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ABSTRACT

This paper describes the technologies that are being developed or extended beyond the current state-of-the-art to achieve Advanced Turbine Systems (ATS) Program goals. The Westinghouse ATS plant is an advanced closed-loop cooled combined cycle, based on an advanced gas turbine engine incorporating novel design concepts and enhancements of existing technologies. The ATS engine is a fuel-flexible design operating on natural gas with provisions for future conversion to coal or biomass fuels. It is based on proven concepts employed in 501F and 501G engines. To achieve the required performance and reliability the engine will include closed-loop steam cooling, advanced materials and coatings, and enhanced component performance. To minimize NOx emissions, an ultra-low NOx combustion system will be incorporated. To ensure technical success, development programs are being conducted on the following: closed-loop steam cooling, advanced materials and coatings, component aerodynamic performance, flow visualization, optical diagnostics, combustion generated noise, and catalytic combustion.

INTRODUCTION

The U. S. Department of Energy, Office of Fossil Energy ATS Program is an ambitious program to develop the necessary technologies, which will result in a significant increase in natural gas-fired power generation plant efficiency, a decrease in cost of electricity and a decrease in airborne pollutants. In Phase 1 of the ATS Program, preliminary investigations on different gas turbine cycles demonstrated that net plant efficiencies greater than 60% could be achieved (Little et al., 1993). The more promising cycles were evaluated in more detail during Phase 2 in order to select the cycle that would achieve all of the program goals (Briesh et al., 1994). The closed-loop cooled combined cycle was selected because it offered the best solution, with the least risk, for exceeding the ATS Program goals of net plant efficiency, emissions, cost of electricity, reliability, availability, and maintainability (RAM), and commercialization in the year 2000.

The Westinghouse ATS plant is based on an advanced gas turbine design combined with an advanced steam turbine and a high efficiency generator. To promote achievement of the challenging performance, emissions, and RAM goals, current technologies are being extended and new technologies developed (Bannister et al., 1995). The attainment of the ATS performance goal necessitates advancements in aerodynamics, sealing, cooling, coatings, and materials technologies. To reduce emissions to the required levels demands a development effort in the following technologies: pre-mixed ultra low NOx combustion, catalytic combustion, combustion instabilities, and optical diagnostics. To achieve the RAM targets requires the utilization of proven design features, with quantified risk analysis, and advanced materials, coatings, and cooling technologies. Phase 2 research and development projects currently in progress, as well as those planned for Phase 3, will result in advances in gas turbine technology and greatly contribute to ATS Program success.

The ATS engine is the next frame in the series of successful utility turbines developed by Westinghouse over the last 50 years. During that time, Westinghouse engineers made significant contributions in advancing gas turbine technology as applied to heavy-duty industrial and utility engines (Scalzo et al., 1994). Some of the innovations included single-shaft, two-bearing engine design, cold-end drive, axial exhaust, cooled turbine airfoils, and tilting pad bearings. Today, these features are incorporated by all major gas turbine manufacturers in their designs. In the

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st, enhancements in Westinghouse gas turbine rformance and mechanical reliability were made in ntinuous steps (see Figure 1). The evolution of large gas bines started with the introduction of the 45 MW 501A gine in 1968. Continuous enhancements in performance re made up to the 100 MW 501D5 introduced in 1981. e next engine developed was the 160 MW 501F roduced in 1991 (Scalzo et al., 1988, Entenmann et al., 90, Entenmann et al., 1992). The 230 MW 501G, the est engine in the 501 series, is the initial step in ATS gine development (Southall and McQuiggan, 1995). Each cessive engine design was based on the proven concepts d in the previous design (see Table 1).

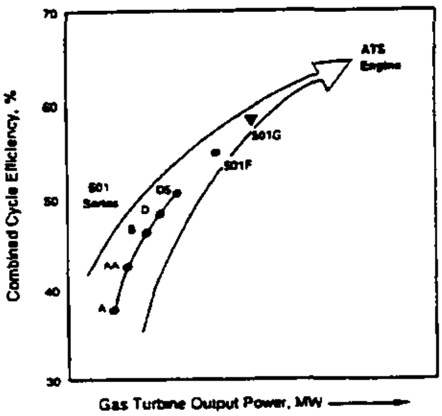

Figure 1. Evolution of Large Westinghouse Gas Turbines

Table 1. Proven Design Features

Feature	501D	501F	501G	ATS
earing Rotor	X	X	X	X
tage Turbine	X	X	X	X
Combustors	X	X	X	X
iz. Split Casing	X	X	X	X
d End Drive	X	X	X	X
gle Row 1 Vanes	X	X	X	X
ling	AC	AC	AC/SC	AC/SC
k-in Enclosure	X	X	X	X

C = Air Cooling   SC = Steam Cooling

The 501F produces 160 MW at a simple cycle ciency of 36%. Its combined cycle net efficiency is about %. This performance level was achieved by roximately 110°C (200°F) increase in firing temperature, npared to the previous engine, advanced compressor and bine design, advanced airfoil cooling design, and roved materials.

The current production engine, 501G, introduced in the ing of 1994, produces 230 MW. Its combined cycle net ciency is 58%. This engine incorporates further ancements in materials, cooling technology, and nponent aerodynamic design. The 19:1 pressure ratio npressor uses advanced profile high efficiency airfoils. combustion system incorporates 16 dry low NOx combustors. The combustor design is similar to the 501F with the same flame temperature and hence the same low NOx emission. The four-stage turbine uses full 3-D design airfoils and proven aeroderivative materials and coatings.

Achieving the ATS Program's challenging goals will require breakthroughs in several key technologies as well as advances in a broad range of current technologies. At the ATS performance level, the key issues are turbine component cooling, coating systems, and emissions control. Of almost equal importance in achieving the ATS Program goals will be the advances in component aerodynamic performance, sealing technology, and materials. To mitigate the key technical issues and to bring about the necessary advances in the supporting technologies, several development programs were proposed for ATS Phase 2. These programs include developments in combustion, emissions, cooling, aerodynamics, sealing, coating, and materials technologies.

Westinghouse's strategy to exceed the ATS Program goals is to build on the proven technologies used in the successfully operating fleet of its utility gas turbines, such as the 501F, to extend the technologies developed for the 501G and to overcome the identified technical barrier issues through a concerted development effort in ATS Phases 2 and 3 (see Figure 2).

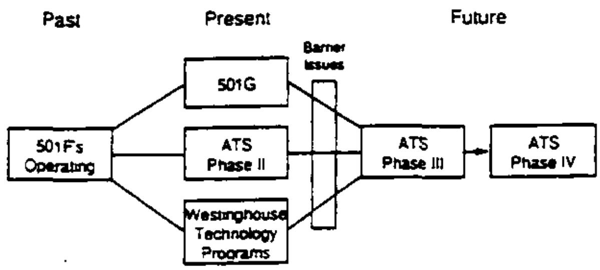

Figure 2. ATS Technology Development

The objectives of the ATS Program Phase 2 were to select the ATS cycle and to enhance or develop technologies required to achieve the ATS Program goals. This paper describes progress on the different technology development programs being conducted by Westinghouse in support of the ATS Program.

ATS DESCRIPTION

The ATS plant is based on the advanced combined cycle. The gas turbine, generator, and steam turbine will be connected together in an in-line arrangement with a clutch located between the generator and the steam turbine.

The gas turbine exhaust gases will pass through the three-pressure level heat recovery steam generator (HRSG) before being exhausted through the stack. The steam turbine will employ advanced 3-D aerodynamic design methods. The 60-Hz two-pole generator will be an extension of the Westinghouse hydrogen-cooled modular design concept.

The ATS engine is an advanced 300 MW class design incorporating many proven design features used in previous Westinghouse gas turbines and new design features and technologies required to achieve the ATS Program goals. The compressor design philosophy is based on that used in the advanced 501G compressor. The aerodynamic design uses 3-D viscous codes, controlled diffusion airfoils and reduced airfoil thicknesses. Variable stators are incorporated in the front stages to improve starting and part-load operation.

The combustion system has 16 combustors of lean-premixed multistage design. To limit NOx emissions to less than 10 ppmv, nearly all of the compressor delivery air is premixed with the fuel. Therefore, closed-loop steam cooling is used to cool the combustors and transitions.

The four-stage turbine is an extension of the advanced 501G turbine design. The design is based on 3-D design philosophy and advanced viscous analysis codes. The airfoil loadings are optimized to enhance aerodynamic performance while minimizing airfoil solidity. The reduced solidity results in reduced cooling requirements and increased efficiency. To further enhance plant efficiency, the following features are included: turbine airfoil closed-loop cooling, active blade tip clearance control on the first two stages, improved rotor sealing, and optimum circumferential alignment of airfoils.

Preliminary ATS engine and plant designs were completed and design reviews were held to verify the design concepts.

CONVERSION TO COAL-FIRED ATS

A number of advanced, coal-fired power generation technologies have been under development that could be applied to a natural gas-fired ATS. These include a broad range of coal gasification technologies (fixed bed, fluid bed, and entrained bed), second generation pressurized fluidized bed combustion (PFBC), and direct coal-fired turbine. Two advanced, coal-fueled technologies have been selected for consideration as coal-fired ATS: air-blown integrated gasification combined cycle (IGCC) with hot gas cleaning based on the KRW fluidized bed gasifier; and second generation PFBC (Newby et al., 1995).

The selection of a coal-fired reference system for the conversion of the natural gas-fired ATS to a coal-fired ATS was based on performance potential (power conversion and emissions), cost potential, and status of development. Estimated thermal conversion performance, cost potential, and the status of development of several advanced, coal-fueled technologies that could achieve the desired turbine inlet conditions were considered. Comparisons were made to natural gas-fired turbine cycles, as well as to conventional coal-fired power plants, atmospheric pressure fluidized bed combustion, and pulverized coal combustion boilers with flue gas desulfurization. The air-blown IGCC technologies with hot gas cleaning and second generation PFBC appeared to have the greatest thermal performance and cost potential with combined cycle efficiencies in the range of 51 to 53%. Their respective status of development is categorized as early demonstration which is suitable for the ATS Program.

PHASE 2 COMPONENT DEVELOPMENT PROJECTS

To resolve the identified technical barrier issues and achieve ATS Program goals, an extensive R&D program is in progress in the fields of combustion, cooling, aerodynamics, leakage control, coatings, and materials.

COMBUSTION

Due to the strict emissions requirements for the ATS engine, combustion development will be one of the critical areas requiring significant effort. To address this issue the following development programs were initiated: combustor flow visualization, combustor cylinder flow mapping, optical diagnostics probe development, combustion instability/noise investigation, and catalytic combustion component development. In addition, several dry ultra low NOx combustor development programs are in progress. The combustor verification tests are being performed in the full-scale, high pressure test facility located at the Arnold Engineering Development Center, Arnold AFB, Tennessee (Pillsbury et al., 1995).

Flow Visualization.

Air flows inside the combustor cylinder and into the combustors are very complicated. These flows have a significant effect on the losses, and hence on engine performance, and on the flow distribution inside the combustors and hence on the combustion processes. The latter effect is especially critical in the dry ultra low NOx lean-premix combustors, which rely on the correct fuel/air ratios within a very narrow tolerance band, for low NOx production and operational flame stability. Flow tests are being carried out on plastic models of the ultra low NOx baskets in the single-can rig and the sector rig at Westinghouse Casselberry Labs. Flow visualization, detail flow mapping tests and effective flow area measurements are being performed. Hot wire anemometry, as well as other conventional measurement techniques, are employed.

Combustion Cylinder Flow Mapping

Flow mapping and flow visualization tests are in progress on a half-scale plastic model of a combustion cylinder at Clemson University. Higher order effects inside the combustor are being investigated. Detailed measurements of pressures, velocities and flow angles inside the combustion cylinder, and especially around the combustor baskets, are being obtained. Included in this investigation is the effect of struts, cooling air return pipes, op-hat length increase and cooling air extraction. In addition to studying and optimizing the flow around the combustors, the objectives of these tests are to optimize the performance of the compressor exit diffuser, reduce the diffuser exit dump loss, and hence improve the ATS engine performance. In support of the experimental program, a CFD analysis of the combustion cylinder flow field was carried out.

ptical Diagnostics Probe Development

Optical diagnostics allow measurement of pertinent arameters, such as the composition and concentration of ombustion products, in addition to velocities and flow ngles, without disturbing the main flow. A laser-induced uorescence fibre optic probe is being developed. Probe aluation at high pressures and temperatures is in progress. is probe will be a very useful tool in enhancing mbustion development productivity. It will be used in ld flow and fired tests.

mbustion Instability Investigation

Lean-premix combustion system will be employed to hieve the NOx emissions goals. The lean combustion with inherent flame instability results in more combustion nerated noise, and hence, in vibration problems in the mbustion system as well as in the downstream mponents. A program to develop the theoretical ckground for combustion instabilities, to carry out periments to aid in the understanding of the problem, to velop a generalized analysis procedure, and to develop bility criteria is under way.

An active noise control system is being developed to minate combustion instabilities. It consists of a sensor to tect the combustion instabilities, signal processor, dback algorithm generator and a fuel modulation valve d controller (see Figure 3).

tra Low NOx and Catalytic Combustion Development

Westinghouse is developing several ultra low NOx mbustors (Foss et al., 1994) in order to achieve single nt NOx emissions (see Figure 4). The ATS combustor ll be based on the most successful candidate. Catalytic mbustion is expected to play an important role in hieving ultra low NOx emissions at the ATS engine firing nperature. The catalyst allows ultra lean-premix mbustion without flame instability and flame outs. erefore, NOx production is restricted to low single digits ATS firing temperature with stable operation. velopment is in progress to gain theoretical understanding catalytic combustion, to design a catalytic combustion stem and to develop a practical catalytic combustor. A talyst coated pilot has been tested in an ultra low NOx combustor with excellent results. A catalytic combustor with exhaust gas recirculation to preheat the catalyst to the required temperature is being designed.

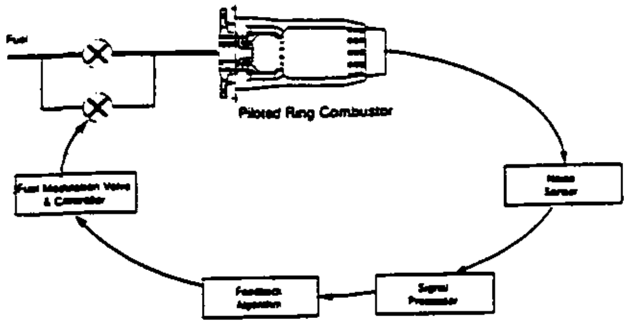

Figure 3. Active Noise Control Loop

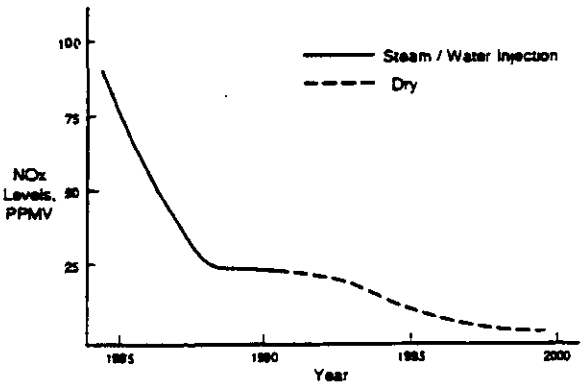

Figure 4. NOx Emission Levels

COOLING

Closed-Loop Steam Cooling

The one concept that will result in the greatest increase in the ATS plant cycle efficiency is closed-loop steam cooling. There are several challenges to a successful closed-loop steam cooling system. These include maintaining airfoil surface metal temperatures without outside film cooling, limiting wall temperature gradients, preventing corrosion and fouling of internal components over long periods of time, bringing coolant to and out of rotating components, leakage control, and cold start before steam from the downstream HRSG is available.

The major contributor to plant efficiency increase with closed-loop steam cooling is the elimination of cooling air injection into the turbine flow path. This results in an increase in gas temperature downstream of the first stage vane and hence an increase in gas energy level during the expansion process. A secondary contributor is the elimination of mixing losses associated with cooling air ejection. The combination of these effects results in a significant increase in ATS plant efficiency. In addition, NOx emissions will decrease because more air is available for the lean-premix combustor at the same burner outlet temperature. Achieving acceptable blade metal temperatures in a closed-loop cooling design is a challenge due to the absence of cooling air film to shield the turbine airfoil and shroud wall, and no shower-head or trailing edge ejection to provide enhanced cooling in the critical leading and trailing edge regions. To produce an optimized closed-loop cooling design, the following approaches are utilized: (1) airfoil aerodynamic design tailored to provide minimum gas side heat transfer coefficients, (2) minimum coolant inlet temperature, (3) thermal barrier coating applied on airfoil and end wall surfaces to reduce heat input, (4) maximized cold side surface area, (5) turbulators to enhance cold side heat transfer coefficients, and (6) minimum outside wall thicknesses to reduce wall temperature gradients and hence the internal heat transfer coefficients required to cool the airfoil.

Preliminary investigation was carried out on the following turbine airfoil closed-loop cooling concepts: thin wall shell/spar (see Figure 5), thin wall casting, and peripheral spanwise cooling hole. More detailed calculations will be carried out to select the final ATS airfoil cooling design.

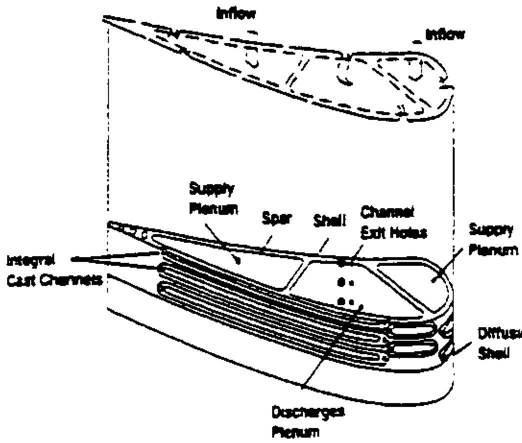

Figure 5. Conceptual Design of Shell/Spar Vane

Shroud Cooling

The airfoil end walls or shrouds present a cooling challenge. The combination of flat burner outlet temperature profiles and uncertainty in the end wall heat transfer coefficients make it difficult to arrive at an optimized shroud cooling design analytically. To address this issue, plastic model tests are in progress to optimize first stage turbine vane shroud film cooling design (see Figure 6). The thermochromic liquid crystal technique is being used to measure the surface temperatures, and hence, the heat transfer coefficients.

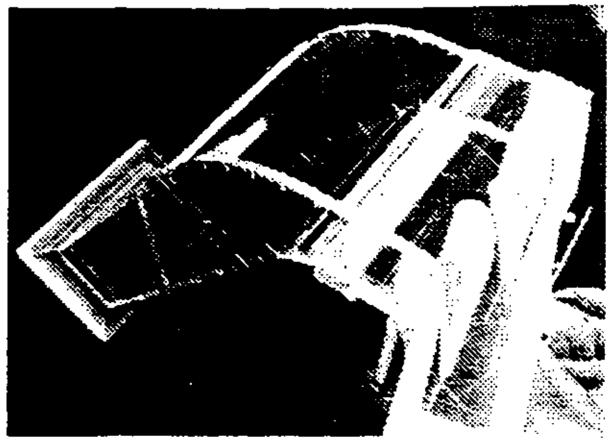

Figure 6. Vane Cascade

Serpentine Channel Cooling

To maintain low blade metal temperatures, complicated multipass serpentine cooling schemes are being developed. Plastic model tests are being conducted to verify and optimize the design prior to incorporation into the engine. Two models are being tested: one model simulating the multipass mid chord region of the blade and the second model representing the trailing edge portion of the blade. Tests are being carried out at different cooling air flow rates. The internal heat transfer coefficients and pressure losses are being measured. Tests on the trailing edge model have been completed, and testing on the mid chord model is in progress.

Integral Shroud Cooling

Turbine blades in the rear stages are designed with an integral interlocked tip shroud to enhance performance and to improve mechanical integrity by providing vibration damping. Metal temperatures and stress levels on these blades may necessitate cooling not only the airfoil, but also the tip shroud. A new approach is required in the cooling design, casting, and machining to cool the shrouds with cooling air, which will heat up through the blade before reaching the tip shroud. To effectively cool the tip shroud, alternative cooling schemes and manufacturing processes were investigated. An optimized shroud cooling design was developed. This cooling concept consists of a cast cavity in the bottom portion of the blade and a mid chord radial hole to supply tip shroud cooling air. The velocities and internal heat transfer coefficients in this hole are low. Therefore, the cooling air heat pick-up is minimized before the cooling air enters the cooling holes machined in the shroud.

AERODYNAMICS

High Efficiency Compressor

High efficiency compressor design is being developed for the ATS engine. The ATS compressor design philosophy is based on that used for the advanced 501G compressor. The aerodynamic design uses 3-D viscous codes, reduced number of stages, controlled diffusion airfoil design, reduced airfoil thickness and advanced sealing. An optimization study was carried out to optimize each stage efficiency individually so as to maximize the efficiency for the design pressure ratio with the minimum number of stages. Compressor performance at off-design conditions was investigated to ensure adequate surge margin over the entire operating range. Blade and stator profile definition is in aerodynamic/mechanical design iteration. All airfoils are custom shaped using controlled diffusion design process. The design objective is to satisfy all mechanical constraints with minimum airfoil thickness so as to optimize efficiency.

LEAKAGE CONTROL

Active Blade Tip Clearance Control

Turbine blade tip clearance has a significant effect on the performance of highly loaded front stages. For each one percent tip clearance increase (based on blade height), the stage efficiency may decrease by up to 2 percent. Even if the initial cold blade tip clearances are set at minimum values, during transients, such as rapid starts and emergency shutdowns, the blade tips are ground off. This results in increased hot running blade tip clearances, which get progressively worse with time. To optimize turbine efficiency, an active tip clearance control system is being developed. The objective of such a scheme is to maintain large tip clearances on start-up and to reduce them to minimum acceptable values when the engine has reached steady-state operating condition (see Figure 7). A conceptual design of an active tip clearance control system is in progress.

Brush Seals

To reduce air leakage, as well as hot gas ingestion into turbine disc cavities, brush seals, which have the potential to reduce leakage by up to 90%, will be incorporated in appropriate locations in the turbine and compressor (see Figure 8). This will enhance engine efficiency as well as mechanical integrity of the turbine components. To incorporate an effective, reliable, and long-lasting brush seal system into a heavy-duty industrial gas turbine, a development program was initiated. Preliminary investigation was carried out to evaluate the benefits, potential seal locations, and validation testing required to apply brush seals to industrial gas turbine engines. Conceptual seal design at one engine location was carried out. Tribopair tests were carried out on five different bristle alloys, two rotor materials, and two rotor surface finishes to determine which tribopair results in minimum wear. The optimum tribopair was selected for seal leakage testing.

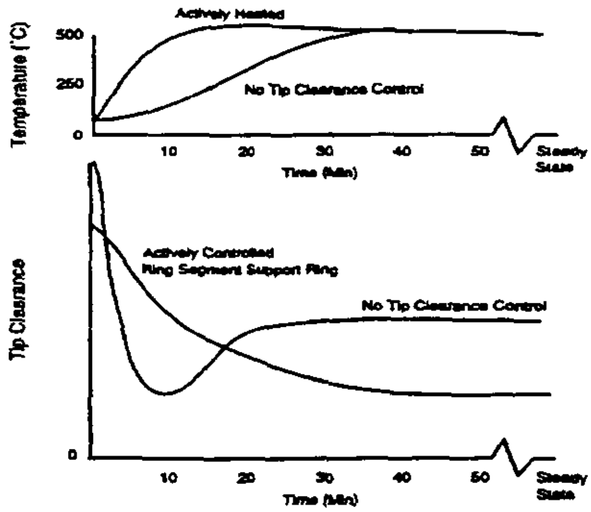

Figure 7. Support Ring Temperature and Tip Clearance with and without Active Control

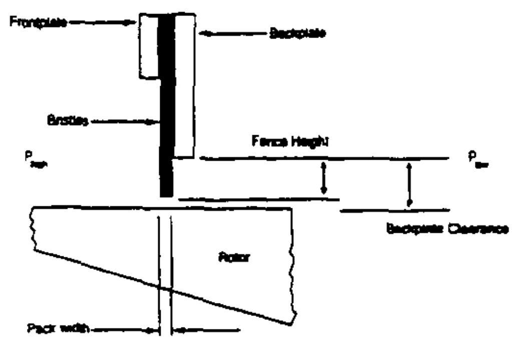

Figure 8. Brush Seal Schematic and Terminology

COATINGS

Thermal Barrier Coatings

Ceramic thermal barrier coatings (TBC) are important to the success of the ATS program. TBC low thermal conductivity effectively insulates the metal substrate and provides up to 11°C (20°F) metal temperature reduction per .025 mm (.001 in.) coating thickness. While TBCs have been used widely on stationary components, use on rotating components has been limited. Field testing of coated blades must be carried out to determine comparative coating longevity and effectiveness of air plasma spray (APS) and physical vapor deposition (PVD) thermal barrier coatings. Three batches of first stage turbine blades coated with APS TBC, PVD TBC, and with only the metallic overlay coating were installed in an operating engine. During engine inspections, these blades will be examined insitu or removed for coating performance evaluation.

Advanced Coating Development

The ATS engine turbine component coatings must be capable of operation for 24,000 hours. To ensure this, a program is in progress to develop an advanced coating system. Different bond coats are being evaluated under accelerated oxidation test conditions. New candidate ceramic coating materials are undergoing testing. The objective of this program is to combine the optimum bond coat with the best performing TBC to provide a coating system with maximum service life at the ATS operating conditions.

MATERIALS

Single Crystal Blade Development

To enhance performance and reliability, single crystal (SC) blades will be incorporated in the ATS engine. A casting development program was carried out to demonstrate castability of large industrial turbine blades in CMSX-4 material. Existing 501F engine blade tooling was used to cast single crystal blades. The castings were evaluated by grain etching, selected NDE methods and dimensional inspection methods to determine their metallurgical acceptability. After several trials, excellent results were obtained on a solid and a cored stage 3 shrouded blade, thus demonstrating that SC blades are castable in CMSX-4 alloy. Further process development is still needed to improve the yield.

A development program is in progress to optimize post-cast heat treatment, evaluate effects of grain defects, generate SC material design data, and further develop the casting process.

Ceramic Components

Ceramic components have potential for reducing cooling requirements, enhancing engine performance and improving turbine component reliability. Investigations are being carried out into the applicability of ceramic components, such as combustors, transitions, and ring segments into large industrial gas turbines. A conceptual design of turbine ring segments using ceramic matrix composite material is in progress.

FUTURE ACTIVITIES

The Phase 2 technology development efforts have progressed sufficiently to demonstrate that the ATS program goals are obtainable in the 5-year time frame. The Phase 2 technology developments currently under way and those planned for Phase 3 should resolve the identified technical barrier issues, so that the performance, emissions, and RAM objectives may be achieved. A concerted combustion development effort will be initiated to provide the ATS engine with an environmentally benign combustion system limiting NOx emissions to single digits, while achieving stable, reliable, long service life operation. The cooling program will lead to optimized cooling designs which should result in enhanced performance, mechanical integrity, and reliability. The advanced coating systems and materials development programs will make a significant contribution to enhanced turbine component reliability as well as increased performance. The aerodynamics and advanced sealing developments are intended to ensure that the ATS program thermal performance goal of greater than 60% net plant efficiency will be achieved. Significant technological advancements have already been achieved in Phase 2. The technology development programs planned for Phase 3 will build upon these successes and advance the gas turbine state-of-the-art, thus making a major contribution to the success of the Westinghouse ATS Program.

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..KW-COMBUSTION TURBINES. STEAM COOLING. AIR COOLING. COAL-FIRED TURBINES. COMBUSTION. ULTRA LOW NOx SHROUD COOLING. ADVANCED COOLING.

..AB-This paper describes the technologies that are being developed or extended beyond the current state-of-the-art to achieve Advanced Turbine Systems (ATS) Program goals. The Westinghouse ATS plant is an advanced closed-loop cooled combined cycle, based on an advanced gas turbine engine incorporating novel design concepts and enhancements of existing technologies. The ATS engine is a fuel-flexible design operating on natural gas with provisions for future conversion to coal or biomass fuels. It is based on proven concepts employed in 501F and 501G engines. To achieve the required performance and reliability the engine will include closed-loop steam cooling advanced materials and coatings, and enhanced component performance. To minimize NOx emissions, an ultra-low NOx combustion system will be incorporated. To ensure technical success, development programs are being conducted on the following: closed-loop steam cooling, advanced materials and coatings, component aerodynamic performance, flow visualization, optical diagnostics, combustion generated noise, and catalytic combustion.

TP-96173

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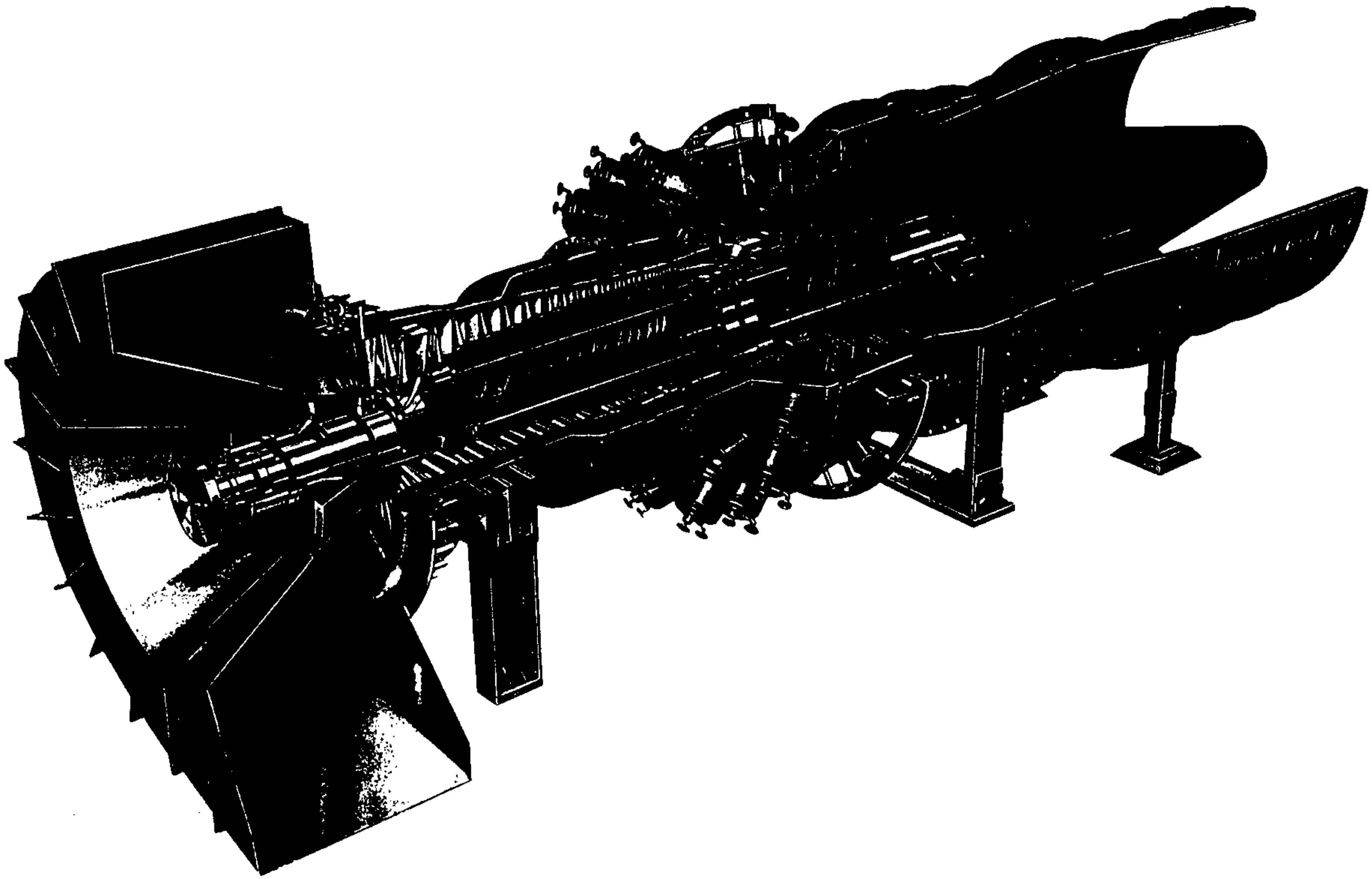
501G Overview



501G Major Design Goals

- High efficiency
- Largest Power Density
- Low cost of electricity
- Proven advanced design concepts
- High reliability
- Low life cycle costs

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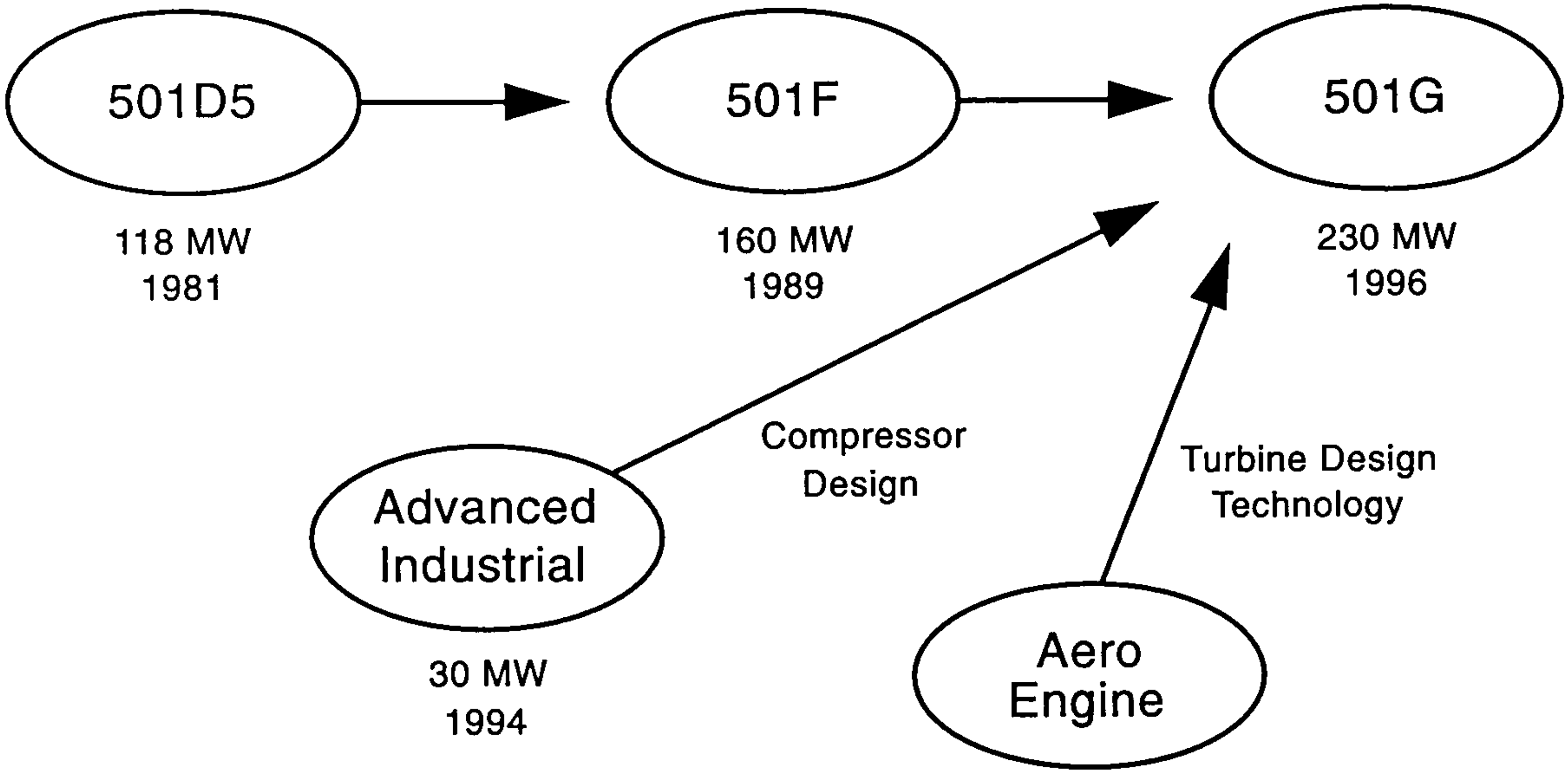
501G Design Approach

- Capitalize on Alliance strengths and resources
  - MHI, FiatAvio
  - Rolls-Royce
- Maintain time-proven fundamental design
- Utilize state-of-the-art technologies
  - Aero computer design codes
  - Advanced materials including thermal barrier coatings
- Aggressive concurrent engineering
  - Engineering / Manufacturing / Vendors
- Independent Design Oversight Board

501G Heritage



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501G Design Approach

- Maintain metal temperatures within proven experience
- Comprehensive component verification programs
  - Complete compressor
  - High Temperature Demonstration Unit
- 3-D flowfield analysis for optimum efficiencies
- RAM and Risk Analysis programs for maximum reliability
- Manufacturing considerations a primary objective
- Low emissions program to satisfy world-wide needs

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501G: World Leader in Performance

	Simple Cycle	Combined Cycle*
Power, MW	230	345
Efficiency, (LHV)	38.5%	58%
Heat rate - Btu/kWh	8,860	5,883
Airflow, Lb/Sec	1,200	
Exhaust Temp., °F	1,100	

* Mature Rating

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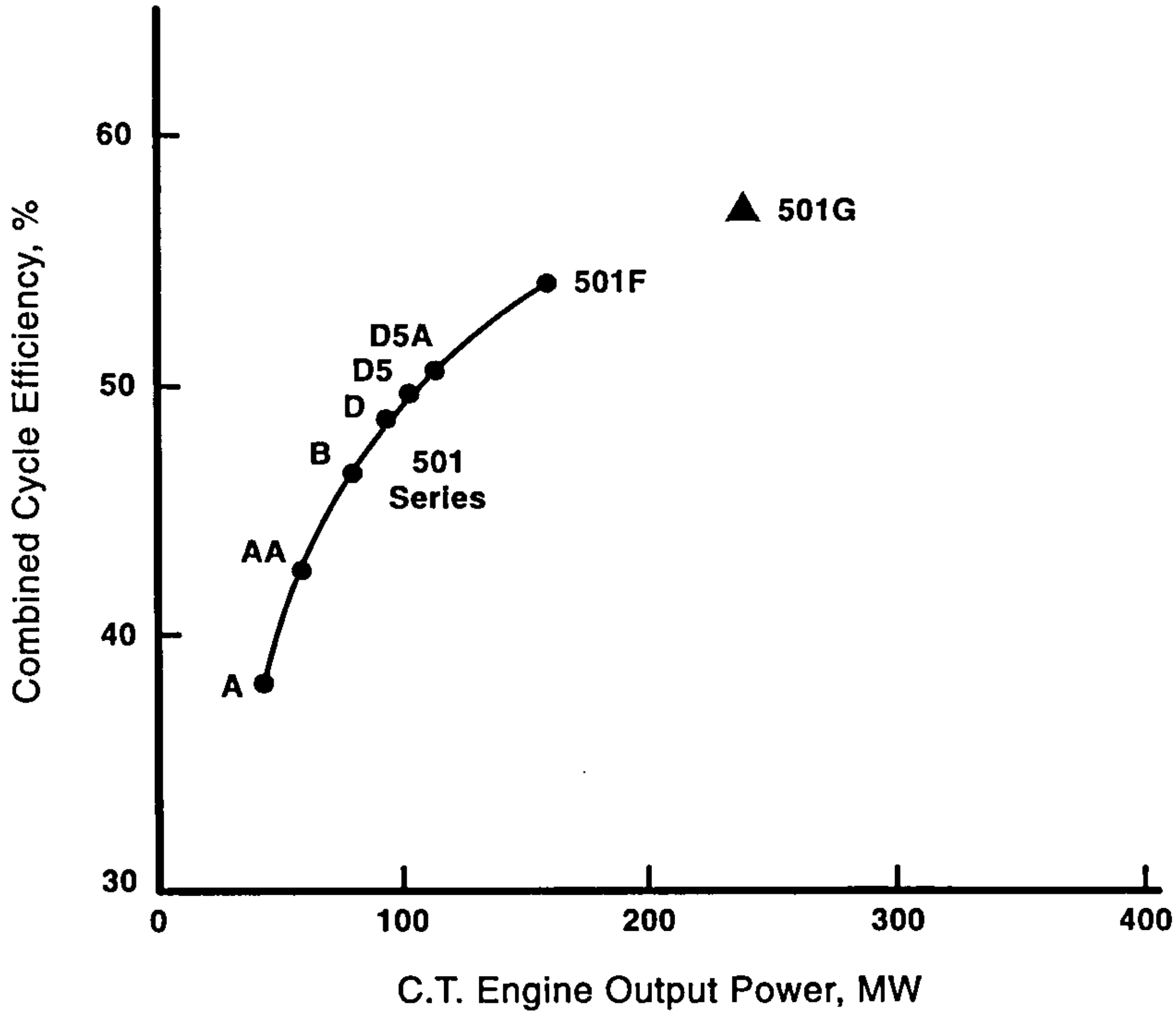
501G Evolution
World Leader in Performance



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501G Emissions for 1996 Shipment

	Natural Gas (Dry)	Oil (Water Injection)
• NOx	< 25 ppm	< 42 ppm
• CO	< 10 ppm	< 25 ppm
• UHC	< 10 ppm	< 15 ppm

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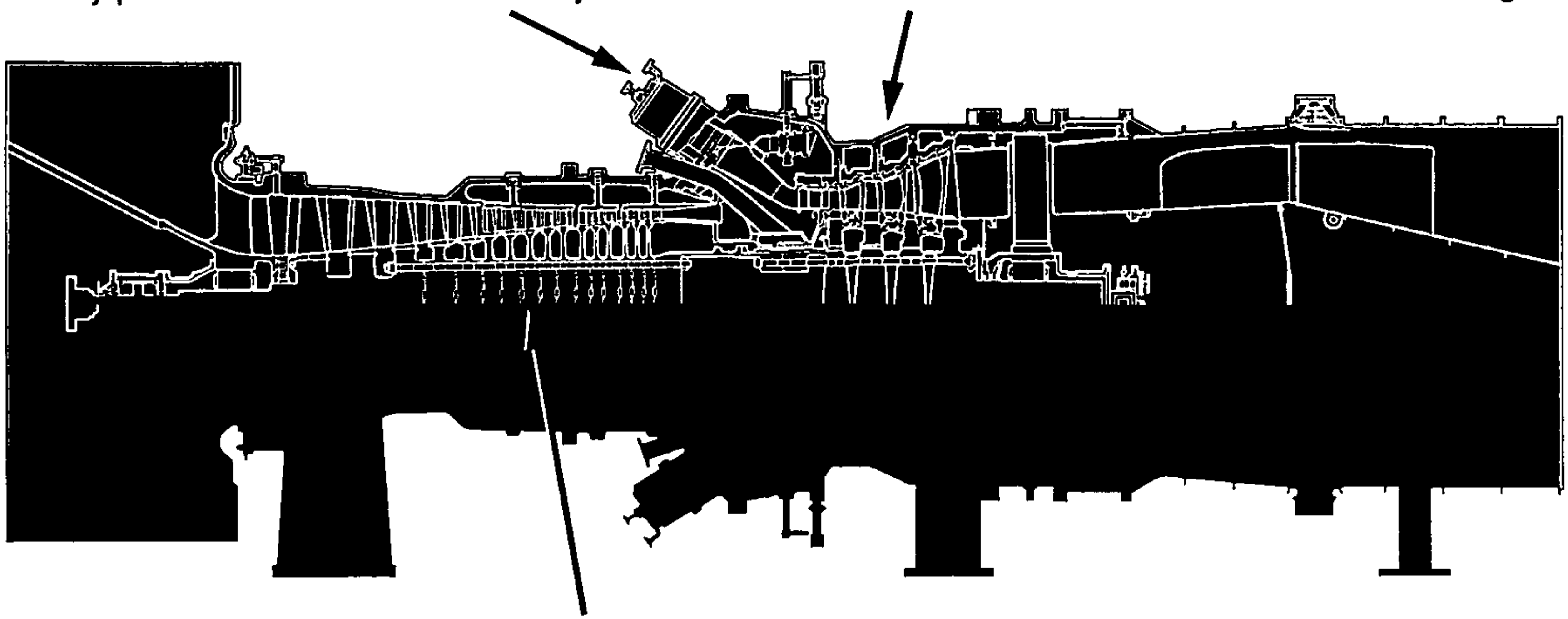
501G Design Features

