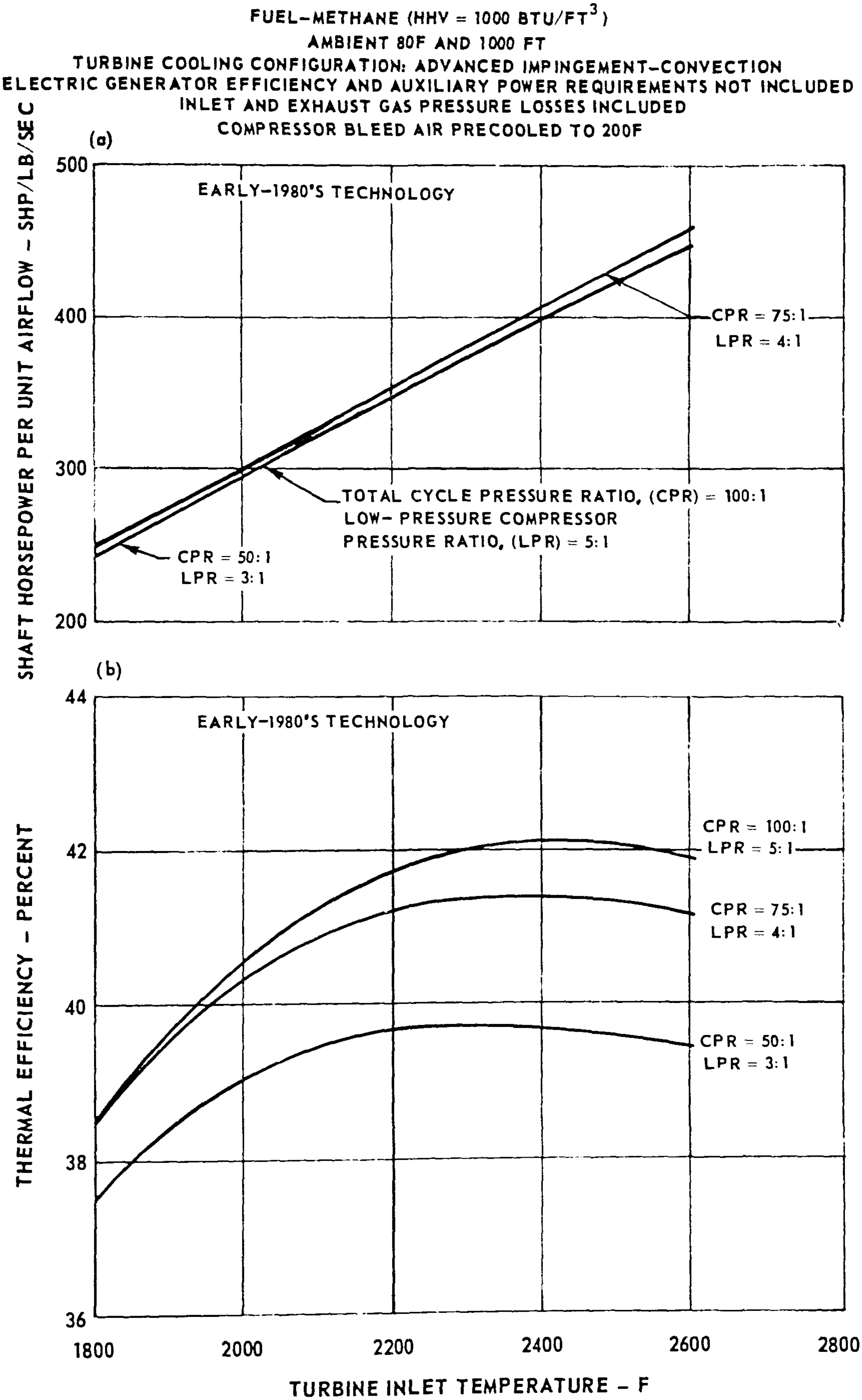


FIG. 37. ESTIMATED 1980 - DECADE COMPOUND-CYCLE BASE-LOAD GAS TURBINE PERFORMANCE

FUEL–METHANE (HHV = 1000 BTU/FT³)
AMBIENT 80F AND 1000 FT
TURBINE COOLING CONFIGURATION: ADVANCED IMPINGEMENT–CONVECTION
ELECTRIC GENERATOR EFFICIENCY AND AUXILIARY POWER REQUIREMENTS NOT INCLUDED
INLET AND EXHAUST GAS PRESSURE LOSSES INCLUDED
COMPRESSOR BLEED AIR PRECOOLED TO 200F

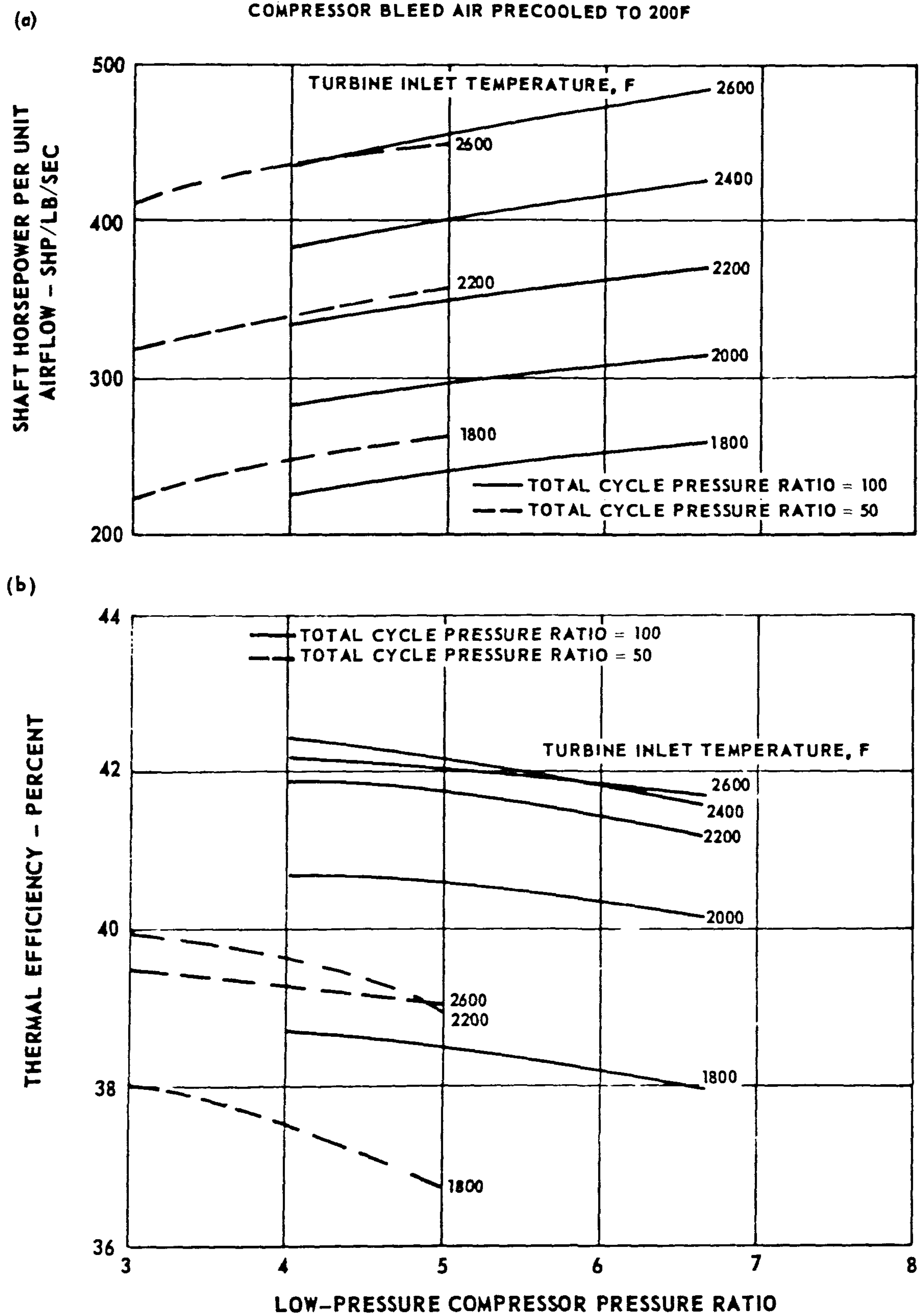


FIG. 38. EFFECT OF LOW–PRESSURE COMPRESSOR PRESSURE RATIO ON COMPOUND–CYCLE GAS TURBINE PERFORMANCE

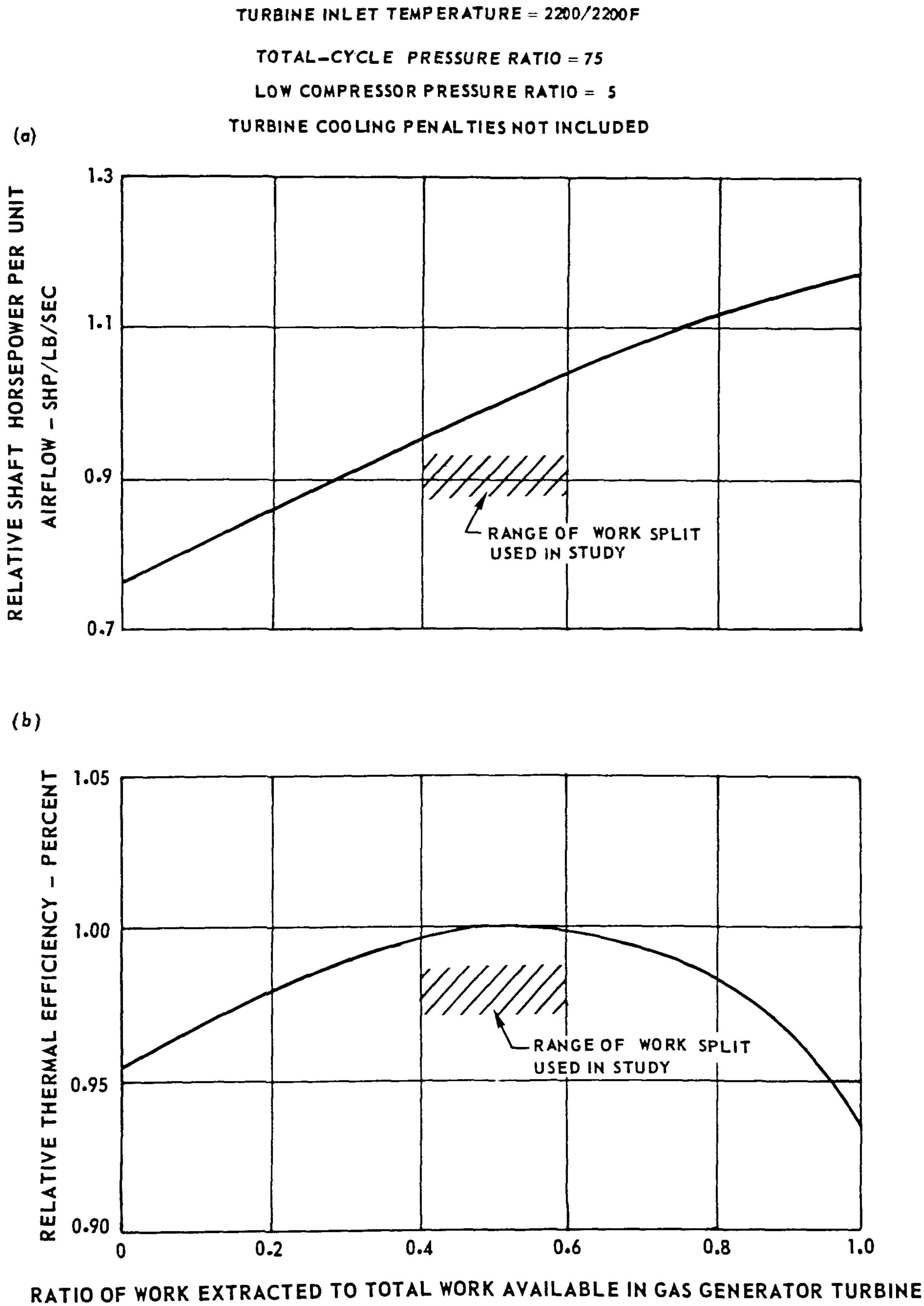


FIG. 39. EFFECT OF WORK SPLIT ON COMPOUND-CYCLE PERFORMANCE

SELLING PRICE IN 1970 DOLLARS BASED ON CONSTANT TOTAL MARKET;
INCLUDES DEVELOPMENT, ASSEMBLY, AND TEST COSTS,
GENERAL AND ADMINISTRATIVE EXPENSES, AND
PROFIT IN SELLING PRICE OF EACH UNIT
COMPRESSOR BLEED AIR PRECOOLED TO 200F
BLADE STRESS LEVELS AS INDICATED

