**ADVANCED
TURBINE
SYSTEMS
PROGRAM**

**Proceedings of the Advanced Turbine
Systems Annual Program Review Meeting**

**November 9-11, 1994
Oak Ridge National Laboratory**

**Office of Energy Efficiency
and Renewable Energy
Office of Fossil Energy
Washington, DC**

In Memoriam

*Dr. Holmes Andres "Sandy" Webb, Jr.
1943-1994*

The Advanced Turbine Systems Program suffered a major loss on September 8, 1994, in the crash of USAir Flight 427 near Pittsburgh, which took the life of Dr. H.A. ("Sandy") Webb, Jr. The ATS Program exists today largely due to the efforts and dedication of Sandy.

Sandy was born to Dr. Holmes Andrew and Doris Johnson Webb on January 15, 1943, in Abilene, Texas. After graduating from Robert E. Lee High School, he earned a Bachelor of Science in Mechanical Engineering from Texas Tech University in 1965. He received his Ph.D. from Northwestern University in 1972. Dr. Webb then served as an assistant professor at Texas Tech University for one year, and as a research engineer at E.I. DuPont de Nemours & Co. for six years.

Sandy joined Morgantown Energy Technology Center (METC) in 1978. During his distinguished sixteen-year career at METC, he was responsible for programs concerned with developing advanced power-generation technologies. Dr. Webb was most recently Product Manager for Advanced Heat Engines, which included direct and indirect coal-fired turbines and diesels and the ATS Program.

We will deeply miss his leadership, expertise and friendship.

The plane crash also claimed the lives of Randy Dellefield, William Langan, and Manville Mayfield who were employees of METC.

ls
DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

MASTER

DISCLAIMER

**Portions of this document may be illegible
in electronic image products. Images are
produced from the best available original
document.**

Foreword

The U.S. Department of Energy, Office of Energy Efficiency and Renewable Energy, Office of Fossil Energy, held its Second Annual Advanced Turbine Systems Program Review on November 9-11, 1994, in Arlington, Virginia. The goals of this eight-year program are to develop cleaner, more efficient, and less expensive gas turbine systems for utility and industrial electric power generation, cogeneration, and mechanical drive units. Projects are funded jointly by the office of Fossil Energy and private industry.

During the three-day meeting, presentations on energy policy issues were delivered by representatives of regulatory, industry, and research institutions; program overviews and technical reviews were given by contractors; and on-going and proposed future projects sponsored by university and industry were presented and displayed at the poster session. Panel discussions on distributed power and Advanced Gas Systems Research (AGTSR) education provided a forum for interactive dialog and exchange of ideas. Exhibitors at the conference included the U.S. Department of Energy; Solar Turbines, Incorporated; Westinghouse Electric Corporation; Allison Engine Company; and General Electric.

This proceedings contains copies of the majority of the papers and posters presented during the sessions in the order listed on the agenda.

William P. Parks, Jr.
Acting Director,
Industrial Energy Efficiency Division
Advanced Turbine Systems
Office of Industrial Technologies

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

INTRODUCTION

These proceedings summarize the annual activities of the Advanced Turbine System (ATS) Program. The objective of the program is to develop more efficient gas turbine systems for both, utility and industrial power generation (including cogeneration). The program is developing base-load power systems for commercial offering in the year 2000. Although the target market is natural gas, advanced turbine systems will be adaptable to coal and biomass fuels. All advanced turbines will exhibit: (1) ultra-high efficiencies, (2) environmental superiority, and (3) cost competitiveness.

The ATS ^{*}"Comprehensive Program Plan" outlines an eight-year program, estimated to be \$700 million consisting of approximately \$450 million government share and approximately \$250 million cost-shared by industrial participants. Turbine systems burning natural gas offer environmentally sound and economical power generation and cogeneration. Since the U.S. demand alone will require up to 15 gigawatts per year of new and replacement capacity after the year 2000, the availability of ATS to fill a share of this need will save significant amounts of fuel and benefit the environment.

Implementation of the ATS program will also keep U.S. manufacturers on the cutting edge of turbine technology for power generation application and enhance the nation's economic competitiveness. The ATS program has four elements; innovative cycle development, utility system development demonstration, industrial system demonstration, and a technology based development. The basic program strategy is to fund teams lead by U.S. turbine manufacturers to develop advanced turbine systems from concept and component development to full-scale prototype demonstration.

^{*}"Report to Congress Comprehensive Program Plan for Advanced Turbine Systems", February 1994, DOE/FE-0279-1

Contents

Session 1: Policy and Strategic Issues

- 1.1 Energy Industry Changes: The Role of Advanced Combustion
Turbine Technology—George Touchton, Electric Power Research
Institute 3
- 1.2 Regulatory Issues Facing Gas Turbine Facilities—Eric Wong for Charles
Imbrecht, California Energy Commission 15

Session 2: Program Element Overviews

- 2.1 Overview of Westinghouse's Advanced Turbine Systems Program—
Ronald L. Bannister, Westinghouse Electric Corporation 21
- 2.2 Advanced Turbine Systems Program Industrial System Concept
Development—David A. Rohy, Solar Turbines
Incorporated 27
- 2.3 METC Combustion Research Facility—Jack Halow, Morgantown Energy
Technology Center 33
- 2.4 A Collaborative Vision: The Advanced Gas Turbine Systems Research
Program—Daniel Fant, South Carolina Energy Research &
Development Center 39
- 2.5 Materials/Manufacturing Support Element for the Advanced Turbine
Systems Program—Michael A. Karnitz, Oak Ridge National
Laboratories 45
- 2.6 Distrusted Power: A Contractor's Perspective—David E. Eppinger,
Fluor Daniel 51
- 2.7 ATS For Distributed Utility Generation: A Case Study—Dan Whitney,
Sacramento Municipal Utility District 57

Contents

(Continued)

Session 3: ATS Program Technical Review

- 3.1 Advanced Turbine Systems Program: Phase 2 Development Projects-
Ihor S. Diakunchak, Westinghouse Electric Corporation 67
- 3.2 Advanced Turbine Systems Program Industrial System Concept
Development-Karl Karstensen, Solar Turbines
Incorporated 75
- 3.3 ATS Turbine Modification for Coal and Biomass Fuels-
Richard Wenglarz, Allison Engine Company 90
- 3.4 Ceramic Retrofit Project-Mark van Roode, Solar Turbines
Incorporated 97

Session 4: ATS Project Related Activities

- 4.1 Long Term Testing- Matt Ferber, Oak Ridge National Laboratories 121
- 4.2 Dry Ultra,Low Emissions Combustion System for Allison Industrial
Engines-Mohan Razdan, Allison Engines Company 139
- 4.3 High Performance Steam Development-Tom Duffy, Solar Turbines
Incorporated 147
- 4.4 Turbine Airfoil Manufacturing Technology-Land Based Turbine Casting
Initiative-Michael A. Karnitz, Oak Ridge National
Laboratories 165

Session 5: University/Industry Consortium Project Review

Materials Session

- 5.1 Functionally Gradient Material for Thermal Barrier Coatings in Advanced
Gas Turbine Systems-Arnold R. Marder, Lehigh University 179
- 5.2 Life Prediction of Advanced Materials for Gas Turbine Application-
Sam Y. Zamrick, Penn State University 185

Contents

(Continued)

5.3	Reduction of Turbine Endwall Total Pressure Loss and Heat Transfer Using the Iceformation Design Method-Ronald S. LaFleur, Clarkson University	207
-----	--	-----

Heat Transfer Session

5.4	Vortex-Generator Induced Enhanced Heat Transfer in Gas Turbine Coolant Channels-Sumanta Acharya, Louisiana State University	221
5.5	Advanced Turbine Cooling, Heat Transfer, and Aerodynamics Studies- Je-Chin Han, Texas A&M University	235
5.6	Emissions Models for Lean Premixed Turbines-Arthur M. Mellor, Vanderbilt University	253
5.7	Combustion Modeling in Advanced Gas Turbine Systems- Paul O. Hedman, Brigham Young University	273
5.8	Lean Premixed Combustion Stabilized by Radiation Feedback and Heterogenous Catalysts-Robert W. Dibble, University of California, Berkeley	293
5.9	NOx Abatement in Advanced Gas Turbines-Paul E. Sojka, Purdue University	319
6.0	Actuators for Combustion Control: New Approaches and Materials- Uri Vandsburger, Virginia Polytechnic Institute	355

Poster Session: New Research Topics

Aerodynamics Group

P.1	Experimental and Computational Studies of Film Cooling with Compound Angle Injection-Terry W. Simon, University of Minnesota	383
P.2	Improving Aerodynamics of the Intercooler Flow Path for the Development of High Efficiency Gas Turbines-Ajay K. Agrawal, University of Oklahoma	395

Contents

(Continued)

- P.3 Examining the Role of Turbulence on Internal Heat Transfer-
Paul A. Dellenback, University of Wyoming 405

Heat Transfer Group

- P.4 Measurements of Surface Temperature and Heat Transfer Using a
UV-Induced Phosphor Fluorescence Thermal Imaging System-
Mingking K. Chyu, Carnegie Melon University 413
- P.5 Advanced Multistage Turbine Blade Aerodynamics, Performance, Cooling
& Heat Transfer-Sanford Fleeter & Patrick Lawless, Purdue
University 421
- P.6 Steam as Turbine Blade Coolant: Phase I Development of the Experimental
Facility-Abraham Engeda & John Lloyd, Michigan State
University 425

Materials Group

- P.7 Combustion Chemical Vapor Deposited Coatings for Thermal Barrier
Coatings Systems-W. Brent Carter, Georgia Institute of
Technology 437
- P.8 Compatibility of Gas Turbine Materials with Steam Cooling-
Vimal Desai, University of Central Florida 449

Combustion Group

- P.9 The Role of Unmixedness, Strain Rate and Length Scale on Performance
and Emissions in a Combustion System-Scott Samuelsen &
John LaRue, UCI Combustion Laboratory, University of California,
Irvine 459
- P.10 Advanced Concepts for High Efficiency and Low NO_x Gas Turbine Combustor
Development-Ashwani K. Gupta, University of Maryland 473

Contents

(Continued)

Appendices

Appendix A: Agenda	A1
Appendix B: List of Participants	A9
Appendix C: List of Speakers	A27
Appendix D: In Memorium	A33

Session 1

Policy and Strategic Issues

ENERGY INDUSTRY CHANGES:

THE ROLE OF ADVANCED COMBUSTION TURBINE TECHNOLOGY

George Touchton
Manager, Natural Gas Fired Central Stations
Electric Power Research Institute

EPRI IN A CHANGING WORLD

EPRI is voluntary not-for profit organization which carries out a broad program of technology research, development and delivery on behalf of its investors and stakeholders. EPRI and the global electric energy industry are in turbulent transition from a stable past to a challenging future, and EPRI has reevaluated both its vision and mission in light of these sweeping changes.

EPRI's newly minted vision is to lead in electrification through global collaboration. This vision adds global collaboration to the traditional leadership in electrification. In realizing this vision, EPRI intends to excel in its mission of discovering, developing, and delivering high value technology advances through networking and partnership with the electricity industry. Significantly, networking and partnership with the electricity industry has been explicitly recognized as the cornerstone of EPRI's future relations. As a leading example, EPRI's long-standing cooperation with the Department of Energy was recently formalized through a Sustainable Electric Partnership Agreement signed by DOE Secretary Hazel O'Leary and EPRI President Richard Balzhiser.

EPRI is realizing its vision of global collaboration through expanded opportunities for membership moving with, and leading, the changing domestic and global electricity energy market. The core of EPRI is

still traditional utility membership, which covers the full gamut of large to small domestic utilities, and government agencies such as TVA. Presently, as in the past, EPRI represents approximately 70% of all megawatts generated in the US.

In recent years, EPRI has extended opportunities for affiliation to international electricity industry companies, and at this time has five affiliates, PowerGen, a generation company in the United Kingdom, KEMA, the electric energy industry R&D arm in the Netherlands, AEM, the municipal utility of the City of Milan, British Columbia Hydro, a Canadian generating company, and State Electricity Commission of Victoria, an Australian utility.

Finally, and most recently, EPRI has extended membership opportunities to other electricity industry participants. Class V membership covers the burgeoning, some would say exploding, new generation market — co-generators, independent power producers, exempt wholesale generators, and self-generators. In the context of advanced combustion turbine technology, EPRI is a users' organization — perhaps the world's broadest based user organization.

ENERGY INDUSTRY CHANGES

As the electricity industry makes the

turbulent transition from the calm well mapped landscape of the past to a fragmented, and as yet uncertain, future, three themes guide and constrain it. The first theme is competition. The mood of the populace, government, and industry is swinging toward more accountability and cost reduction. Market forces should be allowed to work to deliver the widest variety of services at the lowest cost to the consumer. Regulation, where it exists, should aid and accelerate the movement to market competition. Increased competition and decreased regulation drive pressures for accountability, and for cost reduction. Business practices and technologies will be closely screened for their effect on the bottom line and for their cost impacts.

Environment is the second theme. Despite pushes for lower costs, pressures for improved environmental performance will increase — not diminish. New and old industry players must take a proactive role in issues such as EMF, air, water, and solid wastes, toxics, global warming, and others. As in the past, many of these issues will extend beyond the boundaries, and the scope of any one player or group of players and will demand national or international collaboration.

The third theme, the focus of the DOE Advanced Turbine Systems program, is technology. However, today as never before, technology development cannot stand alone — delivery is the market test of success. Only if technology produces a competitive advantage for the user will delivery be successful. Research and development are vital, and must be done, but the research and development community must constantly evaluate the pull from the market for each product.

Market Drivers

Users of technology are diverse today, and their diversity is increasing (as noted under EPRI membership). This growth in diversity is driven by the fragmentation of the served markets. Today's list includes utilities, cogenerators, independent power producers, and exempt wholesale generators. Wholesale power is the present focus of competition among generators, but that focus will change to include specific served loads. In the past all retail sales went through utility transmission and distribution systems to the consumer, but now generators can deal directly with large customers, and soon — by 2002 — if the California Public Utility Commission ruling takes hold, they will be able to deal directly with small consumers, with the transmission and distribution companies receiving a fee for "wheeling" the power to the user.

The challenges for all these users will be to:

- Hone and keep the competitive edge
- Take charge of environmental issues
- Achieve environmental excellence
- Manage assets as well as liabilities
- Focus on profitable markets
- Reduce costs and keep them down

Identifying and deploying critical technologies will be A Key Success Factor.

Power Markets

In wholesale power, the usual competitive focus has been on building and operating new capacity, and the players have been utilities versus independents and cogenerators. The independents and

cogenerators have enjoyed considerable success, and today build about 50% of all new U. S. generation. However, a new competitive focus is emerging. Service to — and direct contracts with — wholesale loads are the focus of this competition, and the players will be generating companies (GENCOs), exempt wholesale generators (EWGs), independents (IPPs), and cogenerators (COGENs).

In this emerging wholesale power market, new capacity will be chosen to serve specific new loads — not to maintain some theoretical "generation mix" criteria. Often these new loads will be the subject of capture or retention efforts. The larger loads such as bulk power contracts with large industrial customers, can be classified into base, peaking, or mid-range and will often be served from central stations. Distributed generation will play a key role in responding to smaller customers, increasing the utilization of transmission and distribution resources, and negotiating prices for purchased power.

Finally, as new competitors crowd the market, generators will be faced with keeping their existing capacity competitive and maximizing return on their sunk investment. Repowering offers many opportunities in this area, and advanced combustion turbine technology is the dominant repowering technology for older fossil-fired steam resources. Repowering can take several forms. In "site repowering" one completely replaces all generation resources with a combined cycle plant, retaining only site permits and services. In "HRSG repowering" the boiler of the steam plant is replaced with a combustion turbine and heat recovery steam generator keeping the turbine cycle. In "feedwater heater repowering," steam cycle feedwater is heated in a combustion turbine heat recovery steam generator. This option is especially attractive

for super-critical plants. Finally, in "hot-windbox repowering," the exhaust from a combustion turbine is by ducted into the air side of the Rankine cycle boiler, replacing the air heater and fans.

CT User Market Drivers

Users of advanced combustion turbine technology who are driven by the market, will be winners. Their first priority will be reducing supply and delivery costs. PG&E, one example among many, expects to reduce 1995 costs by \$285 million. However, cost reduction is only the entry requirement. Winners will also master proactive pollution prevention at least cost, and they will manage fuel supply links by coordinating fuel supply and future generation planning, and exploiting present fuel supply and generation synergies.

Advanced turbine technology contributes to each of these winning strategies. It reduces cost directly by reducing fuel costs through improved efficiency. It reduces pollution absolutely due to its efficiency, and comparatively when it displaces solid or liquid fuel generation. Advanced combustion turbine technology can generate credits when it is running for later trade or consumption, and it integrates easily into environmental dispatch schemes. It is a multi-fuel technology burning gaseous, liquid, and solid (when combined with gasification) fuel. This enables a generator to exploit present low gas prices with a secure bridge to alternatives if needed. Finally, its rapid start-up minimizes both cost and pollution.

Thus advanced combustion turbine technology will help winners deliver least cost power for their targeted market —

whether it is peaking, mid-range, or base load.

Technology Drivers

Each of the advantages of advanced turbine systems has a technical challenge. Reaping the full benefits of this technology will require:

- Improving gas turbine systems across a range of sizes and applications
- Greater real time dynamic operation of grid delivery system
- Expansion and more real time dynamic operation of gas delivery system

Users will need a balanced portfolio of advanced turbine plants in a wide range of sizes and for a range of applications from distributed generation to central station generation. They will need these plants for cogeneration, new green field sites, re-powering, and unattended remote applications. Because of their flexibility and rapid response advanced combustion turbine plants will facilitate and be closely integrated with real time dynamic operation of the grid delivery system. In order for this to happen an expansion of the gas delivery system will be needed, a synergy that EPRI and GRI have identified and studied for some time now. In addition there are opportunities for siting flexible gas-fired generation which exploit the capabilities of existing gas and electric transmission systems. The San Diego Gas and Electric Company has identified several such opportunities in their studies of the inter-cooled aeroderivative combustion turbine.

Collaborative Advanced Gas Turbine (CAGT) Initiative

The Collaborative Advanced Gas Turbine Initiative was begun by a user (PG&E) in response to an identified long term need for high efficiency gas-fired combustion turbine systems and a mid-term need for efficient, gas-fired mid-range generation which could be installed at remote sites to improve transmission system utilization. EPRI took over the leadership and administration of this program when PG&E reduced its involvement for business reasons. The program has identified the intercooled aeroderivative machine as meeting mid-range generation goals and providing unique generation capabilities. It has high simple cycle efficiencies — about 45 to 47% — in the size range of 100 to 125 MWs. Further, when combined with humidification the ICAD machine can meet the long range efficiency goals of 60+% in a size range of about 200 to 250 MW.

Humid Air Turbine (HAT) Cycle

Humidified cycles have been a major emphasis of EPRI and its members for many years. The idea is quite simple and elegant. Evaporate water into the compressor discharge to get the cooling effect of the heat of vaporization while increasing the mass flow to the turbine. The fact that you raise the water to pressure as a liquid minimizes the compression energy needed. Further, since the water begins at a low temperature, the saturator can utilize low-level cycle heat, such as inter-cooler heat, in the expansion (energy producing) step. A significant barrier to commercialization of HAT technology has been the development costs for specialized compressors and turbines. This cost barrier is overcome in the Cascaded Humidified Advanced

Figure 1: Adding humidification to the combustion turbine cycle captures waste heat from the inter- and after-coolers and exhaust and increases the mass flow through the expander.

Turbine (CHAT) which uses an auxiliary spool to match existing (W501D) compressors and expanders. CHAT has been studied by EPRI, TVA and Energy Storage Power Systems, and these results were recently reported at the EPRI Gasification Conference:

- Capital costs of \$400/kW compared to \$500/kW for a combined cycle
- Efficiency of 51.4% compared to 53.7% for the combined cycle
- Much better part load efficiency than combined cycle.

Combustion Turbine Technology Summary

Figure 2 shows the full range of efficiencies and sizes for combustion turbine based technology. The gaps in size and efficiency between the simple and combined cycle products are created by the physics of the situation. Adding an efficient bottoming cycle to a heavy frame machine increases power by about 50% and increases efficiency

Figure 2: The sizes and efficiencies of current and advanced combustion turbine technologies show characteristic features. ICAD technology is position uniquely.

by almost 20% points. Adding an efficient bottoming cycle to an aeroderivative increases power by about 30%, and increases efficiency a little over 10% points.

The intercooled advanced aeroderivative (ICAD) machine effectively fills both size and efficiency gaps. ICAD technology positions a simple cycle (i.e. — no bottoming cycle) machine at about 100 - 120 MWs at about 9 - 12% points better efficiency than an equivalent size simple cycle heavy frame machine. This ICAD machine should have many of the advantages of a simple cycle including: simplicity, potential for remote siting and dispatch, and no steam cycle equipment. Further, in combined cycle the ICAD is competitive with heavy frame technology such as the GE Frame 6FA. Finally, the ICAD turbine provides an excellent stepping stone to ATS goals, and should meet or exceed them when combined with humidification.

Compressed Air Storage with Humidification (CASH) Cycle

The CASH cycle adds underground storage to the humidified turbine cycle, as shown in Figure 3. The compressor is separated from the expander, and is driven from system power. Thus, the expander (energy producer) does not have to

Figure 3: The Compressed Air Storage with Humidification (CASH) Cycle offers unique advantages for base-load and peak shaving at competitive costs

drive the compressor when drawing air from the cavern. This dramatically (by approximately 30%) increases its power output. This output is available for very rapid peaking, since the compressor load can be dropped and diverted to the line virtually instantaneously. Thus CASH offers unique advantages to generators who serve large base loads and/or provide bulk power, while at the same time providing system peak shaving to minimize off-system purchases. CASH also provides a unique potential for continuous operation of generation equipment. It is a multi-fuel technology, fully

compatible with natural-gas firing or with integrated gasification.

EPRI has obtained a patent on the CASH process, and is seeking partners to demonstrate its advantages.

Combustion Turbine Technology for Distributed Generation

EPRI has a very active effort in distributed generation. In fact, it is so important

Figure 4: Distributed generation can utilize a variety of combustion turbine sizes depending on the level of transmission and distribution resources served.

that a special Solution Team has been formed with participants from generation, delivery, and commercial technologies to deal with the cross-cutting issues raised by this technology. Figure 4 shows the relationship of generation technology sizes needed for various levels of distributed generation. Preliminary results from EPRI's studies show that:

- Gas-fired technologies can compete with bulk power in niche service situations

- Emerging fuel cells and advanced gas turbines have potential to be broadly competitive with bulk power
- Distributed generation must focus on areas with high value industrial and commercial customers

Distributed generation is most attractive in special service situations, such as high outage or hard-to-serve areas, alternate fuel opportunities, and environmentally sensitive areas. Transmission and distribution upgrade deferrals may be valuable in places, but may be limited. The main opportunities are for feeders, transformers, and substations. The primary feeder system is typically the cost-effective area for T&D deferral.

Integrated system planning of distributed generation is difficult, and the best plan is to adopt local area plans targeted around or near high-value customers.

F-Class Turbine User Needs

Utility managers and planners face a dilemma. The new F-Class technology offers lowest cost of electricity and emissions levels possible on gas, but there is significant risk because of the unproved performance of the new machines. EPRI's durability surveillance project gives EPRI investors access to information and products to mitigate the risks of applying this new technology. Durability surveillance is a joint effort between EPRI, the host utilities, and the original equipment manufacturers which is aimed at monitoring an early installation of each of the F-Class machines to verify performance and to discover and correct problems from peaking and base-load service.

The hot gas path is the heart of the

machine, and durability surveillance includes monitoring actual blade metal temperatures in real time using laser pyrometers, which provide real time temperature measurements of each blade on each pass. New low-NO_x pre-mixed combustion systems are particularly susceptible to pressure pulsations and those are monitored, in addition, to vibrations and thermal performance. The plant's performance is tracked using EPRI's E-MAP — efficiency monitoring and analysis program.

The operating history of the machine from these monitoring and diagnostics are combined with detailed hot gas path inspection data and analytical methods to develop models for life expenditure as a function of fuel, time and operating regime. Thus durability surveillance brings the advantages of collaborative R&D to bear on the users' bottom line by reducing the risk of applying new technology.

EPRI is involved in partnerships with users and manufacturers to ensure the durability reliability and availability of the present F-Class combustion turbines and the emerging G-Class. EPRI looks forward to the working in cooperation with the DOE ATS Phase IV demonstrations for both large and small sizes, building upon the experience and field smarts from the F and G experience.

In durability surveillance work, there are EPRI projects with Potomac Electric Power Company at their Station H, on a GE Frame 7F peaking turbine, with Florida Power and Light Company at their Martin Station, on four GE 7FA combined cycle plants, and with Jersey Central Power and Light on the GT24 machine of ABB — the first F-Class machine employing sequential combustion (or reheat) technology. EPRI is negotiating with Metropolitan Edison and

Siemens to apply durability surveillance on their Portland Station V84.3 peaking machine, and looking for EPRI member partners for the Westinghouse F and G machines. Also durability surveillance is being applied at the gasification repowering of PSI's Wabash River Plant on GE Frame 7FAs firing medium Btu coal gas.

EPRI's Material Center for Combustion Turbines (MCCT)

The MCCT, a joint EPRI/Southwest Research Institute initiative, performs materials research to develop, support and maintain life management products for optimizing the life of CT components, especially the hot section components. Emphasis is on degradation and life modeling of coatings and coated blading in advanced CT's in combined cycle applications. Life models are developed from laboratory testing, from field evaluations, from metallurgical analysis of service-run parts, and from mechanical testing and analysis of service-run parts. Results are then incorporated into upgraded versions of existing EPRI life management computer codes, such as REMLIF (Remaining Life) for nozzles and blades, and in new products, such as a failure atlas and a materials property database. In addition, the Center supports development of the other EPRI products and computer codes by supplying the necessary material data and behavior laws.

The SOAPP Workstation

The State-of-the-Art Power plant workstation is a computer application which allows the user to screen and assess the benefits of advanced combustion turbine technology for his particular site, design requirements, and financial parameters as

shown in Figure 5. The program incorporates levels of information. At the most elementary level are tutorials on topics such as dry Lo-NOx technology. This level of information can be used by technical interest groups such a public utility commission staff, and technical communicators. The program incorporates multiple levels of information. Tutorials are at the most basic level. This level of information can be used by technical interest groups such a public utility commission staff, and technical communicators.

Figure 5: Assessing the benefits of a technology such as Advanced Turbine Systems for a particular user requires evaluating it with regard to the users' particular fuel, site, and economics.

In addition to technology transfer functions, the program has many technical levels, which include heat and mass balances, plant layouts, construction schedules, and many others. An interactive cost evaluation, layout and schedule analysis is illustrated in Figure 6. The user has an economic summary screen, a plot layout screen, and a schedule screen open. As she adjusts various parameters such as discount factor the screens are automatically updated. This capability is particularly important for repowering studies, where many parameters interact to determine the over-all feasibility of an application.

Figure 6: Using the SOAP Workstation, a user would be able to conduct a detailed analysis of an Advanced Turbine System for her particular parameters such as fuel, discount rate, and site constraints. The result of such an analysis would be similar to this example. The analysis is fully interactive.

The SOAP CT Workstation can be applied to great advantage to technology transfer and market development for advanced combustion turbine technology.

EPRI USER NEEDS ASSESSMENT

EPRI is moving to be even more user oriented through stepping up and formaliz-

ing its role of collecting and organizing user needs. This effort has benefited in the past, and can benefit in the future from cooperative efforts coordinated between DOE, GRI and EPRI. EPRI has sponsored or is sponsoring, a number of workshops collectively or independently. Collecting user needs in a standard (but not rigid) format will serve the user community and manufacturer, institute, and government R&D efforts, including the

DOE ATS Program.

The process is to hold workshops and meetings, and collect and publish user consensus needs. An example of this is the Workshop on Small Combustion Turbines, co-sponsored with GRI and DOE, which is planned for early 1995. In addition, EPRI is planning a Workshops on:

- Hybrid Generation Technology involving the renewable resource and the fossil generation communities
- International workshop on F-Class technology involving those presently using, installing, and planning for this technology

As an example, the participants in the F-Class combustion turbine process would be the utility and non-utility developers, builders, and owners of plants. They would bring to the table their experience in operating existing equipment, including the F, G, and advanced aeroderivative machines which will be the precursors of the ATS. Manufacturers will be invited to provide valuable input and to review function for their technology, in the much the same way that EPRI now collects and disseminates information through the State-of-the-Art Power Plant initiative. Ultimately, this information would be incorporated into the SOAPP Combustion Turbine Workstation technology modules.

ADVANCED COMBUSTION TURBINE TECHNOLOGY COLLABORATIVE STRATEGIES

The bottom line is that the greatest benefit to be had from advanced combustion

Figure 7: Successful application of new combustion turbine technology will require collaboration between users, manufacturers, and entities such as DOE, EPRI, and GRI.

turbine technology will come about not only through new cycles, materials, control systems, and the like — though these are absolutely essential — but also through a constant and rigorous screening against market-driven user needs. As shown in Figure 7, collaboration will also be essential. Institutes like EPRI and GRI and agencies such as DOE can maximize that benefit by constantly reexamining their mission, by constantly communicating between themselves, manufacturers, and users, and by building bridges between the talents and technology of the former and the diverse needs of the latter.

BIBLIOGRAPHY

1. Armor, A. F., Touchton, G. L., Cohn, A.,
"Powering the Future: Advanced Combustion
Turbines and EPRI's Program", EPRI Coal
Gasification Power Plants Conference, San
Francisco, CA, Oct. 21-23, 1992.
2. Pace, S., Sopocy, D., et. al., "State-of-the-Art
Power Plant (SOAPP)," Sargent and Lundy
Engineering Conf., Chicago, IL, Spring 1992.
3. Ondryas, I. S., Pugh, J. E., Piulle, W.,
Touchton, G. L., "Performance, Operation and
Maintenance Management System
"POMMS": EPRI's Monitoring System for
Gas Turbines and Combined Cycle Plants",
IGTI-Vol. 9, p. 347, ASME COGEN-TURBO
Conf., Portland, OR 1994.
4. Van Laar, J. A., Bruns, G. A., "State-of-the-
Art Approach for Evaluating Repowering
Options", Sargent & Lundy Fossil Generation
Conference, Chicago, IL, Spring 1994.
5. Armor, A. F., Touchton, G. L., "Integrity of
Combustion Turbine Power Plants: The EPRI
Program", ASME COGEN TURBO POWER,
7th Congress & Exposition, Bournemouth,
U.K., Sept. 21-23, 1993

Luncheon Address: Regulatory Issues Facing Gas Turbine Facilities

**Eric Wong
California Energy Commission**

On behalf of California Energy Commission Chairman Charles Imbrecht, thank you for the invitation to speak today. Chairman Imbrecht asked that I address you on the important issues of electricity rates, regulatory reform, and industry restructuring.

The dramatic April 20, 1994 proposal by the California Public Utilities Commission to reform electric utility regulation produced a nationwide debate over the future structure of the electric utility industry and the role that increased competition and market forces should play in that industry. This debate is aided by the participation of some of the finest minds in the country -- nationally recognized academics, utility executives, consultants of international repute, potential market participants, and regulatory professionals from around the country. Notably, the FERC recently announced their desire to learn more about existing power pools and California's experience.

California is indeed the beneficiary of this impressive debate. We have witnessed the development of some extraordinary proposals for industry restructuring. While there has been a great deal of convergence in the last month of the fundamentals of a competitive market and the market mechanisms necessary to an operating pool, much more work needs to be done. The outcome may well become the model for reform throughout the nation.

When the California Public Utilities Commission's "Blue Book" was issued, we and other observers were skeptical that California

could simultaneously lower electricity rates for all consumers while maintaining the financial integrity of the investor-owned utilities, let alone do so by allowing direct access over a phased period without shifting costs between different customer classes. However, over the last few months, proposals have emerged that will not only allow retail customers to exercise choice fairly and quickly but simultaneously improve the efficiency of the wholesale market as well.

In 1985, Chairman Imbrecht testified before the FERC, pointing out the need for "wholesale wheeling" -- open, non-discriminatory access by all market participants to the transmission grid -- to allow increased wholesale competition. His testimony then recognized that expanding wholesale competition had long been stymied by two seemingly insurmountable obstacles:

- First, potential wholesale competitors could not get open, non-discriminatory access to critical facilities, such as transmission lines, in order to move their power over the interconnected grid.
- Second, transmission owning utilities under traditional cost-based regulation lacked any incentive to be "willing buyers" of their competitor's power or "a willing party" to upgrade their grid to provide additional capacity to third parties. The result was that competitors were denied opportunities to acquire and/or sell their power.

Transmission, access, market-based or incentive transmission service pricing, and market

power control were key reasons for the birth of the electricity title of the National Energy Policy Act of 1992. It comes as no surprise to any of you that one "short" sentence in EPACT would become the wellspring for the national debate that we are experiencing today. This sentence follows the act's prohibition on FERC to issue any order on mandatory retail wheeling. Surfacing as a savings clause, the act, through this sentence, appears to not "affect any authority of any state or local government under state law concerning the transmission of electric energy directly to an ultimate consumer". Some would say, EPACT declared this to be a State's right, not a Federal rights issue. Others have clearly articulated that congress never intended for states to have authority in this area. FERC has also asserted their authority over retail wheeling but diplomatically wants a Federal/State solution to this complex issue.

While the jurisdictional issues will likely end up in the courts, the national debate goes on. Ironically, whether and how to allow retail competition is producing a better solution to the wholesale market dilemma. First San Diego Gas and Electric and then Southern California Edison have developed a promising proposal that will in addition to allowing retail direct access, open up the wholesale market to expanded competition. This proposal or "POOLCO" works on the simplest of principles: if a new competitor can sell power to California at less than it now costs to generate electricity, that power generator should be guaranteed access to the grid and allowed to sell that less-expensive power to the utilities or their customers. And to make sure that beneficial wholesale and retail transactions occur for any market participant without discrimination, the utilities are willing to divest themselves of the operational centers that dispatch available transmission and utility-owned generation. That is the functional equivalent of divesting generation and transmission from two of the state's largest vertically integrated monopolies.

For too long, transmission-owning utilities claimed there was open access, but they controlled the terms and conditions under which access was granted, and more often, denied. Prying open this "black box" became absolutely essential to the creation of credible competitive markets. The most straight-forward solution to this problem is to create an independent, unbiased system operator to provide open, non-discriminatory, and comparable access to transmission and essential ancillary services. Even if nothing further is accomplished in California to implement direct access and customer choice, opening up the wholesale markets through the creation of an independent system operator would rate as a major achievement.

At this time I need to tip my hat to the efforts of the members of the Western Systems Power Pool (WSPP), who on a separate effort, are looking at enhancements to encourage further wholesale electricity trading within the WSPP. This project, known as "hub enhancements" and led by San Diego Gas & Electric, Southern California Edison, Pacific Gas and Electric and other WSPP utilities, is compatible with the retail direct access efforts of the POOLCO proponents and will be complementary to increasing competition in the active wholesale market of the Western US, Canada, and Mexico.

But there is no reason to stop at this impressive reform. The beauty of the POOLCO approach is that the concept that improves wholesale markets -- the independence of the system operator -- also facilitates customer choice and retail competition in ways that are both efficient for market participants and fair to all customer classes. You will hear in the ensuing months, both pro and con about POOLCO or the concept of an independent pool operator. There are both misconceptions and misunderstandings. I encourage you to be relentless in your questioning until you fully understand the facets of each proposal and why they will or not work.

For now, we are convinced that the merits of opening up the "black box" through an independent system operator -- the heart of the POOLCO proposal -- are unassailable and can withstand intense scrutiny.

We believe that the resulting electricity system would be at least as reliable and stable as today's electricity system, but the increased competition would likely drive down both wholesale and retail electricity prices. How soon and how much prices might fall, we do not know.

We at the Energy Commission are also seriously examining important ancillary issues associated with the consulting role of energy efficiency programs, resource diversity, and research and development in a competitive electricity market. Our approach will be to determine how best to obtain the societal benefits and economic efficiencies that are possible from these policy initiatives that have long been embodied in California's Statutes and Federal Law. At this time we are optimistic that there are workable solutions to all of these concerns and that an efficient, competitive electricity market can accommodate these solutions.

Thank you.

Session 2

Program Element Overviews

OVERVIEW OF WESTINGHOUSE'S ADVANCED TURBINE SYSTEMS PROGRAM

CONTRACT INFORMATION

Contract Number	DE-AC21-93MC30247
Contractor	Westinghouse Electric Corporation 4400 Alafaya Trail Orlando, Florida 32826-2399 (407) 281-3270 FAX (407) 281-5014
Contractor Project Manager	Ronald L. Bannister
Principal Investigator	Ihor S. Diakunchak
DOE Project Manager	Donald W. Geiling
Period of Performance	August 5, 1993 to November 4, 1994
Schedule and Milestones	

FY1994 Program Schedule

	O	N	D	J	F	M	A	M	J	J	A	S
Mgt. Plan	—											
NEPA Information	—	—										
Select GFATS			—	—	—	—	—	—	—	—	—	—
Conversion to CFATS					—	—	—	—	—	—	—	—
Market Study							—	—	—	—	—	—
System Definition								—	—	—	—	—
Integrated Plan						—	—	—	—	—	—	—
Component Development							—	—	—	—	—	—

OBJECTIVE

In cooperation with the U.S. Department of Energy's Morgantown Energy Technology Center, the Westinghouse Electric Corporation is working on an 8-year, four-phase program to develop and

demonstrate a highly efficient (greater than 60% plant cycle efficiency), environmentally superior (NO_x < 10 ppmv), and cost-competitive (busbar energy costs 10% less than current technology) utility Advanced Turbine System (ATS) for baseload utility scale power generation. The

natural gas-fired gas turbine must be capable of being adapted to operate on coal or biomass fuels. Availability is to be equivalent to modern advanced power generation systems, and the concept must be commercially available by the year 2000.

BACKGROUND

In Phase 1, Westinghouse concluded that a plant efficiency greater than 60% can be obtained with an advanced combined cycle, which includes an increased gas turbine firing temperature (2600°F), increased component efficiencies, and reduced cooling air usage (Little et al., 1993). In Phase 2, in which Westinghouse is currently working with DOE, gas turbine cycle efficiencies were reevaluated using established principles such as intercooling and recuperation, and newer concepts such as thermochemical recuperation and partial oxidation. Also, efficiency enhancements within the ATS cycle were evaluated to determine the best approaches to raising overall thermal plant efficiencies (Bannister et al., 1994). Within Phase 2, Westinghouse has developed a conceptual gas turbine design for an advanced combined cycle. Equipment requirements and technological barrier issues have been identified. This paper presents an overview of Westinghouse's current status on the ATS program.

PROJECT DESCRIPTION

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 reevaluate cycle efficiencies. Results of some of the analyses are summarized in this paper. In addition, efficiency enhancements within the ATS selected cycle have been evaluated. 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 (Briesch et al., 1994).

The Phase 2 Westinghouse ATS program is defined within eight major tasks. As indicated in the program schedule, Task 1 (Management Plan), Task 2 (National Environmental Protection Act [NEPA]), Task 3 (Select National Gas-Fired Advanced Turbine System [GFATS]) and Task 4 (Conversion to a Coal-Fueled Advanced Turbine [CFATS]) have been completed. The remaining tasks have been started including the gas turbine flange-to-flange conceptual design within Task 6.

CYCLE STUDIES

Baseline Cycle

In order to evaluate different technologies and concepts applicable 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 was modeled to provide a baseline cycle. The combustion turbine rotor inlet temperature (RIT) was set at 2600°F to approximate near-term temperature capabilities. Compressor pressure ratio was set at 18. High pressure steam conditions entering the steam turbine were specified at 1450 psi and 1000°F, and the hot reheat steam temperature was also set at 1000°F. 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 feedwater recirculation flow.

Steam Cycle Enhancements

The basic reason for raising the steam pressure and temperature of the Rankine cycle is to improve the potential thermal efficiency. The optimum steam cycle was determined to be at 1800 psi with 1100°F high pressure superheat steam and 1100°F reheat steam (both 100°F over the baseline temperature). This results in a 0.5 percentage point increase in net plant thermal efficiency.

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 is a detriment to cycle efficiency. First, air is ejected from the turbine air foils earning a disruption in the surrounding flow field. Secondly, the cooling air is ejected from the airfoil into the gas path which results in irreversible pressure losses due to non-ideal mixing. The third loss mechanism is caused by the reduction in gas path temperature which reduces the work output of the temperature. Finally, the turbine cooling air must be pumped to pressures significantly higher than gas path pressure to assure that cooling flow rate is sufficient.

By using closed-loop steam cooling, the loss mechanism can be eliminated while still maintaining turbine material temperatures at an acceptable level.

Higher Compressor Pressure Ratio

Aircraft gas turbine engines are designed with high overall pressure ratio to maximize simple cycle efficiency. For Westinghouse engines the simple cycle efficiency curve is relatively flat above a pressure ratio of approximately 40. In a

combined cycle, steam cycle efficiency decreases with increasing gas turbine pressure ratio due to the reduction in gas 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.

Compressor Intercooling

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. 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 amount of extra fuel energy added is about equal to the simple cycle efficiency. Intercooling adds output at approximately the simple cycle efficiency.

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.

Combined cycle efficiency is 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. Installation of a recuperation system on a the baseline configuration results in an increase in thermal efficiency of one percentage point.

Other Cycles

Some of the other innovations studied included use of a reheat combustion turbine, design of a partial oxidation combustion turbine and the concept of thermal chemical recuperation. The analytical results for these cycles were interesting, but the concepts are not needed to obtain an ATS thermal cycle efficiency greater than 60 percent.

ADVANCED TURBINE SYSTEM

The cycle analyses discussed in this paper were used along with an in-depth evaluation of emissions, cost of electricity, reliability-availability-maintainability, and program schedule requirements to select the best ATS. Features of the Westinghouse single-shaft combined cycle include:

- Advanced Aero/Heat Transfer/Materials Technology
- Burner Outlet Temperature in Excess of 2700°F
- Closed-Loop Steam Cooling
- Single Crystal and Directional Solidified Airfoils

- Improved Thermal Barrier and Anti-corrosion Coatings
- Ceramic Ring Segments
- Active Tip Clearance Control
- Brush Seals
- Reduced Compressor Dump/Combustor Loss
- Increased Row 4 Blade Exit Area
- Reduced Inlet and Exhaust Losses
- Fuel Preheating
- Advanced Steam Turbine Design Technology
- 1800 psi/1100°F/1100°F Steam Cycle

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 (DCFT). 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 pressurized fluidized bed combustion (PFBC). The rationale for these selections is summarized in this section.

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 state 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 (AFBC) and pulverized coal combustion boilers with flue gas desulfurization (PC/FGD). The air-blown IGCC technologies with hot gas cleaning and second generation PFBC appeared to have the greatest thermal performance and cost potential (Newby et al., 1995). Their respective status of development is categorized as early demonstration which is suitable for the ATS program.

CONCEPTUAL DESIGN OF ATS ENGINE

The ATS engine will be the next model in the series of large heavy duty combustion turbines developed by Westinghouse. Westinghouse combustion turbine genealogy began in 1943 when the first wholly American designed and manufactured jet engine went on test (Scalzo et al., 1994).

From the first industrial turbine introduced in 1949 (1.3 MW) to the 501G (230 MW) introduced in 1994, improvements in turbine performance and mechanical efficiency were both evolutionary, where small improvements were made continuously, and revolutionary, where new designs were produced to achieve a large step change in engine output power and efficiency, as well as mechanical integrity and reliability. Since 1949, Westinghouse has made the technological advances, as required, in the areas of combustion, aerodynamic design, cooling design, mechanical design, leakage control, and materials to advance combustion turbine power generation for simple cycle and combined cycle applications.

When Westinghouse started the ATS Program in 1991, the fifth generation combustion turbine

(501F) was rated at 160 MW with a combined cycle thermal efficiency of 54 percent. As part of the Westinghouse ATS Program, the 501G engine was introduced that has a combined cycle efficiency of 58 percent.

Information on the 501G and a discussion of the component development programs under way to support the ATS engine are given in a comparison paper presented at the ATS Annual Review Conference by I. Diakunchak entitled, "ATS Program, Phase II Development Projects."

CONCLUSIONS

This paper presents an overview of the Westinghouse ATS Program. The single-shaft combined cycle plant conceived by Westinghouse during this Program will meet or exceed the goals of the DOE-sponsored program. The 501G just introduced by Westinghouse shows early influences of ATS. Some key components to support the ATS program have already been evaluated and incorporated into advanced design processes.

ACKNOWLEDGEMENT

The work described in this paper is being conducted under a DOE sponsored program, Advanced Turbine Systems Program-Conceptual Design and Product Development, 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.

REFERENCES

Bannister, R. L., Cheruvu, N. S., Little, D. A., and McQuiggan, G., 1994, "Development of Requirements for an Advanced Gas Turbine System," ASME Paper 94-GT-388.

Briesch, M. S., Bannister, R. L., Diakunchak, I. S., and Huber, D. J., 1994, "A Combined Cycle Designed to Achieve Greater Than 60 Percent Efficiency," *Modern Power Systems*, Oct., pp. 23-28.

Little, D. A., Bannister, R. L. and Wiant, B. C., 1993, *ASME Cogen Turbo Power '93 Proceedings*, IGTI - Vol. B, pp. 271-280.

Newby, R. A., Archer, D. H. and Bannister, R. L., 1995, "Conversion of an Advanced Natural Gas-Fueled Combustion Turbine to Coal-Base Fuel Applications," to be presented at the 1995 ASME Turbo Expo Conference, June 5-8, Houston, Texas.

Scalzo, A. J., Bannister, R.L., De Corso, M., and Howard, G. S., 1994, "Evolution of Heavy-Duty Power Generation and Industrial Combustion Turbines in the United States," ASME Paper 94-GT-488.

**Advanced Turbine Systems Program
Industrial System Concept Development
D. A. Rohy**

CONTRACT INFORMATION

Contract Number DE-AC21-93MC30246

Contractor Solar Turbines Incorporated
2200 Pacific Highway
P.O. Box 85376
San Diego, CA 92186-5376

Contract Project Manager Karl W. Karstensen

Principal Investigators	Graham J. Benjamin	Michael D. Stephenson
	Richard S. Brent	Luke H. Cowell
	David J. White	Michael D. Fitzpatrick
	Joel S. Otsuka	Jane F. Simpson
	John F. Talbott	Fritz Langenbacher
	Chaur S. Wen	

DOE Project Manager Charles T. Alsup, Jr.

Period of Performance August 3, 1993 to November 30, 1995

Schedule and Milestones

FY94 Program Schedule

Task	O	N	D	J	F	M	A	M	J	J	A	S
1. Project Plan	XXXX											
2. NEPA Report	XXXX	XXXX										
3. System Selection	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX					
4. Coal Conversion	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX					
5. Market Study		XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX
6. System Definition		XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX
7. Program Plan												
8. Critical Components							XXXX	XXXX	XXXX	XXXX	XXXX	XXXX

OBJECTIVE

The U.S. Department of Energy (DOE) has set new standards for industrial gas turbines through the development of the Industrial Advanced Turbine System Program. Their leadership will cause the most energy efficient and environmentally friendly power system in this size class to be developed for all industrial market needs. Already the DOE has created a positive effect by encouraging gas turbine system manufacturers to reassess their product and technology plans using the new higher standards as the benchmark. The results of this program should lead to the U.S. based gas turbine industry maintaining international leadership with a resulting increase in high paid jobs.