- 16 Can-Annular Dry Low NOx Combustors
- 2600 °F Class Turbine
- Steam Cooled Transitions
- By-pass Valve for Part Load Stability

- 4 Stages of Full 3-D Blades and Vanes
- 15% Fewer Blades and Vanes than 501FA
- 3-Stages Air Cooled
- Aeroderivative Proven Materials and Coatings



- 17 Stage Advanced Profile Airfoils
- 19.2 to 1 Pressure Ratio
- Bolted Rotor Construction
- One Variable-IGV to Maintain Part Load Performance

501 Evolution

Engine	501A	501B	501D	501D5	501D5A	501F	501G
Commercial Operation	1968	1973	1976	1982	1995	1992	1997
Power, MW	45	80	95	107	119	162	230
Rotor Inlet Temp., °F	1615	1819	2005	2070	2150	2350	2583
Air Flow, Lb/Sec	548	746	781	790	832	961	1200
Pressure Ratio	7.5	11.2	12.6	14	14.2	15	19.2
No. Comp. Stages	17	17	19	19	19	16	17
No. Turbine Stages	4	4	4	4	4	4	4
No. Cooled Rows	1	3	4	4	4	6	6
Exhaust Temp., °F	885	907	956	981	1004	1083	1100
Heat Rate (Btu/kWh)							
LHV - ISO Gas							
Simple	12,600	11,600	10,925	10,040	9,930	9,550	8,860
Combined	9,000	7,350	7,280	7,055	7,024	6,250	5,883

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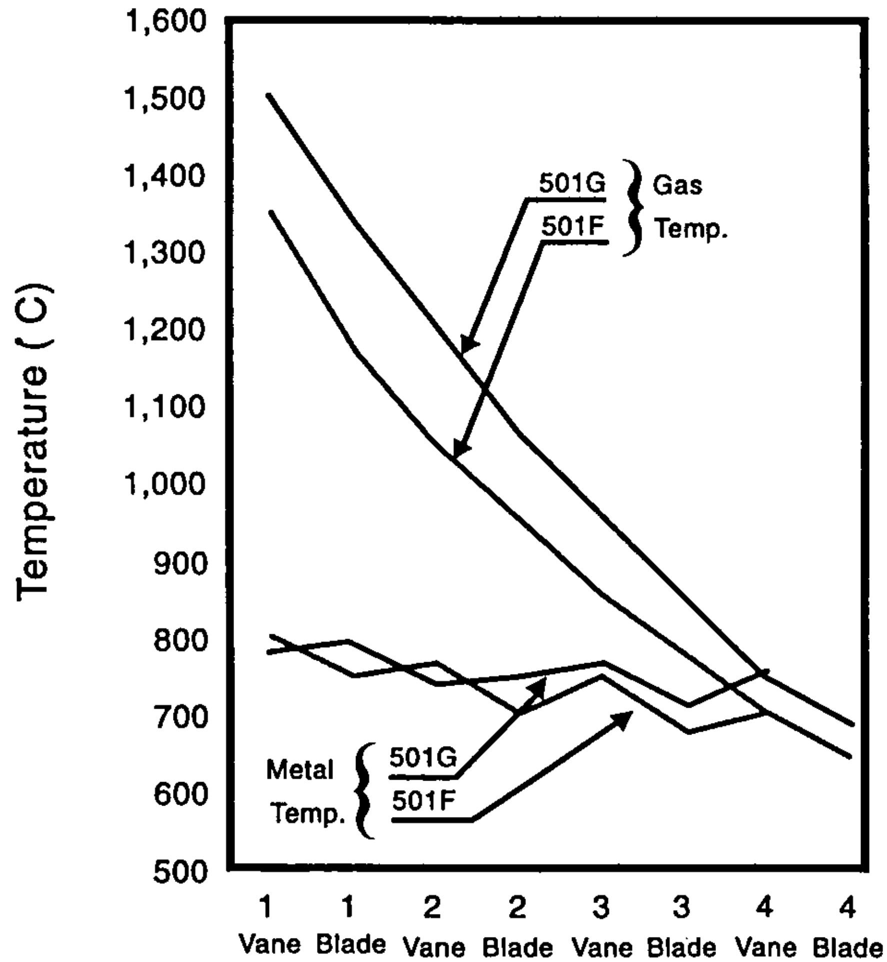
501G Proven Design Features

- Compressor-end drive
- Two-bearing rotor
- Axial exhaust
- Tilting pad bearings

Proven Design Features

Feature	251	501D	501F	501G
2-bearing rotor	✓	✓	✓	✓
4-stage turbine		✓	✓	✓
Individual combustors	✓	✓	✓	✓
Horizontal split casing	✓	✓	✓	✓
Cold end drive	✓	✓	✓	✓
Single Row 1 vanes	✓	✓	✓	✓
Cooled & filtered rotor air	✓	✓	✓	✓
Walk-in enclosure	✓	✓	✓	✓

Proven Blade and Vane Metal Temperatures



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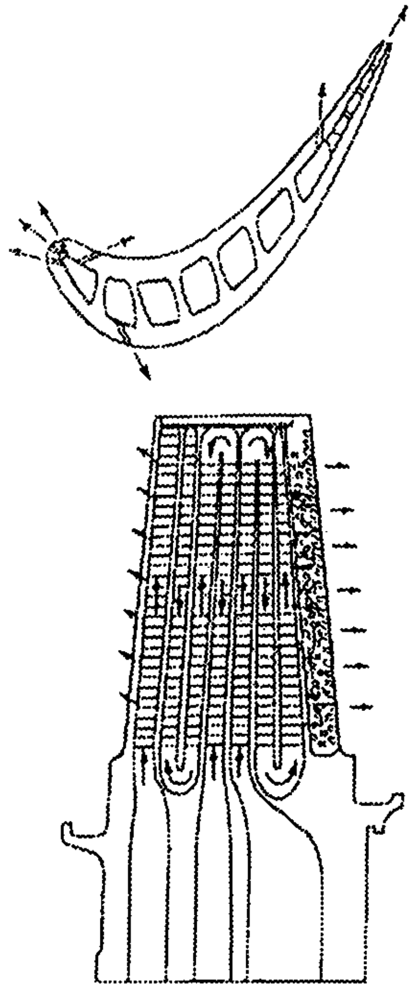
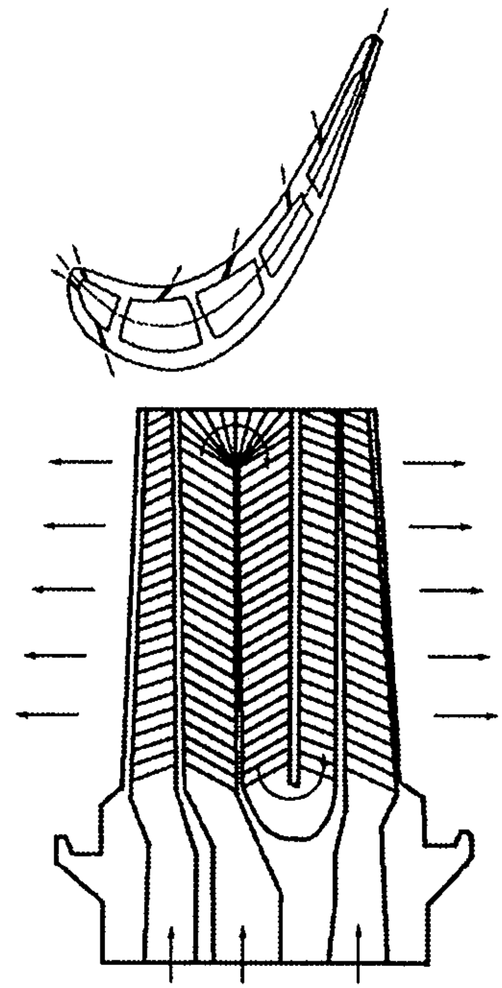
Row 1 Blade Evolution



501F



501G

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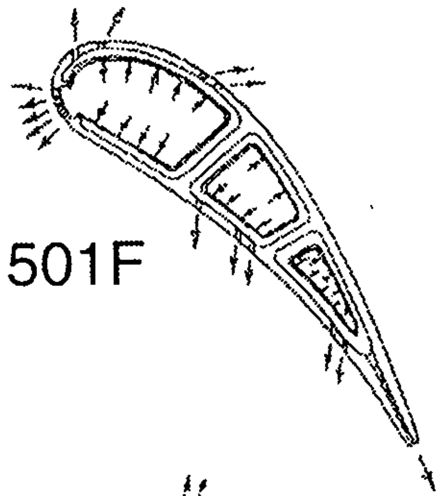
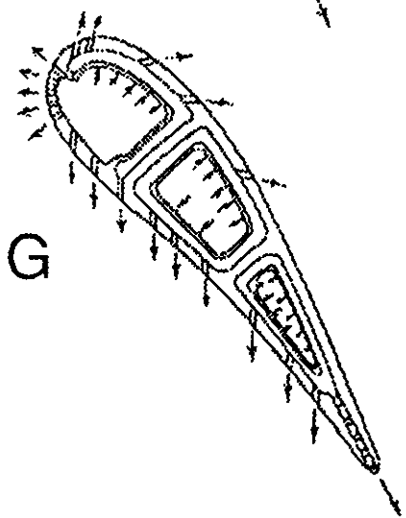
Row 1 Vane Evolution



501F



501G

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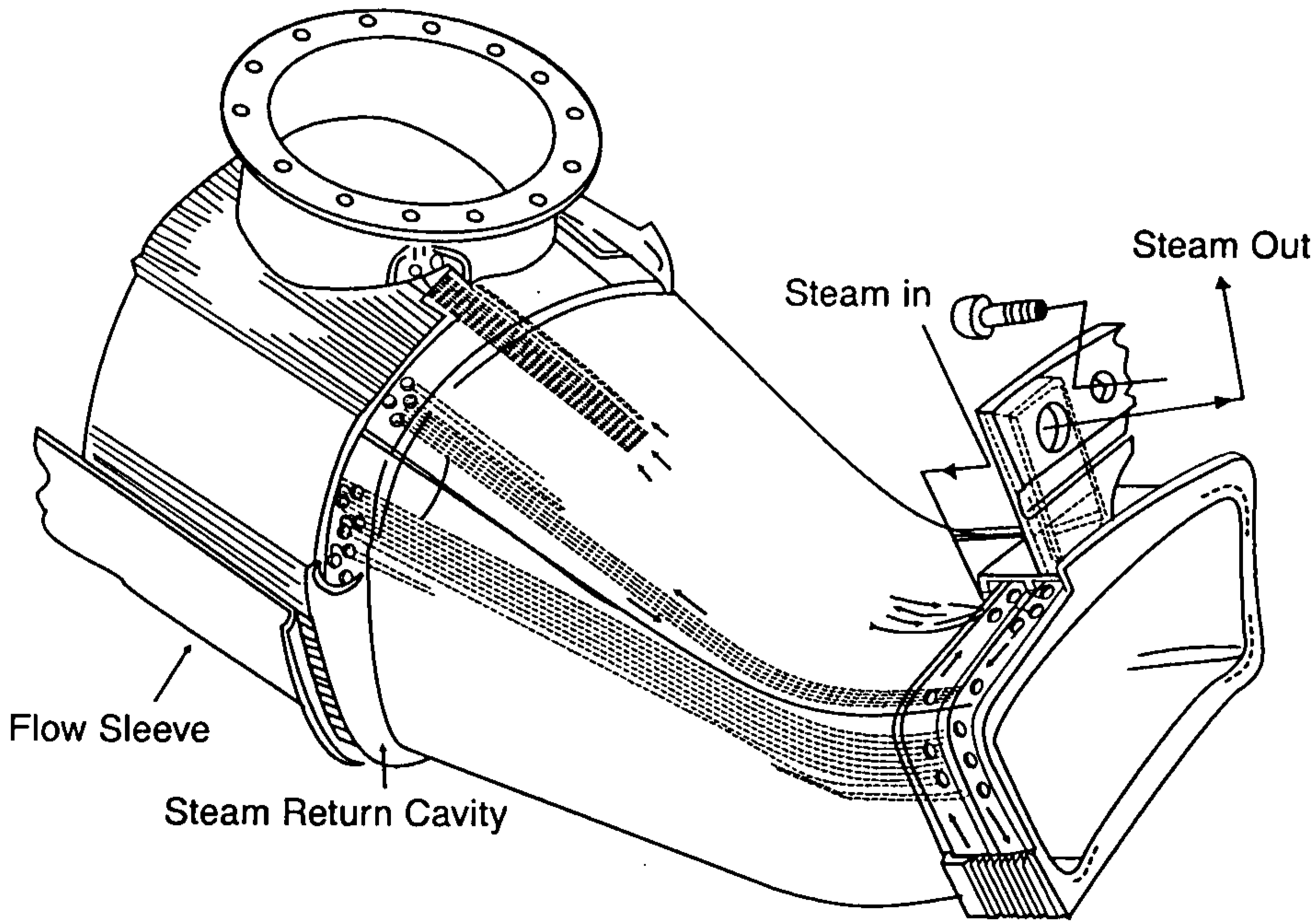
Transition - Steam Cooling



501G Lowest Life Cycle Costs

Number of Critical Parts

	501F	501G
Combustors	16	16
Row 1 Vane	32	32
Row 1 Blade	72	54
Row 2 Vane	48	36
Row 2 Blade	66	50
Row 3 Vane	48	42
Row 3 Blade	112	101
Row 4 Vane	56	42
Row 4 Blade	100	90
Total Blades & Vanes	534	447
Total Cooled Blades & Vanes	378	315

501G Inspection Intervals

Inspection Type	Hours	Starts
Combustor	8,000	400
Turbine	24,000	1,200
Major	48,000	2,400

Gas Fuel

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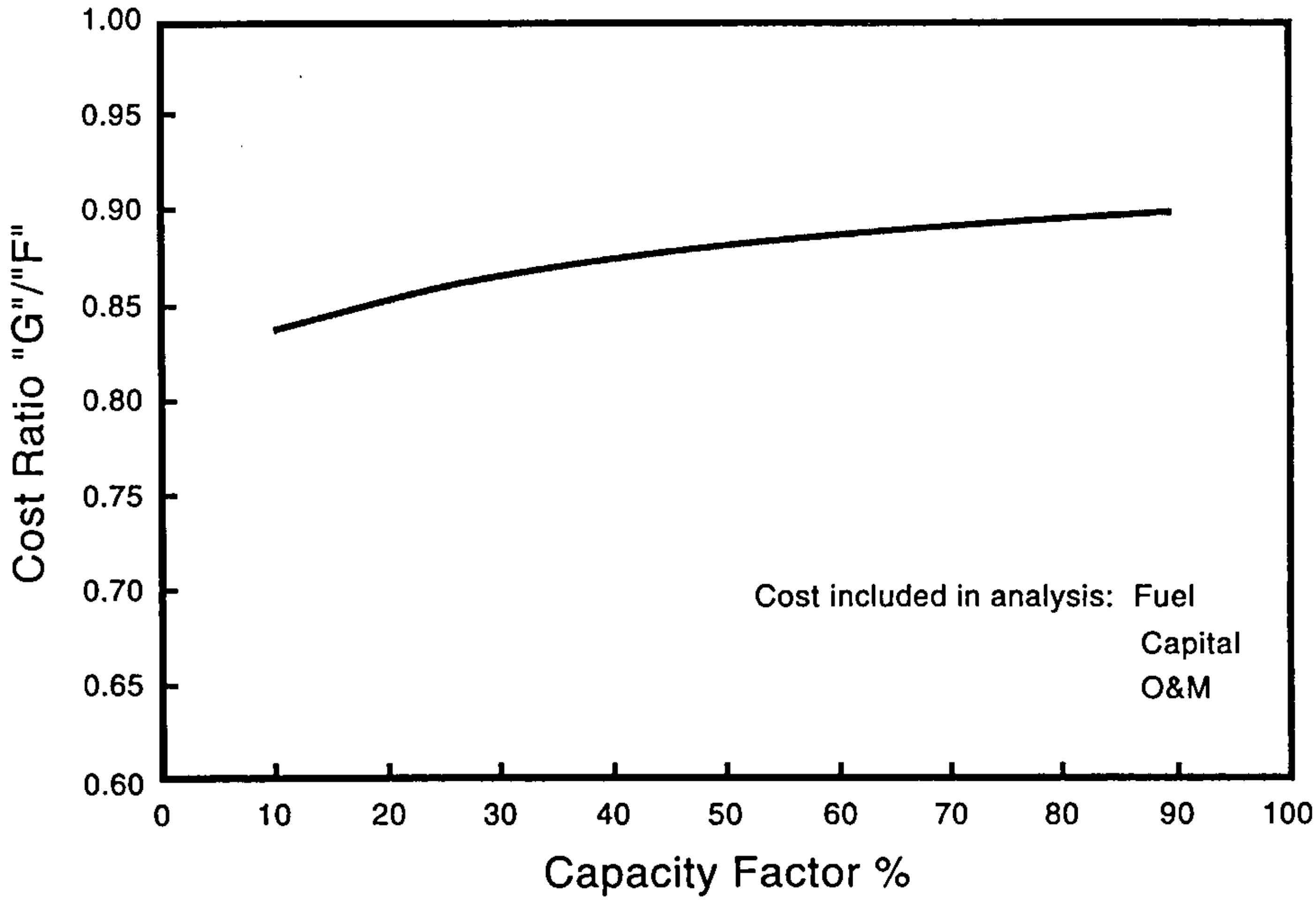
Relative Annual Cost of Power

"F" vs "G" Technology - Combined Cycle Plants



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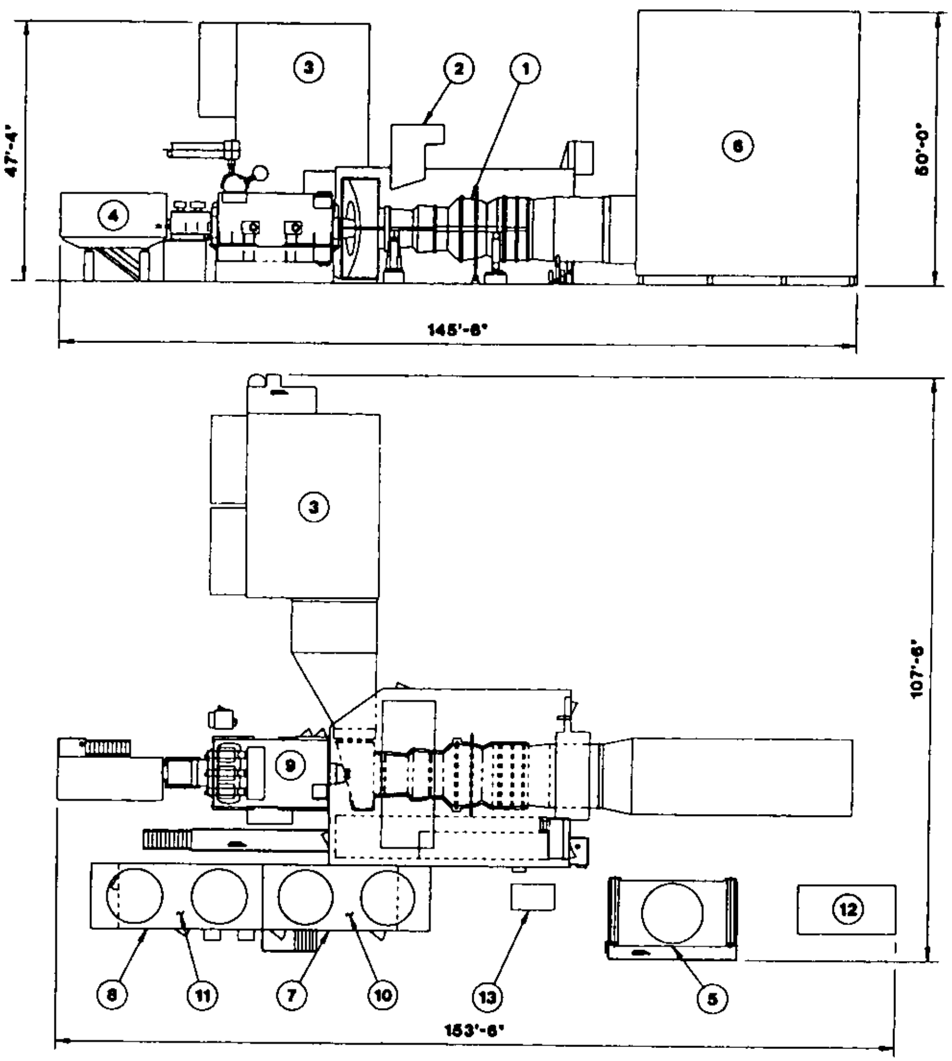
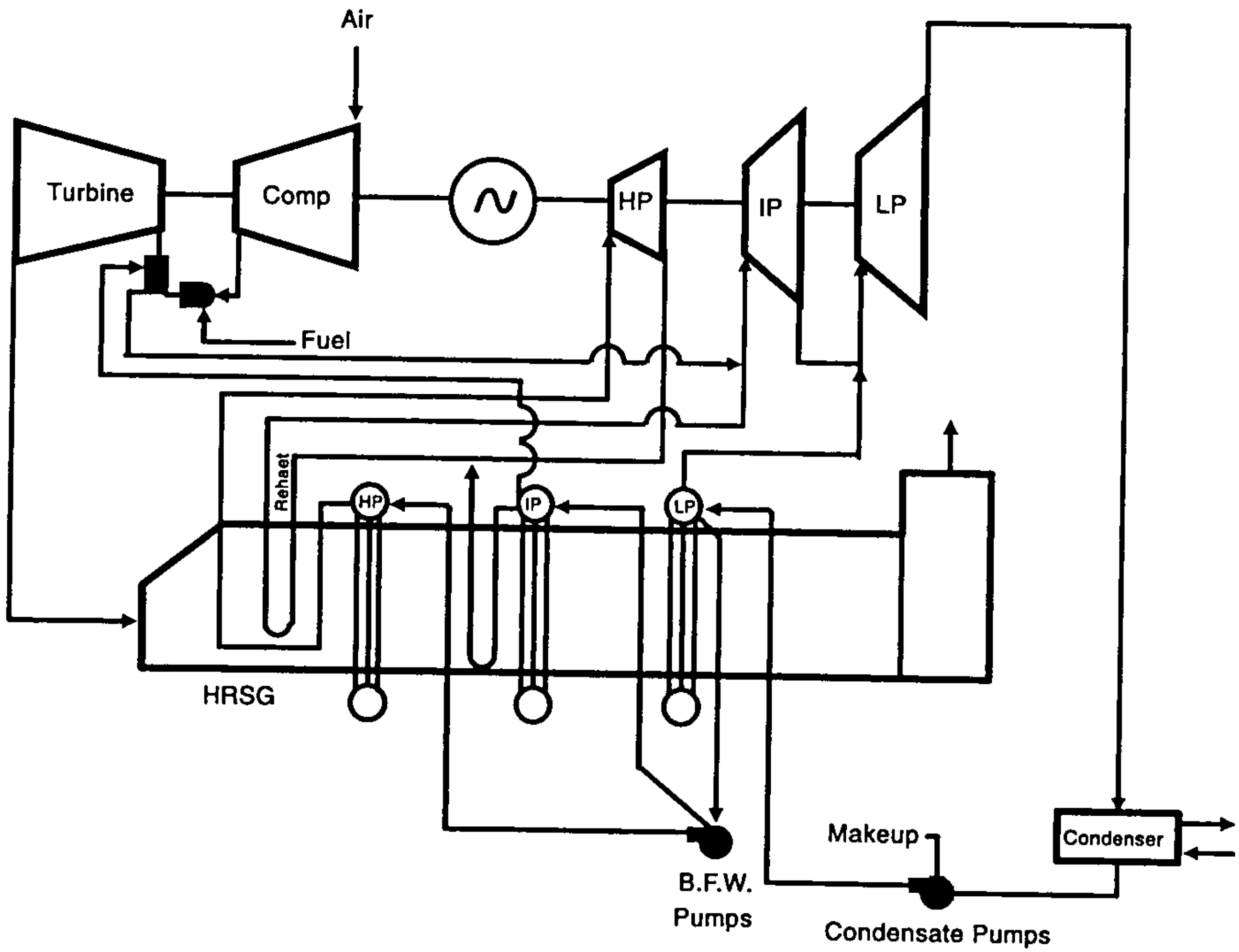
501G EconoPac - Standard Configuration



Legend

1. Combustion Turbine
2. Combustion Turbine Enclosure
3. Turbine Air Inlet Filter
4. Starting Package
5. Air-to-Air Cooler
6. Exhaust Stack
7. Mechanical Package
8. Electrical / Control Package
9. Generator
10. Lube Oil Cooler
11. Generator Glycol Cooler
12. CO_2 Fire Protection Storage
13. Compressor Wash Skid

501G 1X1 Reference Plant



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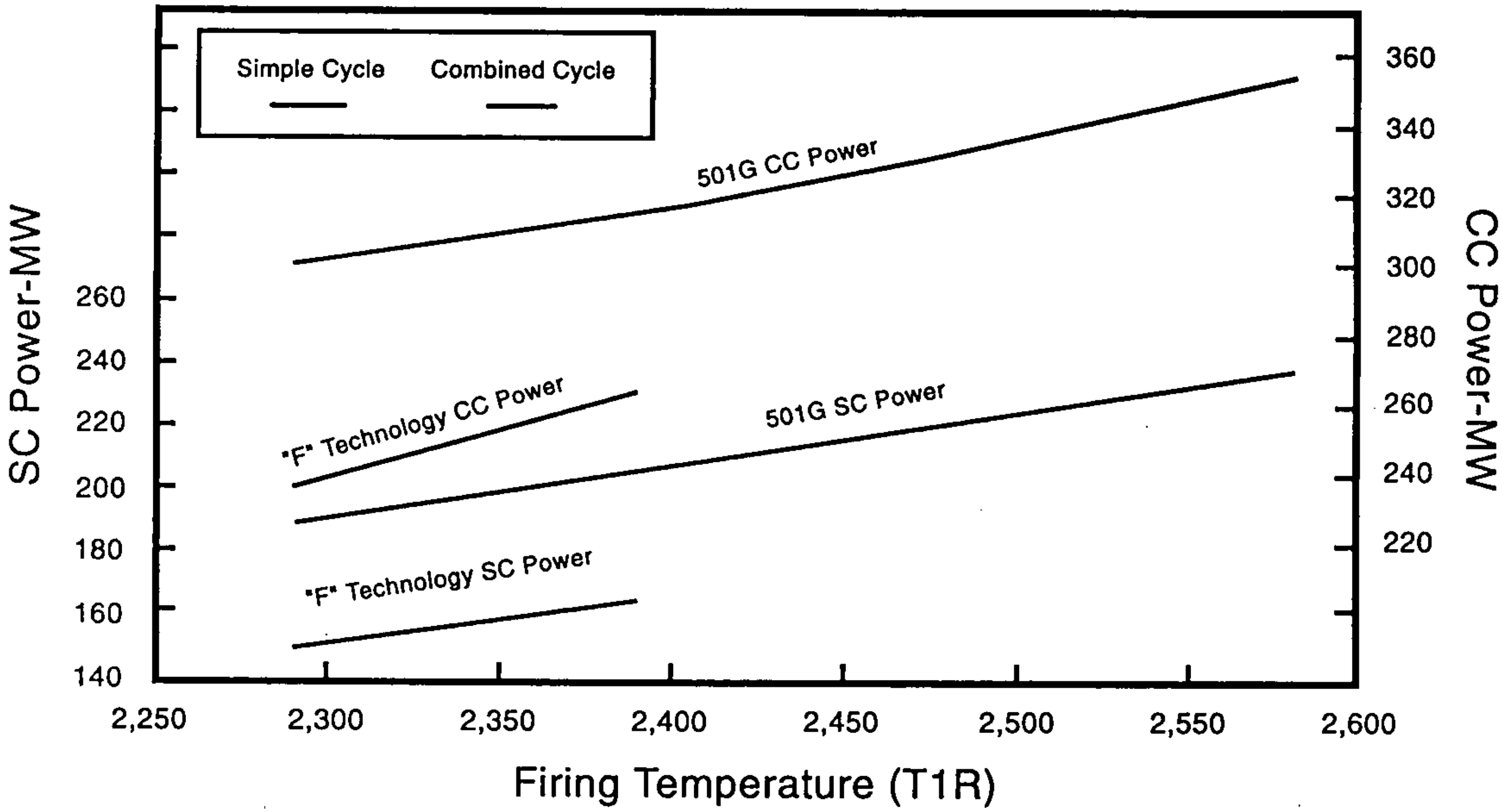
501G Power vs Firing Temperature



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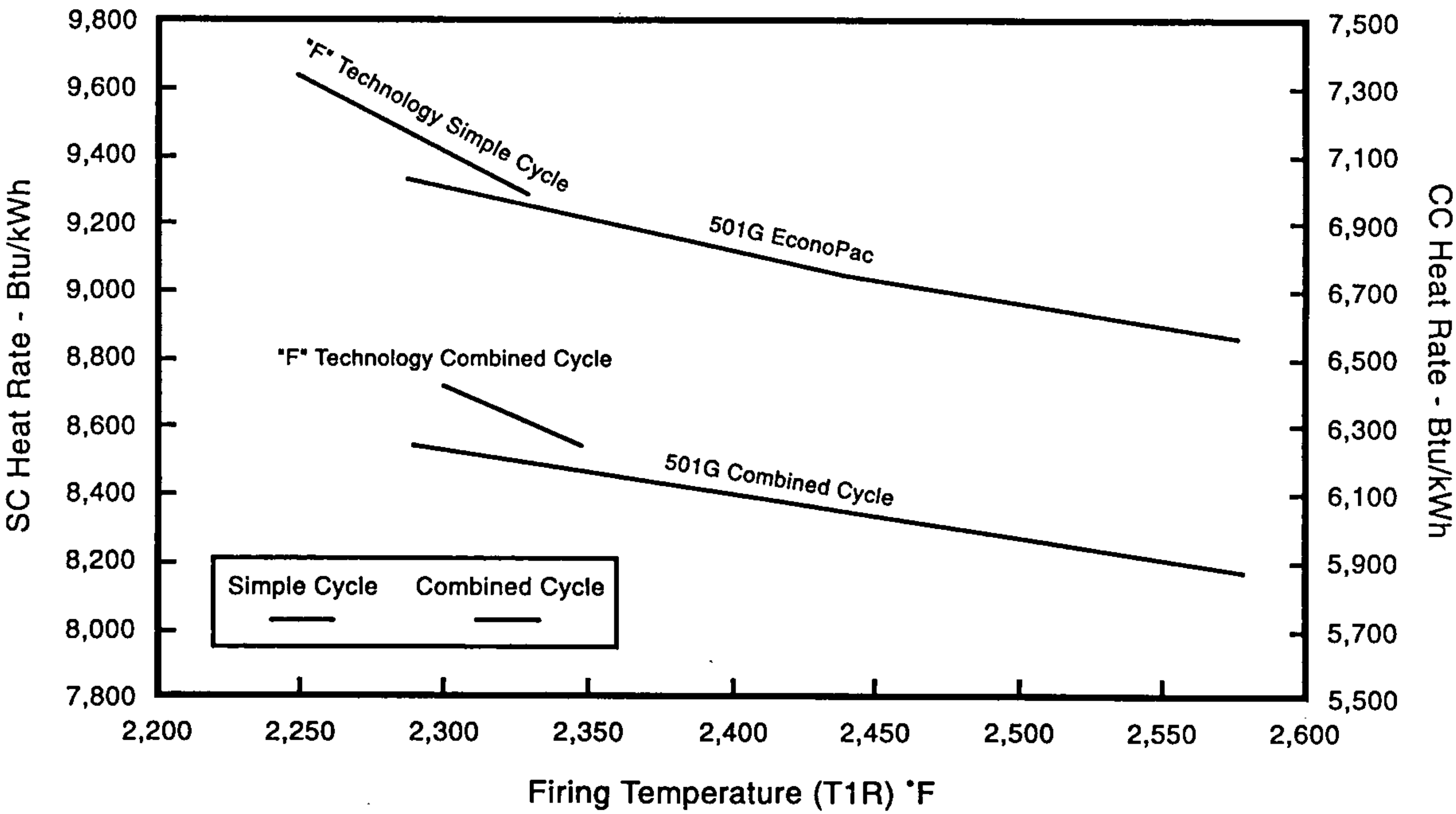
501G Heat Rate vs Firing Temperature



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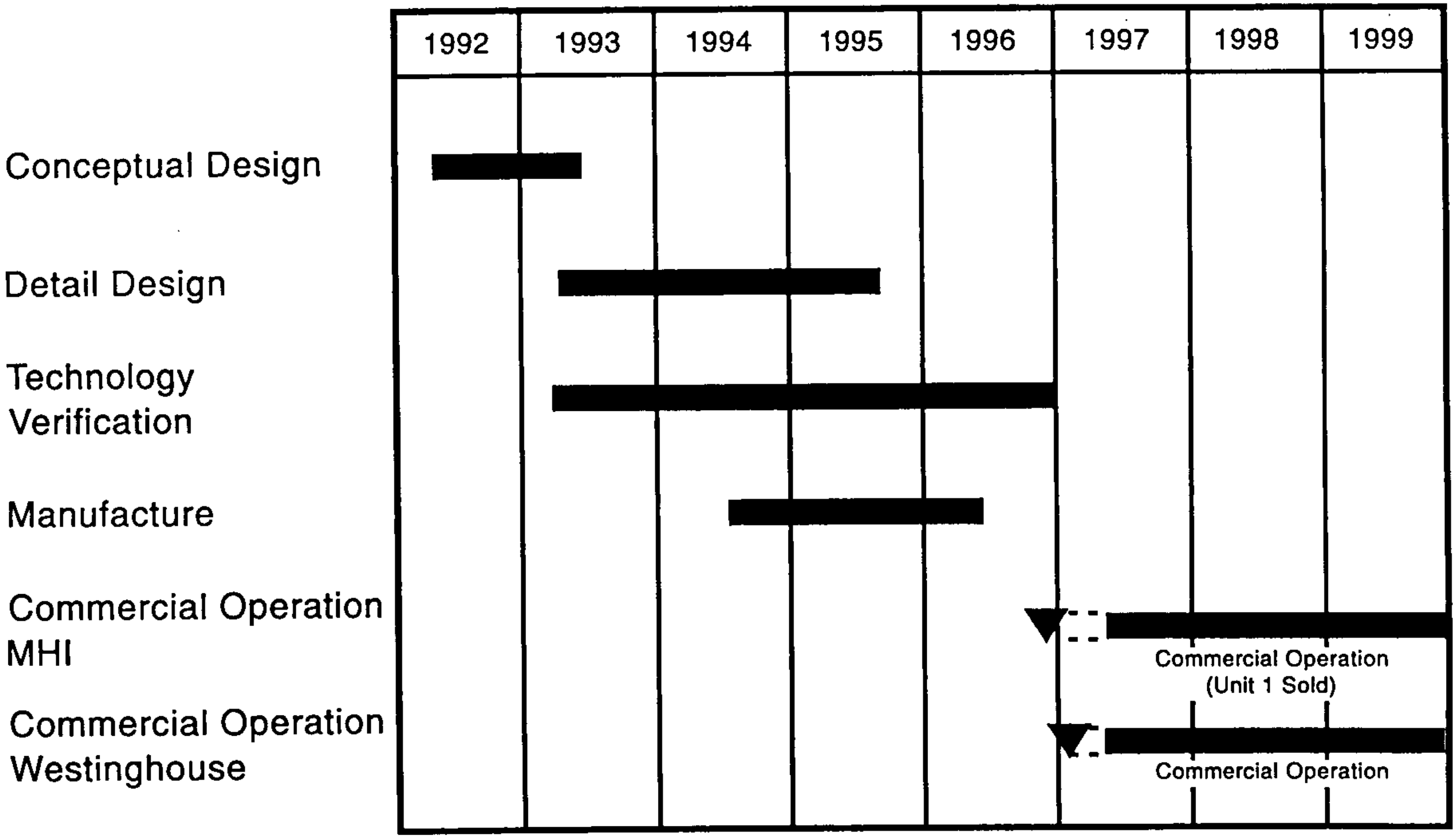
501G Component Verification Tests

- Combustor (full scale, stability, emissions)
- Transition steam cooling
- Model turbine aerodynamic
- Film cooling of vane and blade (cascade test)
- HTDU (0.6 scale of Row 1 vane and blade)
- Model blade tip cooling
- Scaled compressor
- Full load engine

501G Development Schedule



Combined Cycle Plant Economic Analysis

Financial Performance for a Non-Utility Generation Project
(Assumes 25 year levelized power rates, includes SCR)

90% Capacity Factor

	Project IRR 25 Years	Average Debt Coverage Ratio
"F" Technology	20.0%	1.52
"G" Technology	35.2%	2.18

50% Capacity Factor

	Project IRR 25 Years	Average Debt Coverage Ratio
"F" Technology	20.0%	1.52
"G" Technology	33.4%	2.11

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Financial Analysis Key Assumptions

3% degradation on capacity
2% degradation on heat rate
Fuel cost = $3.00/MMBtu
25 year plant life
4% general escalation rate
No sale to steam host
Base capacity factor = 90%
Construction period = 27 months ("F" & "G" technology)
Owner's contingency = 5% of turnkey cost
Permitting, legal, misc. costs = $5 Million
Federal income tax rate = 34%
State income tax rate = 8%
Debt/Equity = 85/15
Debt term = 15 years
Debt interest rate = 10%
Required debt reserve = 6 months debt service
Levelized power rate set to have "F" Technology IRR = 20%

501G: The Economic Answer

- Highest Efficiency
- Largest Power Blocks
- Lowest Life Cycle Costs
- Lowest Cost of Electricity

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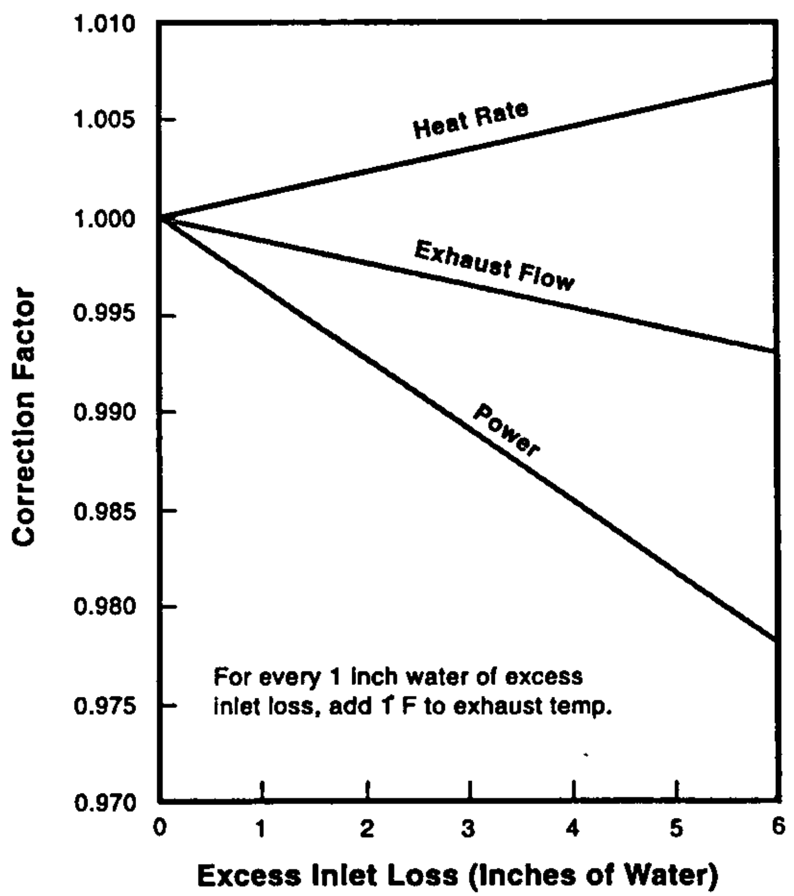
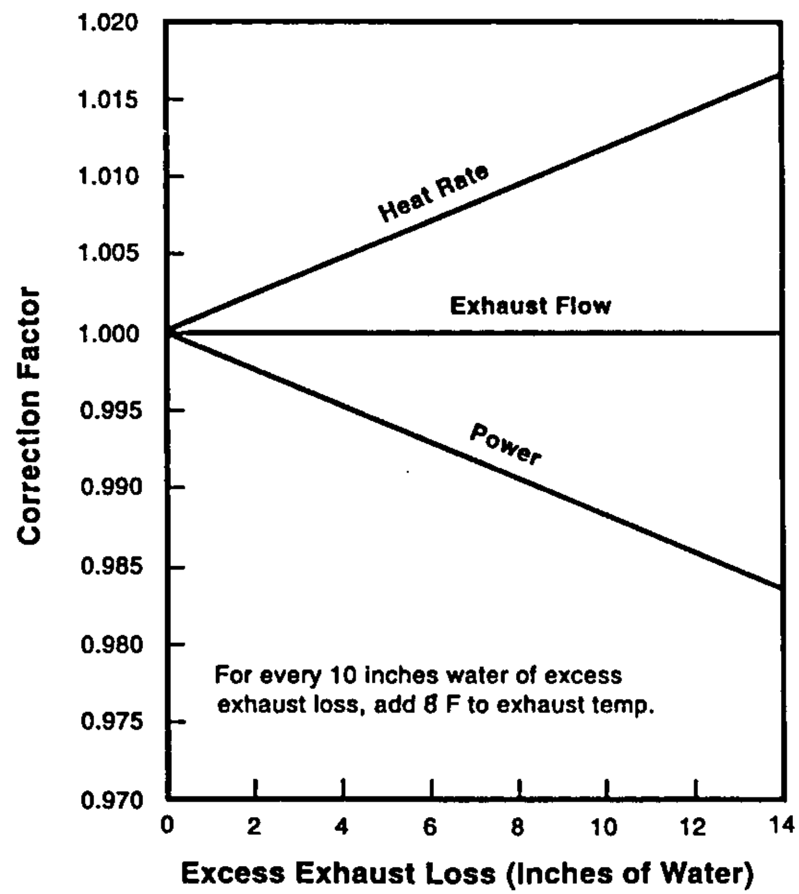
501G Performance Correction Curves





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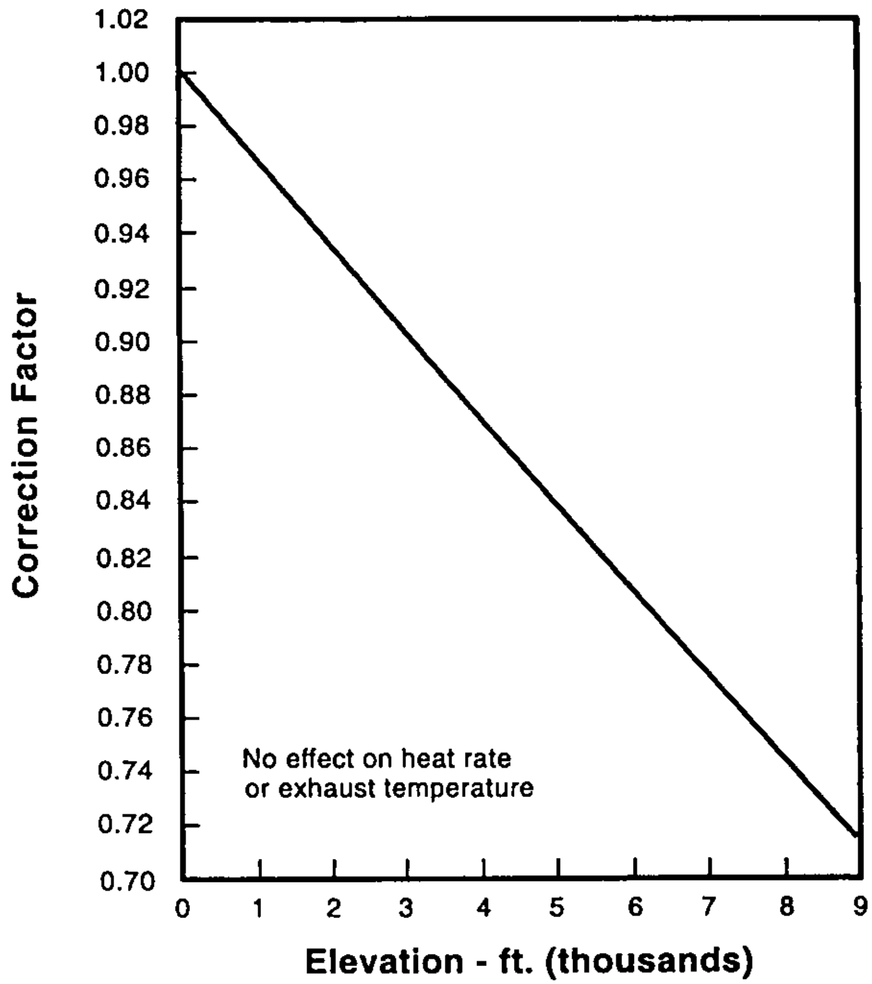
501G EconoPac System Performance



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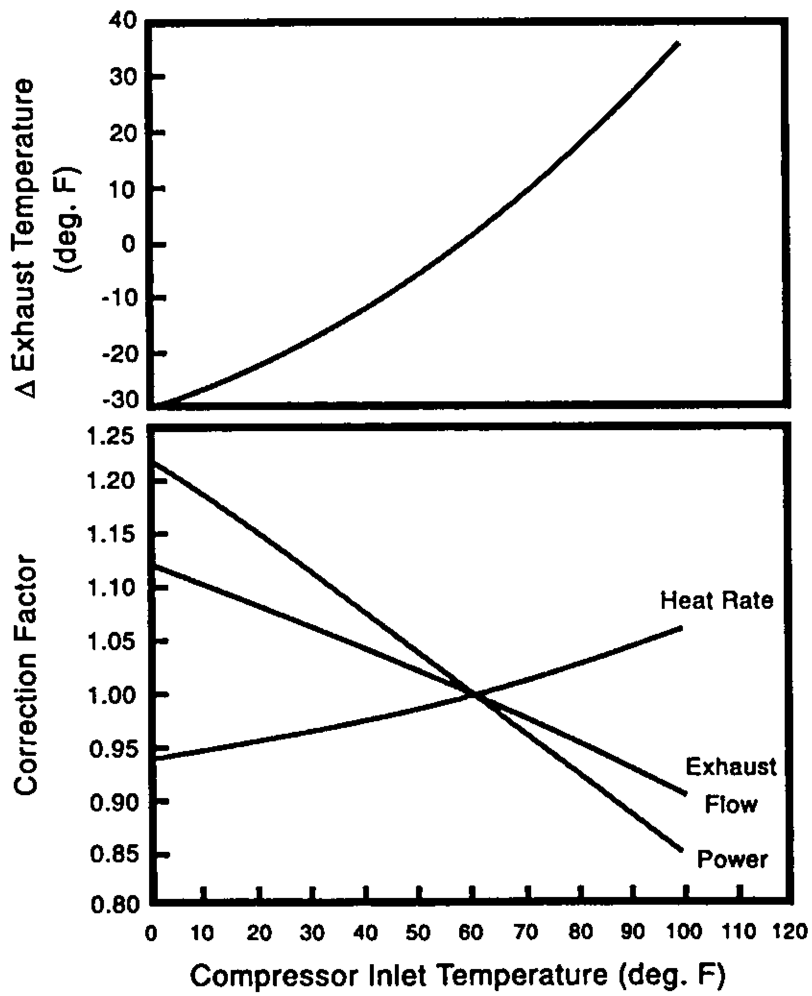
501G EconoPac System Performance



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WESTINGHOUSE COMBUSTION DEVELOPMENT
1996 TECHNOLOGY UPDATE

RICHARD J. ANTOS

COMBUSTION TURBINE ENGINEERING
POWER GENERATION TECHNOLOGY DIVISION

Technical Paper TP-96003

December 1995

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Westinghouse Combustion Development
1996 Technology Update

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Westinghouse Electric Corporation
Power Generation Business Unit
Orlando, Florida

Abstract
Westinghouse has aggressively incorporated advanced technologies into the Combustion Turbine family of engines. One area of intense development has been in combustion with a focus on reducing emission levels at increasing firing temperatures while maintaining world class operating reliability and availability.
Proactive combustion development programs have resulted in multiple options for achieving dramatic reductions in emissions with Ultra Low NOx combustion systems. This paper presents a description of the program organization, advanced technologies, and results of development efforts to date.