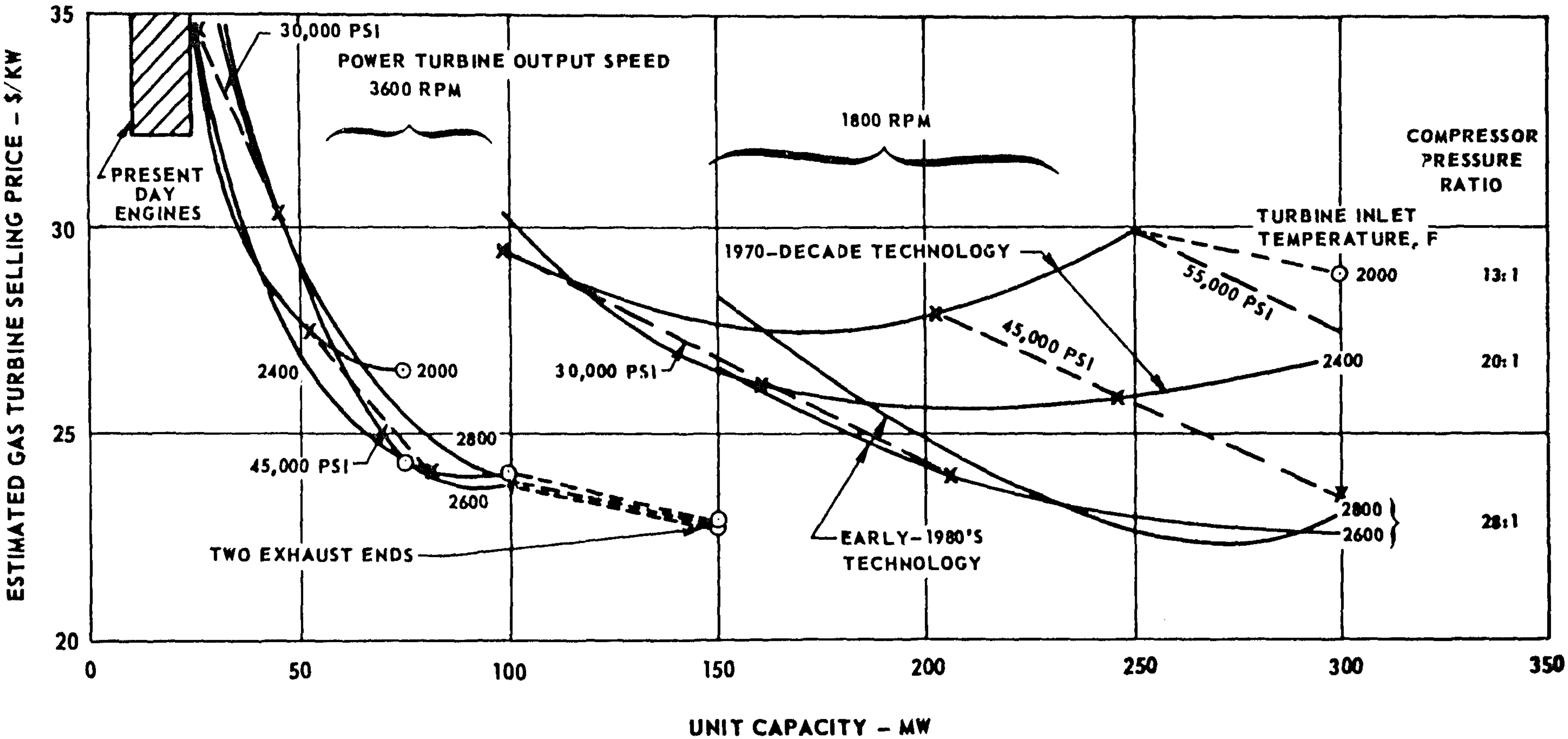


FIG. 40. EFFECT OF GAS TURBINE UNIT CAPACITY ON SELLING PRICE

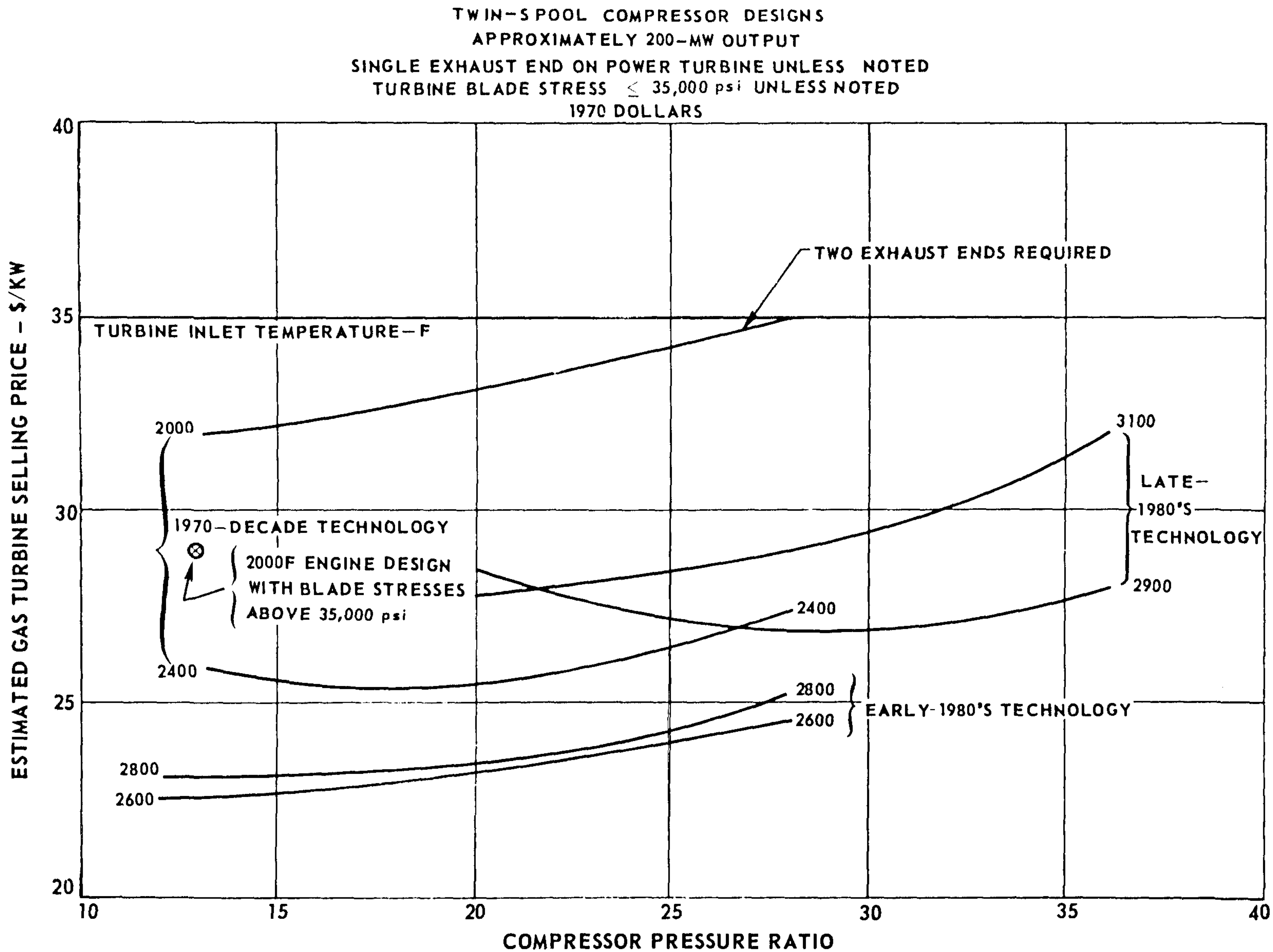


FIG. 41. EFFECT OF COMPRESSOR PRESSURE RATIO AND TURBINE INLET TEMPERATURE ON GAS TURBINE SELLING PRICE

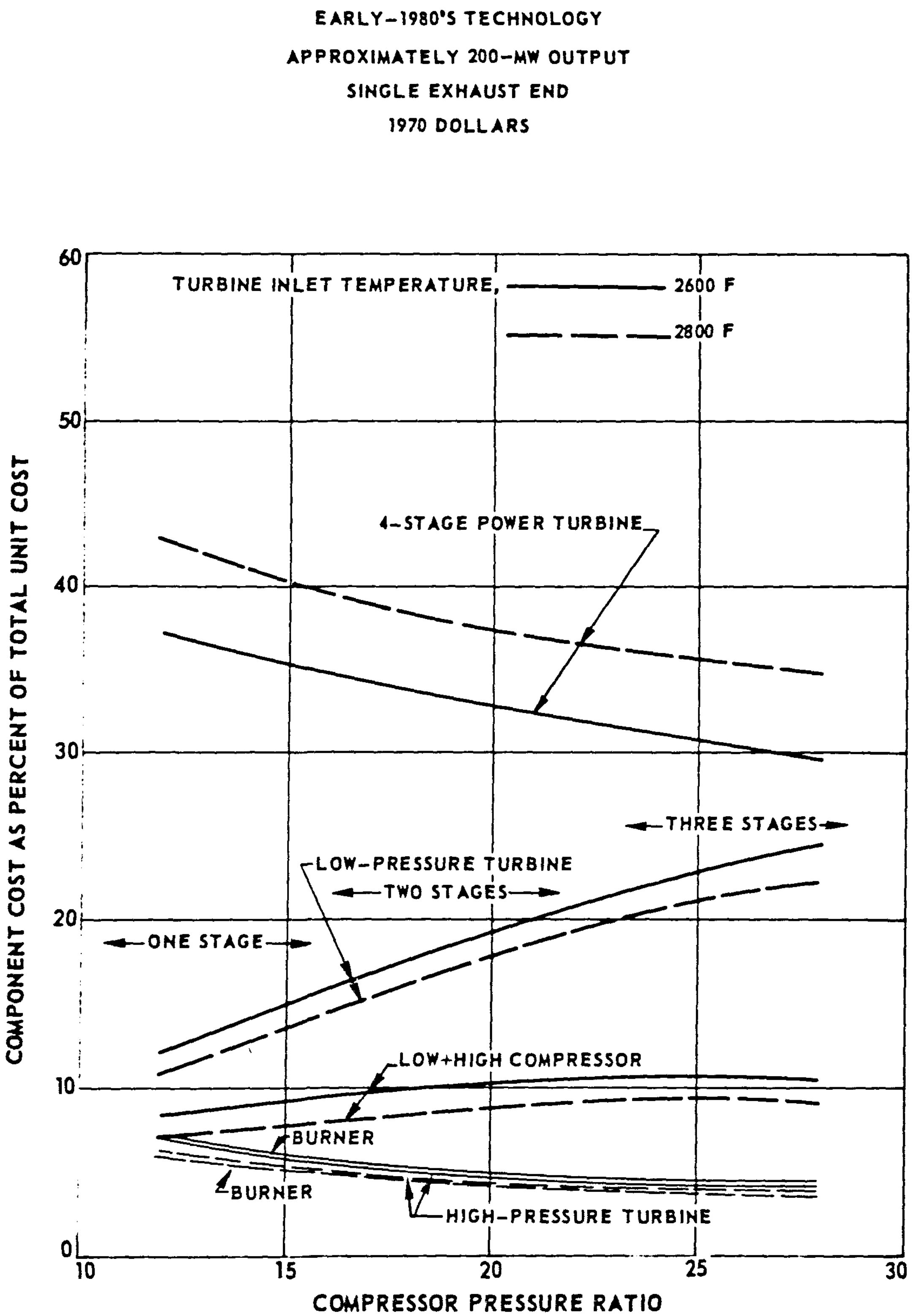


FIG. 42. COMPONENT COST DISTRIBUTION OF ADVANCED GAS TURBINE ENGINES

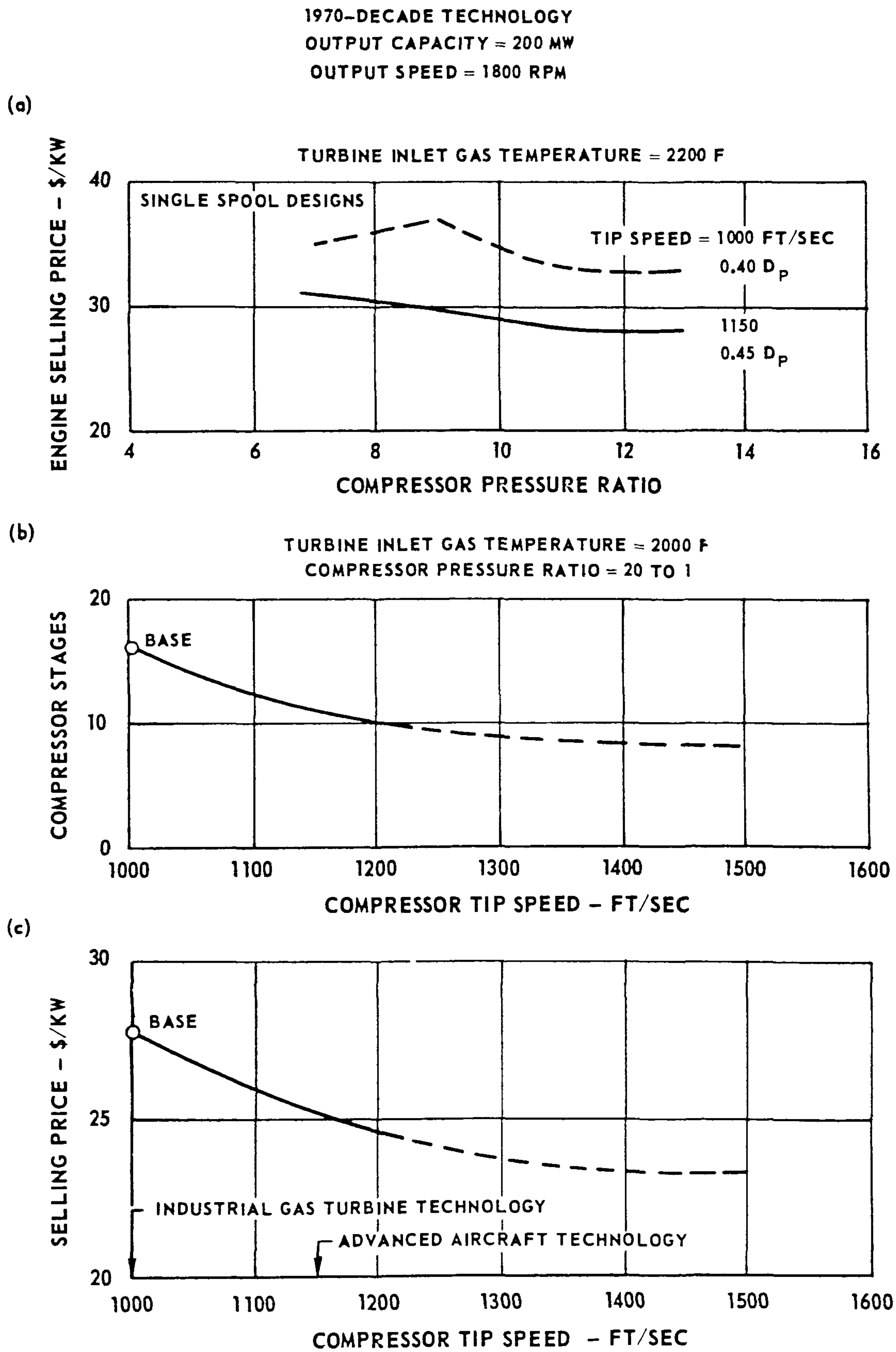


FIG. 43. EFFECT OF COMPRESSOR DESIGN PARAMETERS ON ENGINE SELLING PRICE

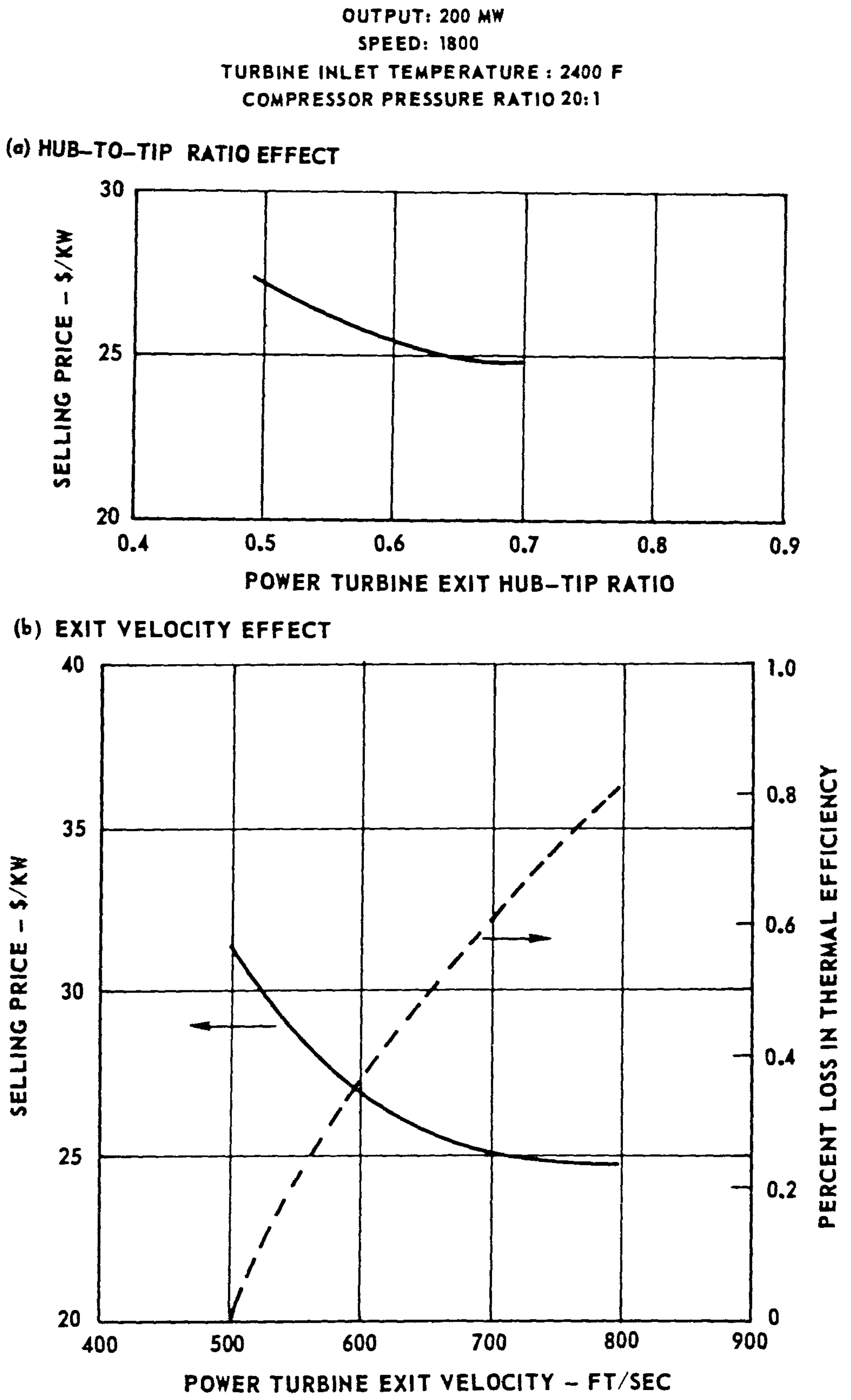


FIG. 44. EFFECT OF POWER TURBINE DESIGN PARAMETERS ON ENGINE PERFORMANCE AND COST

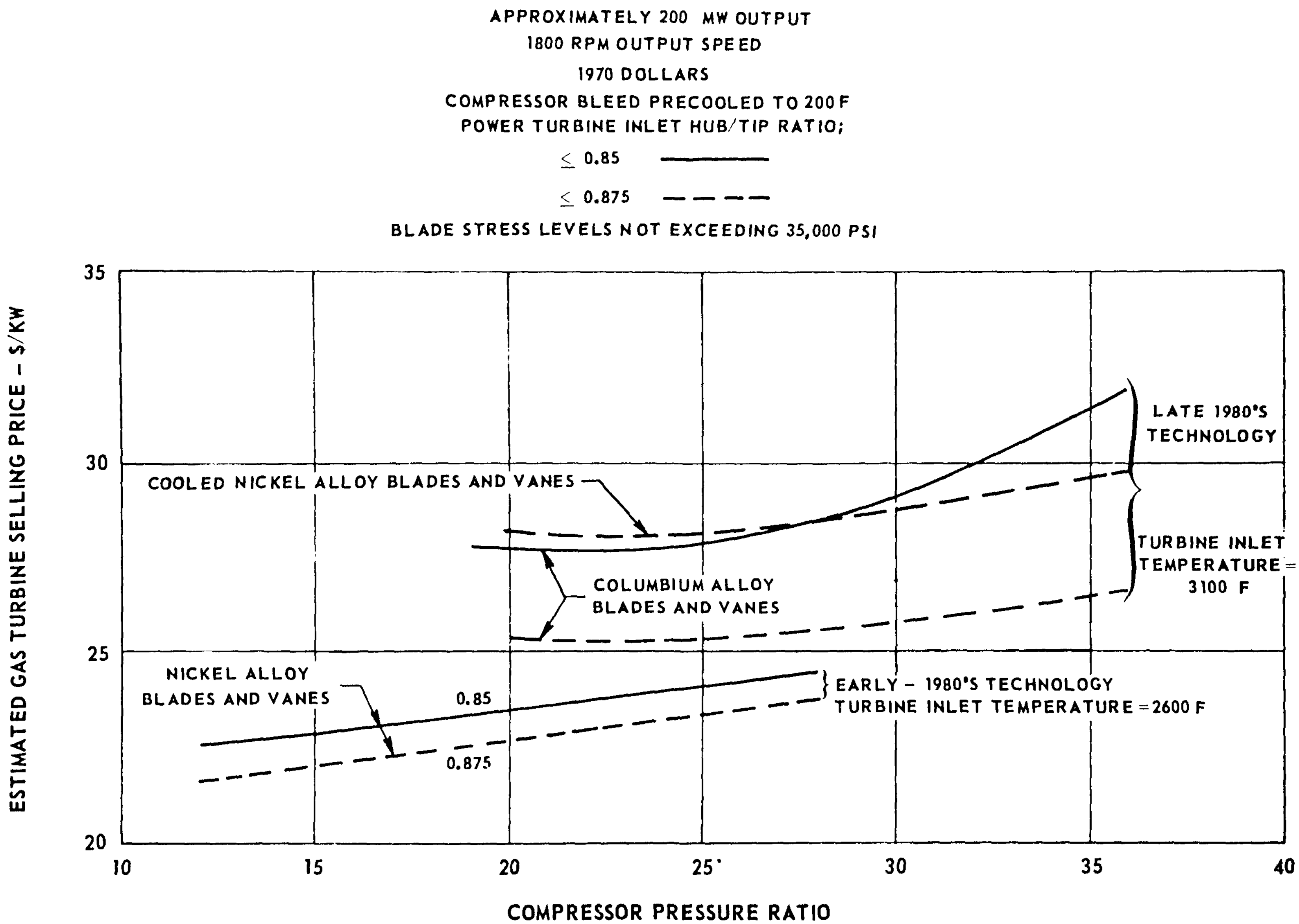


FIG. 45. EFFECT OF POWER TURBINE MATERIALS AND DESIGN PARAMETERS ON SELLING PRICE

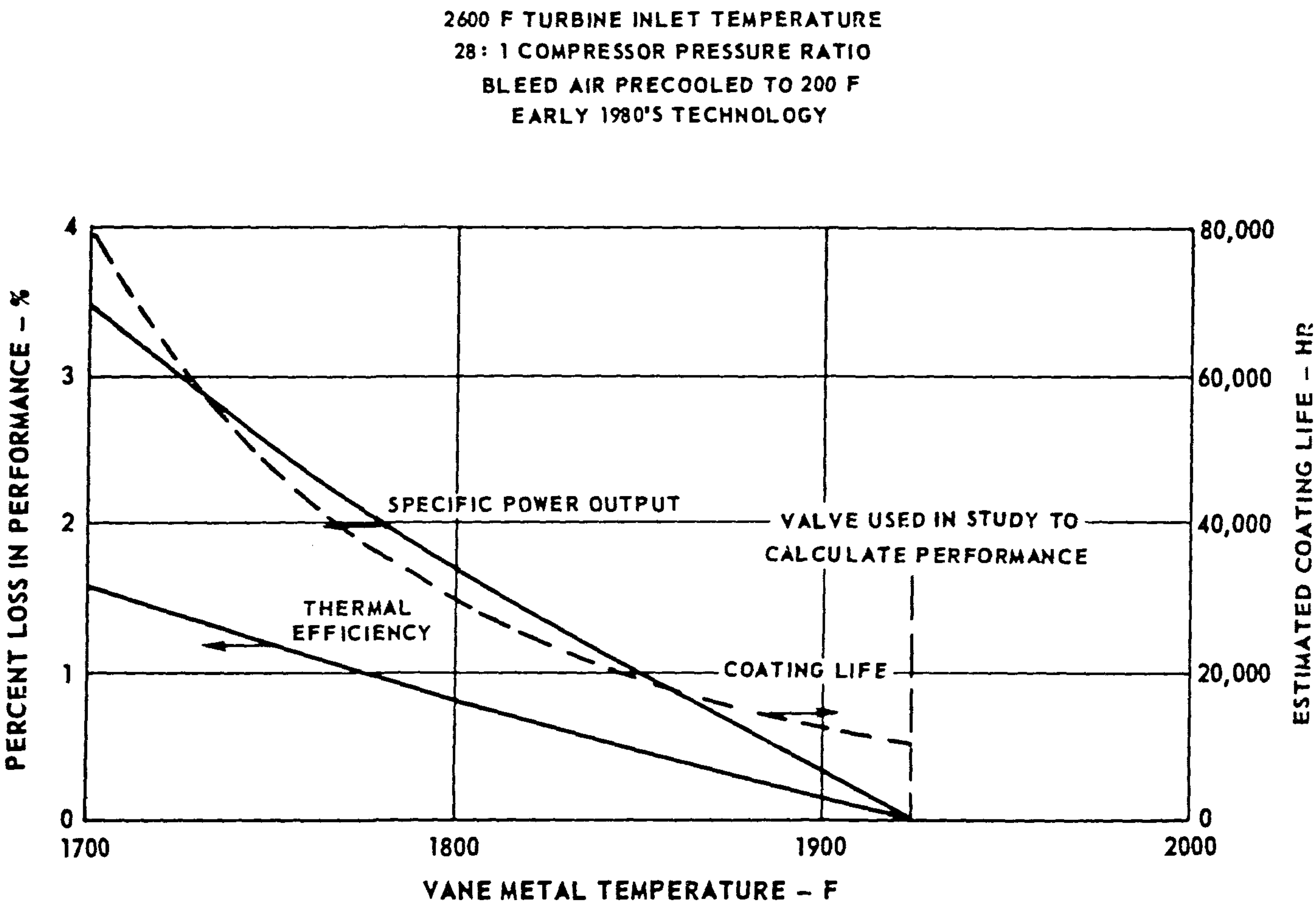


FIG. 46. EFFECT OF VANE COOLING REQUIREMENTS ON ENGINE PERFORMANCE AND COATING LIFE

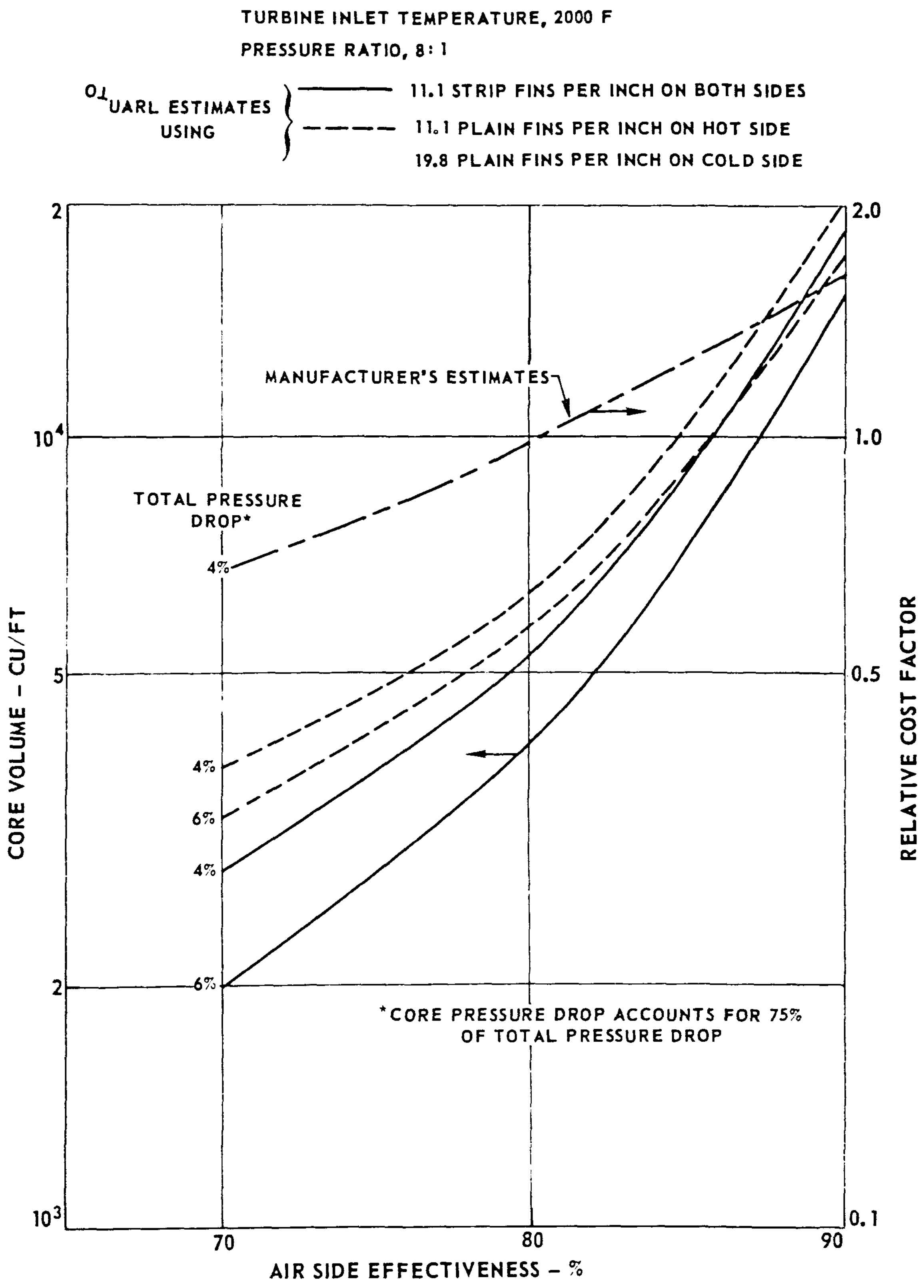


FIG. 47. VARIATION OF REGENERATOR SIZE WITH EFFECTIVENESS

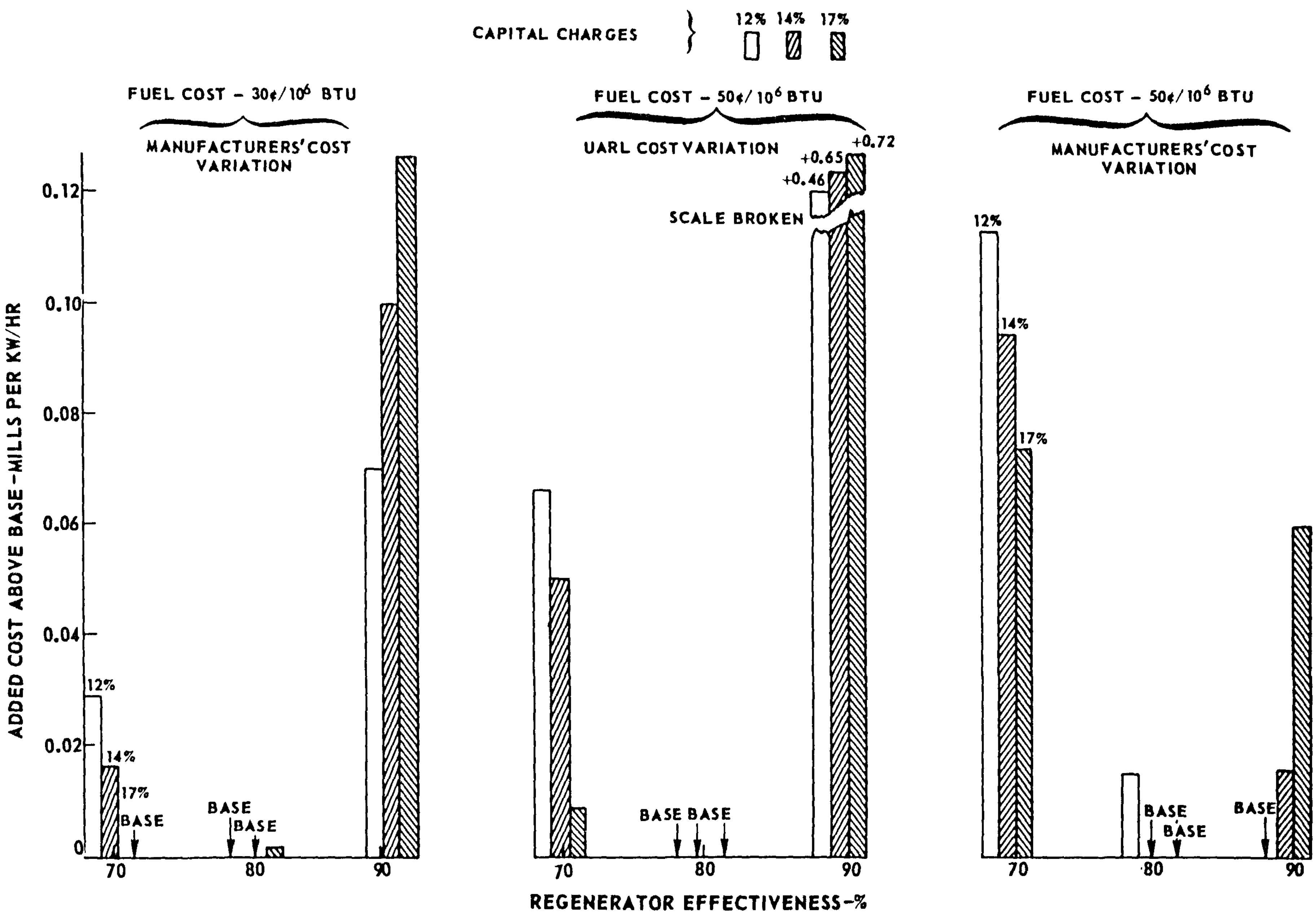


FIG. 48. INFLUENCE OF REGENERATOR EFFECTIVENESS ON POWER COSTS

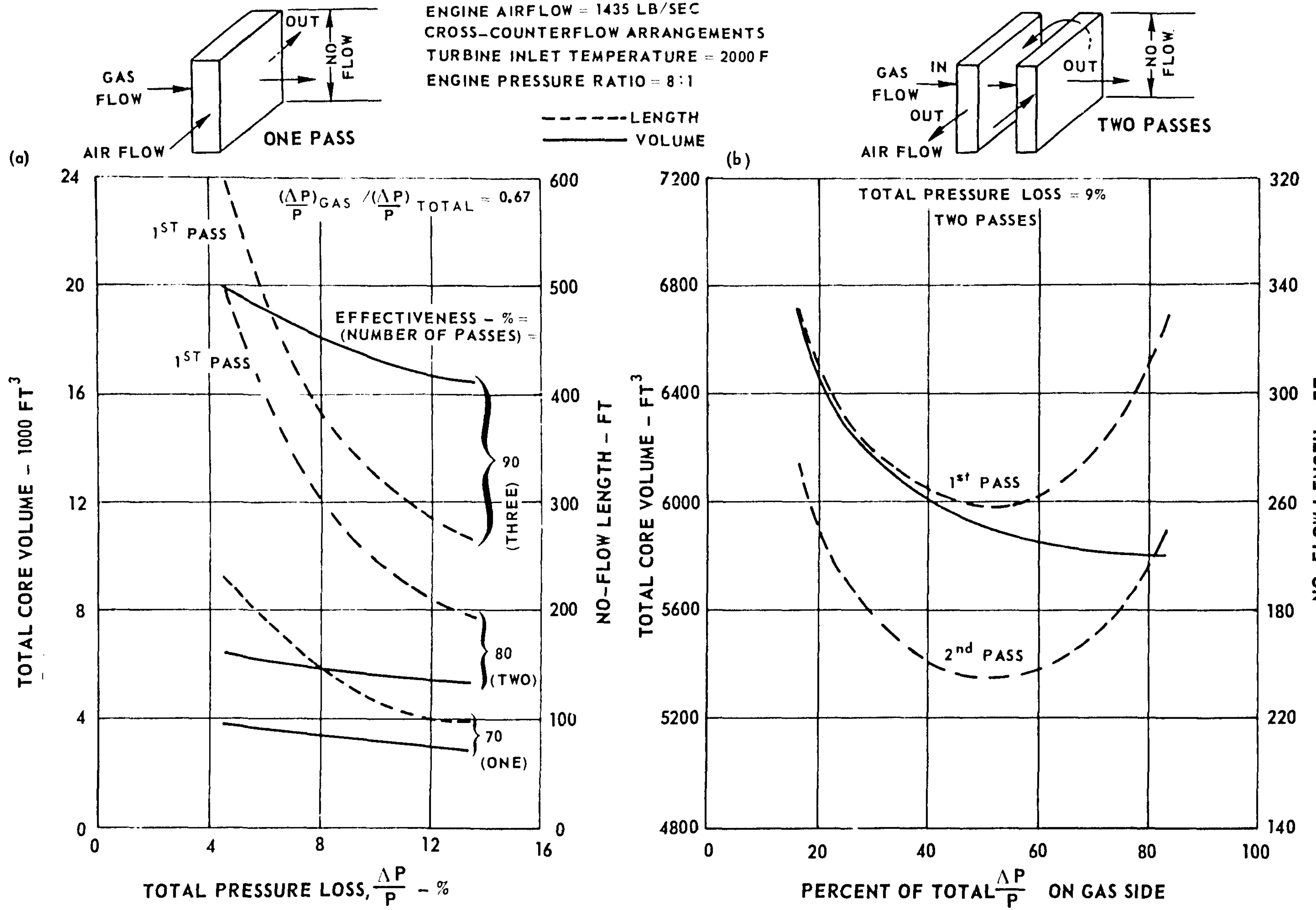


FIG. 49. EFFECT OF PRESSURE LOSS PARAMETERS ON REGENERATOR SIZE CHARACTERISTICS

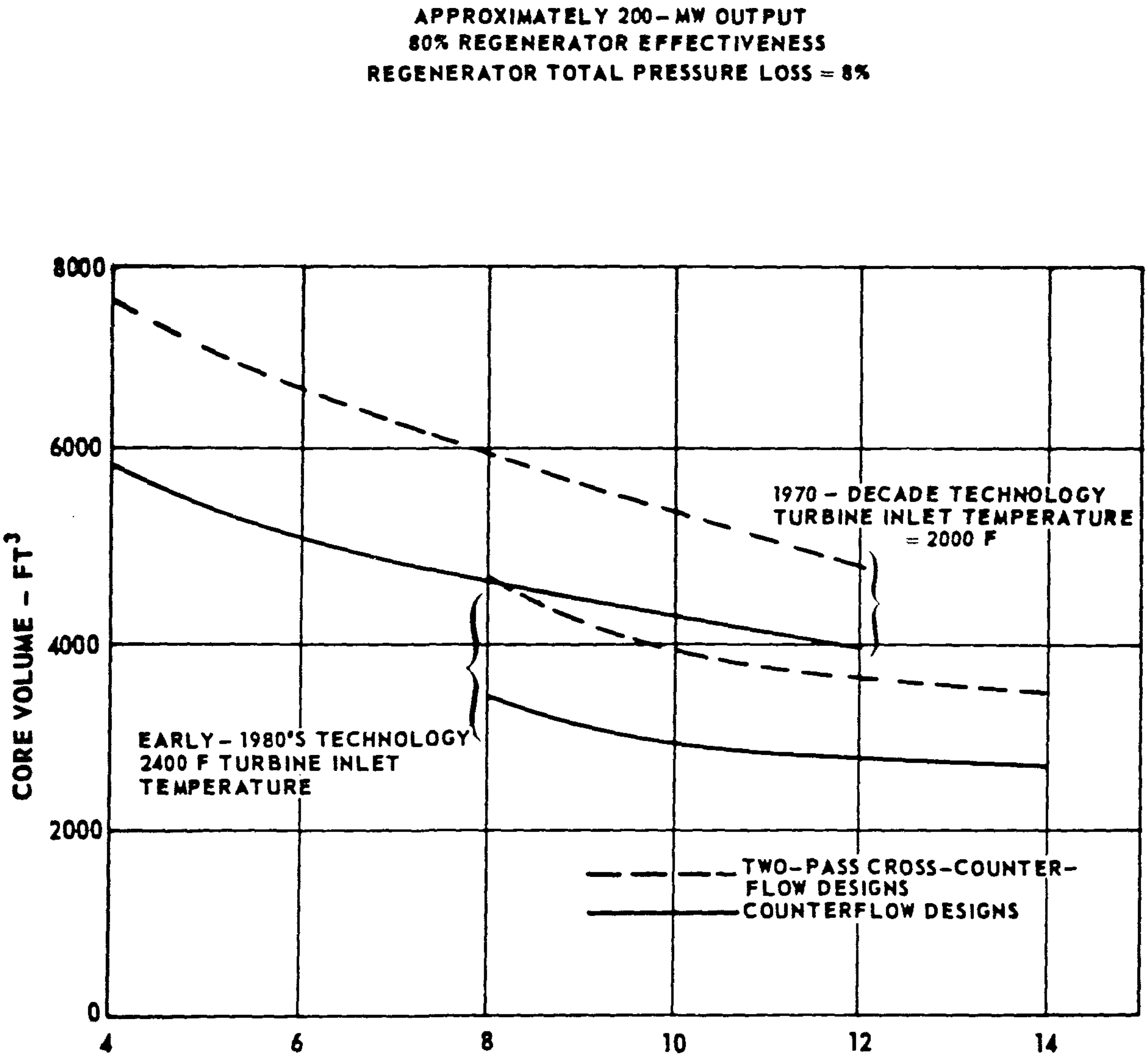
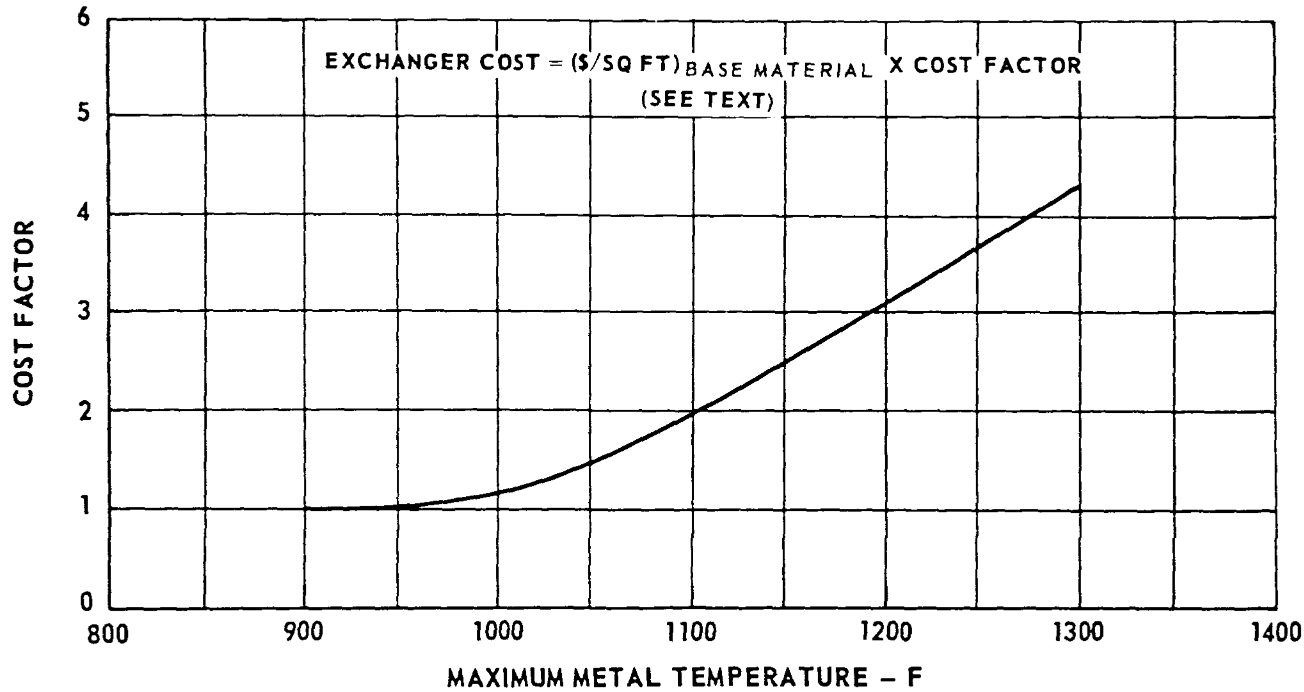