Solar Turbines is pleased to be able to participate in this program. In a parallel paper at this conference the details of our ATS Phase II program are discussed. This paper will describe Solar's approach to meeting the DOE objectives for the ATS Phase III program.

BACKGROUND

The DOE has stated in its request for ATS Phase III proposals, "...Commercial Implementation of the innovative technology is the principle goal of the Solicitation...". The required program goals are:

- Efficiency - A 15 percent improvement compared to the best technology in its class, available in 1991.
- Environment - Emissions will be lowered to achieve acceptance in severe non-attainment areas.
- Fuel Flexibility - Natural gas-fired systems must be adaptable to future firing with biomass and coal derived fuels.

- Cost of Power - Busbar energy costs should be 10 percent less than current state-of-the-art turbine systems, meeting the same environmental requirements.
- Reliability, Availability, and Maintainability - Equivalent to current state-of-the-art turbine systems.

Maximizing Program Value

The ATS program is designed to foster national goals of (1) global competitiveness of U.S. industry, (2) public health benefits, and (3) the expansion and retention of industrial employment through the higher efficiency of affordable electric power. Other components of national value include increased product exports and reduced fuel imports. To achieve these goals the industrial partners must maximize the potential market volume, the market penetration by ATS technologies, and the technical performance gains. Market penetration has two components: market acceptance and time to market. Solar has taken each of these elements of program value into consideration when developing its plans for its ATS offerings.

ATS Markets

Market Size. The total market size for industrial gas turbine systems has doubled since 1985. this growth in market size is expected to continue at least through the year 2005 with a reduced growth to the year 2020. Acceptance of gas turbines into the growing power market is due to high availability, low initial cost, short delivery times and the capabilities for dual fuel operation and unmanned operation.

The potential ATS applications within this large market are summarized as:

- **Electric power generation:** Industrial ATS can be used as a prime mover in an electric power generation system to meet the demands of electric utility customers. The owners and operators of ATS electric power plants can be investor-owned utilities, rural electric cooperatives, municipal utilities, and independent power producers. In the broadest context the ATS can be used in both traditional (i.e., baseload and cycling) and new (i.e., distributed generation) applications. The continued growth of electricity demand at or above the nation's Gross Domestic Product (GDP) through the remainder of this decade and beyond will create major opportunities for the ATS owing to its high efficiency, low capital cost, low prices for natural gas, short lead times for installation, and compliant environmental emissions.
- **Oil and natural gas production, transmission and storage:** These industry sectors, which depend extensively on pumping equipment and systems, represent clear opportunities for the ATS with a capacity between roughly 5,000 hp and 30,000 hp. The ATS, with its efficiency of nearly 45 percent, its low capital and maintenance cost, and its compliant emissions performance is a very competitive option for the oil and natural gas industries.
- **Marine propulsion:** Propulsion applications in the commercial "fast" ferry and "fast" freight merchant marine industries appear to be the greatest opportunities for the ATS in marine applications, since these applications emphasize fuel efficiency and power density. These applications are designed to use liquid fuels.
- **Industrial prime movers:** The ATS can be used for industrial manufacturing power

generation to meet on-site plant requirements, including mechanical shaft power for compressors and pumps in petrochemical and other process energy intensive applications, and in electric/thermal cogeneration to serve a portion of on-site electric and process steam demands in manufacturing plants which depend extensively on low-cost reliable electricity supply. Because the ATS will have system efficiencies approach 45 percent, the amount of thermal energy contained in the engine exhaust is relatively small compared to traditional cogeneration systems. Consequently, an ATS cogeneration system (e.g., unfired waste heat boiler) will have a higher electric to process steam production ratio (i.e., 5:1 versus 2:5) than traditional systems using less efficient gas turbine (e.g., less than 30 percent). Duct firing can be used to augment thermal energy output when required.

As currently designed, the ATS has many attributes (e.g., low emissions, high availability) which will make it equal, if not superior, to most current and advanced gas-fired power systems available in the market place after the year 2000, the projected cost performance clearly distinguishes it advantages in contrast to its expected competitors.

Product Characteristics and Market Buying Criteria

Industrial prime mover applications, in general terms, have the following characteristics:

- Prime movers are dispersed at or close to the site where their output is used.
- The prime mover, and often the entire site, is either unmanned or manned by personnel whose primary assignment is elsewhere or unrelated to the prime mover.

- The output of the prime mover is a critical element in the production of the site product but, unlike production machinery, must be transparent to the user.
- The prime mover is a significant cost element to the user.

These application characteristics will combine to produce a set of buying criteria similar to the set shown in Figure 1. These criteria vary somewhat from industry to industry and trends can be observed with time. Several such trends can be noted in Figure 1. For example, emissions as a buying criteria, is growing rapidly in importance as regulatory activity increases. In non-attainment areas, emissions already heads the list, but still does not necessarily overpower RAMD or cost. In the present economic milieu, first cost ranks very high but could yield to lifetime cost where fuel costs become high either in their own right or as a result of a tax on fuel usage.

Market Penetration

ATS program benefits will be maximized by ensuring market acceptance and getting the product to market early. These two issues combine to maximize the penetration of the ATS benefits into the marketplace. The markets discussed in the preceding section show a common characteristic - risk aversion. While the benefits of new technology are seductive, each wants to see proof of the technology at "another site". Solar's approach to this issue is to involve customers in the evaluation of the technologies early in the program. Because of Solar's large fleet of existing gas turbine systems (over 9,000), technologies can be introduced singularly and early into the field to gain acceptance. The process also ensures that early benefits are realized from each technology.

Two examples of this process are Solar's dry low emission technology and the DOE/Solar ceramic retrofit components. Dry low emissions were

Figure 1. Buying Criteria

RE94238M

introduced into the marketplace in 1993. They were designed for both current product and for retrofit into existing units. As of this date both markets have accepted this technology and orders are evenly split. Acceptance has been achieved by careful attention to the customer needs. A similar process is planned for ceramic components. These components are being designed for retrofit into Solar's Centaur 50. Once proven in a field test engine, the components will be introduced to both new and retrofit markets.

Customers look to Solar for upgrades to their equipment at the time of overhaul and understand the increase in value to their operations.

Another facet of market acceptance is the company's reputation to stand by the customer. Solar's extensive field service network and service tools ensure that any problems encountered during the introduction of new technology are promptly resolved.

Technical Performance Improvement

The market characteristics and customer buying criteria are essential inputs to the decision on product characteristics. Solar performed an extensive QFD (Quality Functional Deployment) analysis to quantify the customer needs in the industrial marketplace. A second set of key inputs came from the program goals, especially early commercialization of ATS technologies. Commercialization implies market acceptance. Thus, Solar evaluated what could be placed into the marketplace by the year 2000. Evaluation of numerous cycle against these criteria led Solar to select the optimized, recuperated gas turbine as the drier for its advanced turbine system. The selected turbine meets or exceeds all program goals. The technologies employed have or will be individually tested in full scale tests reducing the technical risk.

The optimized recuperated gas turbine is designed to provide the highest efficiency and lowest emissions. It is not a simple cycle engine with a recuperator added to it. The pressure ratio, firing temperature and cooling flows are carefully selected to meet program goals. The resulting gas turbine has a shaft efficiency of 45 percent (LHV) and an emission goal of 5 ppmv (15% O₂) on natural gas fuel. This performance is possible because of the incorporation of the following technologies:

- High efficiency compressor and turbine aerodynamics.
- High effectiveness recuperator.
- Ceramic combustor liner and first stage turbine blade.
- Advanced thermal barrier coatings.
- Advanced single crystal alloys.
- Catalytic combustion.

Industrial customers also require high thermal efficiency over a broad load range. The Solar ATS will achieve this with a variable area turbine nozzle (VATN). With this component the engine delivers almost flat heat rate over the 50 percent to 100 percent load range. A second advantage of the VATN is the flat or constant combustor firing temperature over the same load range. Because of this characteristic it is easier to maintain low emissions over the same load range.

Advanced Turbine Systems includes more than the gas turbine driver. Systems include air inlets, exhaust silencers, gear boxes, generators, lube oil systems, fuel gas compressors, and controls. Each of these items have an effect on system efficiency, cost and availability. Some have an effect

on air quality because of fugitive emissions. Calculations on a typical gas turbine system shows that 8 percent of the engine output is not available at the busbar. Solar has addressed each item in the system reducing its parasitic power consumption and fugitive emissions. When these technologies are implemented the optimized recuperated gas turbine system should have ~95 percent of shaft energy available at the busbar.

SUMMARY

The Department of Energy has set new standards for industrial gas turbine systems. These aggressive goals are designed to meet customer and regulatory needs in the year 2000. Solar, after thoroughly examining all customer needs and integrating with ATS program goals, has chosen an optimized recuperative gas turbine system. All components in the system are being addressed to meet and exceed all goals and customer needs.

METC Combustion Research Facility

John S. Halow
Daniel J. Maloney
George A. Richards
Morgantown Energy Technology Center
Morgantown, WV 26505

OBJECTIVES

The objective of the Morgantown Energy Technology Center (METC) high pressure combustion facility is to provide a mid-scale facility for combustion and cleanup research to support DOE's advanced gas turbine, pressurized, fluidized-bed combustion, and hot gas cleanup programs. The facility is intended to fill a gap between lab scale facilities typical of universities and large scale combustion/turbine test facilities typical of turbine manufacturers. The facility is now available to industry and university partners through cooperative programs with METC.

BACKGROUND

Consistent with the energy policy act, a major DOE program, the Advanced Turbine Systems (ATS) Program has been initiated to develop more efficient gas turbine systems for both utility and industrial electric power generation (U.S. Department of Energy 1992). The program targets base load power systems for commercial offering in the 2000 to 2002 time frame. The program will target a 15 percent improvement in efficiency, reductions in NO_x , CO_2 , CO, and unburnt hydrocarbon, and provide for a 10 percent lower cost of electricity than current technology.

The basic strategy of the program is to fund teams led by U.S. turbine manufacturers to develop ATS while in parallel to develop a technology base to address key technical issues. A university/industry consortium administered by the South Carolina Energy Research and

Development Center will provide cross-cutting technical support for the ATS program. Under this consortium, the two rounds of research proposals were selected through a competitive process. Other activities including projects in the METC combustion facility will contribute to this technology base development of the ATS program.

High pressure combustion research is also important to other DOE programs. Integrated gasification combined cycle (IGCC) systems and second-generation, pressurized, fluidized-bed combustion (PFBC) systems use gas turbines/electric generators as primary power generators. The turbine combustors play an important role in achieving high efficiency and low emissions in these novel systems. These systems use a coal-derived fuel gas for the turbine combustor. The METC facility is designed to support coal-derived fuel gas fired combustors as well as the natural gas fired combustors.

PROJECT DESCRIPTION

In 1990, an existing METC building, Building 6, that had been used to dry and prepare coal for the METC gasifier and other projects was decommissioned. The need for a modern, high-pressure combustion facility was recognized and the decision was made to modernize and upgrade B-6 to meet this need.

The primary considerations that dictated the renovations included safety in the event of an accident, the ability to remotely operate the test rigs, availability of substantial volumes of high

pressure preheated combustion air, availability of high pressure methane and other fuels, and a modern high speed data acquisition system. The renovations required to meet these needs were extensive. These included addition of blast walls to create isolated test cells, a separate control room, installation of a large ventilation system capable of reducing the risks from accidental release of gaseous fuels and dissipating the heat released by the combustors, installation of the needed high pressure utilities to the test cells, addition of a modern data acquisition system, and numerous smaller changes to the building.

The renovations were completed in January of 1993. A pressurized pulsed combustor began operating in July and several others are expected to be in operation by the end of 1994.

BUILDING LAYOUT

Figure 1 shows the physical layout of the facility which consists of two primary test cells,

areas 103 and 105, a staging area, fuel handling room, control room, and a clean area. The test cell 105 is a high bay area with a ceiling height of 30 feet. It is capable of holding at least two test units. Test cell 103 has a ceiling height of 20 feet on the north side and a height of 10 feet on the south half. The staging area is used to assemble combustors for tests and to do minor repair and fabrication tasks. It is also used as a test area for cold flow visualization models and other small ambient pressure and temperature rigs. The distributed control system in the control room will be used to remotely operate rigs that are in either the 103 or 105 test cells. The common computer system is readily configured to control the various rigs. The fuel handling room was originally intended for use with coal water slurries and liquid fuels but currently it houses a methane compressor and some auxiliary equipment.

Figure 1. B-6 Combustion Facility

UTILITIES

Table 1 summarizes the utilities and services available in the main test cells in the facility. Basically the facility was sized to support a 10,000,000 Btu/h combustion process. One such process or a combination of smaller processes could be serviced at one time. Up to 160,000 scf/h of air is available for combustion, atomization and other needs. High pressure water is available as city, deionized or process cooling water.

High pressure air entering the building passes through a manifold where a portion or all of the air can be diverted to preheaters. The preheaters can deliver air at 700 psig at 1,050 °F. Preheated air is metered and then can be delivered to either of the two high pressure cells.

CURRENT PROJECTS

Currently two combustion rigs are operating in B-6 and one additional project is under construction for the facility. Space is available in the test cells for at least one additional test rig.

A pressurized pulsed combustor began operating in July of 1993 in the 105 test cell. The combustor will carry out pulsed combustion of natural gas at pressures up to 10 atmospheres. Figure 2 illustrates the components in this combustor. The objective of the test program is to test various pulsed combustor geometries to determine pressure gain performance, high pressure operating characteristics, evaluate

formation of pollutants, and obtain information on scaling laws for this class of combustors. The information obtained will be used to evaluate the potential of pulsed combustors for achieving pressure gain in gas turbine systems.

A high-pressure steady flow rig is currently completely fabrication. The objective of this rig is to test novel, steady-flow, pressurized combustors that produce very low NO_x and other emissions. Figure 3 is a schematic of the steady flow rig. The rig will be able to test burners in the range of 1 to 10 million Btu/h. It has a modular design that allows alternate burners to be installed in the rig.

An evaporation rig currently is in startup in area 105 of B-6. Figure 4 illustrates the basic components of this rig. This rig will test the concept of water injection in an externally fired cycle. The specific technical issue that the unit will address is evaporation rates of water droplets in high pressure flows.

FUTURE WORK

The METC B-6 combustion facility is a modern laboratory with sufficient ancillary equipment and utilities to simultaneously support multiple combustion projects. METC plans to work cooperatively with industrial and university partners on these projects.

REFERENCES

"Report to Congress, Program Plan, Advanced Turbine Systems," U.S. Department of Energy, 1992.

Table 1. Building B-6 Services

Air Supply -- Total of 160,000 scf/h		
• 700 psig combustion air @ up to 1,050 °F		156,000 scf/h
• 900 psig atomization air		5,700 scf/h
• 700 psig purge air		
• 125 psig plant air		
Water Systems		
• 500 psig city water		12 gal/min
• 500 psig deionized water		5 gal/min
• Process cooling water (closed loop)		250 gal/min
• Low pressure steam		1,000 lb/h
• 90 psig city water		
Fuels -- Up to 10,000,000 Btu/h		
• 600 psig natural gas		700 scf/h
	(10,000 scf/h available 3/95)	
• 600 psig synthesis gas @ 1,500 °F		Planned
Gas Analyzers		
• Mass spectrometer		
• O ₂ /CO analyzers		
• Gas analyzer (CO ₂ , NO _x , HC, H ₂)		
• N ₂ O analyzer		
• H ₂ O analyzer		
Control and Data Acquisition		
• Low speed: 100 channels @ 0.5 Hertz		
• High speed: 8 channels @ 100,000 Hertz		
Safety Features		
• Ventilated test cells		
• High pressure nitrogen		
• Low pressure nitrogen		
• Video monitoring systems in test cells		
• Remote control and operation of rigs		
• Blow out panels room 105		
• Toxic gas monitors and alarms		
• Combustible mixture monitor and alarm

M93052677

Figure 2. Pressurized Pulse Combustor

M93001812E

Figure 3. Pressurized Burner Test Facility

Figure 4. Evaporation Unit

A Collaborative Vision: The Advanced Gas Turbine Systems Research Program

CONTRACT INFORMATION

Contract Number	Cooperative Agreement DE-FC21-92MC29061
Contractor	South Carolina Energy R&D Center Clemson University 386-2 College Avenue Clemson, SC 29634 803-656-2267 803-656-0142 (fax)
Contract Project Manager	Daniel B. Fant
Principal Investigator	Lawrence P. Golan
DOE Project Managers	Dr. Jack Halow/Dr. Norm Holcombe
Period of Performance	September 30, 1992 to Year 2000 (and beyond)

OBJECTIVES

The Advanced Gas Turbine Systems Research (AGTSR) program is an ambitious, industry-driven research program with several objectives in mind:

- establish the technology-research base to support the Advanced Turbine Systems (ATS) prototype program for both industrial and utility applications and actively seek to transfer research to useful technology for ATS and future growth power generation systems that utilize stationary gas turbines in a combined-cycle -- transferring research does not mean to have upper management simply endorse it, but to develop long-lasting and on-going collaboration with industry at the working level.
- enhance turbomachinery-power generation research activities at American universities and

promote industry-university research collaborations -- with professors and their students developing projects focused on industry-defined needs.

- enhance and develop new power system course development - applicable to stationary gas turbines, combined cycle and cogeneration areas - and sponsorship of undergraduate, first-year graduate design and research activities, such as co-ops and internships at industry labs or in-house METC, DOD and NASA facilities.
- co-sponsor seminars and workshops that directly contribute to the needs of the ATS and AGTSR programs.

From a global standpoint, AGTSR is a synergistic program that has begun to spark a strong university-industry alliance, enabling closer ties and interactions with respect to focused research activity and new educational

opportunities to promote the next frontier of land-based gas turbines, enhance U.S. competitiveness in the global market, and prepare us to meet 21st century demands for advanced electrical power generation systems.

BACKGROUND INFORMATION

The AGTSR program was initiated by a collaborative research vision among government, industry and academia with a vested interest in expanding the U.S. R&D base to enable advanced power generation systems and higher efficiency stationary gas turbines. AGTSR is an integral part of the ATS program and is being supported by DOE's Morgantown Energy Technology Center (METC) through a cooperative agreement, started in September, 1992, with the South Carolina Energy R&D Center (SCERDC). SCERDC is responsible for organizing, administering and managing the AGTSR program which is defined as a university/industry consortium that is expected to continue past the year 2000. The program was developed at an opportune time as the projected growth of power needs in the U. S. is substantial, however gas turbine research funding within American universities is dwindling. In the United States, funding for enabling technologies is at an all-time low with the main focus directed toward critical technologies to address specific problems on a fairly short-term basis. At best, this approach to research will foster evolutionary design changes only, that is, near-term improvements resulting from incremental changes in existing designs. The AGTSR program is industry-driven but nurtures both evolutionary and revolutionary design ideas, either at the system or component level, with a strong advocacy for university-industry interactions established from the very onset of

any research endeavor.

PROJECT DESCRIPTION

ATS prototype development, industry, government, American universities and the U.S. economy as a whole will benefit significantly from the AGTSR program. New utility turbine systems will be extremely efficient and environmentally superior. Energy costs are envisioned to be at least ten percent less than current state-of-the-art turbine systems. AGTSR Request For Proposals (RFP) are solicited on an annual basis with industry defining and prioritizing research topics that are extremely relevant to their needs and considered high payoff technology areas from the designers' perspective. The Industry Review Board (IRB) reviews the university proposals and selects superior ones for subcontract. Proposal evaluation criteria consist of: technical merit and relevance to ATS goals; engineering and educational value, and leveraging of existing facilities; timeliness of results, transition ideas and collaboration with industry; and qualifications and gas turbine related experience of research team. The participating industrial members are Allison Engine Company, General Electric, Solar Turbines, TurboPower and Marine Systems Division of United Technologies, and Westinghouse Power Generation. Other members are the Electric Power Research Institute and the Gas Research Institute which represent the electric utility and industrial market segments that will purchase the advanced design gas turbines developed under the ATS program. The IRB is a very important facet of AGTSR, as it not only guides the technical thrust of the program but facilitates university-industry collaboration and insures that university research studies make relevant technology contributions to

the ATS program.

Research work by the university participants will be performed on existing or upgraded facilities, so program funding will be spent mainly on R&D activities. The work will generally be non-proprietary, with intellectual rights retained by the performing universities. Participation in the program is open to accredited engineering schools in the United States. Membership requires that the university have some experience and capability in the gas turbine and/or power generation areas, and that the university administration indicates support with respect to teaching release time for their faculty's involvement with the program. There are no membership fees for university performing members.

RESULTS

As of September 1994, sixty-five universities have been granted performing member status, representing 34 states and 5 major cost-sharing corporations. To date, there have been two RFP announcements with a total of 95 proposals submitted for competition. There are now 23 research subcontracts in place, ranging from turbine heat transfer phenomena, materials for thermal barrier coatings, to combustion modeling efforts. The educational part of the AGTSR program motivates students and nurtures industry co-ops to work more closely with companies and gain real-world engineering and design experience. At present, three trial programs are underway, one each at the University of Minnesota, Clemson and Clarkson University, dealing with undergraduate honors research projects. New program directions and initiatives are established by means of focused workshops, such as Vanderbilt's recent

combustion meeting and an upcoming heat transfer workshop scheduled for March, 1995 at Hilton Head Island, SC.

Research results from the AGTSR program are made available through quarterly technical progress reports and two-page executive summaries which are distributed to the IRB members -- copies of full reports are kept on file at SCERDC and may be requested by the IRB on an as-needed basis. The executive summaries are very useful to the IRB as they help them stay abreast of new ideas, accomplishments, capabilities and/or collaborations. Annual contractor meetings are also required to review the AGTSR university-industry consortium in detail.

Research topics for the third RFP are in the process of being deliberating with IRB and METC. This RFP will be more specific than the previous two and will emphasize high priority topics pertaining to stationary gas turbines and the combined-cycle area. As identified at the Vanderbilt Combustion Workshop, combustion instability may be a high-payoff research area for this round and potential research issues under this topic were identified by METC and are currently being reviewed by the IRB. If combustion stability is included as a topic, we would encourage university and industry to build research paths for larger scale combustion testing, taking advantage of in-house combustion research facilities at METC. One or two other topics will also be defined -- possibly in aerodynamics - efficiency losses, or advanced cycle analyses.

The dates for the AGTSR Advanced Heat Transfer Workshop have now been finalized. It will take place at Hilton Head Island, SC on March 1, 2 and 3, 1995. The session organizers are Drs. James Leylek, James Liburdy and Ting Wang of Clemson University. In addition to the power generation community, we are seeking DOD and NASA participation related to their in-

house heat transfer research, facilities and research issues pertaining to advanced aeroengine development. We are also planning to invite a high-level DOE-METC representative to give a keynote dinner address on ATS and future METC energy programs.

SCERDC has also submitted an AGTSR technology brief for the 1995 IGTI Technology Report, which is distributed worldwide and is a great way to advertise and promote the AGTSR program. Four AGTSR "Update" Newsletters have been published since the start of the program and were distributed to all university and IRB members, METC and interested DOD and NASA personnel.

FUTURE WORK

Future activities include changing our budget and progress report requirements for new and existing university awards, expanding our educational program and IRB membership to include small and moderate size companies, and nurturing more collaboration with industry; possibly at the international level with respect to advanced combined-cycle utility applications.

SCERDC would like to expand the educational "mission" of the AGTSR program. The three trial programs that recently got started will be presented and discussed during the AGTSR Educational Panel at the ATS review in November, 1994. A lot of fresh ideas should emerge from this panel meeting to give us the impetus to "grow" this part of the program in the future, possibly calling it the AGTSR Fellowship Program. This should be a "low-maintenance, low-paperwork" addition, as too much administrative paperwork would detract from the "mission" deterring professors from participating in the program. Indirect costs should not be

applied to the "educational" part of the program; or at least kept to a minimum, such as a 10 percent flat rate.

In the same vein, we would also like to change AGTSR budget and progress report requirements to possibly quarterly financial invoices and semi-annual technical reports. One report after six months and an annual report plus the ATS annual review should be adequate to highlight, describe and discuss any new research collaborations, transitions and significant accomplishments.

SCERDC is interested in pursuing the idea of a special RFP announcement for HBCU (Historically Black Colleges and Universities) schools, in collaboration with Oak Ridge National Laboratory and DOE-METC. An additional program for HBCU schools is an excellent way to ensure their participation in research and educational activities related to the ATS-AGTSR program.

AGTSR would also like to nurture more research "transition" paths, both horizontally and vertically -- where "horizontal" refers to university-to-university interactions and "vertical" (the high payoff transition) refers to university-to-industry collaboration where new design ideas, research results and concepts are incorporated into industrial technology programs -- such as the ATS prototype program!

The AGTSR program is scheduled to continue through the year 2000. The planned funding profile will permit issuance of yearly RFP's to address all the important technology and research barriers identified by the IRB.

REFERENCES

Abhari, R., CALSPAN, Private Communication.

Allen, R. and P. Micheli, "The Advanced Gas Turbine Systems Research Program: A Model for Industry, University and Government Cooperation", IGTI, The Hague, Netherlands, June 1994.

AGTSR, *Combustion Technology Workshop Proceedings*, Vanderbilt University, February 1994.

Bajura, R. A., H. A. Webb, and E. L. Parsons, "A Proposed Plan for an Advanced Turbine Systems Program," In *Proceedings of Workshop to Define Gas Turbine System Research Needs--II*.

Goldstein, R. and T. Simon, University of Minnesota, Private Communication.

LaFleur, R., Clarkson University, Private Communication.

Mellor, A. M., Vanderbilt University, Private Communication.

Report to Congress--Comprehensive Program Plan for Advanced Turbine Systems, July 1993, Report Number DOE/FE-0279.

Webb, H. A., and R. A. Bajura, "Development of Advanced Turbine Systems Meeting Tomorrow's Needs," presented at the Energy Daily Conference: *Advanced Combustion Turbines: Looking Past the 90's*, June 25-26, 1992, Washington, DC.

Ng, W. and W. O'Brien, Virginia Polytechnic Institute, Private Communication.

**MATERIALS/MANUFACTURING SUPPORT ELEMENT
FOR THE
ADVANCED TURBINE SYSTEMS PROGRAM**

CONTRACT INFORMATION

Contract Number	<u>ED 32 01 00 0</u> Field Work Proposal
Contractor	Oak Ridge National Laboratory P. O. Box 2008 Oak Ridge, TN 37831-6065 615-574-5150 615-574-6098
Contract Project Manager	Michael A. Karnitz
Principal Investigators	Michael A. Karnitz Eugene E. Hoffman William P. Parks
DOE Project Manager	Eugene E. Hoffman
Period of Performance	July 1992–September 2000

ABSTRACT

The Technology Base development portion of the Advanced Turbine Systems (ATS) program contains sub-elements which address generic technology issues for the advanced turbine system. One sub-element is the materials/manufacturing technology program to be directed by Oak Ridge National Laboratory, with work to be performed predominantly by industry. This sub-element includes materials issues, such as the long-term mechanical property testing and materials exposure testing, which are important for all turbine manufacturers but not specific to any one individual manufacturer's project. Projects in this sub-element are aimed toward hastening the incorporation of new materials and components into gas turbines.

A materials/manufacturing plan was developed in several stages with input from gas turbine manufacturers, materials suppliers, universities, and government laboratories. The plan was developed by a small advanced materials and turbine technology team over a six-month period. The technology plan calls for the initiation of several high priority projects in FY 1995.

The technical program for the materials/manufacturing element focuses on generic materials issues, components, and manufacturing processes. Categories include: Coatings and Process Development, Turbine Airfoil Development, Ceramics Adaptation, Directional Solidification (DS) and Single Crystal (SC) Airfoils Manufacturing Technology, Materials Characterization, Catalytic Combustor Materials, and Technology Information Exchange.

BACKGROUND INFORMATION

In 1993, the Department of Energy (DOE) initiated a program to develop advanced gas turbines for power generation in utility and industrial applications. The Advanced Turbine Systems (ATS) Program is jointly sponsored and managed by DOE Fossil Energy (FE) and DOE Energy Efficiency and Renewable Energy (EE). The program plan (Ref. 1) outlines an eight-year program which will involve turbine manufacturers, utilities, industrial end-users, national laboratories and universities. The objective of

the ATS Program is to develop ultra-high efficiency, environmentally-superior, and cost-competitive gas turbine systems for base-load application in utility, independent power producer, and industrial markets. The primary focus of the program is on natural gas-fired turbines, but adaptation to coal and biomass fuels is a design consideration. The program includes two major elements: innovative cycle development and developments to increase the turbine inlet temperature. One of the supporting elements of the ATS Program is a materials/manufacturing technologies task. The purpose of this element is to address material issues for both utility and industrial gas turbines. This paper describes the ATS materials/manufacturing plan that was completed in April 1994 (Ref. 2).

DOE-Oak Ridge Operations Office (ORO) and the Metals and Ceramics Division of Oak Ridge National Laboratory (ORNL) are coordinating the effort for the materials/manufacturing element. These Oak Ridge organizations are involved in various national DOE materials programs including: (1) Ceramic Technology Program, (2) Fossil Energy Materials Program, (3) Continuous Fiber Ceramic Composite Program, (4) Advanced Industrial Concepts Materials Program, and (5) Basic Energy Science Welding Program. Other Federal laboratories also have extensive materials/manufacturing expertise that will be used to support this initiative.

OBJECTIVES OF MATERIALS/MANUFACTURING ELEMENT

Materials and related manufacturing technologies are two key activities required to meet the Advanced Turbine Systems (ATS) Program goals. Several turbine manufacturers have stated a need for turbine inlet temperature as high as 2600°F (300°F higher than current heavy frame machines) to achieve the efficiency goals. To achieve these temperatures for the extended operating periods, there is a need to utilize new materials developments. The turbine manufacturers also indicated a need for effective interaction between themselves, materials suppliers, national laboratories, and others. Thus, the ATS Program Plan incorporates an element to address materials/manufacturing

issues for both industrial and utility gas turbines.

The primary objective of the materials/manufacturing element is to provide manufacturing technologies support to meet the ATS goals. The ATS prime contractors are responsible for the technologies necessary for their demonstration projects. The materials/manufacturing support is intended to complement the ATS team effort and provide expertise not available to any one single contractor.

TECHNICAL PLAN

The technical program for the materials/manufacturing element focuses on materials issues, components, and manufacturing processes with emphasis on generic developments. The sub-elements for the program were divided into seven categories as shown in the work breakdown structure in Figure 1.

Coatings and Process Development

This element will develop coatings and application processes to provide thermal barrier coatings (TBCs) for land-based gas turbines. The TBC/bond coat system was noted by all the manufacturers as a particularly high priority issue.

Reliable, higher performance TBCs will be developed to meet ATS temperature requirements. Coatings must be stable for times commensurate with ATS goals. Dependable TBCs would enable increased turbine inlet temperatures while maintaining airfoil substrate temperatures at levels to meet ATS life goals.

Current bond coats serve a dual purpose of providing oxidation and corrosion resistance to the metal substrate (and in some instances, the internal air-cooling passages of the airfoil) and bonding of the TBC. Corrosion requirements for land-based gas turbines require this work to be coordinated with work described under Materials Characterization and efforts on reducing the effects of sulfur described under Manufacturing Technology for Scale-up of Current DS and SC Turbine Airfoil Processes. Bond coats will be developed that have demonstrated reliability to allow increases in turbine inlet temperature

dependent on the TBC system while maintaining airfoil substrate temperatures at levels to meet ATS life goals.

Analytical models will be developed to enhance the reliability of dependence on the TBC systems to provide turbine inlet temperature increases with long life required to meet ATS goals. Testing will be used to validate the model's capability of predicting long-term behavior. The model would also provide a basis for developing diagnostic techniques for detecting impending coating failures.

Current manufacturing processes will be modified as required to process the reliable, higher performance TBC systems. Such processes will ensure that the benefits of TBCs and advanced air cooling airfoil designs are additive and cost effective.

Inspection and repair techniques will be developed in parallel with efforts described above that are applicable to the maintenance of ATS gas turbines in the field. Again, the dependence of TBCs systems to provide turbine inlet temperature increases with long life required to meet ATS goals will require development of procedures/devices to ensure preliminary indication of impending TBC failure. Life prediction methodology and sensor techniques will be developed, validated, and demonstrated.

Innovative coating concepts will be studied where a significant payoff can be forecast in terms of reliability, cost, or physical properties.

Development of Turbine Airfoils for ATS

Modification of current alloys will be investigated to meet specific requirements of ATS e.g., corrosion and long-term creep and fatigue properties. Special consideration will be given to alloy needs for closed circuit steam cooling.

Alloys will require modification /improvements in: casting process parameters, mold materials and design, core materials, implementation of computer modeling and rapid prototyping being developed elsewhere.

New concepts such as Castcool®, fabricated blades, etc. will be examined as alternate approaches for meeting the requirements for ATS airfoils in large gas turbines. To be viable, any new concept would have to afford advantages in terms of cost and/or cycle performance characteristics.

Ceramics Adaptation

Monolithic ceramics and ceramic composites processing technology will be adapted to ATS hot section hardware such as: shrouds, vanes, hollow (air cooled) stage 1 vane airfoil segments in metal bands, combustors and transition ducts, and seals for closed circuit steam cooling.

Monolithic ceramics and ceramics manufacturing technology will be adapted by process modeling and control and component inspection methodology. Long-Term Thermal and Mechanical Reliability will be studied with regards to oxidation and corrosion and the effect of transient stresses.

Manufacturing Technology for Scale-Up of Current DS and SC Turbine Airfoils

This element focuses on the modifications/improvements required to produce DS and SC airfoils for land-based gas turbines. Manufacturing technology is included in the three previous sub-elements. Manufacturing issues were consistently cited as one of the high priority technology areas by the gas turbine industries.

Alloy melting practice or treatments to reduce sulfur content in alloys will be studied. This work will be coordinated with the effects of reduced sulfur on oxidation resistance and resistance to spallation in TBCs. The manufacturing technology addressed will include: modifications/improvements in the DS and SC casting parameters, mold materials and design, grain orientation control for large single crystal airfoils, ceramic core materials and design, and mold and core removal and casting clean-up procedures that prevent recrystallization in SC airfoils. Post casting processing e.g. hot isostatic pressing and heat treatments and in-line and post manufacturing quality assurance and associated inspection techniques will also be addressed.

Materials Characterization

Long-term high temperature materials properties are needed for life prediction and reliability and process modeling for ATS. The database on materials selected by industrial and utility gas turbine contractors needs to be expanded. In addition, the generation of 100,000 h creep and fatigue data by standard methods is impractical to meet nearer term ATS needs. Accelerated testing methods and modeling will be investigated to provide reliable forecasted long-term properties. Substantiation of these procedures with microstructural analysis of long-term exposed similar materials from the field will be investigated.

The use of higher temperature airfoil and disk alloys developed for aircraft gas turbine use in ATS gas turbines will require evaluation of longer term microstructural stability.

The characterization of the long-term stability of TBCs subjected to time-dependent thermal and mechanical loads is primary concern since TBC failure may lead to catastrophic failure of the underlying alloy. Current characterization techniques are often based on semi-empirical tests which evaluate coating spallation effects. In order to generate data which can be used to establish life prediction methodologies for TBCs, the thermomechanical characterization tasks will be used to evaluate the failure mechanisms for representative thermomechanical cycles. A major thrust of this task will be to establish standardized test geometries and measurement procedures.

A life prediction analytical model will be developed for life prediction and reliability using the long-term data generated. Modeling will also be applied to large disks as required for ATS.

If protected from externally-induced physical damage, the intrinsic lifetime of TBCs depends on the chemical stability of the bond coat/TBC interface. Of primary concern are the effects of oxidation and salt-induced hot corrosion on the mechanical strength and stress distribution of this interface. Additional concerns relate to the interdiffusion of component elements, such as sulfur, to the interface. Corrosion studies of reference TBC/bond coat systems will be

conducted under prototypic ATS temperature and environmental conditions to identify degradation mechanisms operative at the bond interface and to model their kinetics to enable predictions of long-term (100,000 h) effects. Test variables will include effects of reactive metal additions to the bond coat and pretreatments to remove sulfur from the substrate alloy.

Catalytic Combustors Materials

The applicability of a simple catalytic combustor that could operate at the higher projected temperatures required for ATS is limited by the current state-of-the-art materials. New materials will be explored that could meet these stringent demands.

Technology Information Exchange

This element will provide a mechanism to transfer information on materials issues developed at NASA, DoD, and other DOE materials programs to the ATS Program. Many of these materials efforts will have a significant impact on the ATS Program.

SUMMARY AND CONCLUSIONS

The technical program for the materials/manufacturing element is focused on mid-to-long range R&D as identified by the gas turbine industry. For this element to be successful, it is necessary that industry have a lead role in executing the technical agenda. The majority of the projects will be directly with the gas turbine industry and many others will be lead by gas turbine suppliers. Federal laboratories will also have projects in the materials/manufacturing element, and these projects will also be required to be coordinated with the gas turbine industry. Projects will be selected based on competitive solicitations and other contractual mechanisms. Requests for proposals will be issued for the industrial materials/manufacturing projects, and informal competitions will be held for the federal laboratory projects. All of the projects will require teaming with a gas turbine company operating division. The involvement of universities will also be encouraged for both the industrial projects and the federal laboratory projects.

Two major solicitations were issued in FY 1994.

One solicitation was for the development of thermal barrier coating systems and the second was for manufacturing of single crystal turbine blades for both utility and industrial size gas turbines. Proposals have been received and are currently under evaluation. Work will be initiated in both of these project areas in early FY 1995. It is anticipated that additional solicitation in other high-priority project areas will be initiated in FY 1995-96.

REFERENCES

1. "ATS Program Plan", Office of Industrial Technologies and Office of Fossil Energy, Department of Energy, July 1993, DOE/FE-0279
2. "Materials/Manufacturing Plan for Advanced Turbine Systems Program", Office of Industrial Technologies, April 1994, DOE/OR-2007

Figure 1.1. Work Breakdown Structure of the Research Tasks in the ATS Materials/Manufacturing Element

**ADVANCED TURBINE SYSTEMS
ANNUAL PROGRAM REVIEW**

**Distributed Power
A Contractor's Perspective**

by

**David B. Eppinger
Director, Business Development
Fluor Daniel**

November 9, 1994

Distributed Power - A Contractor's Perspective

I am pleased to be here today to participate in this exciting discussion on distributed power generation. Being a new resident of California, I can say that there is one thing in common between ATS and Los Angeles - Wayne Gretsky. When asked how a relatively small and lightweight person could succeed in the National Hockey League, he would answer something to the effect that "I don't skate to where the puck is ... I skate to where the puck is going to be!"

The results of the ATS program is "where the puck is going to be" and Fluor Daniel has been successful in the power industry by anticipating and positioning for such occurrences. In essence, this is why Fluor Daniel is interested in participating in the development of Advanced Turbine Systems.

FLUOR DANIEL'S EXPERIENCE

Fluor Daniel has enjoyed a long and diverse role in the power generation industry. Our clients have included utilities, private power developers and industrial firms. Our history with gas turbines started with utilities who installed peaking plants industrial clients who used gas turbines for their energy intensive operations such as in refineries, chemical plants, pulp and paper plants, pipelines and off-shore platforms.

Over the past 30 years we have completed more than 70 gas turbine projects, with a total of 280 gas turbine units producing over 12,000 MW of electric power. These projects ranged in size from 3 MW to 1,370 MW and include leading edge gas turbine projects such as the Midland Cogeneration Venture, the Bergen Repowering Project, the Doswell IPP Project and the enhanced oil recovery projects in Bakersfield, CA.

Enough of the commercial on Fluor Daniel, lets look at the outlook for power generation and some of the market factors that exist today and explore how ATS fits into these developments.

OUTLOOK FOR POWER GENERATION

I don't think anyone here today doubts that the power industry is undergoing the greatest and most rapid change that we have seen in our life times. The PURPA act started the deregulation process and the Energy Policy Act of 1992 accelerated and confused it! I used to think that the California PUC is leading the way to competition, but with the emergence of power brokers and "marketing phenomenon" such as Destec's "1-800-SHOCKED" number for information on power competition, I'm not so sure.

Distributed Power - A Contractor's Perspective

The emergence of power brokers may indicate that the days of long term power purchase agreements are coming to an end and the era of merchant plants emerging. In addition, intense global competition, communication and intelligence networks, will ensure that events that occur in places such as Asia and Latin America, will impact the US power market. The capital cost of power plants the selling price of electricity have tended downward significantly in the past two years. While we hope that these trends will level off soon, we expect the effects to be permanent. New and well capitalized players such as oil & gas, pipeline and mining companies are emerging in the power market. Environmental issues such as emissions, noise and EMF make it difficult to site new generation and transmission as does the "NOPE" movement (Not On Plant Earth). In addition, even though laws and regulations are being designed to encourage market based economics, many manufacturing and commercial enterprises need consistent and reliable power. What does this all mean in terms of the power generation market outlook and the role of ATS?

We believe that the long term viability of power generation companies is dependent upon its ability to be competitive in its market. Not only must capital costs be competitive, but efficiency, fuel pricing, emissions financing and operations & maintenance costs must be competitive and sustainable over time. The ATS program addresses the issues of efficiency, emissions and equipment O&M costs and firms like Fluor Daniel can address the issues of plant capital and O&M costs. Distributed generation can also help provide consistent and reliable power by being able to locate generation in small increments, where the power is needed. As transmission systems become increasingly difficult to site and permit, this flexibility will become more valuable. Distributed generation can also help industrial firms capitalize on the true value of cogeneration.

The ownership of distributed generation will gravitate to the firms that bring to most added-value to such projects. These will probably include generation companies that need system support to enhance the quality of power, oil & gas companies who can supply and transport fuel long-term, and possibly O&M companies.

PAST IMPEDIMENTS TO SMALL COGEN

Cogeneration plants tend to be custom built and relatively high in capital cost in terms of \$/KW installed. It is difficult to reduce engineering costs through reference designs or modularization. Before PURPA cogen plants were built to maximize energy efficiency and minimize energy costs; however, after PURPA they were built to maximize revenue. This drove industrial firms and private power developers to construct the largest plant possible. Also, the owner costs of developing a cogen project such as permitting, financing, contracts development, land acquisition or leasing did not vary significantly with the size of

Distributed Power - A Contractor's Perspective

the project. Hence it was generally uneconomic for a private power developer to develop a small cogen facility.

While private power developers were contracting to supply the energy needs of many industrial firms, many utilities were actively trying to keep from losing such industrial loads and actively campaigned against cogeneration through regulatory processes and incentives such as cogeneration deferral rates. As such, many industrial firms who would be ideal candidates for cogeneration, were able to achieve the objective of lower energy costs by contracting for lower cost electrical power.

FLUOR DANIEL'S LONG TERM INTEREST IN SMALL POWER AND ATS

As mentioned at the beginning of this paper, Fluor Daniel has been successful by being able to anticipate "where the puck is going to be" and we believe ATS has a high probability of creating the future market for gas turbine projects. We believe it will have application in both as stand alone generation and industrial cogeneration. Fluor Daniel is active in the utility and private power business, but is also active in many industries that can benefit from highly efficient and cost effective gas turbine cogeneration systems. The ability to package our services and provide both the plant technology as well as the energy technology only increases our value to our clients.

Fluor Daniel has developed many tools such as intelligent CAD systems and reference plant specifications that minimize the costs of gas turbine power plant facilities. Our process design capabilities are also well suited for optimizing cogeneration cycle designs. As a full services contractor, Fluor Daniel can provide the full engineering, procurement, startup and testing needed to bring a plant on-line and demonstrate its design capacity and heat rate. We also can apply our skill in managing all elements of a project from design to startup to minimize the project schedule and further reduce the capital cost.

REMAINING CHALLENGES

For distributed power to become a significant part of the power industry, these plants must be cost competitive on a life-cycle basis. They must also be able to operate in both base load and dispatchable modes in order to adapt to cogeneration and cycling applications. In order to meet the need for high quality power, some form of value pricing needs to be developed to recognize the value of located distributed power where it is needed, when it is needed.

Distributed Power - A Contractor's Perspective

From the perspective of private ownership, the costs of developing, permitting and financing distributed power must be minimized in order make such projects economically attractive. Permitting processes need to be streamlined and standardized to reduce both the time and cost of this activity. Contracting to sell the power must also become a simplified process. Other owner related costs such as legal and financing fees, fuel supply and transportation and land acquisition must be also be minimized.

SUMMARY

In summary, distributed generation may have a significant role in the gas-fired generation industry of the future. However, for this to occur, the capital costs and life-cycle costs must be competitive and the value recognized (and paid for!) for being able to locate small increments of generation where it is needed when it is needed. The major equipment manufactures must complete the development of efficient, reliable gas turbines and contractors must develop the reference plant designs and fast-track schedules needed to reduce plant capital costs. Finally, regulatory reform must occur to streamline the permitting and approval processes such that owner's costs to develop distributed generation can be minimized.

ATS FOR DISTRIBUTED UTILITY GENERATION: A CASE STUDY

CONTRACT INFORMATION

Contract Number: None

ABSTRACT

Until 1990, the Sacramento Municipal Utility District (SMUD) had operated much like a classic integrated utility committed to large central station generation. At that point it was contractually separated from having its operations integrated with those of a large Investor Owned Utility; and significantly, its customer-owners had just elected to shutdown over half of its energy supply, the Rancho Seco Nuclear Generating Station. This paper discusses the issues facing the utility at this juncture and describes the approach used to diversify the energy supply while establishing the necessary basis for independence and economical power generation.

SMUD's response to these challenges was to construct what could now be described as an integrated approach to distributed resources, including cogeneration, demandside management and advanced and renewable technologies. Some of the projects which are underway might be considered as an order of magnitude larger than what is often considered as "distributed" generation, but SMUD's unique circumstance has necessitated the larger plants. What is specific to the goals and benefits of distributed generation are the reasons for the type of generation selected and how it is being integrated into the community and the electrical system.

New generation is being sited throughout our metropolitan service area by co-locating it with industrial processes and hosts able to take advantage of economically priced process steam. The benefits being realized by the community are significant. The new facilities are replacing process steam boilers, with the net effect being that reduced amounts of air pollutants are being emitted. While this is important, probably the most impressive benefit is that over 2,500 jobs have been retained within the community, not counting the new jobs created by the four co-generation facilities.

Concurrent with development of these replacement generators, SMUD has initiated aggressive Demand Side Management and Energy Efficiency programs. The direct benefit is that we expect to keep peak load growth flat for at least ten years, and in addition improve the efficiency of energy use. Since SMUD's customers are the direct beneficiaries of such savings, the effect is to retain capital and economic strength within the community.

An important adjunct to these actions has been the implementation of an aggressive

search for advanced and/or renewable sources of electric generation. Biomass, fuel cells, photovoltaic, solar-thermal and wind generation are all being actively developed for use in cost effective applications. Advanced technologies such as Electric Vehicles and high efficiency gas turbines are being supported and encouraged for applications within our service area.