Background
Westinghouse and it's alliance partners have been leaders in the industrial combustion turbine development area for decades. Westinghouse's initial low emission combustion development began in the 1970's (Ref. 1). Westinghouse was also a leader in the early development of catalytic combustion with full scale testing occurring in the 1970's and early 1980's (Refs. 2 and 3). Initial Dry Low NOx systems were installed in MW701D in the early 1980's (Ref. 4). These innovations have provided a strong background facilitating the state of the art Ultra Low NOx systems that are now being implemented into in-service and production engines. The evolutionary approach to new technology introductions has allowed Westinghouse to continue to make product improvements while maintaining excellent reliability and availability.

Combustor Design Process
The Westinghouse design process (Figure 1) has been tailored for rapid, thorough development using state of the art techniques such as computational fluid dynamics (CFD) analysis and rapid prototype manufacturing utilizing SLA processes, intertwined with strategic use of atmospheric and high pressure test facilities. Inherent in this process is an optimization of operating parameters and a complete verification program including operation in an actual engine. The process is focused on maximizing the benefits of advanced analytical approaches such as CFD with the traditional testing approach.

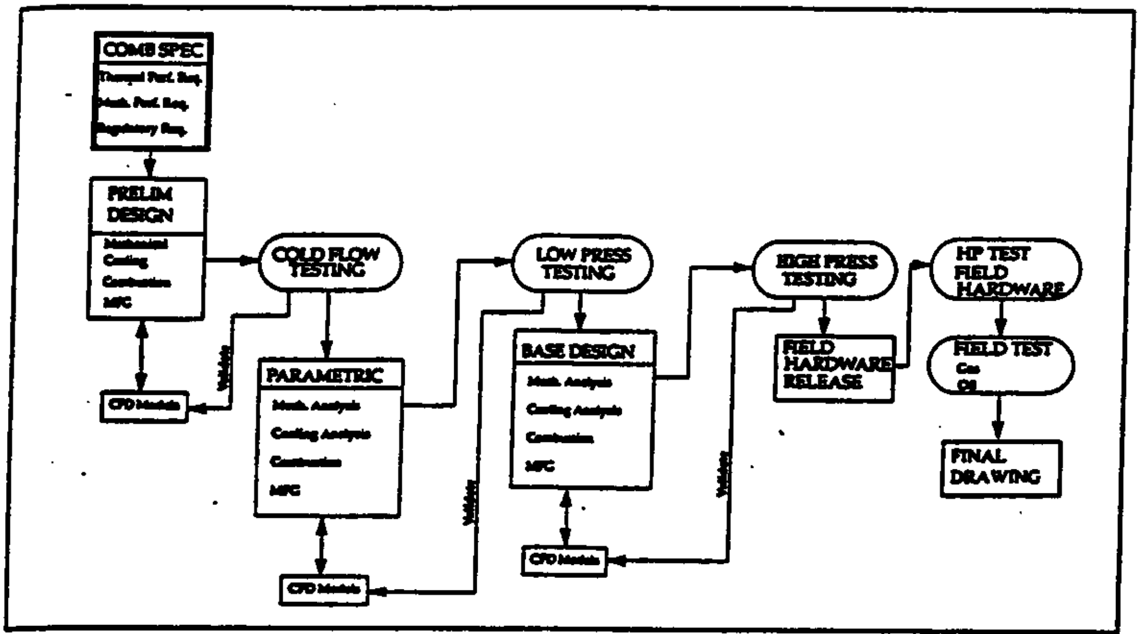

Figure 1 Flowchart Showing Combustor Development Process

CFD Analysis

Computational codes for airfoil flow analysis have been available for many years. However, the added complexity of including combustion reactions along with air and fuel chemistries has traditionally made combustion development a testing-focused technology. More recently though, the development and increased availability of high speed computers and advanced CFD codes have allowed computer modeling to be utilized in the day to day design process of advanced combustion systems. The use of CFD allows for much quicker turnaround in design concept evaluation, less expensive iterations, and most importantly more efficient and productive testing. An example of a CFD model is shown in figure 2. This model is utilized extensively in optimizing the Dry Low NOx design for specific applications.

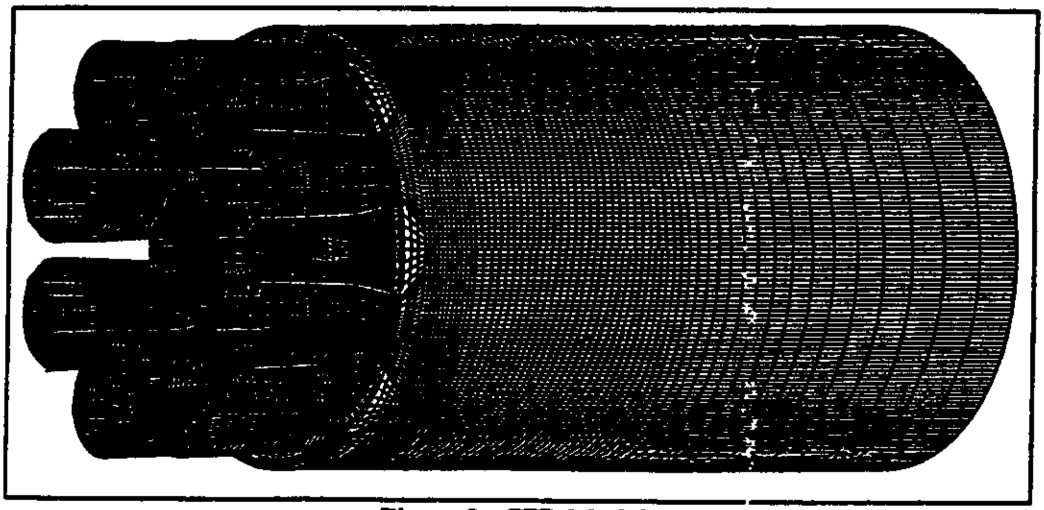

Figure 2. CFD Model

Test Facilities

Westinghouse utilizes a comprehensive array of test facilities to compress the development time and cost of combustion system development. The facilities range from airflow visualization for single combustors and turbine casing annular sectors to full scale, pressure, flow and temperature testing. The majority of these facilities are in the Westinghouse laboratories located near Orlando, Fl. Additionally, Westinghouse also has a full pressure test rig (Figure 3) installed at the Arnold Engineering Development Center in Tullahoma, Tennessee. A photograph of the test rig is shown in Figure 4. This Air Force testing ground provides the high pressure, high flow rates required for engine replication combustor testing (Ref. 5).

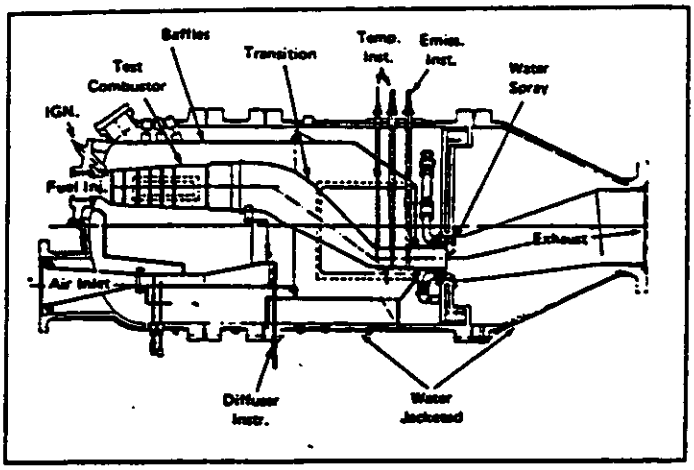

Figure 3

Dry Low NOx Design Philosophy

NOx emissions are governed by two primary mechanisms in combustion turbines, the conversion of atmospheric nitrogen at high temperatures (thermal and prompt NOx) and by the conversion of nitrogen in the fuel (called fuel bound nitrogen,). Fuel bound nitrogen is typically limited to heavy fuel oils or to coal derived gas fuels. Although this paper will focus on natural gas oriented combustors, Westinghouse is also a leader in developing combustors for fuel bound nitrogen laden gases. The Multi Annular Swirl Burner combustor utilizes a rich-quench-lean concept that greatly minimizes the fuel

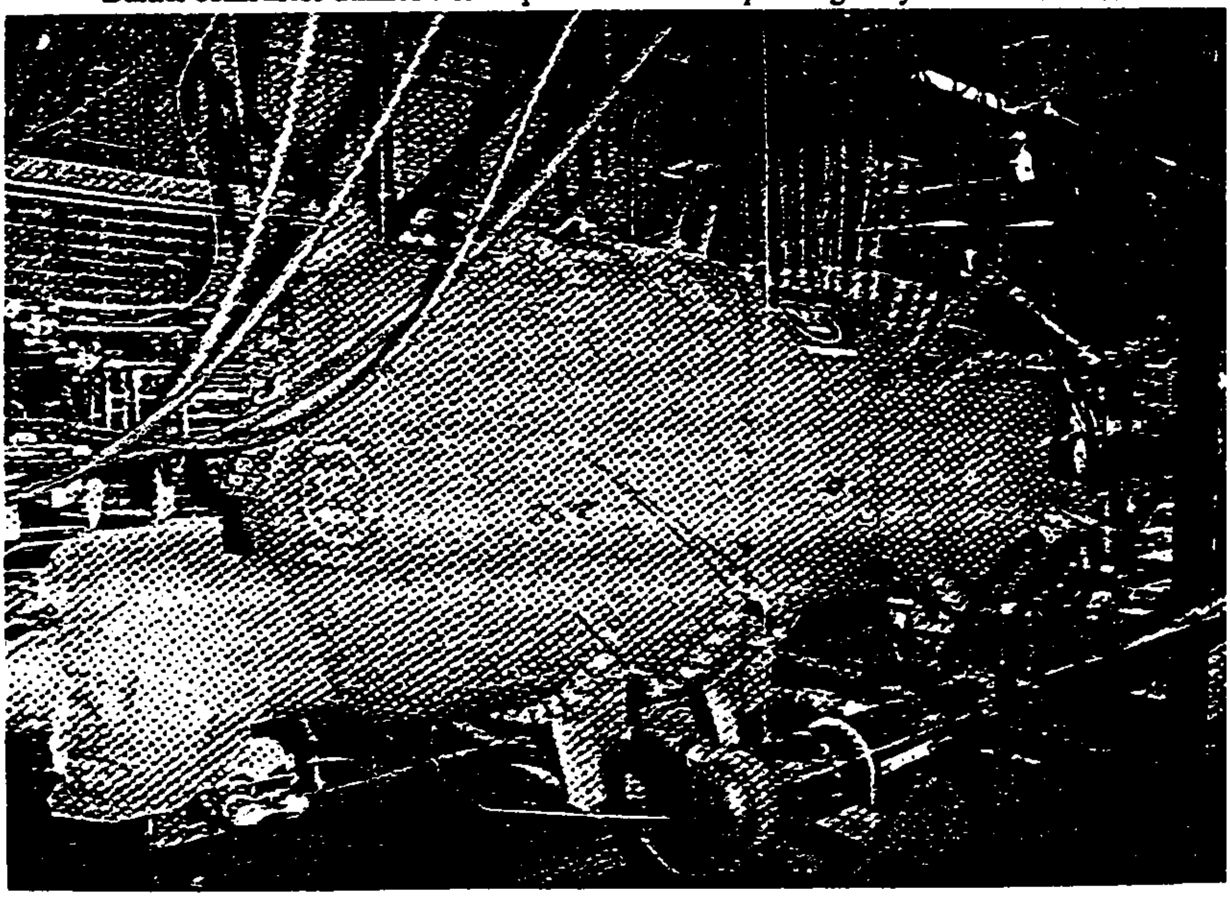

Figure 4 High Pressure Test Rig

bound nitrogen transformation to NOx (Ref. 6). For natural gas fuel the overriding factor in preventing NOx formation is the flame temperature. Flame temperatures in conventional diffusion flames approach 3500F, and in a typical engine would produce about 200ppm of NOx. Dry Low NOx combustors reduce the flame temperature through the partial premixing of fuel and air before ignition (Ref. 7). Figure 5 demonstrates that NOx production is directly related to flame temperature and which in turn is related to the local fuel to air ratio. Premixed combustors operate in a lean fuel/air condition resulting in a lower flame temperature and hence lower NOx. The current DLN system in service is the second generation system that utilizes lean premixed fuel mixing zones surrounding a central pilot. The central pilot provides a stability to the basic system. Third generation ULN systems that utilize fully premixed (no pilot) have been developed and are entering service.

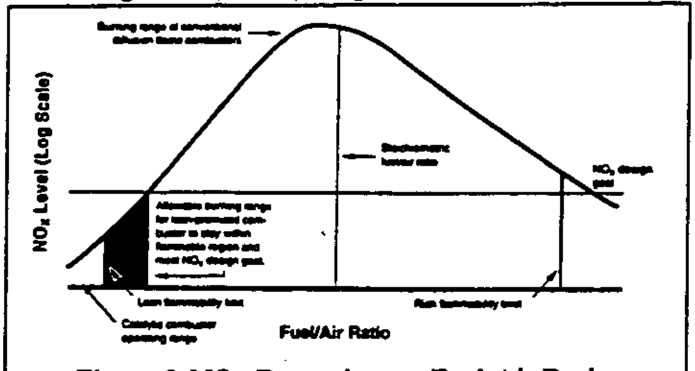

Figure 5 NO_X Formation vs. Fuel Air Ratio

Dry Low NOx Combustion Systems

The second generation dry low NOx combustion system was first introduced into service in 1992 and has an excellent performance record. Lead units now have over 20,000 hours of operation and have not required any shop repairs or replacements. This impressive performance is achieved by maintaining the combustor flame activity in a quiet, stable mode. A Dry Low NOx combustor (Ref. 8) is shown in Figure 6. The

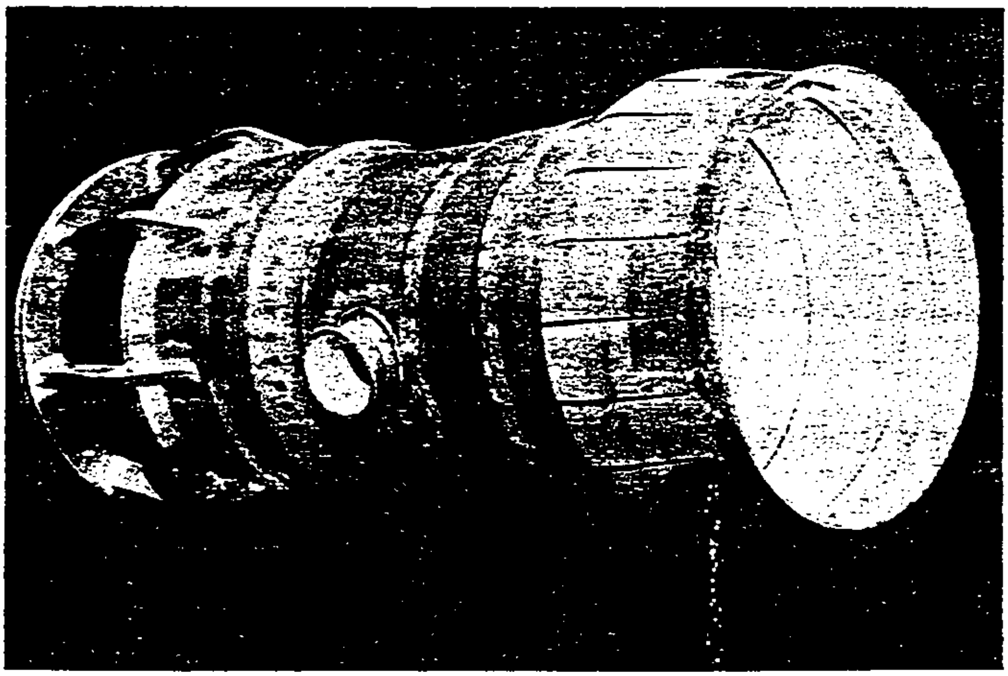

Figure 6

design includes the capability of performing on gas or liquid fuel including startup on either fuel as well as fuel transfers in either direction. The Dry Low NOx design is utilized in engine applications where as low as 15ppm NOx on gas is required.

Current Westinghouse 60 Hz engines include:

Engine	Power (MW)	Number of Combustors	Reference Firing Temperature (F)
251B12	48	8	2100
501D5	107	14	2070
501D5A	120	14	2150
501FA	160	16	2330
501G	230	16	2583

Continuing the evolutionary design approach (Ref. 9), all current engines utilize can-annular designs that facilitate use of common designs for different engines. Figure 7 shows essentially the same Dry Low Combustor in 501D5, 501D5A,701F, and 501F.

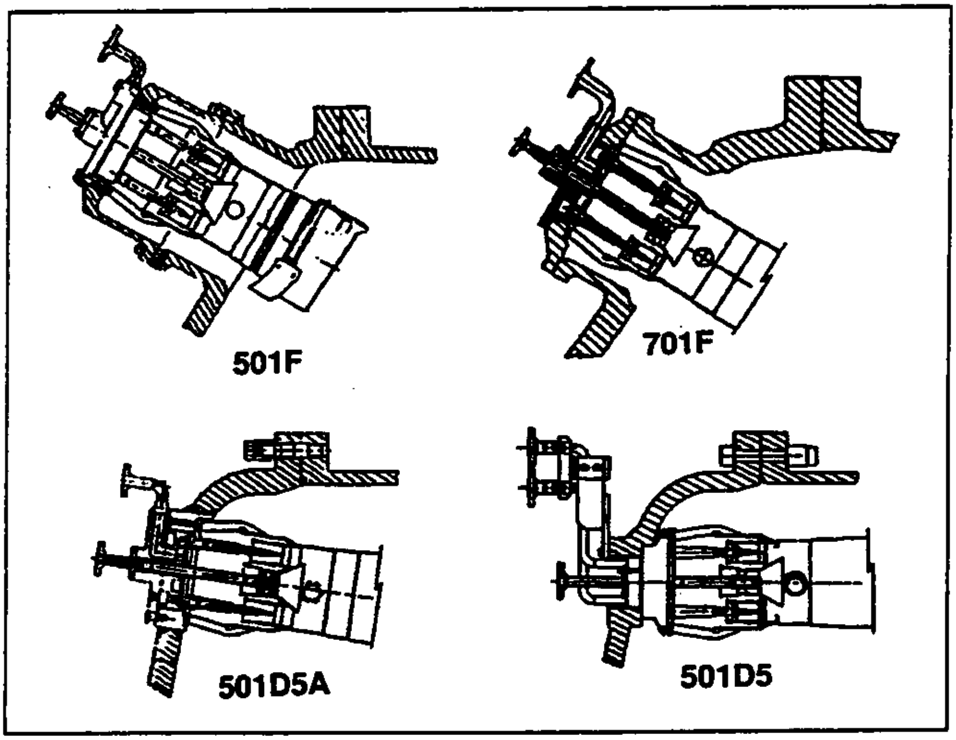

Figure 7

This design philosophy allows the experience base for all Dry and Ultra Low NOx combustors to be additive. The 501G is the newest addition to the Westinghouse 60Hz single shaft heavy duty combustion turbine product line (Ref. 10). This engine offers the option for Ultra Low NOx combustors in the initial production.

Power Augmentation

Power augmentation is defined as adding water or steam to the operating cycle to increase the power output of the engine. With diffusion flame combustors this was achieved by injecting diluent directly into the combustor to also reduce NOx levels. The addition of diluent increases the power output of the engine but generally has a detrimental effect on cycle efficiency. With DLN combustors the operating regime is closer to the lean limit and cannot accommodate large additions of diluent. Thus the augmentation is accomplished by injecting the water or steam into the combustor casing away from the combustor where it still provides the extra power but only a small portion of the diluent finds it way into the flame zone. Augmentation of the power output of an engine utilizing steam augmentation has been in commercial service for over a year. An

8000 hour inspection of this unit revealed that the use of power augmentation has not impacted the excellent operational performance of the combustor. As with non-power augmentation units, the combustors were inspected and did not require any rework prior to reinstallation.

Ultra Low NOx Program

Ultra Low NOx development activities continue on several fronts to positively impact new and existing engines. The comprehensive program includes focused development in advanced premix systems, dry low emission on liquid fuels, active stability enhancement, optical sensor measurements, as well as catalytic combustion.

Relative to premix combustors, Westinghouse has developed a full product line of Ultra Low Emission combustors. Multiple approaches were pursued to provide optimum performance for the wide range of operating conditions required for the fleet of new and operating engines.

Fully Premixed Designs

Piloted Ring

The Piloted Ring Combustor is shown schematically in Figure 8. The combustor consists

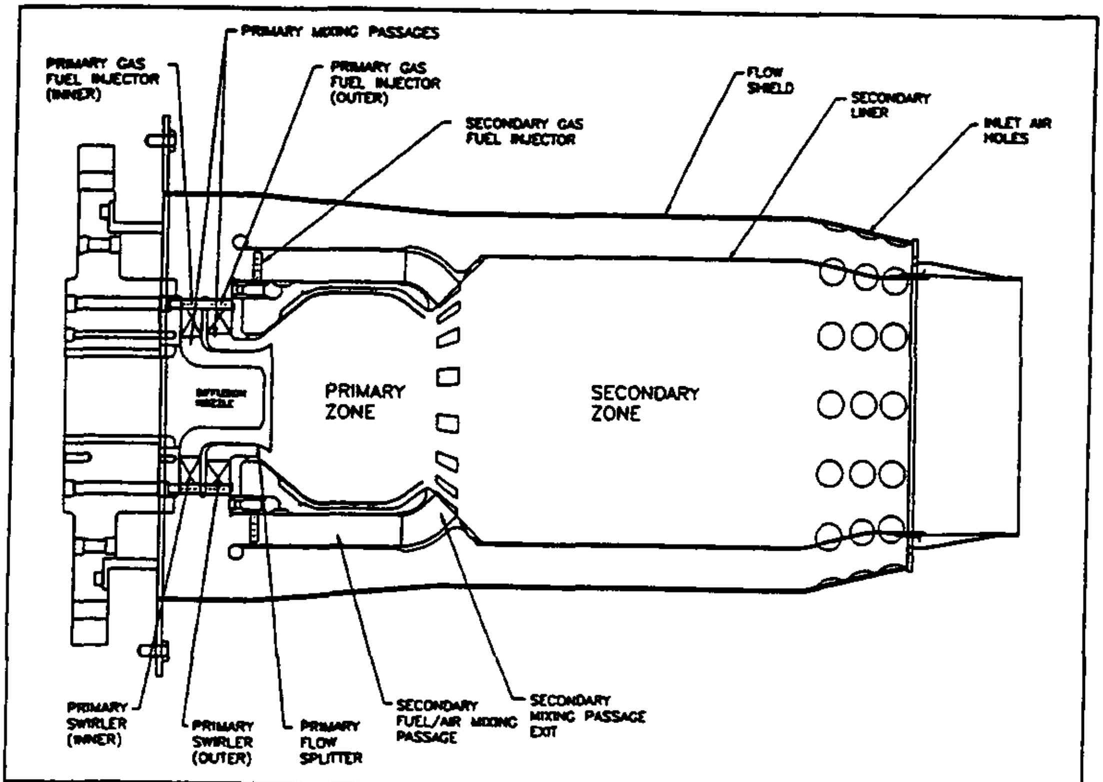

Figure 8 A Westinghouse Ultra Low NO_X Combustor

of two axial stages. The primary stage utilizes two counter rotating swirlers to premix fuel and air and stabilize the primary combustion zone. In the secondary stage a long premixing annulus provides the necessary length to achieve ideal mixing of the natural gas and air before injecting it into the secondary combustion zone. The combustor operates in a fully premixed mode for ultra low emissions, but can also operate in the diffusion mode with natural gas or oil supplied from a center nozzle.

This design has gone through an exhaustive development process including fully integrated cold flow air rig modeling, computational fluid dynamics (CFD), fired atmospheric, mid-pressure, and full pressure rig tests (Ref. 11). The Westinghouse

piloted ring design is being released for 1996 installation. This design has a heritage as the RB211 DLE combustor (Ref. 12).
This Westinghouse Piloted Ring combustor demonstrates single digit NOx levels over a wide range of operating temperatures and is focused on Ultra Low Emission high temperature units.

MultiSwirl Combustor
The MultiSwirl combustor utilizes annular, parallel stages to feed a well-premixed fuel and air mixture to a multitude of individual swirlers for final mixing and flame holding. A center pilot is used for starting and low load stabilization. Figure 9 schematically shows the combustor configuration. This combustor has demonstrated excellent stability and emissions over a very wide range of conditions.
This combustor, shown in Figure 10 installed in our high pressure rig, has passed the pre-operation testing. Installation in a 251B10 unit will occur late in 1995 with commercial operation scheduled for January 1996. The startup process is very similar to the Dry Low NOx design and is shown in figure 11. This design operates below the NOx production flame temperature and thus produces very low emission levels at elevated temperatures. The multitude of flames also provides a very high combustion efficiency resulting in extremely low secondary emissions.

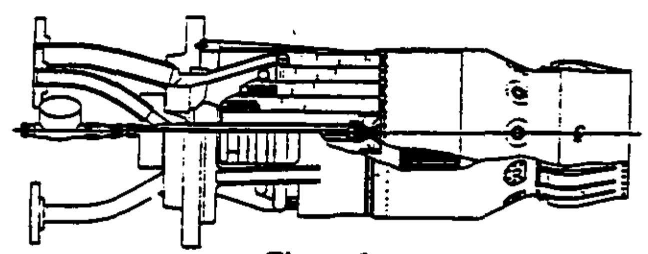

Figure 9

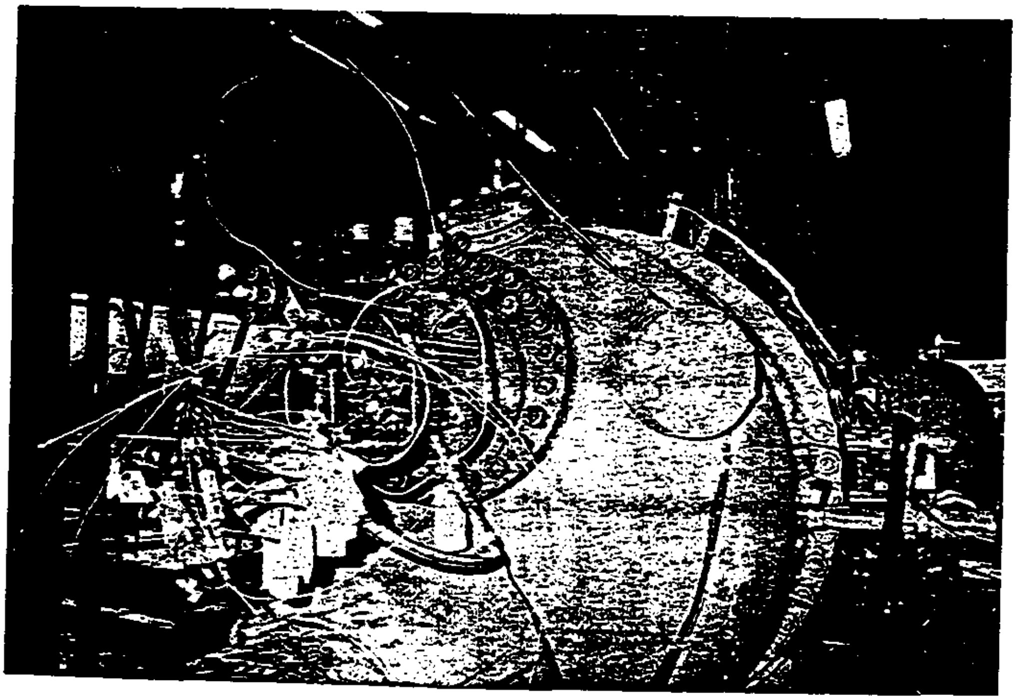

Figure 10

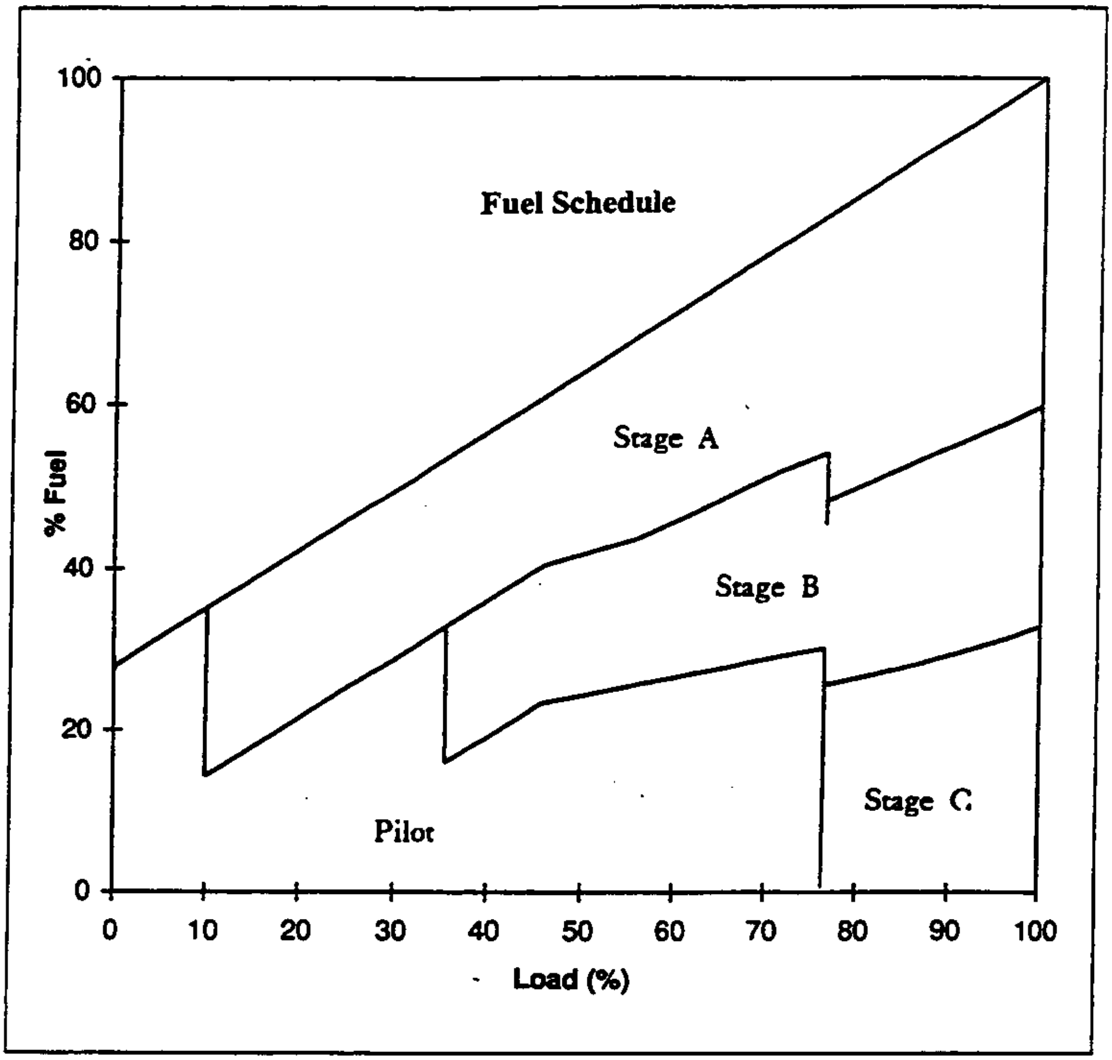

Figure 11

Environmentally Benign

The reduction of emissions in combustion turbines has been dramatic. In the period of about five years the acceptable rate of dry emissions has been reduced from 75ppm to common levels of 25ppm and less than 10ppm in certain site locations. The technologies that have driven this process have required a completely new approach to combustion design. Whereas traditional diffusion flame combustors were designed around an optimum stability point, low emission designs must operate near the lean limit. Advances in combustion technology and control system designs have made these dramatic improvements possible.

The next technology step is aimed at achieving environmentally benign designs. The goal of the Environmentally Benign program is to result in negligible impacts on local contaminant levels. Presently there are some sites where the volatile organic compound emissions exiting the turbine appear lower than the surrounding environment. To achieve this next profound step in NOx reduction requires that new technologies be incorporated. In our development activities, Westinghouse is focusing on several techniques that should increase the stability range. Included in these programs are projects such as active control of pressure oscillations, catalytic combustion, and optical mixing and flame characterization. These techniques, while still in the development stage, offer a great potential to extend the operating range of current designs. Our short term goals are to prove these concepts by adapting the techniques to our current designs with a longer term intent towards full optimization of these technologies.

Fuel Options

Westinghouse Combustion Turbines perform on a variety of fuels ranging from heavy to light liquids and gases. Fuel flexibility is a key attribute of combustion turbines and Westinghouse experience is both broad and deep. Engines in service include those running on crude oils, coal gas, butane, propane, naphtha, blast furnace gas, process gases laden with hydrogen, as well as the more conventional natural gas and Number 2 distillate fuel. The combustion technology involved in burning such as wide range of fuels is continuing to advance as new higher performance engines are introduced, and dry low NOx combustion matures into Ultra Low capabilities.

Summary

Westinghouse's comprehensive program to reduce emission levels from combustion turbines are proceeding toward the ultimate goal of environmentally benign designs. The combined goal to lower emissions with higher performance, higher temperature engines provide a particularly difficult challenge to the combustor designers. Westinghouse is utilizing state of the art tools that include both analytical and testing improvements that make these challenging goals achievable.

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A COMBINED CYCLE DESIGNED TO ACHIEVE GREATER THAN 60 PERCENT EFFICIENCY

M. S. BRIESCH, Plant Thermal Systems; R. L. BANNISTER, Emerging Technologies; I. S. DIAKUNCHAK, Combustion Turbine Engineering; D. J. HUBER, Plant Thermal Systems

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ABSTRACT

In cooperation with the U.S. Department of Energy's Morgantown Energy Technology Center, Westinghouse is working on Phase 2 of an 8-year Advanced Turbine Systems Program to develop the technologies required to provide a significant increase in natural gas-fired combined cycle power generation plant efficiency. In this paper, the technologies required to yield an energy conversion efficiency greater than the Advanced Turbine Systems Program target value of 60 percent are discussed. The goal of 60 percent efficiency is achievable through an improvement in operating process parameters for both the combustion turbine and steam turbine, raising the rotor inlet temperature to 2600°F (1427°C), incorporation of advanced cooling techniques in the combustion turbine expander, and utilization of other cycle enhancements obtainable through greater integration between the combustion turbine and steam turbine.

INTRODUCTION

The purpose of the Department of Energy's Advanced Turbine Systems (ATS) contract is to investigate technologies and innovative concepts applicable to natural gas-fired combined cycle power generation systems which will allow thermal efficiencies greater than 60 percent,1* while providing electricity at significantly lower cost than current combined cycle power plants and operating with much reduced environmental impact. Additionally, reliability-availability-maintainability (RAM) will be competitive or superior to current power generation systems. The ATS cycle fired with natural gas is to be commercially available by the year 2000. The final ATS design must also be able to be adapted to operate on coal-derived or biomass fuels for the post-2005 power generation market.

Westinghouse has developed a product line that mirrors the general gas turbine inlet temperature trend shown in Figure 1. To make gas turbines more competitive than steam turbine plants, it has been necessary to develop efficient cycles. Over the years, Westinghouse has performed many engineering studies to determine optimum cycles to minimize the cost of electricity. Some of the more promising cycles (intercooled, multiple shafts, reheat, steam injected, and water injected at various locations) have been studied in detail (Scalzo et al., 1994). In the final analysis, the simple cycle gas turbine combined with a steam bottoming cycle (a synergistic combination of the Brayton and Rankine cycles) was developed to increase overall cycle efficiency. Stephens (1952) and Baldwin et al. (1965), are two references which summarize earlier Westinghouse work on optimizing plant cycle efficiency.

Current large natural gas-fired combined cycle power generation systems are capable of net efficiency levels in the range of 54 percent. Within the ATS program, Westinghouse has been given the opportunity to re-evaluate cycle efficiencies using older, established concepts, such as intercooling and recuperation, and newer concepts, such as thermochemical recuperation (Little, Bannister, and Wiant, 1993). In addition, efficiency enhancements within the ATS selected cycle are to be evaluated to determine the best approaches to raising overall thermal plant efficiencies to greater than 60 percent while adhering to the other ATS program goals. The concepts considered in the Westinghouse analyses are required to be capable of demonstration within a three to four year time frame. From a baseline cycle definition, this paper reports on how different concepts will affect the overall plant thermal efficiency. Results given in the paper show that a plant efficiency of greater than 60 percent is achievable.

BASELINE CYCLE

In order to evaluate different technologies and concepts applic-

* All plant efficiency values in this paper are based on lower heating valve (LHV).