FIG. 50. EFFECT OF COMPRESSOR PRESSURE RATIO ON RECUPERATOR SIZE

(a)



(b)

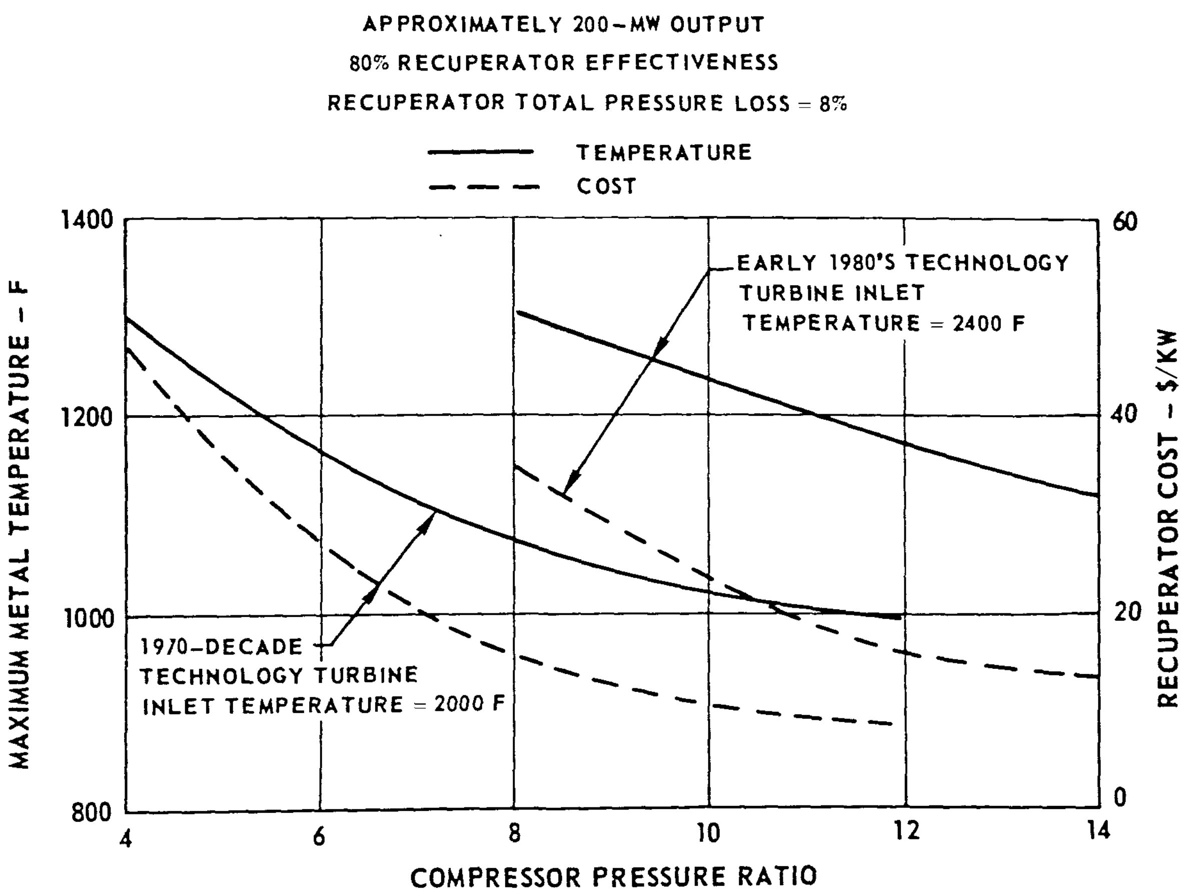


FIG. 51. EFFECT OF TEMPERATURE AND COMPRESSOR PRESSURE RATIO ON RECUPERATOR COST

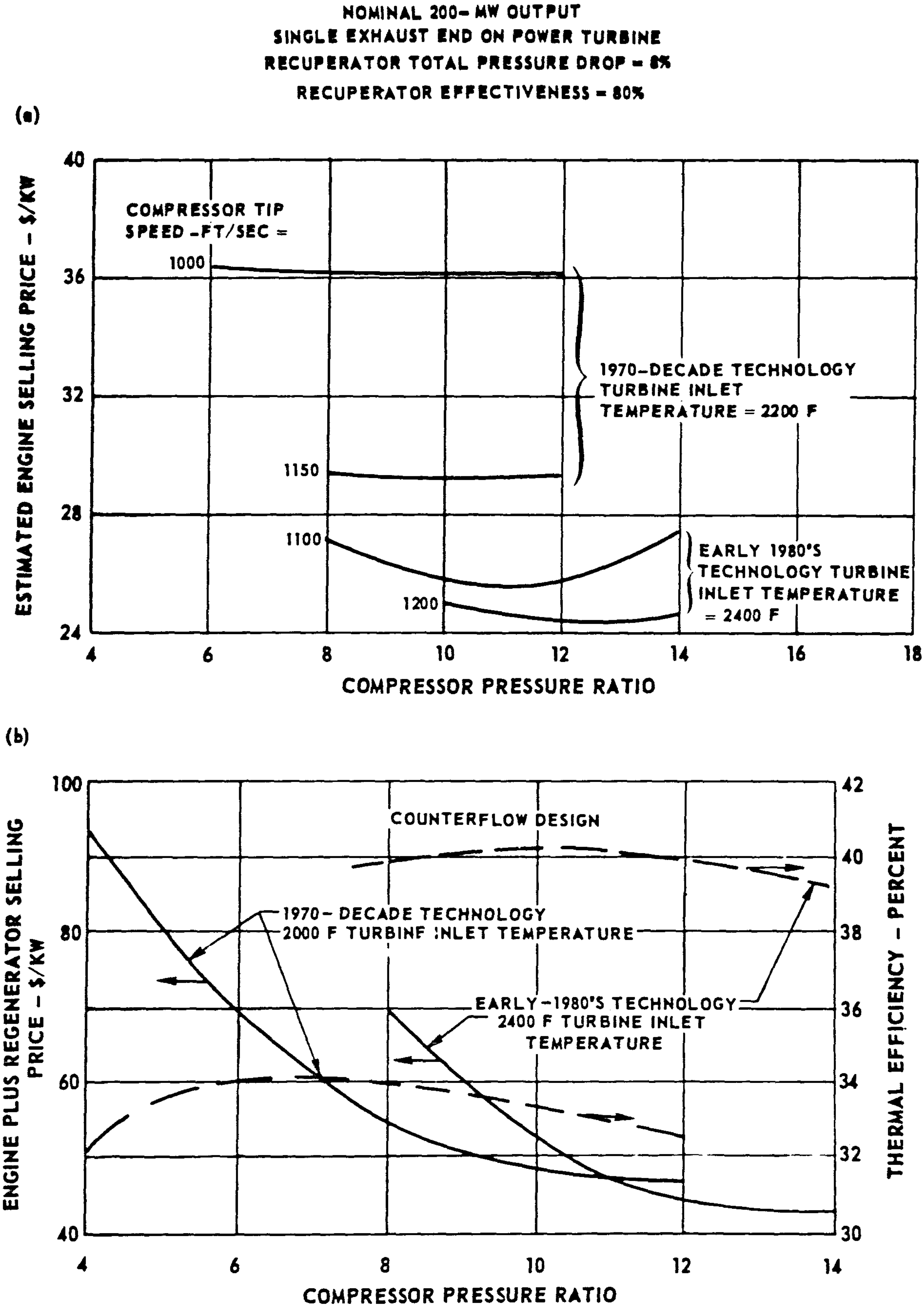


FIG. 52. EFFECT OF DESIGN PARAMETERS ON REGENERATIVE-CYCLE GAS TURBINE ENGINE SELLING PRICE

EARLY 1980'S TECHNOLOGY

TURBINE INLET GAS TEMPERATURE 2400 F

COMPRESSOR PRESSURE RATIO 32:1

AIRFLOW 1176 LB/SEC

SEE TABLE XX FOR REFERENCE CHARACTERISTICS

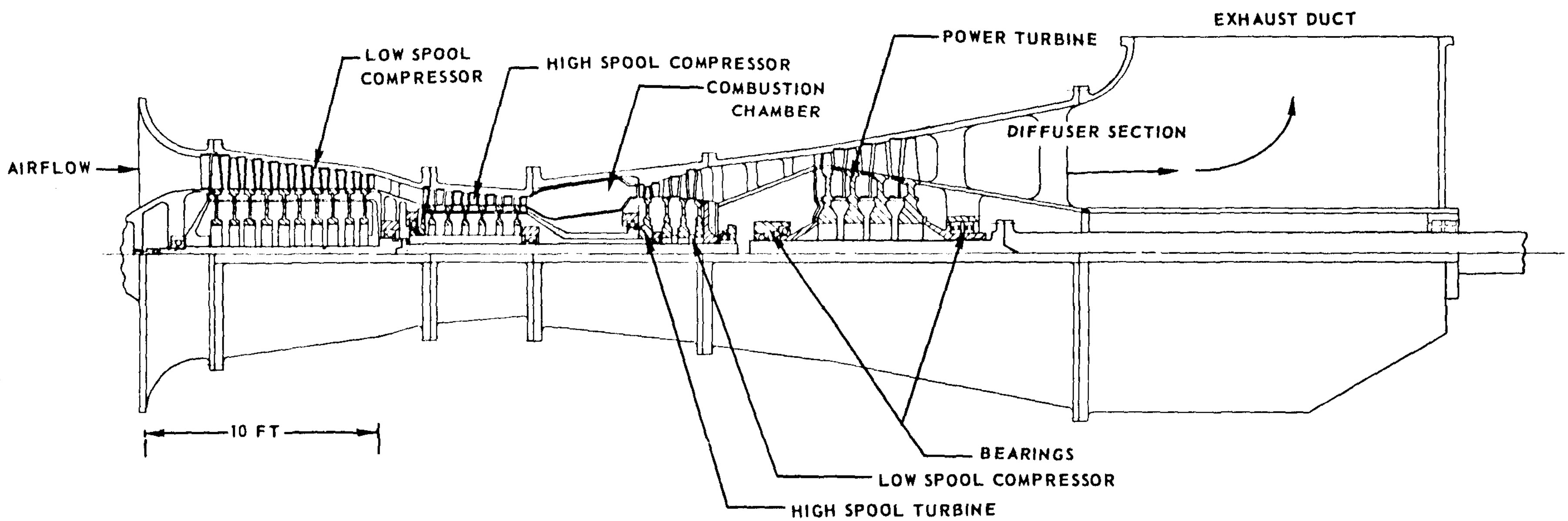
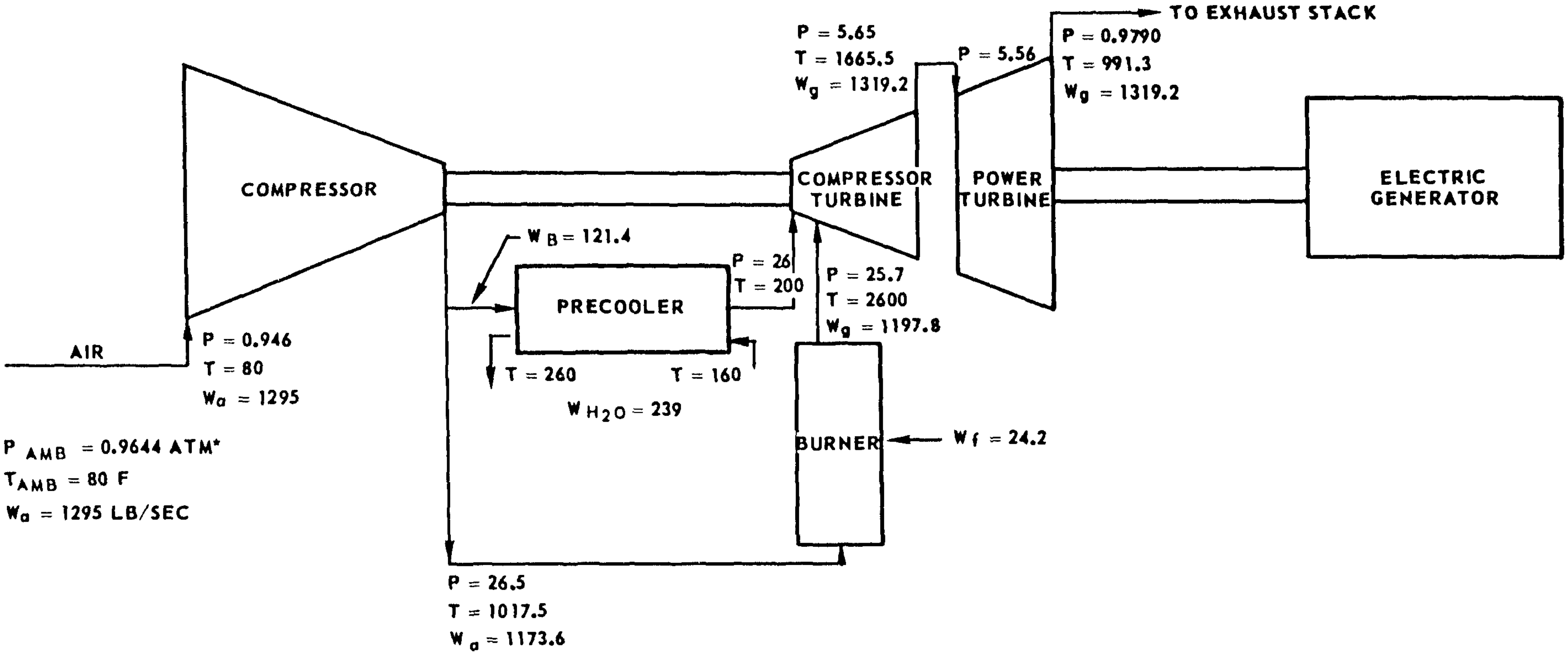


FIG. 53. CONCEPTUAL DESIGN OF 200 – MW BASE-LOAD GAS TURBINE ENGINE

NOMINAL 250 MW
TURBINE INLET TEMPERATURE = 2600 F
COMPRESSOR PRESSURE RATIO = 28 TO 1
EARLY 1980'S ENGINE DESIGN
COMPRESSOR BLEED AIR PRECOOLED TO 200 F

SEE TABLE VI FOR REFERENCE



P = PRESSURE IN ATMOSPHERES
T = TOTAL TEMPERATURE IN F
W = FLOW RATE IN LB/SEC
*1 ATM = 14.7 PSIA

FIG. 54. HEAT BALANCE FOR SIMPLE-CYCLE GAS TURBINE

ELEVATION
EARLY 1980'S TECHNOLOGY

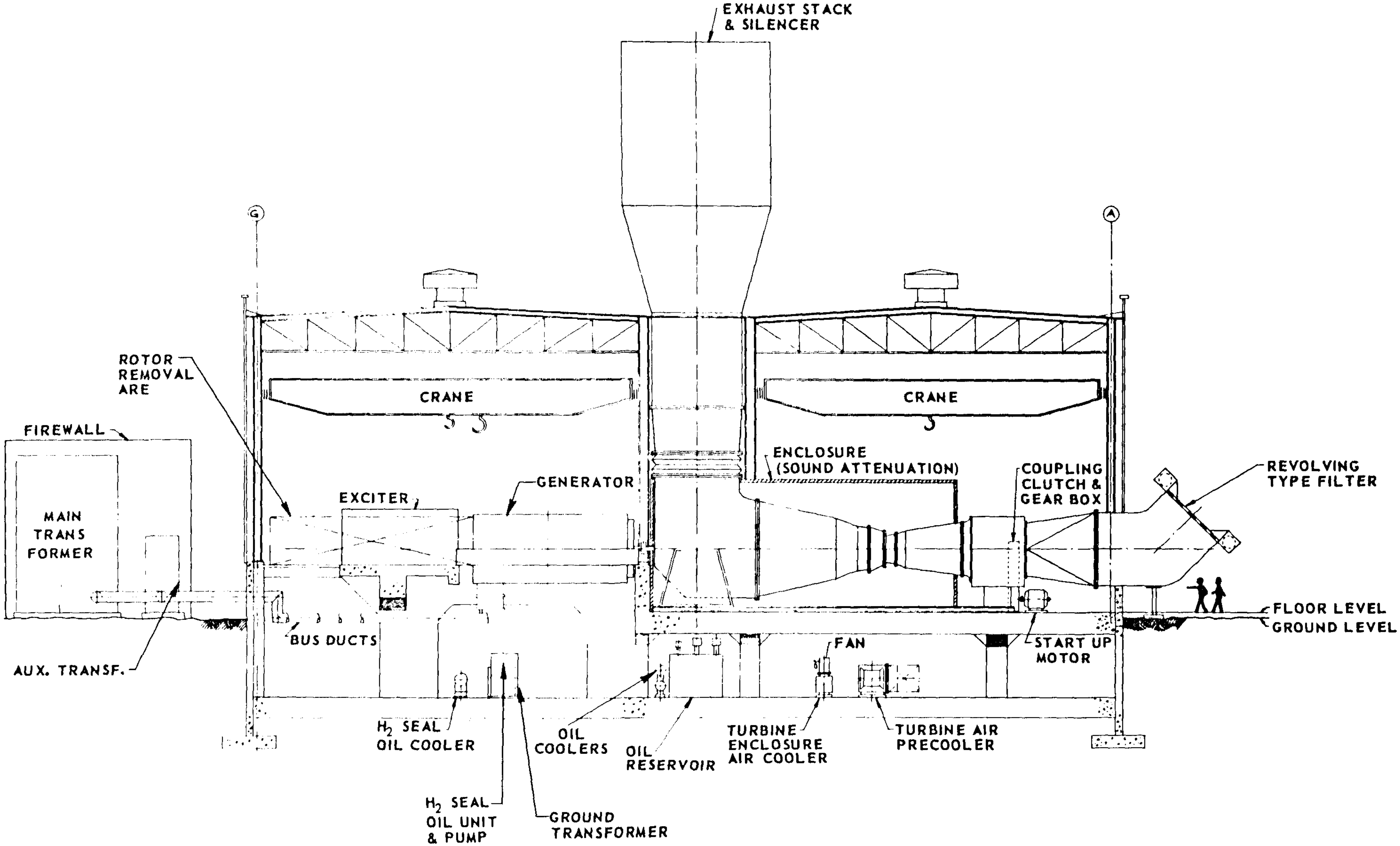


FIG. 55. 1000-MW GAS TURBINE POWER PLANT

PLAN VIEW

EARLY 1980'S TECHNOLOGY

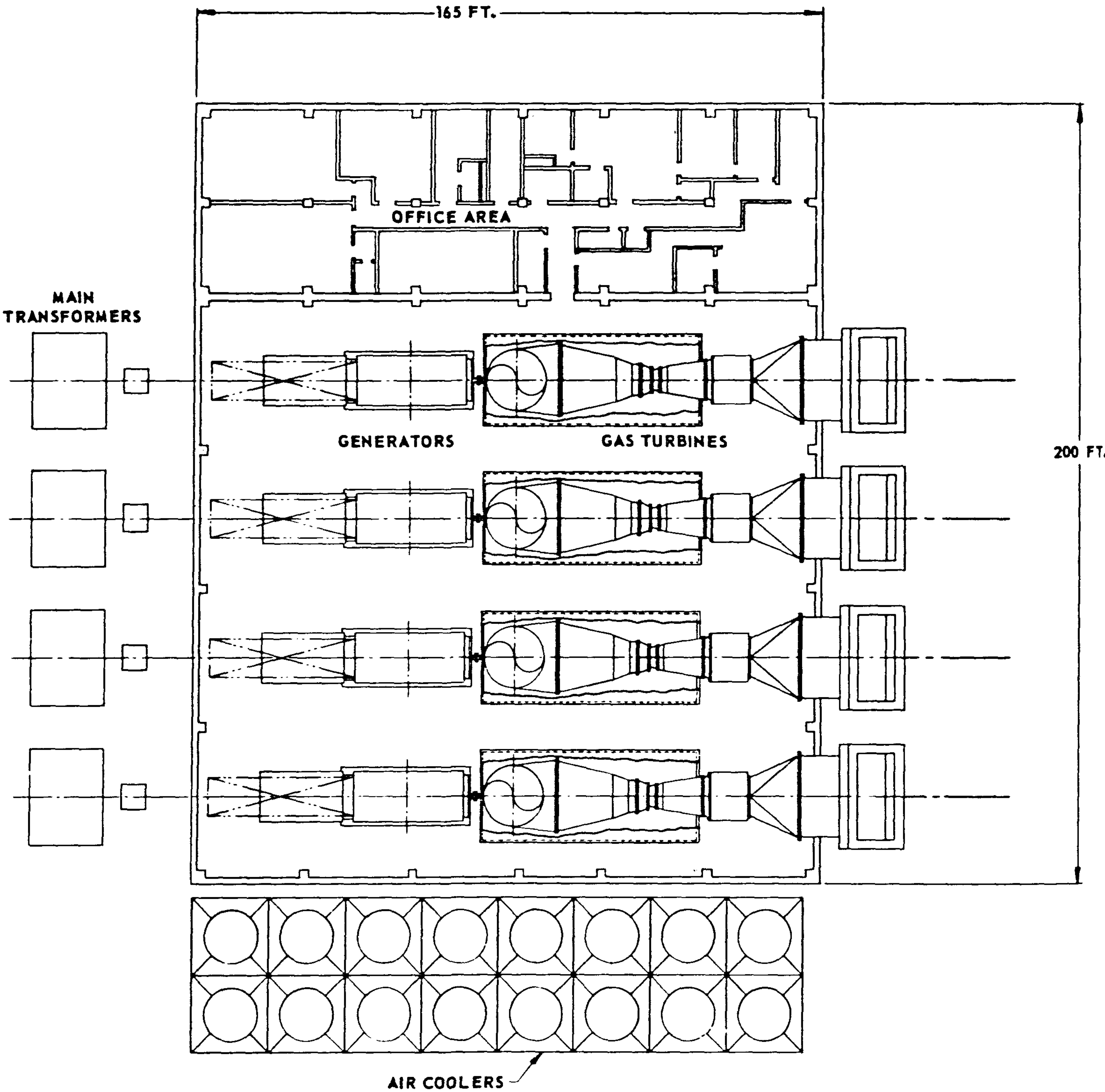


FIG. 56. 1000-MW GAS TURBINE POWER PLANT

NOMINAL 200 MW
TURBINE INLET TEMPERATURE = 2400 F
COMPRESSOR PRESSURE RATIO = 10 TO 1
EARLY 1980'S TECHNOLOGY
COMPRESSOR BLEED AIR PRECOOLED TO 200 F
REGENERATOR TOTAL PRESSURE DROP = 8%
REGENERATOR EFFECTIVENESS = 80%