The Sacramento area is not heavily industrialized, consequently it is not easy to develop single generation projects which can benefit from co-generation. It is for this reason that SMUD is very interested in the synergies that accrue when a relatively small scale, but high efficiency, gas turbine can be cost effectively sited with smaller industrial hosts. As this occurs the benefits of distributed generation will be available to even more of our customers, the effect of which should make the community's industrial base ever stronger.

Today utilities are being faced with an uncertain future. Some form of deregulation and a more competitive marketplace is going to be in the future of every electric utility. The utilities which are able to offer rate stability and quality customer service, are going to be valued by their customers. Such utilities are going to persist. Utilities unable or unwilling to be innovative and leaders in the industry are likely to be consumed by those who offer that "something extra". The technologies that define "distributed generation" are tools in the hands of a utility that can make it competitive in the marketplace of today and tomorrow.

ATS FOR DISTRIBUTED UTILITY GENERATION: A CASE STUDY

CONTRACT INFORMATION

Contract Number: None

BACKGROUND

For the 20 years prior to 1990 the Sacramento Municipal Utility District (SMUD) integrated its' generation resources and loads with those of a large private electrical utility. Concurrently we constructed and operated a 900 MWe nuclear plant and enjoyed the benefits of central station generation and the economies of scale. In 1989 SMUD's customer-owners elected to shutdown its nuclear plant, an action which overnight denied use of half of the Districts generation resources. Compounded by the associated costs and required rate increases, the municipal utility was challenged with a hostile takeover and management instability. In this environment it was necessary to think "outside the box" and introduce innovative concepts which would bring about rate stability and restore public confidence in the utility. This paper tells of the strategy and accomplishments which have resulted in a largely distributed utility able to accomplish these goals.

RESPONSE TO THE CHALLENGE

The lights did not "go-out" in Sacramento when the nuclear plant and integrated utility operations ended. Power purchase contracts had been put in place as a bridge to the future. These contracts would supply power at fairly competitive prices, though after five years escalation factors came into play which were going to cause another round of customer rate increases. The strategic goal of the utility then became; how to reduce this impact and reestablish a strong competitive position within the community.

The immediate action was to summarize our power requirements and to issue a Request For Power (RFP) to the industry as it had been decided that we could not afford the infrastructure, time and cost of creating our own generation design and construction capability of the magnitude required. The response to the RFP was inspirational, over 8,500 MWe offered from some 93 projects.

The diversity of projects and their capabilities was impressive, with most of the offerings were submitted by IPP's, Independent Power Producers.

These offerings fit our strategic objectives perfectly. One of our original goals was to reduce the risk of ownership and operation of generation, another was to diversify, that is, to never again place "all of our eggs in one (generating plant) basket". Out of the competition came a "short-list" of projects which met our target economic and "externality" criteria, for we evaluated the projects for their overall impact on our ratepayers.

We then took the "short-listed" projects, and our concept of bringing them into the system, to the community. Many meeting were held to obtain input on which projects and what types of resource and operating scenarios were important to our customer-owners and the community at large. From this it was decided to develop three "blocks" of resources:

Block 1	Four 100-150 MWe Gas Turbine based combined cycle/co-generation plants totaling 550-600 MWe, to be online by 1997.
Block 2	Develop out-of-area resources to take advantage of 400 MWe of economical transmission access.
Block 3	Allocate 300-450 MWe to Advanced and Renewable Generation to be on line in the post 2000 period.

In addition, it was decided to expand our already important Demand Side Management program to achieve a total of 600-800 MWe of DSM and Energy Efficiency load reductions, thereby effectively eliminating load growth for at least 10 years.

STATUS OF THE PROGRAM

The first 100 MWe co-generation plant is under construction and will be online in 1995. The other three plants are in the final steps of licensing and permitting and should go into the financing and construction phases within the next few months.

SMUD has instituted a wide range of DSM and Energy Efficiency programs which are on-track toward their goal of providing us with a cost effective 600 MWe "Conservation Powerplant". Importantly, the concepts of energy efficiency have also affected the selection and design of the new gas turbine power plants. They will be among the most efficient in the country. Likewise, improved efficiency is a foundation of our Advanced and Renewable Technology (A&RT) development projects.

As a component of our A&RT program we have been a charter member and supporter of the Collaborative Advanced Gas Turbine Project (CAGT). This effort develops a cost effective and very high efficiency utility power plant from advanced aeroderivative gas turbine components. In addition we are actively developing many renewable resources which are powered by solar, wind, geothermal and bio-mass derived energy. Significant within the A&RT program is the development of appropriate uses and applications for the developed technologies.

DISTRIBUTED GENERATION AND THE UTILITY

In the past four years SMUD has moved toward becoming a utility with distributed generation resources. As a component of the effort we have looked at the impact of such a strategy on the electrical transmission and distribution systems, and where there might be infrastructure savings on facilities. We have found that such savings can be achieved, but there are often competing system requirements which do not allow full implementation. On the other hand, we have found significant benefits for both the utility and the community by simply having a diversified and distributed generation supply. These benefits are likely to be all the more valuable to the utility as we enter the era of utility deregulation and competition for electrical customers.

SMUD is a case study in the benefits of distributed generation. Our new co-generation plants have already been directly responsible for retaining two large industrial customers within our service area, and with them, over 2,500 jobs. In addition, two new industrial firms have been brought into the area to utilize the process steam available from the co-gen plants. Furthermore, a significant improvement of the local air quality will result as the best available control technology is being built into each new plant. In fact, each of the power plants at existing sites will be emitting fewer pollutants than did the boilers simply supplying process steam before. This has not been done without additional costs, but the air quality value to the utility and the community, has more than justified the expense.

Sacramento is not a heavily industrialized region, and as such we have pretty well exhausted the available large steam hosts,

those able to utilize more than 100,000 pounds of steam per hour. Still the competitive advantages achieved by those hosts where we have been able to locate plants, have been very important to the community as well as SMUD. For this reason this author is suggesting that utilities will need to look at ways to offer this advantage to another tier of customers, those who could use 10,000 to 30,000 pounds of steam per hour. Of course this translates to a gas turbine plant in the under 25 MWe class. While many utilities might "turn up their noses" at such increments of capacity, those who wish to retain good customers by providing service and inducements, should consider the many benefits of such distributed generation. Sizeable benefits are likely to accrue to a utilities long-term competitiveness, while at the same time bring important improvements to the community and the environment. In order to make this scenario work it is essential that comparatively low-cost, and relatively high efficiency gas turbine generation be available.

HOW TO MAKE DISTRIBUTED GENERATION WORK

High efficiency must not automatically translate into complexity, high costs and/or additional infrastructure. One of the lessons of IPP generation is that power plants need not be staffed as a vertically integrated, self-sufficient operation. Today it is not unusual to find IPP operated combined-cycle/co-generation plants staffed at the level of one person per ten megawatts of capacity. Alternatively, there are many utility central stations which are burdened with staffing levels over one person per Megawatt! A competitive marketplace will surely force some dramatic changes!

Consequently SMUD could use a 25 MWe

standardized generation package that utilizes highly efficient components configured to operate reliably without continuous operator oversight. We have been operating 650 MWe of remote and "distributed" hydroelectric generation for decades without having operators at the plants. In addition we have similar experience with our wind farm, two fuel cells and 20 acres of photovoltaic generation. Today's controls, surveillance and telemetry systems provide the necessary levels of safety, protection and efficiency to insure reliable and cost effective operation of these smaller sized co-generation plants. This must be a goal of any "Advanced" turbine system intended for the distributed generation market.

A facet of advanced technology that needs to be challenged is the drive for "optimization". Many of the new innovations that we are proposing to introduce into utility service provide significant improvements over previous technology, and in particular, the typical or average generation unit. In this regard it is important that the improvements get into the field, not that every last increment of improvement be incorporated. As an example, we can look at the experience of every "good" utility engineer to prove that he was not asleep in "thermo", and therefore he needs to improve upon the Rankine (steam) cycle being adapted to his new plant. What we find is that practically every resulting steam turbine is unique to the application because there are so many variables with which to "play". In truth, there is very little improvement which can be made in optimizing the steam cycle, for usually the optimization algorithm is looking only at a few criteria, such as unitized first cost, life cycle costs, all assuming "base-loaded" generation profiles. Of more importance is how the unit will be able to support the multiplicity of utility and steam host needs. The unit typically needs to be good at a lot of things, and only above

average in order to be highly placed in the dispatch order. SMUD's four new co-generation plants will have heatrates (efficiency) which are within +/- 200 BTU/kWh of each other, depending upon ambient and steam host conditions. How will we ever be able to place them in the dispatch order? Honestly, it is not a problem, for the "market" in which they have to compete is nominally 2,000 BTU/kWh higher, a measure of the benefit of constructing a highly efficient plant! It is for reasons such as this that SMUD is supportive of Advanced Turbine Systems, and feels that there is a need to get associated technologies into the field, and available to as many customers as possible.

THE FUTURE "UTILITY"

There are many utilities which are taking a hard look at their circumstances and reconsidering how it is that they fit into the business of producing and distributing electricity. We read in the trade journals of major utilities which are considering a complete realignment in which they will no longer produce electricity, rather their role would be to distribute and service retail customers. Consideration of such changes is being encouraged by a number of recent pronouncements by regulators, along with the increasing access to large transmission grids and a multiplicity of power generators.

Over the past twenty years there have been a number of events and trends which have made utilities become much more risk adverse. Consequently it is not difficult to see why a utility might consider a future in which it did not have to raise capital for generation expansion, be threatened with interminable delays in siting and licensing, and then retroactively penalized for "imprudent" decisions and commitments. It might be viewed that providing power "just-in-time",

with access to a plethora of generators, might describe a much more certain future.

These comments and considerations result in a utility which is in business to provide a commodity, or service, to its customers.

When one focuses on that aspect of the business, then it is soon recognized that there are different levels and needs within the electric users. Some require an extremely high standard of reliability and/or quality of supply, while others may simply be looking for cheap electricity on an "as available" basis. A successful future utility will need to be able to match the generation resources to the customers needs. The yardstick for success is in retention of the customer, for in the future customers will have the ability to self-generate, or buy from others. It is in this regard that this paper is proposing that distributed generation facilities will be needed and very useful to utilities. Furthermore, there will develop a need for a new class of generating facility; comparatively small in size, built in modules, highly efficient and with its operation tailored to the needs of the customer.

CONCLUSION

Properly sized and highly efficient distributed generation capacity provides a new and valuable tool to a utility in the competitive marketplace. It is a vehicle with which utilities can improve and expand the services they provide. In this fashion distributed generation provides additional value to utility customers, retains utility load and improves the efficient use of infrastructure and investment in plant and facilities.

Session 3

ATS Program Technical Review

ADVANCED TURBINE SYSTEMS PROGRAM PHASE 2 DEVELOPMENT PROJECTS

CONTRACT INFORMATION

Contract Number	DE-AC21-93MC30247
Contractor	Westinghouse Electric Corporation 4400 Alafaya Trail Orlando, Florida 32826-2399 (407) 281-3270 FAX (407) 281-5014
Contractor Project Manager	Ronald L. Bannister
Principal Investigator	Ihor S. Diakunchak
DOE Project Manager	Donald W. Geiling
Period of Performance	August 5, 1993 to November 4, 1994

OBJECTIVES

The objectives of the Advanced Turbine Systems (ATS) Program Phase 2 were to select the ATS cycle and to develop technologies required to achieve ATS Program goals.

BACKGROUND INFORMATION

The Department of Energy-sponsored ATS Program is an 8-year program to develop the technologies required to provide a significant increase in natural gas-fired combined cycle power generation plant efficiency.

In ATS Program Phase 1, Westinghouse carried out a preliminary investigation on different cycles to demonstrate that overall thermal plant efficiency greater than 60% could be achieved (Little et. al., 1993). In Phase 2, different cycles were evaluated for the natural gas-fired ATS plant to select one that would achieve all of the program

objectives (Briesch et. al., 1994). Some of the cycles considered in this study, such as recuperation, reheat, and thermochemical recuperation, had higher estimated plant cycle efficiency compared to the advanced combined cycle concept. However, the advanced combined cycle was selected as having the best potential for exceeding the ATS program goals of overall plant efficiency, emissions, cost of electricity, RAM, and commercialization in the year 2000. The Westinghouse ATS plant will incorporate an advanced gas turbine coupled with an enhanced bottoming cycle and a high efficiency generator. To achieve the challenging ATS program performance, emissions, and RAM objectives, new technologies are being developed and existing technologies advanced beyond the current state-of-the-art. The attainment of ATS performance goals will require advancements in cooling, coating, materials, aerodynamics, and sealing technologies. The reduced emissions

requirement necessitates a development effort in the different areas of combustion technology, such as flow visualization, optical diagnostics, combustion instabilities, and catalytic combustor components. Materials and coating systems development are being carried out to ensure that the RAM targets are achieved. The research and development projects in ATS Phase 2 and beyond will advance the gas turbine state-of-the-art and resolve identified technical barriers.

The ATS engine will be 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, first cooled turbine airfoils in an industrial engine, and tilting pad bearings. In the past, enhancements in Westinghouse gas turbine performance and mechanical reliability were made in continuous steps (see Figure 1). The evolution of large gas

turbines started with the introduction of the 45 MW 501A engine in 1968. Continuous enhancements in performance were made up to the 100 MW 501D5 introduced in 1981. The next engine was the 160 MW 501F introduced in 1991. The 230 MW 501G is the latest engine in the series. Each successive engine design was based on the proven concepts used in the previous design (see Table 1).

Table 1. Proven Design Features

Feature	501D	501F	501G	ATS
2-Bearing Rotor	X	X	X	X
4-Stage Turbine	X	X	X	X
Ind. Combustors	X	X	X	X
Horiz. Split Casing	X	X	X	X
Cold End Drive	X	X	X	X
Single Row 1 Vanes	X	X	X	X
Cooling	AC	AC	AC/SC	AC/SC
Walk-in Enclosure	X	X	X	X

AC = Air Cooling

SC = Steam Cooling

FIGURE 1. EVOLUTION OF LARGE WESTINGHOUSE GAS TURBINES

The current production engine, the 501F, produces 160 MW at a simple cycle efficiency of 36%. Its combined cycle net efficiency is higher than 54%. This performance level was achieved by approximately 110°C (200°F) increase in firing temperature, compared to the previous engine, advanced compressor and turbine design, advanced airfoil cooling design, and improved materials.

The 501G engine, introduced in spring of 1994, produces 230 MW. Its combined cycle net efficiency is 58%. This engine incorporates further advancements in materials, cooling technology and component aerodynamic design. The 17-stage 19:1 pressure ratio compressor uses advanced profile high efficiency airfoils. Its performance has

been verified on a scaled engine test. The combustion system incorporates 16-can annular dry low NOx combustors. The combustor design is similar to 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 enhancements 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 included developments in combustion, emissions, cooling, sealing, mechanical design, coating, and materials technologies.

ATS DESCRIPTION

The ATS plant will be 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 (see Figure 2).

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 and appropriate materials for the advanced inlet steam temperature. In addition to the 3-D design philosophy, the steam turbine will incorporate other enhancements. The 60-Hz two-pole generator will be an extension of the Westinghouse hydrogen-cooled modular design concept.

FIGURE 2. ATS PLANT LAYOUT

The ATS compressor design philosophy will be based on that used in the advanced 501G compressor but with additional design enhancements, such as brush seals to minimize leakage under the stator shrouds. As a result of the pressure ratio selected for the ATS engine, variable stators in two or three front stages will be utilized for starting and part-load operation.

The combustion system will have can-annular combustors of lean premixed multistage design with catalytic components. To obtain less than 10 ppmv NO_x emissions, nearly all of the compressor air will be premixed with the fuel.

The turbine will be an advancement of the four-stage turbine designed for the 501G engine. The efficiency will be optimized by employing 3-D design philosophy and advanced viscous analysis codes. The airfoil loadings will be increased above previous levels so as to optimize airfoil efficiency while minimizing airfoil solidity (chord/pitch). The increased loading will result in reduced cooling requirements and improved plant efficiency. The following additional enhancements are planned for the ATS design: steam cooling of turbine airfoils, reduction of blade tip clearance, and improved rotor sealing.

To enhance the plant efficiency, closed-loop steam cooling (CLSC) will be used in the design of turbine airfoils, transitions, and baskets. The heat capacity of steam is 1.3 times that of air. Therefore, less steam than air will be required to cool the turbine components. The major benefit of CLSC is the elimination of cooling air ejection into the 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 benefit is the elimination of mixing losses associated with cooling air ejection. The combination of the above effects will increase the ATS plant efficiency significantly. In addition, the NO_x emissions will

decrease because more air will be available for the lean premix combustor at the same burner outlet temperature.

PHASE 2 COMPONENT DEVELOPMENT PROJECTS

To ensure that the ATS Program goals are achieved, an extensive R&D program was initiated in the fields of combustion, cooling, sealing mechanical design, materials, and coatings. The results obtained from these projects will help in resolving the technical barrier issues and will contribute to the success of the ATS Program.

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 combustion 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, Westinghouse-funded dry low NO_x combustor development programs are in progress.

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

flow mapping tests 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 being carried on a half-scale plastic model of the 501F combustion cylinder at Clemson University. Higher order flow effects inside the combustor cylinder are being investigated. Detailed information on pressures, velocities, and flow angles inside combustor cylinder, and especially around the combustor baskets, will be obtained. Included in this investigation will be the effect of struts, cooling air bleed port, cooling air return pipes, diffuser exit swirl, flow shields around the combustor baskets, top-hat length, and ultra low NOx baskets. In addition to studying and optimizing the flow around the combustor baskets, 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.

Optical Diagnostics Probe Development. Optical diagnostics allow measurement of pertinent parameters, such as the composition and concentration of combustion products, in addition to velocities and flow angles, without disturbing the main flow. An optical diagnostic probe is being developed to aid in combustion system optimization. Rig and engine tests using the optical diagnostic probe will be carried out.

Combustion Instability Investigation. Lean-premix combustion system will be employed to achieve the NOx emissions goals. The lean combustion with its inherent flame instability results in more combustion generated noise and hence in vibration problems in the combustion system as well as in the downstream components. A program to develop the theoretical background for combustion instabilities to carry out

experiments to aid in the understanding of the problem, develop a generalized analysis procedure, and to develop stability criteria is under way.

Catalytic Combustion. Catalytic combustion will play an important role in achieving ultra low NOx emissions at firing temperatures in the range of 2700°F. The catalyst allows ultralean-premix (fuel and air) combustion without flame instability and flame outs. Hence, NOx production is restricted to low single digits at ATS firing temperatures with stable combustor operation. Development is being carried out to gain theoretical understanding of catalytic combustion, to design a catalytic combustion system using the "clean sheet approach," and to develop a practical catalytic combustor.

Ultra Low NOx Combustor Development. Described below are three ultra low NOx combustors being developed by Westinghouse.

1. **Multi-Swirl Combustor:** This combustor consists of small swirlers arranged in four concentric rings in a single plate. A pilot located at the center of the four swirler rings can be operated in a semi-diffusion mode, premixed, or a combination of both. Premixed fuel/air is fed to each swirler ring through four annular passages (one per swirler ring). At the entrance of each annular passage, gas is injected through a fuel ring into the inlet airflow subsequently becoming premixed before entering the combustion zone. Mid- and full-pressure testing has shown that this combustor can restrict NOx emissions to single digits with low-pressure dynamics. Mid-pressure tests demonstrated that a stable flame can be maintained with the pilot flow shut off. Although this combustor design allows for the pilot to be run at very low levels of fuel flow, the pilot is primarily responsible for most of the NOx produced. To achieve

low NO_x levels, a catalytic pilot was developed to allow operation below the lean extinction limit. This pilot has been high pressure tested in the multi-swirl combustor and has demonstrated an extremely stable flame.

2. **Multi-Nozzle Combustor:** This combustor consists of eight premixed nozzle/swirler assemblies arranged around a diffusion pilot nozzle/swirler assembly. The premixed swirlers are designed to enhance mixing without recirculation whereas the pilot swirler provides strong swirl to produce a recirculation zone and, hence, good flame stability.
3. **Piloted-Ring Combustor:** This combustor comprises a pilot and two separate premixed zones arranged axially, that is the primary and secondary zones. Premixed fuel and air enter the primary zone where combustion is stabilized by a swirl-produced recirculation zone and a centrally located pilot. The secondary zone is downstream of the primary zone and is fed premixed fuel/air through an annular passage surrounding the primary zone. Recent mid-pressure rig tests demonstrated single-digit NO_x emissions at ATS burner temperature.

Cooling

Closed-Loop Steam Cooling. The one new 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 steam to and out of rotating components, leakage control, and cold

start before steam from the downstream HRSG is available. In ATS Phase 2, a conceptual design of closed-loop steam cooled airfoils is being carried out.

Shroud Cooling. The airfoil end walls or shrouds present a cooling design challenge. Model tests are needed to optimize the cooling design of vane shrouds. To address this issue, model tests are being performed to optimize the shroud air film cooling design.

Integral Shroud Cooling. In order to improve performance and reduce blade vibratory stresses, the Row 3 turbine blade incorporates an integral interlocked tip shroud. The blade is cooled to maintain the blade metal temperature at an acceptable level. To cool the tip shroud, alternative cooling schemes and manufacturing processes were developed. The blade casting development program is being carried out.

Serpentine Channel Cooling. To maintain blade metal temperatures, complicated multipass serpentine cooling schemes are being developed. Plastic model tests are being performed 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 will be carried out at different cooling air flow rates. The internal heat transfer coefficients and pressure losses will be measured.

Mechanical Design

The last stage turbine blade is another component requiring development effort. To achieve minimum exhaust losses, the annulus area of the last stage turbine blade must be as large as possible. This results in a long, highly loaded, last stage blade. To provide mechanical damping, this blade incorporates an integral interlocked tip shroud. In this type of a blade design, there is a

possibility that two blade natural frequencies may occur between adjacent harmonics. These two frequencies can become coupled, resulting in excessive vibratory stresses. A three-phase program is being carried out to resolve this issue. In Phase 1, a review of the design rules for last stage turbine blades, the reevaluation of advantages/disadvantages of interlocked shroud versus free standing last row blades, and the evaluation of the effect of stimuli and damping on last row blade vibratory stresses are being carried out. In Phase 2, evaluation of alternative blade and shroud designs including part span shrouds and a blade casting development program on the selected design will be performed. In Phase 3, vibration test of the last stage blade will be performed to verify the design. The Blade Vibration Monitoring (BVM) technique will be employed to measure vibration characteristics of each individual blade in the row.

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. 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. A conceptual design of an active tip clearance control system is being carried out in ATS Phase 2.

Brush Seals. To reduce air leakage, as well as hot gas ingestion into turbine disc cavities, brush seals will be incorporated in appropriate locations in the turbine and compressor. This will enhance engine efficiency as well as mechanical integrity of the turbine components. The incorporation of an effective, reliable, and long-lasting brush seal system into a heavy-duty industrial combustion turbine requires a development program. Phase 1 involves preliminary investigation and a conceptual brush seal design.

Materials/Coating Development

Single Crystal Blade Development. To enhance performance and reliability, single crystal blades have been considered for application in the ATS engine. A program is in progress to demonstrate castability of large, unshrouded and shrouded blades in CMSX-4 material. Existing 501F engine tooling have been used to cast stages 1 and 3 single crystal blades. The blades will be evaluated by grain etching, selected NDE methods, and dimensional inspection methods to determine their metallurgical acceptability.

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 approximately 11 to 14°C (20 to 25°F) metal temperature reduction per .025 mm (.001 in.) coating thickness. While TBC's have been used widely on stationary components, use on rotating components has been limited. Field testing of coated blades is being carried out to determine comparative coating longevity and effectiveness of air plasma spray and physical vapor deposition thermal barrier coatings.

Ceramic Components. Ceramic components have potential for reducing cooling requirements, enhancing engine performance and improving turbine component reliability. In ATS Phase 2

program, stages 1 and 2 turbine ring segment designs using ceramic matrix composite materials are being carried out.

Turbine Aerodynamic Development

An internally funded turbine aerodynamics development program was carried out to optimize stage efficiency and minimize cooling requirements. The program objectives were: to demonstrate turbine airfoil design with high suction surface diffusion, to verify the turbine efficiency estimation method, and to validate 3-D viscous analysis of a high diffusion airfoil. In past turbine airfoil designs, conservative suction surface diffusion values were used. If the airfoil loading could be increased without penalizing efficiency, then the airfoil numbers and, hence, the wetted surface area could be reduced. The lower airfoil surface area results in reduced cooling requirements and increased engine performance. Increasing the aerodynamic loading pushes the boundary layer on the suction surface closer to separation. The increase in suction surface diffusion was made possible through the use of advanced aerodynamic design codes which predict accurately the airfoil surface velocity distribution while maintaining sufficient margin before boundary layer separation. To verify the increased loading and the validity of the design codes, a single-stage turbine was designed, model turbine hardware at 0.32 scale was manufactured and extensive rig tests were carried out. Based on the measured stage efficiency, flow visualization, and vane and blade exit radial circumferential traverse results, it was proven that increased airfoil loading did not result in suction surface flow separation and that the aerodynamic codes used in the design accurately predicted the flow conditions and stage efficiency.

CONCLUSIONS

The Phase 2 efforts on ATS gas turbine development have progressed sufficiently to

demonstrate that the ATS Program goals are obtainable in the 5-year time frame. The Westinghouse advanced gas turbine will utilize a steam cooled blade path in conjunction with advanced aeroderivative materials and coatings to meet ATS Program performance and cost of electricity objectives with minimum risk in availability. The engine will be designed with a minimum number of parts using advanced 3-D aerodynamic, heat transfer, and mechanical design codes to reduce both manufacturing and life-cycle costs. An environmentally benign combustion system is being developed to limit NO_x emissions to single digits.

REFERENCES

Little, D. A., Bannister, R. L. and Wiant, B. C., 1993, "Development of Advanced Turbine Systems," ASME Cogen Turbo Power '93 Proceedings, IGTI - Vol. 8, pp. 271-280.

Briesch, M. S., Bannister, R. L., Diakunchak, I. S., and Huber, D. J., 1994, "A Combined Cycle Designed to Achieve Greater Than 60 Percent Efficiency," ASME Cogen Turbo Power '94 Proceedings, IGTI - pp. 299-308.

Scalzo, A. J., Bannister, R. L., DeCorso, M., and Howard, G. S., 1994, "Evolution of Heavy-Duty Power Generation and Industrial Combustion Turbines in the United States," ASME Paper 94-GT-488.

ACKNOWLEDGEMENTS

The ATS concepts discussed in this paper were documented under DOE Contract No. DE-AC21-86MC23167. The program is administered for DOE by the Morgantown Energy Technology Center under the guidance of Mr. Donald W. Geiling.

Advanced Turbine Systems Program Industrial System Concept Development

CONTRACT INFORMATION

Contract Number DE-AC21-93MC30246

Contractor Solar Turbines Incorporated
2200 Pacific Highway
P.O. Box 85376
San Diego, CA 92186-5376

Contract Project Manager Karl W. Karstensen

Principal Investigators	Graham J. Benjamin	Michael D. Stephenson
	Richard S. Brent	Luke H. Cowell
	David J. White	Michael D. Fitzpatrick
	Joel S. Otsuka	Jane F. Simpson
	John F. Talbott	Fritz Langenbacher
	Chaur S. Wen	

DOE Project Manager Charles T. Alsup, Jr.

Period of Performance August 3, 1993 to November 30, 1995

Schedule and Milestones

FY94 Program Schedule

Task	O	N	D	J	F	M	A	M	J	J	A	S
1. Project Plan	XXXX											
2. NEPA Report	XXXX	XXXX										
3. System Selection	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX						
4. Coal Conversion	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX						
5. Market Study		XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX
6. System Definition		XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX
7. Program Plan												
8. Critical Components							XXXX	XXXX	XXXX	XXXX	XXXX	XXXX

OBJECTIVE

Phase II

The objective of Phase II of the Advanced Turbine Systems Program is to develop conceptual designs of gas fired advanced turbine systems that can be adapted for operation on coal and biomass fuels. The technical, economic and environmental performance operating on natural gas and in a coal fueled mode is to be assessed. Detailed designs of and test work relating to critical components are to be completed and a market study is to be conducted. (Reference 1)

BACKGROUND INFORMATION

Throughout its 35 year history as a supplier of industrial gas turbines, Solar has maintained a careful surveillance of the marketplace into which the company's product is sold. This effort has paid off in the growth of Solar into the world's leading supplier of industrial gas turbine systems.

Based on this ongoing evaluation of the marketplace, Solar established short- and long-term product development goals which coincide exactly with ATS goals as set by the DOE in four areas:

- Reduced exhaust emissions
- Increased RAMD*
- Reduced cost of power
- Increased thermal efficiency

*Reliability, Availability, Maintainability, Durability

Entering into Phase II of the ATS Program, Solar has quantified goals in these four areas as shown in Table 1.

Table 1. Advanced Turbine System Goals

Parameter	Solar's ATS
Thermal Efficiency	50 %
Exhaust Emissions NOx	8 ppm
Exhaust Emissions CO & UHC	15 ppm
Cost of Power (COP)	10 % Reduction from Today
Reliability, Availability, Maintainability & Durability (RAMD)	Equal to or Better than Today

PROJECT DESCRIPTION AND RESULTS

Reference 2 identified an intercooled and recuperated (ICR) gas turbine as the power plant best suited for development to meet ATS objectives. The technologies that will need to be incorporated into an ICR to meet these objectives are within the capability of development by a manufacturer of industrial gas turbines, i.e., Solar. More importantly, these technologies are likely to find ready acceptance in the industrial gas turbine marketplace, i.e., characterized by moderate risk and reasonable cost. Many of these technologies will also find application in Solar's current product line and as retrofit improvements in Solar's 9200-unit installed fleet.

Recuperation of the classic Brayton cycle gas turbine is a well-known efficiency improvement that has been applied at levels of success primarily established by the cost and durability of the required heat exchange device. At any given level of peak temperature a recuperated Brayton cycle will have an optimum overall pressure ratio. At pressure ratios above this optimum, recuperation capability is reduced by the narrowing gap between exhaust temperature and compressor discharge temperature.

The addition of one or more stages of intercooling into the compression process of the

Brayton cycle will significantly reduce the work input required by the compression process. The combination of intercooling and recuperation into an ICR cycle produces an increase in both specific power and thermal efficiency. The thermal efficiency improvement derives from an increased temperature differential between the exhaust gas stream and the compressor discharge air. This enables the recovery of a greater portion of the thermal energy in the exhaust gas. In a modern ICR, thermal efficiency continues to increase with pressure ratio and with temperature.

Task 3 of Solar's ATS Phase II program is entitled, "System Selection." (Tasks 1 and 2 fulfilled certain non-technical contractual requirements). With the ICR cycle identified as the best ATS candidate, Task 3 consisted of setting cycle parameters, primarily firing temperature and pressure ratio. These parameters establish the maximum attainable thermal efficiency within practical material development prospects and within the probable bounds of acceptance in the marketplace.

Based upon these "developability" and "salability" considerations, a maximum firing temperature was set at 2600°F, and a practical pressure ratio of 20:1 was selected. The temperature limit was established by Solar's current view of the near term capability for both ceramic airfoils and for metal airfoils with advanced cooling technology. The 20:1 pressure ratio limit is dictated by manufacturing considerations arising from the very small last stage compressor airfoils which will result from the application of intercooling to the smaller sizes of gas turbine in an industrial class (<10 MW). High pressure ratios have been associated with difficulty in achieving stable combustion in low emission burners. Installation expense and system efficiency is also compromised by the size and power consumption of fuel gas compressors required for use with high-pressure-ratio (>20:1) gas turbine systems.

ICR Gas Turbine Arrangement and Performance

Solar's ICR gas turbine as shown in Figure 1 is characterized by the performance map of Figure 2. Based upon component performance attainable by industrial gas turbine design and on intercooler and recuperator performance as stated below, this cycle optimizes at 2500°F TRIT and a pressure ratio of 16:1. Cooling air bleed used in calculating this map is associated with advanced cooled metal airfoils – 10 to 13 percent of inlet air flow. This ICR features a low-cost, single-spool arrangement which allows the use of a single-stage driver turbine for the gas producer section. A more costly two-spool design will show a small gain in thermal efficiency as overall pressure ratio is increased above 16:1. A two-spool design optimizes at 2500°F TRIT, also, and produces no higher design-point efficiency than the one-spool version.

Figure 1. ICR Gas Turbine With Free Power Turbine

Recuperator

Solar brings to the ATS program, a unique, fully developed, compact, high-performance recuperator known as a Primary Surface Recuperator (PSR). In a heat exchanger, a surface is designated primary when the heat flow is in a

Figure 2. ICR Thermal Efficiency vs PR at Lines of Constant TRIT One-Spool Configuration. The optimum efficiency is shown to be at Rc of 16:1, with a TRIT of 2500° F.

transverse direction as across a sheet or across a tube wall. (A secondary surface transmits heat in a lateral, high resistance direction, as in a fin.) 100 percent of the heat transfer surface in Solar's PSR is primary as shown in Figure 3. This results in the extremely compact, high performance recuperator compared with other types (Figure 4).

Figure 3. Primary Surface Recuperator Sheets

Figure 4. 2750-kW (3690-hp) Gas Turbine Recuperator Comparison

The primary surface sheets shown in Figure 3 are formed of very thin (typically 3- to 5-mil) foils of stainless steel or superalloys. The sheets are assembled into air cells as shown in Figure 5 by welding around their perimeter. Air cells are then welded together into a core assembly as shown in Figure 6. The full PSR assembly results from the subsequent attachment of air manifolding, gas ducting and support structure as dictated by the physical layout of the gas turbine system.

Figure 5. PSR Air Cell

Figure 6. PSR Core Assembly

All fastening is welding--no brazing or other methods are employed which can reduce the strength or ductility of the parent material. All welding is external -- none is required in the interior of the core.

Recuperator life is set by the creep life and oxidation resistance of the sheet alloy. Both are temperature dependent and, in Type 347 stainless steel, limit gas inlet temperature to approximately 1230°F for life capability equal to typical core engine time between overhaul (TBO) life. The use of superalloy sheet raises this temperature limit, but increases cost along with it. The designer of recuperated gas turbines soon recognizes that recuperator temperature (life) limit is juxtaposed onto the temperature (life) limit of the turbine airfoils in a well-matched design. Thus, if firing temperature is increased, a recuperator material improvement is required along with a turbine airfoil material/cooling improvement unless the overall pressure ratio of the machine is increased.

The recuperator applied to Solar's ICR design has been sized for 90 percent effectiveness and static pressure loss ($\Delta P/P$) of 0.02 cold-side and 0.05 hot-side.

Intercooling

The high air flow rate of the gas turbine and its sensitivity to static pressure loss in the flowpath will result in a very large device if direct (air-to-air) intercooling is considered. Further, there will be a significantly large parasitic power requirement for coolant (air) pumping.

These considerations quickly lead to the choice of indirect intercooling for the gas turbine. A single type of compact heat exchanger can be selected for the engine flowpath unit and a number of options made available for the secondary cooler. Three such options are shown in Table 2, which illustrates that the choice of a secondary cooling device is highly dependent upon conditions at the installation site.

In some segments of the marketplace, intercooling can present a commercial challenge. In many areas of the world, the gas turbine has won favor by its total independence from water, either as a required site feature or as a maintenance requirement; make-up requirements, chemical treatment, etc. These areas, which have welcomed the advent of dry NOx control, may resist water-based intercooling.

The intercooler applied to Solar's ICR is sized for 85 percent effectiveness and air-side static pressure loss ($\Delta P/P$) of 0.03. Cycle performance is calculated based on coolant entry temperature equal to ambient air temperature.

Compressor Description

For the optimum 16:1 pressure ratio in Solar's ICR Gas Turbine the low pressure (LP) compressor will have three stages. Variable inlet guide vanes and second stage stator vanes are employed to allow adjustment to match flow vectors at lower rotational speeds during start and low power operation. The

high pressure compressor has eight stages, again with a variable inlet guide vane and second stage stator row.

The compressor selected for Solar's ICR gas turbine comes from an internally funded aerodynamic technology program known as "ACE" – Advanced Component Efficiency. The compressor portion of this program is based on state-of-the-art compressor design codes and will feature controlled diffusion airfoils (CDAs). In addition to improving compressor efficiency over current Solar designs, the use of CDAs will reduced the number of airfoils used (lower cost) and the airfoils, in turn will be larger and more rugged (improved RAMD).

Turbine Description

The design of all turbine stages is optimized for aerodynamic performance while maintaining mechanical integrity over the required life. The performance of the turbine stages – one gas producer stage and three power turbine stages – is based on Solar's "ACE" aerodynamic technology program mentioned earlier.

Turbine Cooling and Material Selection

Any Brayton Cycle efficiency gain to be realized with increase in peak (firing) temperature is very

dependent upon the energy expended in cooling the turbine materials to the temperature level required for commercially practicable life. Normally this energy expenditure takes the form of air flow bled from the compressor which then bypasses the combustion process and varying portions of the expansion process. This is true of all Brayton Cycle forms, simple or complex.

In complex cycles there are additional opportunities for improving cooling effectiveness. In a recuperated cycle, for example, air to be returned early in the expansion process can be bled from the recuperator exit rather than the compressor exit. While this may increase the actual cooling air bleed flow, it can improve thermal efficiency by using this flow to recover additional exhaust energy for use in the expansion process. Where cooling air temperature is of critical importance, any intercooled cycle has a lower compressor discharge temperature and bleed air can be further cooled by a second pass through a dedicated section of the intercooler.

Given a firing temperature selected on the basis of cycle optimization or other consideration, cooling air requirement becomes the key variable in determining thermal efficiency. This effect is shown for Solar's ICR gas turbine in Figure 7. At 2500° F, present cooling technology requires some 20 to 22 percent of the gas turbine's inlet flow to be bled for

Table 2. Available Types of Secondary Gas Turbine Intercoolers

Type	Advantages	Challenges
Liquid (closed-circuit) - Air	Controlled coolant environment Small environmental effect	Cooled air temperature above ambient
Evaporative (cooling tower) Type	Cooled air temperaure below ambient	Make-up water required Water treatment Vapor plume Fouling avoidance
Direct Liquid Sink (sea water or stream)	Lowest attainable cooled air temperaure No visible bulk	Thermal pollution Corrosive coolants Fouling

Figure 7. Thermal Efficiency of an ICR Gas Turbine as a Function of Cooling Air Flow

cooling. Metallic blade cooling technology which can be developed in an ATS program will reduce cooling air bleed sufficiently to provide a 14 percent improvement in thermal efficiency – a major portion of the 18 percent affordable by commercially practicable introduction of ceramic blades.

In any technology program, the key element in reducing risk is the readiness of alternative technology items where future progress may be less than crystal clear. Solar believes that successful attainment of the DOE's ATS goals to be sufficiently vital that two alternatives are warranted. Table 3 shows the primary choice together with both alternatives for key hot section elements. The primary choice in each case is the one affording the best overall attainment of ATS objectives – low emissions, high RAMD levels, reduced cost of power and increased thermal efficiency. Alternative materials may also become attractive in terms of providing a shorter time-to-market for the ATS or in an improved payoff via greater opportunities for retrofit to the existing gas turbine fleet.

System Definition

Task 6 of Solar's ATS Phase II requires the physical description of the Advanced Turbine

System selected in Task 3. Most of the work in Task 3 was completed without reference to a specific power output of the ATS. On the few occasions when a physical size was required for reference, an airflow (and hence, physical size) was selected which would produce 16,000 shp from the ATS cycle under study at the time. A gas turbine of this size would power a generating system of approximately 12 Mwe.

At such time as the market study enables a choice of the exact size of an ATS to be proposed, the size (or sizes, if more than one is indicated) can be established through application of mechanical laws of scaling. These scaling laws operate as follows on gas turbine aero components:

- For constant rotor linear tip speeds, geometrically similar components will operate with:
 - Identical Pressure Ratios
 - Identical Efficiencies
 - Identical Stresses
 - Comparable Metal Temperatures
- Rotational speeds will scale inversely proportional to linear dimension.
- Air flow and power will scale with the square of linear dimension.

Heat transfer components such as recuperators or cooled airfoils will scale differently because the quantity of heat in place at any given location will vary with the cube of a linear dimension and the size of the heat transfer path scales with the its square. This does not affect the application of the rules for scaling aero components and heat transfer designs will be "thermally" similar but geometrically somewhat dissimilar.

Scaling is not applied to standard hardware such as fasteners and tubing. It may also be intentionally set aside where performance

Table 3. Summary of Preliminary Material Selections for Critical ATS Hot Section Components

Engine Component	Primary Selection	First Alternative	Second Alternative
GP Turbine Section a. Blades b. Nozzles c. Disks d. Seals	Ceramics ¹ Ceramics ² Dual-property disk FGM [†] ceramics	Cooled CMSX-4 Cooled CMSX-4 Coarse-grain U-720 Zr [†] -based coatings	Cooled CMSX-4 + TBC [†] Cooled CMSX-4 + TBC [†] Waspaloy McrAlY coatings
Recuperator	Dual material Inconel 625/347 SS	Inconel 625	Haynes Alloy 230
Combustor	Carbide or BZP [†] ceramic	Metallic w/FGM [†]	Cooled superalloy ³
Ducts	Monolithic ceramic (SiC [†])	Oxide/oxide CFCC [†]	Cooled superalloy ³
<p>Notes:</p> <p>¹Candidates are nitride and carbide ceramics; cooled or uncooled depending on turbine stage.</p> <p>²Candidate are toughened nitride, carbide and CFCC ceramics; cooled or uncooled depending on turbine stage.</p> <p>³Candidates are Hastelloy X Haynes Alloy 230 and oxide dispersion strengthened superalloys.</p> <p>[†]TBC = Thermal Barrier Coating FGM = Functionally Graded Material Zr = Zirconium BZP = Barium Zirconia Phosphate SiC = Silicon Carbide CFCC = Continuous Fiber Ceramic Composite</p>			

advantages are available in the larger scales through relatively reduced airfoil leading and trailing edge thickness, endwall clearances and surface finish.

Gas Turbine Power Plant Integration

The concept cross section of Figure 8 shows an ICR gas turbine for a 12 MWe ATS system. This two shaft gas turbine has a variable area power turbine nozzle. The combustion system is a gas fueled, nine can annular ultra-lean premix design retrofittable to catalytic combustion. The approach to structural design follows proven industrial gas turbine practice. Structures are sized to provide the

necessary rigidity to preserve optimum alignment of components through the entire range of transient and steady state thermal expansions.

The industrial gas turbine practice of using hydrodynamic mainshaft bearings is continued in Solar's ATS rather than changing over to rolling element bearings. While the latter offer a small reduction in power loss, they lack the damping capability, long life and general robustness of hydrodynamic bearings. Solar's tilting pad bearing designs also have an ability to accept levels of misalignment that would result in severe overloading of rolling element bearings.

Figure 8. Solar's ICR Core Gas Turbine Cross-Section

Mainshaft sealing in current Solar gas turbines is by air-buffered labyrinth seals. This type of seal is low in cost and rugged under operational conditions. Acceptable alternatives, such as face seals, will be sought during the ATS development program that will eliminate the loss to the cycle resulting from buffering air bleed and will also reduce fugitive emissions from the oil tank.

Advance Turbine System Integration (Packaging)

The level of achievement of ATS goals by action limited to the basic prime mover will fall well short of that attainable by treatment of the entire system. This system includes all prime mover support systems – control, lube, fuel and starting – ancillary equipment – inlet and exhaust systems and enclosures – as well as driven equipment. Solar is a system integrator (packager) of its own gas turbine prime movers and, as such, has complete design control over the system. This control allows

optimum matching of all system components in maximum support of ATS goals.

Energy conservation by means of controlling parasitic power losses is one such opportunity for Solar as its own system integrator. Figure 9 shows a typical gas turbine generator set. Fully eight percent of the shaft power developed by the core gas turbine is consumed in support systems and ancillary equipment. Opportunities that Solar will pursue in the ATS program toward reducing these losses include:

- Demand-driven variable speed lube and servo pumps, eliminating the loss deriving from dumping excess oil flow over relief valves.
- The design of less restrictive inlet systems.
- Integration of the reduction gear and generator eliminating one entire bearing and its associated power loss.

Figure 9. Typical Industrial Power Generation System Energy Distribution

Cost reduction opportunity also exists throughout the gas turbine system. In the system described in Figure 9, fully 60 percent of the total in-plant manufacturing cost lies outside the boundaries of the core gas turbine.

Figure 10 illustrates the integration of an ICR gas turbine into an ATS system.

Development of Critical Technologies

The DOE has recognized that, if new technologies are to be incorporated into an ATS in a timely manner, an early start on their development is good insurance. Accordingly, such technology development has been written into Phase II contracts. This element of Solar's Phase II contract – Task 8, titled "Design and Test of Critical Components. Task 8 includes materials development in support of the ATS recuperator and turbine disks, advanced ceramic materials development, low emissions combustion research and advanced control technology.

Materials Development – Recuperator

The temperature capability of the recuperator sheet material – presently Type 347 stainless steel – limits attainable thermal efficiency at all levels of pressure ratio. Currently it limits the cycle firing temperature for any given pressure ratio. This limiting exhaust temperature is approximately 1230°F for Type 347 stainless steel. A low-cost alternative to be examined in Task 8 is one or more of the high temperature ultrafine precipitate strengthened (HT-UPS) austenitic stainless steel alloys under development at Oak Ridge National Laboratory.

Large increments of temperature capability can be obtained through the use of nickel-base alloys such as Inconel 625 or Haynes 230. Cost of these alloys approaches four times the per-pound cost of Type 347 stainless steel. Task 8 will determine the prospects for using such alloys only in regions where temperature will be above the limit for Type 347, thus minimizing cost. Methods of joining the two

Figure 10. ICR Gas Turbine Integrated into an ATS System

alloys will be examined along with the formability and other critical characteristics of the joined pair.

Materials Development - Turbine Disks

The current material of choice for turbine disks is a forged high temperature – usually nickel based – alloy. This material is chosen for its high low-cycle fatigue strength required for survival of the large strain excursions involved in gas turbine start-run-stop cycles. Near the rim at the point of blade

attachment, these materials lack the creep resistance of cast versions of the same or a similar alloy. Adequate creep life of this region of the disk is maintained by cooling with compressor bleed air, at the cost of cycle efficiency. Increasing cycle firing temperature requires more cooling air. Flattening of combustor outlet temperature profiles as required to maintain low stress levels in ceramic airfoil at a constant average temperature will also raise disk rim temperature and require additional cooling.

Task 8 efforts will develop a method of producing a strong bond between a turbine rim section cast of creep-resistant high temperature alloys and a center section which maintains the low cycle fatigue properties of the present materials. Once the right alloy pair and bonding process is identified, its characteristics will be defined to the gas turbine designer for application to Solar's ATS as well as for introduction to the balance of Solar's product line.

Materials Development – Ceramics

Going beyond the wealth of ceramic technology that will flow into the ATS from Solar's Ceramic Stationary Gas Turbine (CSGT) Program with the DOE, Task 8 will focus in two specific areas:

- Candidate ceramics for use in association with the catalytic combustor and its associated ducting will be identified. In ATS machines, these materials will serve to free cooling air applied to metallic ducts for use in emissions reduction.
- A second focus will be on accurately determining the life characteristics of ceramic composites such as carbon- or silicon carbide-reinforced silicon carbide. The end goal is the potential use of these materials for turbine blade airfoils for operation in the 2500°F region without the requirement for cooling air.