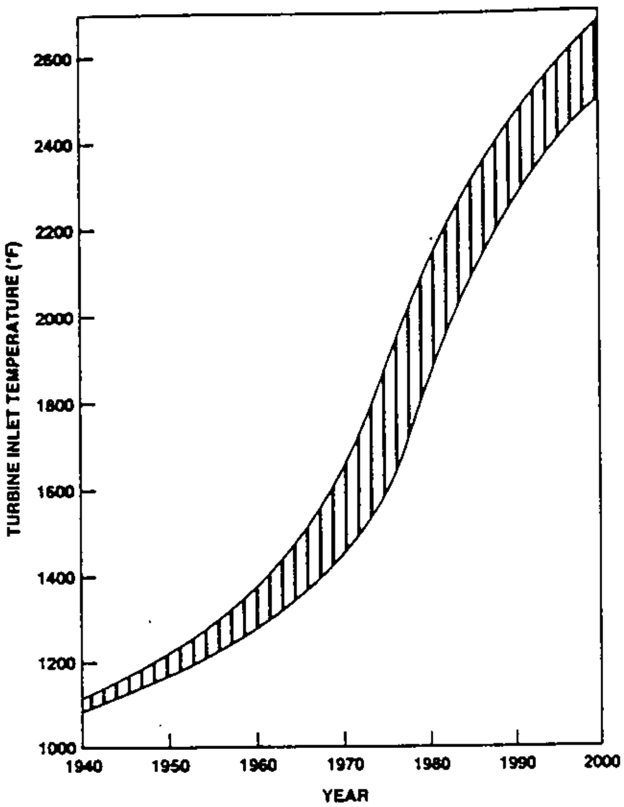

FIGURE 1. GAS TURBINE INLET TEMPERATURE TREND

able to combined cycle power generation systems, a baseline combined cycle configuration first had to be developed to provide a basis for comparison of all the cycle concepts and technologies to be considered. For this purpose, a conventionally configured combustion turbine coupled with a three-pressure level reheat steam cycle (Figure 2) was modeled to provide a baseline cycle. The combustion turbine rotor inlet temperature (RIT) was set at 2600°F (1427°C) to approximate near-term temperature capabilities (Bannister et al., 1994). Compressor pressure ratio was set at 18. High pressure steam conditions entering the steam turbine were specified at 1450 psi (10.2 MPa) and 1000F (538°C) and the hot reheat steam temperature was also set at 1000F (538°C). Note that this configuration utilizes turbine rotor cooling air heat to produce additional low pressure steam in the steam cycle via a heat exchanger located in the heat recovery steam generator (HRSG). Also, the natural gas fuel is preheated by feed water recirculation flow.

COMPONENT IMPROVEMENTS

Incorporation of several component improvements, available through recent technological developments and advanced design techniques, into the power generation system of Figure 2 results in significant efficiency gains.

The application of advanced compressor analysis and design tools results in increased efficiency compressors. Additionally, the application of brush seals in the compressor, as well as in the combustion turbine expander and the steam turbine, results in further plant efficiency improvements. At the compressor inlet, an advanced inlet design improves efficiency via reduced pressure loss.

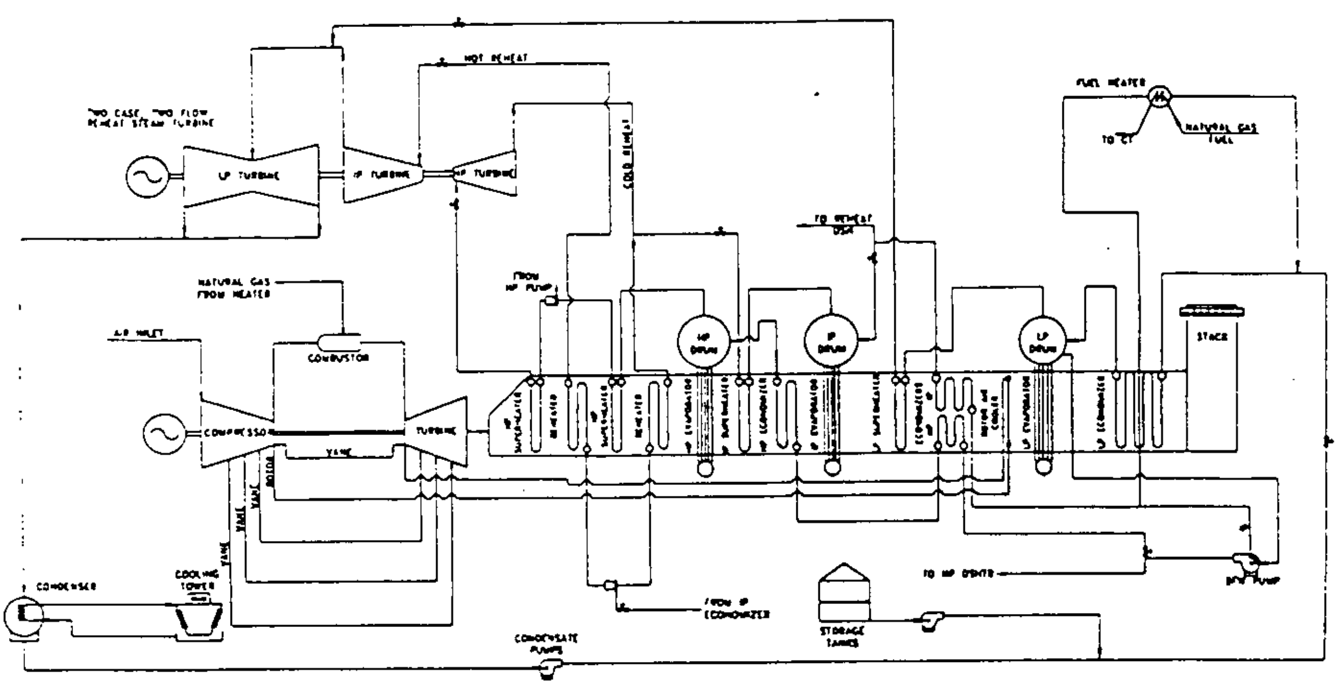

FIGURE 2. BASELINE COMBINED CYCLE

Westinghouse is currently engaged in full-scale rig tests of several diffuser / combustor configurations. These tests will allow the design of diffusers and combustors with reduced pressure loss, which will improve plant efficiency.

Several advanced turbine technologies may be applied to the combustion turbine expander. These include the application of single crystal and ceramic components to increase material operating temperatures and reduce turbine cooling requirements, active blade tip clearance control to minimize tip leakages and improve turbine efficiency, and the use of a three-dimensional (3-D) airfoil design philosophy. By utilizing 3-D analytical design tools, airfoils can be designed which are more efficient than those designed with 2-D design tools because they can accommodate more effectively the complex flows inside a turbomachine. The use of a 3-D airfoil design philosophy may also be applied to the steam turbine to enhance its efficiency.

There are two generators in the baseline configuration. One of these is the combustion turbine generator, while the other is the steam turbine generator. Current generator designs are capable of higher efficiency than those chosen for the baseline cycle. While the combustion turbine generator of the baseline configuration is of sufficient size to cost effectively apply this technology, the steam turbine generator is not. By utilizing a single shaft arrangement, however, the smaller steam turbine generator is eliminated and the remaining single generator may be designed at the higher efficiency.

When the component improvements listed above are all incorporated into the baseline cycle, the net plant thermal efficiency is increased by approximately 2 percentage points.

STEAM CYCLE ENHANCEMENTS

The basic reason for raising the steam pressure and temperature of the Rankine cycle is to improve the potential thermal efficiency (previous Westinghouse studies to optimize steam cycle configurations for subcritical and supercritical applications are summarized by Ernette and Silvestri, 1990 and Silvestri et al., 1992). The first cycle variations investigated within this study were modifications to the baseline cycle in which the steam cycle was enhanced. The results of these studies indicated that increasing either high pressure steam superheat temperature or reheat steam temperature by 50°F (28°C) results in an improvement in combined cycle thermal efficiency of 0.1 percentage point. Increasing high pressure steam pressure from 1450 psi (10.2 MPa) to 1800 psi (12.6 MPa) results in an increase in net plant thermal efficiency of 0.1 percentage point. A further increase in pressure to 2400 psi (16.8 MPa) yields only an additional 0.05 percentage point in thermal efficiency, while adding to the cost of the high pressure steam system. Also, since the steam turbine size is set by the exhaust energy of the combustion turbine, increasing steam pressure reduces the blade heights in the high pressure steam turbine. For 2400 psi (16.8 MPa) high pressure steam, the resulting blade heights are much smaller and less efficient than for the 1800 psi (12.6 MPa) steam. Therefore, the optimum steam cycle was determined to be at 1800 psi (12.6 MPa) with 1100°F (593°C) high pressure superheat steam and 1100°F (593°C) reheat steam (both 100°F (56°C) over the baseline temperature). This results in a 0.5 percentage point increase in net plant thermal efficiency and also in a slight increase in output due to the increased efficiency of the steam cycle. The steam temperatures were limited to 1100°F (593°C) for this study. The steam temperature of 1100°F (593°C) plus a reasonable steam superheater approach T determined the gas turbine exhaust temperature and this set the baseline cycle pressure ratio.

ROTOR AIR COOLER HEAT UTILIZATION

The Westinghouse 501F combustion turbine combined cycle provides two options for rotor air cooler heat utilization. The first option is an air-to-air cooler to cool the rotor cooling air after it exits the compressor and prior to its introduction into the rotor. The rotor air heat is rejected to the atmosphere via an air-to-air cooler. The other option is to cool the rotor air via an air-to-exhaust gas heat exchanger located in the heat recovery steam generator upstream of the low pressure evaporator, as was done in the baseline cycle. With this configuration, the rotor air cooler heat is recovered by the steam cycle, which produces low pressure steam with the heat. This results in higher plant efficiency than that of the air-to-air cooler method since the rotor air cooler heat is recovered by the low pressure steam system.

Another concept involves removing the HRSG rotor air cooler used in the baseline configuration and installing a rotor air cooler which exchanges heat with the incoming natural gas fuel (after the fuel has been preheated by feed water recirculation flow). This returns the rotor air heat back to the combustion turbine, which then requires less fuel to achieve the desired rotor inlet temperature. Therefore, the rotor air heat is recovered at the combustion turbine efficiency (typically about 40 percent), which is much higher than the low pressure steam system efficiency. This is, therefore, a much more effective recovery of the heat than is obtained via low pressure steam production, and results in an increase in net plant thermal efficiency of 0.4 percentage point over the baseline configuration.

CLOSED-LOOP STEAM COOLING

Most current technology gas turbine engines utilize air to cool the turbine vanes and rotors. This allows the turbine inlet temperature to be increased beyond the temperature at which the turbine material can be used without cooling, thus increasing the cycle efficiency and power output. However, the cooling air itself is a detriment to cycle efficiency in four ways. First, it is ejected from the turbine airfoils causing a disruption in the surrounding flow field. This increases the airfoils' irreversible pressure losses and results in a reduction in turbine efficiency. Secondly, since the cooling air is ejected from the airfoil into the gas path, the resulting mixing of the cooling air into the gas path results in irreversible pressure losses due to the non-ideal mixing of the streams, which have very different velocity vectors. The third loss mechanism is caused by the reduction in gas path temperature that accompanies the mixing of the cooling air into the gas path. This reduction in temperature reduces the work output of the turbine and, therefore, compromises cycle efficiency. Finally, the turbine cooling air must be pumped to pressures significantly higher than that of the gas path pressure at the location it is injected. This is done to assure that the cooling flow rate will be sufficient during certain operating conditions where the ratio of coolant pressure to gas path pressure drops below its design level. While some of this pressure is recovered by the turbine, there are internal losses as the cooling air passes from the compressor to the turbine gas path. The additional pumping work required to raise the cooling air to the required pressure is the associated loss.

The effect of cooling air on cycle efficiency is shown in Figure 3. This figure shows the potential increase in combined cycle thermal efficiency for fractional reductions in cooling or leakage flows. In this figure, the lines labeled 'Fixed P/P' show the effect for fixed compressor pressure ratio (i.e., the turbine expander is increased in size to handle the additional flow resulting from the reduction in cooling or leakage) and the lines labeled 'Fixed Expander' show the effect for fixed expander size (the compressor pressure ratio is allowed to rise so that the additional flow caused by the reduction in cooling or leakage can be accommodated by a turbine of the original size). Note that, although turbine leakage generally carries a larger efficiency penalty than turbine cooling per given amount of flow, the fact that the amount of leakage flow is far less than the amount of cooling flow results in far less efficiency benefit from reducing turbine leakage flows by a given fractional amount compared to reducing turbine cooling flows by the same fraction of their baseline value.

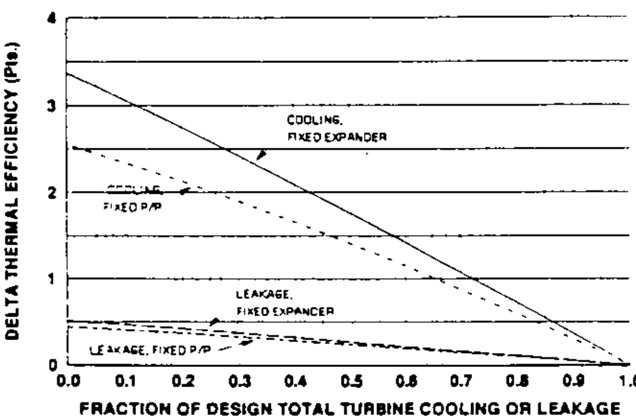

FIGURE 3. TURBINE COOLING AND LEAKAGE, EFFECT ON COMBINED CYCLE THERMAL EFFICIENCY

By using closed-loop steam cooling, the loss mechanisms described above can be largely eliminated, while still maintaining turbine material temperatures at an acceptable level. In combined cycles, the steam used for cooling the combustion turbine hot parts will be taken from the steam bottoming cycle. This steam is then returned to the bottoming cycle after it has absorbed heat in the closed-loop steam cooling system. For an advanced bottoming steam cycle, closed-loop steam cooling would route cold reheat steam from the exit of the high pressure steam turbine to the combustion turbine vane casing and rotor. The steam is passed through passageways within the vane and rotor assemblies and through the vanes and rotors themselves, then collected and sent back to the steam cycle intermediate pressure steam turbine as hot reheat steam. This approach to turbine cooling relies solely on convective heat transfer. Since no steam or cooling fluid is ejected from the airfoils, aside from a small amount of steam leakage through the rotor seals, there is very little influence of the cooling steam on the airfoil flow fields, and hence minimal mixing losses. Also, the reduction in gas path temperature is minimized, since the convective heat flux across the airfoils is relatively small. Typically, first vane cooling air mixing reduces the gas path temperature approximately 100°F to 150°F (56°C to 83°C). For closed-loop steam cooling however, the reduction in gas path temperature is only about 10°F to 15°F (6°C to 8°C), or one tenth of the reduction of conventional cooling techniques. Application of closed-loop steam cooling to the baseline configuration yields a 2 percentage point increase in combined cycle efficiency.

INCREASED TURBINE INLET TEMPERATURE

Since thermal efficiency increases with increasing turbine inlet temperature, the potential benefits of increased turbine inlet temperature was investigated. The turbine inlet temperature for the baseline cycle was increased 300°F (167°C) to 2900°F (1593°C). This resulted in a cycle output increase of 10 percent, and combined cycle thermal efficiency increase of slightly more than 1 percentage point. The reason that the performance increase was relatively small for such a large increase in turbine rotor inlet temperature is that, since the (air) cooling technology remained constant as the temperature was increased, large amounts of additional turbine cooling air were required to maintain turbine material operating temperatures at an acceptable level. This increase in cooling flow decreased cycle efficiency by the three mechanisms described earlier, and this significantly offset the benefit of increasing the turbine rotor inlet temperature. Therefore, since efficiency and output would have been increased much more if turbine cooling was held at the same level as in the baseline cycle, increased turbine operating temperature must be accompanied by corresponding advancements in turbine cooling. However, even if cooling technology advancements were available to allow operation at much higher rotor inlet temperatures, the formation of NOx at these higher temperatures would result in unacceptable emissions characteristics.

INCREASED COMPRESSOR PRESSURE RATIO

Commercial aircraft gas turbine engines are designed with high overall pressure ratio. This is done to maximize the simple cycle efficiency. For the ideal Brayton gas turbine cycle, the cycle efficiency is a function solely of the cycle pressure ratio and increases with cycle pressure ratio (the ratio of specific heats of the working fluid also effects the cycle efficiency, but for air-breathing cycles there is little that can be done to change this property). In fact, for this ideal cycle, the maximum efficiency occurs at the point where the compressor exit temperature is very close to the combustor exit temperature. Of course, the cycle output for this case is very close to zero.

In real cycles, the effect of non-ideal components causes the peak efficiency pressure ratio to decrease significantly from that of the Brayton cycle. Figure 4 shows the effect of compressor pressure ratio on simple cycle performance for a family of engines based on the combustion turbine of the baseline configuration. Also included in Figure 4 are the corresponding steam cycle and combined cycle efficiencies. Note that the simple cycle efficiency curve is relatively flat above a pressure ratio of approximately 40. This indicates that it is nearing the peak simple cycle efficiency. The steam cycle efficiency is seen to decrease with increasing combustion turbine pressure ratio. This is due to the reduction in combustion turbine exhaust temperature, which in turn reduces the maximum steam temperature and pressure and the steam's availability, and results in lower steam cycle efficiency. The effect of all of this on combined cycle efficiency is that it peaks around a pressure ratio of about 20, but remains approximately constant for a relatively large increase in compressor pressure ratio.

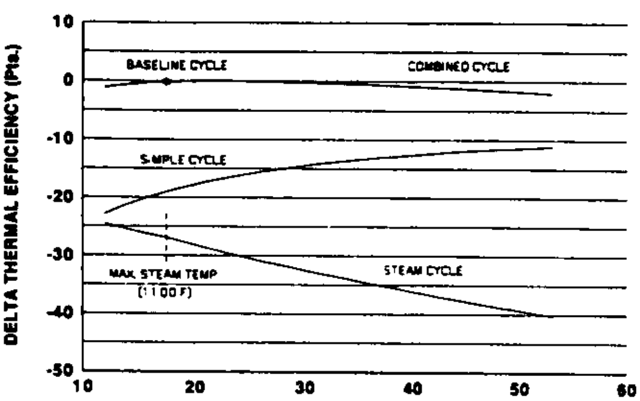

FIGURE 4. COMPRESSOR PRESSURE RATIO, EFFECT ON THERMAL EFFICIENCIES

Reducing the compressor pressure ratio below the baseline value of 18 results in a significant decrease in combined cycle efficiency. This is due to the fact that, since the maximum steam temperature considered for this study was 1100°F (593°C), any decrease in pressure ratio below the baseline value of 18, where the maximum steam temperatures are reached, will increase the turbine exhaust temperature while maintaining the steam temperatures at 1100°F (593°C). This results in a much smaller increase in steam cycle efficiency than that obtainable by allowing the steam temperatures to rise with the turbine exhaust temperature. Figure 4 shows the significant reduction in the slope of the steam cycle efficiency line at this point, which causes the combined cycle efficiency decrease.

The maximum specific work output of the Brayton cycle occurs at a significantly lower value of pressure ratio than that for the peak efficiency. Figure 5, which includes intercooled cycle characteristics to be discussed later, shows the specific output of the cycles in Figure 4 (solid lines). The simple cycle peak specific output occurs at a pressure ratio of approximately 18. Also, note that the steam cycle specific output is reduced as pressure ratio is increased, since there is less exhaust energy available to the steam cycle. The combined cycle specific output, which is merely the sum of the simple cycle and steam cycle specific outputs, decreases for increasing pressure ratio across the entire range shown.

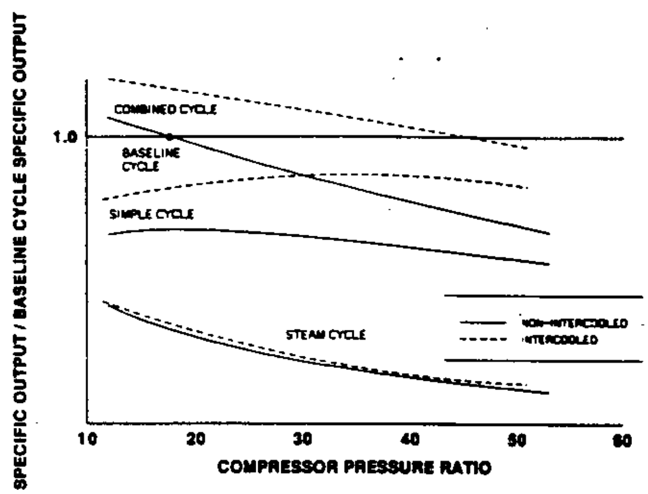

FIGURE 5. COMPRESSOR INTERCOOLING, EFFECT ON SPECIFIC OUTPUT

It, therefore, is optimum to select the lowest pressure ratio at which the peak combined cycle efficiency level is obtained. In this way both efficiency and output are maximized, and the cost of electricity is minimized (the baseline cycle is designed in this fashion).

COMPRESSOR INTERCOOLING

The most typical arrangement for compressor intercooling involves removing the compressor air flow halfway through the compressor temperature rise, sending it through an air-to-water heat exchanger, and returning it to the compressor for further compression to combustor inlet pressure. The heat removed from the compressor air flow by the intercooler is rejected to the atmosphere, because, at the pressure ratios considered in this study, the heat is of too low quality to be of use to the cycle.

Another intercooling concept is to spray water droplets into the compressor. As the air is compressed and increases in temperature, the water evaporates and absorbs heat. This results in a continuous cooling of the compressor. Note that for this concept the heat absorbed by the water is also rejected to the atmosphere, since this water is never condensed by the cycle but instead exhausted with the stack gases as low pressure steam.

Compressor intercooling reduces the compressor work, because it compresses the gas at a lower average temperature. Since the combustion and steam turbines produce approximately the same output as in the non-intercooled case, the overall cycle output is increased. However, since the compressor exit temperature is lowered, the amount of fuel that must be added to reach a given turbine inlet temperature is greater than that for the non-intercooled case. The ratio of the amount of compressor work saved to the amount of extra fuel energy added is about equal to the simple cycle efficiency. It can therefore be said that intercooling adds output at approximately the simple cycle efficiency. Since combined cycle efficiencies are significantly greater than simple cycle efficiencies it would be expected that the additional output at simple cycle efficiency would reduce the combined cycle net plant efficiency for the intercooled case. Figure 6 verifies this expectation and shows that this trend is the same for a wide range of cycle pressure ratios. Note that the simple cycle shows almost no change in efficiency for intercooling, which is expected since output is added at approximately the simple cycle efficiency.

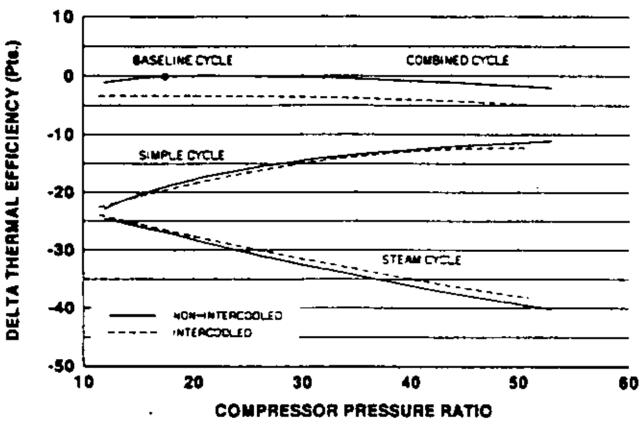

FIGURE 6. COMPRESSOR INTERCOOLING, EFFECT ON THERMAL EFFICIENCIES

Figure 5 shows the effect of intercooling on specific output. Since the compressor work requirement is reduced while the gas and steam turbine work outputs remain approximately the same as in the non-intercooled case, the net power output of both the simple and combined cycles is increased. Also, since the turbine exhaust temperature is increased slightly due to the exhaust gas composition effects, the steam cycle specific output is also increased slightly.

RECUPERATION

In recuperative cycles, turbine exhaust heat is recovered and returned to the combustion turbine combustor, usually via a heat exchange between the turbine exhaust gases and the compressor exit air flow. The discharge from the compressor exit is piped to an exhaust gas-to-air heat exchanger located aft of the combustion turbine. It is then heated by the turbine exhaust and returned to the combustor. Since the resulting combustor air inlet temperature is increased above that of the non-recuperated cycle, less fuel is required to heat the air to a given turbine inlet temperature. Because the turbine work and the compressor work are approximately the same as in the non-recuperated cycle, the decrease in fuel flow results in an increase in thermal efficiency. This is especially true for the simple cycle, since the heat recovered by recuperation is rejected to the atmosphere in the non-recuperative case. For combined cycles the efficiency is also increased, because the combustion turbine recovers the recuperated heat at the simple cycle efficiency, which is larger than the 30 to 35 percent thermal efficiency of the bottoming steam cycle, which recovers this heat in the non-recuperated case.

Installation of a recuperation system on the baseline configuration results in an increase in thermal efficiency of 1 percentage point. The steam cycle in this recuperated cycle has a lower efficiency than the steam cycle in the baseline configuration, because the recuperator exit temperature is significantly lower than the turbine exhaust temperature. However, the effect of reduced steam cycle efficiency is smaller than the effect of recovering the recuperated heat at the combustion turbine efficiency.

The thermal efficiency of the recuperated cycle can be increased further. Since the combustor inlet flow is smaller than the turbine exhaust flow (due to the removal of the turbine cooling air prior to combustion) and has a higher specific heat (due to the combustion products of the fuel), the heat capacity of the turbine exhaust flow is somewhat higher than that for the burner inlet flow. This means that the recuperated cycle described above does not fully utilize the quality of heat available in the turbine exhaust. By placing a steam superheater in parallel with the recuperator, the remainder of the available turbine exhaust heat can be recovered at its maximum quality. The maximum steam temperature can then be raised to that of the baseline cycle (1000°F (538C°), and the cycle efficiency is increased by an additional 0.1 percentage point.

Since recuperative cycles return exhaust energy to the combustion turbine, less energy is available to the steam cycle, and the resulting steam turbine output is lower than that of the baseline configuration. However, the combustion turbine output is approximately the same as in the baseline cycle (minus losses in the recuperation system). This means that recuperative cycles carry a significant output penalty, with this penalty being proportional to the amount of recuperation performed.

INTERCOOLING WITH RECUPERATION

From a simple cycle standpoint, the combination of intercooling with recuperation eliminates the problem of the reduced combustor inlet temperature associated with intercooled cycles. The simple cycle then gets the benefit of the reduced compressor work and, at all but high pressure ratios, actually has a higher burner inlet temperature than the corresponding non-intercooled, non-recuperated cycle. This results in a dramatic increase in the simple cycle efficiency.

However, the bottoming cycle receives even less energy than in the recuperated cycle, since the recuperator removes much more heat from the turbine exhaust than in the recuperated, non-intercooled cycle. The additional heat removed corresponds to the heat rejected to the atmosphere by the intercooler. This means that we have merely displaced the lost energy of the intercooler by taking energy from the bottoming cycle. This is in addition to the energy already removed from the bottoming cycle in the non-intercooled, recuperative cycle. The result is a very low recuperator exit temperature, which in turn translates into a low generated steam pressure (low availability) and a low efficiency steam bottoming cycle. Thus, while simple cycle efficiency can be increased to over 50 percent, the combined cycle efficiency is reduced for the entire useful range of compressor pressure ratios. For conventional air-to-water heat exchanger intercooling of the compressor, the combined cycle efficiency is reduced approximately 1.9 percentage points at the baseline cycle pressure ratio. For continuously cooled compressors utilizing water droplet spraying into the compressor, the combined cycle efficiency is reduced by 0.4 percentage point.

Since the continuously cooled compressor with recuperation is not far below the baseline level of combined cycle thermal efficiency, it is worth investigating further optimization of the steam cycle in this case. As mentioned earlier, steam may be superheated in parallel with the recuperator, yielding more efficient recovery of the heat available to the steam cycle. Utilizing this approach, along with a two-pressure level steam cycle, results in an increase in net plant efficiency of 0.8 percentage point over the baseline efficiency level.

Another approach to optimizing the intercooled recuperative cycle is to place a saturator between the compressor exit and the recuperator entrance, as illustrated in the combined cycle shown in Figure 7. This saturator, also called an aftercooler, evaporates water into the compressor exit flow, resulting in a lower temperature, higher mass flow entering the recuperator. While this may be seen as a way to better balance the heat capacities of the hot and cold streams in the recuperator and thereby increase the amount of heat recuperated by the combustion turbine (in addition to the fact that the air flow is at lower temperature), it is important to realize that all of this additional recuperation is accomplished by the evaporation of water in the saturator. Since this water is never condensed by the cycle but instead is rejected through the stack as low pressure steam, the additional amount of energy recuperated is not recovered anywhere in the cycle. Furthermore, the recuperator exit temperature is reduced even further than in the intercooled recuperative cycle, resulting in an even lower efficiency steam cycle. Finally, note that, since the heat capacity of the cold side recuperator flow now closely matches that of the hot side flow, parallel steam superheat cannot be utilized to increase the steam cycle efficiency. The application of this concept to the baseline cycle results in a decrease in thermal efficiency of 2 percentage points.

REHEAT COMBUSTION TURBINE

Reheat combustion turbines utilize a sequential combustion process in which the air is compressed, combusted, expanded in a turbine to some pressure significantly greater than ambient, combusted again in a second combustor, and finally expanded by a second turbine to near ambient pressure. For a fixed turbine rotor inlet temperature limit, the simple cycle efficiency is increased for a reheat combustion turbine compared to a non-reheat cycle operating at a pressure ratio corresponding to the second combustor's operating pressure. This is because the reheat cycle per-

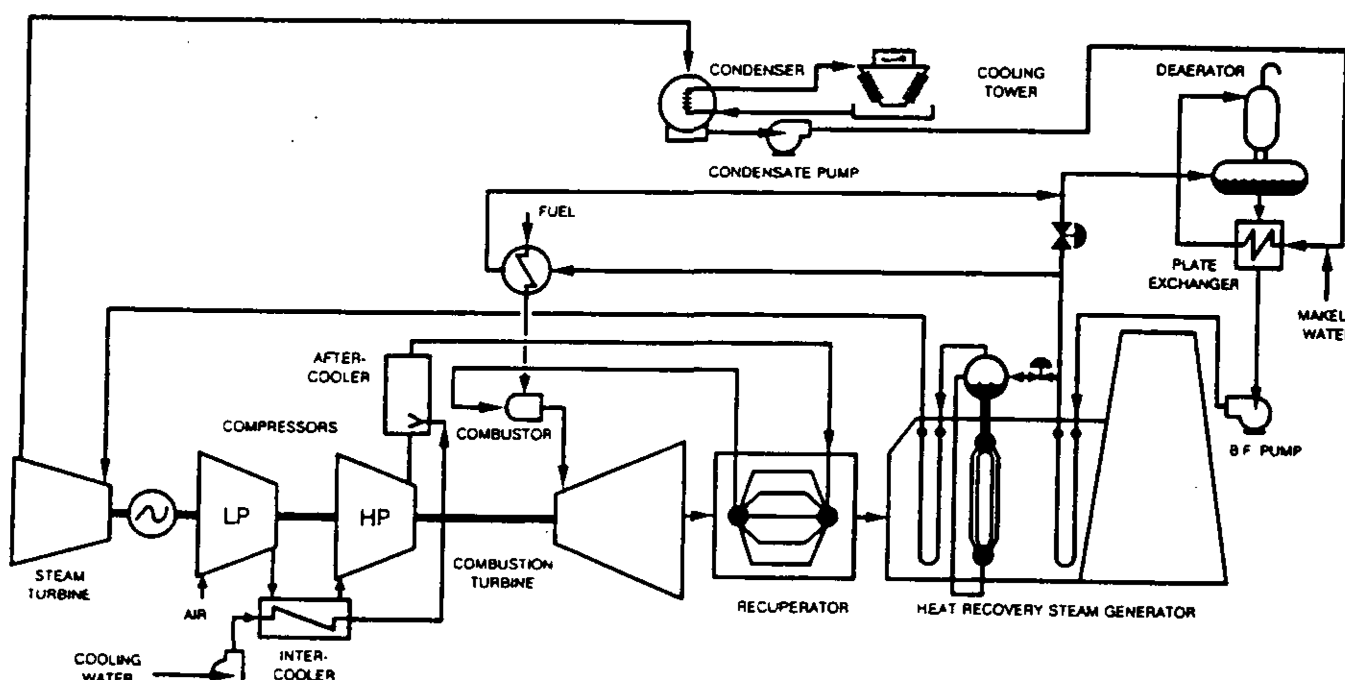

FIGURE 7. INTERCOOLED, AFTERCOOLED (EVAPORATIVE), RECUPERATIVE COMBINED CYCLE

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forms some of its combustion and expansion at a higher pressure ratio, which increases simple cycle efficiency (see section on increased compressor pressure ratio). From a purely thermodynamic standpoint, the average temperature at which heat is added is raised, thus raising the Carnot efficiency of the cycle. For combined cycles the turbine exhaust temperature can be controlled by the selection of second turbine inlet temperature and expansion ratio. This in turn allows control over the efficiency of the steam bottoming cycle.

Another beneficial feature of reheat combustion turbine cycles is that they exhibit higher output for a fixed compressor flow rate and turbine inlet temperature than non-reheat cycles. This is due to the fact that they burn sequentially with an expansion between the combustors. This allows for the addition of more fuel (in the second combustor) without violation of the turbine rotor inlet temperature limit. Also, since this results in a higher fuel-to-air ratio than in non-reheat cycles, they burn closer to stoichiometry and exhaust lower concentrations of excess oxygen.

Applying combustion turbine reheat to the baseline cycle, with the second combustor operating at the exit pressure and temperature equal to those of the baseline cycle, results in nearly identical turbine exhaust temperatures, and therefore the steam cycle of the combustion turbine reheat case is not compromised and is identical to the steam cycle of the baseline cycle. However, the compressor pressure ratio has been increased to 36. This necessitates the addition of 6 compressor stages, and an additional combustor and turbine stage located upstream of the second combustor. The simple cycle efficiency is increased nearly 2 percentage points from the baseline simple cycle level. Since steam cycle efficiency remains at the level of that in the baseline cycle, the combined cycle efficiency is increased 1.3 percentage points.

An investigation was made into the application of intercooling and recuperation to reheat combustion turbine cycles. The results showed the same reduced efficiency as for non-reheat combined cycles.

THERMOCHEMICAL RECUPERATION

In a thermochemical recuperative (TCR) power plant, a portion of the stack exhaust (flue) gas is removed from the stack, compressed, mixed with natural gas fuel, heated with exhaust heat from the combustion turbine, and mixed with the air compressor exhaust as it enters the combustor. As the mixture of natural gas and flue gas is heated by the combustion turbine exhaust, an endothermic reaction occurs between the methane and the carbon dioxide and water in the flue gas. This reaction occurs in the presence of a nickel-based catalyst, and results in the production of hydrogen and carbon monoxide. For complete conversion of the methane, the effective fuel heating value is increased approximately 30 percent. Therefore, the natural gas / flue gas mixture absorbs heat thermally (as it is heated) and chemically (via the endothermic reaction), resulting in a larger potential recuperation of exhaust energy than could be obtained by conventional recuperation, which recovers energy by heat alone. In fact, with full conversion of the natural gas fuel to hydrogen and carbon monoxide, up to twice the energy recuperated by the standard recuperative cycle may be recovered.

The endothermic reaction described above is accelerated for low excess oxygen in the reacting mixture, low pressures, and high mass ratios of recirculated flue gases to methane. Therefore, in order to take full advantage of this concept, the engine is controlled by running the combustor at near stoichiometric fuel-to-air ratios (10 percent excess air at the combustor exit) and using flue gas recirculation to quench the combustion products down to the desired turbine inlet temperature. This maximizes flue gas recirculation and minimizes excess oxygen in the flue gas. For typical cycles utilizing this control philosophy, the resulting recirculation rate of flue gas is over 50 percent of turbine flow. This means that both the air compressor flow rate and stack exhaust flow rate are less than half that of conventional cycles with the same turbine size.

Another advantage of the thermochemical recuperation / flue gas recirculation (TCR/FGR) concept is that, because the fuel has a low adiabatic flame temperature and operates with very low levels of excess oxygen in the exhaust, the resulting emissions of NOx and CO are much lower than those for conventional design power plants.

When applied to the baseline configuration, thermochemical

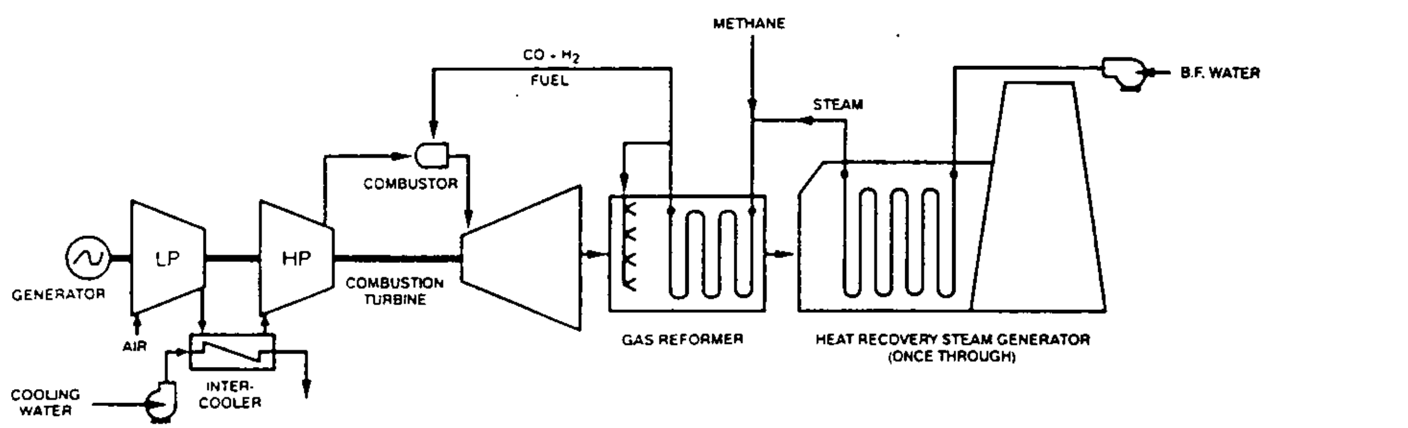

FIGURE 8. THERMOCHEMICAL RECUPERATION CYCLE

recuperation yields a combined cycle thermal efficiency over 2 percentage points greater than that of the baseline cycle for several different TCR/FGR configurations. Additionally, there are many possible cycle configurations, other than those described above, which can utilize thermochemical recuperation. For example, steam reforming may be used instead of flue gas reforming, as illustrated in Figure 8.

STEAM INJECTION

To apply steam injection to the baseline configuration, both the high pressure and low pressure steam systems are eliminated, leaving only the intermediate pressure steam system. The intermediate pressure superheated steam is then routed into the combustion turbine combustor inlet. Therefore, all steam turbines are eliminated, along with the condenser and the cooling tower. This results in a cycle which is significantly less expensive to build than the baseline cycle and it provides a means to reduce NOx emissions via the large amount of steam injection. However, the cost associated with demineralization of the large cycle make up flow must also be considered.

For a given turbine flow area, the compressor flow size must be reduced significantly, due to the addition of a large amount of steam, which now must pass through the turbine in addition to the compressor exit air flow. Since the steam is generated with exhaust heat, which is not used in a simple combustion turbine cycle, this concept provides a significant portion of the combustor inlet gas flow with very little work requirement (a small amount of energy is needed to run the feed water pumps which pump the water to the intermediate steam pressure). Therefore, from a simple cycle standpoint, the work of compression is much reduced due to the fact that some of the combustor inlet gas flow is compressed as a liquid. Also, the turbine output is increased significantly, since the average specific heat of the working fluid is increased considerably by the presence of the steam. The simple cycle efficiency and output are therefore increased by steam injection.

Compared to combined cycles, however, elimination of the high pressure steam system results in generating more intermediate pressure steam, but this steam is at significantly lower availability due to its lower pressure. Also, since the low pressure steam system has been eliminated as well, the exhaust stack gas temperature is much higher, as is the associated heat loss. Finally, since the steam injected into the gas turbine is effectively throttled to its partial pressure upon mixing with the compressor exit air without doing any work, and is only expanded to its partial pressure in the exhaust stack (which is significantly higher than typical low pressure steam turbine exit pressure), the resulting steam expansion ratio is much smaller than that of the conventional steam turbine cycle. These losses result in a reduction in combined cycle efficiency in the range of 5 to 8 percentage points. Net plant output, while much higher than the baseline configuration simple cycle output, is less than the combined cycle output.

CONCLUSIONS

Through component improvements and cycle innovations, the goal of achieving a 60 percent efficiency natural gas-fired power plant for industrial application can be met within an 8-year time frame. The selected cycle and efficiency enhancements will be cost competitive, as a result of not only high thermal efficiency, but also low plant construction and maintenance costs and high plant power output. Additionally, the cycle will demonstrate reduced environmental impact through innovations to reduce emission levels.

This paper has outlined a number of technologies and cycle innovations which will allow Westinghouse to design and build an ATS plant for power generation which meets or exceeds the goals of the DOE-sponsored program. This advanced power plant will demonstrate the continued evolution of Westinghouse large frame gas turbines as depicted in Figure 9. Starting with the 501A engine through the ATS engine, gas turbine output versus combined cycle efficiency is plotted. The data shows the step change that will occur with the successful completion of the ATS Program.

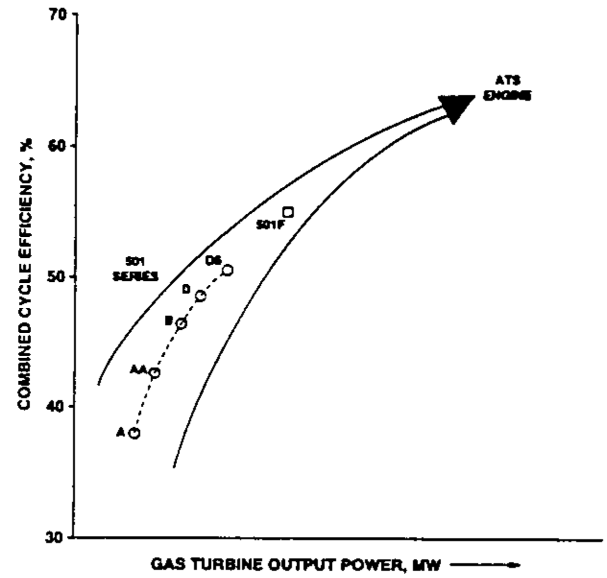

FIGURE 9. EVOLUTION OF LARGE WESTINGHOUSE GAS TURBINES

ACKNOWLEDGMENTS

The innovative cycle and gas turbine component concepts applicable to raising the efficiency of natural gas-fired combined cycles to greater than 60 percent were studied under DOE Contract No. DE-AC21-93MC30247. The program is administered through the Morgantown Energy Technology Center under the guidance of METC's Program Manager, Mr. Donald W. Geiling.

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DEVELOPMENT REQUIREMENTS FOR AN ADVANCED GAS TURBINE SYSTEM

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ABSTRACT

In cooperation with U.S. Department of Energy's Morgantown Energy Technology Center, a Westinghouse led team is working on the second part of an 8-year, Advanced Turbine Systems Program to develop the technology required to provide a significant increase in natural gas-fired combined cycle power generation plant efficiency.

This paper reports on the Westinghouse program to develop an innovative natural gas-fired advanced turbine cycle which, in combination with increased firing temperature, use of advanced materials, increased component efficiencies and reduced cooling air usage, has the potential of achieving a lower heating value plant efficiency in excess of 60%.