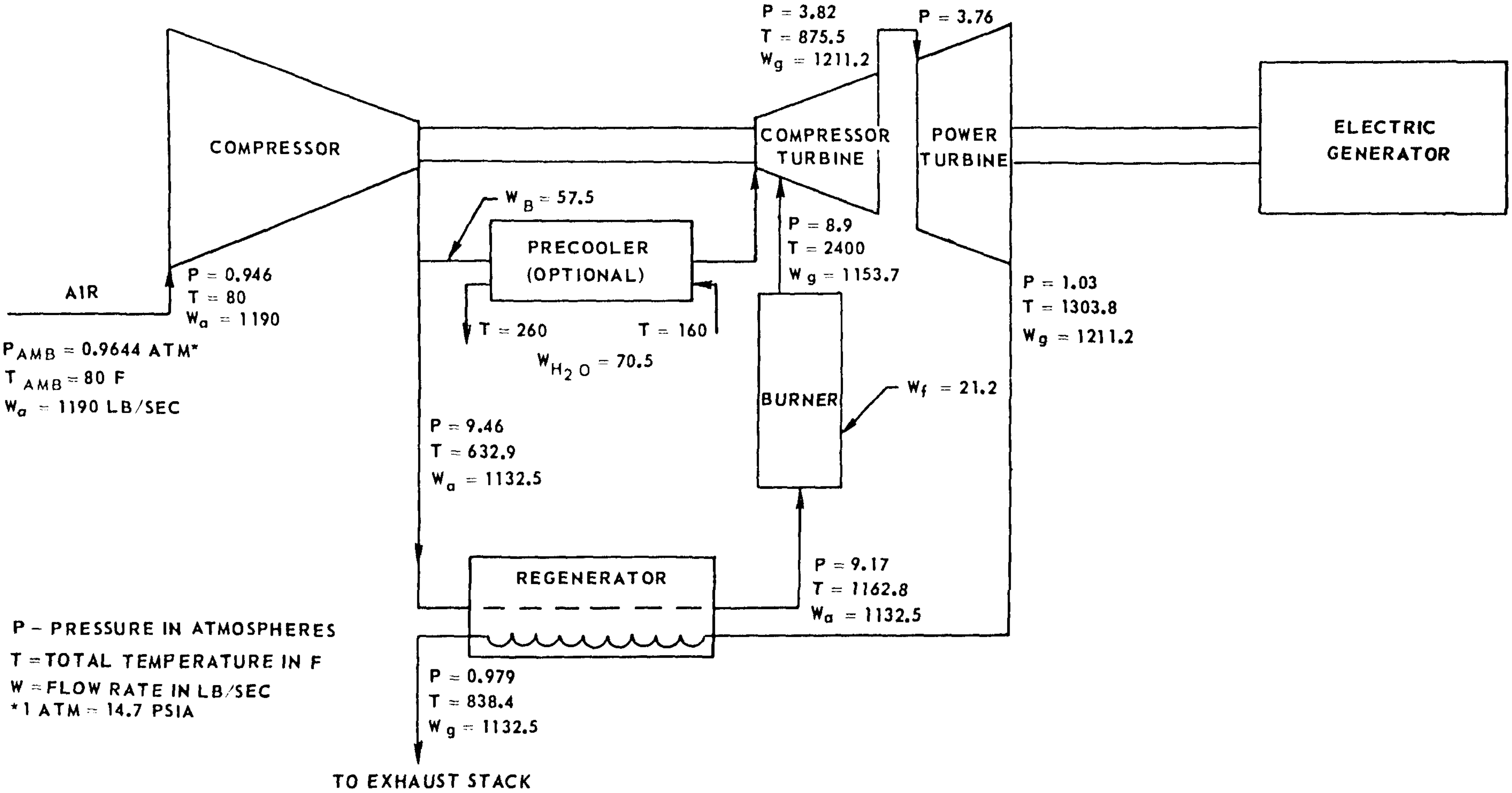


FIG. 57. HEAT BALANCE FOR REGENERATIVE-CYCLE GAS TURBINE

EARLY 1980'S TECHNOLOGY

CAPACITY 200 MW   TURBINE INLET TEMPERATURE 2400F

COMPRESSOR PRESSURE RATIO 12:1   REGENERATOR EFFECTIVENESS 80%

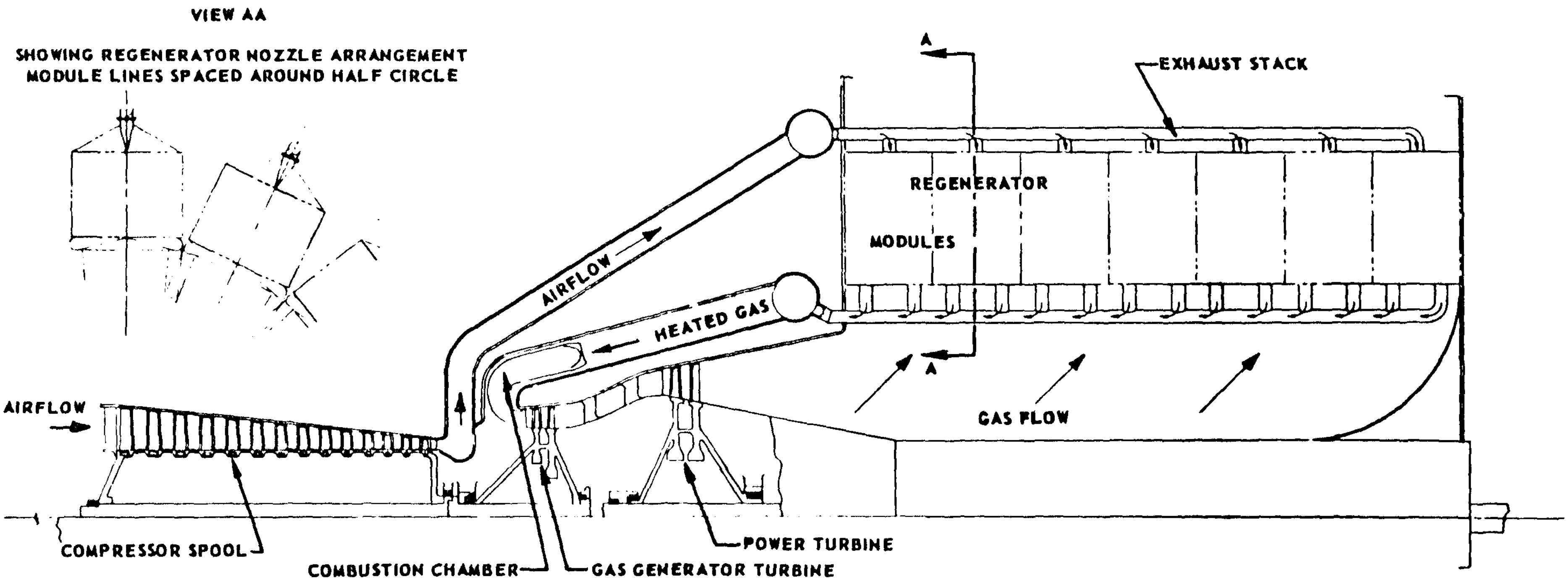


FIG. 58. REGENERATIVE – CYCLE ENGINE FLOW PATH

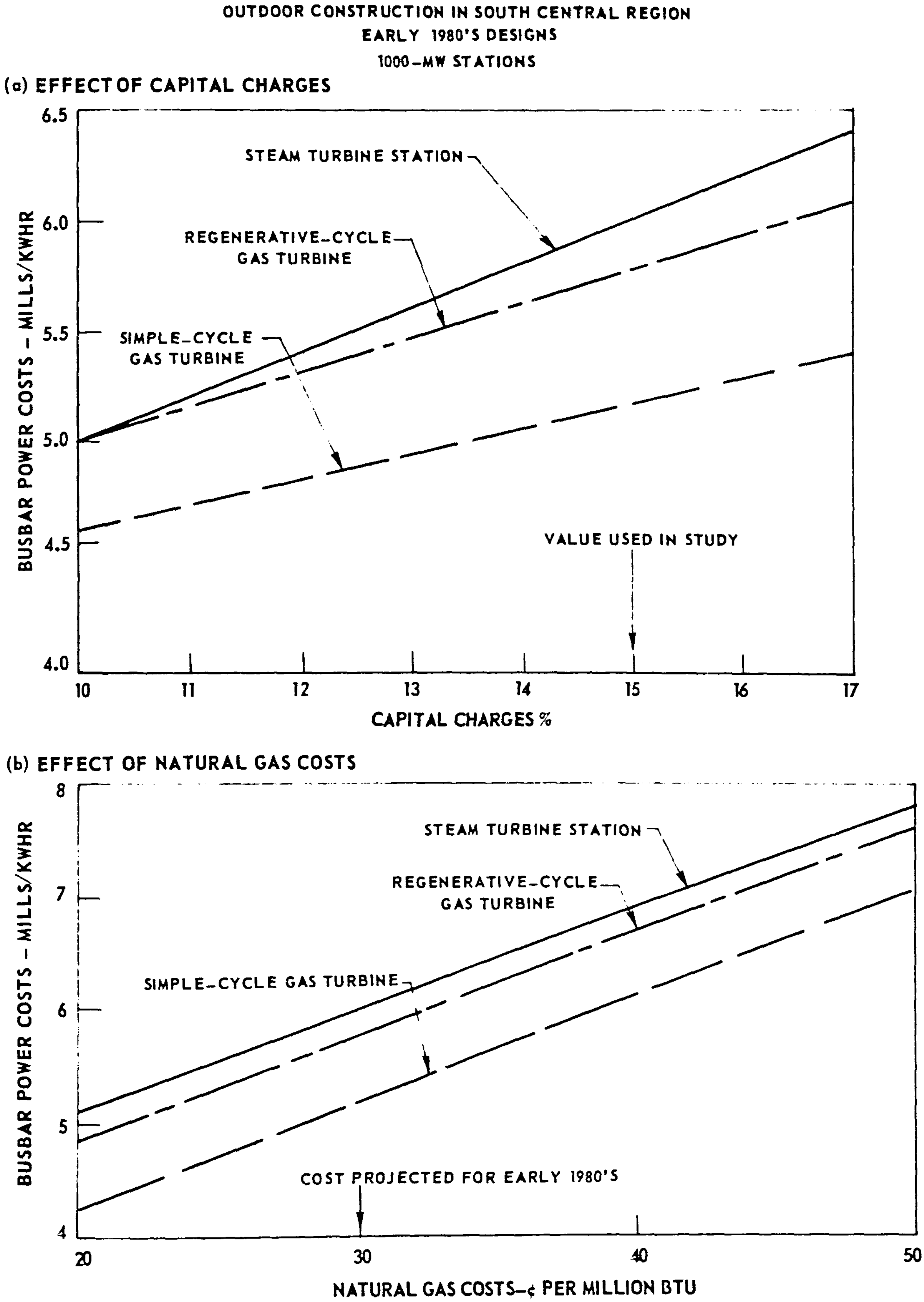
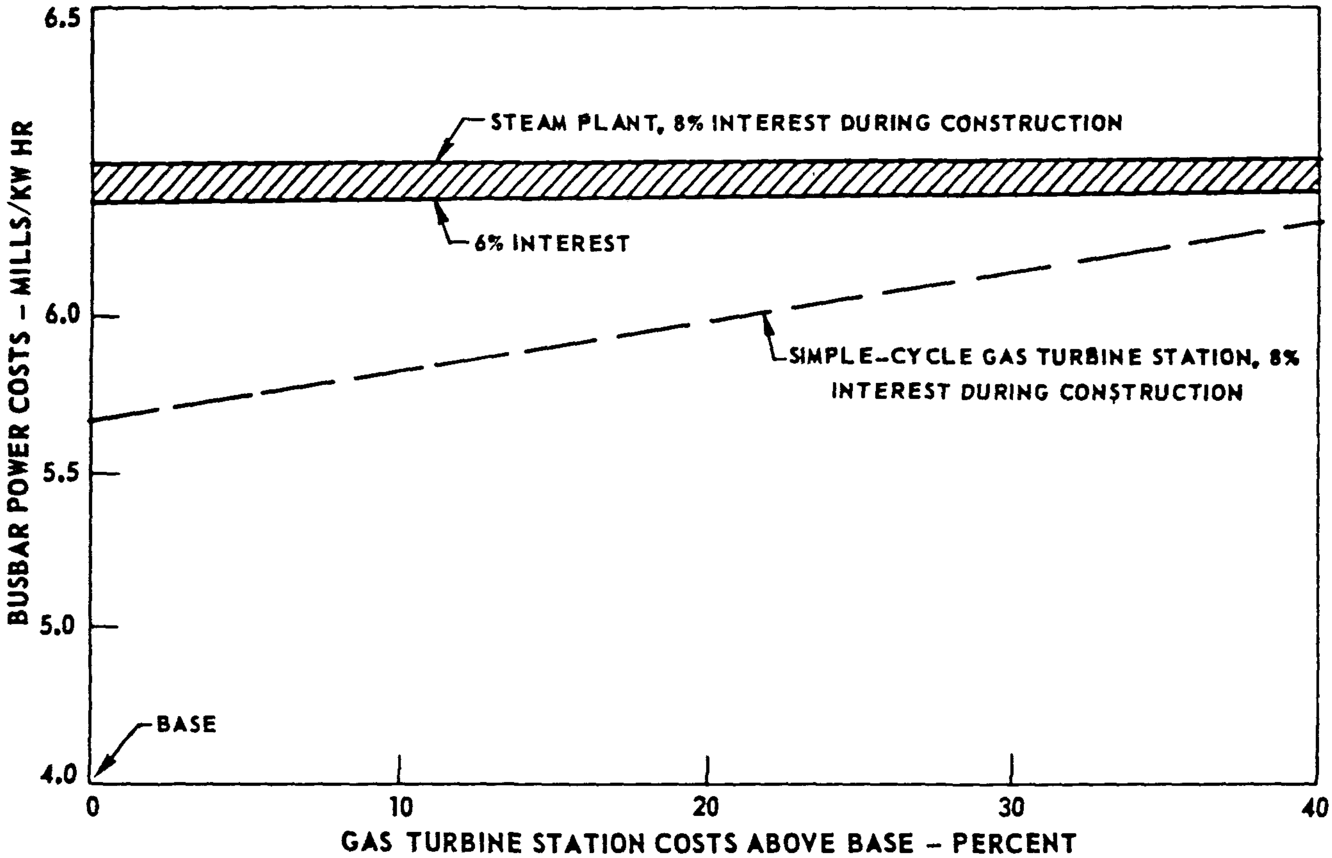