Component Development - Combustion

As a part of Task 8, Solar is evaluating a subscale ATS combustor in an existing high pressure test rig (Figure 11). The objective of this test is to determine performance of the catalyst bed and the fuel-air premixer under ATS conditions. This program will next progress to a full-scale single can of the multi-can annular catalytic combustor intended for the ATS. Working with a subcontractor supplying the catalyst in a ceramic

Figure 11. Catalytic Combustor Rig

matrix, Solar will develop the combustor to meet the ATS requirement of 8 ppmv of NO_x and 15 ppmv of CO and hydrocarbons.

A second requirement of combustion systems intended for high temperature operation is a reduced pattern factor – less variation of individual temperatures around the average level. As average temperatures increase, peak levels increase proportionately and, unless pattern factor is reduced, can exceed the temperature capability of downstream components which are otherwise entirely adequate at the average temperature level. Addressing this concern will be an integral element of the catalytic combustor program.

Component Development - Recuperator

In addition to the recuperator material task previously discussed, a second task deals with recuperator thermal and flow performance required in support of ATS cycle performance goals. The geometric parameters of the primary heat transfer surface can be varied in order to provide the combination of heat transfer effectiveness and static pressure loss required by the cycle. In the case of the ATS, the goal is to preserve the 90 percent thermal effectiveness used in prior applications of the primary surface recuperator (PSR) while

reducing the static pressure loss in both air and exhaust gas streams. This will entail a tradeoff in overall recuperator volume.

The design of the new surface was accomplished using codes derived both empirically from previous test work and by application of "textbook" heat transfer principles. The first laboratory validation of the new surface utilizes a transient heat transfer test rig shown in Figure 12. Known as the "f and j" rig from the Friction Factor (f) and Colburn Modulus (j) derived from its data,

Figure 12. Transient (f and j) Performance Test Rig

this rig is a duplicate of one developed by Prof. A. L. London of Stanford University. Utilizing sample cores 4 x 4 x 4 inches in size, this rig has evaluated over 80 different heat transfer surfaces in connection with PSR development.

Final verification and mapping of recuperator heat transfer and flow loss performance is accomplished utilizing the steady flow rig shown in Figure 13. This rig, using a 12 x 12 x 12 inch core sample is now in the process of mapping the performance of the core design chosen for Solar's ATS.

Component Development - Autothermal Fuel Reformer

Autothermal fuel reformation consists in reacting a fuel such as methane with steam in order

to produce a "reformed" fuel with a high hydrogen content – on the order of 70 percent or greater. Hydrogen has a wider range of combustion stability than methane. Being lighter and more reactive than methane it mixes more readily with air and can be burned at much lower fuel/air ratios. This results in lower NO_x production. The reformed fuel also has a much lower ignition energy which makes low temperature catalytic combustion possible – another factor contributing to low NO_x production.

Figure 13. Steady State Performance Test Rig

When the steam required for the reformation of gas turbine fuel is raised using energy from the gas turbine exhaust gas stream, cycle efficiency is improved leading to the term "chemical recuperation." Chemical recuperation is applicable to the Brayton cycle in most of its forms, including simple cycle, recuperated and ICR.

A fully developed autothermal reformer (ATR) will bring to an ATS two fundamental benefits:

- Reduced NO_x emission without some of the risks associated with other methods, such as reduced combustion stability.
- An increment of thermal efficiency based on additional waste heat recovery.

For this reason, Solar has included in ATS Phase II both fundamental and advanced development work on an ATR that can be added to a commercialized ATS whenever marketplace economics and preferences will dictate. This strategy has, in fact, contributed to the choice of a can-annular combustor design. Solar's vision is that of an ATR integrated into the upstream region of a combustor can as illustrated in Figure 14. The combustion process downstream of the ATR can either be a lean premix design, as shown, or a catalytic process.

Figure 14. Integrated Autothermal Reformer and Combustor

Solar's current efforts are centered on the screening of various reformer catalyst options and on determining the effect of reformer operating conditions upon catalyst performance. This work is being conducted in the tube reactor shown in Figure 15 with typical results shown in Figure 16. Here, hydrogen concentration is seen to increase less rapidly with temperature above 1300°F and to increase with steam/carbon ratio. Catalyst screening for properties such as sulfur tolerance are also being determined in this rig.

Performance of an ATR under gas turbine operating conditions will be developed using a rig now under construction. This rig utilizes an injector

for premixing of fuel and steam. A pilot combustor partially oxidizes the fuel to provide the heat required for preheating the steam/fuel mixture to the reformation temperature. Following a mixing section, the reforming catalyst produces the high hydrogen secondary fuel which is delivered to a post-combustor where its heat content and emissions characteristics can be determined. A general view of this rig is shown in Figure 17.

Figure 15. Tube Reactor for Reforming Catalyst Screening Tests

Figure 16. Test Results of Hydrogen Produced as Functions of Temperature and Steam to Carbon Ratio (R)

Figure 17. Aerothermal Fuel Reforming Test Rig

Component Development - ATS Control System

In order to fully realize all of the benefits of ATS, program activity has to proceed beyond the boundaries of the gas turbine itself and through the entire system. Within the system all support subsystems must be optimized for full support of the ATS goals designed into the gas turbine, itself.

One such subsystem of critical importance is the control system – not just a gas turbine control but an ATS system control. During Phase II, Solar will design an advanced Man/Machine Interface (MMI) which will apply microprocessor technology so as to provide improved efficiency and RAMD along with reduced cost and emissions. Important features of this system are described below:

- The system will provide easy communication with other system/plant computers and control equipment over industry standard networks. This will allow optimization of the complete system of which the ATS is a part, improving efficiency and reducing emissions.
- Automatic intelligent reduction/analysis of control system data into advisory information.

This will assist the operator in the diagnosis of plant operation and to set maintenance schedules which will result in higher levels of RAMD. He will also be able to optimize plant operation for increased efficiency, lower emissions and lower cost of power.

- Long range communication ability will be improved by utilizing media such as telephone lines, microwave links and communications satellites. This will allow operation, data collection and analysis and diagnostics and maintenance to be performed on multiple units from a central location. The system will also interface with Solar's Customer Services Center to provide rapid response to problems.
- Modular options, easily integrated at low cost into the customer's MMI will be provided based on the customer's needs. These can include remote starting and operation, interfacing with existing plant and process controls, on-line economic analysis and many others.
- Easy, in-field reconfiguration of the MMI will minimize the cost of future updates that may be required by changing requirements of the system.

Other ATS-related Technology Development Programs at Solar

Ceramic Stationary Gas Turbine (CSGT)

This DOE-sponsored program with Solar as the prime contractor and eight cost sharing subcontractors is a technology demonstrator program with the retrofit of ceramic components into existing industrial gas turbines as its ultimate goal. Starting with the development of design methodology for these components, this program has produced detailed designs for components to be tested in a Solar Centaur® 50 gas turbine. Component rig and engine tests will precede final

validation of CSGT technology in a 4,000 hour field test. Commercialization of the retrofit design for the Centaur 50 and other gas turbine models will follow successful completion of this field test.

Materials/Manufacturing Development Programs with ORNL

Solar is involved in two materials/manufacturing programs with the Oak Ridge National Laboratory (ORNL). One of these has Solar as a subcontractor -- along with other gas turbine manufacturers -- to prime contractor Howmet. This program, awarded in September, 1994, has as its goal the development of lower cost, high performance single crystal components. Field demonstrations of the resulting components are scheduled in the fourth year of the program, making Solar's ATS field test a candidate host for these tests.

The second ORNL program involves the development of improved performance thermal barrier coatings for industrial gas turbines. Solar, as a candidate prime contractor, has submitted a proposal in response to RFP DE-RPO5-94OR22242, along with subcontractors including Howmet, NASA, Allied Signal, the University of Connecticut and Chromalloy. Timing of this program is also such that resulting coatings can be a part of Solar's ATS field Test.

In addition to their key contributions to ATS, both of these programs will provide spinoff technologies for inclusion in Solar's current product line as well as for retrofit into Solar's fleet of more than 9200 gas turbine systems sold around the world.

SUMMARY

Solar approached Phase II of the ATS program with the goal of identifying a system that would be capable of 50 percent thermal efficiency.

Technologies for this system were selected on the basis of a high probability of successful development within the time frame of the overall program and, more importantly, a high probability of early acceptance in the industrial gas turbine marketplace. This industrially optimized ATS was identified as an intercooled and recuperated (ICR) gas turbine at 16:1 overall pressure ratio and fired at 2500°F TRIT.

FUTURE WORK

Solar has been heavily involved in ATS starting with DOE's CSGT Program, continuing with ATS Phase II, other DOE programs as well as internal programs. Fiscal 1995 will see the completion of formal documentation of the Phase II Market Study and System Definition as well as full continuation of the nine technology programs included in Task 8. Technology developed in these nine areas will provide a rolling start for ATS Phase III and, at the same time, feed into current Solar product as retrofit improvements. Solar's overall goal is early and abundant public derivation of benefits provided by DOE funding.

REFERENCES

1. Report to Congress, "Comprehensive Program Plan for Advanced Turbine Systems," U.S. Department of Energy, Office of Fossil Energy, Office of Energy Efficiency and Renewable Energy, July 1993.
2. White, David J., "Gas Fired Advanced Turbine System," Solar Turbines Incorporated and U.S. Department of Energy. 1993.
3. Solicitation for Cooperative Agreement Proposal (SCAP) for "Industrial Advanced Turbine Systems (ATS) Development and Demonstration," DE-SC21-94MC31173, Amendment 001. U. S. Department of Energy, September, 1994.

Advanced Turbine System (ATS) Turbine Modification For Coal And Biomass Fuels

CONTRACT INFORMATION

Contract Number DE-AC21-93MC29257

Contractor Allison Engine Company
P.O. Box 420
Indianapolis, IN 46206-0420
(317) 230-2575
(317) 230-3691 (Fax)

Contractor Project Manager Dale W. Mukavetz

Principal Investigators Richard Wenglarz
Nirm Nirmalan
Tom Daehler

DOE Project Manager Diane Hooie

Period of Performance August 3, 1993 to February 3, 1995

Schedule and Milestones

1994 Program Schedule

	J	F	M	A	M	J	J	A	S	O	N	D
Define CFATS Systems Repts												
Define Dep/Cor Repts												
Evaluate Cooling Approaches												
Determine Turbine Performance												

OBJECTIVE

One overall goal of the Advanced Turbine System (ATS) program is alternate fuel flexibility. The natural gas-fired ATS turbine designs are to be adaptable to future firing with coal-derived and biomass fuels. Specific current program objectives include:

- definition of a coal-fueled advanced turbine system (CFATS) based on the gas-fired advanced turbine-system (GFATS)
- estimate performance and emissions for the CFATS

BACKGROUND INFORMATION

Oxygen blown coal gasification plants using cold gas scrubbing can produce turbine fuel gases that contain very low levels of impurities approaching those levels in natural gas. However, high capital cost and complexity, in addition to efficiency penalties for cold gas scrubbing, are so significant that acceptable economics are an issue for these plants. Economics are particularly an issue for plants based on industrial-sized turbines, which do not have the economies of scale of large utility power plants.

Air blown gasification or direct-fired turbine plants using hot gas cleanup with ceramic filters offer significantly lower capital costs and higher energy conversion efficiencies for alternate fuels than oxygen blown plants using cold gas scrubbing. However, the levels of impurities entering the turbine are higher for hot gas cleanup. Depending on the filter operating temperature, thermodynamic analyses and gas stream measurements indicate that vapor phase alkali concentrations from 1 to 10 ppmw can pass the hot gas filter and enter the turbine. These alkali levels are as much as two orders of magnitude greater than what is allowed by conventional fuel specifications to alleviate turbine corrosion. Furthermore, measured solid particulate levels of 10 ppmw and higher have passed hot gas ceramic filters in tests, such as those at the Grimethorpe and Tidd PFBC plants.

Consequently, approximately 3 to 5 ppmw of coal ash might reasonably be expected to enter a commercial engine operating in a coal gasification or direct-fired turbine plant using a hot gas filter. This particulate level is similar to that produced by washed and treated (with additives of control corrosion) residual oils, the highest ash fuels that have been commercially used in gas turbines.

Residual oil fueled turbines have usually operated in peaking service. At intervals as frequent as 200 operating hours, deposition in the turbine flow path can require taking turbines out of service for water washing for periods as long as a day. While acceptable for peaking service, such frequency and duration of turbine outages would not likely be acceptable for cogeneration, base load, or intermediate load electrical generation plants.

The residual oil operating experience described is for commercial turbines typically operating at inlet temperatures of less than 1038°C (1900°F). An objective of the ATS turbine is much higher rotor inlet temperatures, up to 1427°C (2600°F). Tests such as at Westinghouse (Whitlow, et. al, 1981) have shown that higher gas temperatures greatly increase deposition rates. At the highest test gas temperature (1260°C [2300°F]), deposition rates were 30 times higher than at a temperature corresponding to past commercial turbine experience (1038°C [1900°F]). The high deposition rates were caused by an increased fraction of molten ash at the higher gas stream temperature.

The magnified effect of increased gas temperature on deposition has also been shown for lower ash coal fuels (Mulik, et. al, 1985). Figure 1

TE94-1951

Figure 1. Deposits Formed at First Stator Conditions Representing 1260°C (2300°F) Inlet Temperature Turbine

shows deposits formed at 1260°C (2300°F) in about 100 hr from a coal-derived liquid fuel, which produced about 2 ppmw particulate loading in the gas stream.

PROJECT DESCRIPTION

The high rates of deposition (and corrosion) for high gas temperatures result from molten ash particles that stick on airfoil surfaces and are more chemically reactive than solid or gaseous ash phases. Since nearly all coal ash would be molten for the 1427°C (2600°F) rotor inlet temperature goal for the GFATS, extreme deposition and corrosion will occur unless measures are taken to protect the CFATS. Extreme cooling of CFATS hot section airfoils can be used to alleviate deposi-

tion and corrosion. Extreme cooling of surfaces freezes molten ash particles in the vicinity of airfoils, substantially reducing ash sticking, deposit strength, and rates of corrosion.

The following describes efforts to:

- specify the degree of cooling needed to protect the CFATS from deposition and corrosion
- evaluate alternate approaches to provide the specified degree of cooling
- determine the cooling flow rates needed to evaluate the performance of the CFATS

RESULTS

Cooling Requirements

In the late 1970s and early 1980s, extensive tests were conducted to determine the effects of decreased surface temperatures on reducing deposition and corrosion and, thereby, improving turbine fuel flexibility. At typical turbine inlet temperatures, low turbine alloy surface temperatures were found to result in low corrosion rates from fuel contaminants such as sulfur, sodium, and vanadium. Corrosion rates rapidly increased at surface temperatures above the 538°C to 649°C (1000°F to 1200°F) range (EPRI, 1981).

Cascade tests (Rose, 1982) with ash bearing fuels showed that extreme cooling of airfoils to the 427°C to 538°C (800°F to 1000°F) temperature range significantly reduced deposition rates and greatly facilitated on-line deposit cleaning by nut-shelling. Cascade tests using coal-water fuels (Horner, 1989) have also demonstrated the benefits of airfoil surface cooling to below 538°C (1000°F) on reducing deposition rates and improving cleanability of deposits. Cleanable deposits and tolerable corrosion rates are also known to occur in coal boiler plants with surface temperatures less than about 538°C (1000°F).

Based on the above information, an average airfoil surface temperature of less than 538°C (1000°F) has been specified for the CFATS turbine rows where the gas temperature exceeds 927°C (1700°F). The cooling of downstream expander rows may not be necessary, as indicated by coal-fired PFBC pilot plant cascade tests in which tolerable deposition and corrosion occurred below the 927°C (1700°F) temperature range.

Alternate Cooling Approaches

Three cooling alternatives have been considered to provide the specified degree of cooling for coal fuels:

- air cooling
- steam cooling
- two-phase, water evaporating cooling

Air Cooling. As inlet temperature increases, gas turbine cycle efficiency optimizes at increasing pressure ratios. This results in high compressor discharge air temperature, exceeding 538°C (1000°F) for the ATS rotor inlet temperature of 1427°C (2600°F). Consequently, a CFATS plant would require an intercooler, if compressor discharge air is used to cool the turbine airfoils to less than 538°C (1000°F).

Steam Cooling. Steam is superior to air as a medium for high levels of cooling (Wu and Lewis, 1984). This is because steam has a higher specific heat than air and relatively little pumping is required to provide pressurized steam. Since steam is produced in cogeneration and combined cycle plants, it is available for gas turbine cooling at relatively little additional equipment cost.

Two-Phase Cooling. A very high degree of surface cooling results from water evaporation at that surface. A two-phase cooling approach using compressor discharge air to atomize water into small droplets that impinge and evaporate on the interior surfaces of turbine airfoils is being evaluated by Allison. For stator vanes, the atomization is accomplished interior to the airfoils. For rotor blades, the atomization is accomplished exterior to the airfoils, with the centrifugal pumping effects of the rotating blades drawing the air and droplet stream into the blades. This bypasses the difficulties of transporting water across a moving interface into the rotating blades, but increases the challenge of properly distributing the droplets for impingement on the interior blade surfaces. Two-phase cooling requires very high water purity to minimize deposition of minerals on the interior airfoil surfaces.

Coolant Flow Requirements

Cooling of airfoils in the CFATS turbine to average temperatures of less than 538°C (1000°F) has been specified for those expander rows where gas temperatures exceed 927°C (1700°F). Using this specification, two stator rows and two rotor rows would need to be cooled to 538°C (1000°F) for the

ATS turbine with a rotor inlet temperature of 1427°C (2600°F).

First order analyses were conducted to assess the feasibility of the three methods described for cooling the coal-fueled ATS turbine. Table 1 shows the calculated coolant flow rates for the airfoils in the four highest temperature rows in an ATS turbine with a rotor inlet temperature of 1427°C (2600°F). The values for air and steam coolant flows consider film/impingement cooling for the first row vanes, film/convection cooling for the second row blades, impingement cooling for the third row vanes, and convection cooling for the fourth row blades. For both air and steam, the coolant temperature is assumed to be 232°C (450°F). For two-phase cooling, temperatures of 93°C (200°F) and 260°C (500°F) were assumed for the water and atomization air at the atomization nozzle inlets.

Table 1 shows that the mass flows of steam needed for cooling are only about 57 to 66% of that needed using compressor discharge air. As expected, air cooling flows for the four airfoil rows are excessive, requiring about 35% of the total compressor discharge flow.

Cooling of all four rows with steam was also determined to require excessive flows. The steam flows required are over 40% higher than can be generated from the available heat in the turbine exhaust using a heat recovery steam generator. Cooling the fixed stator vanes with two-phase flows and the rotor blades, end walls, and seals with steam was then considered. Considering leakage flows also, the total steam requirement was determined for this case to be about all that could be generated from the turbine exhaust. Steam cooling would increase turbine power due to a

Table 1. Percent Ratio of Airfoil Coolant to Compressor Flow Rates for 538°C (1000°F) Surface Temperature*

Coolant	First stator vanes	First rotor blades	Second stator vanes	Second rotor blades
Air	13.6	10.5	8.1	3.1
Steam	8.9	6.9	4.6	1.8
Two-phase-water/air	0.70/2.0	0.82/2.6	0.48/1.9	0.50/1.7

* ATS turbine with 1427°C (2600°F) rotor inlet temperature

higher mass flow, but would decrease revenues in a cogeneration system due to steam being diverted to cooling the turbine. Results from the previous DOE/Allison Advanced Coal-Fueled Gas Turbine Systems Program (Wenglarz, 1994) indicate the loss in revenues would prevent the CFATS from achieving an acceptable internal rate of return (IRR) for the operator of a cogeneration plant. For example, Figure 2 shows the cogeneration plant IRR penalty versus the airfoil surface temperature of steam cooling two rows only of a coal-fueled turbine (1093°C [2000°F] inlet temperature). In this case, the 538°C (1000°F) surface cooling results in a loss of about 80% of the steam revenues (compared to 100% for a CFATS plant) and an IRR below a commercially attractive value of 20%.

The above considerations indicate that two-phase cooling of all four rows appears to be the most feasible of the methods explored for cooling the 1427°C (2600°F) CFATS turbine.

FUTURE WORK

The flow rates and temperatures of air, steam, and unevaporated water entering the turbine flow path at each row for two-phase cooling have been estimated. Using this information, turbine performance computer codes will be used to calculate the performance (power output and efficiency) for the CFATS turbine. Emissions estimates for the CFATS turbine system will be based on measurements made during tests of the rich-quench-lean

Figure 2. IRR for Cogenerator Systems Using Steam for Cooling of a 1093°C (2000°F) Inlet Temperature Turbine

coal combustor for the DOE/Allison Advanced Coal-Fueled Gas Turbine Systems Program.

REFERENCES

EPRI, 1981. *Water-Cooled Gas Turbine Development Program*, Final Report, EPRI AP-1889.

Horner, M. W., et. al., 1989. *Advanced Coal-Fueled Gas Turbine Program*, U.S. Department of Energy report, DOE/MC/23168-3040.

Mulik, P. R., Wenglarz, R. A., Spengler, C. J., and Whitlow, G. A., 1985. Coal-Derived Liquid Fuel Performance Under Gas Turbine Conditions, *Energy Progress*, Vol 5, No. 3, pp 136-142.

Rose, R. S., 1982. *Advanced Cooling Full-Scale Engine Demonstration Program*, Final Report, EPRI AP-2605.

Wenglarz, R. A., 1994. *Advanced Coal-Fueled Gas Turbine Systems*, Final Report, Vol 1: *Coal-Fueled Turbine Technologies and Systems*, in press.

Whitlow, G. A., Lee, S. Y., Mulik, P. R., and Wenglarz, R. A., 1981. The High Temperature Combustion of Residual Fuel Oil, *EPRI/DOE 2nd Conference on Advanced Materials for Alternate Fuel-Capable Heat Engines*, Monterey, CA.

Wu, C. S., and Louis, J. F., 1984. A Comparative Study of the Influence of Different Means of Cooling on the Performance of a Combined Cycle, *J. of Engineering for Gas Turbines and Power*, Vol 106, pp 750-755.

CERAMIC RETROFIT PROGRAM

CONTRACT INFORMATION

Contract Number DE-AC02-92CE40960

Contractor Solar Turbines Incorporated
2200 Pacific Highway
San Diego, CA 92101
(619) 544-5000

Contractor Project Manager Mark van Roode

Principal Investigators Paul F. Norton
William D. Brentnall

DOE Project Manager Stephen Waslo

Period of Performance September 25, 1992 to September 30, 1998

FY95 Program Schedule

	S	O	N	D	J	F	M	A	M	J	J	A
Test Engine												
Test Facilities												
Ceramic Components												
Low Emission Combustor												
4000-Hour Engine Test Plan												
Management & Reporting												

RE94231M

Table 1. CSGT Program Team

TEAM MEMBERS	RESPONSIBILITIES
GAS TURBINE MANUFACTURERS:	
Solar Turbines Incorporated (Solar)	Program Management Engine Design Materials Evaluation Component Testing Engine Testing Technical and Economic Evaluation Technology Integration Commercialization
Sundstrand Power Systems (SPS)	Life Prediction Design Support Materials Evaluation Support
CERAMIC COMPONENT SUPPLIERS:	
AlliedSignal Ceramic Components (CC)	Rotor Blade (GN-10 Si ₃ N ₄)
Kyocera Industrial Ceramics Corporation (KICC)	Rotor Blade and Nozzle (SN-253 Si ₃ N ₄)
Norton Advanced Ceramics (NAC)	Rotor Blade (NT164 Si ₃ N ₄) Combustor Liner (NT230)
Carborundum	Combustor Liner (Hexoloy® SA)
NGK Insulators, Ltd.	Nozzle (SN-88 Si ₃ N ₄)
Babcock & Wilcox (B&W)	Combustor Liner (alumina/alumina CFCC)
DuPont Lanxide Composites (DLC)	Combustor Liner (SiC/SiC CFCC)
TEST LABORATORIES	
Caterpillar Technical Center (CAT TC)	Non-Destructive Evaluation (NDE) of Ceramic Components
Argonne National Laboratory (ANL)	
University of Dayton Research Institute (UDRI)	Long Term Testing of Ceramics
END USER	
ARCO Oil & Gas	End User Representation Commercial Application Cogeneration Field Test Site

Figure 1. Centaur 50 Gas Turbine. Components for Ceramic Substitution are Indicated

PROJECT DESCRIPTION

Program Strategy

The program strategy selected by the Solar led team provides a strong focus on near-term ceramic turbine technology demonstration and lowering barriers for its acceptance by the marketplace.

Program Activities

Phase I of the program, which was started in September of 1992 and completed in April of 1993, involved concept and preliminary design for the ceramic Centaur 50 engine and for its first-stage blade and nozzle, and combustor liner components. Commercially available silicon nitride and silicon carbide monolithic ceramics and SiC/SiC and alumina/alumina continuous fiber-reinforced ceramic matrix composites (CFCC's) were the ceramic materials selected for the component designs. The design philosophy focused on developing component geometries with relatively low stresses and

temperatures for the anticipated 30,000 hours of service life of a production-ready ceramic stationary gas turbine.

Performance improvements resulting from the reduction in cooling air requirements and increase in turbine rotor inlet temperature (TRIT) from 1010°C (1850°F) for the all-metal Centaur 50 baseline to 1121°C (2050°F) for the ceramic uprate are estimated to be 5.6% in thermal efficiency and 25.9% in output power in simple-cycle operation. These gains were estimated for a hot section in which first-stage metallic blades and nozzles, and the cylindrical portions of the combustor liner were replaced with ceramic parts without an attempt to redesign the hot section to optimize the performance for the operating conditions of the ceramic uprate. When fully optimized, performance improvements as outlined in the BACKGROUND section could be realized. An economic evaluation showed significant performance improvements and economic benefits of the ceramic gas turbine technology once implemented in retrofits of current engines and in future engine designs.

Phase II, which started in April of 1993, and is scheduled for completion by September 1996, addresses detailed engine and component design, ceramic specimen and component procurement, fabrication and testing. Ceramic blades, nozzles, and combustor liners will be tested in subscale test rigs and in a gasifier rig which is a modified Solar Centaur 50 engine. The Phase II efforts also involve long term testing of ceramics, development of appropriate nondestructive evaluation (NDE) methodologies for part evaluation, and ceramic component life prediction.

Phase III, to be performed from October 1996 though September 1998, will include the manufacturing and instrumented engine testing of the field engine, the actual 4,000 hr field test at the cogeneration site, and pre- and post-testing component evaluation.

Management and reporting is an integral task for all phases of the program.

RESULTS

Phase I: Turbine and Component Preliminary Design

The Phase I work has been presented at conferences [4-6] and has been fully documented in a Final Report [7]. Where appropriate results of the Phase I work are being referenced in the discussion of the Phase II tasks described below.

Phase II: Final Design, Material and Component Testing

Task 7: Test Engine Design and Procurement

The Task 7 efforts involve the detailed design for the CSGT engine and its components. The Phase I conceptual and preliminary designs were the starting point for the Task 7 work.

Performance Analysis

Aerothermal analysis provided performance goals for the program engine, an increase in turbine rotor inlet temperature (TRIT) from 1010°C (1850°F) to 1121°C (2050°F), an increase in output power of 25.9%, and a thermal efficiency increase of 5.6% in simple-cycle operation. A NO_x emission level of 25 ppmv will be demonstrated, with a stretch goal to achieve 10 ppmv or lower. The thermal efficiency increase is relatively modest because the retrofit scenario does not incorporate optimizing the flow path and aerodynamics of the hot section to take full advantage of the increased volume flows at the higher firing temperatures. The output power increase is substantial and is primarily attributed to the higher turbine rotor inlet temperature (TRIT) in the CSGT engine compared to the all-metal Centaur 50 baseline.

Component Design Approach

From a design perspective one of the challenges of the CSGT program is to interface the ceramic hardware with the remainder of the metallic structure of the Centaur 50 engine. The overall strategy was to design for component stresses and temperatures that will ensure acceptable reliability for the service life of the CSGT engine. A summary of critical design stresses and temperatures is listed in Table 2. The design strategy must take into account the thermal expansion mismatch between metallic and ceramic components and its effect on movements at metal/ceramic and ceramic/ceramic interfaces.

The ceramic designs have progressed on the basis of three major tenets:

- i) use of balls/races and curved surfaces at ceramic interfaces in order to increase the precision and quality of the load distribution between surfaces.
- ii) use of rolling elements to avoid direct relative movement of flat, plane ceramic interfaces under high loading conditions.
- iii) avoidance of tapered ceramic interfaces or ceramic compression joints, which under thermal "ratcheting" or cyclic conditions, lead to higher stress conditions.

Specific solutions have been generated as part of the design strategy. The selected conceptual component designs selected were taken through preliminary design using detailed finite element analysis (FEA) applying the boundary conditions estimated for the Centaur 50 CSGT at 1121°C (2050°F) TRIT.

Crucial in the FEA boundary conditions was the combustor hot-spot temperature envelope which was significantly different from the scaled up radial profile of the baseline Centaur 50 engine used in the

Table 2. Critical Steady State Component Temperatures and Stresses

Component	Peak Temperature	Peak Stress
Blade	1116° C (2041° F)	165-214 MPa ($\approx$ 24-31 ksi)
Nozzle	1297° C (2366° F)	179 MPa ($\approx$ 26 ksi)
Combustor Tile	1204° C (2200° F)	190 MPa ($\approx$ 28 ksi)
Combustor Ring	1204° C (2200° F)	75 MPa ($\approx$ 11 ksi)
Combustor CFCC (Fiber)	1150° C (2100° F)	78-163 MPa (11-24 ksi)

Task 1 conceptual design studies. The new hot-spot envelope was designed to provide a 'worst case' (conservative) estimate of the combustor temperature envelope impinging on the stage 1 nozzle. Also, this combustor envelope reflects adequate cooling flows for acceptable stresses on the secondary metallic hot section components. While the latter objective was successfully met, the selected radial profile resulted in significant increases in the stress levels at critical locations in the blade and nozzle airfoils of the primary ceramic components. The combustor radial profile selected to guide ceramic component design is shown in Figure 2. The combustor pattern factor was

established as 0.2. The impact of the profile and pattern factor are particularly important for the nozzle because high stresses can arise as a result of a combination of hot-spot conditions (maximum profile) at the nozzle airfoil trailing edge adjacent to localized cool areas (minimum profile) near the nozzle shrouds.

The candidate materials and suppliers selected for the component design are listed in Table 1. The materials were selected based on the following criteria: (1) low probability of failure of the supplier's material under engine conditions defined by preliminary design, (2) fabrication experience with similar component configurations and ability to deliver components in a timely fashion, and (3) potential for establishing volume manufacturing. Ceramic materials and supplier selection is being reviewed at various milestones in Phase II as materials property and life prediction data become available.

Figure 3 shows a schematic of the Centaur 50 CSGT hot section showing the three primary components targeted for ceramic substitution: the first-stage rotor blade, the first-stage nozzle, and the combustor liner.

Figure 2. Combustor Exit Temperature Envelopes for First Stage Blade and Nozzle

Ceramic Blade Design

The firtree root attachment of the current Centaur 50 metallic first-stage blade was modified to

OBJECTIVES

The overall objective of the DOE Ceramic Retrofit Program (Ceramic Stationary Gas Turbine (CSGT) Development Program) is to improve the performance of stationary gas turbines in cogeneration through the selective replacement retrofit of metallic hot section parts with uncooled ceramic components. The successful demonstration of ceramic gas turbine technology, and the systematic incorporation of ceramics in existing and future gas turbines will enable more efficient engine operation, resulting in significant fuel savings, increased output power, and reduced emissions.

BACKGROUND

The performance of current industrial gas turbines is limited by the temperature and strength capabilities of the metallic structural materials in the engine hot section. Because of their superior high temperature strength and durability, ceramics can be used as structural materials for hot section components (blades, nozzles, combustor liners) in innovative designs at increased turbine firing temperatures. The benefits include the ability to increase the turbine rotor inlet temperature (TRIT) to about 1200°C (1192°F) or more with uncooled ceramics. Once the basic ceramic turbine technology has been demonstrated the potential to optimize the flow path and aerodynamics at these higher TRIT's may result in further performance improvements.

It has been projected that fully optimized stationary gas turbines would have a ~20% gain in thermal efficiency and ~40% gain in output power in simple cycle compared to all metal-engines with air-cooled components [1]. Annual fuel savings in cogeneration in the U.S. would be on the order of

0.2 Quad by 2010. Emissions reductions to <10 ppmv NOx are also forecast [2].

In September of 1992, the Department of Energy, Office of Industrial Technologies, initiated a program aimed at achieving energy savings and environmental emissions reductions for cogeneration applications by selectively replacing metallic blades, nozzles, and the combustor liner in existing industrial gas turbine engines with ceramic components. The technology base for this program is provided by the significant advancements in ceramic fabrication developed under past ceramic turbine programs for transportation and military applications.

Solar® Turbines Incorporated is the prime contractor under a six-year, three-phase program. The Solar-led team also includes major suppliers of ceramic turbine components, nationally recognized test laboratories, and a cogeneration end user. The Solar team and the responsibilities of the team members are shown in Table 1.

The engine selected for the program, the Solar Centaur® 50 (formerly known as the Centaur Type 'H') will be retrofitted with first-stage ceramic blades, first-stage ceramic nozzles, and a ceramic combustor liner. The engine hot section is being redesigned to adapt the ceramic parts to the existing metallic support structure. The program engine with the components targeted for ceramic insertion is shown in Figure 1. The ceramic turbine technology to be developed under this program is a key enabling technology to realize the performance and environmental goals of the Advanced Turbine Systems (ATS) program. The overall program approach, design strategy, and materials selection has been presented previously [3]. This review summarizes progress on the program from the start of Phase II through August, 1994.

Figure 3. Schematic of Centaur 50 CSGT Hot Section

accommodate the ceramic root-to-metal disk interface. Two ceramic blade designs have been developed. One is based on a "dovetail" type root attachment (55° angle) using a compliant layer to ensure uniformity of pressure face loading. The second is a "pinned" type root attachment wherein a single pin retains the blade in the disk. The airfoil has been slightly modified to eliminate a vibrational interference. Figure 4 shows solid models representing the two ceramic blade configurations.

Steady state and transient stress analyses have been performed for the "dovetail" configuration. Maximum stresses appear at steady state conditions due to the relatively slow start-up ramp and the fast shutdown ramp. Steady state temperature and stress profiles for the dovetail blade are shown in Figure 5. The stress profile includes the contributions of centrifugal, aerodynamic and thermal loading. The maximum principal stress (MPS) occurs in the neck region at a value of about 214 MPa (≈ 31 ksi). Similar analyses have been conducted for the pinned

root design. The maximum stresses in a modified version of the pinned root design of Figure 4 resulted in stresses of 255 MPa (37 ksi) in the neck region.

**(A)
Pinned Root Blade**

**(B)
Dovetail Blade**

Figure 4. Solid Models of Two Ceramic Blade Designs

(A) Temperature Distribution

(B) Stress Distribution

Figure 5. Steady State FEA Temperature and Stress Distribution for First Stage Turbine Dovetail Blade

Reliability analysis using NASA CARES and SPSLIFE [8,9] life assessment computer routines have been performed during various stages of blade design. Early analysis data on preliminary "dovetail" designs similar to the final design indicated about 98% probability of survival (POS) over 30,000 hrs of component life for a single as-processed blade with machined root using NT154 silicon nitride. NT154 is a predecessor material to NT164, one of the candidate materials for blade fabrication. Slow crack growth was identified as the life limiting

failure mode. The POS improved to 99.99% for a fully machined blade in which the reacted surface layer of NT154 (which lowers as-processed strength) was effectively removed [10]. This suggests that localized surface treatment through abrasive flow machining or some other means of surface finishing could be used to improve the POS for this blade design.

Ceramic Nozzle Design

The conceptual nozzle design was derived from the current (cooled) first-stage nozzle design of the Centaur 50 engine. For ease of fabrication, the current metallic two-airfoil nozzle was replaced by two ceramic single-airfoil nozzles, and the current integral tipshoe was decoupled from the outer nozzle shroud in the metal nozzle.

The nozzle design presented several challenges. These include minimizing stresses at the critical hot-spot locations in the airfoil, the attachment to the metallic support structure, and the vibrational response due to the mounting scheme, and minimizing blade clearances.

The stress profile in the nozzle is essentially the result of the response to the combustor exit temperature profile and pattern factor. Ideally, a shallow profile is preferred because the associated nozzle stresses will be low, but limitations of the metallic first stage disk and of the metal nozzle case precludes such a profile. A conservative design approach has been used which assumes a "worst case" scenario represented by the temperature gradient and associated stresses resulting from a combination of hot spot conditions on the trailing edge and lowest temperature conditions near the shrouds of the nozzle. This worst-case scenario corresponds to about a 537° C (966° F) maximum temperature difference (see Figure 2).

Several nozzle configurations have been analyzed including a single shrouded configuration

attached to the nozzle case via an intermediate ceramic ring attachment as well as several two-shroud nozzle designs attached to the first-stage diaphragm. All integral nozzle designs based on the current aerodynamics of the Centaur 50 first-stage metallic nozzle have high stresses in the 393-483 MPa (57-70 ksi) range in the critical trailing edge region. These stresses are known to be unacceptable in view of the target component design life of 30,000 hours, representing an average Time-Between-Overhaul (TBO) for a gas turbine in the field.

NASA CARES/LIFE and SPSLIFE life prediction analyses performed on a representative single shrouded nozzle design developed in Phase I gave Probability of Survival (POS) values that were unacceptable. Slow crack growth and creep were found to be the limiting failure modes for one of the silicon nitride materials evaluated under the program and silicon carbide was found to be limited by slow crack growth. Details of the life prediction analysis have been reported [10]. The life prediction results indicated that stress and/or temperature levels on the nozzle airfoil needed to be lowered to the 140-210 MPa (20-30 ksi) range to provide acceptable service life.

A number of potential design solutions were evaluated to lower the stress levels. Cooling of the airfoil can be used to lower the temperature in the critical trailing edge region and thereby reduce the temperature gradient and resulting nozzle stress. It was estimated that stresses could be reduced to about 172 MPa (25 ksi) using this approach. However, significant development activities were envisioned to assess the efficacy of the cooling approach, and to determine the impact on component strength of incorporation of cooling apertures. This approach was therefore abandoned in favor of a simpler design solution requiring less development efforts.

Various segmented designs were also evaluated. FEA showed steady state stresses in the trailing edge location as low as 103 MPa (15 ksi) could be achieved in multi-piece design concepts. Although stresses are perhaps lower than in any other configuration the segmented design approach was abandoned because of considerations of cost, and concerns about contact stresses and leakage.

The design solution ultimately adopted was based on a strategy to lower the stresses in the critical airfoil trailing edge region through redesign. The elements of the redesign process included shortening of the airfoil chord, and axial and tangential displacements. The redesign and optimization process was an iterative effort in which aerodynamic, mechanical design and analytical considerations were taken into account to provide an overall balance of properties.

The final design solution satisfying the reliability goal of the program was an airfoil bowed in the tangential and axial directions as seen in the solid model of Figure 6. Maximum steady state stresses in the airfoil are about 179 MPa (26 ksi). Steady state temperature and stress maps for the "bowed" nozzle design are shown in Figure 7.

Figure 6. Solid Model of the "Bowed" Nozzle Design

(A) Temperature Distribution

(B) Stress Distribution

Figure 7. Steady State FEA Temperature and Stress Distribution for "Bowed" Nozzle Design

Detailed life prediction work on the "bowed" nozzle design using the NASA CARES/LIFE and Sundstrand's SPSLIFE codes is in progress. Preliminary data indicate acceptable POS values for several of the selected nozzle materials indicating that the "bowed" design will be sufficiently viable for fabrication of the first generation components.

Ceramic Combustor Liner Design

The CSGT combustor, developed to fit the Centaur 50 gas turbine, incorporates a ceramic liner in the hot cylindrical wall surfaces of the metallic SoLoNOx combustor. The layout schematic of Figure 3 pictures the combustor in cross section and displays a generic liner "insert" in position. Concern about "sticking" of the monolithic ring and tile segments resulted in a decision to lower the maximum liner temperature from the initial target of 1371°C (2500°F) to 1149-1204°C (2100-2200°F). The uncooled ceramic liner walls will enable the combustion system to demonstrate operation in a stable lean firing condition. The current metallic SoLoNOx combustor has NOx emission levels in the 42-25 ppmv range. The DOE program goal is to demonstrate NOx emissions levels of 25 ppmv or lower, and Solar has set a NOx emissions goal of 10 ppmv or lower. As for the nozzle, critical factors are minimizing stresses, and attaching the ceramic liner to the metallic support structure.

Three ceramic insert designs for the CSGT combustor are being evaluated: integral continuous fiber-reinforced ceramic matrix composite (CFCC) cylindrical liners, concentric monolithic rings, and axial monolithic tiles. The three designs have a common envelope in the combustor configuration shown in the hot section schematic of Figure 3. The CFCC design consists of a set of two concentric integral cylinders. The outer cylinder is surrounded by a layer of refractory fiber which acts as both a mechanical and thermal barrier for the CFCC which will reach an operating temperature of ~1149°C (2100°F). On the back side of the refractory is a metallic structure which applies a radially inward force to retain the CFCC in position. The inner CFCC liner is retained by the same mechanism. Finite element temperature and stress analyses for the CFCC liner design are in progress. Figure 8 shows FEA temperature and FEA maximum principal stress maps for the inner combustor liner

NCV 2 194
 9:14:12
 PROF N 2
 POUT 27422
 JIB 1
 JIB 1
 JIB 1
 JIB 1 (AVG)
 SMO - 1 212
 SMOH 1251
 SMOX 11.56
 SMOY 126.5

 XV -1
 TV -1
 ZV -1
 *DIST 7"
 XZ 4
 ANGZ -3
 PRETEND HEIGHT
 L
 1245
 2500
 1755
 5070
 6266
 7521
 8776
 10011
 11266

CSCI COMPOSITE INS LINES 14" DIA DEL IN COMPOSITE M B-again 110494

Figure 8. FEA Maps for DuPont Lanxide Composites SiC/SiC CFCC

A second design approach, uses monolithic silicon carbide ring segments as shown in the solid model of Figure 9A. A series of four 5 cm (2 inch)

Redesign of the metallic components interfacing with the ceramics is an important effort under the program. Redesign is required to ensure thermal matching of adjacent ceramic and metal parts. Several design changes were also made to accommodate the heat management in view of the increase in TRIT from 1010° C (1850° F) to 1121° C (2050° F) and elimination of cooling in the first stage blade and nozzle. The design and materials changes have been summarized in Table 3. The changes were based on a design service life target of 10,000 hours which was selected to provide sufficient margin for the 4,000 hours Phase III field test. The incorporation of rim seals on the first-stage disk, a significant change from the current design, was necessary to provide adequate stress rupture margin on the disk which will experience an increased heat flux because of the TRIT increase, the altered combustor radial profile, and the elimination of blade cooling. The rim seals can be seen on the layout in Figure 3.

(A) Ring Design

(B) Tile Design

Figure 9. Solid Models of Monolithic Combustor Designs

Table 3. Materials Changes in Secondary Components

Component	Current Material	New Material
Stage 1 disk	V-57	Waspaloy
Front and Aft Rim Seals	Not in Current Centaur 50	Waspaloy
Stage 2 disk	V-57	Waspaloy
Stage 2 blade	IN-738LC	DS MAR M-247
Stage 2 diaphragm	N-155	Incoloy 903
Nozzle Support Housing	D-5B	Incoloy 909

Another major activity has been the development of a FEA model of the static structure of the engine. The model predicts the expansions of the hot section assembly and enables the prediction of thermal movement during start up, shut down, and trip conditions. It also predicts all clearances in the hot section. The latter information is important, for it enables the designer to optimize efficiency and avoid undesirable component contact.

Task 8: Design and Procurement of Test Facilities

Task 8 involves the set up and modification of all test facilities for the program. The actual testing is performed under Tasks 9 and 10.

Testing Methodology

Figure 10 illustrates the integrated design and test philosophy of the CSGT program. In this approach design analysis is iterated with life prediction, testing, and post-testing component evaluation.

Figure 10. Schematic of Iterative Design/Testing Strategy

Initially, simulated components are being evaluated in test rigs to prove key design concepts such as blade root configuration and compliant layers for the blade and fixturing to the metallic support structure for the combustor liner. Subsequently, the findings from these tests are being fed back into the design of the first component prototypes which will first be evaluated in proof testing, and subsequently be tested in the gasifier rig. The results of the first generation prototype testing will be used to modify the component designs. Based on the design modifications, second generation parts will be manufactured, which will also be tested in the gasifier rig. Second generation parts with superior performance will be used for the 4000 hour engine field test in Phase III of the program. The test sequence for the three components is illustrated in Figure 11.

Figure 11. Flowpath of the Sequential Testing of the Three Program Components

To CSGT ceramic component testing is aimed at generating empirical stress/temperature distribution data on subscale and full scale components under simulated engine operating conditions. Proof testing in which a load is applied equal to a computed overstress condition is an important aspect of the overall test strategy.

The component maximum stress under the proof load will be greater than the analytically predicted engine operating design stress and the overstress factor eliminates defective components (e.g. with processing related flaws) that could have resulted in failure in subsequent engine testing. Ideally, these tests should closely approximate the analytically predicted component stress fields to obtain the most comprehensive results. However, it is recognized that exact duplication of the engine operating conditions is generally not achievable in a rig simulation. Therefore, the test must focus on a duplication of the stress field in the most critically stressed portions of the component.

Figure 12 shows the combustor can rig. It will be used to evaluate subscale combustor cans of the candidate materials for the program. Figure 13 shows the gasifier rig which is a full production

Figure 12. Subscale Can Combustor Test Rig

Figure 13. Gasifier Rig

Centaur 50 engine mated to a water brake dynamometer which replaces the generator to absorb the electrical load. The gasifier rig engine was factory tested and met all requirements for a production Centaur 50 engine including emissions.

The gasifier rig test matrix is designed to evaluate each of the ceramic component sections individually (i.e. first stage blades, first-stage nozzles, combustor) and together. The gasifier rig testing will start at a firing temperature of 1010°C (1850°F) which is the TRIT of the baseline all-metal Centaur 50 engine. In subsequent testing the firing temperature will be increased in 56°C (100°F)

increments to a final TRIT of 1121°C (2050°F). In the final rig test all ceramic components will be combined in one build and tested as were the individual ceramic components. A successful engine build will then be subjected to cyclic (start/trip shutdown) testing for periods of up to 50 hours during Phase II. Each engine build will be disassembled and the components will be inspected visually and by metallography and nondestructive evaluation.

Task 9: Ceramic Component Iterative Design, Manufacturing, and Testing

Work under Task 9 includes fabrication of specimens, subscale and full scale components, testing of specimens, simulated blades, and prototype blades and nozzles. Testing of subscale and full scale combustor liners is performed under Task 10.

Rotor Blade: Materials Properties, Fabrication and Testing

The three blade suppliers selected in Phase I are fabricating blades of the two designs developed under Task 7. The dovetail blades are being fabricated by AlliedSignal Ceramic Components of their GN-10 Si_3N_4 material and by Norton Advanced Ceramics of their NT164 Si_3N_4 material. Kyocera Industrial Ceramics Corporation is fabricating the pinned root blade from their SN-253 Si_3N_4 material. These suppliers have also fabricated attachment specimens and flexure bars for evaluation of basic properties under the blade operating conditions.