INTRODUCTION

Today's gas turbine systems feature high fuel-to-electricity efficiencies. Efficiencies, on a lower heating value (LHV) basis, for large natural-gas-fired combined-cycle systems for the utility market have been demonstrated at 54 to 55%. Even though manufacturers will make improvements in the 1990s, efficiency levels will reach a plateau. Cycle innovations, plus gas turbine design changes will achieve LHV efficiencies in the 60% range for natural gas-fired utility machines.

Advanced Turbine Systems (ATS) using natural gas or coal-derived fuels are candidates for the post-2005 power-generation market. An ATS is a highly efficient, environmentally-superior, and cost-competitive gas turbine system for base-load application in the utility, independent power producer, and industrial markets. The final design must be fuel flexible in that it will operate on natural gas, but also be capable of being adapted to operate on coal, coal-derived, or biomass fuels. Reliability-availability-maintainability (RAM) is to be equivalent to current turbine systems and water consumption is to be minimized to levels consistent with cost and efficiency goals.

Westinghouse, a pioneer of gas turbine development, has developed a product line that mirrors the general gas turbine inlet temperature trend shown in Figure 1. Combined cycle efficiency improvements have followed the general advance in gas turbine technology reflected in this inlet temperature trend.

For example, in 1970 a Westinghouse supercharged, evaporative cooled, Model 301 (1450°F [788°C] turbine inlet temperature) gas turbine exhausting into a heavy fired, reheat boiler achieved an annual average efficiency (LHV) of 39.6%. This West Texas Utilities, San Angelo Power Station reportedly held the distinction of achieving the highest operating efficiency in the U.S. for a number of years (Stephens et al., 1990).

Today, with a gas turbine burner outlet temperature (BOT) of about 2460°F (1350°C) [2300°F (1260°C) rotor inlet temperature (RIT)], a Westinghouse 501F based combined cycle is in the range of 54 - 55% (LHV) plant efficiency, ISO with: 1) dry low NOx; 2) three -pressure level, reheat, heat recovery steam generator (HRSG); 3) rotor cooling; and, 4) fuel gas heating.

The bottoming cycle has been fine tuned to convert as much of the turbine's exhaust heat into electricity as possible. Additionally, the gas turbine incorporates all the material, aerodynamic and cooling technology gains of the last 25 years.

As suggested by the Department of Energy (DOE), the target of 60% combined cycle efficiency by the year 2000 is shown to be a challenging target in Figure 2, where cycle efficiency is plotted against the turbine inlet temperature trend of Figure 1.

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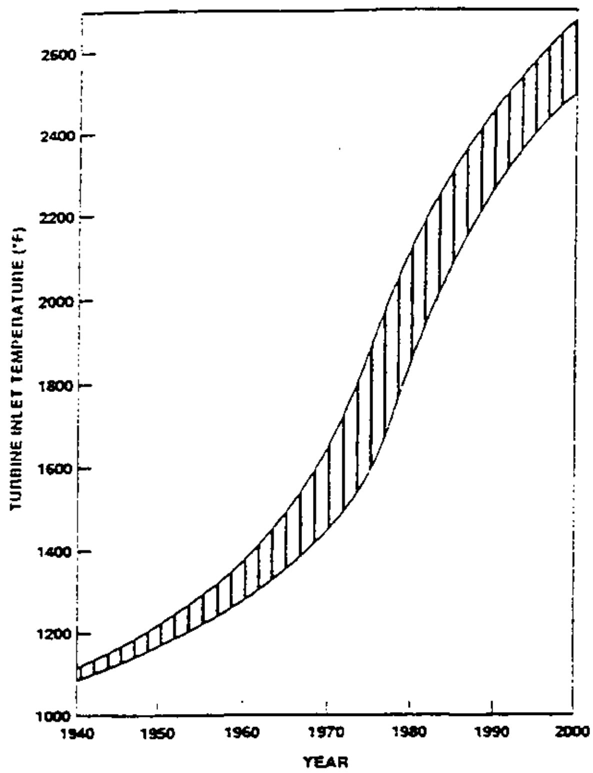

FIGURE 1.
GAS TURBINE INLET TEMPERATURE TREND

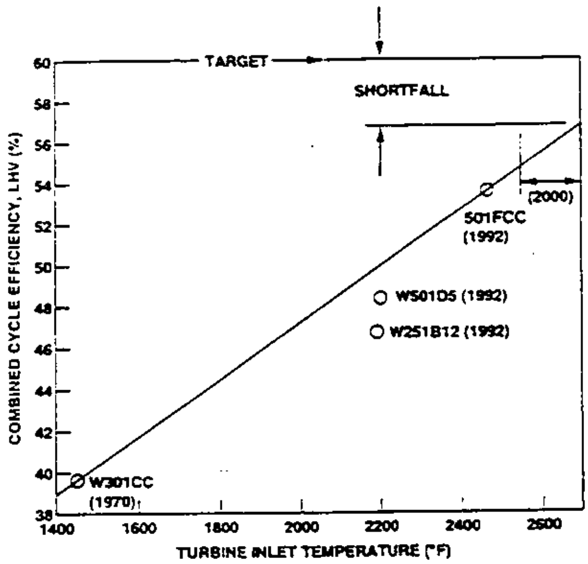

FIGURE 2.
COMBINED CYCLE EFFICIENCY TREND

PROPOSED REFERENCE SYSTEM

In the first phase of the ATS program, the effect on efficiency of reconfiguring the gas turbine based combined cycle was studied, to see if a step gain in efficiency was possible. As reported in a previous paper (Little, Bannister, Wiant, 1993), it was found that a significant increase in combined cycle efficiency could indeed be expected by reconfiguration into an intercooled, recuperative combined cycle, with closed loop steam cooling as shown in Figure 3. All rotating components will be directly coupled to form a single shaft running at 3600 RPM, and the generator rotor will be double ended so that the steam turbine and gas turbine can drive from opposite ends.

Intercooling could take the form of: 1) surface heat transfer where heat is extracted from air leaving a compressor section and transferred to cooling water; 2) evaporative intercooling where the temperature of air leaving a compressor section is reduced by evaporation of water into the air stream; 3) either one intercooler [low pressure (LP) and high pressure (HP) compressors], or two intercoolers [LP, intermediate pressure (IP), HP compressors]; or, 4) continuous evaporative cooling in all stages of a single compressor.

Recuperation could take the form of: 1) surface heat transfer between turbine exhaust gas and HP compressor delivery air; 2) evaporative recuperation where heat extracted from turbine exhaust would both heat and evaporate water into HP compressor delivery air and; 3) chemical recuperation whereby turbine exhaust energy would be transferred back into the turbine through a reformed natural gas fuel. In most cases, intercooling and recuperation would not displace entirely the bottoming cycle, but would instead reduce its size.

The initial ATS study, although preliminary in nature, showed increases in advanced turbine system efficiencies ranging from 0.5 to 4.0 percentage points depending on the cycle configuration and assumptions made. A change in configuration from the basic single shaft gas turbine to that of the reference system is thus anticipated to provide about 2.0 percentage points increase in system efficiency. Closed loop steam cooling of hot gas path components is anticipated to improve cycle efficiency approximately 1.5 percentage points.

Within the reference system configuration, additional efficiency gains can be made by: 1) high temperature developments in materials, coating systems, and cooling techniques to allow the attainment of a 2600°F (1427°C) rotor inlet temperature; 2) reduction in cooling air requirements; 3) application of advanced aerodynamic design techniques to further enhance component efficiencies; and, 4) reduction in leakages, clearances and parasitic losses.

It is expected that higher firing temperature in combination with cooling air reduction should yield about 2.5 percentage points in efficiency, while advanced aerodynamics, and reduced leakages, clearances, and losses would give a further

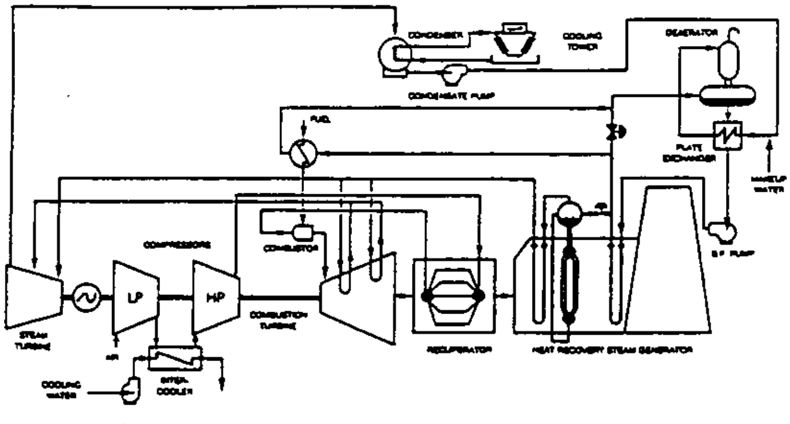

FIGURE 3.
PROPOSED REFERENCE SYSTEM (ICRCC)

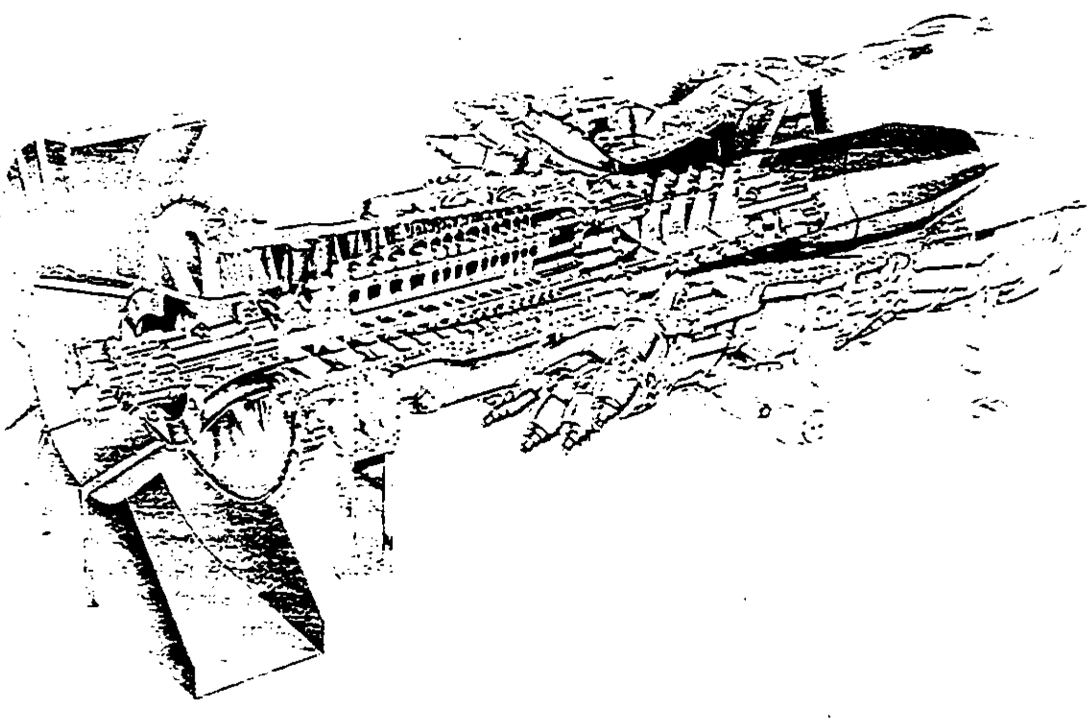

FIGURE 4.
WESTINGHOUSE 501F GAS TURBINE

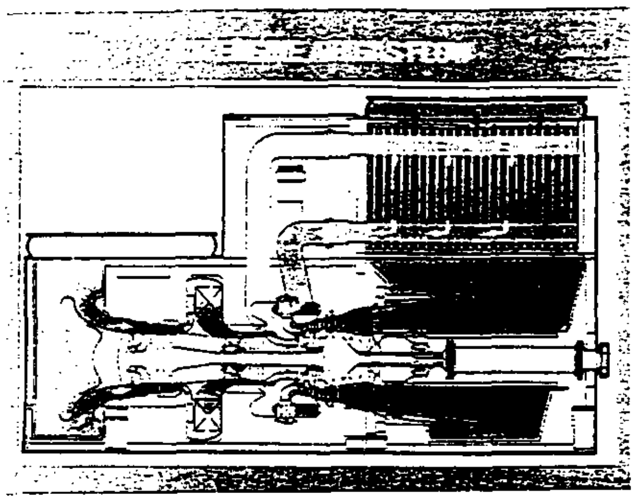

FIGURE 5.
WR21 ENGINE SYSTEM
WITH COMPACT SURFACE INTERCOOLER

1.0 percentage point. Therefore, within the reference system structure, 60% (LHV) target ATS efficiency is obtainable.

CYCLE STUDIES

Intercooling between compressor stages to reduce compression power has long been a feature of centrifugal compressor installations. The radial delivery of compressor air from each impeller lends itself to flow through an external water-to-air cooler before being introduced to the following impeller. Stages in the axial compressor of the industrial gas turbine are extremely close coupled, as shown in the 501F longitudinal (Figure 4), making intercooling impossible without significant configuration changes.

Intercooling between separate axial compressors has been done successfully in the past, where air is ducted from the LP compressor to a large water-spray evaporative cooler, and returned to the HP compressor inlet. More recently the compact water-to-air surface intercooler configuration, shown in Figure 5, is being developed by Westinghouse/Rolls-Royce for the U.S. Navy intercooled, recuperative (ICR) engine program (Crisall and Parker, 1993).

In the ATS program, either of these concepts may be adopted, depending upon the results of detailed thermodynamic cycle analysis comparing evaporative to surface intercooling. Additionally, the number of positions through the compressor where air is intercooled will be studied. It may be that double intercooling could prove most effective.

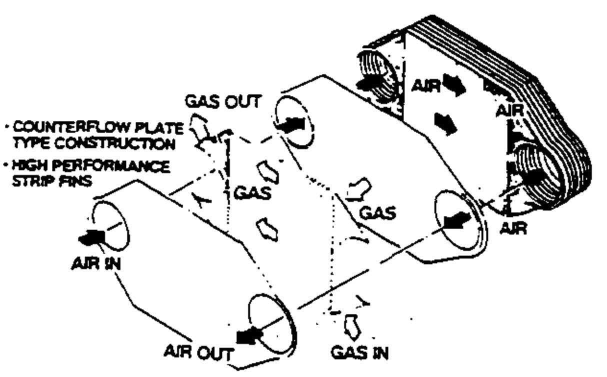

FIGURE 6.
RECUPERATION CORE DESIGN FEATURES

The major problem in the entire intercooling area is the conflicting requirements of the compression and intercooling functions. Velocity through the blading is high, approximately 640 ft/s (195 m/s) between stages, while for cooling (either evaporative or surface) velocity must be very low to reduce pressure loss and maximize residence time. In surface intercooling, a face velocity of approximately 32 ft/s (9.8 m/s) would result in a $\Delta P/P$ for the core elements of about 1%, and a effectiveness of over 90%. Diffusing, turning and reaccelerating air through a 20:1 area ratio will require careful engineering to minimize loss, as pressure loss cuts cycle efficiency and defeats the purpose of intercooling. A trade-off between area ratio, loss and effectiveness will have to be taken.

Evaporative intercooling requires an enormous pressure vessel. It would be a great advantage if evaporation could be done in a shorter length, in fact the ultimate scheme could be to evaporate in the compressor itself.

There is the possibility of introducing a water mist into the air flowing through the compressor. If the droplet size were of the order of 5 microns, collisions with the compressor blades would be minimized; such droplets would follow the flow streamlines. (It should be noted that the LP ends of steam turbine expanders have coped well with water droplet concentrations of up to 14% of the total flow.) A continuous cooling of the compression process by droplet evaporation might then be achieved. A preliminary analysis of such a compression process indicates a 25-30% reduction in the work of compression; a corresponding reduction in the temperature increase over the compressor; and, perhaps surprisingly, a reduction in the water used over a staged inter- and after-cooler process.

Techniques have recently been developed to produce such fine droplets, to maintain a uniformly distributed spray, and to

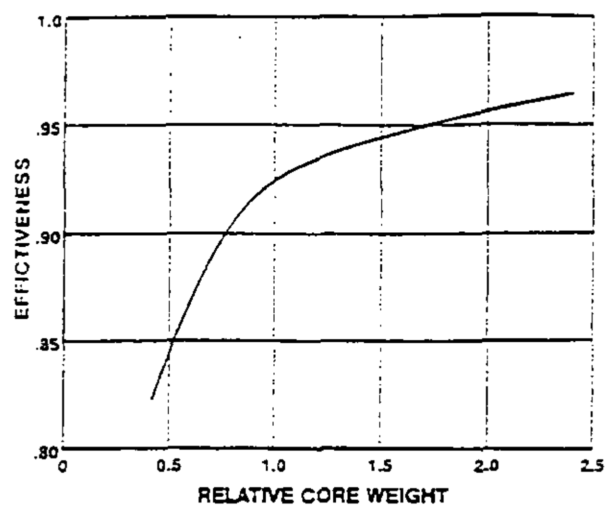

FIGURE 7.
RECUPERATION WEIGHT VS. EFFECTIVENESS

Techniques have recently been developed to produce such fine droplets, to maintain a uniformly distributed spray, and to avoid droplet coalescence. Incorporation of the water distribution system within the thin stationary aerofoils is an obvious means of access into the tightly spaced compressor stages, which will not disrupt the main flow. Spraying of these tiny, 5 micron, droplets into the air stream with minimum momentum mixing penalty would be the objective.

Designing the aerofoil shapes to properly account for the evaporation of the water mist in a finite distance downstream of the introduction points will require careful analysis of the time required for evaporation. Vaporization will be a function of water droplet temperature, injection velocity, air temperature, pressure, velocity and humidity. The modeling and experimental verification of the processes, just described, will be key prerequisites to being able to incorporate a design procedure into existing compressor blading design systems. Continuous evaporation within the compressor appears to be an attractive innovation which could improve the performance and reduce the cost of the ATS.

In the last 30 years, Westinghouse has supplied recuperative gas turbines from 5000 hp (Model 52) to 35,000 hp (Model 352) to the industrial market, thus, although recuperation is an innovation to the 2600°F (1427°C) RIT combined cycle, the surface recuperator itself is tried and proven. AiResearch, a Westinghouse team member, who will size, specify and supply the recuperator for the ATS, has designed and produced recuperators for large industrial gas turbines ranging in power levels from 6,000 to 60,000 hp. Gas turbines with AiResearch recuperators are being used for gas pipeline pumping, electric utility peaking power, and chemical plant power generation.

The standard core shown in Figure 6, utilized by AiResearch for large industrial recuperators, has a stainless-steel, brazed, plate fin construction. It is designed for 5,000 start/stop cycles and 120,000 hours of operation with exhaust gas temperatures up to 1200°F. The design has proven to be completely compliant by extensive laboratory tests and over 1,000,000 hours of field experience.

Higher temperature capabilities, if needed in the ATS engine, will be facilitated by the substitution of a new material which has 1300°F (704°C) temperature capability. Material development work is currently being done at 1500° to 1800°F.

AiResearch has performed a rough sizing for the ATS recuperator, at possible system operating conditions to achieve an effectiveness of 0.942 with a combined pressure drop of no more than 3.45% at an exhaust flow rate of about 1146 lb/s (520 kg/s). The high effectiveness, low pressure drop, and high flow rates dictate the size of the design, as shown in Figure 7. The requirements can be met by a system of counterflow heat exchangers, each handling part of the total flow of air and of gas.

The temperature drops and pressure losses in the piping leading from the combustor shell to, and from, the recuperator must be added to the estimated core performance to assess the full impact on the cycle efficiency. In the case of the Westinghouse Model 352 unit, air leaves the combustor shell, and returns to the unit through pipes located both above, and below, the horizontal split line on the shell (as shown in Figure 8 for the lower half). Within the shell itself, hot and cold air are separated by a flexible dividing wall. Turbine casing structural connectivity was obtained by the use of ribs (struts), each bolted at one end to the compressor cylinder, and at the other end to the turbine cylinder. The horizontally split combustor shell could thus be designed to contain the pressurized gases, yet be flexible.

Additionally the lessons learned in the Model 352 unit that will be factored into the ATS design, include the absolute necessity of sealing all clearances in the dividing wall and in the transition-basket, and transition-row 1 vane interfaces to ensure that compressor delivery air is actually forced to flow to the recuperator. Often in the past, when recuperated engine efficiency targets were not met, the cause was determined to be the bypass of air around the recuperator (in the combustor shell region). Also, the minimization of (using closed loop steam cooling in combination with thermal barrier coatings) and accounting for, heat pickup by compressor delivery air as it flows around the hot transitions while heading to the recuperator, needs to be considered. Every Btu picked up by compressor air reduces the heat that can be transferred from the exhaust, and thus, reduces cycle efficiency.

Initial studies suggested the possibility that evaporative recuperation may improve ATS efficiency, since more exhaust heat could be transferred into the topping Brayton cycle through moisture evaporation. In the current study, if the

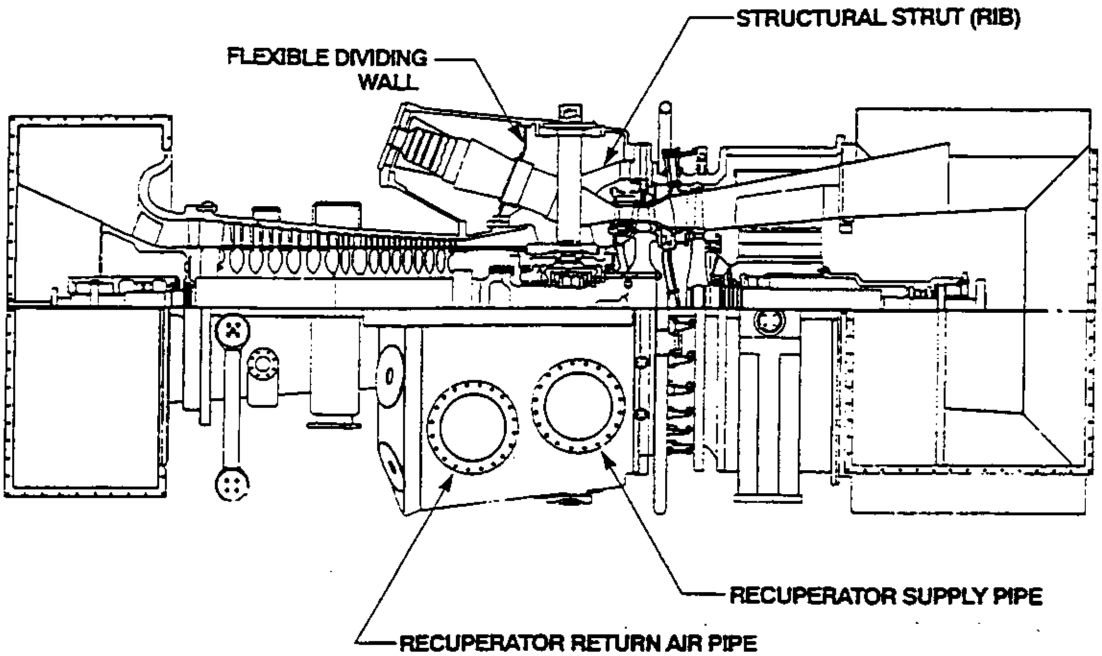

FIGURE 8.
MODEL 352 RECUPERATED GAS TURBINE

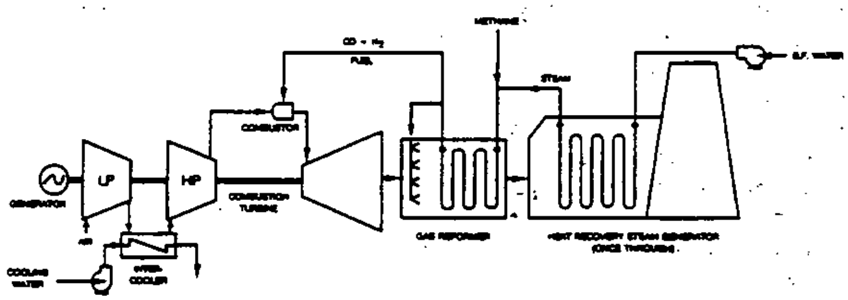

FIGURE 9.
INTERCOOLED, THERMOCHEMICAL RECUPERATIVE CYCLE

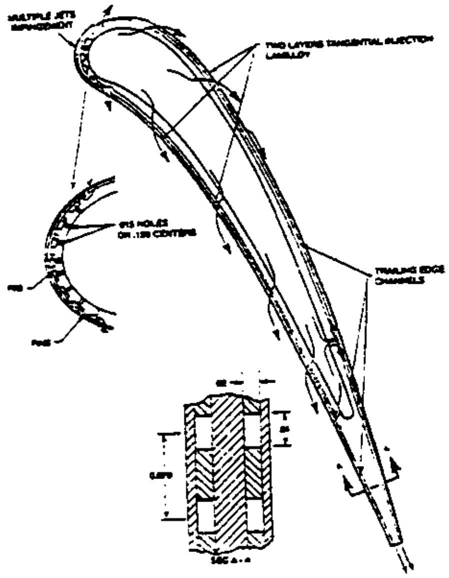

FIGURE 10.
COOLING GEOMETRY,
AIR-COOLED FIRST STAGE VANE

detailed thermodynamic cycle evaluation confirms this idea, then the possibility of water injection in the recuperator cores (either within the air side fin sections, or in the outlet headers) will be investigated.

Thermochemical recuperation of gas turbine exhaust energy, as shown in Figure 9, was also considered. Here only the gas turbine would produce power (no longer a combined cycle) by burning reformed, recuperated fuel. Methane and sufficient steam in the presence of a catalyst, and at appropriate temperature, will reform into a fuel consisting of H_2 and CO. None of the latent heat of vaporization of steam generated in the HRSG is lost to the cycle, and so the potential exists for efficient exhaust energy recuperation.

In current reforming technology, methane conversion efficiency is a function of pressure, temperature and excess steam/ methane ratio. Therefore, the reformed fuel could consist of CO, H_2, and CH_4 in various ratios depending on process variables.

COOLING REQUIREMENTS AND POSSIBLE ALTERNATIVES

Currently the first 3 (of 4) stages in the 501F turbine are cooled. First stage vane cooling is accomplished by a combination of impingement, film and convective cooling. First stage blade cooling incorporates serpentine, ribbed, and pin passages with external film. Both of these highly complex

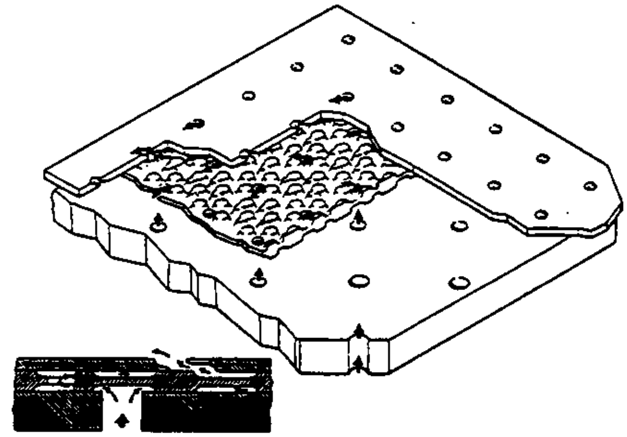

FIGURE 11.
SCHEMATIC OF TYPICAL LAMILLOY GEOMETRY

cooling schemes may be further refined, but potential cooling air reductions may be small. Alternatively, several approaches which have been investigated in the past may be reconsidered.

One of the most promising of these is shell/spar construction. Shell/spar vanes and blades may be formed by bonding a thin non-porous aerofoil shaped metal sheet (the shell) to an inner hollow cast support member (the spar), or by casting. Cooling air for the assembly is supplied from alternate cavities within the spar (the supply cavities). It flows along chordwise convection cooling channels between the spar and the shell and returns to the remaining spar cavities (the discharge cavities). The heated or spent cooling air is then routed from the discharge cavities to the main gas stream through exit holes in the shrouds or aerofoil.

Westinghouse, together with Allison/General Motors (GM), made an extensive study of air cooling approaches suitable for industrial size gas turbines at the elevated firing temperature of 2650°F (1454°C) and 3000°F (1650°C) under a U.S. Energy Research and Development Administration High Temperature Turbine Technology program. This work identified the feasibility of reaching these temperature levels using an advanced transpiration type air cooled technology.

One candidate for the row 1 vane is shown in Figure 10. This transpiration design consists of two layers of sheet material diffusion bonded to a hollow, cast spar. In the leading edge region, the sheet material is solid to minimize susceptibility to erosion and deposition. It is cooled by a multiple diffusion bonded scheme. In the trailing edge region, the outer layer of the sheet material is solid for aerodynamic and fabrication reasons and is cooled via chordwise channels located in the inner layer of the skin. Over the rest of the airfoil, the skin is tangential injection Lamilloy® which is a porous material that produces quasi-transpiration cooling. A cutaway drawing of a typical Lamilloy geometry is shown in Figure 11 (Lamilloy is a registered trademark of Allison Division/GM.) Preliminary studies have demonstrated the improved heat transfer effectiveness of Lamilloy over conventional multipass cooling.

Spanwise holes which taper from hub to mid height represent an improvement over constant diameter holes used in rotor blade cooling. With this technique, increased cooling in the critical mid height region is obtained by reducing over cooling in the hub region resulting from the lower coolant temperature as the cooling air enters the hub region. Increased hub region flow area obtained by tapering reduces heat transfer in that region, thus reducing temperature rise in the coolant as it flows radially outward.

Another variation in cooling hole geometry is to integrally cast circumferential ribs periodically spaced along the length of each cooling hole which act as flow turbulators to produce turbulent rather than laminar heat transfer for the cooling air. Field tests have been run on a Model 501 engine comparing smooth and turbulated holes. These tests verified that the heat transfer coefficient doubled by the use of the turbulators. A combination of turbulators and tapered holes could provide a very significant improvement in the cooling effectiveness required for the first stage rotating blade and possibly the second stage rotating blade.

Westinghouse policy has been to use high strength ferritic steel for rotating parts such as discs and torque tubes. The advantages of the low alloy, time-proven NiCrMoV turbine disc material over Ni based alternatives are better fracture toughness, lower cost, availability in industrial size forgings, and higher strength at the 750°F (400°C) design temperature. This material can only be used up to approximately 800°F (427°C) for temper embrittlement concerns.

For the ICR cycle, the HP compressor discharge temperature will always be sufficiently low so that it can be used, uncooled, for rotor cooling. The use of compressor discharge or interstage bleed air to cool engine parts reduces engine efficiency by up to 0.5% for each percent of cooling air utilized, depending on the coolant source and flow path entrance conditions. In the 501F engine, some 60% of the cooling air is used for aerofoil cooling. Most of the remainder leaks into the turbine hot gas flow through clearances in the parts which bound the coolant flow path as it proceeds to its destinations throughout the engine. These clearances are required to permit differential expansion of parts during temperature transients in order to avoid low cycle fatigue damage.

Another area that has a large potential for cooling air reduction is the reduction of the combustor outlet hot spot that the turbine blading must account for during design. In the case of the stage 1 vane, some vanes in the row will be directly impacted by the hot spot. This hot spot will be carried downstream, but will be attenuated somewhat by the averaging process due to blade rotation. Even so, the hot spot temperature may also have a special effect on the first rotating blade due to time averaged migration to the pressure side. Once the amount of hot spot reduction is known, the amount of cooling flow reduction can be estimated and the improvement in cycle efficiency predicted. Hot spot reduction will be estimated from ongoing tests of premix lean-burn combustors employing reduced cooling.

CLOSED LOOP STEAM COOLING

Another innovation which will provide increased ATS cycle efficiency at constant RIT, is closed loop steam cooling. Closed loop steam cooling involves directing a portion of the dry steam raised in the HRSG through the walls of hot end components, such as combustor baskets, transitions, and vanes, prior to expanding it through the steam turbine. The steam is superheated as it removes heat radiated and convected from the hot gas path. The idea of using steam to cool a gas turbine was suggested in the 1960s (Arsen'er et al., 1990).

When a comparison is made of the implications of replacing cooling air with bottoming cycle steam, while maintaining constant RIT, it is found that temperatures in the combustion reaction drop significantly because the same quantity of fuel is burned with much more air (original quantity, plus all the air that was used for basket wall, transition wall, and row 1 vane wall cooling). Thus, emissions may drop because the premix lean-burn, dry low NOx combustion will be even more lean provided flame stability is maintained.

The total pressure available at the 1st rotor inlet for expansion through the turbine will be higher, because no momentum mixing loss between cooling air and mainstream gas flow will have occurred. Therefore, extra power will be produced by the gas turbine. Also, the hot spot temperature from each basket will be reduced since much less $\Delta T/T$ was required. This will reduce both row 1 vane and blade cooling requirements. Additional kilowatts will be generated in the steam turbine from the higher temperature (but lower pressure) steam expanding through the steam turbine.

Use of steam cooling in the row 1 vane would raise ATS efficiency 0.8 percentage points. Inclusion of the heat extracted from the baskets and transitions would further raise efficiency. Steam cooling for row 2, 3 and 4 vanes would also produce more power, and would additionally boost ATS efficiency, since exhaust temperature would rise, thus reducing fuel consumption (more heat transferred across recuperator to compressor delivery air), and recuperator exhaust temperature would also rise resulting in the raising of more steam in the HRSG. The net effect could be to raise ATS efficiency another 0.5 percentage point or more.

Closed loop steam cooling is a very attractive means of boosting ATS combined cycle efficiency. The technical challenges, are significant, but are all understood and lend themselves to design and analysis. Factors that must be considered include: 1) control requirements to ensure an uninterrupted supply of steam; 2) steam pressure loss minimization and leakage elimination; 3) heat transfer requirement and steam capacities; 4) steam cooling paths through components and casting limitations; and, 5) maintaining constant thermal stress to assure required low cycle fatigue life.

The possibility of applying closed loop steam cooling to the rotating blades should also be investigated. A conceptual design of steam transfer into, and retrieval from, the shaft, as well as routing through the discs and blades will be investigated. The complexity and development effort will be weighed against the potential gain in cycle efficiency to determine the advisability of pursuing this option.

ADVANCED AERODYNAMIC DESIGN

The 501F compressor was designed between 1986 and 1988 by Westinghouse using 20 years of successful design experience on the 501 frame (501AA to 501D5). The experimental knowledge gained in the 501 compressor was incorporated into a design system using appropriately prescribed aerofoil shapes with their loss and deviation characteristics incorporated into a compressible flow. The key element in the design process was to achieve a compressor design which could be built and applied confidently in the engine without developmental testing. This target was achieved using the base of experience which grew as the 501 evolved from the 501AA to the 501D5. The total experience was developed into compatible parts of a system for design and performance analysis which provided the necessary confidence to develop the new 501F compressor.

The design was successful, and resulted in the high efficiency, low loading, high surge margin, 16-stage compressor. For the compressor(s) of the ATS it is anticipated that this traditional Westinghouse approach will be strengthened.

The 501F turbine aerodynamic design benefited from turbine design experience and the application of advanced three-dimensional design techniques. The resulting high efficiency four-stage turbine configuration contributed greatly to the success of the 501F. Within this basic four-stage turbine philosophy, it is anticipated that further aerodynamic refinement can be incorporated.

REDUCTION IN LEAKAGES, CLEARANCES AND MECHANICAL LOSSES

Additional gains in ATS efficiency can be realized through more efficient utilization of cooling and sealing air, and reduction in mechanical losses. It is intended to incorporate ideas which have in many cases been formulated, but not implemented in the past.

Improvement of component performance is possible by reducing clearances in both the compressor and turbine. The use of knife edge seals on disc seal arms which wear into abradable material such as felt metal or filled, honeycomb surfaces that are mounted on the stationary compressor and turbine seal housings, is a method of minimizing running clearance. The process of rubbing the knife edge into the abradable surface during both steady-state and transient operation will produce a groove in the abradable material and establish both a minimum radial and axial running clearance. The final steady-state clearance will be smaller than when seals are designed not to rub. With the knife edge seal rubbing into the abradable material, the frictional heat generated by rubbing is not large enough to cause structural distortion.

Innovative brush seals which are gradually finding acceptance in gas turbines, will be considered for the ATS engine. These seals restrict the flow of air through a gap between stationary and rotating components by filling the gap with a brush of very fine (0.0025 in.[.0064 cm]) alloy wires. The wires are oriented at 45° to the radial direction so they can bend and accommodate transient reduction of radial clearance. Cost versus benefit from reduced leakage flow is the major tradeoff to be considered.

Further clearance reduction beyond wear-in type seals would be to utilize an active clearance control design that would reduce the running clearance caused by transients. It is well documented that the largest clearance required between rotating and stationary parts occurs during startup of a cold engine. If a method can be developed to minimize or eliminate this added clearance then operation with a running clearance set only by steady-state operation can be achieved. Reduction of the transient clearance is important because, in most cases, especially for the blade tips, the transient clearance requirement is more than double that required for steady-state. The effect of elimination of the transient allowance is to reduce the steady-state leakage area and resulting flow by one half.

Some improvement in overall cycle efficiency is possible with a reduction in mechanical and/or windage losses. The use of a directionally lubricated thrust bearing to replace the more conventional flooded design will reduce the mechanical loss. These bearings are being used commercially and could be used for this application. If a concern exists that a sudden loss in oil supply would cause damage to the bearing and rotor, an oil reservoir could be designed to protect the bearing during an emergency shutdown.

Another area for even greater improvement would be the application of active magnetic thrust and radial bearings in place of the conventional oil lubricated bearings. These type of bearings are currently being used in small gas turbines (= 2 MW), and special applications, and are under development for large size applications such as gas turbines. This system would eliminate the need for an oil lubrication system. It would benefit the cycle by reducing mechanical losses for the turbine bearings, oil system pumps, oil cooler fans, etc., and, also eliminate the oil cooler, piping, fire protection, controls, etc.

Another area that translates into a mechanical loss is windage created by any rough or protruding surface, such as uncovered coupling bolts, blade root projections, etc. In any design, attention to minimizing areas that create windage must always be considered.

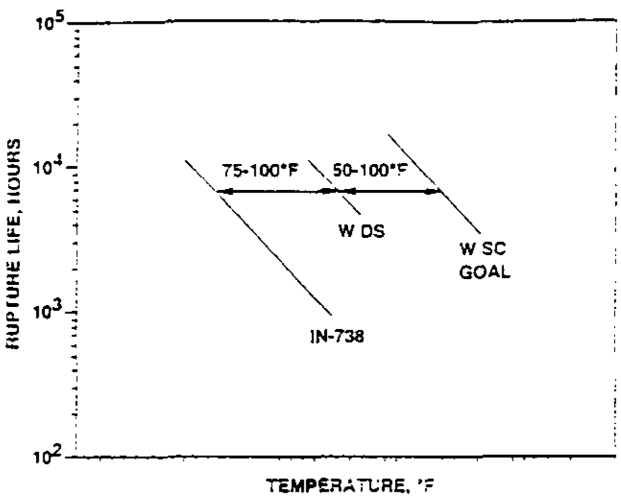

FIGURE 12.
TEMPERATURE ADVANTAGE OF
Ⓦ DS AND Ⓦ SC ALLOYS

HIGH-TEMPERATURE MATERIAL REQUIREMENTS AND NEEDED DEVELOPMENT

Materials play a critical role in the design of engines with higher firing temperatures, since the temperature capability of materials establishes the cooling requirements which in turn affects the engine efficiency. Hot section components for land-based engines typically require materials with superior mechanical properties and good corrosion/oxidation resistance at elevated temperatures. Ni and Co base super alloys with high Cr content are extensively used for this application, the chemistry of these alloys having been adjusted to yield an optimum combination of strength and corrosion/oxidation resistance. At higher firing temperatures, additional protection of hot section superalloy components by coatings has became an absolute necessity. It may not be possible to further adjust the composition of superalloys to obtain higher strength and better hot corrosion and oxidation resistance at elevated temperatures without adversely affecting their stability. The high temperature strength of superalloys arises from a combination of various hardening mechanisms, the most potent being precipitation hardening. The commercial superalloys are solution treated (around 2050° to 2300°F [1121° to 1260°C]) and aged to obtain optimum elevated temperature properties. Thus, the ultimate temperature capability of these materials is limited by the solution treatment temperature. The advent of single crystal casting, which eliminates the need for grain boundary strengthening elements, makes it possible to increase the solution treatment temperature.

The third and fourth generation Ni base single crystal alloys offer a temperature advantage of about 200°F (111°C) compared to conventionally cast IN-738 which is used as a blading material in land based engines. The French have developed a single crystal alloy with 16% Cr which offers a 90° to 100°F (50° to 56°C) temperature advantage at about 22 kpsi (1547 kg/cm^2) stress level when compared to conventionally cast IN-738.

Westinghouse has been developing corrosion resistant directionally solidified (DS) and SC Ni base superalloys for land-based engine applications (Cr levels around 16%). At typical row 1 blade operating stress levels, these new alloys in the DS condition offer a temperature advantage of around 75° to 100°F (42° to 56°C) when compared to equiaxed IN-738, as shown in Figure 12. Westinghouse SC alloys are expected to exhibit 50° to 100°F (28° to 56°C)higher temperature capability than that of the corresponding DS alloys.

Coatings used for land based hot section components fall into the categories of corrosion resistant and thermal barrier. As the name implies, the former is applied for corrosion and/or oxidation protection, while the latter is used to reduce the heat transfer between the gas stream and the substrate of the coated component. It is known that when corrosion resistant MCrAlY coatings are exposed to elevated temperature, they degrade due to inward and outward diffusion of elements present in the coating. The extent of degradation depends both on service temperature and time. Figure 13 shows MCrAlY coating degradation as a function of service exposure at a temperature of around 1700°F (927°C). After 9000 hours of operation, the coating structure reveals denuded zones at the outer surface as well as at the coating - substrate interface. The coating was also locally corroded/oxidized at its outer surface. After 19,000 hours of operation, the coating was almost completely corroded/oxidized, and the substrate was oxidized significantly. These results show that current MCrAlY type coatings do not offer reliable protection at ATS target metal temperatures.

Advanced coatings are needed to realize the full potential of new DS and SC alloys. At present, thermal barrier coatings (TBCs) are being used in service on static components and are being evaluated for application to rotating parts. The TBC is needed to lower the substrate temperature and to improve the reliability of MCrAlY coatings. Though these coatings are being used industry wide, development work is needed to increase their temperature capability for reliable service in ATS engines.

For both corrosion resistant and TBCs, programs to evaluate all available coatings, select candidate coatings, perform screening tests and outline long term development plans will be established.