FIG. 59. EFFECT OF CAPITAL CHARGES AND GAS COSTS ON STATION POWER COSTS

OUTDOOR CONSTRUCTION IN SOUTH CENTRAL REGION
EARLY 1980'S DESIGNS
CAPITAL CHARGES 15%
GAS COSTS 30¢ PER MILLION BTU

(a) SENSITIVITY ANALYSIS TO SHOW EFFECT OF INCREASED GAS TURBINE STATION COSTS ON POWER COSTS



(b) SENSITIVITY ANALYSIS TO SHOW EFFECT OF IMPROVED STEAM PLANT PERFORMANCE ON POWER COSTS

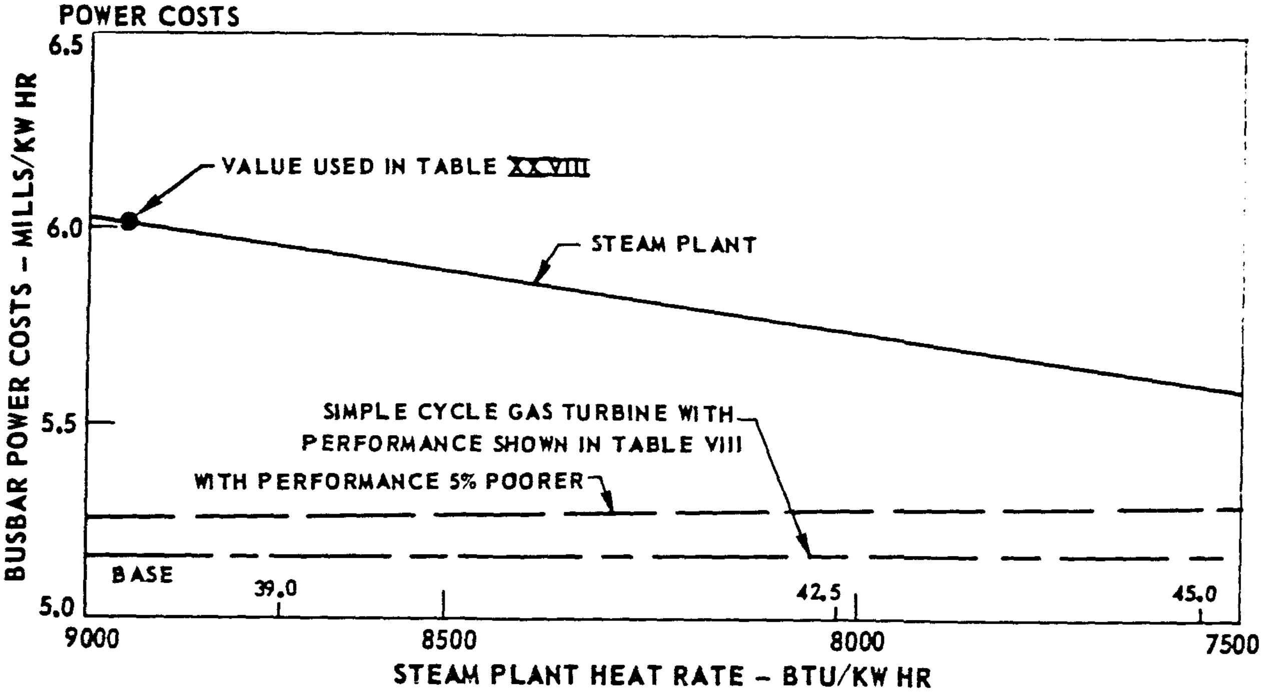


FIG. 60. EFFECT OF GAS TURBINE PERFORMANCE AND COST CHARACTERISTICS ON BUSBAR POWER COSTS

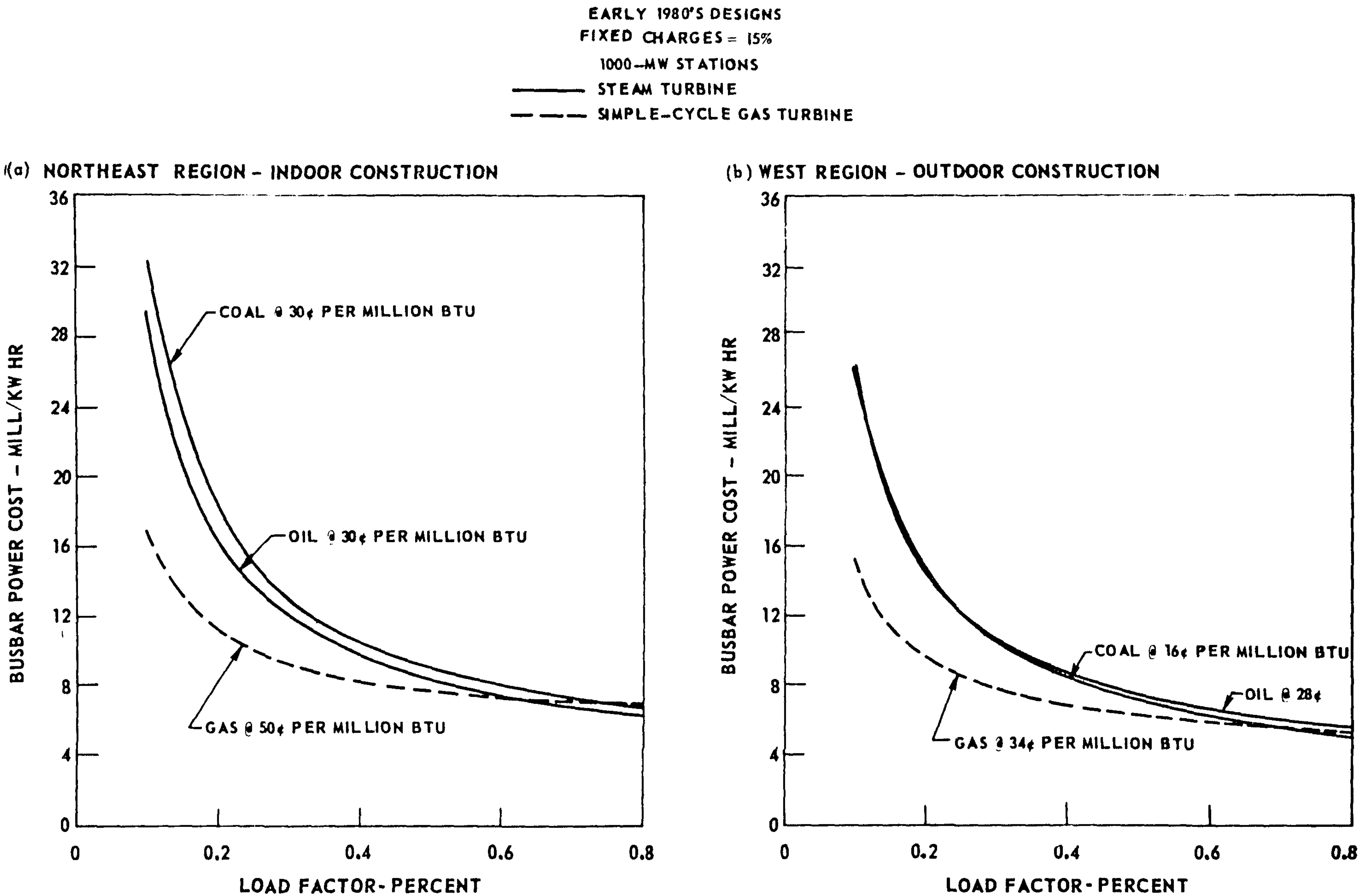


FIG. 61. COMPARISON OF BUSBAR POWER GENERATION COSTS IN SELECTED REGIONS

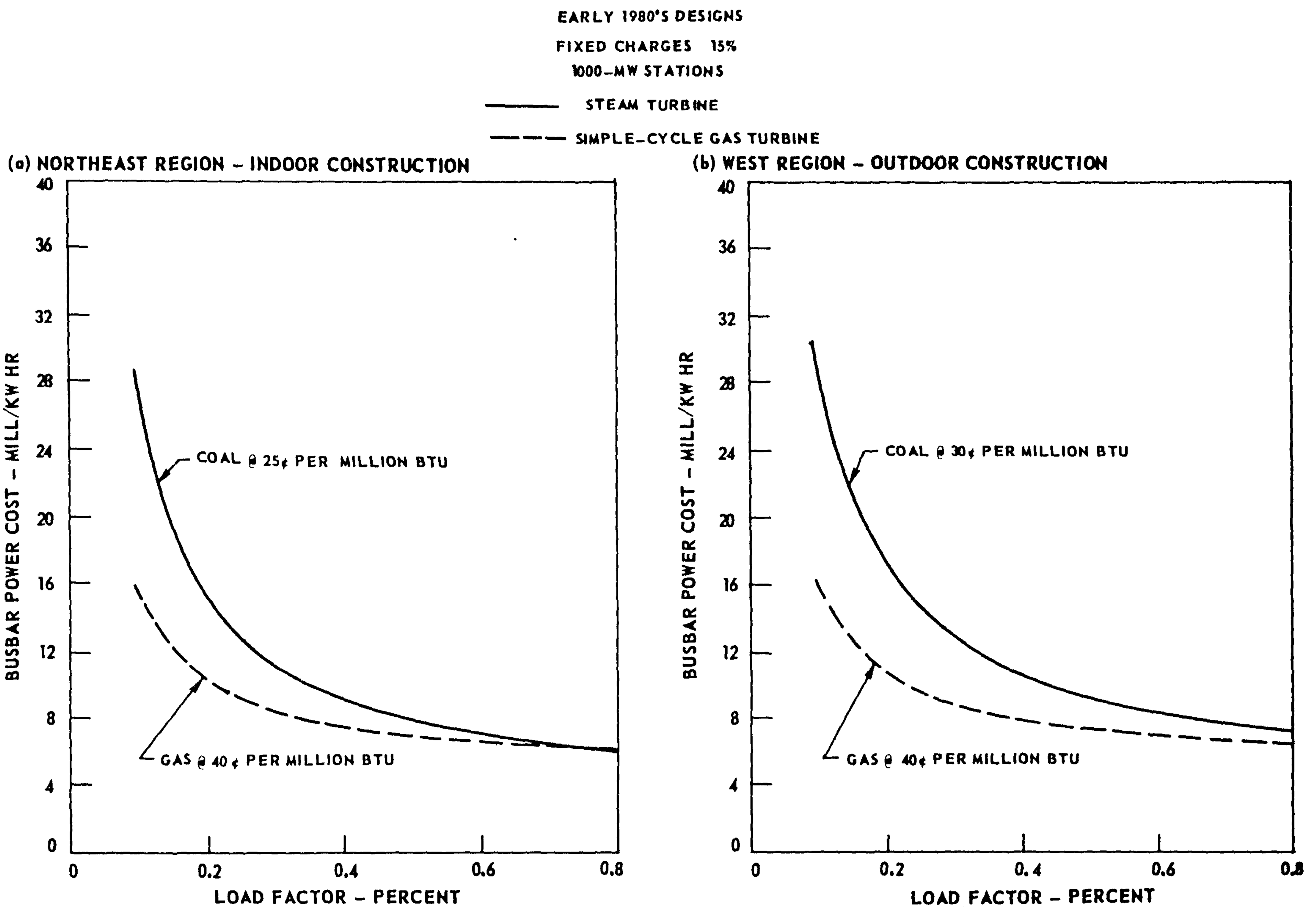


FIG. 62. COMPARISON OF BUSBAR POWER GENERATION COSTS IN SELECTED REGIONS

FORCED OUTAGE RATE = 0.02

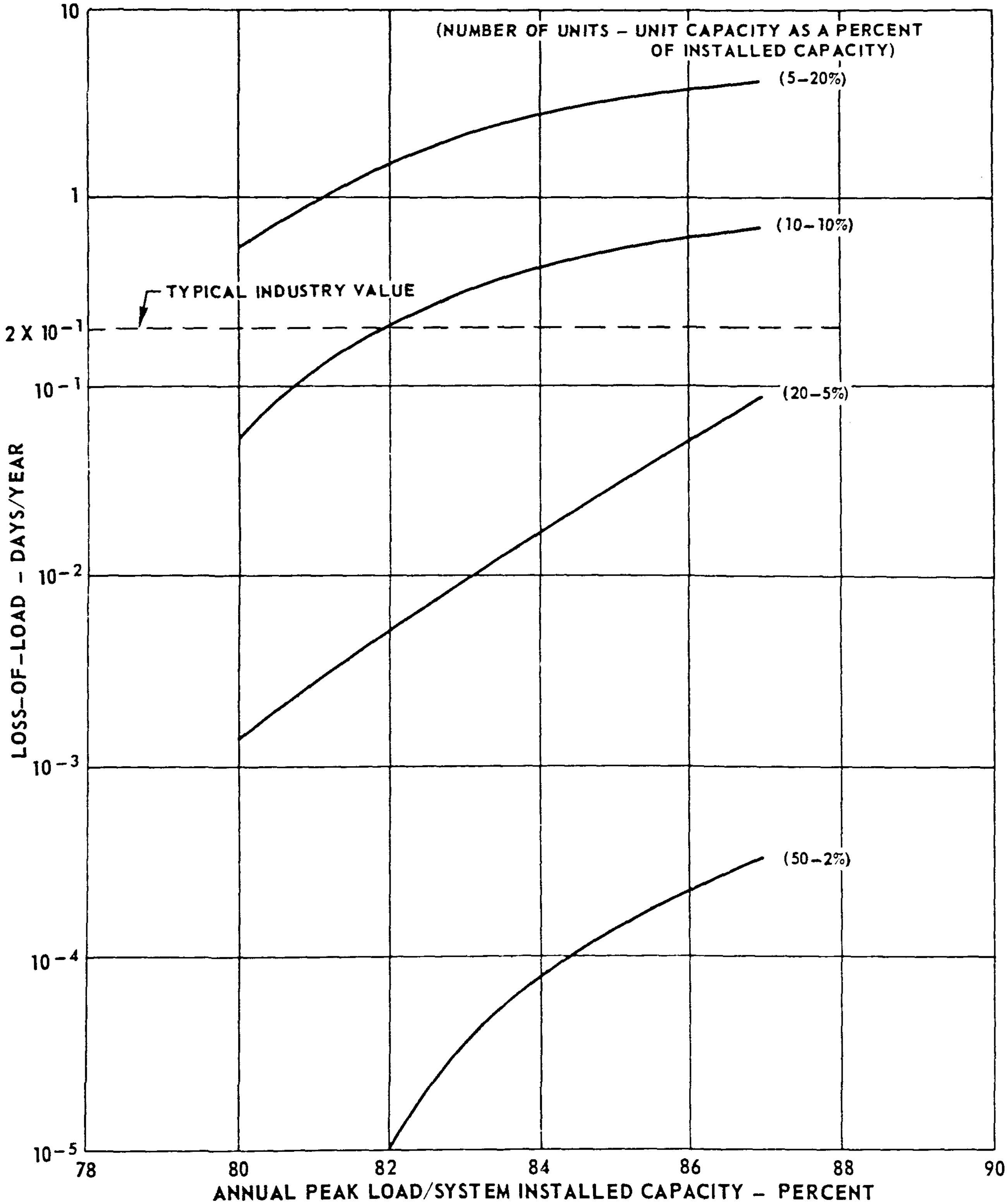


FIG. 63. EFFECT OF UNIT SIZE ON LOSS-OF-LOAD

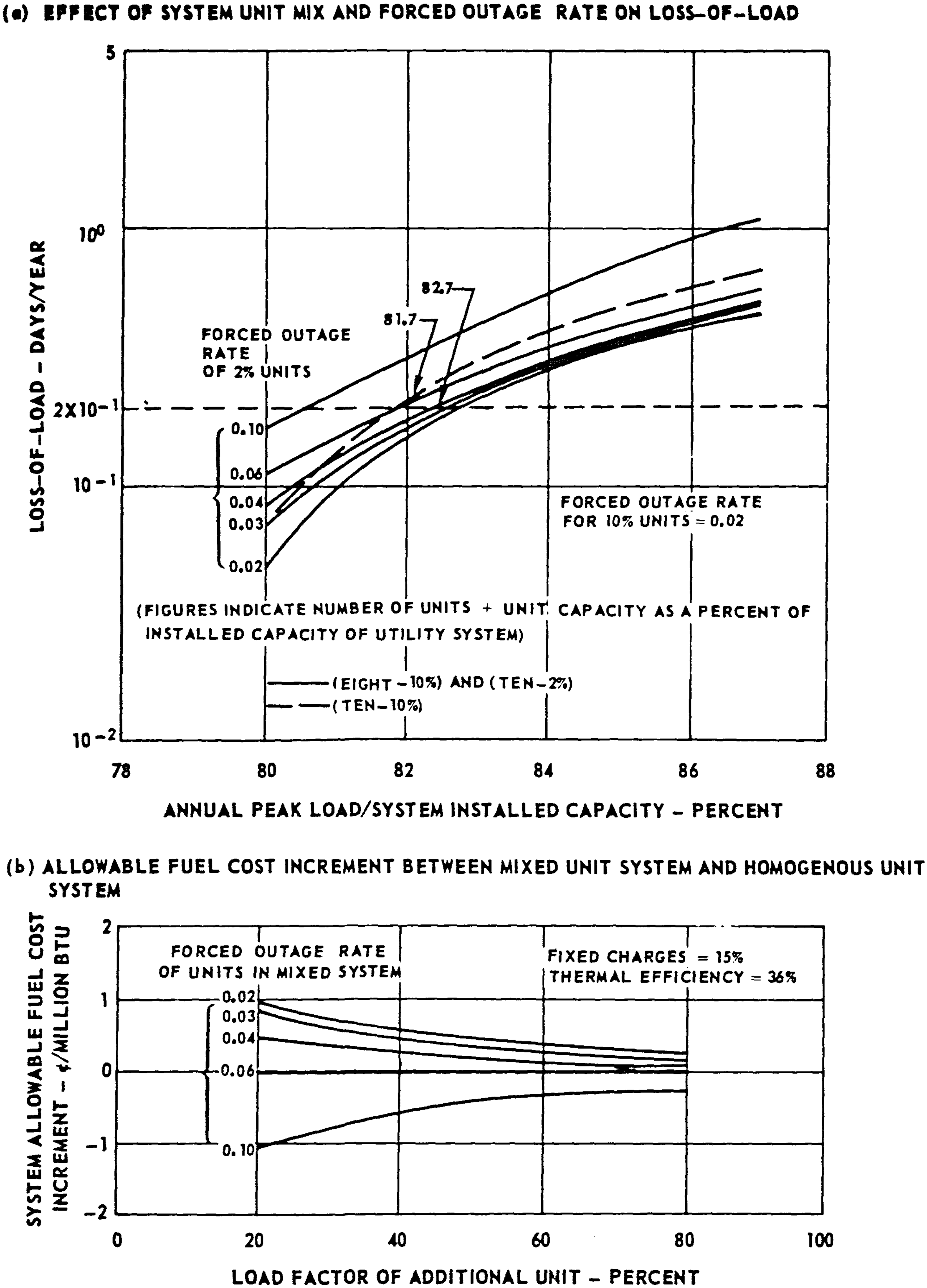
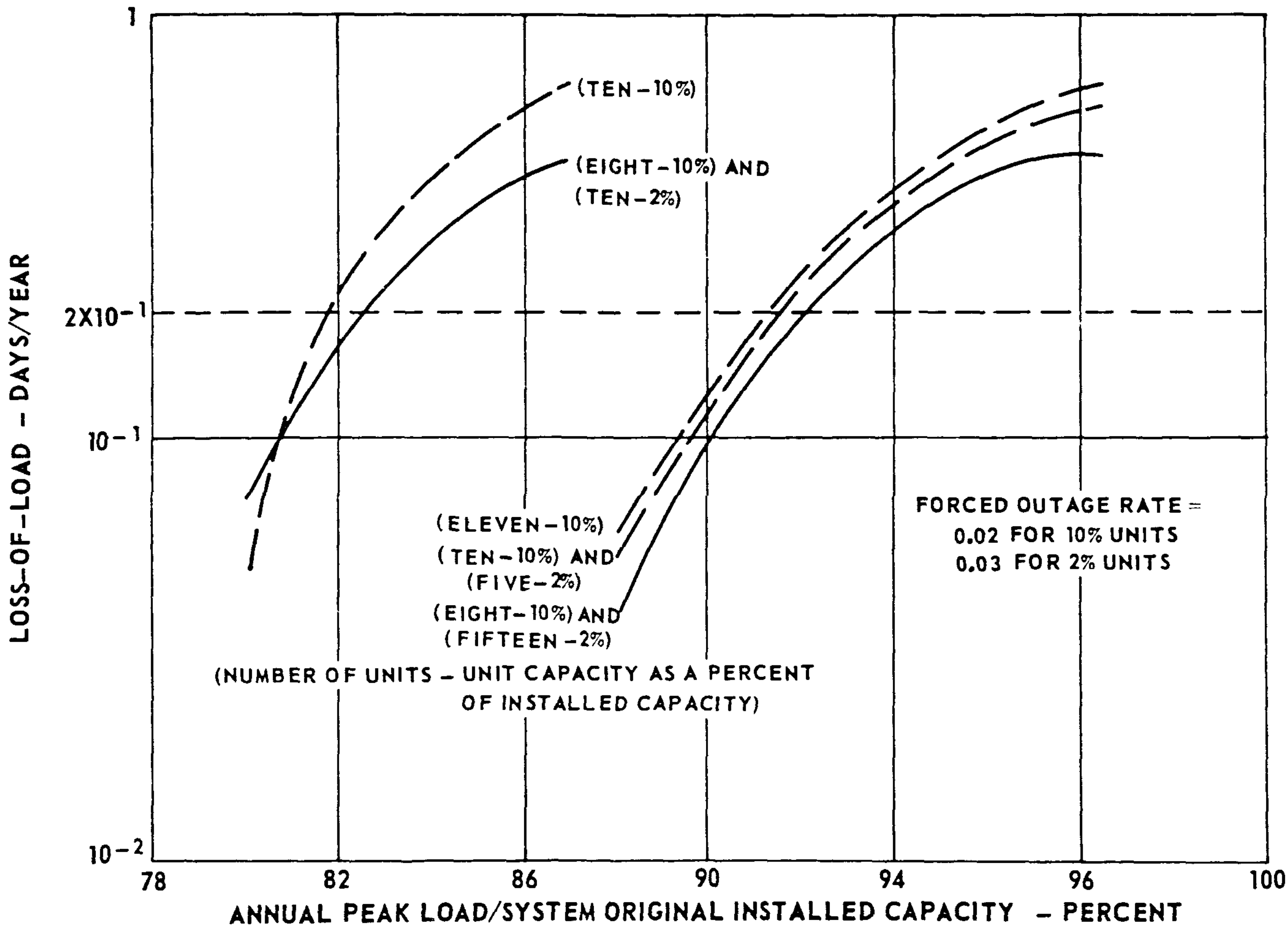


FIG. 64. EFFECT OF SYSTEM UNIT CAPACITY COMPOSITION ON RELIABILITY AND FUEL COST

(a) EFFECT OF EXPANSION UNIT COMPOSITION ON LOSS-OF-LOAD



b) ALLOWABLE FUEL COST INCREMENT BETWEEN MIXED UNIT SYSTEM AND HOMOGENOUS UNIT SYSTEM

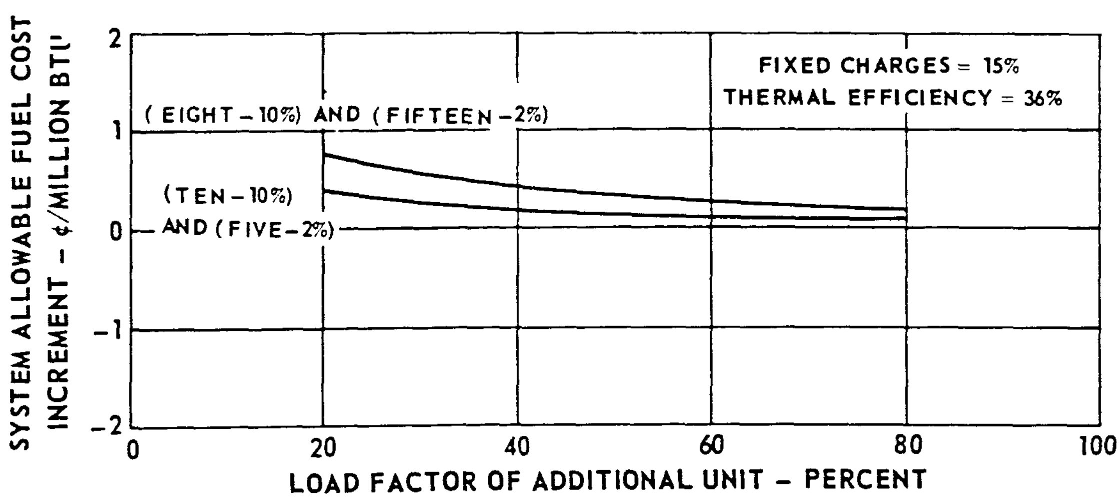
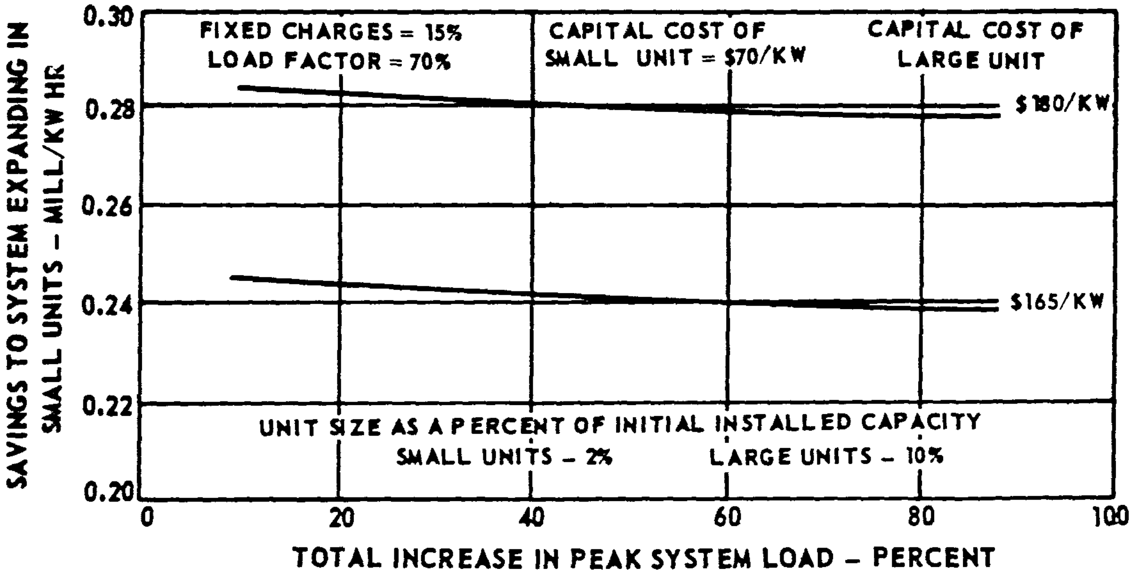


FIG. 65. EFFECT OF EXPANSION AND SYSTEM COMPOSITION ON RELIABILITY AND FUEL COST

a) AVERAGE CREDIT DUE TO SYSTEM EXPANSION WITH SMALL SIZED GENERATING UNITS WHICH MORE CLOSELY MATCH THE DEMAND CURVE



b) ALLOWABLE FUEL COST INCREMENT RESULTING FROM EXPANDING UTILITY SYSTEM IN SMALL UNITS

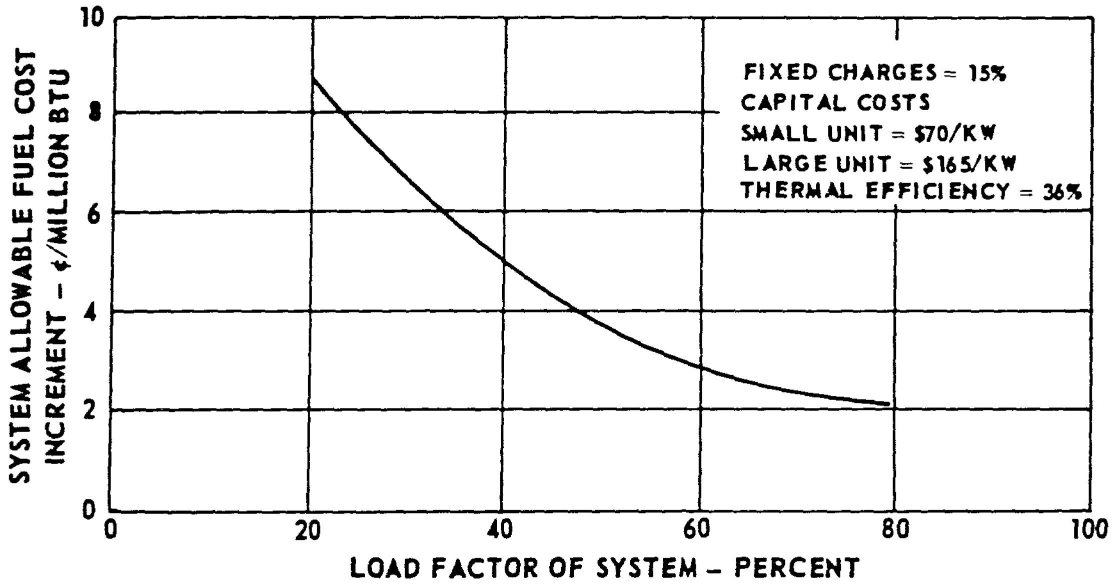
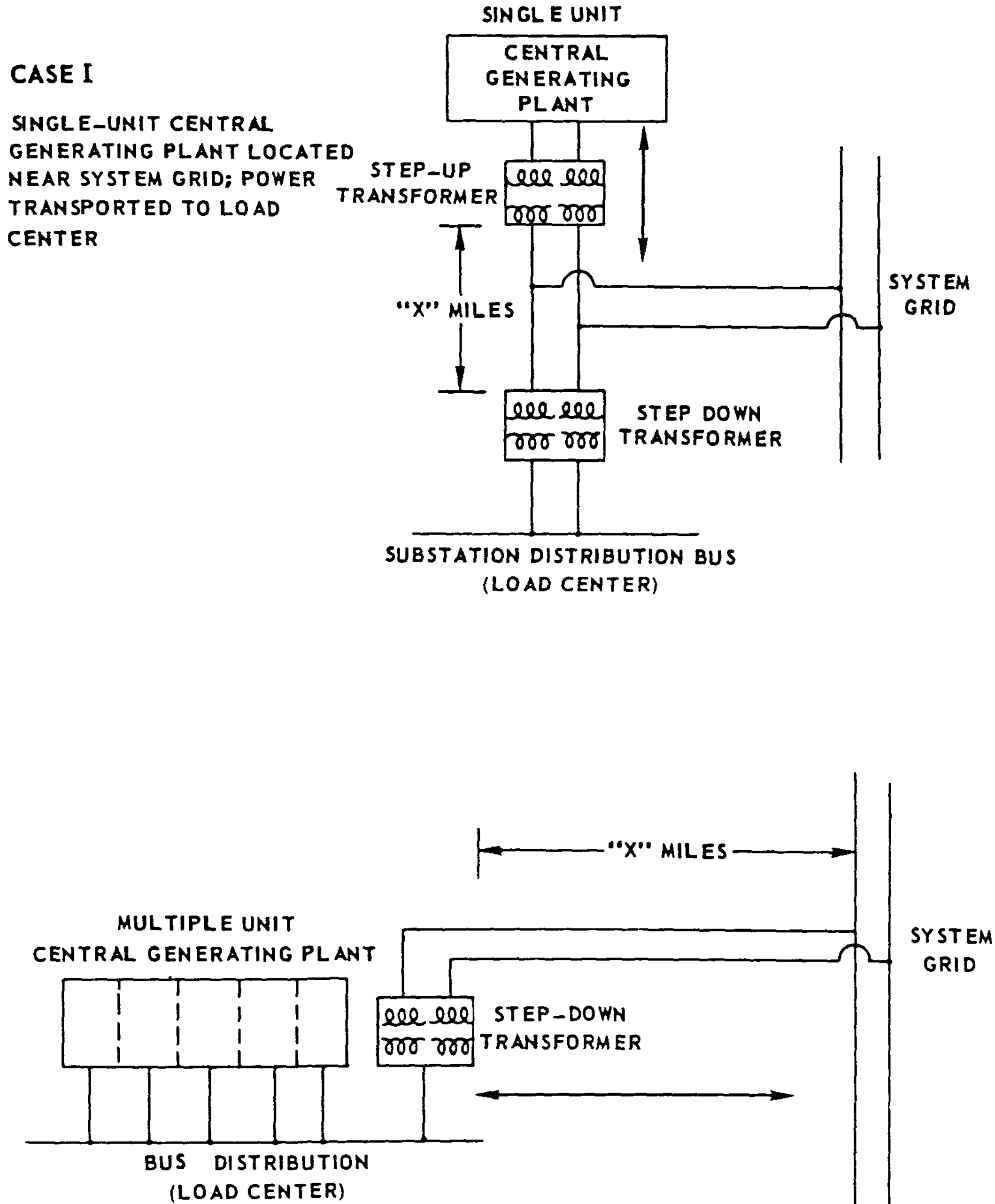