Testing of ceramic inserted blade attachment concepts to a metallic disk involves a matrix of both static and dynamic tests. Because ceramic blade to metallic disk attachments typically degrade due to the combination of: 1) centrifugal (CF) blade loads on a ceramic to metallic interface, and 2) relative motion (tangential sliding) at the loaded interface due to thermal expansion of the disk rim, all CSGT

blade attachment testing focuses on these two primary conditions. Attachment concepts for the dovetail and pinned root blade designs are being evaluated in the test sequence.

Figure 14 shows the double ended attachment test specimens which have been procured from the three ceramic blade suppliers for tensile testing in an Instron tensile test apparatus. The attachments simulate the root configurations of the two blade designs shown in Figure 4. The candidate attachment configurations are being evaluated for:

- 1) full CF blade loads,
- 2) cyclic temperature and CF load conditions, and
- 3) room and elevated temperature failure loads.

Figure 14. Blade Attachment Specimens for Tensile Testing

Tables 4 and 5 summarize the results of attachment testing for the dovetail and pinned root designs to-date. At room temperature the dovetail specimens failed at stresses $\approx 3\text{--}4\times$ the design stress.

The performance of the compliant layer material is being evaluated in the attachment testing of the dovetail configuration. A thin (0.13 mm/0.005 in.) nickel compliant layer performed well during room temperature attachment testing. A nickel-based alloy is being used for testing at 760°C (1400°F), the approximate steady state temperature of the blade root. Testing to-date has shown that the selected compliant layer material performed well under the blade root conditions.

A single test was also performed at the steady state blade root temperature. The failure load in this test was markedly lower than in the room temperature test. Still, a $2\times$ margin over the design stress was observed. Some sticking of the compliant layer was observed in this test. Post-test evaluation is underway to establish whether contact stresses were responsible for the lowering of the failure load at the elevated temperature of the test. The compliant layer will be modified to eliminate the sticking concern.

The pinned root attachment specimens failed at stresses between 540 and 620 MPa (78 and 90 ksi) close to the steady state stress level predicted for the design of Figure 4B. Attachment specimens and pins were provided by each of the three suppliers selected for blade fabrication. To evaluate the effect of surface finish various combinations of attachment specimens and pins with different surface finishes (between 8 and 32 μin) were evaluated. However, the attachment test results did not indicate an effect of surface finish on the failure load.

The pinned root blade design has been modified to lower stress levels to ≈ 255 MPa (37 ksi). It is expected that the design change will provide a margin of safety by a factor of 2-3. Attachment specimens of the modified pinned root blade design have been fabricated and will be tested under conditions similar to those for the dovetail design.

Cold spin testing will be initially conducted on dummy mass blades machined from attachment specimens (Figure 15). Spin tests to rotational speeds that provide simulated engine CF blade loads on the dummy blades will be performed at ambient conditions. Because elevated temperature data can be obtained from attachment testing under conditions approximating the blade root in service hot spin testing is not considered at this point in the program. Cold spin testing will also be performed on first and second generation blades in a proof test. The blades will be tested at speeds that will be

Table 4. Results of Attachment Testing on the Dovetail Blade Root Design

Material	Test Temperature	Failure Stress	Comments
GN-10 Si ₃ N ₄	Room Temperature	221 MPa (32 ksi)	Test performed in duplicate stopped at 100% design load.
GN-10 Si ₃ N ₄	Room Temperature	924 MPa (134 ksi)	Failure in dovetail neck section
NT164 Si ₃ N ₄	Room Temperature	855 MPa (124 ksi)	Failure in dovetail neck section
SN-253 Si ₃ N ₄	Room Temperature	745 MPa (108 ksi)	Failure in dovetail neck section
SN-253 Si ₃ N ₄	760° C (1400° F)	427 MPa (62 ksi)	Failure in dovetail neck section

Table 5. Results of Attachment Testing on the Pinned Root Blade Design

Attachment Specimen Material	Pin Material	Test Temperature	Failure Stress	Comments
GN-10 Si ₃ N ₄	GN-10 Si ₃ N ₄	Room Temperature	550 MPa (80 ksi)	Pin failure, internally initiated
GN-10 Si ₃ N ₄	SN-253 Si ₃ N ₄	Room Temperature	587 MPa (85 ksi)	Fracture in attachment
NT164 Si ₃ N ₄	NT164 Si ₃ N ₄	Room Temperature	540 MPa (78 ksi)	Pin failure in radial machining groove
SN-253 Si ₃ N ₄	SN-253 Si ₃ N ₄	Room Temperature	620 MPa (90 ksi)	Fracture in attachment
SN-253 Si ₃ N ₄	SN-253 Si ₃ N ₄	Room Temperature	540 MPa (78 ksi)	Fracture in attachment

Figure 15. Dummy Blade Specimens for Cold Spin Testing

increased to 120% of the design stress. Blades that pass the proof test will be used in the gasifier rig test. The spin rig designated for the CSGT program has been uprated to ensure adequate safety for the testing envisioned.

Blade fabrication is in progress. Figure 16 shows NT164 Si_3N_4 sintered and pre-sintered blade castings.

Testing in support of the blade development is being conducted at Solar. Because life prediction indicated slow crack growth as the limiting failure mode under service conditions [10] experimental work is being conducted to supply materials property data for life prediction. Table 6 is a compilation of blade materials property data from testing at Solar. The data at 760°C (1400°F), the approximate steady state temperature of the blade root was collected on machined flexure bars since the blade root is machined. At this temperature all three materials have fast fracture and slow fracture strength $\approx 2\text{-}4\times$ the design stress of the two blade configurations in this area. One material, NT164, was also investigated at 982°C (2000°F), a temperature close to the steady state temperature condition on the blade airfoil. Life prediction studies with CARES/LIFE and SPSLIFE for the detailed blade designs and three selected materials are in progress.

Figure 16. Sintered and Pre-Sintered NT164 Si_3N_4 Blade Castings

Nozzle: Materials Properties, Fabrication and Testing

Four materials were considered for the fabrication of the bowed nozzle. Data base and life prediction considerations figured prominently in the selection of SN-88 and SN-253 for prototype nozzle fabrication.

Table 7 shows relative materials property data from testing at UDRI and Solar. Tensile testing at the approximate nozzle hot spot temperature ($1288^\circ\text{C}/2350^\circ\text{F}$) showed the highest fast fracture strength for SN-253. Tensile dynamic fatigue testing at UDRI showed loss of strength at the lower crosshead speed for three of the four materials. SN-88 showed negligible SCG behavior at the nozzle design temperature. Flexure dynamic fatigue testing at Solar gives somewhat lower slow fracture strength as fast fracture strength, but the differences are not as marked as for the UDRI data, presumably because the crosshead speed difference between FF and SF strength is only a factor 10^2 instead of 10^4 in the UDRI study.

Life prediction with CARES/LIFE and SPSLIFE is in progress.

Prototype nozzles will be evaluated in a proof test in which a transient temperature gradient creates

Table 6. Materials Property Data for Blade Conditions

PROPERTY	GN-10		NT164		SN-253	
	MPa/ksi(#)	m	MPa/ksi(#)	m	MPa/ksi(#)	m
FF Strength						
Room Temperature	793/115 (20)	10	917/133 (20)	9	834/121 (20)	15
760° C (1400° F)	676/98 (20)	9	896/130 (20)	7	793/115 (20)	11
982° C (2000° F)	-	-	676/98 (15)	8	-	-
SF Strength						
Room Temperature	662/96 (20)	12	786/114 (20)	8	800/116 (10)	10
760° C (1400° F)	607/88 (20)	25	793/115 (20)	8	752/109 (10)	10
982° C (2000° F)	-	-	572/83 (15)	7	-	-

- NOTES: (a) Room temperature and 760° C (1400° F): Machined flexure bars,
982° C (2000° F): As-fired flexure bars
(b) Number of specimens in brackets (#)
(c) m = Weibull Modulus
(d) FF: Fast fracture strength at crosshead speed of 0.004 cm/s
(e) SF: Dynamic fatigue strength at crosshead speed of 0.00004 cm/s

stresses in excess of the design stress under worst case combustor exit temperature conditions. A rig for this purpose is being designed.

Non-Destructive Evaluation

Development of appropriate non-destructive evaluation (NDE) methodology is an integral aspect of the program. This work is performed in a collaborative effort by the Caterpillar Technical Center (CAT TC) and Argonne National Laboratory (ANL). The initial methodology development work and component evaluations are the responsibility of ANL while CAT TC has responsibility for the coordination of the NDE effort and for technology transfer to Solar.

The Phase I NDE activities focused on a review of techniques currently used for NDE of ceramics.

The methods are principally visual inspection, density measurements, microfocus x-ray radiography, and fluorescent dye penetrant inspection (FPI). Microfocus x-ray radiography is used to examine internal defects (inclusions, voids, cracks) at both the powder compact and final part stages for monolithic components and tensile bars. FPI is used for final part inspection for surface defects (cracks, porous regions) of all monolithic components and test specimens. All of the above NDE techniques will be used for the monolithic components under the program. Laser scattering has proven to be an additional useful technique for monolithics. It provides information on surface cracks with enhanced sensitivity compared to FPI and also enables subsurface examination.

The predominant techniques used for the CFCC materials are through-wall radiography, selected area

Table 7. Materials Property Data for Nozzle Conditions

PROPERTY	SN-88		NT164		Hexoloy® SA		SN-253	
	MPa/ksi(#)	m	MPa/ksi(#)	m	MPa/ksi(#)	m	MPa/ksi(#)	m
FF Strength								
1288°C (2350°F) -UDRI	455/66 (10)	8	441/64 (10)	8	455/66 (10)	8	586/85 (10)	23
1288°C (2350°F)- Solar*	621/90*	9	552/80 (15)	-	503/73*	-	552/80*	-
SF Strength								
1288°C (2350°F)-UDRI	441/64 (10)	25	276/40 (10)	-	393/57 (10)	5	386/56 (10)	27
1288°C (2350°F)-Solar	586/85 (15)	10	503/73 (7)	-	434/63 (15)	-	-	-
Creep								
1288°C (2350°F)-Solar	241/35 (1) 1100 hrs		241/35 (1) 3000 hrs					

- NOTES: (a) UDRI machined tensile bars
 (b) Solar as-fired flexure bars, * data from other studies
 (c) No. of specimens in brackets (#)
 (d) m = Weibull Modulus
 (e) FF: Fast fracture strength at crosshead speed of 0.04 cm/s (UDRI); 0.004 cm/s (Solar)
 (f) SF: Dynamic fatigue strength at crosshead speed of 0.000004 cm/s (UDRI); 0.00004 cm/s (Solar)

computed tomography (CT) and infrared imaging. Infrared imaging has proven to be a useful technique for the detection of delamination of CFCC materials. Figure 17 shows infrared and x-ray CT scans of a subscale combustor liner showing about 10% delamination. There is good correlation between the two methods.

Extensive NDE characterization has taken place at ANL on test specimens and simulated and subscale components fabricated to-date.

Task 10: Low Emissions Combustor

This task involves all the subscale and full scale combustor component testing of the designs developed under Task 7.

Subscale combustor parts have been fabricated. Figure 18 shows subscale combustor liners from the four suppliers. Testing of these liners in the subscale combustor can rig is in progress. In the can rig test the analytically predicted stress distribution of a full size annular liner are duplicated both in steady state and transient operation. A computed temperature distribution is imposed on subscale cans to approximate the desired stress/temperature field. The maximum pressure in the Subscale Combustor Can rig is about 690 Pa (100 psi) which is similar to the pressure in the SoLoNOx combustor during engine operation. Mechanical and/or aerodynamically induced loads are applied directly to ensure that the interface zones are also accurately assessed. The candidate combustor concepts will be evaluated in this rig through detailed post-test

Figure 17. Infrared and X-ray CT Scans of CFCC Combustor Liner

From left to right: DLC SiC/SiC CFCC liner, B&W Alumina/Alumina CFCC Liner, Carborundum Hexoloy® SA tiles, NAC NT230 rings

Figure 18. Subscale Combustor Liner Parts

characterization of the components. Designs and materials that develop observable damage and/or interface distress in short term tests will be eliminated.

Experience has been accumulated on the subscale can rig by evaluating both monolithic and CFCC combustor liners under an in-house program. The can geometries typically used were 20 cm (8 in.) diameter x 40 cm (16 in.) long. Testing times have usually been short, typically 2 hours in duration, but one Nicalon/SiC combustor can was tested and survived for 100 hours in a cyclic test (1500 cycles)

between 704°C and 1204°C (1300 and 2200°F). Current testing will be conducted on can configurations that are 20 cm (8 in.) long.

The atmospheric rig evaluates full size combustor system components to both simulated engine transient and steady state thermal conditions at atmospheric pressure. The rig is electronically controlled to provide accurate and repeatable test conditions for the candidate liner materials and design concepts. All components will be fully characterized following testing to determine component and interface integrity as well as the integrity of the combustor support system. This rig will serve as the proof test rig for all full size combustor assemblies prior to engine installation.

The loop rig evaluates full size ceramic combustor system components to simulated engine transient and steady state temperature and pressure load conditions. The combustor test section is part of a "workhorse" turbine engine flowpath in which compressor discharge flow is diverted outside of the engine into the combustor test section and then returned to the engine turbine section downstream of the combustor test section. This "loop" concept accurately exposes the ceramic combustor components to actual engine operating conditions without the potential of causing secondary damage to downstream components in the event of a combustor failure. Complete "hot wall" combustor performance can be measured in this rig including emissions levels and pattern factor.

FUTURE WORK

Work to-date under Phase II has essentially involved the completion of the detail design of the ceramic parts and associated secondary components, set up of the rig test facilities, fabrication of test specimens, simulated and subscale component fabrication, and extensive materials property evaluation. Fabrication of two blade designs is in

The work for the remainder of Phase II involves the following elements:

1. Completion of the first ceramic engine design.
2. Fabrication by the suppliers of prototype first generation ceramic blades and nozzles.
3. Evaluation of prototype ceramic blades, nozzles, and (subscale) combustor liners in rig testing including gasifier rig testing.
4. Iterative design of program components based on the results of testing.
5. Second generation component fabrication.
6. Evaluation of second generation components in rig testing.
7. Completion of the long term testing of ceramics. For monolithic ceramics this includes dynamic fatigue testing to establish slow crack growth behavior, creep testing at UDRI, and oxidation testing. For CFCC's this involves burst strength testing, tensile testing and stresses oxidation testing. The objective of the testing is to gather a sufficient materials property data base for component life prediction.
8. Evaluation of component designs in life prediction at Sundstrand and Solar.
9. Application of NDE methodology to program components and test specimens.
10. Plan for the Phase III 4,000 hrs engine test at ARCO.

REFERENCES

[1] Anson, D. Sheppard, W.J., Norland, L.D., and DeCorso, M., "Application of Ceramics to Industrial

Gas Turbines; Tasks 2, 3, and 4, Economics, Market Analysis and R&D Plan", Tasks 2, 3, and 4 Final Report, Battelle Memorial Institute, DOE Contract No. DE-AC02-89CE40979, 1991.

[2] Anson, D., Sheppard, W.J., and Parks, Jr., W.P., "Impact of Ceramic Components in Gas Turbines for Industrial Cogeneration", ASME paper 92-GT-393, presented at the International Gas Turbine and Aeroengine Congress and Exposition, Cologne, Germany, June 1-4, 1992.

[3] van Roode, M., Brentnall, W.D., Norton, P.F., and Pytanowski, G.P., "Ceramic Stationary Gas Turbine Development", ASME paper 93-GT-309, presented at the International Gas Turbine and Aeroengine Congress and Exposition, Cincinnati, Ohio, May 24-27, 1993.

[4] van Roode, M., Brentnall, W.D., Norton, P.F., and Boyd, G.L., "Ceramic Stationary Gas Turbine Development Program - First Annual Summary -", ASME paper 94-GT-313, presented at the International Gas Turbine and Aeroengine Congress and Exposition, The Hague, The Netherlands, - June 13-16, 1994.

[5] Anson, D., W.P. Parks, Jr., Evensen, O., and van Roode, M., 1993, "Advanced Small Gas Turbines for Cogeneration", paper presented at the 7th Congress & Exposition Gas Turbines in Cogeneration & Utility, Industrial and Independent Power Generation, Bournemouth, England, U.K., September 21-23, 1993.

[6] van Roode, M., "Ceramic Retrofit Program", paper presented at the Joint Contractors Meeting, FE/EE Advanced Turbine Systems Conference, FE Fuel Cells and Coal-Fired Heat Engines Conference, U.S. Department of Energy, Office of Fossil Energy, Morgantown Energy Technology Center, Morgantown, West-Virginia, August 3-5, 1993.

[7] "Ceramic Stationary Gas Turbine Development Program", 1993, Phase I Final Report, Solar Turbines Incorporated, DOE Contract Number DE-AC02-92CE40960. In print.

[8] Nemeth, N.N., Mandersheid, J.M., and Geykenyesi, J.P., "Ceramics Analysis and Reliability Evaluation of Structures (CARES)", NASA Technical Paper 2916, 1989.

[9] Bornemisza, T., and Saith, A., "SPSLIFE: A User Friendly Approach to the Structural Design and Life Assessment of Ceramic Components", ASME paper 94-GT-486 at the International Gas Turbine and Aeroengine Congress and Exposition, The Hague, The Netherlands, June 13-16, 1994.

[10] Saith, A., Norton, P.F., and Parthasarathy, V.M., "Application of SPSLIFE to Preliminary Design Evaluation and Life Assessment of CSGT Components", ASME paper 94-GT-420, presented at the International Gas Turbine and Aeroengine Congress and Exposition, The Hague, The Netherlands, - June 13-16, 1994.

Session 4

ATS Project Related Activities

LONG-TERM TESTING*

CONTRACT INFORMATION

Contract Number	ED3201000
Contractor	Oak Ridge National Laboratory P. O. Box 2008 Oak Ridge, TN 37830 Phone: (615) 574-5150 Fax: (615) 574-6098
Contract Project Manager	Mike Karnitz
Principal Investigators	Matt Ferber George A. Graves, Jr.
DOE Project Manager	Gene Hoffman
Period of Performance	October 1, 1993 to October 1, 1994

* Research sponsored by the DOE, Assistant Secretary for Energy Efficiency and Renewable Energy, Office of Industrial Technologies, Industrial Energy Efficiency Division and Advanced Turbine Systems Program, under Control DE-AC05-84OR21400 With Martin Marietta Energy Systems.

OBJECTIVES

The primary goal of this research is to determine the long-term survivability of ceramic materials for industrial gas turbine applications. Research activities in this program focus on the evaluation of the static tensile creep and stress rupture (SR) behavior of three commercially available structural ceramics which have been identified by the gas turbine manufacturers as leading candidates for use in industrial gas turbines. Tensile creep data will be generated in air by measuring creep strain as a function of time, applied stress, and temperature. The SR resistance will be evaluated by continuing each creep test until the specimen fails. For each material investigated, a minimum of three temperatures and four stresses will be used to establish the stress and temperature sensitivities of the creep and SR behavior. The test matrix utilized in this program is intended to extend the test conditions investigated by the engine component manufacturers. In the latter case, these conditions focus primarily upon the evaluation of the creep and SR behavior of two candidate materials at one temperature only. Consequently, the temperatures chosen for the current effort will bracket that used by the engine component manufacturers. Because existing data for many candidate structural ceramics are limited to testing times less than 2,000 h, this program will focus on extending these data to times on the order of 10,000 h, which represents the lower limit of operating time anticipated for ceramic blades and vanes in gas turbine engines.

A secondary goal of the program will be to investigate the possibility of enhancing life prediction estimates by combining interrupted tensile SR tests and tensile dynamic fatigue tests in which tensile strength is measured as a function of stressing rate. The third goal of this program will be to investigate the effects of water vapor upon the SR behavior of the three structural ceramics chosen for the static tensile studies by measuring the flexural strength as a function of

stressing rate at three temperatures.

BACKGROUND INFORMATION

DOE-OIT (Office of Industrial Technology) has initiated a program to develop ceramic components for use in industrial gas turbines. The objective of this program is to improve the efficiency of land-based gas turbines that are used for industrial cogeneration. As part of this program, Battelle Columbus completed an assessment and developed a program plan in March 1991. The assessment reviewed the status of ceramic materials in the context of gas turbine engine development and concluded that significant performance gains could be expected from the use of ceramics. Based on the assessment and manufacturers recommendations, a five-year program plan was developed. The program was designed to bring ceramic technology to the point where short-term reliability and engine performance had been demonstrated. This would provide a technology base from which manufacturers' could move forward with engine performance improvements, field testing, and commercial development.

Based on the program plan, OIT issued a solicitation and selected Solar Turbines Incorporated for the development of ceramic gas turbine components. Solar Turbines started working in late FY 1992. One of the critical areas outlined in their program was a long-term materials testing program. Land-based gas turbines are significantly different from automotive gas turbines in that they are designed to operate for 50,000 h or greater (compared to 5,000–10,000 h). Long-term materials tests are needed to determine the survivability of the materials for land-based applications. This paper outlines the long-term testing conducted at Oak Ridge National

Laboratory (ORNL) and University of Dayton Research Institute (UDRI) in support of the new program with Solar Turbines.

PROJECT DESCRIPTION

For long-term testing, ORNL will work with Solar Turbines in identifying materials to be tested. A minimum of three materials will be chosen for evaluation. For each material identified, tensile creep data will be generated in air by measuring creep strain as a function of time, applied stress, and temperature. The SR behavior will be evaluated by continuing each creep test until specimen failure occurs. A minimum of three temperatures and four stresses will be used to establish the stress and temperature sensitivities of the creep and SR behaviors. The temperatures chosen for the ORNL effort will bracket that used by the engine component manufacturers in their "in-house" creep and SR testing effort.

For each temperature considered, the creep and SR behavior will be established using a minimum number of 26 tensile specimens. Five additional specimens will be used to measure the fast fracture strength. The ORNL button-head specimen configuration will be used for these measurements. In order to establish long-term mechanical survivability, the applied stresses will be adjusted to obtain failure times ranging from 10–100, 100–1,000, 1,000–5,000, and 5,000–10,000 h. Initial estimates of these stresses will be based on preliminary mechanical reliability diagrams constructed from flexural dynamic fatigue and fracture toughness data generated at each temperature. Specimens not failing within the allotted time range will be fractured at temperature to determine the residual strength.

For each material investigated, the associated creep and SR data will be used to generate a mechanical reliability diagram describing the time-to-failure and failure mode (creep damage versus slow crack growth) as functions of stress and temperature. The applicability of various creep and SR models (Monkman-Grant, Sherby-Dorn, Larson-Miller, etc.) to the description of the experimental data will also be investigated. The final material selection will be based upon a comparison of these diagrams.

A secondary goal of the program will be to investigate the possibility of enhancing life prediction estimates by combining interrupted tensile SR tests and tensile dynamic fatigue tests in which tensile strength is measured as a function of stressing rate. This technique will be applied to one material at one temperature for interrupted times of 100 and 1,000 h. The data will be used to generate SR curves which will then be compared with actual results.

The third goal of this program will be to investigate the effects of water vapor upon the SR behavior of the three structural ceramics chosen for the static tensile studies by measuring the flexural strength as a function of stressing rate at three temperatures. If water vapor in the environment is found to be a problem, additional tensile creep and SR measurements will be made for verification purposes.

This work will be composed of the following eight Tasks:

Task 1 - Development of an experimental test plan, which outlines the scope of the test matrix.

Task 2 - Creep/SR comparison study between UDRI and ORNL.

Task 3 - Design, procurement, and initiation of tensile test facilities required for the 10,000 hour tests.

Task 4 - Procurement of the flexure and tensile specimens.

Task 5 - Long-term mechanical testing of three candidate materials. Initially, this effort will include the utilization of flexural dynamic fatigue to more clearly define the stress levels for the subsequent tensile creep/SR tests.

Task 6 - Develop and initiate experimental test plan for combining interrupted SR and dynamic fatigue.

Task 7 - Perform flexural dynamic fatigue studies to address the effects of water behavior on SR behavior.

Task 8 - Prepare final report.

RESULTS

Comparison Study

A study comparing the creep and SR measurement capabilities of ORNL and UDRI was continued during this reporting period. Because UDRI will assist ORNL in the completion of the long-term test matrix, any discrepancies in experimental data arising from the test equipment and methodologies used by the two laboratories must be identified and corrected early in the program. The comparison study involves the evaluation of the creep and SR behavior of a hot isostatically pressed (HIPed) silicon nitride (NT154 and NT164)^a at 1370°C, a dense silicon carbide (SA SiC)^b at 1500°C, and an

^a NT154 and NT164 silicon nitride ceramics, Saint-Gobain/Norton Industrial Ceramics Corporation, Northboro, MA.

^b Sintered Alpha silicon carbide, Carborundum Company, Niagara Falls, NY.

aluminum oxide (AD99)^c at 1150°C. Figure 1 compares strain versus time data generated for the NT154 at a stress of 150 MPa. The agreement between UDRI and ORNL is excellent. The difference between respective failure times is well within statistical scatter reported for this material.¹ As shown in Fig. 2, which is a plot of strain versus time for SA SiC, agreement between the UDRI and ORNL results was reasonably good. The differences between respective curves are well within statistical scatter reported for this class of high temperature materials.¹ Based on the comparisons conducted to date, there appears to be no discrepancy in creep strain measurement techniques.

Creep/Stress Rupture Facility

During this reporting period, considerable emphasis was placed upon the specification, procurement, and installation of the eight mechanical test machines required for the long-term testing phase of the program. Although these machines will be primarily used for static stress measurements (e.g., creep and stress rupture), they will also be capable of load-to-failure tests over a wide range of loading rates. Figure 3 illustrates the major components of the creep/stress rupture test machine. The load frame is similar to that used for stress relaxation tests. An electro-mechanical actuator attached to the outer position of a 20-to-1 lever arm provides for application of the mechanical loads. The load train includes the grips for the ORNL button-head tensile specimen (Fig. 4) and special knife-edge couplers which may be adjusted to minimize bending in the tensile specimen. A load cell provides the necessary feedback signal to the electronics used to control the electromechanical actuator. A short two-zone resistance furnace located between the water-

^c AD99 aluminum oxide, Coors Company, Golden, CO.

cooled grips provides for testing at temperatures up to 1500°C. The specimen displacements are measured over a 25.4 mm gage length using a direct contact capacitance extensometer.

When conducting a test, the load is maintained at the desired level by inputting the proper setpoint voltage to the controller where 0 to 10V corresponds to 0 to full-scale load. A second 0 to 10 V signal is used to set the actuator speed. This arrangement allows one to conduct either constant load tests or load-to-failure tests as a function of load rate. Both of the control signals are supplied by a data acquisition system (DAS) and personal computer. The DAS also accepts input signals from the load cell, extensometer, and temperature controller.

The operational tests performed at the seller's plant, Applied Test Systems (ATS), included (1) measurement of the percent bending as a function stress, (2) measurement of the temperature gradients across the gage section of the button-head specimen, and (3) verification of the load controller operation. The percent bending was measured using a PY6 silicon nitride^d button-head specimen having eight strain gages in the gage section (Fig. 4). The four gages located at the specimen midspan were used to measure the angular misalignment.^{2,3} The magnitude of the concentric misalignment or "S" bending^{2,3} was determined from the gages located at the ends of the gage section. A sophisticated strain gage conditioning system with a programmable "front-end processor" was used to simultaneously condition the strain gage signals, calculate the percent bending strain, and record the strain, percent bending error, and the applied load (stress).

Applied Test Systems personnel also demonstrated the use of a newly developed SLVC alignment fixture for the real-time measurement of bending across opposing shanks of the tensile specimen. This device, which was mechanically

clamped to the specimen shanks, basically gave an over-all indication of the longitudinal strain at 90° intervals over the entire gage section. In this sense, it served the same purpose as the four strain gages located at the specimen midspan. The readings from the alignment device were used to minimize bending via adjustments to the knife-edge coupler.

Three separate bending measurement tests, designated as Trials 1 through 3, were conducted. For Trial 1, no adjustments were made to the knife-edge grips. The percent bending decreased as the stress increased. At an applied stress of 150 MPa, the percent bending values for axial and concentric misalignments were 9.2 and 8.1, respectively. These numbers were subsequently reduced to 2.4 and 0.8 by adjusting the knife-edge grips in accordance with the strain readings supplied by the alignment device.

In order to examine the ruggedness of the alignment procedure, the specimen was unloaded and then subjected to two load/unload cycles during which time the percent bending was measured (Trial 2). The percent bending values for axial and concentric misalignments measured at 150 MPa were 4.8 and 2.7, respectively during the first cycle and 4.7 and 2.7, respectively during the second cycle. As a final verification, the specimen was completely unloaded before initiating another test. The percent bending measured at a given stress during Trial 3 was comparable to that measured in Trial 2. The fact that these percent bending versus applied stress curves were not strongly influenced by load/unload cycles indicates that the alignment process was fairly robust. More importantly, the use of the strain readings from the SLVC alignment device to minimize the percent bending (as measured by the strain gages) was found to be a valid approach.

In the next evaluation test, a silicon nitride button-head specimen having three equally spaced Type S thermocouples attached to the gage section was used to measure temperature gradients. Although it was possible to independently control the top and bottom zones on the furnace (through the specification of separate setpoints), no attempt

^d PY6 silicon nitride, GTE Laboratories, Inc., Waltham, Massachusetts.

was made to minimize the temperature gradients. For temperatures above 1000°C, the center and top regions were within 5°C of one another while the bottom region of the gage section was always 10 to 18°C hotter. Preliminary adjustments of the bottom zone controller indicated that the temperature difference between the bottom and center points on the specimen could be reduced to less than 5°C. For temperatures below 1000°C, the temperatures along the top and bottom regions were comparable while the specimen midpoint was 5 to 8°C cooler. This effect was attributed to the presence of an insulating spacer between the top and bottom zones of the furnace. The purpose of this spacer is to eliminate hot spots along the specimen midspan arising from radiation which becomes an important factor only at the higher temperatures (>1000°C). Therefore at temperatures below 1000°C, it should be possible to reduce the temperature gradients between the center and outer regions in the gage section by removing this spacer.

The verification tests of the load controller involved inputting a fixed control signal (0 to 10 V) and then observing the response of the load system, which included a metallic specimen. In general, the load on the specimen was held to within 10 N of the specified setpoint. The minimum and maximum loading rates were also found to be acceptable.

Following delivery of the eight SR/creep machines to ORNL, additional verification tests were conducted including (1) measurement of the percent bending as a function of stress, (2) assessment of the load control capabilities, (3) verification of the computer interface and software, and (4) measurement of extensometer resolution, hysteresis, and linearity.

The percent bending was measured using the same setup described above. Three separate bending measurement tests, designated as Trials 1 through 3, were conducted. In Trial 1, the PY6 was loaded to 7 kN and the grips adjusted to minimize bending. The adjustment was based on the output from the SLVC alignment fixture, which

was mechanically clamped to the specimen shanks. The specimen was unloaded to 1 kN and reloaded to 7 kN. As shown in Fig. 5, this load cycling did not significantly affect the axial or concentric bending numbers both of which remained below the target value of 5%. In Trial 2, the specimen was unloaded to 1 kN and the alignment fixture removed. The specimen was then reloaded to 7 kN. The bending numbers still remained below the 5% target indicating that removal of the alignment fixture had minimal influence on the specimen alignment. In Trial 3, the effect of introducing the contact extensometer upon specimen bending was examined. Even with a 3.0 psi side pressure exerted with the extensometer, the specimen bending (at 7 kN) was at or below the 5% level.

In assessment of the load control capabilities, the setpoint of the load controller was abruptly changed and the load response measured. Little or no over-shoot (under-shoot in the case where the load was abruptly decreased) occurred as the load approached its setpoint.

The verification tests of the computer interface involved confirming that all signals supplied by the test machine were properly received by the data acquisition system. The operation of the data acquisition software, which was developed using a commercial package (LabVIEW® 2^e), was also examined. Initial problems with the stability of the temperature signals was traced to a ungrounded temperature controller in the electronics console. The load signal was found to have an acceptable signal-to-noise ratio as long as the lever arm drive motor was disabled. However, excessive noise was introduced into the load signal when the drive motor was activated. The introduction of analog filters was found to reduce this noise to an acceptable level.

Operational tests with the software were also completed. The LabVIEW® 2 environment allows for the creation of virtual instrument control panels on the computer screen (Fig. 6). For each frame,

© LabVIEW® 2 software manufactured by National Instruments, Austin, TX.

load, extensometer displacement, and the upper and lower furnace zone temperatures are continuously displayed. Digital indicators include local/remote status, specimen break, furnace/water flow interlocks, and electrical supply status (L/R STAT, BREAK, SAFETIES, AND MAINS in Fig. 6). Provisions for digital control of the drive motor and furnace output relay are also available. When the test frame is in the remote mode, the load and actuator speed are controlled by the Setpoint and Speed settings in Fig. 6.

The acceptance tests for the ATS extensometer were less successful. As shown in Fig. 7, extensometer hysteresis was found to be a major concern. Data generated with a second capacitance contact extensometer^f is included in this figure for purposes of comparison. The hysteresis problem is believed to be a result of relative movement of the pivot points associated with the support design of the extensometer rods. Because of these results, the ATS extensometers were rejected and a new order placed for eight capacitance contact extensometers manufactured by the alternate vendor. Verification tests conducted on these new extensometers showed that their performance was within the required specifications.

As-Received Materials and Inspection

The matrix of test specimens for each of the three ceramic materials being examined in this program is given in Table 1. The fabrication of the NT164 is through pressure slip casting for the green state and hot isostatic pressing (HIPing) for the final densification. The microstructure typically consists of β silicon nitride grains having a grain width of less than one micron and an aspect ratio range of 4 to 8. Depending upon processing, the α silicon nitride content can range from 5 to 40%. The SA SiC is a sintered silicon carbide which is fabricated in the green-state by

injection molding. Additions of small amounts of carbon and boron are used to assist the densification during the sintering step. Post-sinter HIPing is also used to improve density and reduce defect size. The microstructure of this material typically consists of a silicon carbide grains ranging size from 2 to 8 μm . Finally, the SN88 silicon nitride is injection molded followed by gas pressure sintering.

All tensile specimens were dimensionally inspected using a commercial coordinate measuring machine (CMM).^g The inspection protocol (Fig. 8) included measurement of the shank diameters, gage diameters, button-to-shank radii, and the coaxiality between the shank and gage regions. The typical measuring time per specimen was 25 minutes which is considerably less than that required for manual inspection with an optical comparator. A comparison of the coaxiality measurements is given in Figs. 9 through 11 for the NT164, SN88, and SA SiC, respectively. The maximum allowable coaxiality of 5.0 μm (designated by the bold horizontal line) was based on the requirement that the contribution to bending arising from the specimen itself be negligible (bending < 1%). Clearly, most of the SN88 specimens exceeded the specified value while the majority of the SA SiC and NT164 specimens were below the 5.0 μm limit. It should be noted that a coaxiality of 30 μm will introduce 3 to 4% of bending from the specimen alone. In general, most of the SA SiC and NT164 specimens satisfied the required dimensional tolerances while fairly significant deviations from the specified tolerances were measured for the SN88 specimens.

Stress Relaxation Testing

Studies to examine the effect of stress

^f Manufactured by Instron Corporation, Canton, MA.

^g LEITZ Coordinate Measuring Machine, distributed by Brown & Sharpe, North Manufacturing Company, Kingstown, RI. 02852.

relaxation upon creep behavior were also initiated this reporting period. Motivation behind the stress relaxation testing is twofold: (1) to investigate the validity of using this technique for accelerating the generation of creep rate-stress data, and (2) to generate data that may be readily used to predict stress relaxation in industrial gas turbine components that are subjected to constant displacement or strain during service. Stress relaxation and its affect upon long-term mechanical survivability is an important consideration in components which are constrained in movement. In the actual experiment, the strain applied to a button-head tensile specimen is increased to a specified value under load control. The test machine is then transferred to strain control to maintain a constant strain, and the relaxation of stress was monitored as a function of time. A stable extensometer signal is crucial in order that a valid stress relaxation test results. The testing apparatus necessary to perform stress relaxation requires that an extensometer signal comprises a portion of electronic, closed-loop control of the testing machine. Due to the high stiffness of these structural ceramics, a stable signal from the high temperature extensometer is imperative; otherwise wide stress fluctuations will result during the relaxation process. A Lexan box surrounds the extensometer and shields its circuitry. A stable flow of air is maintained within the Lexan box during the stress relaxation test and this further stabilizes the temperature of the circuitry. These provisions result in typical strain fluctuations of $\pm 0.1 \mu\text{m}$. As an example, this fluctuation will manifest itself into a stress fluctuation of $\pm \approx 1.5 \text{ MPa}$ for a material with a Young's Modulus of 300 GPa and a button-head diameter of 6.35 mm.

For these test conditions, the total strain (ϵ_{tot}) may be represented as the sum of the elastic (ϵ_{el}) and plastic (creep) strain (ϵ_{pl}), or

$$\epsilon_{\text{tot}} = \epsilon_{\text{el}} + \epsilon_{\text{pl}} \quad (1)$$

Taking the time derivative and recognizing that for

fixed strain, $d\epsilon_{\text{tot}}/dt = 0$, one has

$$d\epsilon_{\text{el}}/dt = - d\epsilon_{\text{pl}}/dt \quad (2)$$

Because the elastic strain rate is equal to $(d\sigma/dt)/E$, where $d\sigma/dt$ is the instantaneous stressing rate and E is Young's Modulus, the plastic strain rate is given by

$$d\epsilon_{\text{pl}}/dt = - (d\sigma/dt)/E \quad (3)$$

Using these expressions, one may determine the creep rate as a function of stress over many orders of magnitude. The advantage of a stress relaxation test is that several decades of creep rates are measured in several hours of experimentation with one specimen; whereas equivalent generated data using static creep tests require numerous specimens (one stress-strain rate data point per specimen) tested over a range of applied stresses.

Figure 12 shows a typical stress relaxation curve generated at 1370°C with NCX-5102^h silicon nitride. Figure 13 compares the $d\epsilon_{\text{pl}}/dt$ versus stress curve determined from the stress relaxation data with corresponding curves obtained from static stress tests at 125 and 140 MPa. The creep rates from the static stress tests were determined at specimen failure (minimum strain rate), at test commencement (maximum strain rate), and when a total of 2810 μe had accumulated. The motivation for including the static creep rates measured at an accumulated strain of 2810 μe is that the data from the relaxation test were also generated when the specimen had accumulated 2810 μe . As shown in Fig. 13, the data generated from the stress relaxation test is situated in the range between the minimum and maximum strain rates. Further exploratory studies of this technique and its applicability are under investigation.

^h NCX-5102 silicon nitride ceramics are manufactured by St. Gobain-Norton Industrial Ceramics Corp., Northboro, MA

Stress relaxation testing of two SN88 (NGK) silicon nitride and two SN253 (Kyocera) specimens was also completed during this reporting period. For each material, one specimen was loaded to an initial stress of 276 MPa (40 ksi) and the other was loaded to 414 MPa (60 ksi); both were tested at 1300°C (2372°F). As a supplement, the stress relaxation behavior of NCX-5102 silicon nitride specimens were also examined at the same test conditions. For both stresses, the rate of stress relaxation increased in the order (NCX-5102 < SN253 < SN88). Although all three silicon nitrides exhibited viscoelastic behavior, it is speculated that the more extensive relaxation of the SN88 may be due to a greater volume fraction and/or less refractoriness of the grain boundary phase. In terms of the creep rate versus stress curves generated from the stress relaxation data, the creep exponents increased in the following order: NCX-5102 > SN253 > SN88. Furthermore, for a fixed stress, the creep rate was largest for the SN88 and smallest for the NCX-5102.

Creep/Stress Rupture Testing

Creep/stress rupture testing of the SN88 tensile specimens was initiated at 1350, 1150, and 1038°C during the present reporting period. The choice of applied stresses was based upon the fast fracture strengths measured at each of three temperatures. Preliminary results from the first test at 1350°C are shown in Fig. 14. Due to unexpectedly rapid creep deformation, the extensometer range was exceeded prior to specimen failure. This extensive creep is consistent with the measured stress relaxation behavior described above. Microstructural analysis of this specimen is underway.

FUTURE WORK

Future efforts will focus upon the completion of the SR/creep measurements of the three materials listed in Table 1. Microstructural analysis and x-ray diffraction will be used to study changes in the microstructure and intergranular phase composition arising from the long-term testing. The effect of water vapor upon the time-dependent mechanical behavior will also be investigated. Finally, methods for combining interrupted tensile SR tests and tensile dynamic fatigue tests, in which tensile strength is measured as a function of stressing rate, will be studied in an effort to enhance life prediction estimates.

REFERENCES

- (1) M. N. Menon, H. T. Fang, D. C. Wu, M. G. Jenkins, and M. K. Ferber, "Creep and Stress Rupture Behavior of an Advanced Silicon Nitride," submitted to the Journal of the American Ceramic Society for publication.
- (2) A. Rudnick, C. W. Marschall, W. H. Duckworth, B. R. Emrich, *The Evaluation and Interpretation of Mechanical Properties of Brittle Materials*, AFML-TR-67-316, Air Force Materials Laboratory, Wright Patterson Air Force Base, Ohio, 1968.
- (3) C. G. Larsen, L. E. Johnson, and L. G. Mosiman, "Gripping Techniques and Concerns for Mechanical Testing of Ultra-High Temperature Materials," MTS Systems Corporation Technical Publication (1993).

Table 1. Summary of Tensile and Flexure Specimen Matrix Required for the Program

Material	Supplier	No. of Tensile Specimens	No. of Flexure Specimens
SA SiC	Carborundum Company Niagara Falls, NY 14302-0832	50	205
NT164 Si ₃ N ₄	Saint-Gobain/Norton Industrial Ceramics Northboro, MA 01532-1545	126	205
SN88 Si ₃ N ₄	NGK Insulators, LTD Nagoya, Japan	50	205

Figure 1. Comparison of Strain Versus Time Data Generated for the NT154 at a Stress of 150 MPa

Figure 2. Comparison of Strain-Time Curves Generated for SA SiC at UDRI and ORNL

Figure 3. Major Components of the Creep/Stress Rupture Test Machine

Figure 4. Strain gage Specimen Used to Measure Bending

Figure 5. Summary of Percent Bending Values Obtained in Trial 1 Tests

Figure 6. Virtual Instrument Control Panel used for Data Acquisition and Control

Figure 7. Comparison of Load-Displacement Curves Obtained with ATS and Instron Extensometers

Figure 8. Schematic Representation of Inspection Protocol used for Ceramic Button-Head Tensile Specimens.

Figure 9. Summary of Coaxiality Measurements for the NT164 Silicon Nitride Button-Head Specimens

Figure 10. Summary of Coaxiality Measurements for the SN88 Silicon Nitride Button-Head Specimens.

Figure 11. Summary of Coaxiality Measurements for the SA Silicon Carbide Button-Head Specimens

Figure 12. Typical Stress Relaxation Curve Generated at 1370°C with NCX-5102 Silicon Nitride

Figure 13. Comparison of the $d\epsilon_p/dt$ Versus Stress Curve Determined from the Stress Relaxation Data with Corresponding Curves Obtained from Static Stress Tests at 125 and 140 MPa

Figure 14. Time-Dependence of Tensile Creep Strain Measured for SN88 Silicon Nitride at 1350°C and 250 MPa.

Dry Ultra-Low Emissions Combustion System for Allison's Industrial Engines

CONTRACT INFORMATION

Contract Number	DE-FG02-93CE41026.000
Contractor	Allison Engine Company P.O. Box 420 Indianapolis, IN 46206-0420 Phone: (317) 230-3683 Fax: (317) 230-3691
Contractor Project Manager	Hugh M. Davis
Principal Investigators	Duane Smith Rahul Puri David Stansel Mohan Razdan
DOE Project Manager	Stephen Waslo
Period of Performance	July 1993 to August 1994
Schedule and Milestones	

FY94 Program Schedule

	S	O	N	D	J	F	M	A	M	J	J	A
Phase I: Combustion System Development												
Task 1: Combustor Preliminary Design	-----											
Task 2: Subcomponent Design, Fab, Test				-----								
Task 3: Full Scale Detailed Design, Fab, Test	-----											
Phase II: Overall Engine System Design									-----			
Phase III: Hardware Procurement Assembly											-----	

OBJECTIVES

The goal of this DOE sponsored program is to demonstrate 9 ppm NO_x emissions on a gas turbine engine operating from 70% to 100% load while burning natural gas fuel. The part load (25 - 70% power) emissions reduction goal is to limit NO_x production to 50% of the levels achievable with water injection, while maintaining emissions of CO and UHC within acceptable limits (50 ppm and 10 ppm respectively). After the successful completion of this program, a subsequent 8000 hour field evaluation test will be conducted.

BACKGROUND INFORMATION

Allison is a world leader in the design, engineering, and production of gas turbine engines for commercial, military, industrial, and vehicular applications. Allison has over 45 years of gas turbine engineering experience and has delivered about half of the gas turbines that power electric generators for either primary or standby electric installations in the world and about 20% of the power units for driving pumps and compressors in the 3-10,000 hp range used in oil and gas recovery. One of the most popular Allison turbine designs is the aero-derivative 501-K, with output ranging from 3000 to 8000 hp.

The current production can-annular combustion system for the 501-K engines uses liners operating in the diffusion flame mode in which fuel/air mixing and combustion processes take place simultaneously. Consequently, regions of the primary combustion zone have stoichiometric mixtures with attendant high gas temperatures that enhance stability and combustion efficiency at the expense of NO_x production. The diffusion flame liners produce NO_x emissions far in excess of the current environmental regulations in many parts of the world. Water injection with a water-to-fuel ratio of up to 1.4 is required to lower the emissions of NO_x below 25 ppm. This method of NO_x control is expensive, imposes durability limits, and incurs a fuel consumption

penalty with a significant increase in CO and UHC emissions.

In recent years, several gas turbine manufacturers, including Allison, have started aggressive programs to develop dry lean premixed low NO_x combustors. In these combustors the fuel and air are premixed to avoid stoichiometric fuel air mixtures that yield the highest temperatures. Since NO_x formation is exponentially dependent on temperature, premixing the fuel and air limits the NO_x formation rate. Under Allison's GRI funded 25 ppm NO_x combustor program, a retrofittable in-line combustion system has been developed. The emissions goals of the GRI program have been demonstrated. The new in-line combustion system will be field tested in late 1994.

Under the current 9 ppm NO_x program, a retrofittable off-centerline silo combustor is being developed. This combustor is also referred to as the 'Green Thumb' combustor to reflect its aggressive emissions goals and its orientation with respect to the rest of the engine (see Fig. 1). The off-centerline combustion system has several advantages over the in-line combustion system being developed under the GRI program. Firstly, the size or volume of the combustor is no longer constrained to be within the space between the compressor and the turbine. Additional combustor volume is essential for reducing CO and UHC emissions in premixed combustor systems which operate at lower temperatures. An off-centerline combustor has greater potential for being a 'universal' combustor that can be used for several engine models (either as is or as a scaled version) and is also easier to access. A single combustor is more conducive to fuel staging which is needed to meet the emissions goals over the range of operating conditions. An off-centerline combustion system also has the advantage of being able to incorporate a catalytic stage that may be needed to meet the increasingly stringent emissions goals of the future.