The target RIT for an ATS engine requires implementation of SC and DS components coupled, with advanced coatings (including TBC) to minimize cooling air/steam usage in the ATS engine. The key is to take advantage of increasing creep

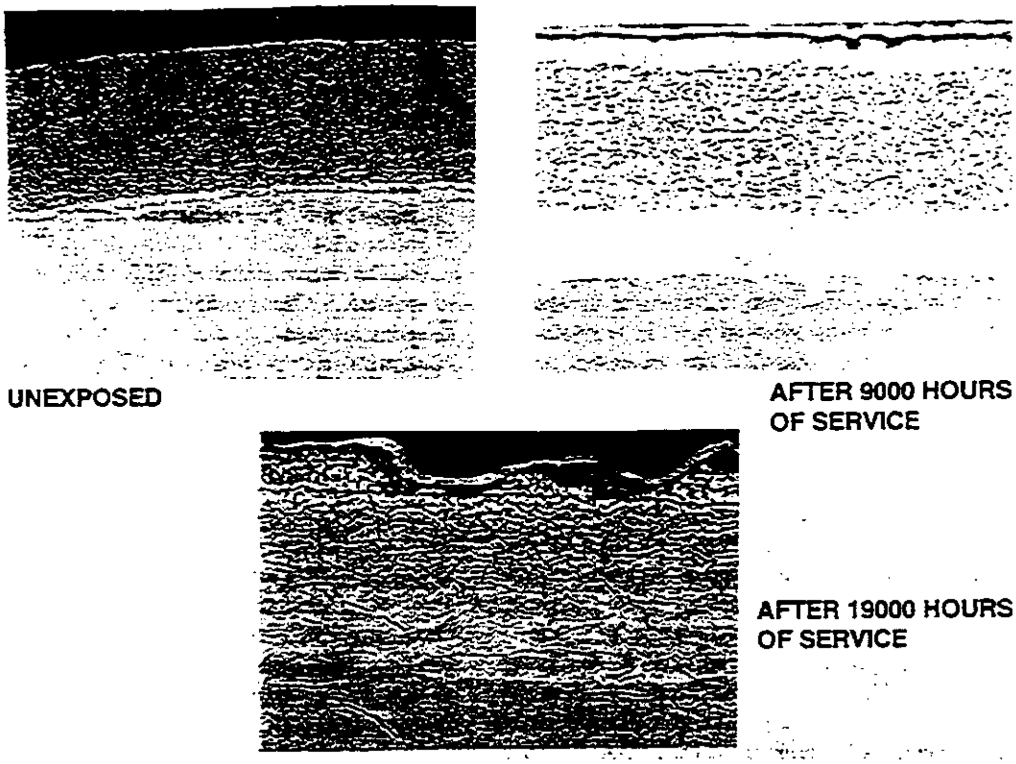

FIGURE 13.
MCrAlY COATING DEGRADATION

strength by letting the blade temperature increase. Corrosion resistant SC materials are expected to exhibit about a 150°F (83°C) temperature advantage compared to current conventionally cast blading materials, while, TBCs typically lower service metal temperature of the coated component by an additional 150°F (83°C).

Ceramics and ceramic composites offer excellent temperature capability and are less significantly affected by the environment (corrosion & oxidation) than superalloys. For continuous advancement in high temperature capability, it will be necessary to consider implementing ceramics, composites, and intermetallics. In these cases, available structural ceramics, and composites will be assessed for their potential application. Ceramics and ceramic composites will be considered for stationary components of the ATS engine.

COMPONENT SPECIFIC MATERIALS

Combustors

Improved metallic materials with TBC or fiber reinforced ceramics will reduce cooling air requirements. Minimization of cooling air flows will reduce NOx emissions.

The use of combustors fabricated from advanced ceramics and fiber reinforced ceramic composites, which provides mechanical integrity, greatly advance the technology. Ceramic composites will allow the use of uncooled or minimally cooled combustors systems. Ceramics will be evaluated as alternatives to conventional metal based combustors for the long-term.

Transitions

Advanced Ni based materials and high temperature coatings coupled with TBC will be considered for the ATS engine. Recently developed oxide dispersion (Y_2O_3) strengthened superalloys offer excellent creep strength, as well as superior oxidation and hot corrosion resistance. These alloys, along with ceramics and ceramic composites, may be evaluated to replace conventional Ni base transitions.

Vanes and Blades

Given the time-frame of the ATS program, intermetallics and composite materials may not be available for the subject application. These materials will be evaluated for future engine applications. Availability of these materials for this application depends on how rapidly the technology advances and how much design data would be generated and available. However, any developmental materials such as intermetallics and metal matrix composites will be evaluated as they become viable within the development time-frame for both vane and blade applications.

Stationary vanes do not have to withstand the rotating stresses that are imposed upon blades, but are subjected to environmental attack and the thermomechanical stresses imposed by thermal cycling. Advanced superalloys that are processed by conventional and improved process methods, including DS and SC casting, are the principal candidates for improved vane materials.

Improved oxidation resistant MCrAlY coatings are candidates for this application. In addition to corrosion resistance, the properties of the coatings need to be evaluated with respect to mechanical stability for candidate vane materials under thermal cycling conditions.

TBCs are expected to help improve the cooling efficiency of vane systems. Key issues regarding the application of TBCs to vanes are the mechanical compatibility of the TBCs with the vane substrate and the interaction of the porous coating with molten corrosive oxides and sulfides. TBCs, primarily those based upon yttria stabilized zirconia, will be evaluated with respect to mechanical and chemical effectiveness for application as improved coatings for vanes.

Intermetallic and composite systems will also be considered for vane applications. These systems are currently limited by the very few intermetallics that display good high temperature ductility and are chemically stable in oxidizing/hot corrosion atmospheres. The most promising systems are those based on NiAl intermetallic. The properties of these systems will be reviewed for prospective vane applications.

Ceramic materials will be assessed for vane applications in the natural gas-fired advanced turbine systems. While ceramics, particularly those based on oxide systems, offer increased temperature capability, the structural integrity of the large sized components that are required in the power generation ATS must be demonstrated. Currently, fiber reinforced ceramic composites for advanced gas filter systems are under development at Westinghouse as part of a DOE sponsored program (Newby and Bannister, 1993).

The most viable candidate materials for improved capability turbine blades are advanced nickel base superalloys that are processed by DS and SC casting methods, in combination with improved oxidation/hot corrosion resistant coatings and thermal barrier coatings.

The study of improved alloy compositions and processes will be based upon current Westinghouse approaches to develop DS and SC alloys for land-based turbine blades. On-going Westinghouse programs have demonstrated that improved high temperature creep and fatigue performance can be delivered by directional solidification of modifications of current turbine alloys. Production of SC alloys has also been demonstrated. The key to utilization of these alloys is to develop coatings that can operate at the high temperature capability of DS and SC materials.

Hot corrosion/oxidation protection of the turbine blades is expected to be provided by advanced coatings based upon existing MCrAlY coatings and oxidation resistant alloy modifications. Candidate coating materials will be assessed based upon their ability to protect against environmental attack and their compatibility with candidate blade alloys. Improved coatings will be evaluated based upon their ability to withstand hot corrosion attack at intermediate temperatures on the cooler portions of the blade, high temperature oxidation attack at the maximum temperatures of the blade, and their impact upon the mechanical performance of the underlaying blade alloy.

The current risks of TBCs are mechanical and corrosion related. The mechanical properties of TBCs must be demonstrated to ensure integrity against spalling under thermal cycling conditions. The corrosion issue is not due to corrosion of the coating itself but derives from the ability of the coating to absorb and retain corrosive molten oxides and sulfides that attack the underlaying metallic blade and coating. Processing of the TBC to avoid this accelerated form of attack will be required to improve the capabilities of TBCs.

Turbine Discs

High strength, low alloy steel discs are used in the turbine section of the 501F with 2300°F (1260°C) RIT. The service temperature of these discs is maintained below 750°F (400°C) through the use of cooling air to minimize in-service temper embrittlement.

COMBUSTION

The extremely low NOx (8 ppmvd @ 15% O_2), CO and UHC emission levels which must be achieved at the higher temperature of the fully developed ATS is a serious challenge. Significant internal R&D is currently being devoted to the achievement of single digit NOx by pursuing the premix, lean-burn strategy.

Alternatively, it has been proposed to investigate catalytic combustion, which in the past did demonstrate single-digit NOx emissions. In the current investigation, a test program is outlined to begin investigating the thermomechanical stability of the catalyst substrate, and the effect on catalyst performance of prolonged exposure to high temperatures. ATS emission control will be discussed in a future paper.

SELECTION OF COAL-FIRED PLANT REFERENCE SYSTEM

Within the natural gas-fired ATS program, a brief effort will be expanded to select a reference coal-fired ATS (Webb, Parsons and Bajura, 1993). Westinghouse will apply its background with advanced coal-fueled power generation (Bannister, Bevc and Newby, 1993) to provide the information needed. The selected coal-fired ATS will be described in sufficient detail to provide information that can be used to identify any conversion issues from natural gas to coal. The natural gas-fired ATS is to be applied to a coal-fueled ATS combined cycle. Modifications to the gas turbine components are to be minimized. Modifications should account for the differences in fuel.

Estimates will result for the reference coal-fired system with respect to thermodynamic cycle efficiency, turbine tolerance factors (e.g., expansion gas particle loading, particle size distributions, and alkali vapor content), turbine maintenance (i.e., blade replacement and cleaning frequency, and environmental emission levels (e.g., particulate, NO_x, CO, SO_2 and CO_2).

MATERIALS FOR COAL-DERIVED FUELS

Conversion of a coal-fired advanced turbine systems operation imposes, more severe environmental constraints on the materials selection for the turbine components. Specifically, sulfur, halides, and low levels of metallic elements, such as vanadium, will be carried by the coal-derived fuel and will present significant corrosion attack to the hot section components of the engine.

Under conditions of increased corrosion in the coal-fired ATS, advanced superalloys will be required to form corrosion resistant chromia scales rather than alumina scales; this capability may significantly impact the availability of higher strength alloys under these conditions.

Ceramic materials are expected to be less significantly affected than metallic materials. In specifically severe cases, it will be necessary to consider ceramics for some selected applications. For these cases available structural ceramics will be assessed for potential applications.

The suitability of coatings will also be significantly affected by the atmosphere. Chemically protective coatings will, in a similar manner to the effect on metals, be required to display corrosion resistance. Coating selection for coal-fired ATS components will be more significantly weighted by the ability of the coating to form corrosion resistant chromia coatings. Coatings will be assessed by evaluating the available coating corrosion data for oxygen/ sulfur atmospheres that are similar to the coal-fired ATS atmosphere.

The conversion to coal as a fuel is also expected to impact the use of TBCs. Although TBCs themselves are expected to be only minimally corroded by the more aggressive nature of this atmosphere, significant effects may be expected on the substrate and bond coat. The acceptability of TBCs will be determined by evaluating the data that define the role of the TBC in accelerating the corrosion of the bond coat and the underlaying substrate. These data will be evaluated with respect to the more corrosion resistant atmosphere.

CONCLUSIONS

The feasibility of achieving a 60% (LHV) efficiency in a natural gas-fired ATS cycle within an 8-year period has been established. Cycle innovations, a 2600°F (1427°C) RIT, reduced cooling air usage, improved component efficiencies and improved material/ coating systems will all be needed to accomplish this goal. The ATS needs to be cost competitive, as well as be fuel flexible to use coal-derived and biomass fuels.

This paper has outlined some of the development work that Westinghouse is pursuing to identify the equipment requirements and technological barrier issues that need to be solved during the component development phase of the ATS program. Critical components will be tested. Subscale assemblies will be evaluated. The laboratory experimental data will lead to the testing of full-scale components and integrated subsystems in the next stage of the ATS program.

ACKNOWLEDGMENT

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..KW-TURBINE SYSTEMS PROGRAM. LOWER HEATING VALUES. NATURAL GAS FIRED.
..AB-In cooperation with U.S. Department of Energy's Morgantown Energy Technology Center, a Westinghouse led team is working on the second part of an 8-year, Advanced Turbine Systems Program to develop the technology required to provide a significant increase in natural gas-fired combined cycle power generation plant efficiency. This paper reports on the Westinghouse program to develop an innovative natural gas-fired advanced turbine cycle which, in combination with increased firing temperature, use of advanced materials, increased component efficiencies and reduced cooling air usage, has the potential of achieving a lower heating value plant efficiency in excess of 60 percent.

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G. S. HOWARD, Westinghouse Retiree

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EVOLUTION OF HEAVY-DUTY POWER GENERATION AND INDUSTRIAL COMBUSTION TURBINES IN THE UNITED STATES

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ABSTRACT

This paper reviews the evolution of heavy-duty power generation and industrial combustion turbines in the United States from a Westinghouse Electric Corporation perspective. Westinghouse combustion turbine genealogy began in March of 1943 when the first wholly American designed and manufactured jet engine went on test in Philadelphia, and continues today in Orlando, Florida with the 160 MW, 501F Advanced Combustion Turbine. In this paper, advances in thermodynamics, materials, cooling, and unit size will be described. Many basic design features such as two-bearing rotor, cold-end drive, can-annular internal combustors, CURVIC[2] clutched turbine discs, and tangential exhaust struts have endured successfully for over 40 years. Progress in turbine technology includes the clean coal technology and advanced turbine systems initiatives of the U.S. Department of Energy.

HISTORICAL PERSPECTIVE

Westinghouse, a major supplier of steam turbines since the beginning of the 20th century (Bannister and Silvestri, 1989), obtained the experience it needed to develop land based power generation combustion turbines through its entry into the jet engine business. In March of 1943, the first wholly American designed and manufactured jet engine went on test at Westinghouse, 15 months after obtaining a contract from the U.S. Navy. The engine (designated WE19A) had a thrust of 1130 lb (513 kg) and weighed 827 lb (375 kg). Without any knowledge of German, British or other United States activity, Westinghouse developed the first American jet engine with an axial compressor, an annular combustor, a turbine and a jet exhaust nozzle. The device resembled the Whittle engine developed in England, but there were major differences, in that an axial flow compressor was used, along with a submerged combustion chamber.

An improved version of the engine, the WE19B, was test flown at Patuxent Flight Test Center in January 1944 as a booster unit on a Chance Vought Corsair. It delivered 1365 lb (620 kg) of thrust, weighed 731 lb (332 kg), and one year later, as the J30, was used to power the Navy's first jet fighter, the McDonnell Douglas FH-1 Phantom. Sixty-one (61) Phantom planes were equipped with the J30 engine.

The J34, a 34-in. diameter engine that delivered 3000 lb (1362 kg) of thrust, was the last production aero engine built by Westinghouse. It was used extensively by the Navy in the McDonnell Banshee. Overall, Westinghouse supplied the engines for 1223 Navy jets before exiting the business in 1960.

Westinghouse's experience with land based gas turbines started in 1945 with the development of a 2000 hp gas turbine generator set (W21) that consisted of a single reduction gear, compressor, 12 combustors and turbine (Putz, 1946). A thermal efficiency of 18% lower heating value (LHV)* was obtained. By 1948, Westinghouse built a 4000 hp gas turbine locomotive with the Baldwin Company that used two of these units. Initial operation was on the Union Railroad burning distillate fuel oil. Later operation was on the Pittsburgh and Lake Erie Railroad using residual oil fuel.

In the late 1940s, the U.S. Navy wanted to develop a back-up for the nuclear submarine power plant. Westinghouse received a contract designated "Wolverine" to develop a gas turbine engine for surface, snorkel, and submerged operations. A gas turbine engine of 7500 shaft horsepower was designed to operate in open, semi-closed, and closed cycle modes. In the semi-closed mode, oxygen was supplied to the cycle from the snorkel. While in the submerged mode (fully closed cycle) oxygen was supplied and CO_2 gas was discharged overboard. Engine testing was done at the U.S. Naval Test Laboratory

[1] Westinghouse Electric Corporation Retiree
[2] Gleason Works

*All efficiency values in this paper will be LHV.

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in Annapolis, MD during the early 1950s. While the power plant met all contract requirements, the success of the nuclear submarine program eliminated the need for "Wolverine."

By 1954, gas turbine power plants were becoming a practical and economic reality for particular applications in the power industry. Westinghouse was offering 3 basic gas turbine designs for power generation which had a rotor inlet temperature of 1350°F (732°C) (Putz, 1954). The units were rated at 3500 kW, 5000 kW (with a regenerator), and 15,000 kW (with a regenerator and intercooler). The latter unit was designed for a full-load simple cycle efficiency of 29% compared to the full-load efficiency of 21% for the 5000 kW unit.

West Texas Utilities helped pioneer combustion turbine base-load generation with a 5000 kW unit that was installed in 1952 (Cox, 1957). In 1954, another 5000 kW combustion turbine was installed. The waste heat from this unit was used for feedwater heating until 1959 when it was incorporated into a fully-fired combined-cycle system. This system generated 39 MW in the steam cycle and 5 MW in the combustion cycle.

The Westinghouse single shaft, two bearing, combustion turbine with a reduction gear was first introduced in 1952 as the W81 at 5.5 MW (Stephens, 1952). In 1967, the W251A introduced major innovations such as minimum peak stress turbine blade roots, and cooled low alloy steel turbine discs. Today there are over 300 gear driven units operating in both simple, combined, and cogeneration cycle modes with the current W251 B12 rated at 48 MW.

The first direct drive combustion turbine (W201) at 20 MW was introduced in 1960. A modified W201 combustion turbine was used in 1960 as part of a blast furnace system at U.S. Steel's Chicago Works. The successful development program included design, development, and manufacture of a special casing configuration to efficiently remove compressor discharge flow and deliver it externally to the engine. (Krapf and Stephens, 1958). Gases from the blast furnace were burned and ducted back to the turbine through a special duct and scroll.

In 1962, Westinghouse designed the first "packaged" combustion turbine power plant, which was delivered to the City of Houma. The "packaged" power plant concept is retained today in the ECONOPAC[3] system. This simple cycle package includes the combustion turbine engine assembly; generator and exciter; starting package; inlet and exhaust systems; and plant auxiliary equipment. It is constructed in modules for easy shipment and installation. The ECONOPAC system is also the base for cogeneration and other heat recovery applications.

In 1967, a supercharged direct drive W301 at 25 MW in a heat recovery application was installed at West Texas Utilities San Angelo's Power station and achieved over 39% in combined cycle efficiency, the highest in the U.S. for a number of years.

The first direct drive W501 combustion turbine went into service in 1968 at the Dow Chemical Company with a rating of 42 MW. Many basic design innovations were introduced on the W501A including unique cooled, filtered rotor cooling systems and tilting pad bearings. By 1974, fifty-seven 501s were operating. The 501 technology has grown steadily over 25 years. The current production engine, W501D5 is now being offered with an ISO rating of over 118 MW and the 501F is at 160 MW. The "F" technology includes increases in airflow and firing temperature, improved component efficiencies, and advances in materials, turbine cooling, and dry low NOx. Currently, there are about 300 direct drive units in operation. A complete listing of all models sold is shown in Table 1 where total service hours is seen to exceed 47 million.

For over 30 years, Westinghouse has had licensee arrangements with several combustion turbine manufacturers, the major ones being Fiat Avio in Italy and Mitsubishi Heavy Industries (MHI) in Japan. These partners have contributed to the development of the combustion turbine with Westinghouse, particularly in the 50 Hz market; i.e., the TG50 of Fiat Avio and the MW 701 of MHI, 50 Hz versions of the Westinghouse W501 Model. During the late 1980s and early 1990s, a tri-lateral alliance was formed between Westinghouse, Fiat Avio and MHI to replace the old licensee/licensor arrangement. The alliance provides for joint development of combustion turbines and sharing of technology. The 501F and the 50 Hz 701F are the first engines developed under this partnership arrangement.

WESTINGHOUSE ENGINE TECHNOLOGY EVOLUTION

Westinghouse model designations initially meant the horsepower output in thousands followed by the number of shafts with letters denoting whether the engine was intercooled, regenerated, reheated, or extracted. Therefore, W201RE would mean 20,000 hp, single shaft, regenerated, with cycle air extraction. Later engine evolution, accelerated by materials and cooling improvements, invalidated such designations.

It should be noted that in this paper, the term "firing temperature" for uncooled engines will be defined as the average temperature leaving the combustors and entering the turbine. This neglects the effect of turbine inlet leakage and first stage vane cooling and will be confined to first and second generation engines. For all engines discussed in the third through fifth generations, rotor inlet temperature (RIT) will be used to define the temperature level. RIT is the mixed out average temperature entering the first rotor blade. The basic reason for the change is to include the effects of cooling air used by the row 1 vane.

Westinghouse combustion turbine evolution will be presented as follows:

First Generation - This period of the mid 1940s was heavily influenced by jet engine and steam turbine philosophies and will be confined to the W21 engine.

Second Generation - This period spanned from the late 1940s to the mid 1960s and generated many major design innovations that have been retained in our present designs.

Third Generation - The "Northeast Blackout" of 1965 initiated this generation that saw an explosion of design effort and introduced air cooling, higher firing temperatures, improved materials, and precision cast components. In addition, innovative concepts such as cooled, filtered rotor cooling air and rotor blade cooling and low alloy turbine discs, were introduced.

Fourth Generation - The fourth generation covered the 1970s and the 1980s, a period that was severely impacted by the "Oil Embargo" and the Fuel Use Act. Essentially the combustion turbine market disappeared until the Fuel Use Act was repealed and the Public

[3] Westinghouse Electric Corporation

TABLE 1. WESTINGHOUSE SUPPLIED COMBUSTION TURBINES

Model	No. of Units	Rating (kW)	Installed Capacity (kW)	Oper. Year	Mech. Drive	60 Hz	50 Hz	Service Hours*
W21	5	1,300	6,500	1949	3	5	0	500,000
W31	6	2,200	13,200	1956	6	0	6	600,000
W41	8	3,100	25,000	1961	4	7	1	800,000
W52	14	3,800	52,400	1956	2	14	0	1,400,000
W62	15	5,500	83,000	1961	13	15	0	1,500,000
W72	17	6,300	107,100	1961	17	1	16	1,700,000
W81	30	5,500	155,400	1952	17	30	0	3,070,345
W82	12	6,000	72,300	1962	12	0	12	1,200,000
W92	29	8,200	231,500	1960	27	29	0	2,900,000
W101	83	7,450	607,600	1961	54	70	13	8,306,630
W121	2	8,950	19,000	1959	1	2	0	200,000
W122	1	10,215	10,200	1967	0	1	0	100,000
W171	39	16,425	509,900	1961	1	28	11	2,184,371
W191	182	17,300	3,044,500	1965	14	127	55	8,596,174
W201	3	18,650	62,600	1960	0	3	0	296,022
251	195	49,100	5,626,700	1967	1	147	48	4,322,303
W301	32	23,850	938,100	1964	0	27	5	1,638,369
W352	7	27,900	190,900	1979	4	7	0	665,227
501	227	159,000	13,929,466	1968	0	227	0	7,192,050
701	8	136,900	1,095,200	1992	0	0	8	0
TOTAL	915		26,780,566		176	740	175	47,171,491

* Service hours are estimated as of 12/31/92

Regulatory Policies Act of 1978 (PURPA) was upheld. The latter part of this generation saw the introduction of increased firing temperatures and improvements in materials, cooling and efficiency. The W501D5, introduced in 1983, became the largest and most efficient 60 Hz single shaft combustion turbine in the world at 100 MW and over 33% efficiency for a simple cycle plant.

Fifth Generation - This generation covers the late 1980s to the present and starts with the 501F advanced combustion turbine with an RIT of 2300F (1260°C). Units in this generation will extend the air flow limits of last row turbine blades, the temperature limits of row 1 turbine vanes, and the flow limits of row 1 compressor blades through the use of advanced materials and cooling.

First Generation

The jet engine designers needed a high pressure, high flow facility, for combustor and turbine aerodynamic development. A lightly loaded 23-stage axial flow compressor was designed to meet the needs of the air supply facility. This test compressor together with a 12 can interconnected combustion system and an 8-stage turbine formed the first industrial combustion engine at 2000 hp. The air flow was 35.5 lb/sec (16.1 kg/sec) at a pressure ratio of 5, a firing temperature of 1250°F (677°C), and a power output of 2070 hp at an efficiency of 18 % (Mochel, 1947). The unit was a single shaft, three bearing design with a design speed of 8750 rpm. The external load was driven from the compressor end of the engine.

After a development period, the engine was first applied (1948) as a twin set in a locomotive built by Baldwin and Westinghouse (Stephens and Young, 1989). The engines burned either diesel fuel or residual oil. The locomotive was used as a demonstrator and ran on several railroads, but was never placed in commercial service. Although the locomotive operated at a cost approximately 5% less than a diesel locomotive, the capital costs were not competitive. The locomotive was subsequently scrapped, but both turbines were sold for further operation. The left hand unit was sold to the city of Larned, Kansas as one of the early decentralized peak shaving plants and the right hand unit became the world's first intercooled steam injected plant driving an air compressor at Hopewell, Virginia. The first base-load service of the engine was as a pipeline gas booster pumper on The Mississippi River Fuel Company System in March 1949. This was a dual fuel application burning either oil or gas with the capability of changing to either fuel at full load, an industry first. This particular engine, the first combustion turbine in the world to achieve 100,000 operating hours, was retired after more than 150,000 hours of operation when the gas line was retired.

W21 combustor baskets were contained in individual pipe-like enclosures connecting compressor discharge to turbine inlet, as shown in Fig. 1, and bore little resemblance to the Westinghouse aviation gas turbine design which was an annular combustor, but it used the wall cooling arrangement (wiggle strips) from the aviation design. The wiggle strip concept of wall cooling was unique; it interposed a corrugated strip of metal between two cylindrical sections of combustor that permitted the injection of cooling air, is structurally superior to a single cylinder because the increased section modulus provides superior buckling characteristics, and proved to be a fundamental cooling feature that is still being used in current Westinghouse turbines. Also, casings were horizontally split (like steam turbines) instead of the axial stacking used by the jets. In addition, late production units introduced the CURVIC coupled rotor structure, as shown in Fig. 2, to align discs and transmit torque.

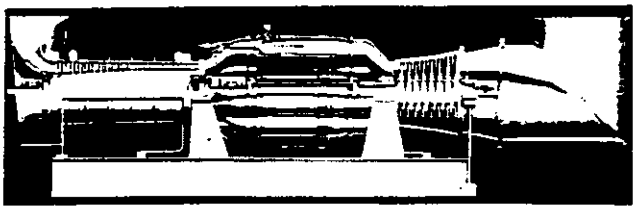

FIG. 1. CROSS SECTION OF W21 COMBUSTION TURBINE

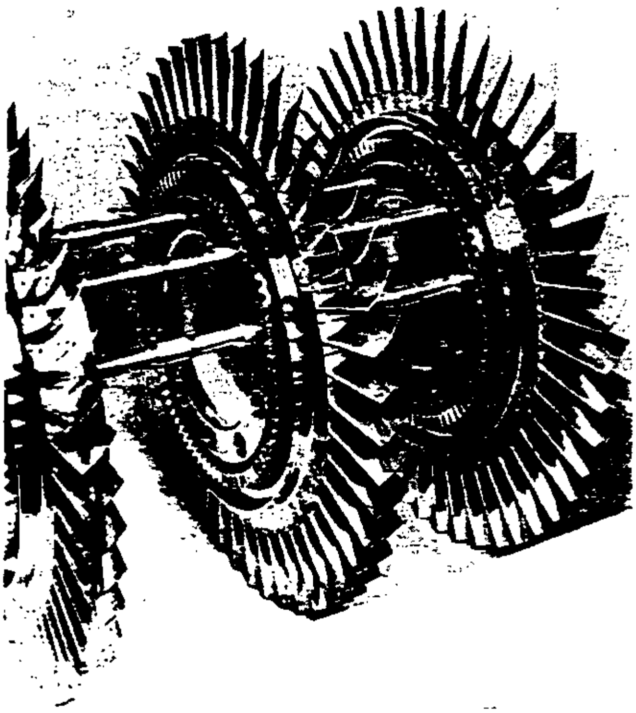

FIG. 2. CURVIC COUPLED TURBINE DISCS

Second Generation

In the late 1940s, Westinghouse formed a design group independent of the aircraft group to develop gas turbines for land based applications. Engineering studies indicated that with the high temperature materials then available a firing temperature of 1350°F (732°C) was feasible for base-load duty (100,000 hr life). In response to a market survey, the first engine developed was a 5000 kW single shaft, simple cycle engine. This engine had a design speed of 5740 rpm, an air flow of about 125 lb/s (57 kg/s) at a pressure ratio of 6.25.
The second generation engine design eliminated the undesirable center bearing and placed the combustion system inside a pressure vessel. Overall, this design had the following improvements:

- Elimination of center bearing removed a fire hazard and simplified shaft alignment.
- Compressor flow from the diffuser provided impingement and convective cooling of the transition duct.
- Flow into this plenum type enclosure attenuated flow disturbances to and from the compressor.
- The annular plenum effect provided damping of combustor pressure fluctuations (This became valuable in later years when limits of combustor stability were approached when using water injection for NOx control).
- Provided for improved cross-flame tube connections.
- Permitted the inclined position for the combustors and transitions, thus, facilitating the center bearing removal by shortening the span length.
- Combustors and cross-flame tubes completely encased in relatively cool compressor discharge air, thereby eliminating a multitude of pressure connections and potential fire hazard.

In addition to the combustion system changes, the 5000 kW (W81) combustion turbine, introduced in 1952, contained the following basic mechanical design features that are illustrated in Fig. 3:

- Individually replaceable turbine and compressor rotor blading without removal of the rotor.
- Inner shrouded diaphragm construction for turbine and compressor stationary blading to permit servicing with rotor in place.
- Support of the exhaust bearing housing with 6 tangential struts that rotate the housing during thermal transients and maintain shaft alignment without the need of cooling.

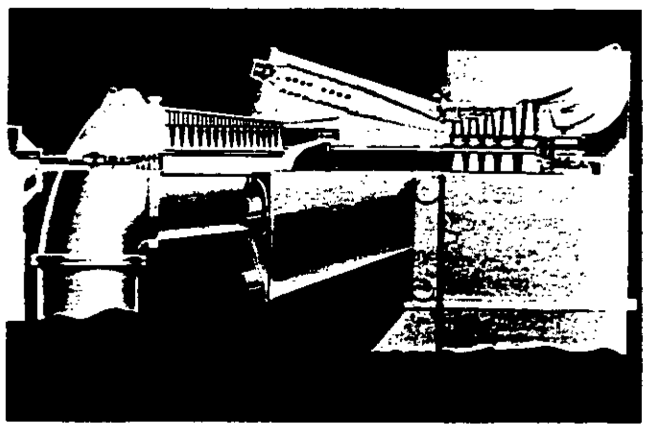

FIG. 3. CROSS SECTION OF W81 COMBUSTION TURBINE

The W81 16-stage compressor design was the result of extensive analytical and laboratory development. While the free-vortex design of the early compressor had the advantage of minimizing the effect of neglecting some terms in the Euler equations, the result was many stages at relatively low blade tip speeds. In addition, theoretical studies indicated that a design of 50% reaction with a flow coefficient of about 0.55 would give a stage of maximum efficiency. By setting up a pre-swirl entering the first rotating blade, the blade relative velocity could be significantly reduced and high reaction obtained. This would allow an increase in blade speed with the consequent reduction in the number of compressor stages required for a given pressure ratio. The vortex pattern selected was a compromise between little change in reaction radially and blade shock losses. This vortex pattern specified the mean stage absolute swirl constant and made the axial velocity relatively constant radially. A positive energy gradient was added from blade hub to tip in the first few stages, the energy addition was kept constant in the middle stages, and the gradient removed in the last stages.

Other design techniques used were to specify flow area blockage factors and to use NACA 65 blade sections on a circular arc mean line for the higher Mach number front stages and NACA (4-digit) sections elsewhere. The resulting design was successful except that the starting surge margin was inadequate; therefore, an interstage bleed was added. The average stage efficiency was about 89%.

The aerodynamic design of the 5-stage turbine was taken mainly from Westinghouse aircraft engine design practice using a free-vortex swirl distribution and constant work radially with a last stage designed for axial velocity outflow. A minimum reaction was maintained at the blade hub sections to prevent suction side separation. All blade sections were designed to provide smooth accelerating flow passages. The average stage efficiency was about 90% and the matching of the compressor and turbine flow characteristics was excellent. The engine produced 5700 kW at the generator terminals at a firing temperature of 1350°F (732°C) and a thermal efficiency of about 21%. The prototype engine, sold to West Texas Utilities, went in commercial service in 1952. This unit was the first utility application of a Westinghouse gas turbine.

Marketing studies indicated there were many differing applications for gas turbines from power generation, transportation, to mechanical drives. Engineering cycle studies were made to determine the best cycle arrangements for the various applications. A summary of those studies at a 1400°F (760°C) firing temperature is shown by Fig. 4. Component efficiencies, pressure losses, and heat exchanger effectiveness for these studies were state-of-the-art.

With the successful completion of the 5000 kW design, a line of mechanical drive engines using the same basic blading as the 5000 kW engine was designed. For pipeline pumping, a unit was designed with regeneration and a free power turbine (Stephens and Bruce, 1955). The regenerative engine produced about 5400 hp at the turbine coupling with an efficiency of approximately 30%. The cycle pressure ratio was 4.2, the compressor speed was 6000 rpm, and the firing temperature was 1350°F (732°C). The regenerative units did not perform in the field as expected, primarily because leakage and pressure loss were greater than expected. This same unit was also modified to be used without regeneration.

To make the engines more competitive with steam plants and other prime movers, it was necessary to use a more efficient cycle. As shown in Fig. 4, a gain of about 12% could be obtained by adding intercooling to the regenerative cycle resulting in 34% efficiency which made it competitive with small steam plants. A 15,000 kW engine was designed as a two shaft, intercooled, and regenerative engine with the load taken from the high pressure shaft. By taking the load from the 3600 rpm high pressure shaft, the low pressure (LP) shaft was made variable speed resulting in little drop off in turbine inlet temperature at part load and, therefore, excellent part load efficiency.

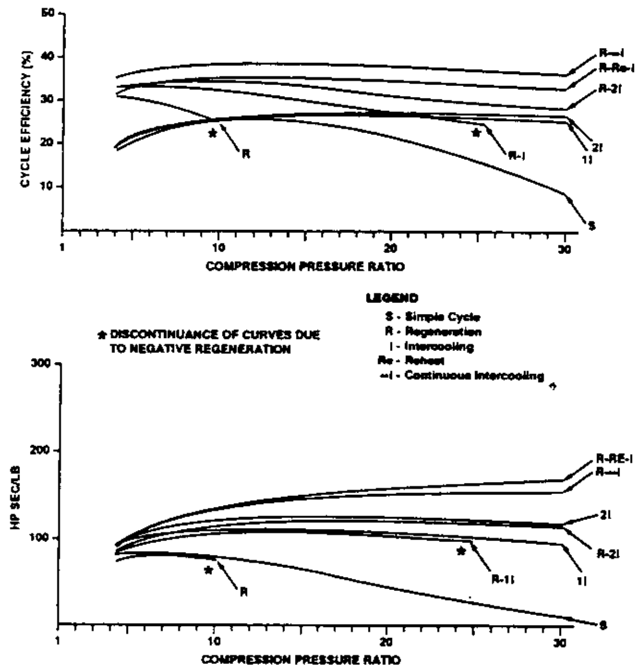

FIG. 4. SUMMARY OF ENGINEERING CYCLE STUDIES AT 1400°F

In 1954, prior to the cancellation of the 15,000 kW project, work was begun on a single shaft simple cycle 3600 rpm gas turbine. This engine was later to become the W201 gas turbine, the first direct drive (no gear) Westinghouse combustion turbine. In the W201, the aerodynamic design of the successful 5000 kW engine compressor was modified slightly by eliminating the energy gradient so that stages could be more easily added or removed from the basic design. In addition, lightly loaded stages for high efficiency were adopted for these engines. Design techniques were also improved by refining empirical factors. As high temperature materials advanced, the allowable turbine inlet temperature was increased, thus improving output and efficiency. By the end of the 1950s, firing temperatures had risen to 1450°F (788°C) at base load. Also, the W31 and the W121 at 3000 hp and 12,000 hp, respectively, both single shaft engines, were designed following the same philosophy developed in the W201.

In the middle 1950s, Westinghouse designed a blast furnace blower to supply 125,000 ft3/min (3540 m3/min)of air at about 2 1/2 atmospheres gauge pressure, and burning blast furnace gas at about 100 Btu/scf (3700 kJ/scm). The blower was a modification of the W201 in which the blast furnace air was extracted from the compressor discharge and the fuel gas supplied to the engine through a shaft driven fuel compressor. The fuel compressor was a modified compressor from the W31 engine. The engine and it's performance are shown in Figs. 5 and 6, respectively. Turbine blading, turbine discs, compressor discs, and recuperator were taken from the 15,000 kW unit. This was a very successful design from an engineering point of view, but the market disappeared when oxygen blown furnaces displaced the old air blown ones.

In the late 1950s, production for the direct drive W201 was initiated. The W201 was the forerunner of the W301 and the W501 line. Also, the W121 was improved to the W171 by eliminating the excess quote margin and increasing firing temperature to 1450F (788°C). In the early 1960s, a zero stage was added to the 14-stage

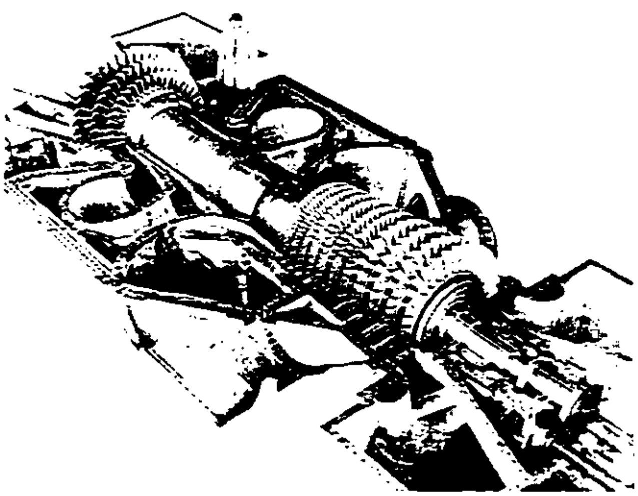

FIG. 5. BLAST FURNACE ENGINE

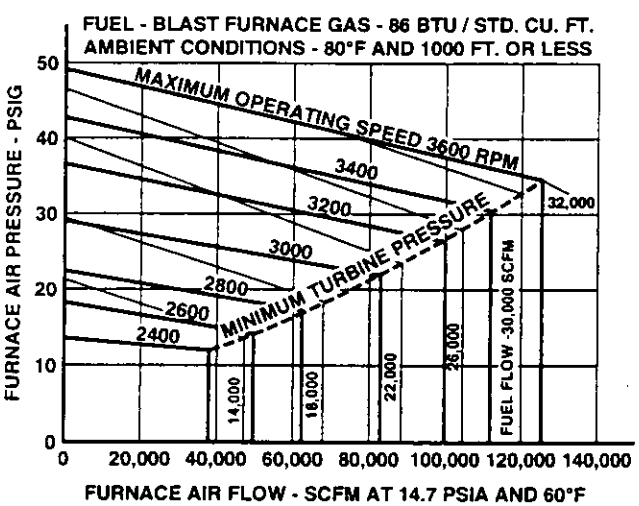

FIG. 6. PERFORMANCE OF BLAST FURNACE ENGINE

W171 compressor to create the W191 and the firing temperature of the W31 was increased to 1450F (788°C) to create the W41.

President Eisenhower's Administration started the "Atoms for Peace Program." An interesting outgrowth of this program was the "Marine Gas Cooled Reactors Program" (designated MGCR) that was expected to power a nuclear powered commercial ship. The power plant was to be a helium cooled reactor as the heat source, coupled with a closed cycled gas turbine. Westinghouse was selected to provide the turbomachinery for the power plant and the design proceeded to development testing and engine design. However, when the reactor development lagged, the program was canceled. The turbomachinery presented no severe aerodynamic problems and was actually much easier to design than air breathing machinery due to the fact that helium is a light monatomic gas and the speed of sound through it is very high. The engine was rated at 20,000 hp, at an expected efficiency of 31%.

Combined gas and steam plants had been developed as early as the locomotive engine, but the 1960s saw the development of the combined cycle. The combined cycle is an ideal thermodynamic solution to the thermal efficiency problem as it combines the best attributes of the Joule-Brayton cycle with that of the Rankine cycle; namely, the addition of heat energy at the high temperature of the gas turbine and the rejection of heat at the low temperature of the steam turbine. In 1967, a Westinghouse supercharged combustion turbine (W301), rated at 25 MW with a firing temperature of 1450F (788°C), was placed in operation at West Texas Utilities San Angelo's Power Station. Hot turbine exhaust gases were used in a heavy fuel, reheat boiler (1500 psi, 1000°F [10.3 MPa, 537°C]) which furnished steam to drive a 85 MW turbine (Cox, Henson, Johnson, 1967). The plant achieved an annual average efficiency of over 39%, the highest in the U.S. for a number of years. In 1990, this plant with over twenty-three years of successful operating experience, continued to demonstrate that good combined cycle efficiency over a broad load range can be obtained with a combustion turbine combined with a heavy fuel, reheat boiler (Stephens et al., 1990).

An interesting feature of this plant was that the forced draft fan of the steam power plant was used to supercharge the gas turbine. Not only did this feature increase the plant capability and efficiency, but the fan was used as a starting device to start the gas turbine by windmilling the compressor up to self-sustaining speed. In addition, the fan could supply the steam boiler while the gas turbine was out of service.

Third Generation

In November 1965, the "Northeast Blackout" ushered in the "Gas Turbine Age" as gas turbine manufacturers were swamped with peaking power plant orders. Development of gas turbines accelerated as technical staffs expanded. Blade and vane cooling, pioneered by the aircraft engine industry, became available to the industrial engine builders. In response to a need for larger, more efficient engines, Westinghouse developed the W251 and the W501 engines.

The 20 MW W251A, introduced in 1967, was a single shaft, two bearing, combustion turbine with reduction gear to accommodate either 60 Hz or 50 Hz for simple or combined cycle applications. Many basic design features proven reliable over many years were retained along with the two bearing, single shaft concept. Most noteworthy of these are the CURVIC coupled turbine discs, tangential exhaust bearing struts, blades and stationary vanes which are field removable with the rotor in place, and a compressor end drive. The evolution of the W251 is shown in Table 2.

The compressor of the W251A was taken from the W191, a rear stage added to allow an increase in engine pressure ratio, and a new 3-stage turbine designed with higher blade speeds to replace the old 5-stage turbine. The first stage turbine vane was cooled with compressor discharge air. The base load RIT was increased to 1575°F (857°C) from 1450°F (788°C) increasing the output power by 16% at constant heat rate. The first engine was placed in commercial peaking service at Detroit Edison in 1967. The W251A introduced the following improvements for the first time in a Westinghouse design:

- New 5-serration, minimum peak stress root design with root extensions to eliminate 3-D stress concentration.
- Precision cast turbine vane segments of cobalt-base material (X45) to eliminate structural weld concerns with the nickel-base welded diaphragm assemblies.
- Single turbine blade ring concept with roll out feature to provide service of turbine stationary parts with the rotor in place.
- Extensive use of turbine cooling including the row 1 vane segment, all three rows of turbine discs, and blade rings together with associated stationary flow path parts.
- Segmented ring segments over the rotor blade tips to allow free thermal expansion.
- Interstage seal housing supported from the vane segments on radially oriented keys to allow independent thermal growth of the seal housing to minimize running seal clearances.
- Use of cooled and filtered compressor discharge air for rotor cooling.
- Use of NiCrMoV turbine disc material.
- Compressor bleed extractions for selected cooling of turbine vanes and disc cavities to provide the lowest cost air for duty required.

The W251AA used the W251A compressor with an added front and aft stage to increase the airflow and pressure ratio while using the same turbine except improved row 1 vane cooling to permit a modest increase in RIT to 1640°F (893°C).

The W501 genealogy begins with the 30 MW W301 which began commercial operation in 1960. Unlike previous smaller, higher speed engines, the W301, like the W201, was directly connected to the generator without a gear.