FIG. 66. EFFECT OF EXPANSION AND SYSTEM UNIT SIZE ON RELIABILITY AND FUEL COST



CASE II

MULTIPLE-UNIT CENTRAL GENERATING PLANT LOCATED NEAR LOAD CENTER; RESERVE POWER TRANSPORTED TO LOAD CENTER

FIG. 67. POWER TRANSMISSION SYSTEMS

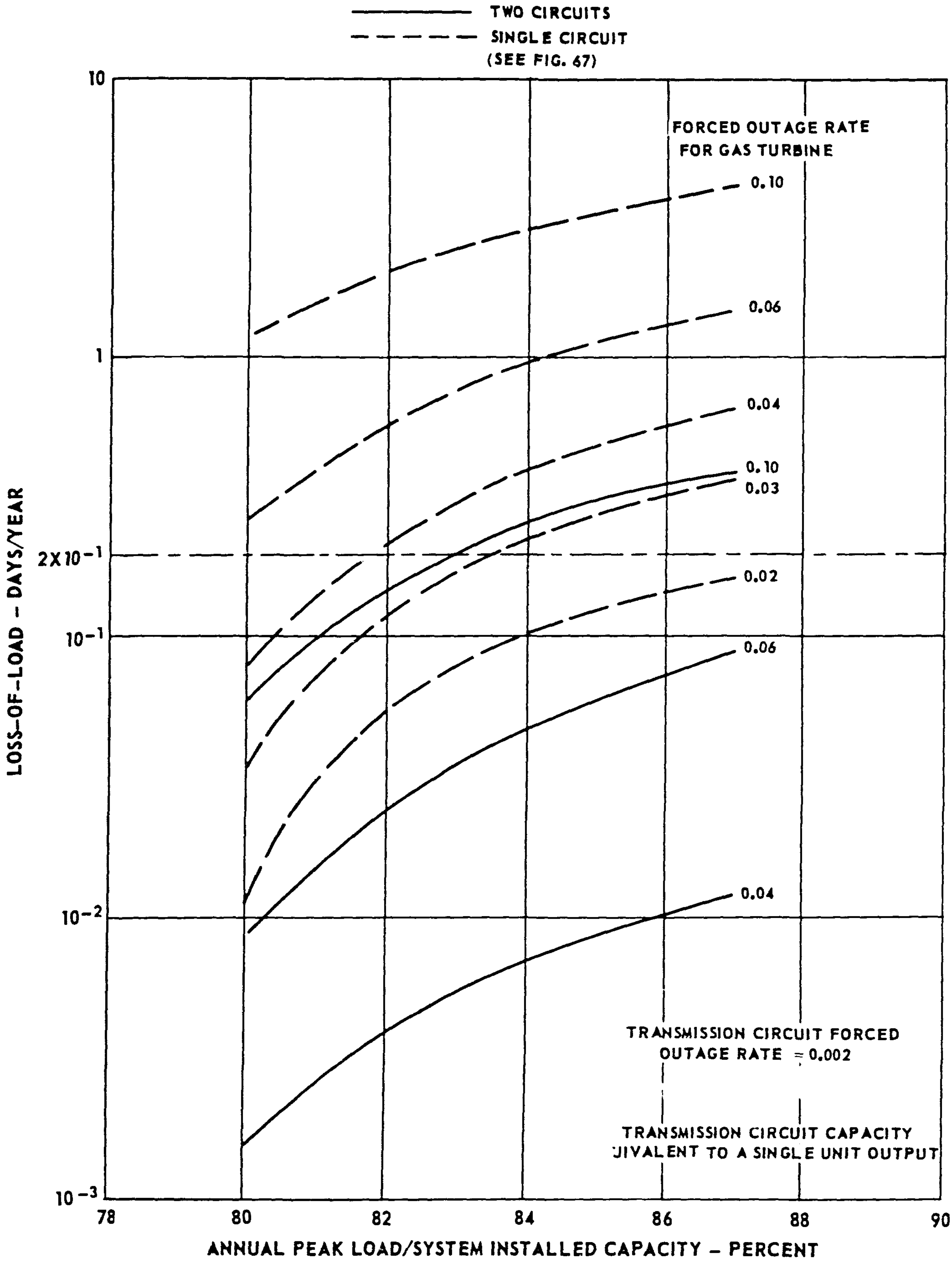


FIG. 68. EFFECT OF FORCED OUTAGE RATE ON LOSS-OF-LOAD

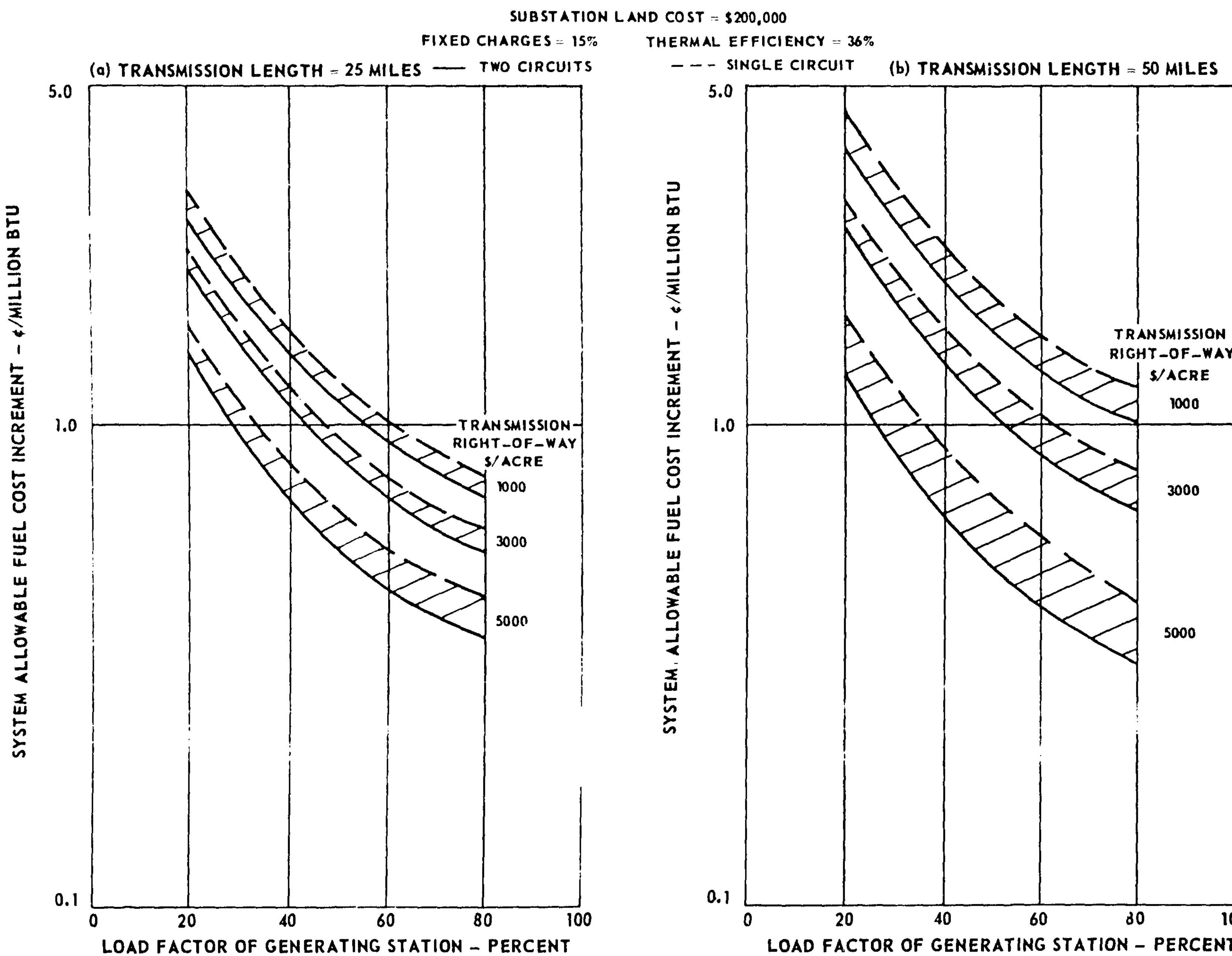


FIG. 69. ALLOWABLE FUEL COST INCREMENT RESULTING FROM THE REDUCTION IN TRANSMISSION REQUIREMENTS

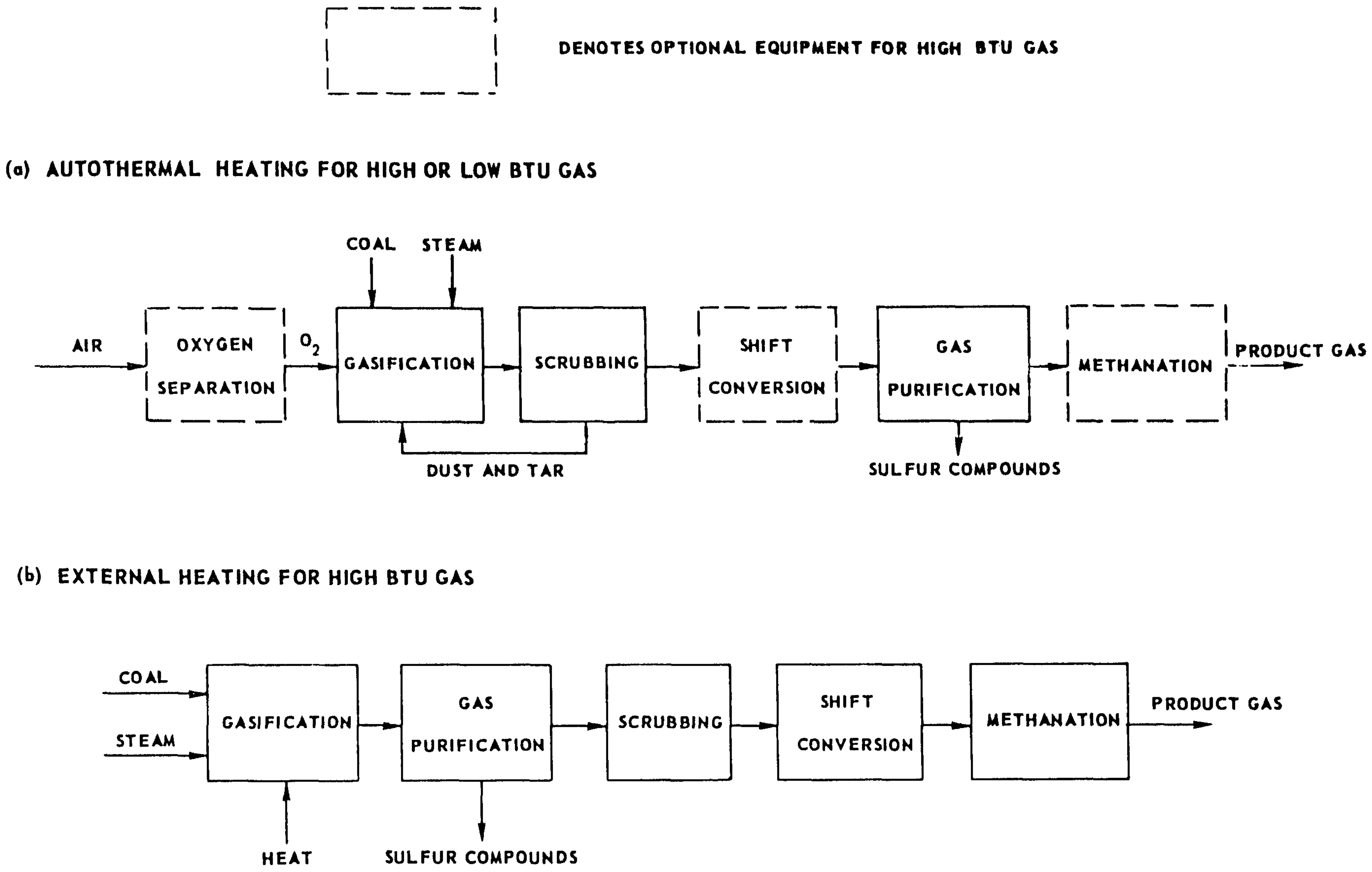


FIG. 70. SIMPLIFIED SCHEMATIC DIAGRAMS FOR COAL GASIFICATION PROCESSES

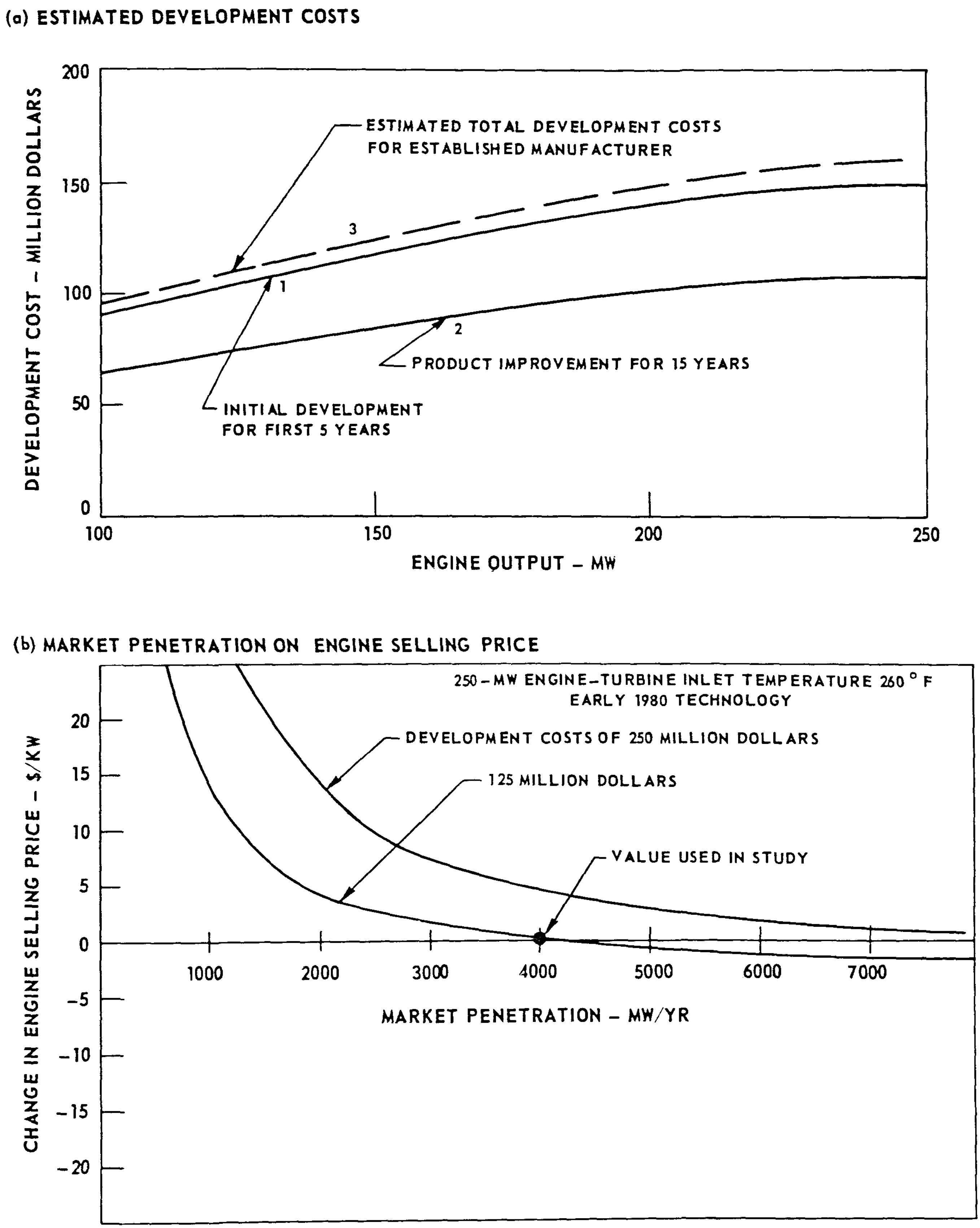


FIG. 71. ESTIMATED DEVELOPMENT COSTS OF ADVANCED GAS TURBINES

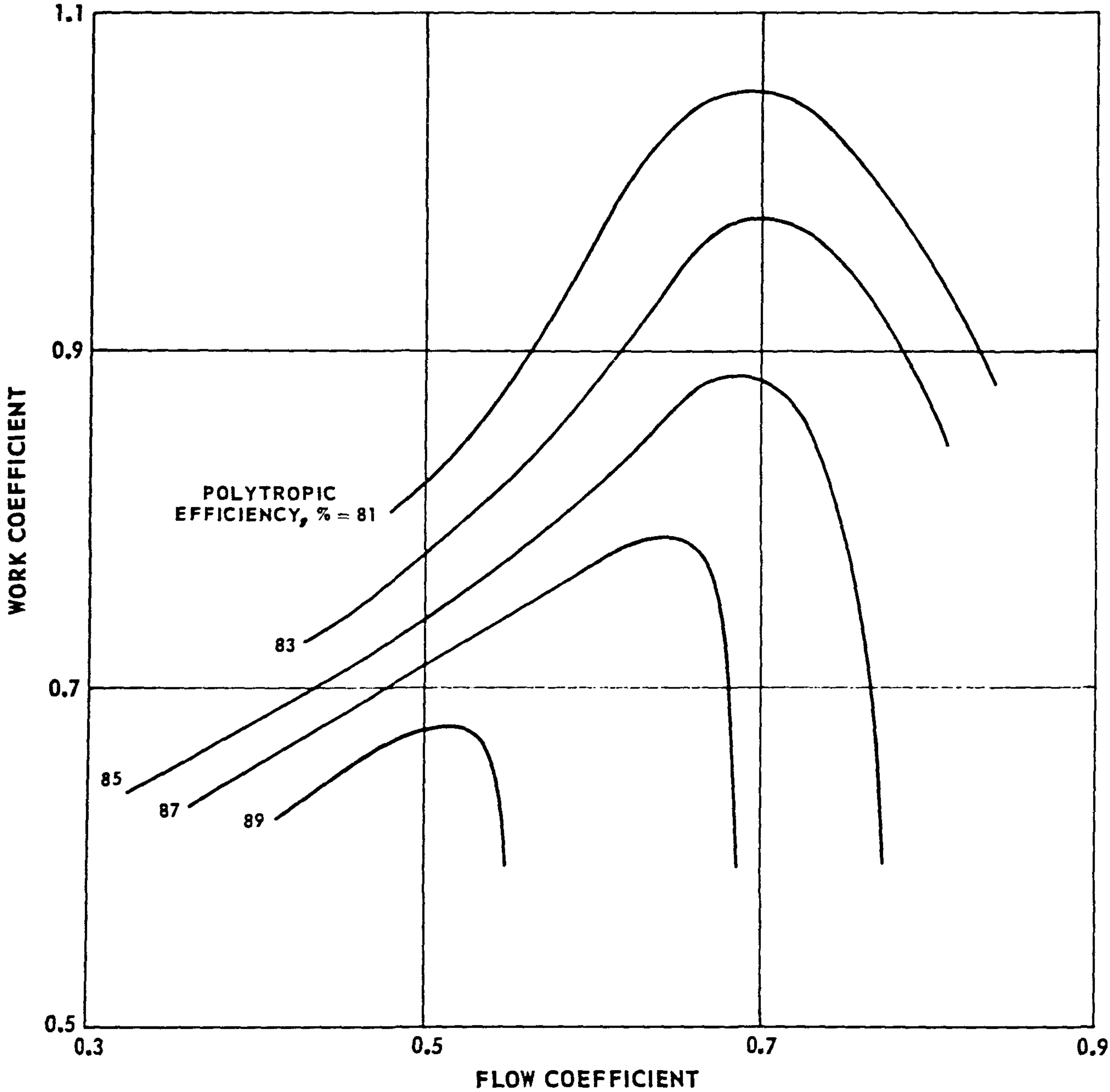


FIG. 72. TYPICAL MULTISTAGE COMPRESSOR EFFICIENCY

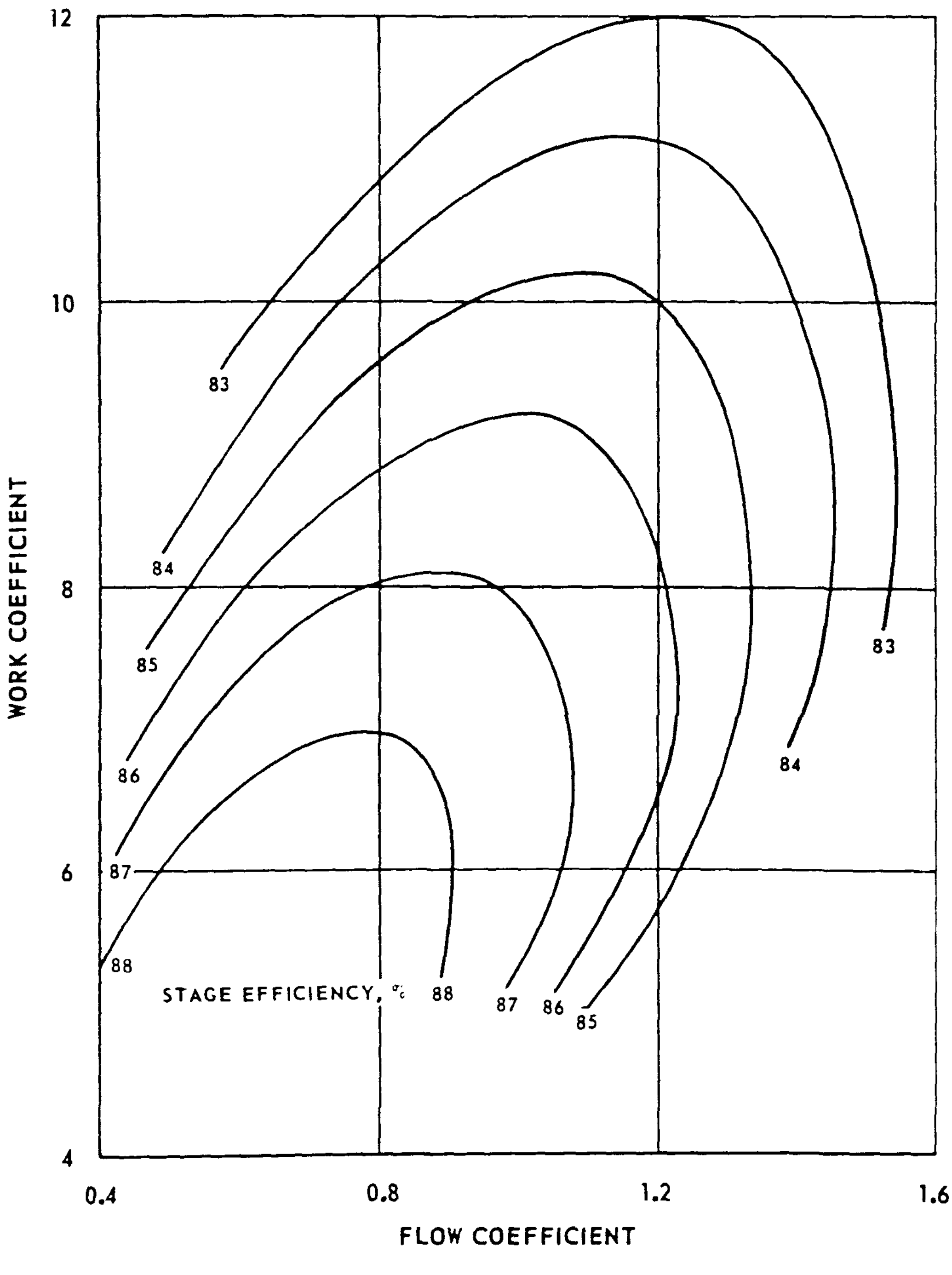


FIG. 73. TYPICAL HIGH-PRESSURE TURBINE PERFORMANCE

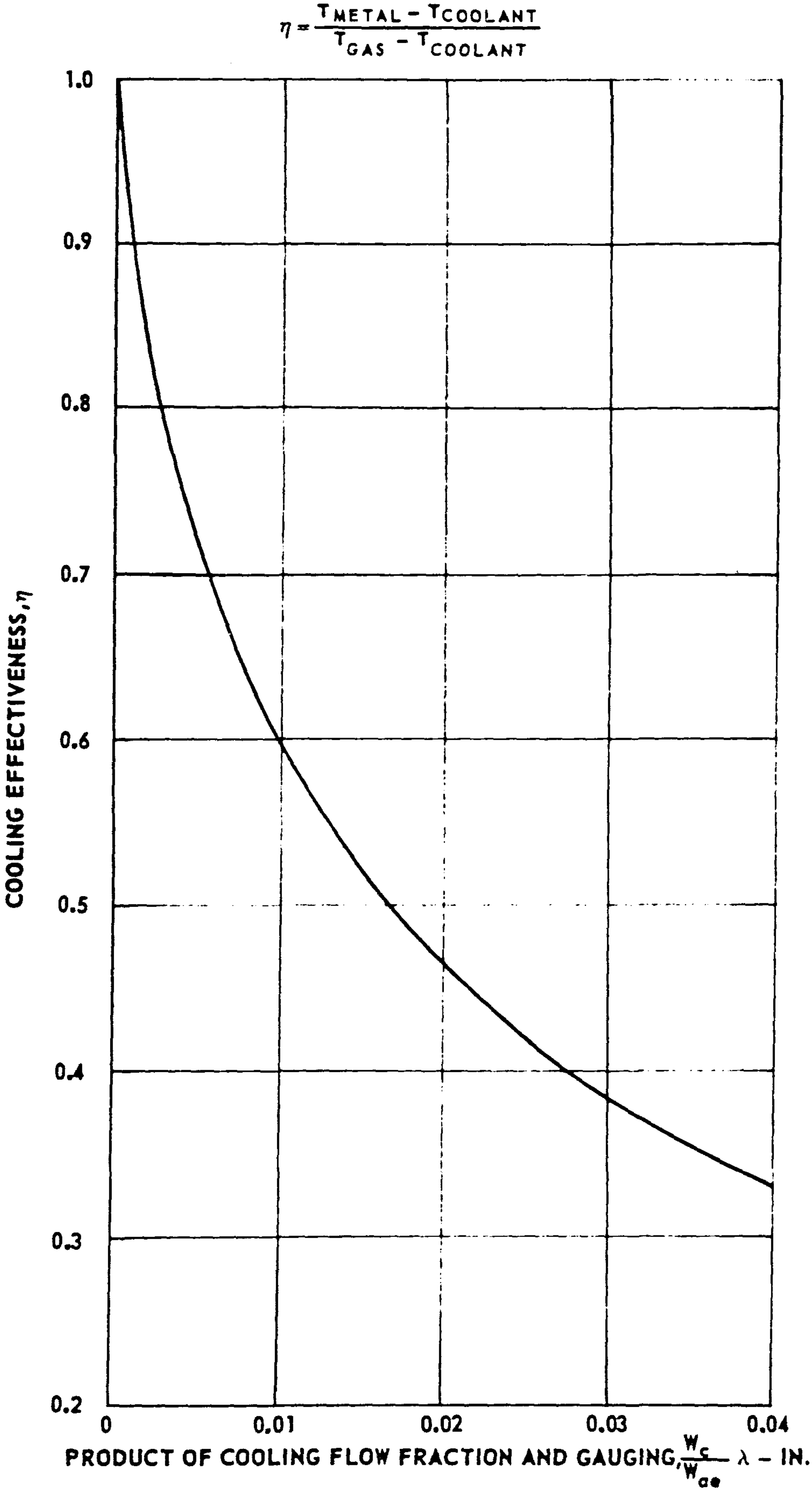


FIG. 74. COOLING EFFECTIVENESS CORRELATION FOR ADVANCED IMPINGEMENT-CONVECTION COOLED BLADES

TWIN - SPOOL CONFIGURATION

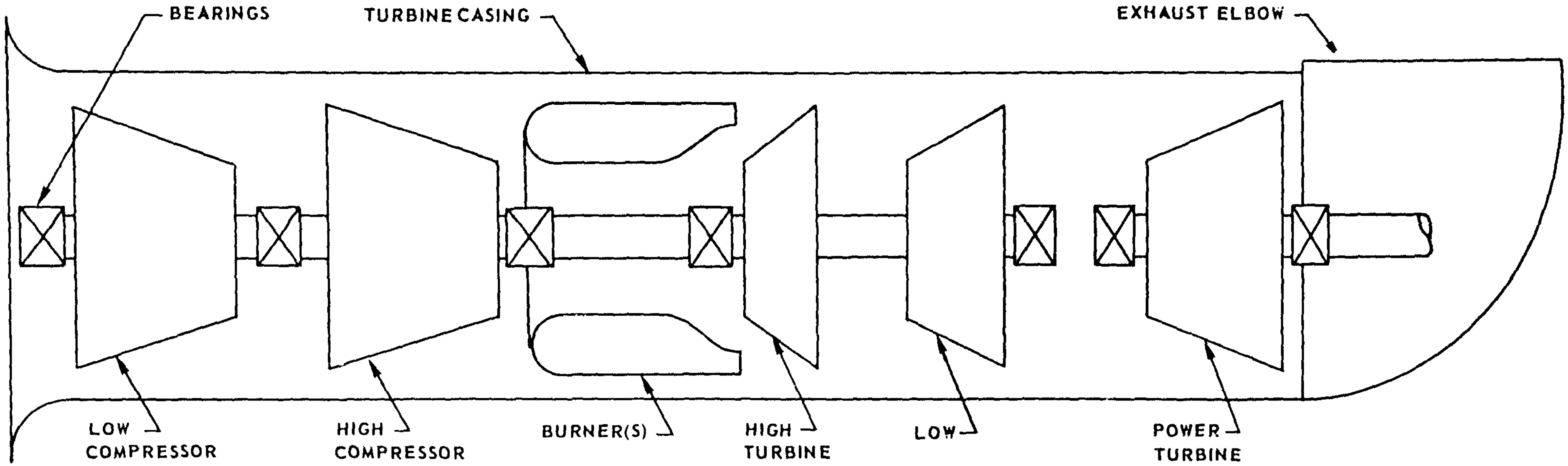


FIG. 75. SCHEMATIC DIAGRAM OF MODEL USED IN GAS TURBINE COST ANALYSIS

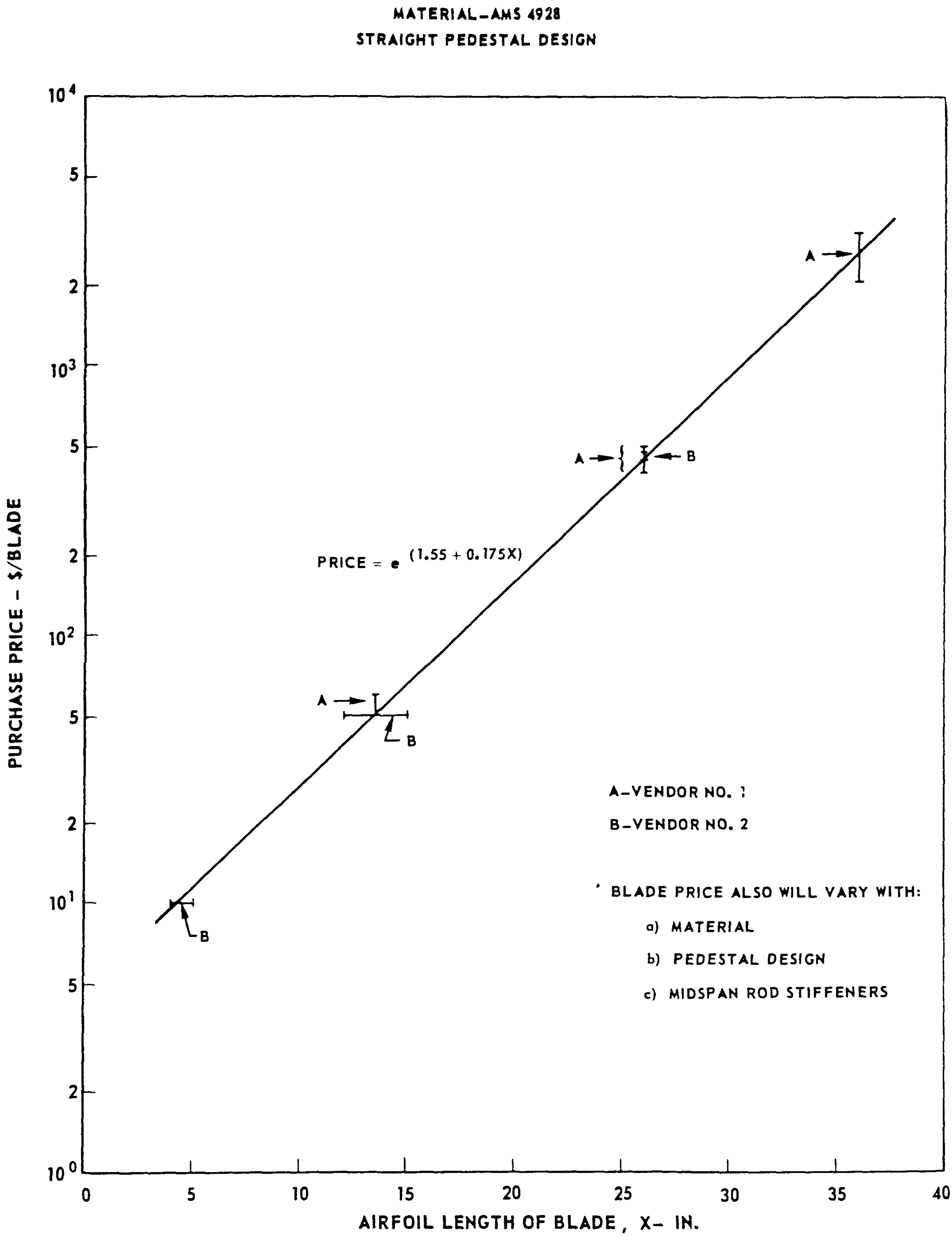


FIG. 76. MANUFACTURERS' PRICES FOR SOLID COMPRESSOR AND TURBINE BLADES

NO CORRECTION FOR PRODUCTION VOLUME

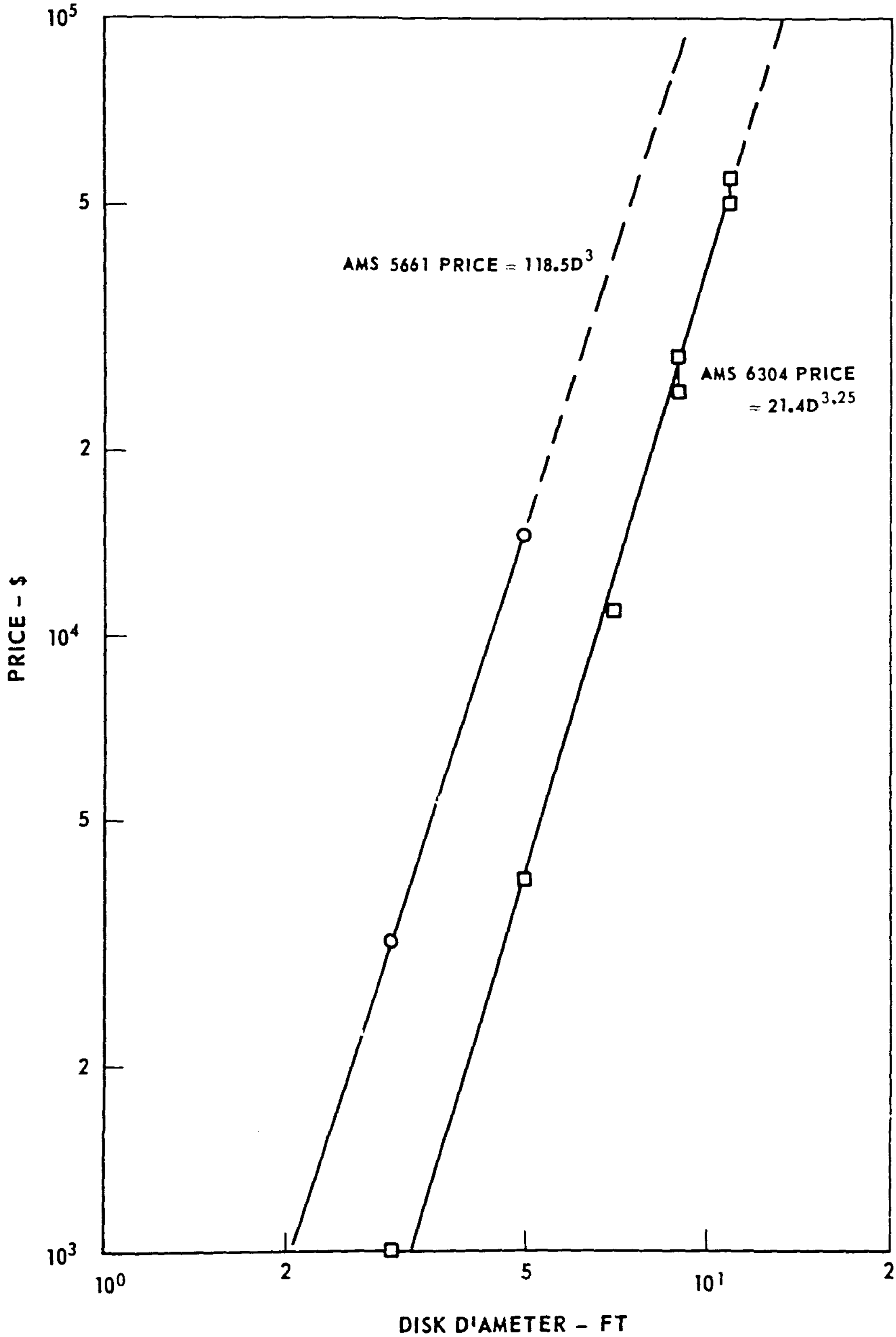


FIG. 77. PRICE ESTIMATES FOR FORGED COMPRESSOR DISKS

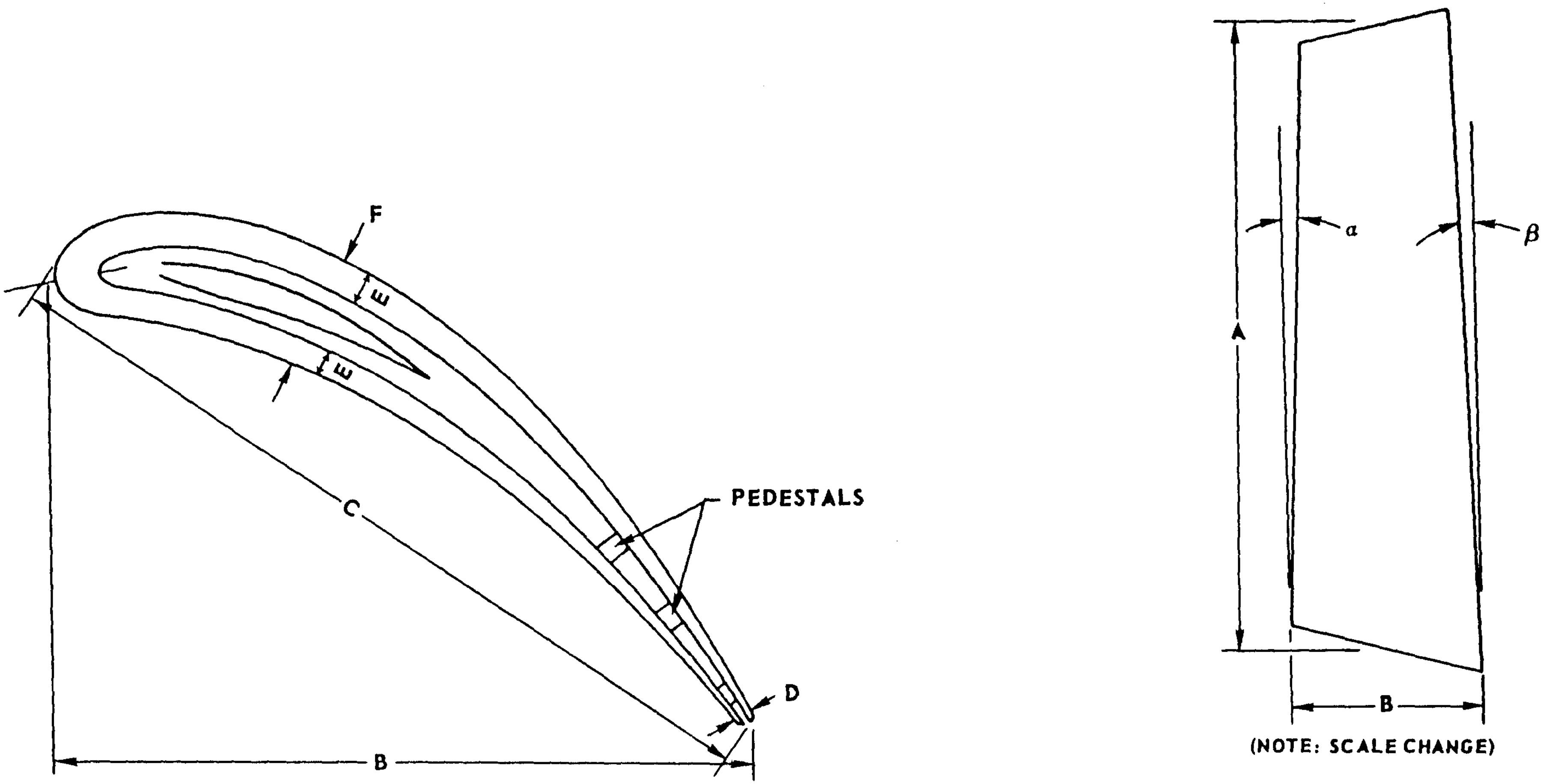


FIG. 78. ILLUSTRATION OF TYPICAL IMPINGEMENT-COOLED TURBINE BLADE DRAWINGS SENT TO BLADE MANUFACTURERS

MATERIAL – B – 1900

AMS 5661

TOOLING COSTS NOT INCLUDED

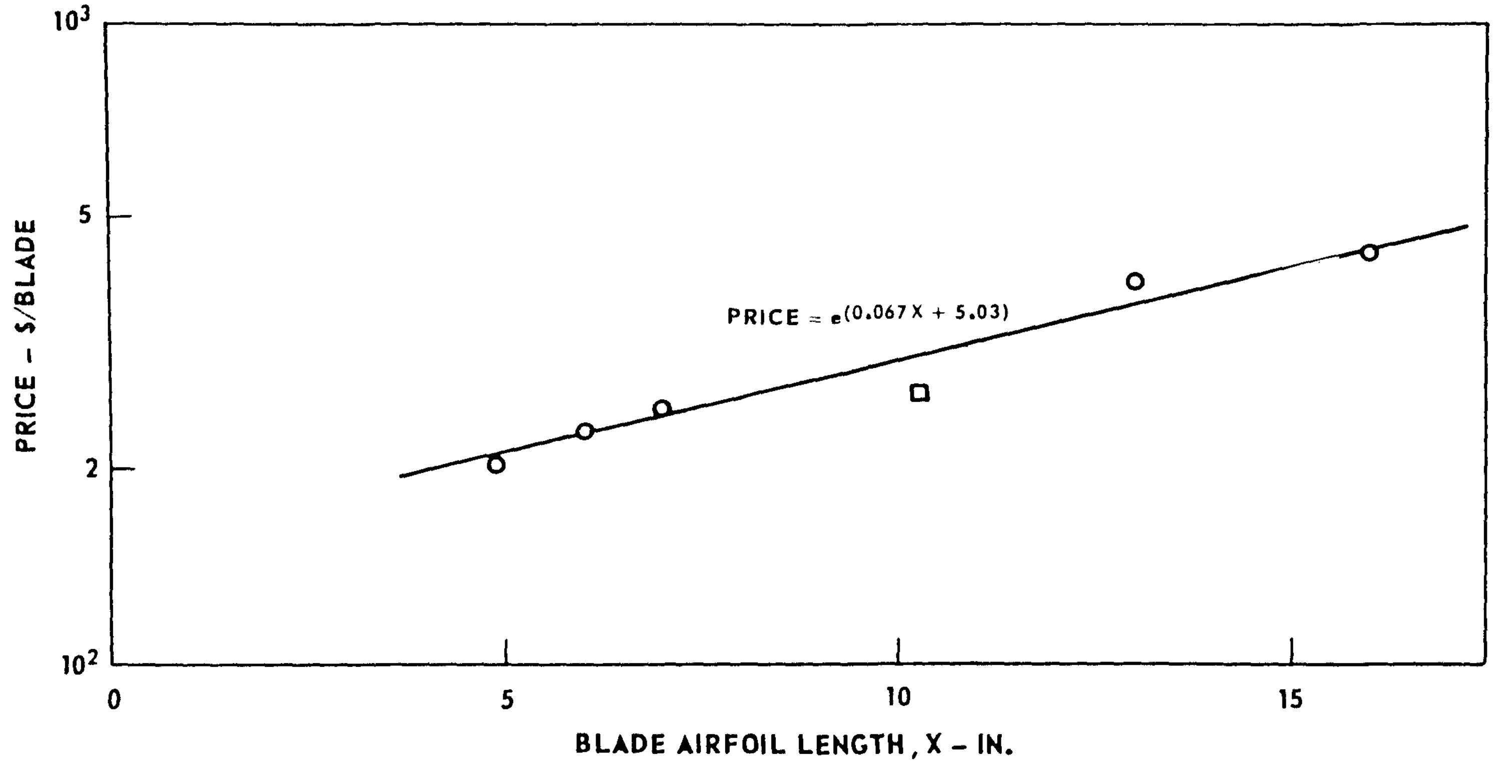


FIG. 79. MANUFACTURERS' PRICES FOR IMPINGEMENT-COOLED TURBINE BLADES

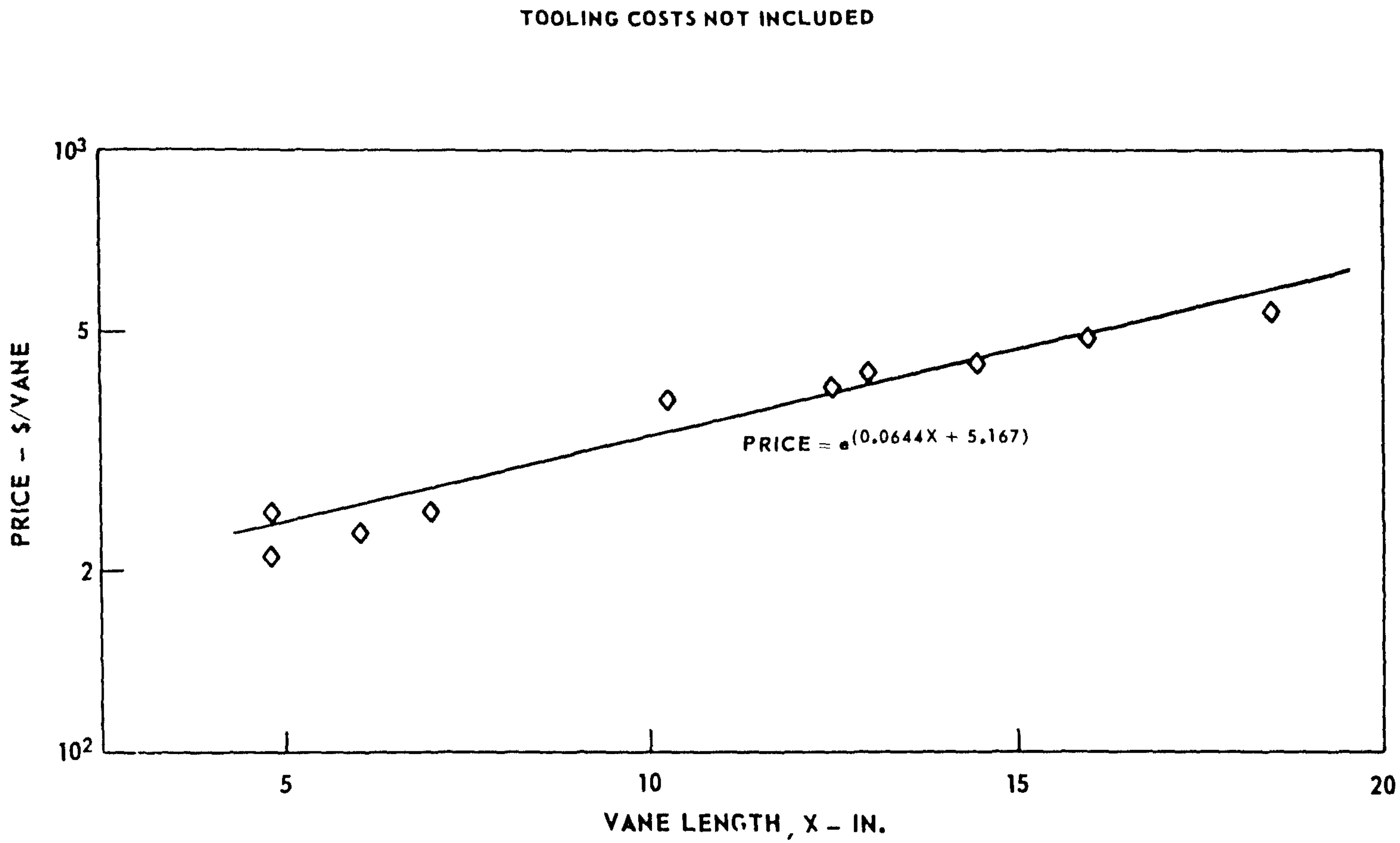


FIG. 80. MANUFACTURERS' PRICES FOR IMPINGEMENT-COOLED TURBINE VANES

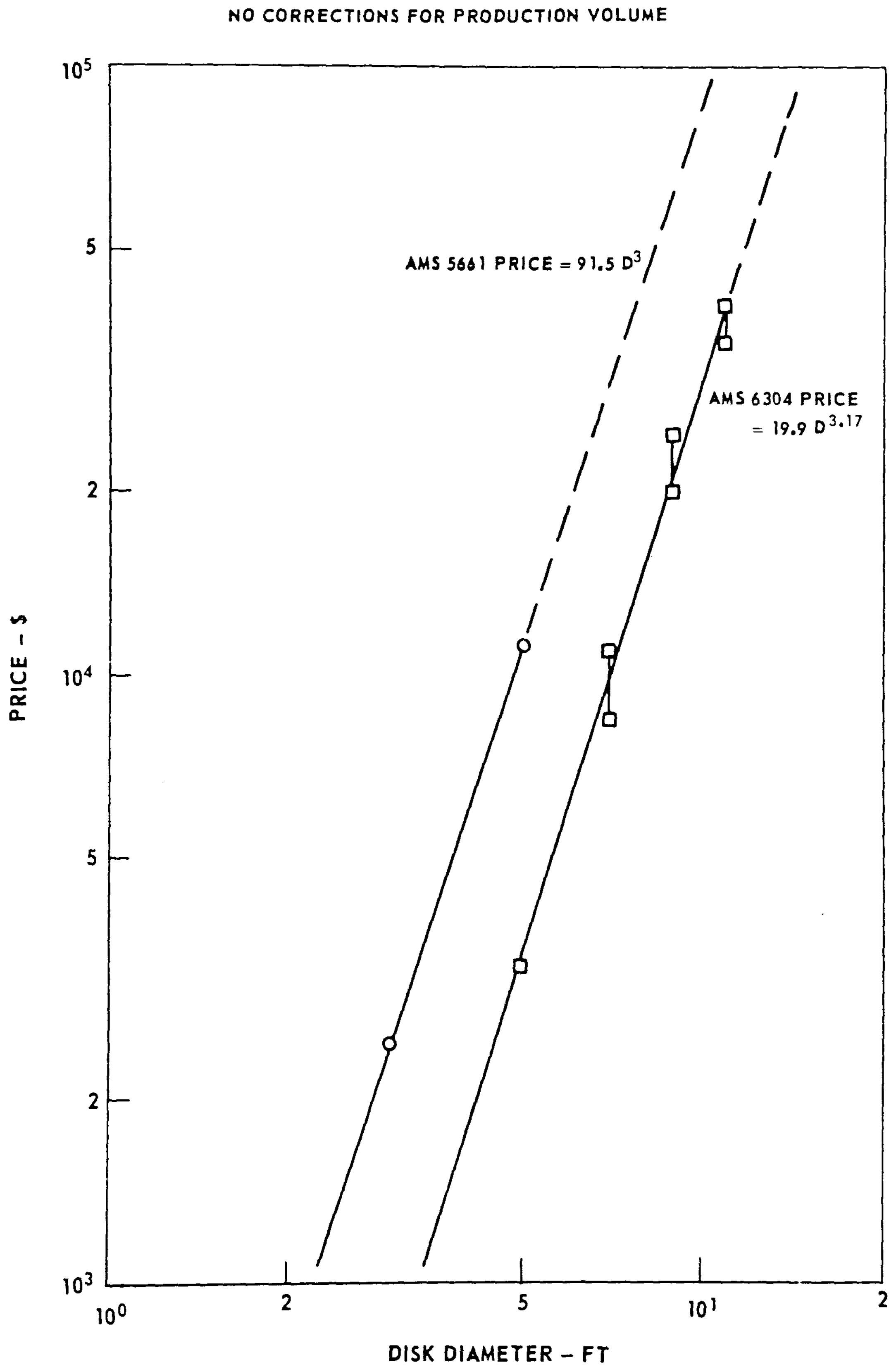


FIG. 81. PRICE ESTIMATES FOR FORGED TURBINE DISKS

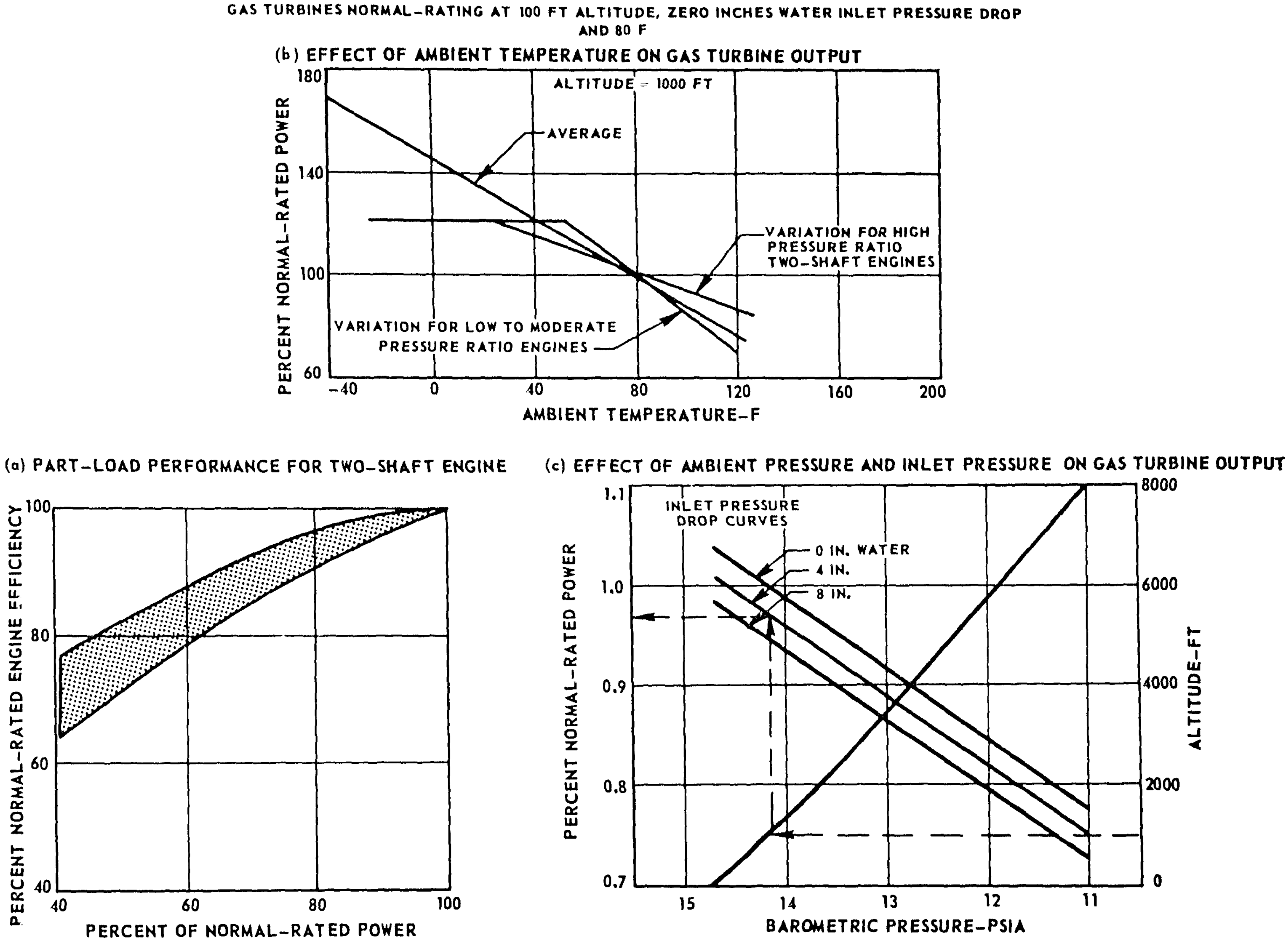


FIG. 82. GAS TURBINE OFF-DESIGN CHARACTERISTICS

SECTION XIII

PUBLICATIONS

As a result of the work performed under Contract No. 14-12-593, the following publications have been produced.

> Utility Applications for Advanced Gas Turbines to Eliminate Thermal Pollution. ASME Preprint 70-WA/GT-9. Presented at the ASME Winter Annual Meeting, November 29-December 3, 1970, New York, New York. Authors: F. R. Biancardi and G. T. Peters.

SECTION XIV

APPENDICES

APPENDIX A

EMISSIONS OF NITROGEN OXIDES FROM GAS TURBINE-TYPE POWER SYSTEMS

Although nitrogen oxide (NO_x) emissions from a power system depend upon many design and operating factors, gas turbine-type power systems usually emit less NO_x (on a ppm stack gas concentration, or pounds per unit heat input basis) than do reciprocating internal combustion engines or large steam power boilers. Gas turbine-type power system NO_x emissions usually range from 75 to 130 ppm (Ref. 138), whereas coal-fired steam boiler NO_x emissions may be as high as 1200 ppm (Ref. 139). The NO_x emissions from oil- and gas-fired steam boilers are generally lower than from coal-fired units, and values as low as 150 ppm have been reported in some gas-fired steam boilers (Ref. 140). However, efforts to reduce NO_x emissions from steam boilers have also resulted in reduced operating efficiencies.