Figure 1. Conceptual schematic of the proposed combustion system.

PROJECT DESCRIPTION

In order to meet the ultra-low, dry emissions requirements of the 501-K 'Green Thumb' combustor, Allison will use a fuel staged, multiple injection, ultra lean combustion concept. By using fuel staging in combination with advanced premixing modules to optimize fuel/air mixing, it will be possible to selectively control combustion stoichiometry over the range of engine operation. This concept provides the potential to minimize the production of CO and UHC at low power operation as well as limit the production of NO_x at high power conditions.

The combustion system is being developed under Phase I of the project. Allison's second generation combustor design codes, including 2-D and 3-D combustor flow models and 1-D and 2-D heat transfer models, have been used to design candidate Lean Pre-Mixed (LPM) modules and combustor liners.

Several LPM configurations were identified computationally by varying the location of fuel injection, swirl angle, and the length of the module. The desirable characteristics that are being sought are optimal recirculation patterns, residence time and fuel/air distribution. These characteristics influence stability, efficiency, flashback propensity and emissions performance.

Various LPM arrangements representing series and parallel staging of modules (to cover the complete power range) within the combustor were also evaluated computationally. The arrangement of the modules inside the combustor has a significant impact on emissions as well as liner wall temperatures. In addition to the above, a feasibility study of various other stoichiometry control techniques was carried out.

LPM modules were first bench tested under atmospheric conditions at the Energy Research Consultants (ERC) facility and then evaluated at

Allison in a high pressure sector rig. ERC is a subcontractor for this project and have advanced laser diagnostic capabilities for flow field measurements. A high pressure sector rig was designed, fabricated and assembled for this project with the flexibility to test both single and multiple modules. Single module high pressure rig tests have been carried out at simulated engine operating conditions. The sector rig tests of the parallel and series staging schemes with the most promising LPM candidate will be carried out shortly. While preliminary layout sketches have been made showing 19 LPM's in a parallel staged arrangement, overall engine system design and detailed drawings will be completed after sector rig testing is carried out to define the best staging approach.

A preliminary control design is also being developed for the 19 LPM's grouped in 6 stages (4, 3, 3, 3, 3, 3 modules per stage). Conceptually, the control design will utilize one Bendix gas valve and six proportional (modulating) valves. The digital control logic will provide fuel demand

signals to the Bendix gas valve as well as the six modulating solenoid valves. This design will permit maximum flexibility in scheduling the fuel/air ratios of the individual stages according to combustor requirements for meeting emission goals and preventing lean blow out. Control enhancements will be included to minimize transients during staging. Control design candidates will be evaluated and optimized using a transient engine control simulation program that includes engine and control performance characteristics as well as a detailed dynamic model of the gas valve fuel manifold and solenoid valves, and the fuel nozzle system.

RESULTS

Design

The most promising candidate for the off-centerline silo combustor incorporates a radial swirler and is designated RSPN. The fuel is introduced through fuel tubes that are located in

Figure 2. Modelling results showing a single central recirculation zone that is created by expanding the flow.

Figure 3. Modelling results illustrating the rapid mixing that occurs in the wake of the fuel tubes of RSPN.

between the vanes of the swirler. The flow is stabilized at the exit of the module by expanding the flow and inducing a recirculation zone. As mentioned earlier, the primary design criteria were a reasonable recirculation zone size and good fuel and air mixing. Figure 2 illustrates a single central recirculation zone that is established at the exit of the module based on our 2-D model. The bottom of the figure corresponds to the centerline of the liner. The swirl angle affects the size of this recirculation zone which in turn influences the residence time and the NO_x production therein. The flow within the swirler was also modelled to study the mixing processes in this module. Figure 3 presents the modelling results for an overall fuel/air (F/A) ratio of 0.03. The location of the vanes and fuel tubes is apparent from the silhouette created by the flow vectors. The aerodynamics with the fuel tubes placed between the vanes lends itself to efficient mixing in the wake of the fuel tubes where the fuel is introduced. This is evident in the presented results as the F/A ratio at the exit of the swirler vanes is close to the overall F/A ratio. The loca-

tion of fuel holes on the tubes was selected such that the shape of the F/A ratio profile minimized flashback propensity.

Figure 4 presents predicted CO contour plots with parallel and series staging of modules at 60% power condition. Since several complexities are involved in setting up and modelling the flow through the liner, the presented predictions are only indicative of trends that can be expected and should not be interpreted in terms of absolute numbers. With parallel staging, the cold air from adjacent modules partially quenches the reacting flow of the adjacent module resulting in enhanced CO emissions. However, parallel staging is more attractive than series staging due to simplicity in manufacturing. Note that the series staged combustor is longer than the parallel staged one. The approach that will be adopted depends on the results of the ongoing sector rig tests.

Figure 5 presents the variations in the equivalence ratio that will be observed with the use of the six stage stoichiometric control strategy.

Parallel Staging: CO ppm

Series Staging: CO ppm

Figure 4. Modelling results comparing CO emissions between Parallel and Series staging of modules.

The large fluctuations in the equivalence ratio may not be acceptable based on the test data obtained on RSPN so far. Additional stoichiometry control strategies that will allow all modules in the combustor stage(s) to operate at partial load with air being diverted to other zones of the combustor are being evaluated. For example, use of two series stages with 11 and 8 modules each, in conjunction with variable guide vanes for added control, can provide a more or less flat primary zone stoichiometric profile with power (Figure 6).

Bench Tests

The modules were tested at atmospheric conditions to determine flow areas, F/A ratios at Lean Blow Out (LBO), fuel concentration profiles at the exit of the modules, emissions at the exit of a quartz liner, and velocity measurements at a few planes inside the quartz liner. In these atmospheric tests, the flow factor was identical to that in the engine at 100% power.

The fuel concentration profile at the exit of the

Figure 5. Variations in the primary zone equivalence ratio as a function of power with a 6 stage stoichiometric control strategy.

module can be controlled, by changing the fuel hole locations on the fuel tubes, to deliver the most desirable combustion characteristics. While the profile in Figure 7a delivered the best NO_x performance under atmospheric tests, the profile in Figure 7b is preferred keeping in mind other issues like flashback resistance. Note that the rig data reported later are with the mixing profile shown in Figure 7b and that the tangential variation in the mixing is small ($\pm 2.5\%$).

The RSPN had the best stability of all modules tested. Modifications of the fuel concentration profile at the module exit only affected flashback propensity without improving the overall emission performance. That is, the modifications impacted the trade-off between the averaged NO_x and CO emissions. Improvements in the NO_x emissions were offset by deterioration in the rate at which CO was oxidized. The uncorrected bench test emissions data with the RSPN module are presented in Figure 8 which illustrates the NO_x - CO trade-off described above.

Figure 6. Minimal variations in the primary zone equivalence ratio with power using a combination of series staging and variable geometry.

Figure 7. Fuel concentration profile can be controlled to optimize desirable combustion characteristics.

A parallel stage, consisting of three RSPN modules (with their centers forming an equilateral triangle), was also tested at atmospheric conditions. The fuel concentration profiles in the three modules was comparable and the distribution of fuel to the various modules was fairly equitable. The LBO performance of each module in this parallel stage was strongly affected by the amount of fuel supplied to the adjacent modules. LBO of a given module also depended on whether one or both of the adjacent modules was lit. This strong interaction between the modules is also apparent in the emissions results which are also presented in Figure 8. The parallel stage had both lower CO and NO_x than the single RSPN when all the

modules were lit and operated at a similar stoichiometry. It is believed that the strong interaction affects the size of the recirculation zones of the individual modules and lowers the NO_x produced therein. The CO is also lower due to reduced liner surface area per unit volume as well as increased residence time due to the bigger volume of the parallel stage liner.

Figure 8. Bench Test data showing the NO_x - CO trade-off for RSPN modules. Parallel staging resulted in better overall emissions.

Figure 9. Rig test data showing NO_x (dry 15% O₂ corrected) as a function of Measured Rig BOT.

Rig Tests

The RSPN module has been tested in the 'Green Thumb' sector rig under simulated engine conditions for the KB-5 and KB-7 cycles. As expected, the module showed higher stability at the elevated pressures and temperatures with a F/A around 0.028 at LBO. Figure 9 presents the corrected NO_x rig data as a function of the measured rig BOT. The data correlates rather well with temperature for all the cycle conditions investigated. The CO and UHC emissions did not exceed 30 ppm and 10 ppm respectively for the NO_x data shown. The data shows that the NO_x emissions are marginally within the 9 ppm goals of this program. This is very encouraging since from the bench testing experience we expect a further reduction in NO_x with staging of multiple modules. The CO and UHC emissions are already low and the reduced liner surface per unit volume of the combustor can only further help these emissions as long as all modules are lit.

FUTURE WORK

Sector rig testing will continue to evaluate the best configurations for the combustion liner. Further refinements to the RSPN module may be carried out as warranted. For example, the swirl angle may be changed to give us more margin on the NO_x emissions with a modified recirculation zone. The overall engine system design will proceed concurrently adapting to the evolving combustor design. The overall engine design includes the design of the turbine transition passage; the engine support; the outer combustion case, dome and heat shield; and, the fuel manifolds. The engine control design will also be adapted to the needs of the combustor to maintain stoichiometric control that will allow the emissions goals to be realized.

High Performance Steam Development

CONTRACT INFORMATION

Contract Number DE-AC02-87CD40812

Contractor Solar Turbines Incorporated
2200 Pacific Highway
P.O. Box 85376
San Diego, CA 92186-5376

Contract Project Manager Thomas E. Duffy

Principal Investigators Phil Schneider
Dave Morrow

DOE Project Manager Steve Waslo

Period of Performance 1991 through 1994 - Phase II and III

Schedule and Milestones

FY95 Program Schedule

Tasks	S	O	N	D	J	F	M	A	M	J	J	A
Turbine Blade and Disc												
• Redesign	XXXX	XXXX										
• Fabrication		XXXX	XXXX	XX								
• Assembly				XXXX	XXXX	XXXX						
HPSS												
• 100 hr Test							XXXX	XXXX	XXXX			
HPSS Site Demonstration												
• Site Selection	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX	XXXX		

OBJECTIVES

- Increase industrial and electric utility efficiency through the development of high temperature steam technology.
- Increase steam temperature to 1500°F at 1500 psig for applications for small combined cycles for industrial cogeneration using supplementary firing in the gas turbine exhaust.
- Design, fabricate and test for 500 hours a full length steam generator test module at 1500°F and 1500 psig.
- Develop production cold drawn final superheater tubes to meet all of the ASME Section I Fired Boiler Code requirements and the economic goals of high performance industrial steam generators.
- Design and fabricate an industrial sized 1500°F heat recovery steam generator matched to a supplementary fired 5 MWe industrial gas turbine.
- Design and fabricate a back pressure steam turbine (BPST) to operate at 1500°F and 1500 psig and produce 4 MWe.
- Develop a high temperature valve to control the system as a combination throttle, bypass, and stop valve.
- Design a 1500°F piping system to minimize thermal stress and costs in a high temperature system.
- Incorporate a low emissions burner.
- Test the High Performance Steam System (HPSS) in a factory development test for 100 hours on natural gas fuel.
- Install HPSS at a permanent site in cogeneration and monitor it for the first 1000 hours.

BACKGROUND INFORMATION

Over 30 years ago U.S. industry introduced the world's highest temperature (1200°F at 5000 psig) and most efficient power plant, the Eddystone coal-burning steam plant. The highest alloy material used in the plant was 316 stainless steel. Problems during the first few years of operation caused a reduction in operating temperature to 1100°F which has generally become the highest temperature used in plants around the world. Leadership in high temperature steam has moved to Japan and Europe over the last 30 years.

Because the majority of the U.S. electricity and much of its industrial heat depends upon steam, the DOE has launched this development to make a step change in technology to 1500°F steam. Gas turbine technology, advancements in high temperature superalloys, manufacturing technology advancements, and a new class of once-through steam generators (OTSG) provided the technology base to start this ambitious program.

Thermodynamic models show that the overall cycle efficiency of a Rankine cycle increases as temperature and pressure increase. The highest possible performance gains can be achieved in a 1500°F steam system when using a topping turbine in a back pressure steam turbine configuration (BPST) for cogeneration. Figure 1 shows the thermodynamics of

Figure 1. Temperature-Entropy of 1500°F Steam Cycle Compared to 900°F Steam in an Industrial Backpressure Steam Turbine Cogeneration System

expanding from 1500°F and 1500 psig compared to typical cycle conditions for small industrial cogeneration (900°F and 900 psig). When expanding steam to 150 psig more than twice the energy is available in the 1500°F case to be converted to electricity (an enthalpy drop of 304 compared to 150/Btu per pound of steam). Because this configuration has the highest thermodynamic advantages with the best economics and many practical development advantages it was selected for the development program. Natural gas was selected as the fuel to minimize costs and emissions although other fuels have been studied in detail (Ref. 1).

Development costs and potential payback to industry of the program also established the small size for the cogeneration system. On natural gas, combustion turbines with process boilers have been the most cost competitive. In large systems combined cycles are standard. In small systems combined cycles are not as competitive and need innovative technologies to bring the performance and economic advantages of combined cycles to

smaller industrial and commercial plants. Developments of the OTSG have greatly simplified the steam generation and systems complexity for small combined cycles. Supplementary firing in combined cycles with BPST arrangements are normally the best economic solution, because they can provide fuel utilization efficiencies of over 90 percent.

Figure 2 schematically shows this simple arrangement for a BPST combined cycle. In this arrangement, the maximum amount of electricity is produced per unit of steam to the process, at an incremental fuel efficiency, to produce electricity of over 90 percent when supplementary fired to 2500°F in the exhaust of the gas turbine. Other combined-cycle systems can be configured by adding a condensing low-pressure turbine to produce much more electricity but the overall fuel utilization efficiency decreases as the heat of vaporization is lost in the condenser.

A high production family of gas turbines, Solar® Centaur®, of 4 to 5 MWe was selected to size the steam generator system. Although some combined cycles based on the Centaur have been installed, the large majority of Centaur cogeneration systems use only a process steam generator, using supplementary firing. This decision is based upon current economic

Figure 2. HPS Combined Cycle With Exhaust Firing and Back Pressure Steam Turbine for Cogeneration

conditions. With traditional technology, the ability to "put-the-steam-to-work" on its way to process heating is only marginal due to low power recovery and consequential costs. High performance steam can be used to essentially double the electrical output of the system and improve its economics.

An exhaust firing temperature of 2500°F was selected on the basis of maximizing performance in cogeneration while maintaining ultra-low emissions capabilities in a lean premixed gas turbine combustion system. At a gas turbine exhaust flow of 38 pounds per second, 56,000 pounds per hour of steam is produced at 1500°F and 1500 psig. In the combined-cycle configuration (Figure 2) the BPST will produce 4 MWe and the gas turbine will produce 5 MWe for a total of 9 MWe combined cycle with 74,000 pounds of desuperheated steam to the process. The use of 1500°F steam in cogeneration in addition to producing more electricity from the topping expansion has two additional benefits. Because the BPST exhaust is at 950°F (to match traditional steam turbines) it can provide superheated process steam for many industrial processes requiring high temperature steam, such as steam reforming. The high exhaust superheat temperature also provides a greater amount of electricity from a low pressure condensing steam turbine if incorporated in the system. An additional 6 MWe from the low pressure turbine produces more electricity efficiently when process steam is not required, for a total of 15 MWe with a 5 MWe gas turbine.

In cogeneration with biomass-produced gas and low-Btu landfill fuels, the HPSS can be configured as a 4 MWe steam plant for cogeneration without the gas turbine or a 10 MWe condensing steam power plant.

In coal plants the HPSS technology has been studied in a separate program (co-funded by EPRI) for repowering old coal steam plants and for new advanced steam power plants. By operating at higher pressures (to 5000 psig) and lowering the temperature to 1300°F, heat rate reductions of 12 percent can be achieved using the same tubes developed by the DOE cogeneration program and many HPSS technology developments. Reference 1 details the materials, manufacturing design, and corrosion issues that have directly supported scale-up to utility systems using HPSS technology for repowering 100 to 300 MWe plants with topping turbines of 25 to 50 MWe. It also discusses the application to topping the steam plant of large combined cycles.

PROJECT DESCRIPTION

Phase II – 500 Hour Proof-of-Concept Test

Based upon results of the Phase I study (Reference 2) a 500-hour proof-of-concept test module was designed, fabricated and successfully tested. Figure 3 shows the module being erected in the test facility. It had four once-through steam generator (OTSG) circuits. It was based upon OTSG patents and designs used in Solar's small combined cycles at traditional steam temperature. Approximately 20 of the low temperature OTSG's are used in combined cycles with plant size from 8 to 70 MWe using multiple units. They are the ultimate in design simplicity and well matched to small combined cycles. The OTSG is also ideally configured for high temperature and pressure steam systems. All tubing in the installed OTSGs is made of alloy 800 superstainless steel. For traditional temperatures, carbon steel fins are used and supply 90 percent of the heat transfer surface. For the HPSS-OTSG the final

Figure 3. Once-Through Steam Generator Module Prior to 500 Hour Proof-of-Concept Test

superheater is made from Inconel® 617. As fin temperatures go above 950°F the material is changed to stainless steel alloy 321. Above 1400°F the fins are made from stainless steel alloy 310. A unique manufacturing process is used to bond the fins to different tube materials. The bonding process is rated for operation to 1800°F.

A natural gas combustion system supplied combustion gases to the module at 2500°F. A total of 517 hours of operation at 1500°F and 1500 psig was completed in the module development tests. Transients were investigated and a total of 38 starts and a shutdowns were accomplished. Relatively good control of metal temperature in each of the final superheater tubes was maintained within $\pm 50^\circ\text{F}$ of 1500°F. Tests also indicated that a higher radiation heat flux to the screen tubes occurred than was estimated analytically.

Post test examination of the final superheater after 500 hours showed an adherent thin chrome-based oxide film, a few microns thick, on both the steam side and air side of the tubes. No evidence of steam oxide scaling or spallation was present. This correlated well with 4000 hour bench loop test results on corrosion of superalloys in steam over 1500°F at 1500 psig. In both tests surface oxide penetration on the steam side was slightly less than on the air side of the tubes.

Phase III – Industrial Prototype HPSS Development

Phase III was initiated when the critical technology 500 hour OTSG module test was successfully completed in 1991. In phase III, the primary tasks were to analyze, design, build, and test a complete HPSS power plant for 100 hours at 1500°F and 1500 psig. Figure 4 shows the complete HPS system in the final test phase with the OTSG at the right and the 4 MWe BPST on a skid using a gear box-driven water brake dynamometer to measure performance in the foreground. Tests have been initiated with the low emissions combustion system, the OTSG, piping and control valve all operated at full rated conditions. The turbine and reduction gear have been operated in the system to about 75 percent of full power. To minimize cost, fresh air firing was used with exhaust gas recirculation providing a similar combustion system input to operating with a gas turbine.

Low Emissions Combustion System

A natural gas, low-emissions combustion system was specially matched to the design criteria of the OTSG. Figure 4 shows a top view of the 110 million Btu per hour combustion system on the top right of the photograph. A 16-foot long reaction zone, six feet wide and sixteen feet high is lined with refractory material. The hot wall configuration prevents cold wall quenching,

and consequent CO formation common in water wall boilers. It allows lean operation with exhaust gas recirculation, at theoretical flame temperatures of 2500° F. High rates of exhaust gas recirculation are used to keep emissions low and steam generator efficiency high when not using the exhaust of a gas turbine. The exhaust gas recirculation can also be adjusted to simulate exhaust conditions into the burner from a gas turbine. Figure 5 shows the burner assembly with stabilizer and fuel injection nozzles for lean premixing. Air is supplied to the windbox from a fan connected directly to the windbox. Figure 6 shows the fan and the location of the exhaust recirculation connection directly downstream of the fan inlet air vortex vane control valve. The

Figure 4. 1500° F HPS in Final Test With OTSG on Right and BPST on Left Driving Reduction Gear to a 6000 HP Water Brake Dynamometer

pressure drop across the valve supplies the percent of the OTSG exhaust into the combustion air supply.

Figure 6. Air Supply Fan With Exhaust Gas Recirculation Connection

Figure 5. Lean Premixed Coen Low Emissions Burner (33 inch diameter, 110 million Btu/hr)

In a combined cycle configuration, the fan would be replaced by the exhaust flow of the gas turbine and no exhaust gas recirculation would be incorporated because the O_2 , CO_2 , and H_2O levels would be about the same.

Steam Generator

The steam generator has a once-through water-steam flow path with horizontal gas flow path. Figure 7 schematically illustrates the flow configuration. A total of 26 circuits are assembled to complete the OTSG heat transfer module. Based upon test results obtained in the 500 hour proof-of-concept-test of Phase II, a number of improvements in the design were incorporated in the industrial prototype OTSG circuits. Simplification of the design was obtained by eliminating an intermediate header and the number of radiation screening bare tubes were reduced. Simplification of seals and supporting structure was also accomplished to improve mechanical design for production. Figure 8 shows one of the 26 circuits used in the prototype unit. Six fins per inch of 0.5 inch high carbon steel are used for over 60 percent of the heat transfer. The last 22 rows use carbon steel fins and show as dark tubes in Figure 8. At the inlet gas side the tubes are bare in the first four

Figure 7. Prototype Once-Through Steam Generator Flow Schematic

rows. The initial superheater uses 321 SS fins and the final superheater has 310 SS fins. Final superheater tubes are made of seamless cold drawn Inconel 617, 1.0 inch outside diameter tubes. The last row is 0.28 inch thick with one row of 0.20 and one row of 0.12 wall thickness Inconel 617 in each of the upstream rows. The remainder of the unit is made of Alloy 800H and Alloy 800 tubing. The majority of the tubing is thin wall 0.083 inch seamless alloy 800 in 26 of the 38 rows in the OTSG. Thin wall alloy 800 is standard in all OTSGs currently installed in combined cycles at traditional temperature and pressure steam conditions.

All of the different wall thickness tubes of Inconel 617 were specially developed for this prototype unit. A high production type process had to be developed because the 500 hour module used bar stock-machined tubes in the final superheater. Final optimization of the tube diameter and wall thickness versus production costs was done by the OTSG designers working with Inco[®] International's tube manufacturing process engineers. Extensive process development produced the tubes to the ASME specification requirements of the Boiler Code. Cold-drawn seamless tubes of this size and high strength material are an important technology developed to cost effectively advance HPSS for commercialization.

Each circuit was hydrostatically tested to the ASME Section I Fired boiler Code, and assembled into the prototype module consisting of 26 circuits (Figure 9). Each circuit inlet and outlet are connected by headers to form a complete steam generator. All outlet tubes are instrumented with two thermocouples. Two circuits, one in the center and one at a wall are instrumented throughout their flow path. The finned area of tubes is 14 feet long and the 26 circuits are about five feet wide. Because each circuit is an individual steam generator, the design can match a 10 MWe gas turbine by

Figure 8. Configuration of a OTSG Circuit Feedwater in at Top Right and Superheated Steam Out at Top Left

doubling the number of circuits. A square frontal area, 14 by 15 feet using 78 circuits, would match approximately a 15 MWe gas turbine and produce a 45 MWe HPSS combined

cycle with condensing turbine. No further development would be necessary for this replication approach to scale up the prototype design.

Figure 9. OTSG Module for Prototype HPSS Showing 25 Circuits for a 56,000 pph System

Piping and Valve to Turbine

Because of the step increase in temperature, all of the high temperature steam components needed innovation, special design, and development. Circuits are connected to a 5.25 inch diameter 1.1 inch thick hot extruded seamless tube header with two 90 degree bends connecting to the control valve. Figure 4 is an overview, and Figure 10 a more detailed view showing the control valve. The header is allowed to slide in the plane of the turbine. The turbine is fixed and the valve is allowed to slide in line with the turbine axis with its weight carried on spring hangers. Inconel 617 rods support the Hastelloy® X valve. By placing the

turbine at the plane of the header and sliding all components on low friction bearings, pipe bending loads are eliminated (except for friction). Total pipe length is less than 10 feet, thus minimizing cost and stress with a simple design approach of allowing all flexing to occur in the long, 1.0 inch final superheater tubes.

The valve has a triple function using a single moving plug that throttles, shuts off, and by-passes steam as required by the electronic controls. Figure 11 is a cross section of the valve showing the desuperheater function built into the low pressure bypass line (vertically down line in Figure 10). A 2000 psi hydraulic system controls the valve with failsafe features to always shut the steam to the turbine. It has a 0.15 second fast trip capability with or without the hydraulic pump operating using accumulators. A mechanical spring is also capable of shutting steam to the turbine in the event of accumulator malfunction. The valve bypass arrangement has the same flow restriction as the turbine nozzle and is designed so that the OTSG always has the same flow area venting to the low pressure exhaust steam system regardless of valve position. Fisher Valve Company has specially developed this valve for the HPSS program and it is designated a special ASME/ANSI B16.34, Section 8 special 9000# Special Class valve.

Steam Turbine

To make a major step advance in steam turbine inlet temperatures, several key design concepts were adapted from industrial gas turbine design criteria. It is critical to costs and technical success that the turbine be as small as possible. Small dimensions reduce pressure stresses, wall thickness and thermal stresses. Complete symmetry is also important to reduce thermal stresses and distortions during transients. The design must also allow the use of gas turbine superalloy materials in the nozzle and

Figure 10. OTSG's Outlet Header Connection to Control Valve With Steam Turbine Connection in Background

RE94196M/S

Figure 11. High Performance Steam Control Valve Cross-Section

superalloy manufacturing process proven for gas turbines.

turbine nozzle superalloy Inconel 939 manufacturing technology.

Figure 12 is a schematic section of the turbine flow path and Figure 13 is computer aided design (CAD) layout of the turbine assembly. They illustrate how the critical design criteria were achieved. A high-speed (30,000 rpm) single-stage impulse turbine with a small diameter overhung rotor, a vertically split case and centered inlet line flow arrangement is an optimum configuration for compactness and symmetry. A turbine efficiency of 75 percent when expanding from 1500°F and 1500 psig to 150 psig back pressure yields a 950°F exhaust temperature. This traditional steam system temperature becomes the cooling for the rotor and outer case. An overhung rotor allows an inner case design that can be fabricated using gas

Steam from the valve enters the turbine through a 5.25 inch diameter line with a 1.1 inch wall thickness made of Inconel 617. An electron beam weld joint connects the hot extruded seamless pipe to a thermal isolation series of three ring forgings welded to form convolutions and to the cast steel outer case made of low alloy steel WC-9.

The outer casing is cooled by exhaust steam to operate at 1000°F. It contains exhaust pressure at 150 psig. A floating sleeve connects the outer case to the inner case and is sealed by ten close fitting seal rings to minimize leakage from the 1500 psig and 1500°F. Steam flows through a total of 24 passages in the investment

Figure 12. Back Pressure Steam Turbine Flow Schematic

Figure 13. Back Pressure Steam Turbine Detail Layout

casting that forms the inner case to direct steam to the nozzle ring. The sleeve inner case and nozzle ring are all made from cast Inconel 939, which is used in gas turbine nozzles at metal operating temperatures in the 1500° F range.

The inner casing and nozzle ring is the most critical component in the turbine. They and the inlet sleeve and inlet-to-case transition are the only components that must operate at 1500° F. Design of the 1500° F inner casing was based on a maximum allowable stress value to not exceed 67 percent of the average stress for rupture in 100,000 hours as guided by Section VIII Division I of the ASME Boiler and Pressure Vessel Code. For hot isostatic pressured castings of Inconel 939 this value is 18 ksi. Because statistical data was lacking, the design allowable stress value was set at 10 ksi, corresponding to 55 percent of the rupture value from the master curve which includes an 85 percent casting allowance. The design and the

casting process were iterated with a total of three castings made before the final design was obtained. Figure 14 shows the casting with test bars cast for laboratory evaluation. Extensive finite element modelling was made of the design to ensure acceptable stress levels for casting process modifications required for production. A total of 24 structural vanes are used in the casting to support the inner and outer walls of the casing. The inner and outer walls and struts are typically 0.38 inch thick. The maximum Von-Mises stress was calculated to be 8,630 psi at the trailing edge of the strut to wall transitions and thus satisfied the allowable stress value with a relatively thin wall design.

Figure 15 is a photograph of the finish-machined inner casing showing the 12 cooling holes that go through struts in the casing. Four radial lugs on the inner casing periphery hold the horizontal and vertical centerline of the inner to outer case. Thermal barrier coatings reduce the

Figure 14. Inner Turbine Case Casting

Figure 15. Inner Casing Assembly With Nozzle Ring Showing Cooling Holes in Cast Struts

amount of cooling steam necessary to keep the outer case at acceptable temperatures. Radiation shielding maintains the temperature of the outer casing less than 1000°F by ducting cooling steam around the wall of the outer case before flowing through the tubes in the inner casing

holes and then across the inlet face of the turbine disc to cool the disc surface with exhaust steam.

Figure 16 is the completed inner casing assembled with the nozzle ring. Fourteen convergent-divergent nozzles have a spouting velocity of Mach 2, or 4000 feet per second. This nozzle is mechanically attached to the casing and the mechanical connections are sealed with a high temperature braze. Extensive manufacturing process development efforts were used in obtaining a nozzle exit angle near 11 degrees to the face of the nozzle in the extremely tough Inconel 939 material.

Figure 16. Supersonic Convergent-Divergent Nozzle Assembly

A bladed disc design of the rotor was used in the initial testing of the turbine (Figure 17). It was made by electrochemically machining a pancake forging made of Inconel 718. Impulse blading is used and is followed by a vaned diffuser. Tandem exit guide vanes raise the static pressure by recovering the high swirl energy in the turbine exhaust. Additional static pressure is recovered by the diffuser downstream of the vanes. The flow then dumps into the exhaust hood collector producing additional static pressure rise. The velocity of the machine exhaust is a modest velocity of 150 feet per

Figure 17. Turbine Rotor Bladed Disc (11.5 Inch Pitch Diameter)

second. Because of the diffuser, the static pressure between the nozzle exit and bucket inlet is significantly less than the exhaust back pressure. This makes possible a cooling steam flow to cool the case and the inlet side of the rotor disc via cooling tubes isolated from the inner case in holes drilled in 12 vanes (Figures 13 and 15). The downstream side of the disc is cooled by exhaust steam leaking to the labyrinth seals on the output shaft.

The bearing housing is made from cast low-alloy steel. It is nested under the hot exhaust collector. Thermal isolation is achieved through the use of a radiation shield with cooling air between the housing and shielding. Air admitted to the bearing housing is also used as a buffer in the honeycomb labyrinth steam/oil seal adjacent to the front bearing.

Figure 18 shows the completed steam turbine assembly installed in the HPSS prior to initial testing. A centerline trunion flexible plate mount to the rails of the skid is shown. This point is the only fixed point in the system with the valve and steam generator header designed to slide in the horizontal plane to accommodate

Figure 18. BPST Installed in HPSS Prior to Initial Tests

thermal expansions. Rigid vertical and horizontal slide guides on the valve support system are designed to react friction forces caused by sliding the system. This configuration eliminates pipe bending loads and minimizes moments around the turbine. A slide key on the bottom of the turbine case allows moments caused by friction to be reacted between the valve guide ways and the turbine case.

HPSS – 100-Hour Development Tests

The HPSS components have been assembled into a development test system (Figure 4). The test is to establish component performance in steady state, part load and off-design conditions. Startup, steady state, transient, and shutdown control methods will be evaluated and developed. Rotor bearing system dynamics and disc and blade vibration characteristics and performance will be evaluated at rated and during startup conditions. A total of 100 hours of maximum power operation at 1500°F and 1500 psig is planned to finish Phase III.

Phase IV – HPSS Cogeneration Site Installation and Test

A long term commercial installation of the developed HPSS is planned for a cogeneration (or power generation site) after completion of the 100 hour development test. It is planned that the DOE project will monitor the first 1000 hours of operation at 1500°F and 1500 psig. Site visits and evaluation by DOE and industry representatives will be part of this demonstration. The same arrangement of components shown in Figure 4 will be used at the site to eliminate then need for piping system redesign. The turbine skid will be redesigned to accomodate an electric generator package, replacing the water brake dynamometer and its gear box with an electric generator matched to the site voltage, and a new reduction gear.

A site is being sought and interested companies should contact the author or DOE.

RESULTS

Phase I Study

Results of the program to date include the completion and publishing of the Phase I study that defined optimum configurations of the HPSS for 4 to 50 MWe industrial cogeneration (Ref. 2). It identified design criteria for cogeneration that would meet the maximum number of industrial sites in the U.S. This study identified combined cycles with a BPST as the highest performance configuration in industrial cogeneration.

Phase II Proof-of-Concept 500 Hour Test of OTSG Module

In Phase II a proof-of-concept 500 hour test was designed and tested to develop the basic technology needed to generate 1500°F and 1500 psig steam in a steam generator matched to supplementary fired gas turbine combined cycles. A total of 517 hours of operation at rated conditions were successfully completed on the module (Figure 3). A total of 38 starts and stops were monitored during the test. Several important design and manufacturing improvements to reduce cost and improve temperature distribution were discovered in the course of testing the highly instrumented test module. These features were incorporated in the design and manufacture of the OTSG prototype built in the current Phase III program.

Phase III

Phase III results are listed below:

- Production Tubes Developed for Final Superheater. Production tube manufacturing processes were developed for the final

superheater tubes. The 500 hour test module used 15 foot long tubes that were manufactured from bar stock with a machining process which was not suitable for production units or ASME Code requirements.

- **ASME Design Approval for 1500°F Steam Generator.** The prototype design based upon the 500 hour test module was approved by the Authorized Inspector for operation at 1500°F and 1500 psig in industrial or commercial sites to the requirements of the ASME Section I Fired Boiler Code.
- **Prototype Steam Generator Tested to 1500°F and 1500 psig.** An OTSG production prototype producing 56,000 pounds per hour steam from natural gas was fabricated and tested (Figure 8). It has accumulated about 50 hours with about 5 hours at 1500°F and 1500 psig. Its performance was as calculated. With a firing temperature of 2500°F and an exhaust of 170°F, it had an efficiency of 96 percent (LHV). Development work on balancing flow within the OTSG resulted in excellent superheater temperature distribution of about $\pm 25^\circ\text{F}$, about an average outlet temperature of 1523°F. Figure 19 shows the actual final calibration and temperature distribution on each of the 26 circuits. Note good correlation between all circuits that had both thermocouples functional. Circuit number 5 thermocouples were not correlating well but the temperature was adjusted to keep the high reading thermocouple within acceptable limits of the design.
- **HPSS Controls.** Excellent outlet temperature and pressure control was maintained by the automatic control system. At 1500°F and 1500 psig, temperature control was within a $\pm 10^\circ\text{F}$ band and pressure did not

Figure 19. Twenty-Six Superheater Outlet Circuit Temperatures

vary more than ± 15 psi. Excellent transient control of the system pressure was obtained with less than a 50 psi pressure transient observed during steam turbine trips at near full steam generator operating conditions. When the entire system is shut down from full firing conditions, the steam pressure drops very rapidly (Figure 20) when fuel, water, and air flows are stopped (as in a loss of electrical power turbine trip). Figure 20 shows a turbine trip from 1430 psig and 1425°F with all systems shut down. Steam temperature shows only a 200°F temperature drop due to swell associated with the rapid pressure drop as the full steam flow is bypassed through the control valve. It maintains high superheat for many hours as it slowly cools and most of the water is boiled out of the OTSG.

Automatic starts and warm-up of system was performed over 10 times. The unit was successfully dry started each time with very low pressure steam exiting the superheater and flowing through the bypass port of the valve to warm these components and dry out the piping. As soon as superheated steam

Figure 20. Maximum Steam System Transient - Pressure and Temperature Transients During Electric Power Trip Test

was sensed, the turbine valve was opened and the turbine warm-up started.

- **Control Valve.** The steam control valve had problems initially with tolerances, causing binding during warm up procedures. A design modification to the internal parts allowed operation with differential temperature of up to 1200°F. A new static seal design was incorporated to control internal leakage with large internal clearances. In addition, electric heating pads were used to preheat the valve to about 800°F prior to system startup. The flow control performance of the valve was excellent. It was tested to 1500°F and 1500 psig and cycled a number of times including during turbine trips. It could maintain accurate closed loop speed control of the turbine from warm-up speed of a few thousand rpm to the full design speed of 30,000 rpm.
- **Piping System.** The design for controlling thermal expansion of the piping, valve and

turbine by allowing the OTSG final superheater tubes to flex was successfully demonstrated.

- **Low Emission Combustion System Tests.** Low emissions from the combustion system on the OTSG were certified by three one-hour steady state tests while operating at 50 percent and 100 percent power. NOx levels were averaged to 27 ppm and CO was 21 ppm, both referenced to three percent oxygen (used for steam generators). At the gas turbine reference level of 15 percent oxygen these are equivalent to nine ppm NOx and seven ppm CO.
- **Steam Turbine Tests.** Steam turbine was operated to full speed of 30,000 rpm and about 75 percent power (4500 hp) with a temperature of 1430°F and 1425 psig inlet conditions. The cooling scheme for the outer case is working well. It maintains the outer case at near exhaust steam temperatures. Cooling steam flow for the rotor nozzle face is adequate to about 50 percent power but needs greater mass flow at higher power. A total of 16 hours was accumulated on the turbine, when during a cooling flow evaluation test at 25,000 rpm the turbine tripped due to high shaft displacement (greater than 4 mils) at the inboard bearing. Subsequent teardown evaluation indicated that two blades had been thrown from the bladed disc. Cracks were also observed on the leading and trailing edge of many blades at the junction with the rim of the disc. The first blade bending resonance is at 23,000 Hz and corresponded to the fourth harmonic of the blade-to-nozzle passing frequencies. Analysis has shown that additional damping is required in the system and is being added to a modified rotor design. Frequencies of the blade resonance are also being changed.

FUTURE WORK

A redesigned rotor with damping using inserted blades with shrouds and a damping wire incorporated in the shroud is scheduled for test in the first quarter of 1995. New blades are scheduled to be received from the supplier in December, 1994. Completion of the 100 hour development tests are scheduled for the second quarter of 1995. This will be followed by a host site demonstration.

Two candidate sites for the installation of the HPSS have prepared preliminary engineering to determine the installation costs of the HPSS. Neither site is firm yet, and other site candidates for a 4 MWe, 75,000 pound per hour of process steam cogeneration system are being sought.

REFERENCES

1. "High-Temperature Steam Topping Cycle Study", Final Report on Project 1403-44, EPRI TR-102059 (1993).
2. "Advanced High Performance Steam Systems for Industrial Cogeneration," Final Report on DOE Contract No. AC02-85CE40746, DOE/CE/40746-T1 (1987).

**Turbine Airfoil Manufacturing Technology
Land Based Turbine Casting Initiative**

CONTRACT INFORMATION

Contract Number

Contractor

Howmet Corporation
1500 South Warner Street
Whitehall, MI 49461-1895
Ph: (616) 894-7250
FAX: (616) 894-7826

Contractor Project Manager

Philip Monaghan

Principal Investigators

Boyd Mueller

DOE Project Manager

Michael Karnitz

Period of Performance

Schedule and Milestones

SECTION 6.4 Program Schedule

OBJECTIVE

The objective of this program is to develop the manufacturing technology of single crystal turbine airfoil castings to a scale that can be economically utilized in industrial and utility gas turbines that will meet the performance goals of the ATS program, and demonstrate high quality, cost-effective quantity production methods.

BACKGROUND INFORMATION

Within aircraft gas turbine engines, single crystal, directionally solidified and equiaxed castings are used (Figure 1). Mechanical and physical properties of components made out of equiaxed polycrystalline nickel base superalloys can be assumed to be isotropic when the best casting practice is used. Directionally solidified (DS) components with columnar grain structure can be treated as transversely isotropic, with no variation of properties in the plane normal to the growth direction. With directional solidification, significantly superior creep resistance and low modulus, parallel to the columnar grain, are exploited. A single crystal (SC) component; however, is truly an anisotropic body with significant variation in properties along different directions of crystal symmetry. To make optimum use of a combination of superior properties in certain directions and remain within the bounds of acceptable variation, control of the crystallographic orientation with respect to the part geometry is required as is the minimization of grain defects.

Figure 1. Single Crystal, Directionally Solidified, and Equiaxed Castings

Casting yields on fully developed aircraft gas turbine single crystal casting processes range up to 95 percent, and are higher for directionally solidified and equiaxed components. The relatively high yields of these processes are the result of an understanding of the relationships between alloy, geometry and process conditions. This understanding was developed iteratively over the past 30 years. While a thirty year iterative cycle may seem excessive; it must be considered that production of investment castings requires expertise in the areas of alloy formulation, waxes, ceramic cores, ceramic molds, melting, casting and material handling.

The alloys used in the aircraft gas turbine industry are produced primarily by vacuum induction melting techniques. To produce high purity superalloys within a very narrow composition specification range, careful control over the incoming feedstock materials, the vacuum refining, ingot casting operations, and ingot handling procedures is required. Appropriate sized superalloy ingot charges are remelted for the vacuum investment casting operation.

The waxes used to produce investment castings are the first step in a series of replications. The waxes must accurately replicate the desired cast product shape and be dimensionally stable during mold set-up and through the Mono-Shell[®] process. The waxes must also be removed from the mold without leaving a residue in the mold or producing undesirable reactions with the mold facecoat.

The ceramic core forms the internal cavity in cooled components. The core in DS and SC processes is exposed to liquid nickel-based superalloys for times in excess of one hour for aircraft gas turbines. During this exposure to liquid metal, the core cannot react with the liquid or be eroded by it. In addition, the core must remain dimensionally stable to accurately reproduce the desired internal features. After the casting has solidified, the core must be removed from the casting without causing metallurgical or dimensional damage to the casting.

Similarly, the Mono-Shell[®] mold must adhere to and replicate the wax pattern and fix the core position without restraining the core excessively such that it fractures due to mismatched thermal expansions. The shell must retain form and core position during the entire casting cycle. This must occur without unduly restricting the casting as it cools, causing hot tears or recrystallized grains. The shell also is in contact with the molten metal for extended periods of time, and as such must not react with or be eroded by the metal. After the casting has solidified and cooled, the shell must be easily removed from the casting without causing metallurgical or dimensional damage to the casting. For the aircraft gas turbine industry, Howmet has developed waxes, cores, and shell systems which are able to meet the requirements discussed above.

For equiaxed castings, microstructure control is attained by mold and metal temperature control, and by controlling heat flow through selective mold insulation with ceramic fiber materials. Heat flow is controlled in equiaxed castings not only to avoid porosity formation, but also to ensure a fine, uniform grain size without producing columnar grained regions.

The key difference between equiaxed casting and directional solidification is in the design of the mold and the equipment features which control temperature. These changes allow the heat to be removed from the casting in a unidirectional manner and a temperature gradient to be maintained ahead of the solidifying interface to prevent extraneous nucleation.

In directional solidification, a mold which sits on a water cooled copper chill plate is surrounded by an induction heated susceptor furnace. After the mold has achieved a preset temperature, it is filled with molten alloy. The molten alloy strikes the copper chill plate through starter holes in the bottom of the mold. The susceptor furnace maintains the alloy filled mold at a temperature greater than the alloy melt point while the alloy in contact with the copper chill has solidified. To initiate the directional solidification casting process, the mold and chill are simultaneously withdrawn from the susceptor furnace at a predetermined rate past a radiation baffle placed at the base of the susceptor. The baffle serves to create a thermal gradient between the hot susceptor and cold furnace walls. This thermal gradient is developed as shown in Figure 2.

The alloy that initially solidified against the copper chill has formed a thin layer of randomly oriented grains (equiaxed). As mold withdrawal

proceeds and solidification continues; competitive crystal growth occurs and those grains most closely aligned with the $\langle 001 \rangle$ direction parallel to the thermal gradient outgrow off $\langle 001 \rangle$ grains. This produces a columnar array of grains with a common vertical $\langle 001 \rangle$ orientation. This process produces a directionally solidified polycrystal airfoil casting as shown in the middle of Figure 1.

A single crystal casting is obtained by inserting a helix located between the starter and the airfoil cavity to select only a single grain to grow into the airfoil (Figure 2b). Starters and helixes allow for primary orientation control only. To control orientation in the longitudinal direction, the secondary $\langle 001 \rangle$ orientation of the crystal is controlled. A "seed" with the desired primary and secondary orientation is used in lieu of a helix selector as shown in Figure 2c. After metal is poured into the mold, the single crystal grain develops by epitaxial growth from the seed with the desired orientation of the "seed". This degree of control over the crystallographic orientations gives the designer further options for maximizing part performance.

In addition to developing unidirectional gradients, additional heat flow control is obtained by varying casting parameters such as mold temperature, withdrawal rate, mold design, and mold properties. An understanding of the effect of, and interactions between, furnace design, mold design, casting parameters and part geometry is necessary to produce high quality investment castings with high yield for aircraft gas turbine engines.

While the aircraft gas turbine industry has made remarkable improvements in yield, quality and cost effectiveness, active research continues in this field. Areas of active research include:

wax research to improve surface finish and dimensional stability, improvements in the consistency and cost effectiveness of core and shell materials and forming processes, feedback control systems for the vacuum casting process, and cellular manufacturing practices to improve work flow and quality while reducing cost. An additional area of active research is to significantly shorten process development cycle times through the application of process modeling and rapid prototyping technologies. These efforts are ongoing and will quickly be integrated into the normal process development/improvement cycles.

Howmet has already applied its turbine airfoil investment casting technology to the production of large equiaxed and directionally solidified land based components. Therefore, the thrust of this proposed program is to work jointly with the OEMs and develop and apply the necessary single crystal casting technology to meet the ATS program goals.

PROJECT DESCRIPTION

Specific developments required to scale aircraft single crystal castings technology up to land base turbine size components have been identified in four technology thrust areas. These four technology thrust areas, while pursuing different disciplines and discrete innovations, constitute a coherent system which encompasses the entire process to reliably produce high quality, cost effective investment casting for land based gas turbines. The four thrust areas are shown schematically in Figure 3 and highlighted below.

Figure 2a Directional solidification (D.S.) casting process, with remelting and pouring performed under vacuum.

Figure 2b, c Various methods of controlling crystal orientation. Addition of a single crystal selector - or "seed" - in a D.S. set-up promotes the production of a single crystal component.

T2-94/1144: 1

**Figure 3. Organization of the Land Based Turbine Casting Initiative
by Technology Thrust Areas**

Low Sulfur Alloy Processing

The environmental resistance of advanced, single crystal, nickel-based superalloys and coatings is dependent upon formation and adherence of an aluminum oxide scale. It is now generally accepted that segregation of indigenous sulfur to the oxide/metal interface seriously degrades scale adherence and that control (i.e., reduction) of the sulfur activity is crucial in order to optimize the environmental resistance. Recent work also suggests that interfacial sulfur segregation degrades the adherence of chromia scales. For land based turbines, which are expected to operate for tens of thousands of hours, coating adherence through the control of the sulfur content is critical.