Table 3 traces the development of the 501 family from the W501A to the current fifth generation 501F model. The W501A was designed to meet a 42 MW need of the Dow Chemical Company by utilizing

TABLE 2. W251 EVOLUTION

Engine	W191	W251A	W251AA	W251B	W251B2	W251B8	W251B10	W251B12
First Startup Date	1961	1967	1969	1971	1973	1978	1983	1990
Power, MW Class	18	20	26	31	34	36	40	48
Rotor Inlet Temp.,F	1450	1575	1640	1806	1806	1911	1985	2100
Air Flow, lb/s	270	270	353	353	353	353	345	372
Pressure Ratio	7	8	10	10	11	11	14	15
No. Comp. Stages	15	16	18	18	18	18	19	19
No. Turb. Stages	5	3	3	3	3	3	3	3
No. Cooled Rows	0	1	1	3	3	3	4	4
Exhaust Temp., F	777	850	850	930	898	962	941	990
Heat Rate, Btu/kWh; (ISO Gas)								
Simple Cycle	13,430	13,775	13,130	12,540	12,265	11,980	10,980	10,600
Combined Cycle	9,600	9,840	9,040	8,600	8,400	7,900	7,400	7,100

the proven, high performance, 15-stage W301 compressor, with an added front and aft stage to increase the air flow and pressure ratio, and a newly designed four-stage turbine to replace the old five-stage turbine. Dow required a cold end drive because their system design was a single train of combustion turbine, steam turbine, and generator with the steam turbine providing combustion turbine starting. Also, the hot end drive was undesirable because a flexible coupling was required. They also preferred two bearings which provided an excellent match with Westinghouse turbine design philosophy. In addition, the W501A introduced the following improvements:

- Variable inlet guide vane to provide exhaust temperature control on heat recovery applications and to improve starting characteristics.
- Individual blade rings with a roll-out feature to provide independent field service of each stator and seal system with rotor in place.
- Unique cooled, filtered rotor cooling concept utilizing "air separator" rotor component to confine rotor cooling air.
- Unique fore and aft blade seal plates to prevent cooling air leakage and service of blades with rotor in place.
- Four-pad, tilting pad bearings to eliminate any possibility of bearing instability from "oil whip" phenomenon.
- Compressor rotor utilizing a hollow shaft with shrunk on discs to reduce stresses and improve rotor dynamics.

The 60 MW W501AA in 1971 used a newly designed, higher flow, higher pressure ratio compressor with the W501A turbine at essentially the same RIT. Another major design change was a reduction in combustor shell diameter in order to facilitate shipping the engine fully assembled, a feature which has been retained on subsequent models. The combustor was set parallel to engine axis, resulting in an s-shaped transition duct. A secondary benefit of this system was a reduction in pattern factor or peak temperature due to increased mixing.

A significant improvement in engine performance was due to the new axial exhaust system. The axial exhaust system allowed waste heat recovery systems to be located directly downstream of the turbine exhaust, avoided high duct losses, and accommodated a longer and more efficient turbine diffuser.

Many engineering studies were made to determine the optimum cycle to minimize the cost of electricity. Some of the more promising ones (intercooled, multiple shafts, reheat, steam injected, water injected at various locations) were studied in great detail. In the final analysis, the simple cycle gas turbine combined with a steam bottoming cycle was the most widely accepted in the market. The missing link was higher firing temperatures. Fig. 7 is a summary of some of those studies. (The question of more efficient cycles is currently being re-visited in the advanced turbine systems program discussed later in the paper.)

As shown by Fig. 7, the combined cycle is superior to the regenerated cycle by a significant amount. The pressure ratio of the gas turbine that yields maximum specific power (power output per

TABLE 3. W501 EVOLUTION

Engine	W501A	W501AA	W501B	W501D	W501D5	W501D5	501F
First Startup Date	1968	1971	1973	1975	1981	1995	1993
Power, MW Class	42	60	80	95	107	118	160
Rotor Inlet Temp.,F	1615	1630	1819	2005	2070	2150	2300
Air Flow, lb/s	548	744	746	781	781	830	960
Pressure Ratio	7.5	10.5	11.2	12.6	14	14.8	14.6
No. Comp. Stages	17	17	17	19	19	19	16
No. Turb. Stages	4	4	4	4	4	4	4
No. Cooled Rows	1	1	3	4	4	5	6
Exhaust Temp., F	885	798	907	982	987	1006	1083
Heat Rate, Btu/kWh; (ISO Gas)							
Simple Cycle	12,600	11,600	11,180	10,925	10,270	10,040	9,590
Combined Cycle	9,000	7,990	7,350	7,025	6,950	6,900	6,600

pound of air flow) also gives optimum combined cycle efficiency. This is a most interesting conclusion and shows that for simple cycle gas turbine engine applications, a high pressure ratio engine is the economic choice, such as for aircraft propulsion. For the same firing temperature levels, the lower pressure ratio gas turbine engine is superior for combined cycle applications. Since the 1970s Westinghouse engines have been designed to be optimum for combined cycle applications and these engines are also excellent for peaking applications, where first cost dominates, since they yield the maximum power for a given amount of hardware. The figure also indicates the dominant role of turbine inlet temperature on both power output and combined cycle efficiency.

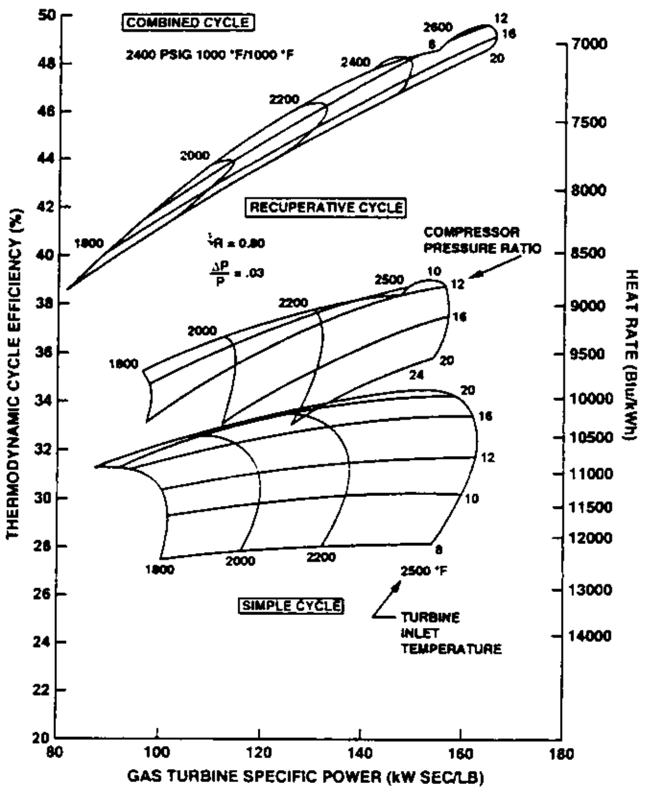

FIG. 7. SUMMARY OF COMBINED CYCLE STUDIES

Fourth Generation

In this era, the W251B used the same compressor as the W251AA, but with RIT increased to 1806°F (986°C). In addition to further improvement in row 1 vane cooling, the row 1 blade and row 2 vane segment were cooled for the first time in a Westinghouse design. At this point material changes were made such as the row 1 blade made as a casting for the first time with cooling passages integrally cast.

The W251B2 represented additional design improvements in the turbine that produced increased performance, while using the same W251AA compressor with no increase in RIT. The continuation of growth represented in this model consisted of adding a single vane with further improvements in cooling, restaggering of the turbine blade path for an increase in pressure ratio, improvements in cooling flow management and an improvement in row 2 vane cooling. The single vane was designed to permit service without removing the casing cover, an industry first.

The W251B8 used the W251AA compressor with performance improvement from an increase in RIT to 1911°F (1044°C) brought about by a redesigned combustion system for improved temperature profile into the turbine, and additional improvements in row 1 vane cooling. Also, the compressor used coated diaphragms for improved performance as well as to retard degradation from ingested contaminants, and rows 1 and 2 turbine vanes and blades were coated for corrosion protection.

The 80 MW W501B, in 1973, was a planned growth step of the W501 model with the principal change being an increase in RIT to 1819°F (993°C). In addition to increased row 1 vane cooling, cooling was added to the row 1 blade and row 2 vane and material was changed, as required, on other rows to compensate for higher gas temperatures. Provision was added for removal of combustors and transition pieces without a cover lift.

The 95 MW W501D was a continuation of the planned growth program for the W501 frame. A further increase in RIT to 2005°F (1096°C) was made possible by advances in turbine cooling technology. Two stages were added to the aft end of the W501B compressor for higher pressure ratio in order to optimize efficiency in combined cycle operations. Cooling was added to the row 2 turbine blade along with increased cooling in upstream stages and material changes as required.

The W501D5 achieved additional performance gains from improvements in component efficiencies, conservation of cooling air energy, and a modest increase in RIT. Compressor performance was improved by increasing the number of diaphragm seal points from two to four, by use of coated diaphragms with improved surface finish, and by restaggering airfoils to improve work distribution. Turbine performance was improved by reducing exit velocity and swirl, by reducing incidence losses, and by better stage work distribution. Stator cooling air was extracted from three compressor bleed points in order to use air at the lowest suitable pressure available. Improvement in detail design reduced internal cooling flow leakage by 12% compared to the earlier W501D. In addition, cooling of the first two vanes and blades. was improved. Other features provided improved reliability and availability by virtue of improvement in parts' life and easier maintenance and inspection such as a single row 1 vane. In the compressor, the aerofoil-shroud joint was redesigned for increased strength in key diaphragm stages. The combustor cooling technique using the standard "wiggle strip" corrugation was improved by the extended lip, as shown in Fig. 8. This extension improved the tenacity of the convective heat transfer coefficient and reduced downstream temperatures by 200°F (111°C) (Scalzo et al., 1983).

The last engine models of this generation were the W251B 9/10 and the W501D5. Both engines used the same basic compressor, the W251 compressor being a 2/3 scale of the W501. In combined cycle applications the engine's potential efficiency approaches 50%. The power capability of these engines are more than 2 1/2 times their

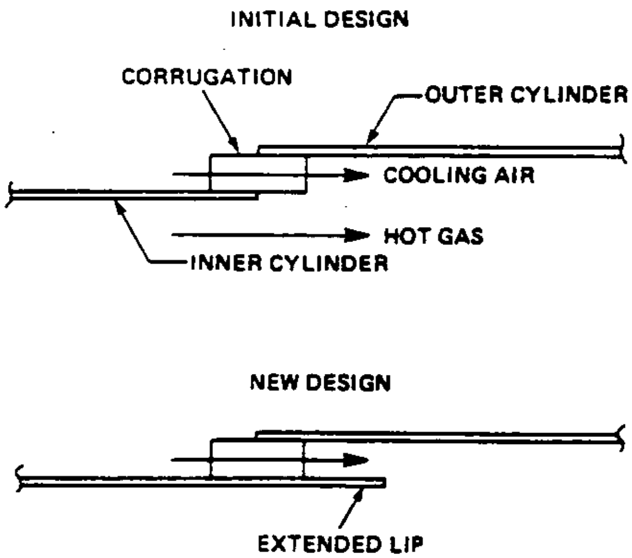

FIG. 8. EXTENDED LIP IMPROVED COMBUSTOR COOLING

third generation antecedents and the efficiency was increased by about 1/3.

The last model engines of this generation utilized streamline curvature calculation methods to solve the Euler equations, with energy loss models developed from a very large experimental data base. The result was compressors with average stage efficiencies approaching 93% and cooled turbines with excellent characteristics.

The power rating of a gas turbine engine is limited by the amount of flow that can be passed by the last stage blade row of the turbine component. Stress capability of the last row turbine blade material at operating temperature and aeroelastic blade limits determines the flow area of the last stage, while leaving axial velocity and density of the exhaust gases, in addition to flow area, determines engine flow. The next generation of engines, the F series and beyond, will be flow limited designs.

Fifth Generation

The fifth generation of Westinghouse gas turbine engines brings us to the present. These engines are designed with the aid of complete 3-D computer codes some of which solve the Navier-Stokes equations. The 501F at 160 MW and 2300°F (1260°C) firing level approaches 55% thermal efficiency in combined cycle applications.

The 501F, shown in Fig. 9, is a 3600 rpm heavy-duty combustion turbine designed to serve the 60-Hz power generation needs for utility and industrial service in the 1990s. It was jointly developed by Westinghouse and Mitsubishi Heavy Industries, Ltd. (Scalzo et al., 1989). Designed for both simple and combined cycle applications, it will operate on all conventional combustion turbine fuels as well as with coal-derived low-Btu gas produced in an integrated gasification combined cycle (IGCC) power plant.

The 501F rotor is of bolted construction supported by two, two-element tilting-pad bearings for load carrying and an upper half fixed bearing. This provides inherent stability of the tilting pad with the

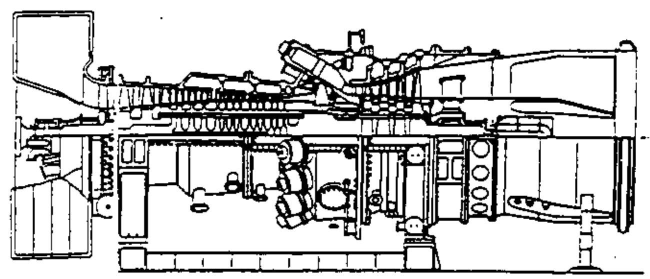

FIG. 9. GENERAL CONFIGURATION OF 501F COMBUSTION TURBINE

reliability of the plain bearing, thus eliminating the top pad fluttering problem that has led to local babbitt fatigue distress. The thrust bearing is a double-acting Kingsbury thrust bearing that uses leading edge groove (LEG) lubrication system.

The compressor rotor is comprised of a number of elements, spigotted and bolted together by 12 through bolts. The turbine rotor section is made up of disks bolted together by 12 through bolts and using CURVIC clutches. This turbine rotor design has amassed over 40 million hours of reliable service in all sizes of combustion turbines. The compressor is a 16-stage axial flow design of 14 pressure ratio that is based on the highly successful W501D5 compressor. A four-stage turbine was selected to maintain moderate aerodynamic loadings even at the increased firing temperature.

Blade rings have been added in the compressor for stages 7 through 16. Similar to those used in the turbine, they have a high thermal response independent of the outer casing, can be aligned concentric to the rotor to prevent blade rubs and minimize clearance, and therefore maximize performance as well as enhance maintainability of stationary parts.

Flow and pressure coefficients of the 501F compressor have been kept similar to the D5 compressor by increasing the mean diameter of the stages to accommodate the 20% increase in flow. In addition, the rear stages of the new compressor have larger diameters to help balance spindle thrust. Interstage bleeds are used for starting and for supplying cooling air to the turbine stationary blading and interstage cooling system. Rotor blades are double circular arc designs in the first four stages. The stators and all other rotor blades are conventional W65 airfoil sections.

The design of the 501F turbine has maintained moderate aerodynamic loadings in spite of the increased inlet temperature by choosing a four-stage turbine with higher peripheral speed compared to the W501D5. The 1st and 2nd stage blades are the free-standing type, while the 3rd and 4th stages utilize integral "Z" tip shrouds. The use of a shrouded system is a departure from past design practice on the 501 series, but has been in use on the W352.

The 1st turbine stationary row consists of precision-cast, single-vane segments. There are precision-cast, two-vane segments in the second turbine station row while the 3rd and 4th turbine stationary rows are precision-cast vane segments three-vane and four-vane segments, respectively.

Cooling circuits for the turbine section, displayed in Fig. 10, are similar to those used on the W501D5. Rotor cooling air is provided by compressor discharge air extracted from the combustor shell. This air is externally cooled and filtered before returning to the torque

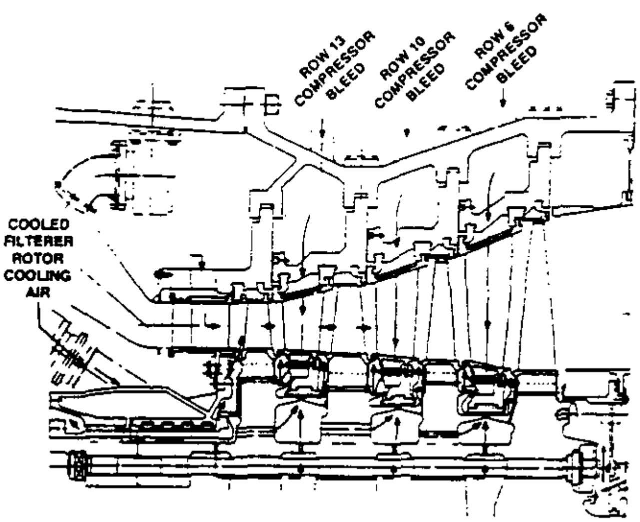

FIG. 10. COOLING CIRCUITS FOR THE 501F COMBUSTION TURBINE

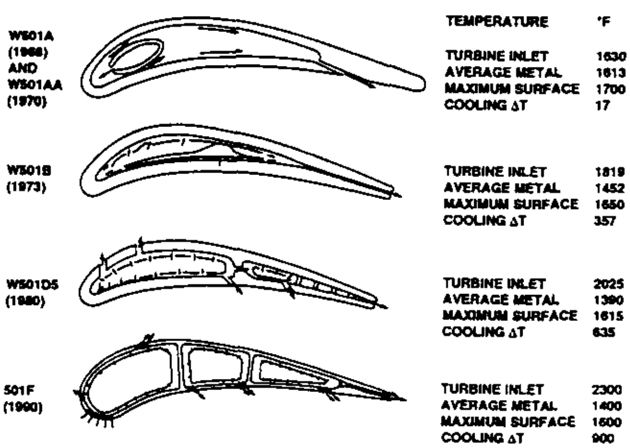

FIG. 11. EVOLUTION OF COOLING TECHNOLOGY

tube casing for seal air supply and for cooling of the turbine disks as well as the first, second, and third-stage turbine rotor blades. The row 1 vane cooling design is shown in Fig. 11. This highly effective configuration, which evolved directly from the W501D5 design utilizes state-of-the-art concepts with three impingements inserts in combination with an array of film cooling exits and a trailing edge pin fin system. The first stage blade is cooled by a combination of film cooling and convection techniques via multipass, turbulated, serpentine passages, with pin fin cooling in the trailing edge exit slots. The cooling circuitry is shown in Fig. 12.

The fifth generation also produced improvements in the W251B10 and the W501D5 models. Introduced in 1990, the W251B12 improvements included modest increases in airflow, pressure ratio, and firing temperature. To accommodate these changes, double circular arc compressor blade profiles were used for the first two blade rows, compressor stages were restaggered, and cooling was improved for the first turbine vane and first rotor blade. Current rating is 48

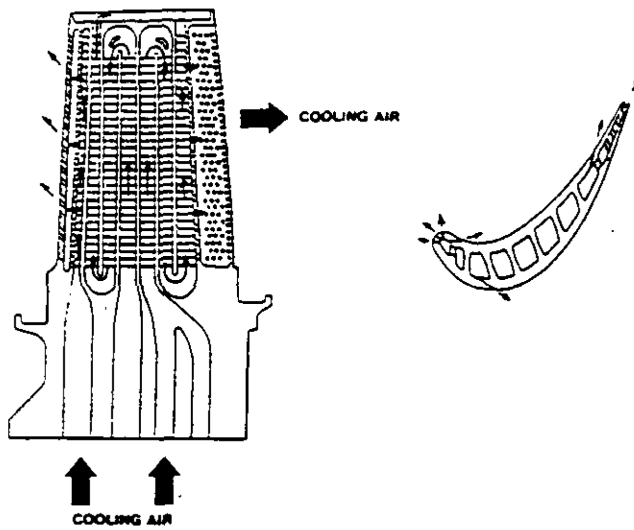

FIG. 12. COOLING CIRCUITRY OF 1ST STAGE BLADE USED IN 501F

MW at a simple cycle efficiency of 32% (Diakunchak, 1989). The improved W501D5, to be introduced in 1995, also will include modest increases in airflow, pressure ratio, and firing temperature. Compressor changes were scaled from the W251B12 and cooling was improved for the first three vanes and the first two rotor blades. In addition, the increase in airflow required that the row 4 blade be changed to a "Z" tip shroud similar to the 501F design. Rating of the improved W501D5 will be 118 MW at a simple cycle efficiency of 34%.

COOLING EVOLUTION

The evolution of the W501 and the W251 was highly dependent upon improvements in cooling technology as shown in Fig. 13. Without cooling, the maximum RIT would be under 1800F (982C). Fig. 11 depicts this evolution for row 1 vane, from the early W501A in 1968 to the latest highly sophisticated cooled design used in the current 501F. Similarly, rotor blade cooling has evolved from spanwise holes of the 501B to a sophisticated multi-pass, turbulated, pin-fin, film cooled design of the 501F. These advances in cooling technology were beyond the published state-of-the-art, hence, a comprehensive development program was required to verify the latest blade and vane cooling designs and to provide a broad base for future designs.

For example, parametric design information on various vane internal cooling features was generated in atmospheric pressure rigs, with the geometry scaled up to effect Reynolds number simulation. (Scalzo, Holden and Howard, 1981). Tests were run over a wide range of test parameters using basic test rigs such as the leading edge rig shown in Fig. 14. The insert nose in the leading edge heat transfer rig can be replaced so as to vary target distance, radius, and hole array. Actual leading edge cooling geometries have been tested as well as a range of geometries for the generation of parametric design information. In addition to atmospheric model tests, design verification tests were run on production vane segments at full operating temperature and

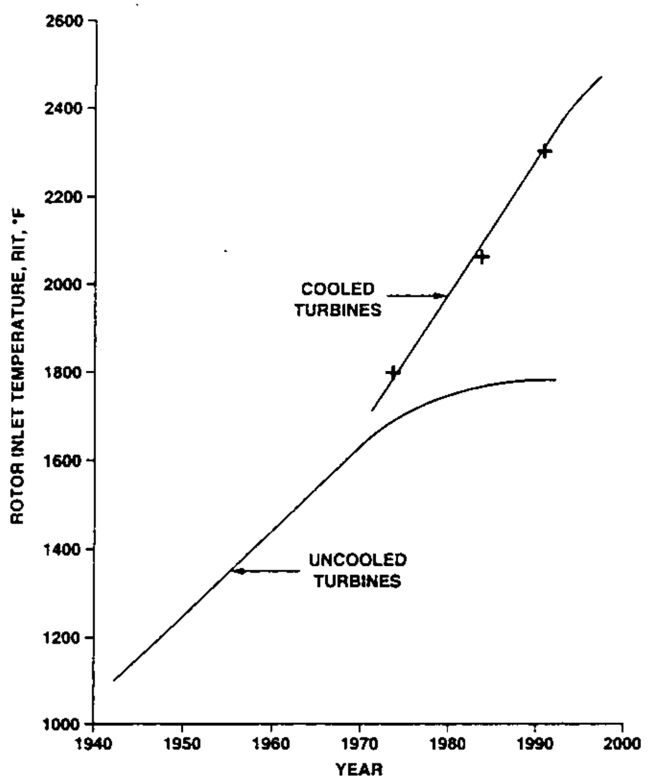

FIG. 13. FIRING TEMPERATURE TREND

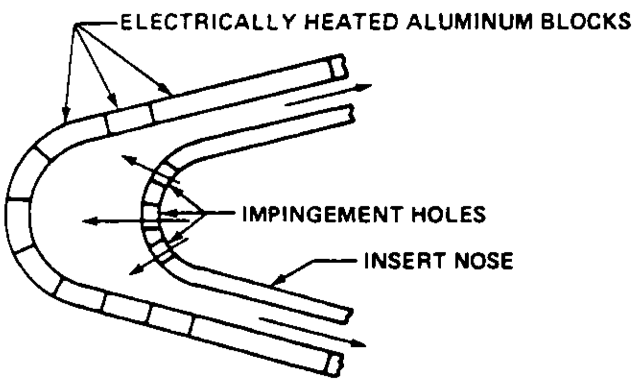

FIG. 14. INSERT NOSE IN THE LEADING EDGE HEAT TRANSFER RIG

pressure in a full-scale vane cooling rig.

Final verification of blade and vane temperatures were obtained from a fully instrumented full-load engine tests utilizing over 1400 pressure and temperature sensors. Comparisons were made to rig data to update the technology data base. Rotating blades and discs were monitored via telemetry (Scalzo, Allen and Antos, 1986; Gabriel and Donato, 1986).

MATERIAL EVOLUTION

In the early years (1950s to 1960s) materials were selected from available in-house steam turbine and jet engine experiences. However, since aero engines enjoy a pristine environment, many of these materials were not the best choice for the more corrosive fuels and operating environments of the land based gas turbine. Also, inspection intervals for aero components are more frequent as dictated by governmental regulations, while intervals in combustion turbines are being driven up to 24,000 hours by commercial considerations. For these and other reasons, material selection criteria, problems encountered, and developed solutions, would fill several volumes and are beyond the scope of this paper; however, a few pertinent comments will be made.

Compressor Area

Steam turbine and aero engine experience at the time of the initiation of the Westinghouse combustion turbine (1948 with the W21) dictated using a 12% chrome material for rotating and stationary blading because of its high strength, good corrosion resistance, and superior damping characteristics. This rationale is still valid today with standard strength 12% chrome material the choice for all compressor rotating blades and diaphragms with the exception of first two rotating blades in the 501F which use 17-4 PH, a 17% precipitation hardened stainless steel. Sermetel 5380DP[4] coating is currently used in the compressor to retard corrosion. To inhibit fretting, CuNiIn coating is used on blade roots. Compressor rotor materials have essentially retained the low-alloy steel materials based on steam turbine experience.

Combustor Area

The 25-20 series and 18-8 series stainless steel materials were used in early engines, however, Hastelloy X[5], a high temperature nickel-base alloy, became the standard for combustor baskets and transition ducts in the late 1960s. This material has been highly successful when kept below 1550°F (843°C) metal temperatures. For turbines with the increased firing temperatures used in the 1980s and later, the transition duct material was changed to IN617[6]; however, the standard combustor basket material is still Hastelloy X. IN617 provided a more stable higher strength material that required no pre or post heat treatment as was required by some sheet alloy candidates.

Turbine Stationary

Early designs utilized welded structures in AISI 310, a 25-20 austenitic stainless steel that had excellent resistance both to corrosion and to oxidation at elevated temperatures, but had limited strength capabilities.

A change to the higher strength, nickel-base, IN713 used successfully by aero engines, met with unsatisfactory results because

[4] Sermetech International Incorporated
[5] Hayes International
[6] International Nickel Company

of a lack of oxidation/corrosion resistance. A change to another precision cast nickel-base alloy, U500[7], resolved the corrosion concern, but the integrity of the welded structure was limited.

In 1967, Westinghouse introduced precision cast, colbalt-base, X45 turbine vane segments. This material was a modification of Haynes Stellite 31 used for years in aero engine turbine blades. It was the standard for precision cast turbine vane segments up to 1975, when the W501D was introduced with Westinghouse's ECY768 material in row 1. ECY768, is a cobalt-base alloy with higher creep strength and oxidation/corrosion resistance than standard X45 material. This improved cobalt-base alloy has become the standard for all row 1 vane segments and some selected row 2 and row 3 vane segments. Beginning in about 1975, diffusion and overlay coatings have been used selectively to enhance oxidation/corrosion resistance, as dictated by application.

Turbine Rotating

Various forged nickel-base alloys, and even 12% chrome materials, were used for early combustion turbine blades. Today nickel-base alloy Inconel X-750 remains the choice for many last row blades. Increases in firing temperatures necessitated changing to the higher strength Udimet nickel-base alloys in the mid 1970s. U520 forgings and cast U500 become the standard materials for front turbine stages. (All turbine blades are free-standing except for rows 3 and 4 of the 501F and row 4 of the improved W501D5.)

The 1980s and 1990s saw a move from forgings to higher strength cast IN738 material for front end blading because the complex cooling configurations could not be produced in machined, forged blades. As with the turbine stationary, diffusion and overlay coatings have been used for rotor blades. Future advanced engines will see a move to directionally solidified (DS) and to single crystal (SC) blading together with advanced coatings.

Turbine Discs

Early combustion turbine discs used a variety of materials including AISI 422, 19-9, and Discalloy 24 (Westinghouse-developed, austenitic alloy similar to A286). Selected cooling of the disc rims was required to prevent creep and notch sensitivity type concerns. To avoid these concerns, Westinghouse developed, in the mid 1960s, a unique cooling system for turbine rotors. High temperature, high pressure compressor discharge air is taken outside the engine where it is cooled and filtered before returning to the turbine rotor. This cooled, filtered air provides the rotor with a blanket of protection from hot blade path gases and supplies filtered cooling air to all disc rims as well as cooled airfoils. Filtration eliminates excessive contaminants that could block critical, intricate cooling passages of today's advanced combustion turbines. A reliable steam turbine material, NiCrMoV, was selected because of its high strength and outstanding fracture toughness. When cooled below 750°F (400°C), this material has infinite life because it is below the creep and aging thresholds. Also, its outstanding fracture toughness accommodated standard ultrasonic qualification and, therefore, did not require overspeed spin pits to qualify the forging as is the case for some materials. Since the mid 1960s, NiCrMoV discs integrated into the Westinghouse filtered, cooled air cooling system have generated over 20 million hours of problem free operation.

[7] Special Metals Company

MECHANICAL EVOLUTION

The evolutionary Westinghouse design philosophy has resulted in operational reliabilities of over 99% and unit availabilities of over 95%. Many of the basic design features have stood the test of over four decades of design evolution. These design innovations were identified in previous sections of the paper; for example, cooled, filtered turbine cooling air with NiCrMoV turbine discs. However, several will be expanded in this section; i.e., single row 1 vane, integral compressor vanes, and minimum peak stress turbine blade roots.

Single Row 1 Vanes

In the early 1970s, the row 1 vane segment of the W251B2 was introduced as a single vane that could be serviced along with baskets and transitions through the combustor basket openings with the casing cover and enclosure roof in place. Other advantages of this design were structural thermal stress at 25 to 50% of a three-vane structure and improved casting quality. (Hultgren, 1981 and Scalzo et al., 1988). Entering a log/log plot displaying the low cycle fatigue (LCF) of a nickel base alloy or cobalt base alloy, LCF is increased by over a factor of 10 when stress is reduced 50%. In addition, a single vane facilitates higher quality castings compared to multi-vane castings. For advanced turbines, the inner shroud is supported from the inner casing to limit axial and circumferential displacements, thus, controlling flow angles.

Compressor Integral Vane

Selected diaphragm rows on current product models incorporate an integral airfoil/shroud design with four point seals on a removable seal carrier as shown in Fig. 15. Critical structural weld joints are eliminated by use of the integral structure. Also, the airfoil thickness/chord ratio is tapered to produce an optimized aero/mechan-

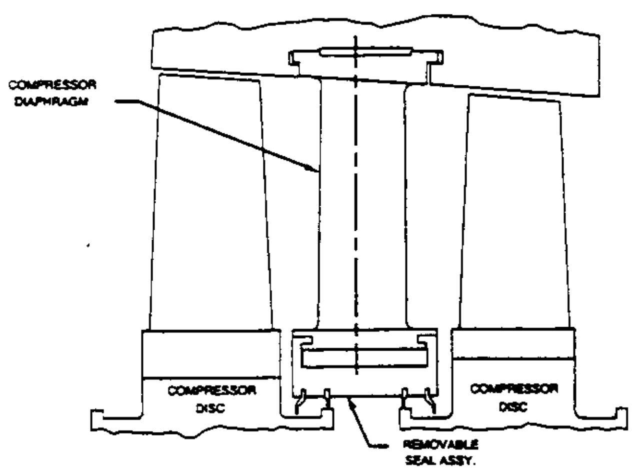

FIG. 15. INTEGRAL DIAPHRAGM DESIGN

ical design; i.e., high at the tip required mechanically and acceptable aerodynamically, and low at the hub, required aerodynamically and acceptable mechanically. Endurance strength of the structure is equal to base-metal properties as shown in Fig. 16. This design also provides for service of compressor diaphragms with rotor in place, as in past designs.

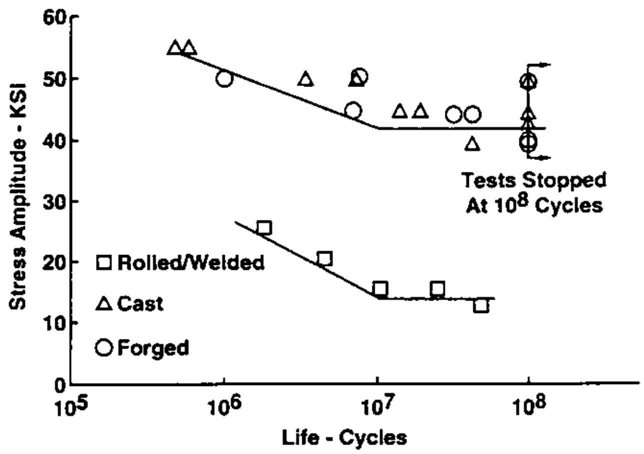

FIG. 16. ENDURANCE STRENGTH OF DIAPHRAGM DESIGNS

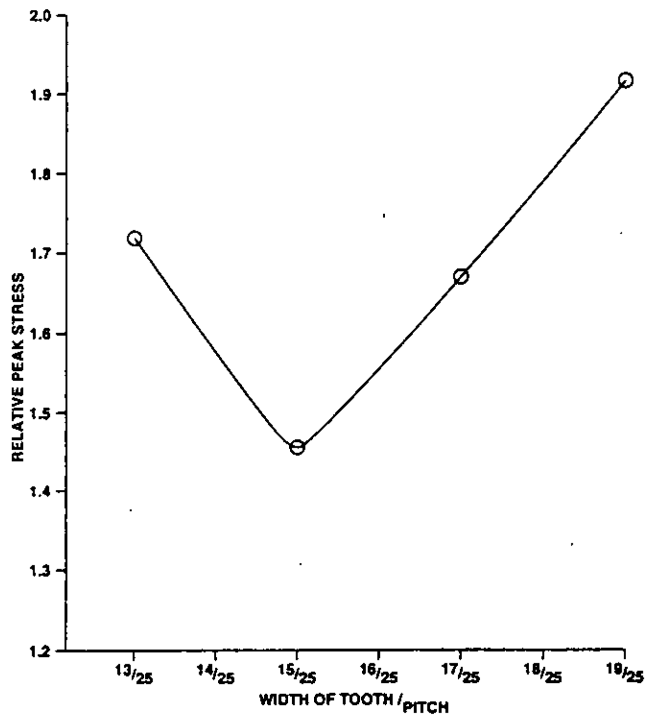

FIG. 17. STRESS VERSUS WIDTH OF TOOTH/PITCH

Turbine Blade Roots

In 1963, one of the top priority design goals was to develop an improved turbine blade root more suitable for higher firing temperatures and increased dynamic loadings expected in the third generation designs. Earlier designs were simply scaled from existing steam turbine and aero technology sources. There was also a desire to incorporate the new root design using NiCrMoV material as described in the preceeding materials section. Three basic objectives were identified; i.e., provide a thermal barrier, eliminate 3-D stress concentration, and minimize peak stresses. Using an extended root or extension isolates the root from flow path temperatures, thus providing a thermal conductive barrier. Also, it eliminates 3-D stress concentration resulting from the transfer of bending or dynamic moments from an airfoil shape to the root neck (Scalzo, 1992).

Minimizing peak stresses required the knowledge of 2-D stress distribution in irregular shapes. Since there were no finite element analysis programs available in the early 1960s, a general method for the determination of 2-D stress distribution in irregular shapes subjected to body and surface forces was developed using the finite difference method to represent the governing fourth order partial differential equation together with suitable boundary conditions. A mainframe computer was used to solve the multitude of equations using the over relaxation technique (Scalzo, 1963). Parametric analyses similar to Fig. 17, developed to determine the optimum tooth shape for minimum peak stress, resulted in Fig. 18. This was used to establish the root configuration for the W251A combustion turbine shipped in 1967, and has been used successfully for every Westinghouse combustion turbine since that time.

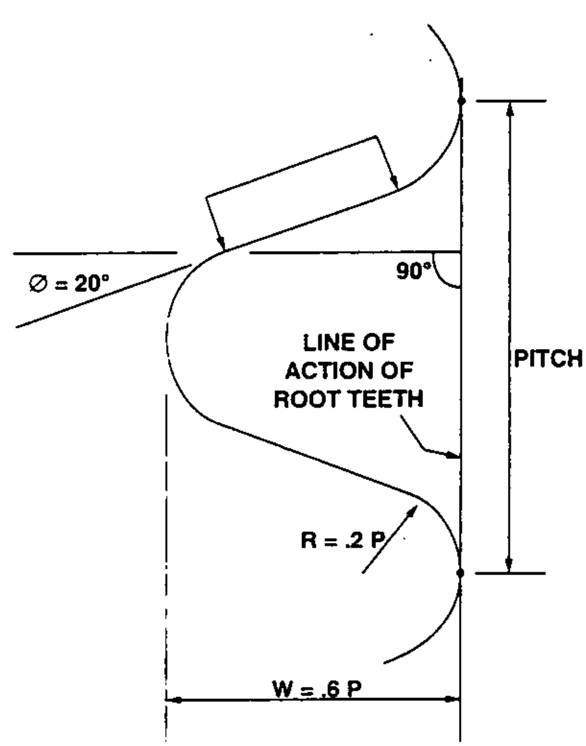

FIG. 18. ROOT TOOTH PROPORTIONS FOR MINIMUM PEAK STRESS UNDER DIRECT LOAD

COMBUSTION DESIGN EVOLUTION

Overall Geometry

In selecting the can-annular design, the W21 designers drew on the early experimental combustor can experience of the jet engine group even though the jets eventually selected an annular combustor. Sets of holes placed at the upstream end provided primary air sufficient for stoichiometric burning and another set of holes placed some distance downstream provided dilution air, bringing the stream temperature down to the required burner outlet temperature (BOT). (Note that BOT is the same value as the "firing temperature" used in the first and second generation.)

All engines have used can-annular combustors except for a silo type combustor installed in the blast furnace unit. The rationale that supports the can-annular design is still valid after 45 years. Advantages of this concept are:

- Shipped fully assembled with engine, thus reducing field cost.
- Facilitated full-scale laboratory testing.
- Can be developed to provide a preferential radial temperature profile, thus satisfying rotating blade requirements.
- Provides a lower "hot spot" by control of circumferential temperature gradients to improve turbine vane life.
- Generates a discrete harmonic content, thus minimizing blade vibratory design requirements.

Fuel Injection

The basic concept of fuel injection and ignition has been maintained from the W21 up to present engines, with the exception of the new advanced low NOx designs. Key elements are a pressure atomizing liquid fuel nozzle with a concentric atomizing air-assist used for starting. Gas fuel is injected via a concentric set of orifices. To prevent undesirable combustor noise, the orientation of fuel injection vectors must be compatible with the essential recirculation stabilization pattern required to anchor the flame front (Scalzo et. al., 1990). In models built after the mid 1970s, mechanical flow dividers were used to meter liquid fuel equally to each combustor. When required for emissions control, water is injected through the air atomizing ports of the liquid nozzle or pre-mixed, steam is injected either pre-mixed with gaseous fuel or via a separate set of orifices.

Ignition

The ignition scheme used on the earliest combustor designs was a spring loaded spark plug in each can which retracted as combustor pressure built-up to operating conditions; and cross-flame tubes provided ignition between cans. This belt and suspenders approach was used because, with the large distance between adjacent cans, cross-firing was not reliable. In later engines, with combustors being nested closer together (W251A model and later), cross firing was more effective and spark ignition was provided for only two cans (for redundancy) and cross-firing lit the others. In early designs the cross-firing occurred through tubes attached to each combustor at the combustor flame zone, with one tube fitting inside the other in a sliding fit. In more recent models this arrangement has been replaced by hard-connected flexible tubing. The retractable spark plug approach remained basically the same with improvements made to the spark plug, the retracting piston, and the electrical power to the plug.

In the earlier engines, flame-on indication was provided by thermocouples protruding into each transition section, which recorded BOT as well as other control input functions. When the BOT increased to that required for the W251AA engine and succeeding models, this arrangement had to be abandoned due to the impact of the higher temperature environment on sensor life. The flame-on detection is now by an optical sensor which views the combustor flame zone through a port. Other control supervisory functions are taken up by thermocouples placed in the blade path downstream of the last turbine blade row. These exhaust blade path thermocouples also monitor each combustor to protect rotating blades against harmful harmonies (Scalzo, 1992).

Special Fuels

Gas turbines have the capability to burn a wide variety of fuels. Combustor and fuel injector design modifications have been able to accommodate fuels which have included: natural gas, blast furnace gas, coal gas, distillate fuel, residual fuel, crude oil, methanol, propane, coal derived liquids, and shale oil (Pillsbury et al., 1974, 1978 and 1979; Seglem and DeCorso, 1980). Burning fuels which have potentially corrosive elements requires that fuel contaminant specifications be set (Wenglarz and Menguturk, 1981). The Westinghouse Research Laboratory was equipped with corrosion evaluation test rigs where actual fuels were burned making accurate determinations of fuel element corrosion levels on the combustor and other turbine hot parts. This work at the Research Laboratory was key to setting the ASTM D2880 and ASME B-133 fuel contaminant specifications now in use industry wide (DeCorso et al., 1971 and Hussey et al., 1973). Use of these various fuels in the future will now be subject to more stringent emission regulations.

Emissions

The first emission concerns to impact combustor designers began with exhaust visibility (smoke) in 1966. As a result, the W171 and W191 combustors were modified to produce a smokeless design (DeCorso, Hussey and Ambrose, 1967). In the early 1970s, EPA began to develop regulations which dealt with NOx emission levels. In local areas of California more stringent emission regulations were passed. In the Los Angeles Air Pollution Control District (LAAPCD) regulations (rule 67) called for particulate emissions of no more than 10 lb/hr (4.54 kg/hr) regardless of plant size, and a NOx regulation which required 32 ppmv from a W501 engine. Particulate levels were minimized by adhering to established fuel specifications (Carl, Obidinski and Jersey, 1975) and limiting sulfur content in the fuel, since the Los Angeles particulate measurement technique counted sulfur compounds. (Many current regulations require NOx to be 25ppmv or lower, regardless of engine size.)

Working in cooperation with the Westinghouse Research Laboratory, a theoretical understanding of NOx formation was gained and a NOx formation model was developed and verified by laboratory and field testing. This model proved to be invaluable in predicting the effect of combustor operating parameters on NOx formation. It was capable of giving NOx emission effects due to combustor operating parameters; and, it also accurately predicted the effect on NOx emission of various fuels, fuel bound nitrogen, water injection, and steam injection (Hung, 1974 and Vermes, 1974). With the NOx model as a guide, combustion laboratory test time was greatly reduced and valuable insight into potential low NOx approaches was provided.

In order to meet the 32 ppm NOx regulation for W501 turbines, water injection for NOx control was designed, tested and proven in the laboratory, and then in a full-scale field test (Ambrose and Obidinski, 1972), in what is believed to be the first application of water injection for NOx control.. Subsequent design and testing also confirmed the use of steam for NOx control, with steam being less effective, pound for pound, than water.