The combustion of a fossil fuel with air in all types of power systems results in the formation of nitrogen oxides. Because 90% or more of the NO_x in the stack gases is present as the relatively unreactive nitric oxide (NO), methods to control NO_x emissions by stack gas removal are quite complex (Ref. 141) and appear to be unattractive unless they might possibly be combined with sulfur oxide stack gas removal. Thus, the most practical method of limiting NO_x emissions is to control their formation in the combustion process itself and in the subsequent processes by which the combustion products undergo cooling before being emitted from the stack.

Theory predicts that values of NO_x concentration in the combustion products are dependent on design and operating conditions, flame temperature, excess air (atomic concentrations), and residence time. The formation of NO_x, which begins with the onset of combustion in the primary combustion zone where temperature is a maximum, continues at a relatively slow pace (due to low chemical reaction rates) as colder bulk gas enters the recirculating flame zone. The NO_x formation cannot proceed until high temperatures are achieved through the combustion of the hydrocarbon fuel with air, and therefore, the hydrocarbon chemistry is virtually completed before NO_x formations begins. The amount of NO_x formed thus depends on the conditions in the primary zone and the subsequent temperature and concentration history of the combustion products with time.

Since the NO_x is formed after the hydrocarbon chemistry has been completed, the potential exists for controlling NO_x while achieving good combustion efficiency. Although present theory is useful in pointing out the approaches which may be employed to limit NO_x emissions, it is not comprehensive enough to explain all the interactions involved in the chemical, thermodynamic, and fluid dynamic processes. Experience has shown that NO_x emissions may vary extensively between power systems having similar operating conditions, and may even be widely different in identical equipment.

Design and operating factors found to affect the power plant NO_x emissions include: fuel type and composition, heat release rate, burner configuration, excess air, and air inlet temperature. All of these factors, in turn, affect flame temperature, excess air, and residence time and thus dictate the amount of NO_x formed. Low-excess air combustion, multiburner combustors, steam and water injection, reduction of air preheating, and recirculation of stack gases have been used with some success on some steam boilers to reduce NO_x emissions. However, the use of these methods usually results in some degradation in operating efficiencies. Stack gas recirculation was used to reduce the temperature in the primary combustion zone and thereby achieve the aforementioned 150-ppm NO_x concentration in the exhaust of gas-fired steam boilers (Ref. 140). Some, but obviously not all, of the above methods could be applied to reduce the NO_x emissions from gas turbine-type power systems without compromising good operating efficiencies.