Two alternative approaches exist to achieve necessary reductions in the sulfur activity of single crystal alloys: 1) addition of rare earth elements, such as yttrium; and 2) development of effective liquid and/or solid-state desulfurization processes. Although the oxidation resistance of yttrium-doped alloys has been demonstrated, interest in desulfurization was motivated by processing difficulties in reliably adding and maintaining specified levels of yttrium homogeneously to single crystal turbine airfoil castings. For larger directionally solidified or single crystal land based components, which can be expected to have longer withdrawal cycles than aircraft turbine components, yttrium retention may become even more difficult.

Howmet has developed an approach to reduce the sulfur content of superalloys during master heat formulation. The liquid-state process developed by Howmet has been used to prepare alloy master heats containing only 1 wppm sulfur. In addition, casting of low-sulfur,

SC aircraft engine hardware with sulfur contents of approximately 1 wppm was demonstrated in recent work. Current experience, however, indicates that lower sulfur contents must be attained to optimize environmental resistance.

A further reduction in the sulfur content has been demonstrated through application of a post-cast desulfurization heat treatment. In this case, sulfur contents less than 1.0 wppm have been achieved. The effectiveness of desulfurization processing for imparting superior environmental resistance has been demonstrated during cyclic oxidation and burner rigs testing. In contrast to untreated PWA 1484, which exhibits significant weight loss, the desulfurized alloy exhibits a slight weight gain associated with formation of a thin, adherent oxide scale. A marked improvement in the hot corrosion resistance of PWA 1484 has also been achieved through application of a solid-state desulfurization treatment. In this case, during cyclic testing of samples coated with Na_2SO_4 , the desulfurized alloy exhibits an increased incubation period prior to the onset of hot corrosive attack.

Casting Process Development/Understanding

Depending on the component configuration, alloy, mold design and casting practice; various kinds of grain defects may appear in single crystal castings. Depending on their appearance and location, many such defects have been classified, including: low angle boundaries, slivers, zebras, bi-grains, freckles etc. These defects are shown schematically in Figure 4. To achieve significant defect reductions requires a fundamental understanding of the conditions under which these defects form. This knowledge is necessary to achieve significant improvement, since what is often required to

eliminate one type of defect may actually cause another defect elsewhere in the component. Only through a fundamental understanding of the relationship between casting process conditions and defect formation criteria, can significant and lasting process improvement occur.

The casting of land based gas turbine engine components may differ significantly from aircraft turbine components owing to their increased size. For example, the interdendritic liquid typically has a density different than that of the bulk liquid. These density differences can drive a thermosolutal convection process and produce solute plumes in the liquid ahead of the solid/liquid interface. If these plumes become severe, they cause freckle defects and macrosegregation. The conditions favoring formation of these plumes increase as the casting size increases. In addition, with increases in casting size, the orientation yield curve may shift towards higher primary angles. The extent of this shift remains to be defined.

Besides the increased variation in primary orientation from component to component, the larger casting size also increases the propensity for fanning out of orientation. The fanning out of orientation, when it occurs without the presence of any defect, is called splaying; and is generally most severe at the far end of large castings. Characterization of splaying has been ignored in aircraft engine components owing to smaller dimensions. However, for large components, it is important to characterize the impact of splaying on properties and assess any debit in performance attributable to splaying. Additionally, the splaying phenomena is also expected to increase the propensity for forming boundary type defects such as low angle boundaries or slivers.

T2-42/559 6

Figure 4. Grain Defects in Single Crystal Turbine Blades

The increased size of land based turbine components also puts additional constraints on the ceramic mold and core systems. For example, in aircraft engine size components, shell creep is not significant due to the small sizes and relatively short withdrawal cycles. However, in large land based applications, shell creep can be severe and significantly decrease dimensional capabilities. Because of the constraints imposed by controlling thermal gradients to produce single crystals, it may not be feasible to simply increase the shell thickness to reduce shell creep. Similar arguments apply to the core used in these large castings. A detailed understanding of the thermomechanical behavior of both shell and core materials is critical to make intelligent choices and fulfill the dual, often conflicting requirements, of

attaining good metallurgical quality and good dimensional stability.

Post Cast Process Development/ Improvement

Handling large single crystal components presents unique problems. For example, recrystallized grain defects can occur if these components undergo mechanical damage, as would result from having two parts strike each other during mold, core, or gating removal. For aircraft engine components, which typically have mold weights of tens of pounds and part weights of one to three pounds, it is easier to avoid handling damage. However, when the mold weight is in excess of one hundred pounds and individual component weight is increased to thirty pounds, handling damage becomes a serious issue. Therefore, special handling procedures need to be in place to prevent damage during these operations.

Nickel-base superalloys derive their performance from precipitation hardening. Both the low temperature strength and high temperature creep resistance are enhanced because of a homogeneous dispersion of fine gamma precipitates in an FCC gamma matrix. In advanced single crystal superalloys, there is 60 to 70 volume percent of gamma in the FCC gamma matrix. The FCC gamma matrix is essentially a nickel based solid solution. In cast single crystal superalloys, the uniform dispersion of gamma is achieved by a post cast heat treatment. Typically, a high temperature solution heat treatment is followed by a cooling heat treatment cycle and then by a lower temperature age to produce a desired gamma dispersion.

As straight forward as this is in principle, the attainment of optimum properties in practice is difficult. The as cast dendritic microstructure is far from being homogeneous, and consequently, different parts of the microstructure go into solution at different temperatures. If segregation of the elements is severe, local areas incipiently melt before complete solutioning is achieved. Also, the cooling rate from the solution heat treatment temperature significantly influences the gamma size which ultimately forms. In general, fast cooling rates are desirable to achieve finer gamma size.

In transitioning single crystal technology to land based turbine components, casting segregation may be increased and the ability to achieve fast cooling rates may be decreased. Increases in casting size increases the propensity for segregation, and variations in cooling rate across the large cross section of a solutioned part may produce a nonuniform gamma size dispersion as well as coarser than desired particle sizes. While the implication of non-optimized microstructure for mechanical performance in aircraft engines is well understood, the effect on long-term creep resistance, critical to the steady state operation of land based gas turbines, is not well defined. Besides the likelihood of gamma size being less than optimized, the increased propensity to form porosity and other phases such as carbides is raised as casting size increases. These microstructural changes are likely to impact fatigue performance. Generally, porosity levels can be significantly reduced by hot isostatic pressing (HIP), but the feasibility and viability of applying this process to large components is not well established.

The large size of land based turbine components also affects the ability to

nondestructively inspect them. The thick sections and long X-ray path lengths typical of these components severely attenuates the signal and reduces the signal-to-noise ratio. This may make it difficult or impossible with current systems to consistently assure a minimum quality standard. To reduce the development cycle time for a new design, it is necessary to monitor and control the critical dimensions of the casting and associated cores and molds. In addition, detailed knowledge of the casting geometry is necessary to plan the drilling of cooling holes during airfoil manufacture. To address these requirements, a new nondestructive inspection system needs to be developed and implemented for land based components. Coupled with this is the need for the data sets generated by these inspection modalities to be in a coherent format that is compatible with the thrust in cellular manufacturing and implementation of electronic data transmission from contractor to supplier to end user. There has to be a direct, easy flow of information relative to the casting of a component and subsequent operations it may be subjected to.

A goal of this program thrust area is to provide a portion of that electronic bridge through the use of a unique digital radiography technique that is transparent to the subsequent receiver of a component; but capable of providing information critical to its processing. Embedded in the digital radiographic technique is the capability to generate computed tomography (CT) scans of component sizes which are currently considered non-inspectable by present state-of-the-art instruments. A direct benefit of having this capability is the potential to significantly reduce product development cycles and increase the robustness of the casting process. There will also be a positive fallout in the areas of dimensional control and defect

detection. These data will also provide information critical to controlling internal dimensions of more advanced internal cooling schemes currently considered inaccessible by standard NDI techniques.

To achieve this goal, there will have to be advances in the areas of X-ray sources, X-ray detectors, and algorithms as identified below:

- Brighter sources will provide better statistical information which is critical to extract defect information from digital radiographs and for CT reconstruction algorithms.
- Area X-ray detectors of sufficient size and sensitivity to allow for efficient data collection and component throughput.
- Algorithms which will allow slight misorientation in component position; thus reducing fixturing/registration problems.
- Data generated in the above inspection routines will be compatible with other media data so that an electronic record can be maintained and supplied to the customer.

Casting Defect Tolerance Levels

The casting yield due to the presence of metallurgical defects is likely to decrease with increasing casting size for the land based turbine. To maintain a high casting yield will require coupling intelligent choices about the effect of defects on performance with improvements in the casting process. Ultimately, all design standards depend on the quantification of the effect of casting imperfections (i.e., defects) on performance. To achieve a higher casting yield, the effect of the

number, size, and location of defects on performance must be understood. Test data will have to be generated to realistically define the defect tolerance in land based engines. Based upon discussions between OEMs and foundry personnel; freckle, orientation, low angle boundary and recrystallized grain defects appear to be the most probable defects and, therefore, the potentially most damaging. The deleterious effect of these defects on mechanical performance will be evaluated so that data based specifications and tolerance limits can be established for the operation conditions of land based turbines.

The team that will perform this program includes companies and experienced individuals in all facets of the design, manufacture, and use of investment castings for aircraft, and land based gas turbines. Howmet will involve three of its casting plants, an alloy plant, and the Howmet Research Center. The industrial and utility gas turbine manufacturers that will be involved are: ABB, GE Power Systems, P&W, Solar, and Westinghouse. This team addresses the entire supply chain to produce land based gas turbine castings including: alloy production, investment casting and end users.

Session 5

University/Industry Consortium Project Review

Functionally Gradient Materials for Thermal Barrier Coatings in Advanced Gas Turbine Systems

CONTRACT INFORMATION

Contract Number	DE-FC21-92MC29061
Contractor	Lehigh University Department of Materials Science and Engineering Whitaker Laboratory 5 East Packer Avenue Bethlehem, PA 18015 (610) 758-4220 (610) 758-4244
Principal Investigator	Dr. Arnold R. Marder
Co-Investigators/Research Students	Dr. Katayun Barmak Dr. Helen M. Chan Dr. Martin P. Harmer Dr. Brian J. Smith Dr. Suxing Wu Stephen W. Banovic Megan C. Slusser
DOE Project Manager	Dr. Paul Micheli Dr. Norm Holcombe
Period of Performance	September 1, 1993 to August 31, 1996
Schedule and Milestones	

FY93-96 Program Schedule

	O-D	J-M	A-J	J-S	O-D	J-M	A-J	J-S	O-D	J-M	A-J	J-A
Literature review	-----											
Processing		-----										
Characterization		-----										
Environmental testing				-----								

OBJECTIVES

It is the objective of this research program to investigate novel compositions and unique processes to utilize Functionally Gradient Materials as thermal barrier coatings in advanced gas turbine engines.

BACKGROUND INFORMATION

New designs for the advanced gas turbine engine for power production will require Thermal Barrier Coatings (TBC) to protect some turbine components. TBC's represent a potential major operational improvement in turbine engine performance by allowing an increase in the maximum inlet temperature for an operating engine. Although thermal barrier technology is over thirty years old, the principal failure mechanism is still the spallation of the ceramic coating at the ceramic/bond coat interface. A solution to this problem is to develop a coating that combines the thermal barrier qualities of the ceramic layer and the corrosion protection provided by the metallic bond coat without the detrimental effect of an interface. A ceramic/metal hybrid material is being developed for just such properties and is called a Functionally Gradient Material¹ (FGM).

FGM's are specifically designed and manufactured ceramic/metal composites that combine a heat-resistant ceramic outer skin with a higher toughness metallic inner layer. In an FGM, there is a continuous, gradual change in composition from 100% ceramic at the surface to 100% metal at the interior. The center of the material is a ceramic/metal mixture that represents a smooth transition in material composition and mechanical properties. As a result, coating spallation based on either a thermal fatigue microcrack accumulation mechanism or corrosion fatigue failure mode would be eliminated because there is no interface separating two dissimilar materials. In addition, novel compositions that

will resist higher temperatures than existing TBC systems without coating degradation can be developed.

PROJECT DESCRIPTION

The technical approach to this research program consists of three tasks:

- Task A: Literature Review Determine potential novel FGM compositions and unique processing techniques based upon a literature review.
- Task B: Processing/Structure Characterization of FGMs Produce FGM coatings by novel processing techniques and develop proper characterization techniques.
- Task C: Environmental Testing of FGMs Evaluate the thermal fatigue properties, corrosion resistance, and optimum coating thickness of potential FGM systems.

RESULTS

Task A: Literature Review

Two literature reviews were completed by the end of the third quarter of this first year. The first, *Functionally Gradient Materials for Thermal Barrier Applications*² presented a review of the current FGM literature and its application as a thermal barrier coating. Various fabrication techniques used to produce graded materials, both as coatings and as free-standing materials, were reviewed briefly and evaluated with respect to their effectiveness in producing a viable FGM. Because the uniqueness of the graded structure may provide the ability to use alternative material systems as TBC's, a section was included that considers possible candidates for future research consideration. FGM characterization and modeling work was also reviewed. The second

review of two alternative FGM fabrications techniques, *Functionally Gradient Materials Produced by Electrodeposition and Reaction Bonded Aluminum Oxide*³, discusses these processes in depth. Explanations of the two processing techniques and how they can be used to produce FGM thermal barrier coatings were detailed.

Task B: Processing/Structure Characterization of FGM's

RBAO Processing. As indicated in the literature, a potential application of RBAO technique is functionally gradient materials (FGMs)⁴. The task of the present work is to develop FGMs by the RBAO technique and further, to apply the FGMs to thermal barrier coatings (TBC's) of turbine components.

By adjusting experimental conditions, such as powder composition, milling intensity, compaction pressure and heating cycle, the RBAO process was successfully reproduced. A powder mixture of 45 vol% Al, 45 vol% Al₂O₃ and 10 vol% ZrO₂ attrition milled for 7 hours was isopressed at 300

MPa and reaction-bonded up to 1550°C in air in a box furnace. A final density of 99.3%TD was achieved, which is the highest density reported in the literature. As shown in Fig. 1, uniform and small Al₂O₃ and ZrO₂ grains (<1μm) make up the dense microstructure of the material.

In order to utilize RBAO materials in TBC's, a tape casting technique is being developed at Lehigh. The process is based on cation exchange in an alginate-containing slip. It is important to inhibit premature oxidation of the Al powder constituent before heat-treatment. In addition to a gelling agent, other organic additives are required to stabilize the slip, inhibit foaming, and minimize the slip viscosity at given solid loading. Gel-cast green compacts of RBAO precursor powders have

Figure 1. Microstructure of an RBAO ceramic body

Figure 2. Tape casting of RBAO precursor powder

been prepared with green densities of $>60\%$, far higher than commonly observed with Al_2O_3 gel-cast green compacts. Fig. 2 shows a 1 mm-thick gel-cast tape of an RBAO precursor powder consisting of 45vol% Al. Tapes of different compositions are to be cut into pieces and laminated together to obtain layered structures which result in graded ceramic profile (i.e., an FGM) after heat-treatment.

Electrodeposition Processing. The electrodeposition laboratory set-up has been completed, with process control computer software acquired to automate the deposition process. A duplex composite coating ($\text{Ni}/\text{Ni}-\text{Al}_2\text{O}_3$) deposited on a pure nickel substrate has been produced with good results using the electrodeposition technique, Figure 3. When viewed in cross-section, the coating/substrate interface is very flat and showed no defects. In observing the two deposited layers (metal region to composite region), a smooth transition can be seen with no defined interface between the two. The coating surface itself is also defect free, with

Figure 3. Light optical micrograph (500x) of duplex $\text{Ni}/\text{Ni}-\text{Al}_2\text{O}_3$ composite coating applied on a nickel substrate (left-etched, right-unetched)

only the Al_2O_3 particles visible under high magnification. Approximately 22 volume percent alumina particles ($1.0\ \mu\text{m}$), with an even distribution, was determined in the coating. Optimization of the process conditions is expected to increase the particulate volume percent in the coating⁵.

Figure 4. Variation of electrodeposited composite coating thickness with current density (45°C , 2 hrs, $20\ \text{g/l}\ \text{Al}_2\text{O}_3$, 600 rpm)

A series of experiments aimed at quantifying the effect of current density on coating thickness was also performed. The change in coating thickness with current density is shown in Figure 4. The deposited coating thickness increased linearly with increased current density while all other variables were held constant. This relationship demonstrates that the deposition rate can be quite high for this process ($>100\ \mu\text{m}/\text{hr}$) and indicates that electrodeposition can be competitive with some other high deposition coating techniques (e.g. thermal spraying, electron beam physical vapor deposition).

Coating Characterization. Metallographic preparation techniques have been developed to characterize the electrodeposited composite coatings via light optical and scanning electron microscopy. Microhardness techniques have also been developed to help relate the properties of the composite layer to the processing technique. Hardness measurements displayed a small deviation in the data from the coating. Micrographs of the indentions confirmed that the deviation is due to slight agglomeration of the alumina particles in the coating. A new wetting agent, as well as a different method for suspending the particles in the electrolyte solution, are presently being investigated to alleviate this problem. Bend testing techniques are also being finalized, with deposition occurring on thinner substrates. The composite layer is placed in tension in order to observe the adhesion of the coating to the substrate and the fracture mode of the electrodeposited layer.

Task C: Environmental Testing of FGM's

Development of a thermal fatigue apparatus has begun in order to evaluate the ability of the coatings to accommodate thermally induced cyclic strains and resist spallation. Samples will be cycled through a flame of commercially available compressed gas mixtures (H_2 , O_2 , and nature gas). Monitoring the coating and substrate temperature will occur throughout each test. Post testing microstructural analysis will be used to evaluate the ability of each coating to accommodate the thermally induced strains and resist fatigue crack propagation. An evaluation of coating spallation resistance will also be made.

Hot corrosion (oxidation) testing will involve isothermal exposure for various lengths of time in a controlled atmosphere tube furnace. Commercially available compressed gas mixtures will be the source of the corrosive gases. The corrosion environments will be selected to produce an oxidation, mixed oxidant (O_2/S_2), and

sulfidation corrosive attack of the coatings. Post-exposure coating characterization will document any microstructural damage and summarize the corrosion resistance of each coating.

FUTURE WORK

Future work will revolve around the completion of processing variables optimization for each technique. Evaluation of the thermal fatigue and hot corrosion (oxidation) properties of the material will also be conducted.

REFERENCES

1. EPRI 1982. Thermal barrier Coatings for Utility Gas Turbines, Report AP-2618, Palo Alto, CA: Electric Power Research Institute.
2. Claussen, N., Wu, S., and Holz, D. 1994. "Reaction bonding of aluminum oxide (RBAO) composites: processing, reaction mechanisms, and properties", *J. Eur. Ceram Soc.*, **14**, 97-109.
3. Smith, B.J. and Banovic, S.W. 1994 Functionally Gradient Materials for Thermal Barrier Coatings: A Literature Review, Report 94-500-17-31, Bethlehem, PA: Energy Research Center and Materials Research Center at Lehigh University.
4. Smith, B.J. and Wu, S. 1994. Functionally Gradient Materials Produced by Electrodeposition and Reaction Bonded Aluminum Oxide: A literature Review, Report 94-500-18-33, Bethlehem, PA: Energy Research Center and Materials Research Center at Lehigh University.
5. Celis, J.P., Roos, J.R., Buelens, C., and Franssaer, J. 1991. "Mechanism of electrolytic composite plating: survey and trends", *Trans. Inst. Metal Finish*, **69** (4), 133-9.

Life Prediction of Advanced Materials for Gas Turbine Application

CONTRACT INFORMATION

Contract Number AGTSR 9301SR012D

Contractor Pennsylvania State University
121 Hammond Bldg
University Park, Pa. 16801
Tel & Fax: 814-865-5241

Principal Investigator Dr. Sam Y. Zamrik

Co-Investigators/research students Dr. Asok Ray (Mechanical Engineering Dept.)
Dr. Donald A. Koss (Material Sc.& Eng. Dept.)
Mark L. Renauld (Grad. Student)
Ravindra Annigeri (Grad. Student)

DOE Project Manager Dr. Paul Micheli / Dr. Norm Holcombe

Period of Performance Sept. 1, 1993 to Sept. 30, 1994

Schedule and Milestones

Sept.1, 1993 to August 31, 1996

	9/93	9/94	9/95	9/96
Literature Search	----- x			
Material Procurement	-----x			
Specimen Fabrication	-----	-----	-----x	
Uncoated LCF Testing	-----	-----x		
Uncoated Hold-Time Testing	-----	-----	-----x	
Coated TMF Testing	-----	-----	-----x	
Microstructural Evaluations				
Analysis, Modeling	-----	-----	-----	-----x
Reporting Periods	-----x-----	-----x-----	-----x-----	-----x-----

OBJECTIVES

The main focus of the research program is directed towards the life prediction modeling of coated advanced gas turbine materials. Emphasis is placed on predicting the lives of gas turbine components whose performance is limited by low cycle fatigue (LCF) under isothermal conditions and thermo-mechanical fatigue (TMF). The program also includes analysis of microscale damage of failed coated and uncoated specimens as well as the deformation response, the fracture mechanism and environmental effects. IN 738 LC has been selected as a basic material which will be followed in the future by directionally solidified (DS) and single crystal (SC) materials..

BACKGROUND INFORMATION

Most studies of fatigue life prediction have been reported under isothermal conditions where the deformation of the material is strain dependent. In the operation of gas turbines, components such as blades and vanes are exposed to a temperature variations in addition to strain cycling. As a result, the deformation and fracture process becomes dependent on temperature as well as on strain. Therefore, the life of the component becomes dependent on a temperature-strain cycling which produces a process known as "thermomechanical fatigue or TMF". The TMF fatigue phenomenon has been modeled on the basis of conventional fatigue life prediction methods, which unfortunately are not sufficiently accurate to quantitatively establish an allowable design procedure. To add to the complexity of TMF, blade and vane substrates are normally coated with aluminide, overlay or thermal barrier type coatings. In such cases the durability of the component may be dominated by the coating/substrate constitutive response and by the fatigue behavior of the coating. The TMF of coated turbine blades or vanes introduces a number of issues depending on the type of temperature/strain phase cycle:

1- the mismatch in deformation behavior between the coating and substrate can

substantially modify the associated stress response of the coating/substrate system.

2- time-dependent inelastic behavior can significantly affect the stress response. For example, creep relaxation during a tensile or compressive loading at elevated temperatures can lead to a progressive increase in the mean stress level under cyclic loading, affecting fatigue lives.

3- the mismatch in elastic and thermal expansion properties between the coating and the substrate can lead to a significant deviations in the coating stress levels due to changes in the modulus

4- the oxidation and corrosion resistance coatings applied to the substrate may act as primary crack initiation sites.

Crack initiation in the coating is a function of the coating composition, its mechanical properties, creep relaxation behavior, thermal strain range and the strain/temperature phase relationship. Of particular importance are the coating ductility, coefficient of thermal expansion mismatch between the coating and substrate which can cause thermally induced strains causing cracking at the surface and creep relaxation.

As a result of the complex constitutive behavior of the coating/substrate system, TMF life prediction methodology has not been developed to explicitly describe the fatigue response of the system.

PROJECT DESCRIPTION

The project is divided into two parts: experimental and analytical. The TMF experimental part is not as straight-forward as the isothermal fatigue testing and becomes very sensitive to temperature/strain programming. Software programs for TMF and heating were developed to include a number of loading schemes such as:

- Compression, or tension or tension-compression cycling combined with in-phase or out-of-phase temperature cycles as well as dwell time effects.

- Combined axial-torsion strain cycles with or without dwell times for in-phase or out-phase strain/temperature cycles.

The experimental testing system consisted of a servo-hydraulic MTS system designed for axial or torsional strain cycling at high temperatures. Only the axial load mode was used in this study. The heating was by induction with three adjustable coils to maintain uniform temperature distribution over the specimen gage length with five thermocouples equally spaced and wrapped around the gage length measuring the temperature distribution. Strain was measured by an axial high temperature air cooled MTS extensometer with wedge probes placed over one inch gage length. The strain and temperature input were computer controlled. A schematic of the TMF system is shown in Fig. 1. The basic TMF testing procedure consisted of three main steps, the start-up process, the second step dealt with the temperature ramp and thermal cycling and the third step was the TMF cycling where coupling and decoupling of strain/temperature is a critical task to avoid temperature overshoot as shown in Fig. 2. Thermocouples and coils placement are shown in Fig. 3. These steps are somewhat broad but each of them contain operations, that must be done in a specific sequence.

EXPERIMENTAL RESULTS

- Material:

The nickel base superalloy IN 738 LC in the solution-treated condition (1120° C for 2 hrs, air cooled), was received in the as-cast bars, one inch in diameter by six and quarter inches long. Tubular test specimens, shown in Fig. 4, were machined by a low stress grind process and were aged in a dynamic vacuum of $\sim 2.7 \times 10^{-4}$ pa. for 24 hrs at 1550°F (843°C) followed by an air cool. The microstructure of the cast material consists of dendritic structure with a high volume fraction of fine-scale gamma-prime particles as well as large second phase particles, mostly primary gamma prime and/or carbides, in the interdendritic regions. No evidence of large-scale (>1mm) porosity was seen, although fine-scale ($\leq 40 \mu\text{m}$) pores were present. The

chemical composition of the material is presented in Table 1.

- Fatigue Tests:

Isothermal fatigue specimens were tested at a temperature of 1500°F (816°C) and cycled at 10 cpm at three total strain ranges: 2.0%, 0.5% and 0.3%. The experimental LCF responses are shown in Fig. 5, which also includes published data from a number of sources. It is interesting to note that the variation in the temperature ranges (750-980°C) has not significantly affected the observed lives. There is an effect on the high temperature LCF due to the frequency of cycling. For example, the low strain-range data reported by Kuwabara [1] at 900°C tested at 0.336 cpm shows a lower life when compared to data by Au [2] at 980°C tested at 6-30 cpm. The difference may be attributed to a strain rate effect (frequency) which enhances the oxidation/creep process at the low cyclic frequency and longer exposure time. This is shown in Fig. 6 where the reported data is replotted in terms of plastic strain range as a function of the cycles to failure. All plastic strain range data show that material such as IN 738 LC with low ductility cannot sustain much plastic deformation leading to a short fatigue life.

- Stress Relaxation Tests:

Two types of hold-time (dwell time) tests were carried out at two total strain ranges of 0.5 and 0.3% with a 90 seconds hold time period. In one test type, the hold time was in tension at maximum tensile strain amplitude, while the other test imposed the 90 sec. hold at maximum compressive strain amplitude. The difference between the tension and compression 90 seconds hold-time tests is the apparent response of the mean stress which becomes compressive in tension hold and tensile in the compression hold. A comparison of lives between the two types of hold-time tests under 0.5% strain range showed that hold times of 90 sec at maximum compression, based on 10% load drop, the LCF life was reduced by a factor of 8. However, when the two type of hold time tests are compared to LCF data, the tensile

hold time showed an increase in life, whereas the compression hold time showed a significant decrease in life by a factor of 5. The compression hold time data of 90-second was also compared to a 2-minute compression hold-time tests reported by Ostegren.[3] as shown in Fig. 7. Little difference in fatigue life was observed between the 90 and 120 seconds tests at the two temperatures;

In analyzing the stress relaxation response, the total cyclic strain range, ($\Delta\epsilon_t$), is the sum of four strain components : an elastic strain range ($\Delta\epsilon_e$), a plastic strain range ($\Delta\epsilon_{pp}$), a creep strain range ($\Delta\epsilon_{cp}$), and a recoverable anelastic strain range ($\Delta\epsilon_a$). The first two strain components, elastic and plastic strain ranges, are the total strain range in high temperature low cycle fatigue which can be evaluated by using the universal slope method. However, the creep and the anelastic strain components are the result of the stress relaxation which occurs during the hold-time period. In analyzing the stress relaxation data (Fig. 8), one observes a rapid load drop which occurs initially within the hold time cycle. The period of rapid stress decay is ≤ 1 sec, followed by a gradual decrease of stress with time (the relaxation process). An analysis of the creep strain data is normally obtained at the stabilized cycle at mid-life, where the amount of stress relaxation remains constant (saturation stage) and becomes independent of the number of cycles. In analyzing the relaxation data, two methods were considered in describing the stress relaxation: (a) time hardening, and (b) strain hardening. The time hardening rule was used because it has a closed form solution.

Stress relaxation during hold times was analyzed by using a stress rate procedure, where the stress rate is normalized by an instantaneous relaxed stress raised to a power n where n is the exponent for primary creep and for IN 738 LC, $n = 4.308$. Using the time hardening rule, the following relation is obtained:

$$-\frac{\dot{\sigma}}{\sigma^n} = A_u t^{m_u} \quad (1)$$

where A_u and m_u are material constants evaluated from the relaxation data. The negative sign is used to account for the decaying stress rate. Integrating the equation, the stress relaxation, σ_r , can be expressed as:

$$\sigma_r = \left[\frac{A_u (n-1)}{(m_u + 1)} t^{(m_u + 1)} + \sigma(o)^{1-n} \right] \frac{1}{1-n} \quad (2)$$

where $\sigma(o)$ is the onset stress relaxation at $t = 0$. Fig. 9 is the plot of Equation (2). The result of stress-rate calculation, performed at mid-life, is shown in Fig. 10 for strain ranges of 0.5, 0.3% .

The creep strain accumulation per cycle, including time-independent plasticity, can also be calculated from the following relation:

$$\Delta \epsilon^{cp} = \frac{\Delta \sigma^{\max} - \Delta \sigma^r}{E} \quad (3)$$

where ($\Delta \sigma^{\max} - \Delta \sigma^r$) is the relaxed stress which is measured from the bottom tip of the hysteresis loop where the effect of mean stress is taken into account. Fig. 11 shows that the magnitude of the creep strain accumulation per cycle as a result of hold-time cycling is constant.

- Creep Rupture Data:

One of the design procedures for life assessment is the stress rupture criterion. The most commonly used is the Larson-Miller parameter (LMP) which estimates the rupture life for a given stress and temperature. The resulting curve normalizes the effect of temperature; however if the effect of temperature and creep strain rate on life for a given stress is to be separated, then the Dorn approach is more appropriate.

The LMP parameter in terms of rupture time, temperature T , and stress σ is expressed as:

$$\text{LMP} = T (C + \log t_r) = f(\sigma) \quad (6)$$

and

$$f(\sigma) = x_1 + x_2 \log \sigma + x_3 (\log \sigma)^2 + x_4 (\log \sigma)^3 \quad (7)$$

The constants are determined by non-linear regression.

The Dorn parameter is expressed in terms of activation energy, ΔH , stress and absolute temperature T as:

$$t_T \exp (\Delta H / R T) = f(\sigma) \quad (8)$$

rearranging:

$$\log t_T - (\Delta H / R T) \log e = \log t_T - C / T = G(\sigma) \quad (9)$$

where

$$G(\sigma) = C_1' + C_2' s + C_3' \log \sigma \quad (10)$$

or

$$\log t_T = C_1' + C_2' \sigma + C_3' \log \sigma + C' / T \quad (11)$$

The creep rupture data of IN 738 LC obtained from literature is plotted in terms of the two criteria as shown in Figs. 12-13.

FUTURE WORK:

Thermomechanical fatigue (TMF) tests of coated and uncoated specimens will be performed and analyzed to assess the effect of the coating on life. Microstructural analysis will be continued to characterize the damage and to identify fracture mechanisms. Future directionally solidified (DS) and single crystal (SC) materials will be investigated under conditions where life is affected by the material durability and resistance to creep damage. Existing life prediction methods will be analyzed on the basis of the data and will be extended as needed.

INDUSTRIAL PARTNERS PARTICIPATION:

The PI of the research program is in direct consultation with our industrial partners: Westinghouse Electric, Power Generation Division in Orlando, FL., Solar Turbine in San Diego, CA. and Allied Signal Aerospace Co. of Phoenix, AZ. Research results will be disseminated through

conferences and seminars. Our laboratory facility has been opened to visitors from our industrial partners and from government laboratories such as Oak Ridge National Laboratory and NASA-LRC to observe our TMF life prediction program and test facility.

REFERENCES

- 1- Kuwabara, Nitta, A., and T. Kitamura, "Thermal-Mechanical Fatigue Life Prediction In High Temperature Component Materials For Power Plant," ASME Int. Conf. on Advances in Life Prediction Methods, Materials Conference, N.Y., 1983, pp. 131-141.
- 2- Au, P., and R. V. Dainty, "Isothermal Low Cycle Fatigue Properties of Diffusion Aluminide Coated Nickel and Cobalt Based Superalloys," Surface Modification Technologies III, Eds., T.S.Sudarshan and D. G. Bhat, TMS, 1990, pp.729-747.
- 3- Ostergren, N., J., "A Damage Function and Associated Failure Equations for Predicting Hold-Time and Frequency Effects in Elevated Temperature, Low Cycle Fatigue," Journal of Testing and Evaluation, Vol. 4, No. 5, Sept. 1976, pp. 327-339.
- 4- Jiao, F. "Low Cycle Fatigue Behavior of IN 738 LC at 1223K," Proc. of Third Int. Conf. on Low Cycle Fatigue and Elasto-Plastic Behavior of Materials," 1992, pp.298-303.
- 5- Strang, S., "High Temperature Properties of Coated Superalloys," Behavior of High Temperature Alloys in Aggressive Environment, Proc. of Petten Int. Conf., The Metal Society of London, 1979, pp.595-611.
- 6- Castillo, R. and A. K. Koul, "Effects of Microstructure Instability on the Creep and Fracture Behavior of Cast IN 738 LC Ni-base Superalloy," High Temperature Alloys for Gas Turbines and Applications," Betz, W., eds, D. Reidel Publishing Co., Dordrecht, 1986, pp.1395-1409.

7- Castillo, R. and A. K. Koul, "Factors Influencing the Residual Creep Life Prediction of Service Exposed Ni-base Superalloys Components," Life Assessment and Repair Technology for Combustion Turbine Hot Section Components, ASM Int., Metal Park, 1990, pp 19-34.

8- Tipler, H., R., "Regeneration of the Creep Properties of a Cast Ni-Cr-base Alloy," AGARD-CP-317, Paper #12.

9- Castillo, R., A. K. Koul and E. H. Toscano, "Lifetime Prediction under Constant Load Creep Conditions for a Cast Ni-base Superalloy," Journal of Engineering for Gas Turbines and Power, vol. 109, 1987, pp.99-106.

10- Kloos, K., H., and J. Granacher, and H. Demus, "Creep Rupture Strength of Cast Turbine Blades under Hot Gas Corrosion," Corrosion and Mechanical Stress at High Temperature, Guttman, V. and Merz, M. eds., Applied Science Publishers, London, 1981, pp. 243-255

11- Hartnagel, and W., R. Bauer and H. W. Grunling, "Constant Strain Rate Creep Tests with Gas Turbine Blade Material under Hot Corrosion Environmental Conditions," Corrosion and Mechanical Stress at High Temperature, Guttman, V. and Merz, M. eds., Applied Science Publishers, London, 1981, pp. 257-272.

12- Villat, M., and P. Felix., "High Temperature Corrosion Protective Coating for Gas Turbines," Sulzer Technical Review, Vol. 3, 1976, pp. 97-104.

13- Nazmi, M. Y., "High Temperature Low Cycle Fatigue of IN 738 and Application of Strain Range Partitioning," Metallurgical Trans. A, ASM, Vol. 14A, March 1983, pp. 449-461.

14- Zamrik, S. Y., "An Interpretation of Axial Creep-Fatigue Damage Interaction in Type 316 Stainless Steel," ASME Trans., Journal of Pressure Vessel Technology, Vol.112, Feb. 1990, pp. 4-19.

15 Gabrielli, F, and M. Marchionni and G. Onofrio, "Time Dependent Effect on High Temperature Low Cycle Fatigue and Fatigue Crack Growth Propagation of Nickel Base Superalloys," Advances in Fracture Research, Proc., of the 7th Int. Conf. on Fracture (ICF7), March 1989, pp. 1149-1163.

TABLE 1: CHEMICAL COMPOSITION OF ALLOY IN 738 - LC

Heat Treatment	Element	Percentage
Solutionize at 2050 °F (1120 °C) for 2 hr followed by aging at 1550 °F (843 °C) for 24 hr.	Cr	15.7 - 16.3
	Co	8.0 - 9.0
	Ti	3.2 - 3.7
	Al	3.2 - 3.7
	W	2.4 - 2.8
	Mo	1.5 - 2.0
	Ta	1.5 - 2.0
	Nb	0.6 - 1.1
	Ni	Balance

Fig. 1. System Configuration

Fig. 2. Temperature Command Signal and Specimen Response

Fig. 3. Cross-Section Of Specimen, Induction Coils and Thermocouples

NOTES: DIMENSIONS IN INCHES
 UNLESS OTHERWISE STATED, TOLERANCES ARE ± 0.002
 GAGE DIAMETER: $0.500 +0.002, -0.000$
 GAGE WALL THICKNESS: 0.060
 COATING THICKNESS: 0.002
 COATING TYPE I: RT21 (NIAI), CLASS I (24-29) WT%Al

Fig. 4. PSU Tubular TMF Test Specimen

Fig. 5. Isothermal LCF Test Data of IN 738 LC

Fig.6. Isothermal LCF Test Data of IN 738 LC

Fig. 7. Isothermal Hold Time LCF Test Data of IN 738 LC

Fig. 8. Analysis of Relaxation Data

Fig. 9a. Mid-Life Stress Relaxation Data - IN 738 LC
Strain Range = 0.5%, Temperature = 1500 °F(816 °C)
Hold Time = 90 sec (Compression), $N_f = 622$ Cycles.

Fig. 9b. Mid-Life Stress Relaxation Data - IN 738 LC
Strain Range = 0.5%, Temperature = 1500 °F (816 °C)
Hold Time = 90 sec (Tension), $N_f = 4959$ Cycles.

Fig.10a. Mid-Life Stress Relaxation Data - IN 738 LC
Strain Range = 0.5%, Temperature = 1500 °F (816 °C)
Hold Time = 90 sec (Compression), $N_f = 622$ Cycles.

Fig. 10b. Mid-Life Stress Relaxation Data - IN 738 LC
Strain Range = 0.5%, Temperature = 1500 °F (816 °C)
Hold Time = 90 sec (Tension), $N_f = 4959$ Cycles.

Fig. 11. Creep Strain Accumulation Per Cycle
Strain Range = 0.5%, Temperature = 1500 °F (816 °C)
Hold Time = 90 sec (compression), N_f = 622 Cycles.

**Fig. 12. Creep Analysis Using Larson-Miller Parameter
For IN 738 LC**

Fig. 13. Creep Analysis Using Dorn Type Parameter
For IN 738 LC

REDUCTION OF TURBINE ENDWALL TOTAL PRESSURE LOSS AND HEAT TRANSFER USING THE ICEFORMATION DESIGN METHOD

CONTRACT INFORMATION

Contract Number	<u>93-01-SR016</u>
Contractor	Clarkson University Box 5725, Department of MAE Potsdam, NY 13699-5725 Phone: (315) 268-3823
Principal Investigator	Dr. Ronald S. LaFleur, Associate Professor
Co-Investigators/Research Students	Mr. Brent Gregory, General Electric, Aircraft Engines Dr. Thomas Fric, General Electric, R&D Center Dr. Lee Langston, Consultant Michael Halter, Research Engineer Juan Araujo, Graduate Research Assistant Patrick Wing, Undergraduate Research Fellow Steve Sperling, Undergraduate Research Fellow
DOE Project Manager	Dr. Paul Micheli / Dr. Norm Holcombe
AGTSR Program Management	SCERDC: Daniel B. Fant / Lawrence P. Golan
Period of Performance	August 1, 1993 to July 31, 1995
Schedule and Milestones	Program Schedule August 1, 1993 - July 31, 1995

Activity	Year	1	2
	<u>Quarter</u>	<u>1</u>	<u>2</u>
I. Design of Turbine/Endwall Iceformation Experiment and Conditions		1	2
II. Turbine/Endwall Iceformation Apparatus Preparation		1	2
III. Evaluation and Preliminary Iceformation Experiments		1	2
IV. Turbine/Endwall Iceformation Design Experiments		1	2
V. GE Performance Testing of Turbine/Endwall Iceforms		1	2
VI. Turbine/Endwall Iceform Shape and Performance Data Representation		1	2

1. OBJECTIVES

Activities during the first year of this project were primarily related to experimental design, fabrication of the apparatus and preliminary testing. The research activities were directed by the following objectives:

1. To design a set of turbine endwall geometries using the iceformation method and modelled turbine guide vane conditions.
2. To evaluate the changes of flow and thermal distributions that correspond to pressure loss reduction and heat transfer reduction.
3. To establish gas turbine endwall design principals based on relationships between iceform endwall features and performance.
4. To operate a university-industry research team that collectively proposes, discusses and decides experimental and design issues.
5. To form a program that encourages undergraduate and graduate careers in the gas turbine research, design and technology fields.

The project is expected to yield tangible reductions in turbine pressure losses and heat transfer by contouring the endwalls. A design methodology will result by relating iceform endwall features to experimental conditions.

2. BACKGROUND INFORMATION

2.1 The Turbine Endwall Design Problem

The contour between turbine blades over the hub or shroud is important because it contributes to aerodynamic and heat transfer performance and energy efficiency of gas turbines. The losses of total pressure, heat and

available energy from the gas flow greatly affect the turbine's performance. The turbine endwall design problem becomes increasingly important as blade loading increases due to lowered number of blade parts per engine and the increasing flow accelerations of reaction stages.

Both the rotor and stator passages of turbines contain necessary pressure gradients for work transmission. Consequently, the passage pressure gradients drive the near-surface boundary layers into secondary flows called horseshoe, passage and counter-rotating corner vortices. The passage secondary flow is an extreme departure from the ideal two-dimensional radial equilibrium conditions assumed during design of the passage aerodynamics. In short span cases (high pressure stages), the secondary flow exists over 25-40% of the passage span [1]. This creates a total pressure loss due to off-design over- and under-turning of the near endwall flow.

The secondary flows in turbines have been studied extensively over the past 30-40 years. A sample of this work includes flow visualization studies and 5-hole pressure probe measurement of turbine velocities and vortices that contribute to total pressure losses. Endwall surface flow visualization studies by Turner [2], Hunter [3] and Langston et al. [4] have shown the complex structure of the turbine/endwall juncture flow. The flow is marked by multiple flow separation lines, flow attachment lines and saddle points of separation consistent with the presence of passage and counter-rotating vortices in the turbine passage. The vortices cause off-design over- and under-turning of the flow at different span (radial) positions and a loss of total pressure in the turbine passage due to momentum deficits in the wake. Quantitative experimental data using five-hole pitot probes have been reported [2-6]. The five-hole probe resolves total and static pressures and three velocity components. The total pressure contour data is a good indicator of turbine aerodynamic performance and the effect of design changes.

2.2 Design Philosophies and Review of Solutions

The design methodology for turbine endwalls can be based on the fundamental lifting bodies juncture with endwalls. The basic lifting body/endwall junction flow has been studied by many investigators including Hasan et al. [7] and Devenport and Simpson [8,9]. All investigations have found that the juncture flow is complex and is not easily predicted using turbulence models. Hasan et al. [7] found that the pressure distribution and horseshoe vortex character were strongly linked. This suggests that to reduce total pressure loss by reduction of vortex motions, the design methodology must radially alter the turbine passage pressure distribution. The three-dimensional pressure gradients can be altered by shaping of the endwall surface. Since the endwall flow is three-dimensional and unsteady, turbine designers are faced with difficult decisions about how to shape the endwall to improve aerodynamic performance, minimize heat transfer and lower the impact on downstream components while maintaining turbine stage design conditions. The difficulty is that the complex turbine flows do not readily provide guidance for design decisions.

According to Hoerner [10], all pressure gradients affect drag and lift of fluid components. Consequently altering the pressure gradients in the near juncture flow may reduce drag and enhance lift by expanding the effective span of the turbine blade. This is especially true when lift (turbine torque) and work output must be gained from the limited spans of high pressure turbine passages. Sieverding [11] provides a review of loss reduction methods up to 1975. A number of endwall designs have been tried to reduce the secondary flow strength by changing near endwall pressure distributions. The designs, which are reviewed below, include the following:

1. Circular radius fillet
2. Prumper fences
3. Camarata and Dring clay geometry
4. S-endwall or "Russian Kink"
5. Iceformation endwall

These design variations are distinguished not only by the geometric result but also by the design mechanism behind the concept.

One idea which seems to be rooted in structural concerns more than aerodynamics is to fill in the turbine/endwall juncture's sharp corner with a fillet. The effect of a fillet on lifting body/endwall juncture flows has been studied by Devenport et al. [12], and Sung et al. [13]. The fillet altered pressure gradients near the juncture corner. Addition of a fillet to the juncture's corner did not definitively reduce total pressure drop. The fillet is not necessarily an aerodynamic or heat transfer benefit.

Since its inception by NACA in 1953, the use of thin metal strips or fences to resist crossflow or channel near surface flow was clarified by Prumper [14]. Prumper used experimental trial and error on over 400 different configurations in a first vane nozzle to evolve an "optimal" fence. Prumper suggested design guidelines such as the fence should start where the crossflow meets the suction surface and the fence height should taper from zero to full height. The results apply only to gas flows below the turbine blade material's melting point and the specific boundary conditions of Prumper's experiment. A new application may require a repeat of extensive trials.

A qualitative idea is to use flow visualization and a clay endwall to mold and shape the surface to minimize the passage crossflow. One example of this is an experimental design study by Camarata and Dring [15]. Their study yielded an endwall contour map that depicts a sharp ridge running down the turbine passage center. Although no quantitative data was obtained, the ridge was observed to inhibit the crossflow that contributes

to the growth of the passage vortex much like one of Prumper's endwall fences. This technique requires repeated trial and error if flow conditions or passage loading changes.

The one endwall design that is widely used in practice employs an axi-symmetric shape or "meridional wall profiling" called a "Russian kink" or "S-endwall" because the surface bends twice like an "s". Examples of S-endwalls that reduce turbine losses by as much as 25% are cited by Sieverding [11], Morris and Hoare [16], Kopper et al. [5] and Boyle et al. [17]. The interesting result is that secondary losses of the uncountoured endwall (shroud) were reduced rather than the countoured endwall (hub) losses. This suggests that changing the passage pressure distribution may be more effective than just altering the near endwall pressure distribution. Consequently, highly countoured endwall shaping that alters midspan pressures may be a better design idea than mild surface alterations.

The iceformation technique was applied to cylinder and turbine endwall junctures by LaFleur [1] after success with sudden enlargement diffusers [18]. The iceformation technique uses a slowly moving phase change boundary to allow the geometry to change in direct response to flow and heat transfer while the flow is altered by the shape change. LaFleur and Langston [19] reported an 18% drag reduction of a cylinder/endwall juncture over range of high Reynolds numbers. LaFleur [1] generated new concept turbine endwall contours. Since these contours were tested using an available three-hole probe, a conclusive comparison of performance could not be made.

2.3 Prior Iceformation Design Studies

The iceformation idea was initiated by the late Professor W.W. Bowley. While studying solidification on the sub-freezing surfaces of a square channel inlet, he noticed that the resulting shapes were smooth and might be "streamlined"

[20]. The mechanism was different than the detrimental aircraft wing-icing problem because the surface was kept cold and the flow relatively warm. The fact that "streamline" geometries were produced led to Bowley's idea that the iceformation process could be used to generate geometries for difficult fluids design situations.