Dry Low NOx

Dry low NOx development schemes were explored in the three areas; i.e., premixed lean combustion, rich-lean combustion, and catalytic combustion. Small scale Research Laboratory tests had confirmed that very low NOx levels were attainable by premixing the air and fuel. Premixing means in this context that the air and fuel are mixed on a molecular level. A practical scheme to use premixing was devised and given the name of "hybrid combustor" because it utilized the dual approach of a premixed fuel/air stream and a pilot flame zone which burned in the diffusion flame mode. The term "hybrid combustor" is now commonly used in the industry to refer to this type of low NOx combustor. An example of an early design which was tested is shown in Fig. 19 (Mumford, Hung and Singh, 1977). This design was the basis for the lean, pre-mix combustor design used in the MW701D combustion turbine (Aoyama and Mandai, 1984). Advanced versions of this design are now operating in the 701F (50Hz scale of the 501F) engines in Japan with NOx emissions less than 25 ppmv. This design is currently being developed further to produce less than 15 ppmv by the late 1990s.

Another approach to NOx reduction utilized the rich-lean burn principle, which did not require premixing and was effective for cases where the fuel contained fuel bound nitrogen (FBN). The premixing or water/steam injection approach is defeated by the presence of significant amounts of FBN. In rich-lean burning, the burning is

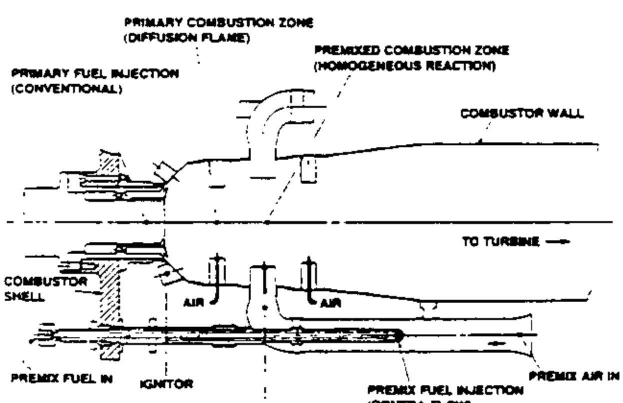

FIG. 19. HYBRID BURNER

staged to occur in a fuel-rich zone where the oxygen deficiency prevents NOx formation; in a second lean zone which follows the rich zone, burning of the products from the rich zone are completed at a temperature low enough to prevent NOx formation. The multi-annular swirl burner (MASB) was developed which had the capability of operating as a rich-lean burner, as well as in the lean burn mode. This combustor was successfully operated at the Waltz Mill, PA coal gas test site (Lew, DeCorso et al., 1981) and is now under development as a topping combustor for an advanced PFB cycle program (Garland and Pillsbury, 1992).

The prospects of ultra low NOx using catalytic combustion was first evaluated in 1971. Tests run at the Research Center confirmed that virtually zero emissions could be attained and that very uniform temperature patterns were possible (DeCorso, Mumford et al., 1977). A very active program included testing of catalytic combustors at full-scale conditions (Hung, Dickson and DeCorso, 1978; Scheihing et al., 1982). Although NOx was in single digits, the mechanical integrity of the substrate was very unsatisfactory. After the oil embargo sharply curtailed market activity, development effort in low emissions tapered off through the early 1980s; however, activity has surged again in the 1990s with advanced designed catalytic systems currently under review.

CLEAN COAL TECHNOLOGY

Combustor design and testing first took place in the mid 1970s under U.S. Government sponsorship for a projected coal gas plant for Public Service of Indiana. Full-scale test facilities were built in which combustors modified for coal gas use were tested with actual coal gas compositions encompassing air blow gasifier and oxygen blown gasifier gas (Pillsbury, 1974). This development supported the joint Westinghouse/Dow Chemical Co. program to burn medium Btu (239 Btu/scf [8890 kJ/scm]) coal gas; first, in a W191 trial engine; and, later in W501D5 turbines at the Dow Chemical Co. IGCC plant in Plaquemine, LA (Hendry and Pillsbury, 1987). These two 104 MW units are operating in a Dow Chemical developed oxygen-blown integrated gasification combined cycle (IGCC) that includes a cold gas cleanup system (Morrison and Pillsbury, 1989). To date, these combustion turbines have operated with average availabilities in excess of 95% (Geoffroy and Amos, 1991).

Most commercial size gasification projects have used high purity oxygen rather than air as the oxidant for the gasifiers. Recent IGCC evaluations have looked at using the combustion turbine air compressor to supply air for the air separation unit. Typically, this air stream is sent to a high pressure air separator unit which produces oxygen for gasification and high pressure nitrogen for combustion turbine NOx control. The diluent nitrogen lowers the flame temperature and, therefore, lowers the NOx.

Other concepts being developed at Westinghouse include incorporating combustion turbines in a coal-fueled combined cycle for second generation pressurized fluidized bed (PFB), and direct coal-fired (DCF) combined cycle configurations. IGCC, PFB and DCF concepts require a fuel gas cleanup system to remove particulates, sulfur, and alkali (Newby and Bannister, 1993). Low NOx levels are obtained through use of advanced combustion processes (Scalzo et al., 1991).

Pressurized Fluidized Bed

Under a DOE-sponsored program, Westinghouse is developing a MASB topping combustor which will be used in the second generation PFB topping combustor (Domeracki, Dowdy and Bachovchin, 1994). The entire combustion air quantity is introduced into the combustor through axial flow, concentric vane rows. The MASB utilizes vitiated air at 1600°F (871°C) for cooling. Thermal NOx levels of less than 10 ppm have been measured for a burner outlet temperature of 2300°F (1260°C).

In addition to the topping combustor, Westinghouse is developing and supplying integrated gas turbine systems that will interface with PFB plants and incorporate the functions of hot gas filtration, alkali vapor removal, hot gas piping and control, and turbine compression and expansion (Newby et al., 1994).

Direct Coal

In a direct coal-fueled concept, ash and sulfur are to be removed from the coal during the combustion process. Air from the combustion turbine driven compressor flows into the pressurized slagging combustor and into a topping combustor. About one-third of the air is directed to the slagging combustor for utilization in the substoichiometric combustion of the coal fuel. Two-thirds of the compressor discharge air goes directly to the topping combustor where the low Btu fuel gas is fired under lean-burn conditions. One advantage of this split is that the majority of the primary combustion process only has to handle one-third of the total air flow required by the combustion turbine. The slagging combustor operates at temperatures high enough to melt or vaporize the inert ash constituents in the coal. Westinghouse has worked with Textron Defense Systems under a DOE contract to demonstrate the feasibility of this concept (Bannister, Newby and Diehl, 1992). Test results confirmed 99% carbon conversion with NOx levels under 50 ppmw. The next step in the development of this process is to integrate various hardware components into an operating cycle in a pilot plant.

CURRENT AND FUTURE TRENDS

In cooperation with U.S. Department of Energy's Morgantown Energy Technology Center, a Westinghouse led team is working on the second part of an 8-year, Advanced Turbine Systems (ATS) Program to develop the technology required to provide a significant increase in natural gas-fired combined cycle power generation plant efficiency (Bannister et al., 1994).

Efficiencies for large natural-gas-fired combined-cycle systems for the utility market have been demonstrated at 54 to 55%. Even though manufacturers will make improvements in the 1990s, pursuing the historic trend, shown in Tables 2 and 3, efficiency levels will reach a plateau for simple and combined cycle plants. Cycle innovations, a 2600°F (1427°C) combustion turbine RIT, reduced cooling air usage, steam cooling, improved component efficiencies, together with improved material/coating systems can achieve combined cycle efficiencies in the 60% range for natural gas-fired utility machines.

ATS using natural gas is to be commercially available by the year 2000. Coal-derived fuel concepts are candidates for the post-2005 power-generation market. The final design will be fuel flexible in that it will operate on natural gas, but also be capable of being adapted to operate on coal, coal-derived, or biomass fuels.

SUMMARY AND CONCLUSION

The Westinghouse evolutionary combustion turbine design philosophy has always maintained reliability and maintainability as major considerations. Many basic design innovations have been retained after over four decades of evolution, thereby attesting to the soundness of their selection. Future advancements in firing temperature, materials, cooling, and cycles must also satisfy reliability/maintainability requirements.

Westinghouse has actively worked with industry and the government for over three decades in developing the use of coal as a clean fuel for power generation. The company is currently actively supporting clean coal technology programs in advanced IGCC, first generation PFB, and second generation PFB plants.

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..AB-This paper reviews the evolution of heavy-duty power generation and industrial combustion turbines in the United States from a Westinghouse Electric Corporation perspective. Westinghouse combustion turbine genealogy began in March of 1943 when the first wholly American designed and manufactured jet engine went on test in Philadelphia, and continues today in Orlando, Florida with the 160 Megawatts, 501F Advanced Combustion Turbine. In this paper, advances in thermodynamics, materials, cooling, and unit size will be described. Many basic design features such as two-bearing rotor, cold-end drive, can-annular internal combustors, CURVIC[2] clutched turbine discs, and tangential exhaust struts have endured successfully for over 40 years. Progress in turbine technology includes the clean coal technology and advanced turbine systems initiatives of the U.S. Department of Energy.

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NEW 200 MW CLASS 501G COMBUSTION TURBINE

G. MCQUIGGAN, Combustion Turbine Development
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POWER GENERATION DEPARTMENT
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NEW 200 MW CLASS 501G COMBUSTION TURBINE

Mr. L. Southall and Mr. G. McQuiggan
Westinghouse Electric Corporation
Power Generation Business Unit
Orlando, Florida

ABSTRACT

The 501G 60-Hz Combustion Turbine has been developed jointly by Westinghouse Electric Corporation, Mitsubishi Heavy Industries, Ltd., and FiatAvio. It continues a long line of large heavy-duty single-shaft combustion turbines by combining the proven efficient and reliable concepts of the 501F with the latest advances in aero technology via the Westinghouse Alliance with Rolls-Royce. The output of the 501G is over 230 MW with a combined cycle net efficiency of 58%. This makes the 501G the largest 60-Hz combustion turbine in the world and also the most efficient.

INTRODUCTION

The 501G is a 3600 rpm heavy-duty combustion turbine designed to serve the 60-Hz power generation needs for utility and industrial service. The 501G combines the efficient, reliable design concepts of the 501F with the latest low NO_x combustion technology and the state-of-the-art cooling utilized in advanced, high-temperature aero engines (Reference 1). The result is an advanced design, high-temperature, efficient, low NO_x, more powerful combustion turbine based on time proven reliable design concepts that will satisfy the large combustion turbine power generation needs for the next decade (see Table 1). Designed for both simple and combined cycle applications, it will operate on all conventional combustion turbine fuels as well as with coal-derived low-BTU gas produced in an integrated gasification combined cycle power plant.Two units are being manufactured with both to begin operation in 1997. It will have an initial ISO rating of 230 MW at a turbine inlet temperature of 1500°C SYMBOL 176 \f "Symbol"C on natural gas fuel. In combined cycle applications, the net thermal efficiency of the plant is 58% (LHV) in power blocks of 340 MW nominal power rating.

This paper describes the features of this latest in a long line of heavy-duty combustion turbines of the 501 model series. Aerodynamic, cooling, and mechanical design improvements are discussed along with the evolutionary changes based on time proven design concepts. Technological advances as well as planned verification test programs are discussed including cascade, model turbine aerodynamic tests, combustor tests, rotating blade vibration tests, and shop test at load.

TABLE 1. 501 EVOLUTION

Engine	501A	501B	501D	501D5	501DA	501FA	501G
Commercial Operation	1968	1973	1978	1982	1994	1992	1997
Power, MW	45	80	95	107	120	160	230
Rotor Inlet Temp., °F	1615	1819	2005	2070	2150	2330	2583
Air Flow, Lb/Sec	548	746	781	790	832	961	1170
Pressure Ratio	7.5:1	11.2:1	12.6:1	14:1	15:1	15:1	19.2:1
No. Comp. Stages	17	17	19	19	19	16	17
No. Turbine Stages	4	4	4	4	4	4	4
No. Cooled Rows	1	3	4	4	4	6	6
Exhaust Temp., °F	885	907	956	981	1004	1083	1100
Heat Rate (Btu/kWh)							
LHV - ISO Gas							
Simple	12,600	11,600	10,925	10,040	9,900	9,610	8,860
Combined	9,000	7,350	7,280	7,055	7,024	6,429	5,883

General Description

Figure 1 illustrates the general configuration of the 501G. Several basic design concepts are evident; such as the two bearing single shaft construction, cold end drive, and axial exhaust. These fundamental time proven concepts used by Westinghouse for over 25 years have now become industry standards.

As in all past 501/701 designs, the single rotor is made up of the compressor and turbine components supported by two tilting pad bearings. For comparison with 501F, see Table 2. The 501G rotor is of bolted construction supported by two 18-inch-diameter, two-element tilting-pad bearings and an upper half fixed bearing. The thrust bearing is a double-acting bearing.

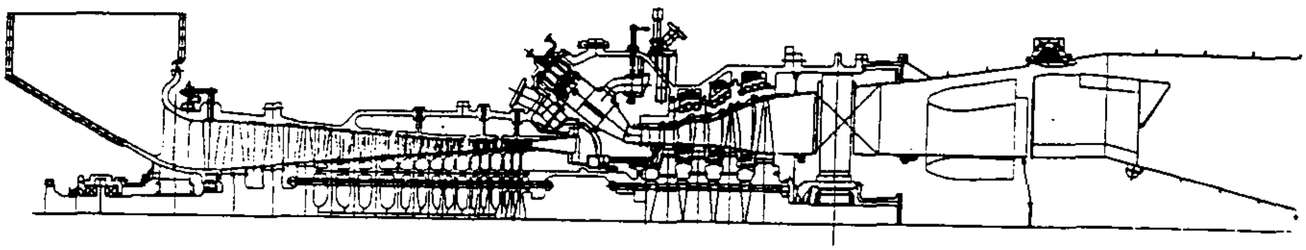


FIGURE 1. 501G

TABLE 2. 501G vs 501F ROTOR DESIGN

	501F	501G
Rotor Span	294.0"	311.6"
Journal Bearings	17.0"	18.0"
	2 Tilting Pads	
Thrust Bearings	22.5"	25.0"
Rotor Weights (Tons)		
Compressor	25.5	37.9
Turbine	21.0	26.9

The compressor rotor is comprised of a number of elements, spigotted and bolted together by 12 through bolts. Alignment and torque transmission is assured by the use of radial pins between the discs. The turbine rotor section is made up of disks bolted together by 12 through bolts and using CURVIC clutches, which consist of toothed connection arms that extend from adjacent disks and interlock providing precise alignment and torque carrying features. This turbine rotor design has amassed over 10 million hours of reliable service in all sizes of combustion turbines.

The air inlet system, which contains a silencer, delivers air to the compressor via a plenum-bell mouth and houses the inlet, main journal, and thrust bearings. The compressor is a 17-stage axial flow design of 19.2:1 pressure ratio that is based on the highly successful 501F. A four-stage turbine was selected to maintain moderate aerodynamic loadings even at the increased firing temperature.

The combustion system consists of 16 can-annular combustors. This low NO_x hybrid design is an improvement on the current highly successful design that has been in commercial operation for over three years in the 701F and 501D5. The presence or absence of flame and the uniformity of distribution of fuel flow between combustors are monitored by thermocouples located downstream of the last stage turbine blades. These can also detect combustor malfunctions when at load while U/V detectors are used to sense ignition during the early starting phase.

All engine casings are split horizontally to facilitate maintenance with the rotor in place. Inlet and compressor casings are of nodular cast iron and cast steel, respectively, while combustor, turbine, and exhaust casings are alloy steel. The inlet bearing housing is supported by eight radial struts, and the aft end bearing housing is supported by tangential struts. Airfoil-shaped covers protect the tangential struts from the blade path gases and support the inner and outer diffuser walls.

Tangential struts respond slowly during transients and maintain alignment of the bearing housing by rotating it as required to accommodate thermal expansion. Individual inner casings (blade rings) are used for each turbine stationary stage and can be readily removed and replaced or serviced with the rotor in place. Similar blade rings are used in the compressor. Another feature of these blade rings is that they have a relatively higher thermal response independent of the outer casing and can be aligned concentric to the rotor to prevent blade rubs, minimize clearance, and maximize performance.

Cooling circuits for the turbine section displayed in Figure 2 are similar to those used on the 501F. They consist of a rotor cooling circuit and four stationary cooling circuits. Rotor cooling air is provided by compressor discharge air extracted from the combustor shell. This air is externally cooled and filtered before returning to the torque tube casing for seal air supply and for cooling of the turbine disks as well as the first, second, and third-stage turbine rotor blades. This filtered air eliminates excessive contaminants that could block critical intricate cooling passages of the rotor blades.

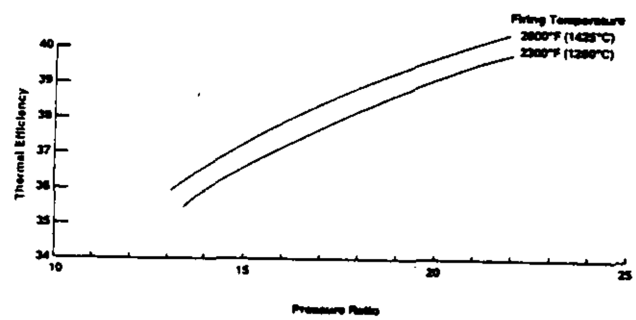


FIGURE 2. SIMPLE CYCLE - THERMAL EFFICIENCY VERSUS PRESSURE RATIO

Direct compressor discharge air is used to cool the row 1 vane while compressor bleed air is used to provide cooling air to turbine blade ring cavities at stages 2, 3, and 4, respectively. The compressor bleed air also cools the stage 2 and 3 vane segments and provides cooling and purging air for the turbine interstage disk cavities preventing the ingestion of hot blade path gases.

The stationary vanes and rotating blades for the first two turbine stages are coated with thermal barrier coatings. Compressor diaphragms are coated to improve aerodynamic performance and corrosion protection. For some environments, compressor rotor blades may be coated for corrosion protection.

CYCLE PARAMETER SELECTIONS

The 501G engine, like all recently designed combustion turbines of the authors' company, is designed for both simple and combined cycle service. The operating firing temperature level is selected to be commensurate with state-of-the-art materials and the latest aero-type cooling schemes. The value selected is 1425 SYMBOL 176 \f "Symbol"C at rotor inlet temperature.

After the turbine inlet temperature has been selected, the cycle pressure ratio can be chosen to maximize simple cycle power output and combined cycle efficiency. Figures 3 and 4, which are plots of simple and combined cycle efficiencies are used to select a pressure ratio of 19.2:1. This results in a cycle that has near maximum simple cycle output with a potential combined cycle efficiency of 58%.

The cycle air flow was set by the turbine exit annular flow area. It can be shown that the last stage blade stress level is directly proportional to the exit annular area. Since the long highly twisted last stage blade is uncooled, the blade material capability and last stage gas temperature then determine the flow capacity of the engine. A flow of 1200 lb/sec was selected, which results in a conservatively stressed blade and high power output.

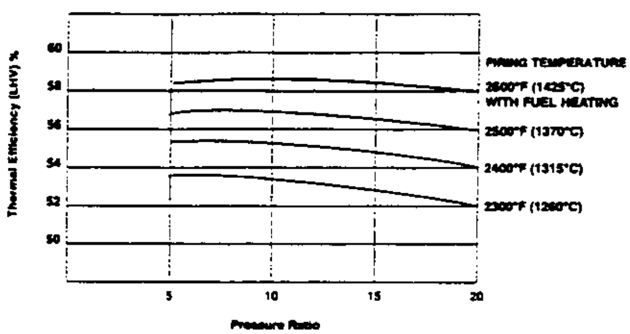


FIGURE 3. COMBINED CYCLE EFFICIENCY

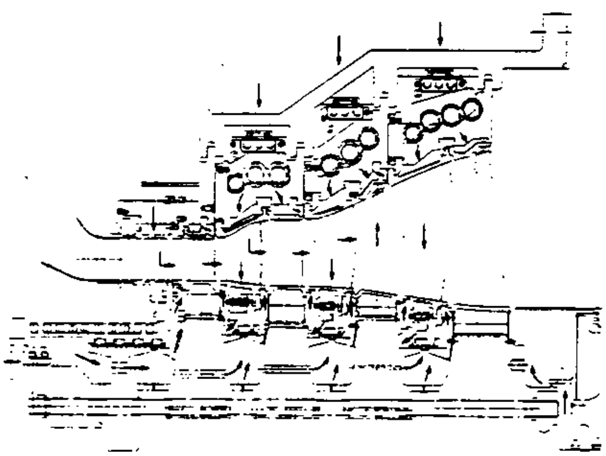


FIGURE 4.

COMPRESSOR DESIGN

The compressor is a highly efficient, 17-stage axial flow compressor patterned after the proven compressor of the W501F. Flow and pressure coefficients of the 501G compressor have been kept similar to the F compressor by increasing the mean diameter of the stages to accommodate the 25% increase in flow. In addition, the rear stages of the new compressor have larger diameters to help balance spindle thrust. Interstage bleeds for starting and cooling flows are in the sixth and eleventh stages, with the fourteenth stage used only for supplying cooling air to the second-stage turbine stationary blading and interstage cooling system. The compressor is also equipped with variable inlet guide vanes, which improve the compressor low-speed surge characteristic and are used in combined cycle applications for improved part-load performance.

Rotor blades are multiple circular arc designs and controlled diffusion airfoils.

Stationary blading is fabricated into two 180 SYMBOL 176 \f "Symbol" diaphragms per stage for easy removal and will maintain the highly efficient inner shroud sealing system currently used on the 501F. These seals will be supported by machined lips on the inner shroud and can be removed to facilitate inspection and maintenance of shrouds and seals. Stationary blading and shrouds are standard strength AISI 403 throughout. Abradable seals will be used to improve sealing.

TURBINE DESIGN

The design of the 501G has maintained moderate aerodynamic loadings by using a four-stage turbine with higher peripheral speed compared to the 501F. Furthermore, improvements in aerodynamic airfoil shapes have been made possible by utilization of a fully three-dimensional viscous analysis code. This sophisticated airfoil design approach was employed to assure that the turbine has the highest practical aerodynamic efficiency and the lowest cooling flow usage. The number of airfoils have been reduced relative to the 501F by approximately 15%. The first two turbine blades are unshrouded. The second and third blades are shrouded as on the 501F.

The first turbine stationary row consists of 32 precision-cast, single-vane segments of IN939 alloy. As in past designs, the row 1 vanes are removable, without any cover lift, through access manways. Inner shrouds are supported from the torque tube casing to limit flexural stresses and distortion, thus maintaining control of critical row 1 vane angles. This is the same method that was used successfully on the 501F. There are 36 precision-cast, single vane segments of IN939 material in the second turbine stationary row. The third and fourth turbine stationary rows are precision-cast vane segments with 14 three-vane and 14 four-vane segments, respectively.

Each row of vane segments is supported in a separate inner casing (blade ring) that is keyed and supported to permit radial and axial thermal response independent of possible external cylinder distortions. Blade ring distortion in the 501G turbine is further minimized by use of segmented isolation rings that support the vane segments and also ring segments over the rotor blades to form a thermal barrier between the flow path and the blade ring. As in all past designs, the interstage seal housings are uniquely supported from the inner shrouds of rows 2, 3, and 4 vane segments by radial keys that permit the thermal response of the seal housings to be independent of the more rapid thermal response of the vane segments

COOLING OF VANES, BLADES, AND DISCS

The cooling system shown in Figure 4 maintains the NiCrMoV turbine disks below 400°C (752°F), which keeps the disk below the creep range and assures long life. Fleet leaders with this disk design are the W501A turbines with up to 150,000 operating hours.

Blade and vane cooling flows have been kept to a minimum while maintaining similar metal surface temperatures as used in the 501F (Figure 5). By using thermal barrier coatings and proven aero engine cooling schemes such as serpentine cooling passages, shower head film cooling, and shroud film cooling along with improved materials, the creep and thermal fatigue margins have been maintained or improved over the 501F (see Figure 6).

The row 1 vane cooling is provided by impingement convection and film cooling. The film cooling holes are fan shaped and the airfoil and shrouds will be coated with a ceramic thermal barrier coating.

The row 1 vane cooling design is shown in Figure 7. This highly effective configuration utilizes state-of-the-art concepts with three impingements inserts in combination with an array of film cooling holes and a trailing edge pin fin system. Film cooling is used at the leading edge as well as at selected pressure and suction side locations. This limits vane wall thermal gradients and external surface temperatures, while providing an efficient re-entry for spent cooling air. Pin fins are employed to increase turbulence and surface area, thereby optimizing the overall edge pin fin exit system.

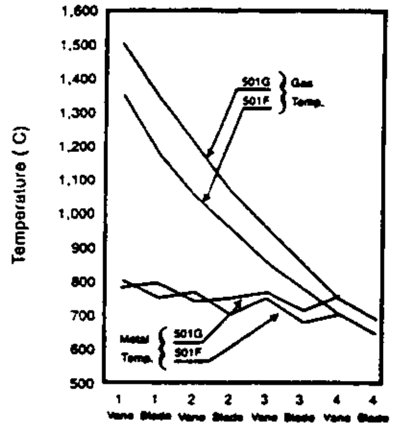


FIGURE 5. 501F AND 501G METAL TEMPERATURE COMPARISON

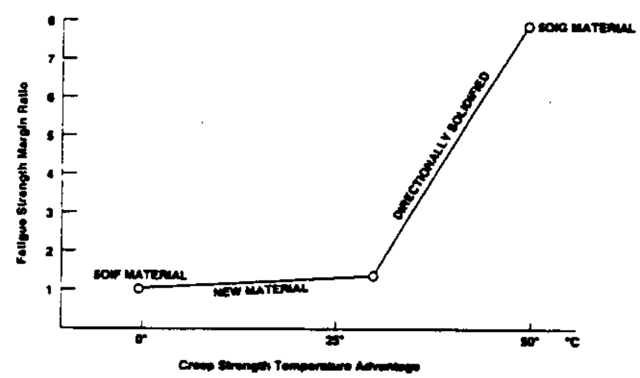
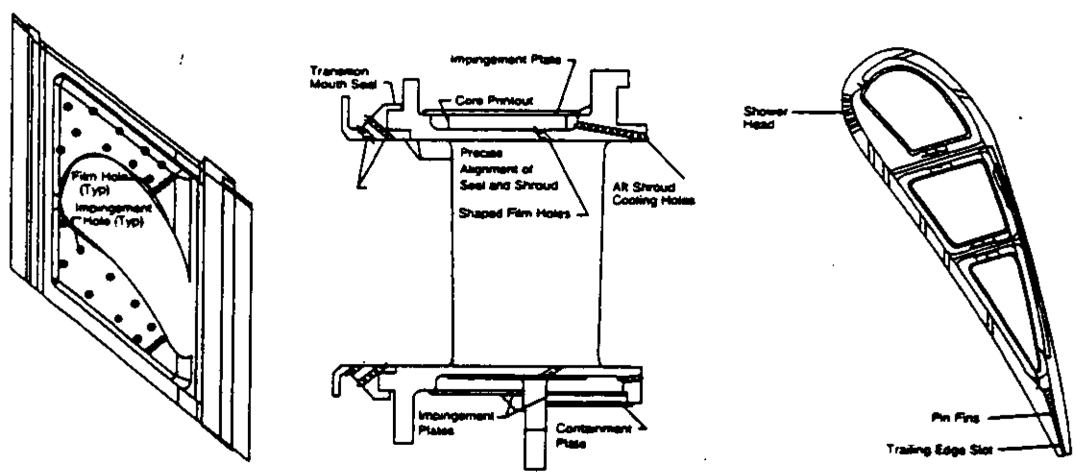


FIGURE 6. DIRECTIONALLY SOLIDIED ADVANCED MATERIALS CREEP AND FATIGUE STRENGTH IMPROVEMENT



Outer Shroud Impingement and Film Holes

Shroud Cooling

Vane Cooling

FIGURE 7. 501G ROW 1 VANE - COOLING

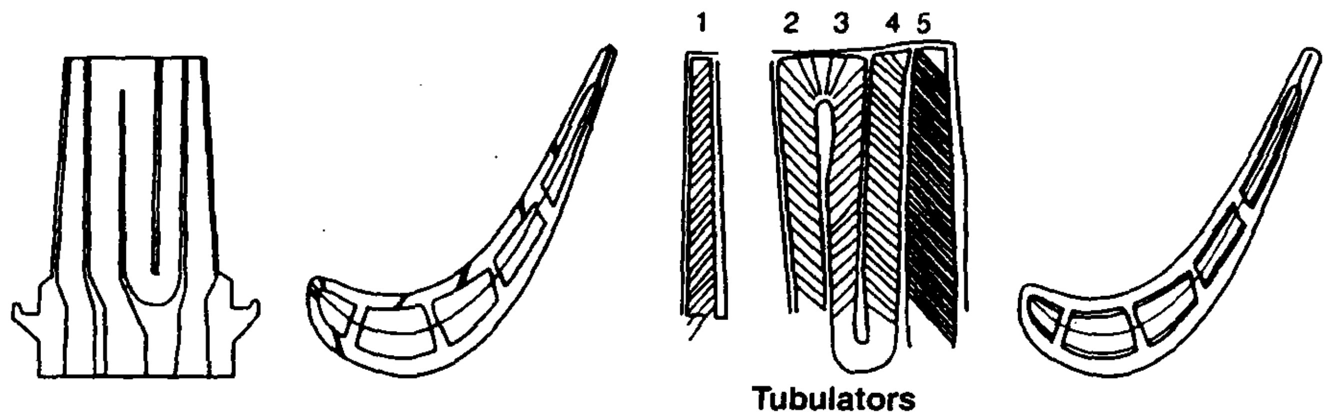


FIGURE 8. 501G ROW 1 BLADE - COOLING

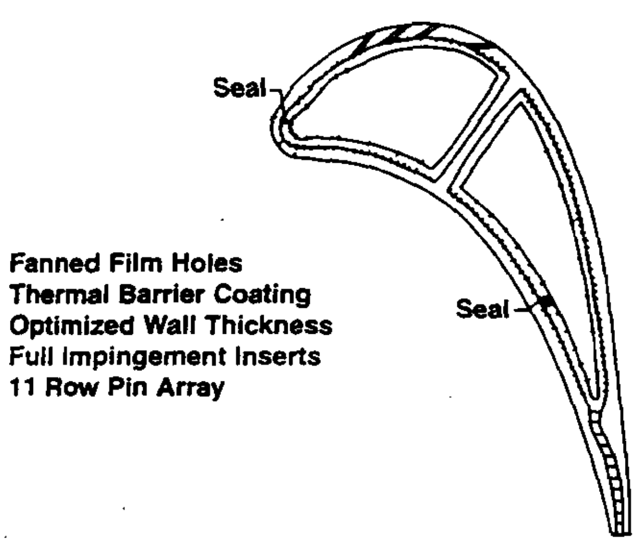


FIGURE 9. 501G ROW 2 VANE - COOLING MID HEIGHT

Particular attention is paid to the inner and outer shrouds because of the flat temperature profile from the dry low NO_x combustor. Cooling of the shrouds will be provided by impingement plates and film cooling as well as convection cooling via drilled holes.

The row 1 blade cooling consists of convectional serpentine cooling with angled turbulators. The film cooling utilizes fan shaped cooling holes for more effective cooling. The blade also features extensive film cooling at the tip to reduce the metal temperature of the squealer tip and platform cooling holes to positively cool the inner platforms. The airfoil will be coated with a vapor deposition thermal barrier coating (Figure 8).

The row 2 vane is cooled by a combination of impingement cooling via the inserts and film cooling using fan shaped cooling holes. The vane will be manufactured as a single vane to reduce thermal stress and to allow the vane to be coated with thermal barrier ceramic coating. As with the row 1 vane, intensive cooling schemes are applied to the inner and outer shrouds using impingement and convection cooling (Figure 9).

The row 2 blade is cooled by convection with no film cooling. The airfoil will be coated with a vapor deposition T.B.C. (Figure 10).

The row 3 blade cooling is unique in that it positively cools the blade tip shroud. Because of the flat profile from the combustor and because of tip leakage from the rows 1 and 2 turbine blades, it was decided to allow for positive cooling of the tip shroud. Analysis has shown that this scheme will have ample margin for cooling the shroud (Figure 11).

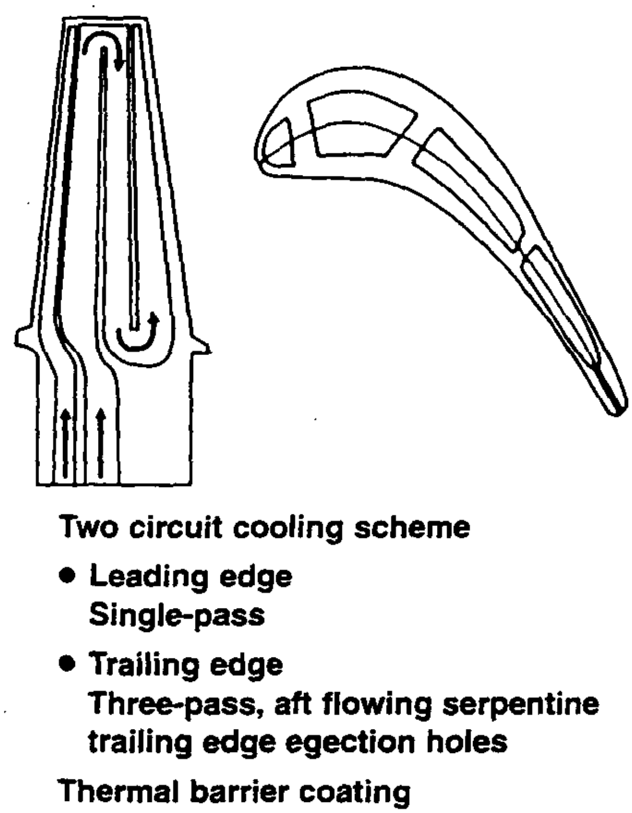


FIGURE 10. 501G ROW 2 BLADE - COOLING

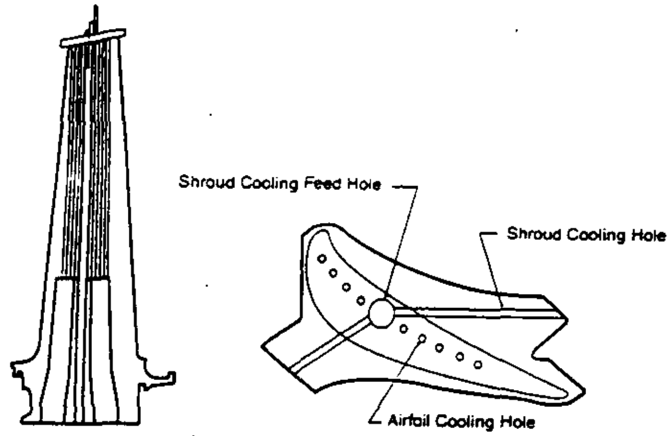


FIGURE 11. 501G ROW 3 BLADE - COOLING

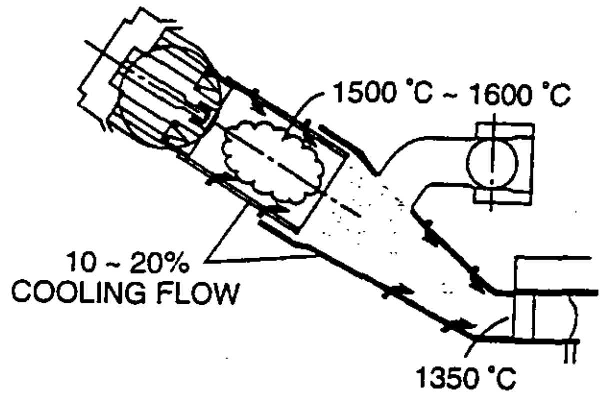


FIGURE 12. 501F COMBUSTION SYSTEM

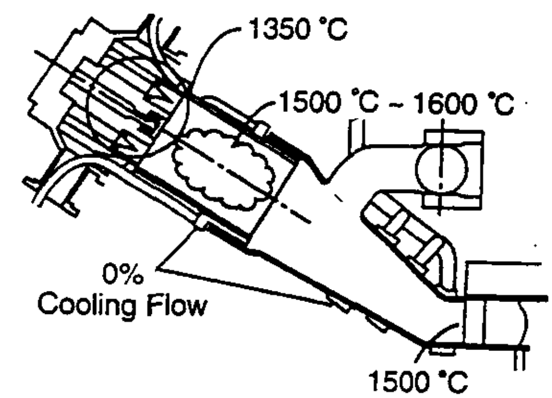


FIGURE 13. 501G COMBUSTION SYSTEM

Compressor bleed air from the 11th stage is used to supply cooling air to the third-stage blade ring cavity. Cooling air is directed to the inlet cavity of a five-cavity multipass convective cooled vane airfoil. Leading edge cavity flow also supplies the interstage seal and cooling system while the fifth pass cavity exits at pressure side "holes" on the vane surface near the trailing edge. The fourth-stage vane is uncooled but does transport sixth-stage compressor bleed air for the fourth-row interstage seal and cooling system.

The rotating blades are precision cast of CM247 for all four rows. All rows utilize long blade root extensions or transitions in order to minimize the three-dimensional stress concentration factor that results when load is transferred between cross sections of different size and shape. The blade roots are the same geometric multiple serration type used on past designs with four serrations used on the first three rows and five serrations used on the rear stage.

501G COMBUSTOR

The combustor is based on the successful dry low NO_X combustor developed for the 501F. This combustor is presently operating at a 25 ppm NO_X level at 1260°C (2300°F) R.I.T. temperature. The 501G combustor will make use of steam cooling to allow for the same NO_X reduction at the higher firing temperature. By eliminating the transition cooling air, virtually all the combustion air is introduced into the primary zone of the combustor so maintaining the flame temperature at nearly the same level as that of the 501F turbine. Therefore, NO_X levels are similar to the 501F level (see Figure (12) and Figure (13). Fuel heating will be used to improve the efficiency of the cycle with the heat coming from the rotor cooling air heat exchanger. The same combustor is designed to burn liquid fuel using water or steam injection to control NO_X levels. Initial emission targets are given in Table 2.

VERIFICATION TESTING

All new advanced technology parts applied in the 501G engine are qualified for engine use by verification tests, including: (a) rotating blade vibration, (b) turbine aerodynamic, (c) combustion tests, and (d) hot turbine cascade testing. Overall engine performance and durability will be verified by engine shop tests.

AERODYNAMICS

The total compressor was tested as a scaled model in a small combustion turbine in 1994. This followed extensive single and dual stage rig testing in 1993.

The turbine design tools have been verified in extensive model and rig tests, and in particular a scale model of the 501G turbine row one vane and blade was tested in 1994 and fully verified the expected aerodynamic performance.

COMBUSTION

The development of the low NO_X combustor is being carried out using full scale pressure and temperature rigs. The initial specification is for a 25 ppm NO_X combustor, with gradual movements to lower levels (see Table 3). The combustor is a dual fuel combustor and will use water or steam to control NO_X when burning liquid fuel.

TABLE 3. 501G INITIAL EMISSIONS TARGETS

	Natural Gas	Oil
• NOx	< 25 ppm	< 42 ppm
• CO	< 10 ppm	< 25 ppm
• UHC	< 10 ppm	< 15 ppm

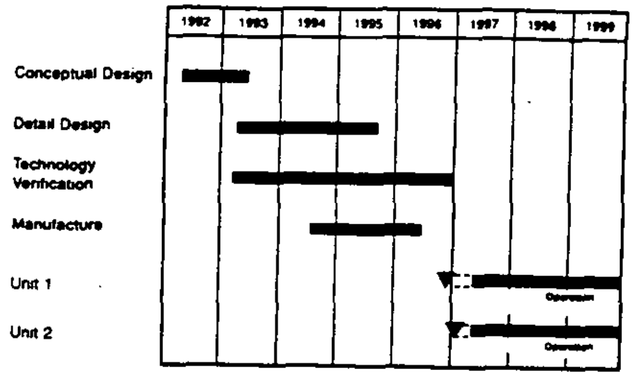


FIGURE 14. 501G DEVELOPMENT SCHEDULE

COOLING

Because of the elevated firing temperatures, it is planned to carry out a cascade test of the row one blade and vane in a full firing temperature rig facility in 1995. This test will verify operating temperatures, especially for the difficult endwall regions, before the engine is introduced into commercial operation.

In addition, cascade testing of model vanes and model tests of blades will be conducted to verify pressure loss in turbine passages and film cooling of shrouds.

SHOP AND FIELD TESTING

Two prototype units will be tested, these will be fully instrumentated engines which will be tested to full power conditions before entering commercial service in 1997 (see Figure 14).

The prototype test is the important final stage for the confirmation of the following:

1. Compressor inlet airflow over entire IGV range
2. Compressor surge margin
3. Engine starting and acceleration characteristics
4. Mechanical integrity of the engine from starting to overspeed including rotor vibration characteristics.
5. Mechanical and thermal performance of the engine over its entire operating range.
6. Reliability of the engine by measurement of gas and metal temperatures, pressure, vibratory stresses, etc.
7. Emission characteristics of the engine and the effects of injection upon emissions and thermal performance.

To provide this data, the engine will be extensively instrumented to measure thermodynamic values, metal temperatures, static and vibratory strains, vibration characteristics, displacements, and other parameters. Dynamic strain gauges will be installed on the turbine blades to verify dynamic responses. The signals from the rotating sensors will be transmitted by a telemetry system. Clearance measurement systems using proximity probes allow stator-to-rotor radial displacement measurements during transients. Data acquisition equipment will be installed to record the special engineering test data. This equipment includes tape recorders, spectrum analyzers, plotters, and chart recorders.

LIFE CYCLE COSTS

Combustion turbines are designed for long life. However, the hot parts especially the cooled turbine components and the combustion equipment, have a finite life because of thermal fatigue cracks, corrosion, oxidation, creep, and other phenomena. In order to reduce the cost of these parts to the customer, the 501G has been designed with the objective of reducing the number of hot parts. The most expensive and most often replaced aerofoils are the row one and row two vanes and blades. Compared to the 501F, these numbers have been reduced by 20%.

Maintenance intervals, despite the higher firing temperatures, have been maintained at the current generation level (see Table 4). This is possible by keeping the same surface metal temperatures as exists in the 501F, and by improving the creep and thermal fatigue resistance by utilizing directionally solidified superalloys.

TABLE 4. 501G INSPECTION INTERVALS

Inspection Type	Hours	Starts
Combustor	8,000	400
Turbine	24,000	1,200
Major	48,000	2,400

Gas Fuel

CONCLUSION

The major market drivers for combustion turbines are capital cost, operating cost, emissions, and fuel availability. To make a step change in the technology of the combined cycle projects, a new advanced design combustion turbine is introduced that is more efficient and cost effective than competitive machines currently available or projected. The 501G engine utilizing advanced aeroengine design and tools, elevated firing temperatures, and proven technology, results in a design that maximizes the output for minimum cost per KW at maximum operating efficiency. By the novel use of steam cooling, the flame temperature is kept at the same level as the 501F; thereby, the new engine will have low levels of NO_x despite the elevated rotor inlet temperature.

The resulting power plant efficiency will improve from today's best of 54% to 58% an improvement of 7.4%. At the same time the size of the engine, compared to the 501F, will be only slightly increased. The 501G engine will produce approximately twice the output of a 501D machine and is 70 MW larger than any other currently available 60 Hz combustion turbine on the market. The high power output will further enhance the maintenance and balance of plant cost advantage of the 501G engine when compared to a plant of similar output but requiring more combustion turbines.

REFERENCES:

1. A. J. Scalzo and Others, "A New 150 MW High Efficiency Heavy Duty Combustion Turbine", ASME 88-GT-162