In present-day gas turbines, the primary zone in the combustor is operated at near-stoichiometric fuel-air ratios, and a great deal of recirculation in the flame zone is designed into the combustor to ensure proper combustion. These characteristics make for a high primary zone temperature and long residence time, both of which tend to promote the formation of NO_x. This type of operation is necessary in aircraft jet engines to vaporize the fuel droplets and ensure stable flame propagation. It is generally difficult to have a lean fuel-air ratio in the primary zone, and to reduce recirculation in the aircraft application, and still meet the requirements for altitude restart capability and flame stability and propagation at low pressures (high altitude and idle power settings). In a stationary gas turbine power system, these requirements do not apply and stable flame propagation with a lean primary combustion zone and little recirculation could be achieved by premixing the fuel and air. This approach would not be considered for the aircraft application because the widely varying operating condition would enhance the possibility of an explosion. Another possible method of reducing NO_x emission in a stationary gas turbine-type power system involves water injection, which is used to cool the primary combustion gases before substantial amounts of NO_x have time to form. It is claimed (Ref. 141) that water injected in a mass equal to the fuel mass would reduce NO_x emissions from gas turbines by 90%.

Based on the previous discussion, and the Corporate- and Government-sponsored work being conducted by UA Research Laboratories and Pratt & Whitney Aircraft, it appears that the principles and techniques required to reduce NO_x emissions from gas turbines are understood. Furthermore, it appears that the NO_x emissions achievable in advanced gas turbine-type power systems can be lower than in present-day engines and that these lower NO_x emissions can be obtained without compromising gas turbine performance.

APPENDIX B

DESCRIPTION OF GAS TURBINE DESIGN PROGRAM

Power plant design parameters and constraints, reflecting the design technology and materials improvements projected in the main body of this report to be available during the 1970 decade, early 1980's, and late 1980's were used to determine the performance and power plant dimensions of simple-cycle power plant designs incorporating either single-spool or twin-spool turbomachinery configurations, regenerative-cycle power plant designs with single-spool configurations, and compound-cycle designs with three-spool configurations.

The primary independent variables (namely, turbine inlet gas temperature, compressor pressure ratio, and regenerator effectiveness) and the various turbine cooling techniques considered are given in Table XVI for each type of power plant design. Turbine blade cooling configurations commensurate with anticipated advances in the state of the art for the projected time period were investigated, and the associated penalty to the gas turbine power system performance was appropriately assessed.

A high-speed digital computer program previously developed under Corporate sponsorship was used to facilitate these parametric studies and also to provide a realistic assessment of turbine cooling flow penalty effects on power plant thermal efficiency and specific output (hp/lb/sec). A brief description of this gas turbine computer program is outlined in the following paragraphs along with a definition of the basic assumptions.

Gas Turbine Design Computer Program

Once the primary independent variables, i.e., the time period for the power plant design, thermodynamic cycle, number of compressor spools, turbine cooling technique, turbine inlet gas temperature, compressor pressure ratio, and power plant airflow rate have been specified in the program, all combinations of the design parameters (incorporated into the program) are investigated, and the appropriate combination of parameters that satisfy specified design constraints are determined. The power plant flow path, pertinent dimensions, number of compressor and turbine stages, turbine cooling flow requirements, and allowable metal temperatures are then computed. After these parameters have been determined, the computer program assesses the effect of the calculated turbine cooling flow rate, including the collective contributions of blade and vane cooling flows and disk cooling flow, on specific horsepower and thermal efficiency.

Basic Assumptions

Certain restraints were necessary in the use of the computer program. For instance, constant-mean-diameter flow passages were assumed for all turbomachinery performance computations to provide some control over the number of possible design concepts. The number of compressor stages in each spool was computed on the premise of a constant flow coefficient per stage. This assumption, together with the preceding one concerning constant mean diameter, resulted in a constant axial velocity and hence constant stage work for each of the respective compressor spools. Compressor stage performance representative of current advanced-design aircraft technology (high stage loadings) is presented in Fig. 72. For the 1970-decade base-load compressor designs the polytropic efficiencies and flow coefficients depicted in Fig. 72 were used, but the work coefficients were derated to 80% of the Fig. 72 values. The early 1980- and late 1980-technology compressor designs were assumed to exhibit the work coefficient and flow coefficient performance depicted in Fig. 72, but at the correspondingly higher polytropic efficiencies presented in Table XVI. The aspect ratio in the first stage of all compressor designs was not allowed to exceed 3.0, thus avoiding the need for costly stiffening rods that would also contribute to increased pressure losses and degradation in compressor efficiency. The last-stage hub/tip ratio for all cmpressor designs was not allowed to exceed 0.93 in order to minimize blade end-losses.

In all twin-spool gas turbine designs, the high-pressure turbine was assumed to comprise a single stage capable of delivering up to a maximum specified value of stage work. The high-pressure turbine performance was described by Fig. 73 for the 1970-decade power plant designs. The early- and late-1980's designs were assumed to exhibit the same work coefficient and flow coefficient performance depicted in Fig. 73, but at the correspondingly higher efficiency values shown in Table XVI.

The low-pressure turbine consists of high-efficiency, stages each capable of providing the same work output and whose number was determined from the appropriate design parameters and constraints. Designing for high stage work and hence a relatively high temperature drop in the high turbine reduces the cooling flow requirements and possibly the high-temperature material requirements in the low turbine. The number of turbine blades and vanes in both the high-turbine and low-turbine stages were estimated on the basis of velocity triangles and lift coefficients comparable with current aircraft engineering design procedures and consistent with the assumed stage efficiency levels. The ratio of blade height to axial width for the unshrouded high-pressure turbine was limited to a range of values between a maximum of 2.5 and a minimum of 1.0. High-turbine minimum mean axial widths of 1.0 and 1.5 in. were assumed for the blade and vane, respectively.

The low-turbine stages and the power turbine stages were assumed to be shrouded, and the maximum blade height-to-axial-width ratio was not allowed to exceed a value of 5.5. Low-turbine minimum mean axial width of 1.5 in. was assumed for both the blades and vanes. The aforementioned basic assumptions are consistent with related aircraft propulsion system design technology.

Turbine blade and vane cooling flow requirements were obtained from correlations of cooling effectiveness (η) with cooling flow as shown in Fig. 74 for advanced impingement-convection cooling techniques. Cooling effectiveness is defined as the difference between average blade or vane metal temperature and cooling air temperature divided by the difference between gas temperature and cooling air temperature. The correlations for advanced impingement-convection cooling shown in Fig. 74 are comparable to the best current cooling designs for aircraft gas turbine propulsion systems. Cooling flow fraction is not used directly in Fig. 74 but is multiplied by the gauging (defined as the airfoil passage throat width). Hence the cooling flow required depends on geometry in addition to temperature.

Disk cooling flow requirements were estimated on the basis of related engineering experience, since accurate determinations of this parameter would have entailed detailed heat transfer and seal leakage analyses. Disk cooling flow requirements of 0.75% per face for the high turbine and 0.375% per face for each stage of the low turbine were assumed for all design conditions. It was assumed that disk metal temperatures could be maintained at from 1200 F to 1400 F or lower with these cooling flows. The contribution of cooling flow to the degradation in turbine adiabatic efficiency was estimated in the following manner. A 1% penalty in high-turbine stage efficiency for each percent disk cooling flow in excess of 1.0% was assumed. Further, a 1% penalty in low-turbine stage adiabatic efficiency for each percent disk cooling flow in excess of 0.25% was also assumed. The blade and vane cooling flows also contribute to a loss in adiabatic efficiency. A 0.5% decrease in stage adiabatic efficiency was assumed for each percent of cooling flow for the blades plus vanes in each stage.

The long-time (1% creep in 100,000 hr), steady-state, creep strength-temperatures properties for the 1970-decade turbine blade alloys (Fig. 24) were estimated from Larson-Miller-type extrapolations of short-time (1000-hr) 1%-creep data corresponding to nickel-base alloys. The long-time, steady-state, creep strength-temperature properties assumed for the early 1980- and late-1980-decade technology turbine blade alloys were estimated by assuming a 20 F/yr improvement in allowable metal temperature with the 1970-decade alloy serving as the reference material. Relatively short-life (1% creep in 1000 hr) aircraft materials have improved at a rate of 30 F/yr. The projected early 1980-decade creep strength properties (see Fig. 24) agree reasonably well with Larson-Miller extrapolations of an advanced nickel-base alloy and a unidirectionally solidified eutectic alloy currently under development for advanced aircraft propulsion systems. Similarly, the projected late 1980-decade material data also compare with preliminary data for columbium alloys as well as with projections for future high-temperature chromium alloys.

The average blade metal temperatures used to determine cooling effectiveness, defined in the preceding discussions, were computed with the aid of the creep strength properties in Fig. 24, used in conjunction with a simplified blade root stress relationship, defined as follows:

$$\frac{S_b g}{\rho_b V_T^2} = \frac{1}{6}\left[1 + H/T - 2\ (H/T)^2 + \frac{A_T}{A_H}\ (2 - H/T - (H/T)^2\right]$$

where S_b = blade root stress, lb/ft^2

g = gravitational constant, 32.2 ft/sec^2

ρ_b = blade material specific weight, lb/ft^3

V_T = blade tip speed, ft/sec

H/T = rotor hub/tip ratio

A_T/A_H = blade taper ratio

The blade allowable stress was assumed equal to the calculated value of blade root stress, and the allowable metal temperature was determined from Fig. 24 as a function of the blade allowable stress. The allowable vane metal temperatures were obtained from Fig. 24, assuming an allowable stress of 5000 psi in the high turbine and 10,000 psi in the low turbine.

Adaptation of Fig. 24 to the computer program placed the emphasis on the use of the best material available during the specified time period to minimize cooling flow requirements as an alternative to placing emphasis on less-expensive alloys with commensurately higher cooling flow requirements. This design philosophy was also used to determine the power turbine configurations. As a result, only a few of the power turbines designed in this study required cooling. The discussion in SECTION VII indicated that avoiding cooling the power turbine appears to be the most economical approach. A maximum allowable blade root stress up to 60,000 psi was imposed on all power turbine designs.

The performance penalties attributable to turbine cooling were computed in the gas turbine design computer program as follows. The mixed flows were calculated at the appropriate stations in the turbine by a mass balance between the main stream and the cooling flows. The temperature of the mixed stream at each station was calculated by a simple heat balance between the main stream and the cooling flows. The effects of cooling flow pumping losses on the net turbine work and the effect of blade and vane cooling flow and disk cooling flow on the adiabatic stage efficiency were also incorporated into the program.

APPENDIX C

DESCRIPTION OF GAS TURBINE COST MODEL

The gas turbine engines being considered as prime movers for advanced-cycle, base-load electric power generating stations which could be commercially available in the next two decades represent significant advances in the state of the art, both from the standpoint of operating conditions and that of unit output power. It is difficult to predict accurately the impact these engines would have on the future of electric power generation without an accompanying economic analysis since these future gas turbine engines could be characterized by entirely new design criteria. By contrast, many present-day industrial gas turbine engines are merely adaptations of aircraft engines in which light weight and high specific power (lb thrust/lb airflow) were emphasized. As a result, few of the general economic correlations developed for evaluating present-day industrial designs could be expected to hold true for future designs, primarily because many of the required geometric and economic scaling factors are not well defined. Further, these future industrial-type gas turbine designs would make use of innovations not applicable to current machinery in order to attain the performance necessary to be competitive with alternative methods of generating base-load electric power. The significance of these observations is that, except for the design of aerodynamic flow passages (primarily the vanes and blades), and use of aircraft-type advanced materials, future base-load large power output gas turbine engines could be designed and manufactured with somewhat different philosophies than are considered common today.

Economic Analysis

Descriptions of engine cost estimating procedures developed under United Aircraft Corporation sponsorship and presented in this section are concerned primarily with the economic analysis which was used to estimate costs and selling prices of future gas turbine engines incorporating all standard accessories such as the fuel system, inlet housing, and exhaust stack. Cost estimates for peripheral equipment such as pumps, generators, housings, mountings, etc., are not included here but were made and taken into account in the overall system costs presented in SECTION VIII.

A survey was made of present-day gas turbine engines manufactured by Pratt & Whitney Aircraft, General Electric, and others to determine whether any correlations between price and engine characteristics existed, and if so, whether these could be used in a gas turbine model synthesized to reflect technological developments anticipated for the forthcoming years. Unfortunately, no general correlations of this type were found to exist, primarily because most of the current engine models were designed for entirely different types of service where, as noted, differing sets of design criteria ultimately dictate final configurations. As a result, it was found necessary to establish a model for the economic analysis which synthesized an engine design and summed the costs of the various major individual components in such a manner that overall costs could be developed in building-block fashion. These costs for the various major individual components were obtained from the literature where possible, but principally from vendors' estimates and from qualified United Aircraft personnel.

It is interesting to note that practically every source of economic data indicated that the primary variables affecting gas turbine engine component costs were: (1) the individual component geometry (primarily its linear dimensions); (2) the material selected; and (3) the production volume. Other variables were estimated to have little effect on cost estimates in a generalized economic program suitable for analyzing many different gas turbine engine designs.

A schematic diagram of the model developed to estimate manufacturing costs of advanced gas turbine engines is shown in Fig. 75. Detailed cost relationships were developed for the low and high compressors, the burner, the high, low, and power turbines, the shafts, the bearings, and the casing and exhaust elbow. The costs of the inlet housing, fuel system, and miscellaneous small parts were lumped together and considered to comprise a constant fraction of the overall manufacturing cost. Provisions in the economic program were made to analyze engine designs incorporating compressors with either constant hub, mean, or tip diameters. Different blade and disk materials can be specified for each compressor and, in addition, provision is made to accommodate a change in blade and disk materials in the high-pressure compressor.

The burner was sized using volume flow, reference velocity, and length-to-diamter (or height) relationships, and either an annular (indicative of advanced engines) or a cannular (indicative of present-day engines) design can be accommodated. Each stage of the turbine section was analyzed individually as were the compressor stages. Varying hub and tip diameters as well as differing blade and disk materials were accommodated. Provisions also were made to estimate costs for impingement cooling in the turbine blades and the use of tip shrouds where necessary.

As many as three shafts (two in the gas generator section and one in the power turbine) were included in the overall analysis. The shaft diameters were calculated by using material stress relationships and rotor torque. Casings for each section of the engine were assumed to be cast in halves, and final machining was assumed to be conducted in the engine manufacturer's facilities. Casing thickness was based on either the minimum thickness required to withstand internal gas pressures, or the minimum thickness required for sound casting techniques. In the former case, a metal working stress of 30,000 psi and a factor of safety of 5 have been assumed. The exhaust elbow was considered to be made from sheet steel and to be similar in design and fabrication to those elbows used on present-day engines. A choice of bearings was considered, including Kingsbury thrust, roller, and ball types, the prices of which were found to vary primarily as the bore diameter of the type selected. In those engine designs not requiring bearings at a particular location, provisions were made to omit that particular component cost estimate. As mentioned, several specific engine components (fuel handling equipment, inlet housing, nuts, bolts, etc.) were combined, and the associated cost of these parts was assumed to be equal to 18% of the total manufacturing costs.

Component Cost Information

Once the basic economic model was established, vendors were consulted to obtain costs for the different major engine parts. Each vendor contacted was supplied with illustrative drawings and a list of materials which were suitable for the part to be manufactured, and then was asked to supply price information in 1969-70 dollars. Where necessary, tables of nominal dimensions were also supplied to the vendors in order to provide them with additional information which would cover a range of parts sizes to be considered in this study.

Compressor Section

Drawings of solid compressor blades with aerodynamic blade lengths from 6 in. to 36 in., were supplied to representative blade manufacturing organizations to assist in obtaining price estimates. These prices (see Fig. 76) are primarily for blades with straight pedestals, (i.e., a constant hub diameter design) manufactured from AMS 4928 material. Mid-span rod stiffeners would be required on blades with lengths greater than 24 in. and length-to-chord ratios greater than 3.0. These stiffeners are quite expensive if cast integrally with the blade, but for the industrial designs considered, round rods inserted through, and welded to each blade would be used and would increase the cost of each blade by only about $2.00. Sloping pedestal blade designs would be approximately 15% more expensive than the prices for constant hub diameter designs, primarily because of the increased machining required on the pedestal. It was recommended that changes in blade materials could be accommodated by assuming that 40% of the blade price would be material cost and 60% labor charge. Different blade materials were handled in the analytical program by taking the overall blade dimensions of length, height, and width, calculating the volume of this block of material, estimating the price of this block if made from the "new" material, then replacing the original material fraction with the new estimate in the basic blade price analysis. Tooling costs for each blade design were estimated at $28,000, based on data obtained from the representatives, a value which was assumed to be written off over a five-year production run. It was assumed that individual compressor vanes would be formed from strip stock at a price of $2.40 per foot. The vanes would be inserted into a vane ring holder, and the cost of material, insertion, and welding was estimated to be $7.10 per inch of rim circumference. Finally, the cost of spacers between adjacent stages was estimated to average $400 per stage.

Drawings for typical compressor disks were sent to a manufacturer of some of the largest forgings in the US. Representatives from this vendor, in turn, supplied the price estimates used in this study (see Fig. 77). These estimates were based on production of at least 50 disks per year in one run. The estimates would increase by 25% for a run of 20 disks, but would decrease linearly for productions rates between 20 and 50 disks. Tooling costs for compressor disks with diameters less than 8 ft were estimated for disks produced from both AMS 6304 and AMS 5661 materials. Above a diameter of 8 ft, open dies would be necessary and these would be maintained by the manufacturer at no cost to the purchaser. It was recommended that the forged disk prices be increased by 30% to cover finished machining costs.

Engine Burner

The costs of the engine burners were estimated to be proportional to the surface area of the burners. Surface area, in turn, was computed for a given burner volume, design reference velocity, and burner length-to-diameter (or length-to-annular height) ratio. Combustion chambers were estimated to cost $400 per ft^2 and $600 ft^2 for cannular and annular burner designs, respectively. Accessories and manifolds would add an estimated $35 per lb of airflow to this cost. Provisions were included in the analytical program to maintain the maximum burner diameter within certain present limits, and adjustments in burner length were made if this overall diameter was exceeded.

Turbine Section

Many of the same costing techniques used for compressor blades were applicable for turbine blades. However, impingement-cooled turbine blades (and vanes) require entirely different manufacturing techniques from those used to produce solid blades and, consequently, separate price estimates were needed. Therefore, blade and vane drawings were supplied to several manufacturers (Fig. 78). Price estimates were received from two of these vendors for blades and vanes, respectively, produced from B-1900A and Inconel 713C materials as shown in Figs. 79 and 80. It was soon evident from these prices that impingement-cooled turbine blades would be considerably more expensive than solid blades with the same dimensions and materials. Tooling costs for blade and vane designs with airfoil lengths less than 11 in. for blades and 7.5 in. for vanes were estimated to be $100,000 each, while tooling costs for blade and vane designs beyond these lengths were estimated to be $150,000 each. All turbine blade tooling costs were assumed to be amortized in five years.

Discussions with vendors revealed that if it were necessary to cast tip shrouds integrally with turbine blades, the price estimates present for solid blades and for hollow blades would be increased by 10% to allow for additional casting complexity and machinery. Machining the root sections of the blades for engine designs with varying hub diameters would add an additional 15% to the turbine blade prices. No estimates were given for the variation of direct manufacturing costs with production volume, but vendors indicated that the volume of blades and vanes produced should be sufficiently high to preclude any upward adjustments in the prices per unit.

Turbine disk drawings were supplied to vendors, and the price estimates summarized in Fig. 81 were obtained for disks with diameters ranging from 3 ft to 11 ft manufactured from two materials, AMS 6304 and AMS 5661. Since most of the dimensions on gas turbine disks would be essentially proportional to disk diameter, irrespective of the absolute value of disk diameter, the fact that the prices of both the compressor and turbine disks would be roughly proportional to the third power of the diameter (or essentially, to the disk volume) is not surprising. The tooling costs for turbine disks would be about the same as those for compressor disks, varying only with disk material. In addition, if disks with diameters larger than 11 ft were foreseen, entirely new forging facilities might be necessary since no commercial facility exists, other than that presently operated by the US Air Force, which would be capable of producing such large parts. The cost of such a new facility could be in the several-million-dollar range, all of which would have to be charged to the production of engine disks should no further commercial applications be developed.

Engine Bearings

Additional vendors were contacted to obtain price estimates for large, anti-friction, roller and ball bearings as a function of bore diameter. All cost data obtained were for oil-damped bearings manufactured from M-50 carburized steel in lots of at least 50 bearings per year. If lots of less than 50 bearings were made, the unit prices would increase by approximately 50%. Estimates received indicated that purchase prices for sleeve bearings and Kingsbury thrust bearings would be approximately 40% and 50%, respectively, of those for roller bearings with the same bore dimensions. To each bearing price must be added a charge for bearing supports, estimated to be approximately 50% of the price of the bearing alone.

Shafts and Casings

The price of turbine engine shafts was estimated to be $10/lb based on available data. Shaft weights were calculated from an estimate of shaft torque, shaft length (a function of engine power rating), and the material density.

Consultations with corporate personnel revealed that, based on recent experience, casings would cost approximately \$1.50/lb. A charge equal to 30% of the raw casting cost was added for setup and in-house machining. As noted previously, casing thickness would be dictated either by the internal gas working pressure or the minimum thickness necessary to obtain sound castings. The cost of the exhaust elbow was estimated to be \$10,000 for an engine with a throughflow of 250 lb/sec, and proportional to the square root of the airflow for larger engines.

Assembly Costs

Assembly and test costs are difficult to assess accurately. Based on recent industry experience, these costs should vary between \$75,000 and \$150,000 per engine, depending on engine complexity and size. In circumstances where complete engines would be too large to ship to the erection site in a completely assembled package, it was assumed that subassemblies would be shipped to the site and assembled in position. Whether the engine were shop-assembled or field-erected, the cost of assembly is included in the manufacturing cost as if all units were shop-assembled. X-ray costs might be as high as \$15,000 per engine inspected, and when extremely large units are built, it would be necessary to X-ray every unit assembled. For small, open-cycle engines, X-ray inspections would not be required as frequently, and consequently, the X-ray cost per engine produced would be less.

Selling Price

The summation of individual engine component costs, assembly and test charges, and X-ray costs, comprise the manufacturing cost per engine. In the analytical program, it was assumed that many of the component parts would be supplied in finished form by the vendors and, where not, appropriate additional machining charges (including overhead) were added. Manufacturing overhead charges were assumed to be similar to those of industrial manufacturing plants, and not to those which have been found ncessary in high-technology organizations.

Although it would be essential that a manufacturer know his production costs, the ultimate electric utility purchaser would be concerned primarily with the final purchase price he must pay for a piece of machinery. Therefore, in order to develop a selling price which must be charged by an engine manufacturer, additonal economic data was specified. These include: the total development cost of the engine (including additions to existing manufacturing facilities needed for engine production); the average engineering expenses needed to support the production and operation of a particular engine design; the gross profit per unit produced; and the general and administrative (G&A) expenses allocable to the particular program. In addition, an estimate of market penetration must be made to schedule production volume and to estimate amortization of tooling costs during production. By taking the simplified approach of allocating development, continuing engineering, profit, and G&A among all engines produced, the selling price was developed directly from manufacturing costs. In this analysis, expenses such as sales and field engineering support are treated as separate costs, and, in all likelihood, the manufacturer would increase the selling price per unit to cover these expenses.

APPENDIX D

GAS TURBINE OFF-DESIGN CHARACTERISTICS

Part-Load Operation

The part-load performance characteristics of advanced-design gas turbines considered in this study would be similar to that shown in nondimensional form in Fig. 82 for present-day, two-shaft (free power-turbine) configurations operating at constant rated output speed. The correlation in Fig. 82, is depicted as a fairly wide band, particularly at the reduced load settings, because of the scatter in the original data points. This scatter is due to differences in the operating conditions (cycle pressure ratio, turbine inlet temperatures, etc.) of the different engine designs represented.

If high performance at part-load power were desired, the power turbine could be equipped with variable inlet guide vanes which would add to the complexity and cost of the engine. In most gas turbine engine designs, the turbine inlet guide vanes are fixed, and part-power is achieved by reducing turbine inlet temperature. Although with this mode of operation engine airflow rate decreases slightly at part power setting, the principal factor contributing to a reduction in output power is the decrease in heat content of the working fluid, resulting from the lower gas temperature. If part-load power is achieved by utilizing variable geometry guide vanes (maintaining choked flow) to control the airflow rate through the engine constant turbine inlet temperature is maintained at a wide range of power settings. Thus, although some degradation in component performance would result from the reduced airflow rate and from changes in turbine inlet guide vane angle, the loss in power plant performance would not be as significant as that indicated in Fig. 82 for the fixed-geometry operation. However, there is a limit imposed by mechanical and aerodynamic considerations to the maximum reduction in airflow rate which can be achieved with a variable-geometry turbine.

The part-load characteristics of the gas turbine are of secondary importance in this study because the power station design load factor selected requires the engine to be essentially base-loaded, i.e., operating most of the time at rated power. To analytically determine the part-load performance of a gas turbine requires the use of a relatively complex engine matching procedure and involves a detailed knowledge of the off-design performance of each component.

Effect of Ambient Conditions

Gas turbine performance is directly affected by operating altitude and ambient temperature. As previously mentioned the performances in this study were computed in accordance with NEMA specifications; i.e., an altitude of 1000 ft above sea level and an ambient temperature of 80 F. Correlations of the effect of changes in ambient temperatures on output power for a number of present-day industrial- and modified aircraft-gas turbine designs are also presented in Fig. 82. The data were plotted for an altitude of 1000 ft and were normalized with respect to a reference temperature of 80 F. Output power decreases at higher operating altitudes due to the lower density of gas and thus reduced mass throughput capability of the compressor. Generalized approximations showing the effects of altitude as well as variations in inlet pressure losses on output power are depicted in Fig. 82c.

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Detailed performance, size, and cost estimates were made for advanced simple-, regenerative-, and compound-cycle gas turbine engines for turbine inlet temperatures of 2000° F and above as anticipated to be commercially available in the next two decades. Conceptual designs for 1000-Mw central power station utilizing gas turbines and comparisons of complete gas turbine and steam turbine power station installed costs and total busbar power costs were made for the various regions of the US.

It is shown that the gas turbines in the 1970 decade could produce electric power at lower costs than steam turbines in the South Central region of the US where natural gas is readily available. Elsewhere in the US the gas turbines would be economically competitive if moderately priced clean fuels are available. Advanced gas turbines will become more competitive in the 1980 decade as anticipated increases in turbine inlet temperature, component efficiences and larger engine designs lead to more efficient and lower-cost engines and power stations.

Although the development costs for large, advanced gas turbines would approach from 100 to 200 million dollars, the total amount that utilities are expected to expend for cooling devices to combat thermal pollution over the next two decades will exceed more than ten times this amount. Thus advanced gas turbines should be given serious consideration for increased research and development support.

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