A number of workers, such as Carlson [21] studied iceformation and generated smooth shapes but did not definitively demonstrate that the shapes were low-drag or of reduced heat transfer. LaFleur [1,18,19], has performed experiments and demonstrated that iceformation produced diffusers and cylinder/endwall juncture contours with superior performance over accepted design solutions. Theories have been developed by LaFleur [1,22-25] to explain how iceformation could be used to design low total-pressure loss and reduced heat transfer contours. The theories form a basis for running experiments and controlling the iceformation process using Reynolds number, flow temperature and substrate temperature [24,25]. Some novel geometries that may have practical value are produced by the iceformation mechanism. For example, LaFleur [1] identified geometric features of a turbine endwall iceform that are insensitive to angle of attack variations.

3. PROJECT DESCRIPTION

This project is focussed on the iceformation method design of turbine endwall shapes as countoured hubs or shrouds. The goal of the work is to demonstrate a method for altering near endwall pressure gradients to beneficially affect the complex juncture flow and reduce total pressure losses and heat transfer. This project uses a linear cascade model of the Energy Efficiency Engine (E³) second stage vane to demonstrate the design methodology, resulting geometries and performance improvements. The intent is to apply the method and results to annular endwalls in gas turbines.

The project has been organized into six tasks as experimental design, apparatus fabrication, preliminary testing, iceformation design experiments, performance testing and processing of results. This paper describes the main activities of experimental design, apparatus testing and preliminary testing followed by design of the instrumentation system. A detailed account is given in the project quarterly reports [26]. The project is a team effort between Clarkson University and General Electric. The research team's communications have been organized by the following procedure:

1. Identify task sub-problem and constraints
2. Develop ideas and a solution proposal
3. Present the solution proposal at a university-industry research group meeting
4. University-industry research group critiques, discusses and suggests direction
5. Decisions are made collectively by the research group
6. New sub-tasks are identified for work by each team member
7. Minutes and reports document discussion, decisions and direction of the research

This mechanism has been effective in our investigation. The interaction between Clarkson and General Electric has been important. The research team has made informed decisions that guide the research to practical results. Students have had first hand exposure to program development and team decision making with the involvement of T. Fric and B. Gregory of General Electric and an expert consultant, L.S. Langston.

The general plan adopts a wind tunnel at Clarkson to supply flow to a linear cascade test section. The test section serves two purposes. First, the test section is used as a water tank to grow an initial ice layer over the cryogenically cooled endwalls. Second, the water is drained to provide the air flow passage in the wind tunnel configuration. Consequently, the initial

ice layer is shaped by the air flow. The test section was engineered and constructed at Clarkson University. Aerodynamic testing will be performed at General Electric Corporate Research and Development Center.

Both undergraduate and graduate students are learning about complex flows in turbomachines, how to improve aerodynamic and heat transfer performance and the impact design methods have on practical turbine engineering problems. The industrial members of the team give the students a perspective they do not get in course work and laboratory work. This experience and perspective will be a benefit to them when working on future gas turbine technology.

4. RESULTS

The results reported for the first year are focussed on the first three tasks of experimental design, apparatus fabrication and preliminary testing. Major hardware includes the test section, iceformation apparatus and instrumentation.

4.1. Test Section

The test section has been designed, fabricated and tested. The design was centered on adopting an appropriate test case and conducting appropriate cascade model testing that simulates a periodic row of vanes. One main item in the experimental design was the definition of vane design problem. Through team discussions, a number of endwall design options were outlined including:

- A. First stator of 78° turning and an efficiency of about 95%
- B. Second stator of 100° turning and an efficiency of about 90%
- C. Steam turbine first vane, 88% efficiency

These options were judged with the criteria of aspect ratio, heat transfer problems, aerodynamic problems and ability to be represented in a cascade arrangement. The second vane concept, case B, was chosen because little has been done to improve this component and it contains a pressure loss and heat transfer signature that could show improvement. General Electric suggested a second vane from the E³ research program stated in a NASA report by Timko [27] could be used for our design problem. Details of this decision included comparison geometries, under-cutting of the exit platform to allow near wall measurements, orientation of heat exchanger edges and leading edge porting of the sideboards. A four-times-scale model was proposed by Clarkson and was adopted with minor modification in the tailboard arrangement. The vane geometry section was taken from the "pitch line" or midspan section of the E³ test. The key issues were the scale of the blade profiles, the size of the test section, the number of blades in the test section and periodicity (the ability of a finite cascade to represent an infinite blade row). One constraint that governed scale is the 10 inch blade span of the GE-Schenectady wind tunnel inlet ducts and test section. The result is as follows

Rectilinear cascade geometry - 4X scale

Pitch = 7.14 inches

Blade height = 10 inches

Axial chord = 7.724 inches

Trailing edge thickness = 0.152 inches

Inlet and exit flow angles are determined by potential flow calculations of the cascade test section geometry performed by General Electric. The Reynolds number based on a vane chord of 10.5 inches is 1.77×10^5 and is similar to the E³ case. The four-times-scale vanes were fabricated at Clarkson. Vane coordinates, provided by General Electric, were used in a numerical control machine to generate an

original vane. This vane was used to produce a mold and cast many acrylic vanes.

The test design and fabrication was guided by the E³ test conditions given by Timko [2]. A proper finite cascade test must be periodic and provide adequate flow velocities. These conditions are provided by Clarkson's test section shown in Figure 1 through the following:

1. Adjustable Tailboards - used to balance the mass flow in each passage
2. Exit Vanes - reduce test section flow resistance
3. Sidewall boundary layer suction system - to reduce vane row end effects, each wall has an independent suction adjustment which is supported by a dedicated suction source.
4. Wind Tunnel Centrifugal Blower - Provides absolute air speeds of 110 ft/sec.

A number of water-related components were added to the test section to enable initial ice layer growth. A gate acts as a water tank wall and seals the inlet area. This allows the filling of the test section volume with chilled water through inlet and outlet plumbing. An initial ice layer is grown using the iceformation system described below. The ice layer's boundary is established by a cut-out region of the test section wall. This region's geometry is defined by the scaled-up E³ platforms and structural-thermal stress concerns on the ends of the vane row.

4.2. Iceformation System

The iceformation system is comprised of cryogenic evaporators for endwalls, a water chilling system, endwall supports and casting equipment. The system chills the water to 3°C in six hours. The chilled water can be pumped back to the insulated holding tank and used for subsequent experiments thereby saving 4 hours on the chilling time.

□ = SUPPORT.

NOTE: A) All the dimensions are in inches.
B) The test section is 10 inches deep.

Figure 1. - Layout of the Clarkson Cascade Test Section

Liquid nitrogen cooled flat plate substrates were designed to maintain an ice layer. The design method required a heat transfer analysis. The variables of concern are maximum ice thickness, flow material (air or water), flow speed or flow Reynolds number, room temperature, liquid nitrogen temperature, heat conductivities and desired steady state ice thickness. This list of quantities contains design constraints and design degrees of freedom. In the typical iceformation experiment, there are two degrees of freedom for control, namely air speed and air temperature. These two degrees of freedom arise in the design theory and closure is provided by two selections, usually flow speed and the desire that the minimum energy dissipation be attained at steady state. The heat exchanger was designed based on an average ice thickness of 1 inch and the liquid nitrogen flow rate was calculated to be about 80 liters per hour per endwall. The design for the heat exchangers was based on a number of simplifying assumptions as follows:

1. The inlet boundary layer is a fully developed turbulent boundary layer of zero pressure gradient.
2. The inlet flow temperature is uniform.
3. The effect of aerodynamic starting length is negligible.
4. The ice layer conducts heat one dimensionally near the leading platform edge.
5. The ice layer and flow are at steady state with each other.
6. At steady state the only two phases are air and ice.
7. The leading platform edge is far enough upstream of the vanes that there are no pressure gradients on the flow.

These assumptions are used together with a theory by LaFleur [23] to determine the boundary conditions required to run the iceformation experiment. The results of the heat exchanger design are the following:

1. The flat plate endwall is cooled using jet impingement of liquid nitrogen. A bank of 37 tubes with 12 one millimeter diameter holes each is used to jet impinge the area of each endwall (about 1' by 4'). Each tube has independent flow adjustment as does each passage area.
2. Based on an E^3 -Reynolds-number-similar air speed of 34 ft/sec, nitrogen jet impingement velocities are 8 cm/sec or higher.
3. At atmospheric pressure, the plates are at a temperature of 78°K or -319°F.

Two evaporators were fabricated and tested. The result is that each endwall can grow about 1 inch of ice within 1 hour. The uniformity of ice layer thickness is a function of adjustments in the evaporator tube flows.

4.3. Instrumentation

Vane airfoil are instrumented with surface static pressure taps over the three regions: leading edge, pressure side and suction sides. Collaboration within the research team provided agreement on 12 leading edge, 14 pressure side and 23 suction side pressure tap tubes. Leading edge pressure tap tube distribution is referenced symmetrical around the vane origin and straddles the stagnation point location predicted by flow calculations performed by General Electric. Suction and pressure side tap distributions are referenced from the edge of the leading edge tube arrangement. Pressure side taps are spaced 15/32" apart along the surface. Suction side taps are spaced 27/32" apart along the surface. The pressure tubing is drilled at different spanwise locations according to surface pressures and gradient contours predicted by the potential flow calculations. Endwall regions have higher number of pressure taps to resolve secondary flow effects on the surface. A 20 channel manometer bank will be used to indicate the leading edge pressure distribution, stagnation

point location, vane surface distributions of static pressure.

Five hole probe selection was based on two different geometries: parallel and perpendicular flow faces. The research group decided to employ perpendicular flow face geometry based upon reviewing the results of Dominy and Hodson [28]. The results of Reynolds number sensitivity suggest probes with pressure taps arranged perpendicular to the probe surface produce less sensitivity to changes in Reynolds number. Also when the probe is positioned at small or zero yaw angles, the dynamic pressure coefficients dependence upon Reynolds number is lower compared to the parallel faced geometry. Another influence of Reynolds number sensitivity is probe cone angle. Comparing three conical and perpendicular probes with different cone angles indicates that a 90 degree cone angle geometry should be used to minimize Reynolds number sensitivity.

The prototype of the Gas Turbine five hole probe design uses 0.20 mm O.D. (0.0078 in.) stainless steel tubing for each pressure port and silver soldered joints. A 1.2 mm O.D. (0.047 in) stainless steel tubing houses the five 0.20 mm tubes in a geometry similar to Ligrani et al [29]. Larger pressure port tubing provides support and a surface for sealing the probe in through the test section wall. Small Tygon tubing is used to connect the exit probe tubing to the pressure transducers of the data acquisition system.

Adequate pressure sensing requirements were discussed by the research group and found dependent upon the following: probe characteristics, wind-tunnel flow speed, boundary layer characteristics, pressure transducer full scale and resolution values. The range of the transducer was sized using the average maximum air speed expected at the exit of the vane row during performance tests. This exit velocity is roughly 27/10 of the inlet velocity (110 ft/sec) or 297 ft/sec. Assuming no losses, this yields a pressure differential of

$$\Delta h_e = \frac{1}{2} \frac{\rho_a}{\rho_w} \frac{1}{g} V^2 = 19.6 \text{ inches of water}$$

This is an approximate pressure differential based on a total pressure of atmospheric (no losses) and one of the four perimeter pressure taps facing nearly 90 degrees to the local flow.

The required pressure transducer resolution can be determined by specifying a desired performance difference between two geometries that must be measured. The research team agreed that a goal for endwall design is to reduce the total pressure losses by 10%. Vane/endwall performance is indicated by the mass averaged total pressure coefficient on an exit axial plane downstream of cascade

$$\bar{C}_{pT} = \frac{\int_0^{\text{span}} \int_0^{\text{pitch}} C_{pT} \rho_a u_x dS}{\int_0^{\text{span}} \int_0^{\text{pitch}} \rho_a u_x dS}$$

$$\text{where } C_{pT} = \frac{P_{T \text{ upstream}} - P_{Te}}{\frac{1}{2} \rho_a V^2}$$

The goal of 10% reduction in loss can be used with assumptions to generate what the desired transducer resolution should be. The first assumption is that the baseline model (model A) is similar the case of Langston et al [4] which indicates a downstream mass averaged total pressure loss coefficient of 0.3. The second assumption is that the iceform model is model B and has a pressure loss 10% lower than model A (the differential of performance is 0.03). The third assumption is that velocity profiles and mass flows of the two models are about the same and mass flow influences are removed from the mass averaged total pressure loss coefficients. The fourth assumption is that

the overall loss reduction is due to a uniform contribution of loss reduction from local points. Applying these assumptions yields the following equation for the loss reduction goal:

$$\Delta h_{TB-A} = h_{TB} - h_{TA} = \Delta C_{PT} \frac{1}{2} \frac{\rho_a}{\rho_w} \frac{1}{g} V^2$$

The inlet water differential is 2.7 inches of water and the goal of total pressure difference resolution is 0.03. Consequently the pressure differential that must be resolved between model A and B is 0.081 inches of water. For confidence in the difference and to insure no overlap in error bars of the model A and B performances, the error on the performance difference could be 50% of the difference or 0.040 inches of water. Therefore the allowable transducer error of each channel would be 0.020 inches of water. The 0.020 inch-of-water resolution is near the limit of available pressure transducers. An electronic, multiple channel, pressure scanner system manufactured by PRESSURE SYSTEMS was selected to meet a 10" range and the 0.020" resolution requirement.

4.4 Important Procedures

The procedure for conducting the experiment begins with aerodynamic balancing. The finite number of vanes in the cascade must simulate an infinite blade row. To do this the following conditions are met

1. The mass flow in each passage must be the same - this is measured using the leading edge pressure taps to locate the stagnation point. This is compared to flow calculations performed by General Electric for our model geometry.
2. The vane geometries must be the same and aligned properly - the vane fabrication and vane mounting is done on a four-times-scale.

These two conditions insure that each vane has the same loading, passage flow and endwall pressure distribution. This provides a computationally and experimentally documented starting point for the endwall design problem.

An initial ice layer is grown by sealing off the test section inlet with the gate and filling the test section with chilled water. The vanes are submerged along with the iceformation heat exchangers in nearly freezing water. Liquid nitrogen flow is turned on and an ice layer grows over both flat endwalls. Each endwall and passage can be independently controlled such that uniform ice layer (periodic) is achieved. Once an acceptable ice layer thickness and uniformity is established, the water is drained and the gate removed.

The use of iceformation in experimental design requires that the flow and thermal boundary conditions be controlled to model the actual operating environment. An additional goal is to specify the thermal control parameter such that energy dissipation is minimized. By minimizing energy dissipation, the viscous dissipation and the thermal dissipation are minimized together, not one or the other. This is a valid gas turbine design goal because the energy dissipation measures the effects of total pressure drop due to viscous dissipation and thermal energy dissipation due to heat transfer.

A theory and example outlined by LaFleur [23,30] show that given flow speed, the thermal parameter can be chosen to minimize energy dissipation. The linear model and a turbulent flow assumption were used to approximate the interface conditions at a point upstream of the turbine cascade. This model revealed two degrees of freedom for controlling the flat ice interfaces (flow speed and air temperature). It was found that the iceformation upstream of the vane row is not controlled by the minimum energy dissipation criterion because the inlet turbulent boundary layer is fully developed (the friction factor and Stanton number are constant over the ice surface). The minimum energy

dissipation criterion may be specified but will not constrain the unknowns. The remaining selection options can be listed for each variable in a menu of choices (U = flow speed, T = temperature and I = ice thickness).

Flow Speed:

- U1. Reynolds number similarity to the E^3 design conditions given by Timko
- U2. Mach number similarity to the E^3 design conditions given by Timko
- U3. Reynolds number similarity to a power generation application of the E^3
- U4. Mach number similarity to a power generation application of the E^3
- U5. Inversely from the iceformation theory after constraining or specifying the thermal parameter and ice thickness
- U6. Using the wind tunnel capabilities to attain a high dynamic head and magnitude of pressure difference measurements

Thermal parameter:

- T1. Specify liquid nitrogen temperature, ice interface temperature and room air temperature.
- T2. Inversely from the iceformation theory after constraining or specifying the flow speed and ice thickness

Ice Thickness:

- I1. Specify the ice thickness based on the mounting of the heat exchanger with the one inch plexiglas wall such that a smooth, flat wall for the air flow is created.
- I2. Specify the ice thickness to match the constraint of the inlet span of the E^3 second vane based on information in Timko.
- I3. Inversely from the iceformation theory after constraining or specifying the thermal parameter and flow speed

Once the experimental control is selected, the air flow is turned on and set to the selected inlet air speed. The air temperature is monitored

and controlled by air circulation through the laboratory. Over time the ice layer geometry will change from a flat endwall to one that is contoured by the flow. The ice thickness near the leading edge is monitored to measure when steady state is reached. Following verification of steady state, the ice contours are removed from the test section and cast using a double casting technique. The result is a plaster or plastic replica of the ice shape. These replicas are tested for performance and studied for geometric features. Performance testing will be based on exit flow measurements using the five-hole-probe and endwall pressure taps.

5. FUTURE WORK

Future work on the laboratory apparatus involves minor modifications and the integration of sub-system procedures.

Future work on the four-times-scale vanes includes polishing and measurement of the Clarkson vane geometry by General Electric. General Electric will produce a database from flow calculations of Clarkson's test model at different row angles of attack. This will provide a guide for adjusting the tailboards to attain vane row periodicity.

Iceformation design tests will be conducted for flow speeds in the range of 35-110 ft/sec. Iceformation design theory suggests air speeds in the range of 35 ft/sec while pressure measurement resolution calculations indicate that speeds over 90 ft/sec are necessary. Air temperature will be a constant throughout testing of air speed variations. Air temperature will be adjusted if the minimum ice thickness of 0.1 inches is violated. If thicker initial ice layers are needed, the evaporators will be remounted on a frame over the test section.

The five-hole-probe will be fabricated and calibrated at Clarkson. A calibration tunnel has been assembled and will be augmented by angular traverse stages for probe calibrations.

The computer-based pressure data acquisition system will be procured and assembled. Procedures for performance testing will be established at Clarkson prior to the performance testing at the General Electric R&D Center.

6. REFERENCES

1. LaFleur, R.S., "Evolutionary Design of an Endwall Junction," Ph.D. Thesis, University of Connecticut, 1988, pp. 254-262 and pp. 281-290.
2. Turner, J.R., "An Investigation of the End-wall Boundary Layer of a Turbine-Nozzle Cascade," Trans ASME, November 1957, pp. 1801-1806.
3. Hunter, I.H., "Endwall Boundary Layer Flows and Losses in an Axial Turbine Stage," Trans ASME, J. Engr. Power, 104, 1982, pp. 184-193.
4. Langston, L.S.; Nice, M.L.; Hooper, R.M., "Three-Dimensional Flow Within a Turbine Cascade Passage," Trans. ASME, J. Engr. Power, 99, January 1977, pp. 21-28.
5. Kopper, F.C.; Milano, R.; Vanco, M., "Experimental Investigation of Endwall Profiling in a Turbine Vane Cascade," AIAA J., Vol. 19, No. 8, August 1981, pp. 1033-1040.
6. Yamamoto, A.; Yanagi, R., "Production and Development of Secondary Flows and Losses Within a Three-Dimensional Turbine Stator Cascade," ASME Conf. 85-GT-20, Houston, Texas, March 18-21, 1985.
7. Hasan, M.A.Z.; Casarella, M.J.; Rood, E.P., 1986, "An Experimental Study of the Flow and Wall Pressure Field Around a Wing-Body Junction, J. Vibr. Acous. Stress and Reliability in Design, Vol. 108, pp. 308-314.
8. Devenport, W.J.; Simpson, R.L., "Flow Past a Wing-Body Junction - Experimental Evaluation of Turbulence Models," AIAA Journal, Vol. 30, No. 4, April 1992, pp. 873-881.
9. Devenport, W.J.; Simpson, R.L., "Time-dependent and time-averaged turbulence structure near the nose of a wing-body junction," J. Fluid Mechanics, Vol. 210, 1990, pp. 23-55.
10. Hoerner, S., Fluid Dynamic Drag, Hoerner Fluid Dynamics, Albuquerque, NM, 1965.
11. Sieverding, C., "Reduction of Secondary Losses In Turbines," Course Note 95, von Karman Institute for Fluid Dynamics, Rhode Saint Genese, Belgium, January, 1975.
12. Devenport, W.J.; Agarwal, N.K.; Dewitz, M.B.; Simpson, R.L.; Poddar, K., 1990, "Effects of a Fillet on the Flow Past a Wing-Body Junction," AIAA Journal, Vol. 28, No. 12, pp. 2017-2024.
13. Sung, C.H.; Yang, C.-I.; Kubendran, L.R., "Control of a Horseshoe Vortex Junction Flow Using a Fillet," Hydromachinery Performance Enhancement for Marine Applications Symposium, Newport, RI. Oct., pp. 13-20.
14. Prumper, H., "Application of Boundary Layer Fences in Turbomachinery," AGARD-AG-164, Boundary Layer Effects in Turbomachines, paper, II-3, pp. 311-331.
15. Camarata, F.J.; Dring, R., "Water Tunnel Study of Endwall Contouring in Turbine Vane Passages Conducted under EOS 730010-746," United Aircraft Research Laboratories, Report UAR-K123, East Hartford, CT.
16. Morris, A.W.H.; Hoare, R.G., "Secondary Loss Measurements in a Cascade of Turbine Blades with Meridional Wall Profiling," ASME paper 75-WA/GT-13, Houston Texas, November, 1975.

17. Boyle, R.J.; Rholik, H.E.; Goldman, L.J., "Analytic Investigation of Effect of End-Wall Contouring on Stator Performance," NASA Technical Paper 1943, Lewis Research Center, Cleveland, Ohio, 1981.
18. LaFleur, R.S., "Exploration of the Iceformation Method Applied to a Diffuser," ASME Trans, J. Fluids Engr., Sept. 1988, Vol. 110, pp. 244-250.
19. LaFleur, R.S.; Langston, L.S., "Drag Reduction of a Cylinder/Endwall Junction Using the Iceformation Method," ASME J. Fluids Engr., Vol. 115, 1993, pp.26-32.
20. Bowley, W.W.; Coogan, C.H., "Two-Dimensional Melting in a Channel," ASME Pub. 67-HT-75, 1967.
21. Carlson, F.M., "An Investigation of the Solidification of a Flowing Liquid on a Circular Cylinder in Crossflow and Its Effect on the Drag Coefficient," Ph.D. Thesis, University of Connecticut, 1975.
22. LaFleur, R.S., "Evolutionary Design Theory Using Dynamic Variation and Thermodynamic Selection," Res Eng Des (1991), Vol. 3, pp. 39-55.
23. LaFleur, R.S., "Evolution theory for optimal control of a Couette iceform design model," Int. J. Heat Mass Transfer, 1992, Vol. 35, No. 10, pp. 2617-2629.
24. LaFleur, R.S., "Selection Game Strategies for Iceformation Method Preprocessing Solutions of Flow and Thermal Parameters," Int. Comm. Heat Mass Transfer, Vol. 19, No. 5, 1992, pp. 607-618.
25. LaFleur, R.S., 1990, "A Basis for Iceformation Design," Proceedings 1990 ASME International Computers in Engineering Conference and Exposition, Boston, MA, pp. 393-401.
26. "First, Second, Third and Fourth Quarter Reports: Reduction of Turbine Endwall Total Pressure Loss and Heat Transfer using the Iceformation Design Method," Subcontract number 93-01-SR016, P.I. R.S. LaFleur, Clarkson University, Report to R. Allen, L. Golan and D. Fant, AGTSR, South Carolina Energy Research Center at Clemson University.
27. Timko, L.P. "NASA Report CR-168289 Energy Efficient Engine - High Pressure Turbine Component Test Performance Report," under contract NAS3-20643, 1989.
28. Dominy, R.G.; Hodson, H.P. "An Investigation of Factors Influencing the Calibration of Five Hole Probes for Three-Dimensional Flow Measurements," J. Turbomachinery, Vol. 115, 1993, pp. 513-519.
29. Ligrani, P.M.; Singer, B.A.; Baun, L.R., "Miniature Five-Hole Pressure Probe for Measurement of Three Mean Velocity Components in Low-Speed Flows," J. Phys. E: Sci. Instrum, Vol. 22, 1989, pp. 868-876.
30. LaFleur, R.S., "Example of the Couette iceform design model: flat plate iceformation," Int. J. Heat Mass Transfer, 1992, Vol. 35, No. 10, pp. 2631-2642.

VORTEX-GENERATOR INDUCED ENHANCED HEAT TRANSFER IN GAS TURBINE BLADE COOLANT CHANNELS WITH ROTATION

CONTRACT INFORMATION

Contract Number	93-01-SR015														
Contractor	Louisiana State University Mechanical Engineering Department Baton Rouge, LA 70803 Phone Number: 504-388-5792 Fax Number: 504-388-5924														
Head Principal Investigator	Sumanta Acharya														
Co-Principal Investigators	Tod A. Myrum Dimitris E. Nikitopoulos Joel Tohline														
Research Students	<table border="0"> <tr><td>Hibbs</td><td>Grad. Student</td></tr> <tr><td>Dani</td><td>Grad. Student</td></tr> <tr><td>Crassier</td><td>Grad. Student</td></tr> <tr><td>Pointel</td><td>Grad. Student</td></tr> <tr><td>Chen</td><td>Grad. Student</td></tr> <tr><td>Burns</td><td>Undergrad. Student</td></tr> <tr><td>Klein</td><td>Undergrad. Student</td></tr> </table>	Hibbs	Grad. Student	Dani	Grad. Student	Crassier	Grad. Student	Pointel	Grad. Student	Chen	Grad. Student	Burns	Undergrad. Student	Klein	Undergrad. Student
Hibbs	Grad. Student														
Dani	Grad. Student														
Crassier	Grad. Student														
Pointel	Grad. Student														
Chen	Grad. Student														
Burns	Undergrad. Student														
Klein	Undergrad. Student														
DOE Project Manager	Dr. Paul Micheli/Dr. Norm Holcombe														
Period of Performance	September 1, 1993 to August 31, 1994														
Schedule and Milestones															

FY94-FY96 Program Schedule

	S	O	N	D	J	F	M	A	M	J	J	A
Test Plan												
Fabrication												
Testing	94	94										
Experiments	94-6	94-6	94-6	94-6	94-6	94-6	94-6	94-6	94-6	94-6	94-6	94-6
Computations	94-6	94-6	94-6	94-6	94-6	94-6	94-6	94-6	94-6	94-6	94-6	94-6
Analysis	94-6	94-6	94-6	94-6	94-6	94-6	94-6	94-6	94-6	94-6	94-6	94-6

OBJECTIVES

The primary objective of the three-year, experimental/computational study is to develop and verify a novel technique for increasing the heat transfer in rotating, ribbed gas-turbine-blade cooling channels. This technique involves placing vortex generators in the vicinity of key rib elements so that the vortex street that forms behind the generators excites the shear layers behind the ribs at subharmonics of their most amplified frequency, thereby enhancing mixing and heat transfer.

In order to monitor the progress of the work, the study has been organized into a set of annual goals, subdivided into experimental and computational objectives. The goal of the experimental portion during the first year, is to demonstrate the viability of using vortex generators to increase the heat transfer in rotating ribbed channels.

This is being accomplished using the naphthalene-mass-transfer technique and the subsequent application of the heat/mass transfer analogy to obtain local Nusselt number results. In order to realize this objective, a sophisticated rotating test facility, complete with a data acquisition system, had to be designed, fabricated, and tested.

On the computational side, the first-year objective is to develop a computational model that can be used to predict local heat/mass transfer coefficients in a three-dimensional flow within a rotating ribbed duct with vortex generators. This involves the finite difference solution of the time-dependent Navier-Stokes, energy, and mass-concentration equations using the direct numerical simulation (DNS) technique.

The second and third year objectives of the experimental portion will be to identify the optimal vortex generator configuration using hot-wire-flow-spectra results and to make mass transfer measurements for the optimal configuration. Regarding the computational portion, the second

and third year objectives involve the development of a fully validated, computationally efficient computer model.

BACKGROUND INFORMATION

Heat transfer in channels with rib turbulators has been studied extensively. Studies have been reported on various rib types and configurations. These and other related studies have all been made for stationary channels. It is well known that in gas turbine blade applications, the centrifugal and Coriolis forces have a significant effect on the flow and transport processes, and that correlations developed for stationary channels are not directly applicable to a rotating channel. In recognition of this, a few studies have examined convective heat transfer in rotating smooth channels. A number of recent studies (Wagner *et al.*, 1991a, 1991b, 1991c, 1992) have reported measurements in rotating smooth and ribbed channels with radially inward and outward flows. However, in these studies, each heated wall of the channel consisted of only four heating elements and only *quasi-averaged* heat transfer results for each element could be measured. Therefore, details of the local interrib heat transfer were not obtained. Wagner *et al.* (1991a, 1991c) clearly demonstrated that the heat transfer in a smooth, rotating channel depends on a variety of parameters: the surface orientation relative to the direction of the flow, the Reynolds number, the rotation number, and a buoyancy parameter, defined as $(\Delta\rho/\rho) (\Omega D/U)^2 (R/D)$, where $\Delta\rho$ is the variation of the density, ρ , D is the duct hydraulic diameter, and R is mean radius of rotation. Due to the Coriolis and buoyancy forces, the heat transfer from the leading and trailing walls is high or low depending on whether the flow is radially outward or inward.

Although, the heat transfer in smooth, rotating channels has been examined to some extent, the heat transfer in rotating ribbed ducts

has received very limited attention. Wagner *et al.* (1991b, 1992) have reported *quasi-averaged* surface heat transfer data in a rotating duct with trips normal and inclined to the flow. They concluded that Coriolis and buoyancy effects must be considered and that the effects of rotation were different depending on the flow direction. However, only *quasi-averaged* data was obtained and no *local* heat transfer or temperature data were presented to delineate the effects of rotation on the flow structure and heat transfer in the separated and reattached regions.

In the majority of the computational studies, the flow is assumed to be incompressible, and therefore, the centrifugal buoyancy forces have played no role in the computations. Where turbulence effects have been considered, the two-equation k - ϵ turbulence model has been generally employed.

No numerical studies have successfully modeled turbulent flow and heat transfer in a ribbed duct that is rotating perpendicular to its axis. This physical problem requires the understanding of not only the effect of rotation on attached boundary layers but also the effect of rotation on the separated curved shear layers.

It is important to note that none of the numerical studies cited used a direct numerical simulation (DNS) approach for this problem. Rather, all of these studies solved a set of time-averaged equations and therefore were unable to predict the time-dependent vortex evolution and pairing processes that occur in the shear layer behind a rib. Since the proposed study is based on the manipulation and control of the shear layer vortices, it will be necessary to follow the formation and transport of these vortices. The time-averaged approach will be inadequate to predict the behavior of the various key dynamical processes. Thus, in the proposed computational work, the DNS approach is employed. To the co-PI's knowledge, this approach has not seen much use in gas turbine cooling applications.

PROJECT DESCRIPTION

In this project, vortex generators will be positioned in the vicinity of the ribs in a rotating ribbed channel. The rationale is to subharmonically excite the rib-shear-layer instabilities with the vortices shed from the vortex generators. Subharmonic forcing has been observed to increase vortex merging and mixing in shear layers. Moreover, Myrum and Acharya (1993) have used vortex generators to increase the average Nusselt number in stationary ribbed cooling channels by 21 percent. In this respect, this study seeks to extend the work of Myrum and Acharya to more realistic turbine blade cooling applications that are complicated by rotational effects.

This study includes both experimental and numerical simulations. The experimental work involves making local naphthalene-sublimation mass transfer measurements for radially inward and outward flows through rotating channels containing ribs (baseline) and ribs and vortex generators. Three-dimensional maps of the local Nusselt number distribution in the channels will be obtained via the heat/mass transfer analogy.

Hot-wire-based flow spectra measurements will be performed to determine the parameter values leading to the desired subharmonic excitation. Spectra measurements will be obtained for several geometrical parameter values by positioning the hot wire behind the vortex-generator-rib pair.

The purpose of the computational work is to complement and generalize the experimental work, with an ultimate objective of developing a computer model that can be implemented by the gas-turbine industry to design turbine blades. Numerical simulations will be used to reduce the number of flow spectra measurements required by determining the parameter values leading to subharmonic excitation. Moreover, the simulations will provide immediate feedback as to the impact of the subharmonic excitation on the heat/mass transfer. In this way, the values of

geometrical parameters can be selected to give the maximum cooling effect. Ultimately, these values will be experimentally validated using flow-spectra and mass transfer measurements. In addition, the computer model will be used to study the effect of buoyancy on the heat transfer rates; this cannot be accomplished using the naphthalene-sublimation technique.

The issues just put forth will now be detailed.

Rotating Test Facility

A rotating test facility, a sectioned side-view of which is shown in Figure 1, is used to simulate the flow in serpentine blade-cooling ducts. This facility has been designed to cover a wide range of rotation (Ro) and Reynolds (Re) numbers. The product of these parameters ($RoRe$) can be varied continuously between zero and 25,000. The highest value can be achieved at 550 RPM by pressurizing the flow to 150 psig.

The test section consists of two 14-in-long ducts of square cross-section (1-in x 1-in) connected by an 180° bend with a short radius of 0.65-in and a long radius of 1.65-in. The radius of rotation at the midpoint of the test section is 35 hydraulic diameters. The flow is directed radially outwards in one duct and radially inwards in the other after going around the bend. This design is a close model of a section of serpentine cooling duct in a gas turbine blade. A cross-stream section of the test section assembly is shown in Figure 3. A high-strength aluminum frame carries eight removable plates that form the inner walls of the test-section ducts, comprising two identical pairs for the top and bottom sides, two identical outer side plates and two identical internal side plates that slide into the frame from the downstream end. Two sets of plates have been manufactured. One set is intended to accommodate the hot-wire measurements. Each plate of the second set (shown in Figure 3), which is made of high strength aluminum, provides a reinforced frame to accommodate casting of a naphthalene wall layer

Figure 1: Sectioned side-view of the rotating experimental facility.

for the mass transfer measurements. Naphthalene cast into these frame-plates is surrounded by a metal rim of at least .020-in. This rim supports the naphthalene and provides a solid reference for the naphthalene surface-profile measurements. When the test section is assembled for naphthalene tests only a small fraction (2%) of the total surface area of the duct will not allow mass transfer to occur. Aluminum has been chosen for light-weight construction and because it does not absorb naphthalene vapor. Thus any exposed metal surfaces will be equivalent to adiabatic ones. The test section is enclosed in a cylindrical pressure vessel (see Figure 1) that can be pressurized to a pressure equal to that in the test section (150 psig maximum). The pressure vessel is screwed onto a

base and sealed via a single O-ring. The test section is bolted onto this base inside the pressure vessel. The base contains the settling chambers and flow straighteners for the conditioning of the flow prior to entering the test section. Air is supplied to the in-going settling chamber through a bulk-head tube-fitting on the outer side of the base, as shown in Figure 1, while the exhaust air leaves

Figure 2: Cross-stream section of the test section.

through a second fitting on the opposite side.

The base, test section and pressure vessel assembly is attached to the shaft of rotation by means of a 3-in steel arm and nest. The shaft is 3.5-in in diameter. A counter-balancing arm of similar geometry with adjustable balancing weights is attached on the shaft pointing in the opposite direction. The shaft is supported by two self-aligning bearings housed in the sides of a containment vessel (Figure 1). The shaft has two blind holes extending well inside the containment vessel on either end. Each blind hole ends in a radial port on the cylindrical surface of the shaft. These ports are connected to their counterparts on the test section base through Teflon tubing. The two ends of the shaft are fitted with two counter-threaded rotating seals. Thus, air is supplied to and removed from the test section from outside the containment vessel, through the shaft itself. Pressure and flow rate is measured at both the supply and exhaust side. One end of the shaft is allowed to protrude several inches outside the

containment vessel and is fitted with an array of slip-rings to bring electrical signals on and off the rotating frame.

The containment vessel is made of steel and is cylindrical in shape with two reinforced end-plates. It comprises two hinged pieces (base and cover) that come together above the shaft bearing-housings and are fitted with 2-inch-thick flanges. A gasket is used for sealing purposes between these flanges, so that vacuum can be drawn inside to reduce the drag on the rotating system. Spring-loaded round seals integrated with the bearings and their housings eliminate leaks along the perimeter of the shaft.

The rotating system is powered by a hydraulic power drive which provides torque with a weak dependence on the speed of rotation that can be varied continuously. A 14-hp electric motor is used to drive a variable-displacement pump which provides the pressure and flow rate driving an hydraulic motor. The speed of rotation can be varied continuously within the range of 40 to 650 RPM by varying the flow rate of the hydraulic fluid through a flow regulating valve. The power is transmitted from the hydraulic motor to the rotating facility main shaft by means of a 1:1 timing-chain-and-sprocket drive. Thus, the rotation speed of the main shaft can be set at will and to a value that is constant to within 1 RPM. The speed of rotation is continuously monitored with a tachometer. By combining a pair of three-way valves, the hydraulic power system also drives an actuator which opens and closes the cover of the containment vessel when the system is stationary.

Naphthaline Sublimation System

The heat transfer results in this study will be deduced from the naphthalene-sublimation mass transfer technique, and the subsequent application of the heat/mass transfer analogy. The naphthalene-sublimation technique involves making the inner surfaces of test section ducts (one for radially inward flow and one for radially

outward flow) out of naphthalene. Local mass transfer rates at these surfaces are then determined by measuring the amount of mass sublimated from different locations on the surfaces and dividing it by the length of a prescribed time interval.

The naphthalene surfaces are fabricated using a casting procedure. To ensure high-quality data, it is imperative that a reliable casting procedure be developed. It is essential that this procedure be capable of generating active surfaces which are smooth, flat, and free of contaminants.

The molds designed for the casting of the naphthalene surfaces consist of two pieces. One of the pieces is an aluminum frame-like structure which houses the naphthalene (the top, bottom and side plates of the test section in Figure 2), while the other is a stainless steel backing plate polished to a mirror finish. To make a casting using this configuration, the aluminum frame is placed on the backing plate. Subsequently, the frame is filled with molten naphthalene. After the naphthalene solidifies, the backing plate is removed, exposing a smooth naphthalene test surface surrounded by the frame.

Since there are two test-section ducts, a total of eight naphthalene test surfaces need to be cast, requiring eight aluminum frame-like structures and eight backing plates.

Another essential part of the naphthalene experiments is the development of a system for measuring the amount of naphthalene that sublimates from a number of x-y locations on each test surface. Conceptually, the measurement system consists of a motor driven x-y table which holds the naphthalene specimens. Overhanging the surface is a strut which holds a linear variable transformer (LVDT) gage head. During the measurement process, the LVDT is brought into contact with the naphthalene surface, and as the table is moved, the LVDT slides over the surface. The voltage output of the LVDT responds to changes in the surface contour of the naphthalene surface, thus providing a contour map of the surface.

The main requirements of the system are as follows:

1. A surface depth measurement repeatability to within 0.0002 in. Measurements of the depth before and after a mass transfer data run enable the determination of how much mass has sublimated per unit data run time, which in turn yields the mass transfer coefficient.
2. The ability to repeatedly return (to within $\pm$ 0.0003 in) to a given x-y location on the naphthalene surface. This is because surface measurements are taken at several x-y locations before and after a data run, so it is imperative that the before and after measurements are taken at the same x-y locations.
3. The ability to obtain a depth measurement at a variable number of preprogrammed x-y locations (> 3000) on each mass transfer surface within 60 minutes. Beyond 60 minutes, the extraneous sublimation loss becomes excessive.

The system consists of the following components:

1. LVDT - Lucas Schaevitz LBB 375TA-100
2. x-y table - Lintech M2-91212
3. Table Control System - CAMAD 2302 Twin Axis Stepper Motor Controller and Custom Software
4. Two American Precision stepper motor packages.

Hot-wire Spectra Measurements

The hot-wire measurements will be obtained using an independent acquisition system of small dimensions that will be mounted onto the rotating frame. The signal is digitized by the A/D converter of the unit (at a maximum of 85KHz) and transmitted in digital form through a serial or parallel connection to a PC on the stationary frame. The serial or parallel connection will be connected to the rotating frame via a slip-ring array. A study of the specifications of the slip-ring array and the communications standards for such connections indicate that this is possible. No noise

is going to be introduced to the data since they will be transmitted in digital form.

The Computations

A Direct Numerical Simulation (DNS) approach is used in the current work, in which, the time dependent conservation equations of the conservation of mass, momentum, energy and species concentration are solved. No modeling is required in this approach, and the solutions are expected to resolve the dynamical features of the flow. An existing three dimensional computer code, currently running on the LSU MasPar parallel machine, was initially adapted for the present purposes.

The calculations, in incompressible flows, are complicated by the fact that a pressure-Poisson equation needs to be solved for the pressure at each step. Three different approaches for the pressure-Poisson solver, using an (1) ADI (Alternating Direction Implicit) technique (2) a 2-D FFT and (3) a direct solver have been examined.

RESULTS

Mass Transfer

Since it is essential to collect all of the data from a given naphthalene test surface within one hour to eliminate extraneous mass transfer during the measurement process, an important test involved determining the maximum speed at which the x-y table, that holds the specimen, could be driven. It was found that if the speed was too high, the lead screws "chattered", and if the speed was too low, the stepping motors overheated (after about one hour) and automatically shut down, rendering the data run useless. The optimum speed was found, and at this speed, all four naphthalene surfaces could be traversed in less than one hour without chatter or overheating. The last test involved determining the repeatability of the apparatus, i.e., the ability to repeatedly return to a given x-y location and to repeatedly obtain the same depth measurement. This test involved taking data off the surface of the x-y table. The depth measurements (zero of the LVDT depth gage in this case) were within 0.0002 in, and the x-y location could be repeated to within ± 0.0003 in.

These are within the tolerances specified at the outset of the study.

After the aforementioned testing procedure was completed, a battery of mass transfer data runs was performed in order to see if the data acquisition procedure and the test rig could yield reliable results. This was accomplished by taking data from a smooth stationary duct, for which mass transfer results are available.

The data obtained from the initial data runs did not display the expected trends. Consequently, an intensive review of the data acquisition procedures was performed. It was found that the measurements of the naphthalene test surfaces obtained before and after the experimental runs were not properly referenced and the method of mounting of the naphthalene-bearing plates on the x-y table was inadequate and a source of uncertainty. As a consequence the plates were not oriented in exactly the same way on the table before and after each data run. To solve this problem, a special mounting device was designed and manufactured to ensure that the plates are mounted and secured with the same orientation before and after each run to within 0.002 in. This device comprises a base plate made out of tooling aluminum mounted on the x-y table. This base plate is equipped with a series of steel dowel pins and diamond pins arranged in pairs and located so that the pins match two of the bolt-holes of each naphthalene-bearing plate. These bolt-holes have been appropriately reamed to match, are located in opposite ends of the plate and have been marked for recognition. In addition, locking bolts have been installed that will allow controlled and repeatable locking of the plates on the tooling-aluminum-plate surface. This is achieved by torquing the bolts to the same level (5 in-lbf) before and after each data run. The measurements made before and after each data run are both being measured relative to a fixed origin on the aluminum frame surrounding the naphthalene surface. Since the frame is unaltered during the experiments, this provides a reliable reference.

Figure 4: Cross-stream Sherwood number distribution

Figure 3 shows plots of the dimensionless mass transfer rates (Sherwood numbers) as function of the streamwise dimensionless coordinate, for three different cross-stream locations on the trailing smooth surface of the outward flow duct of the test section. During this experiment two sides of the duct (trailing and leading) were active and the duct was stationary ($Ro=0$). The Reynolds number for this experiment was 15,358. The local Sherwood numbers are scaled by the average asymptotic value which is used as a reference. These results conform to expectations of high mass (heat) transfer coefficients at the leading edge followed by a decay to an asymptotic value. Our mass transfer results are also compared to heat transfer data from Han et al. (1986) and Hajek et al. (1991). The agreement is reasonable with differences showing up within the first two hydraulic diameters of the channel length. Since the flow in this region is seriously influenced by its conditioning prior to the entry into the test section such differences are not entirely unexpected. Comparison of the trailing side results with those of the leading side (not shown here) indicate a small degree of flow asymmetry at the inlet. This slight asymmetry is also evident when inspecting

cross-stream distributions of the Sherwood numbers on the trailing plate portrayed in Figure 4. Use of the appropriate screens between the settling chamber and the inlet of the test section will reduce the asymmetry in the flow and provide nearly identical mass transfer rates on both plates. The data from $y/D=0.2$ in Figure 3 are also indicative of the repeatability of the mass transfer measurements. The trends for the two experimental runs depicted are nearly identical to within the experimental accuracy.

Spectra Measurements

Spectra measurements behind the ribs will identify the response frequency that characterizes the separated shear layer behind the ribs. Measurements, so far, have been made in the stationary duct. Figure 5, shows the spectra and velocity-time behavior behind the rib placed in the free stream at a $Re_D = 32,133$ (Re_D is based on the duct hydraulic diameter D). The single peak at nearly 50Hz and the periodic velocity signal are characteristic of the Von-Karman street. With the rib on the surface, Figures 6 and 7 show the spectra and velocity at $Re_D = 37,147$ and $Re_D = 52,700$ for a rib height given by $h/D = 5/8$. Frequencies higher than 100Hz have been filtered out. There is a significant reduction in the natural frequency (single prominent peak in Figure 6 at a frequency nearly one fifth that of Figure 5), and the spectrum is more broad-band as the velocity signal is now contaminated with higher frequencies. At the higher Reynolds number the spectral character is modified further and several discrete frequency modes appear. These multiple and discrete frequencies appear to have harmonic proportions. The spectral evidence indicates that large-scale coherent structures characterized by discrete frequencies are present in the shear layer downstream of the wall-mounted rib. Manipulation of these large scale structures through the forcing mechanism currently investigated has the potential of effecting the desired transport (heat transfer).

Figure 5: Velocity signal and spectrum in the wake of a rib in the free stream ($Re_D=32,133$. $h/D=5/8$).

Figure 6: Velocity signal and behind a rib mounted on the wall surface ($Re_D=37,147$. $h/D=5/8$).

Figure 7: Velocity signal and behind a rib mounted on the wall surface ($Re_D=52,700$. $h/D=5/8$).

Computational

Two algorithms are currently being used. The first algorithm is based on a second order (both in the spatial and time directions), explicit, fractional step method. To get better accuracy, in a second algorithm, we are exploring higher order schemes. Currently, some simpler test problems are being studied with schemes ranging from second to fifth order schemes.

streamwise direction and Z is the spanwise direction. At an execution speed of 5.3 microseconds per zone we are able to integrate the fluid motion forward in time approximately 15,000 time steps for every 24 hours of execution time.

The accuracy of this DNS algorithm has been successfully tested via the simulation of a variety of different standard test problems. Of most relevance to the current project, however,

Figure 8: Contours of Z -Vorticity in a Plane Channel

The lower order algorithm, a DNS algorithm for simulating, in full three-dimensional generality, the flow through channels of high-Reynolds-number fluids, has been developed and successfully implemented, on a massively parallel computer. Specifically, the algorithm has been implemented on an 8,192-node MP-1 MasPar computer where it achieves better than 35% of the manufacturer's published peak performance rating, providing us with a simulation tool that runs at better than 300 MFlops. We are now routinely running simulations with a Cartesian (X,Y,Z) grid resolution of $(256,64,64)$ - where, X is the

are the tests that have been run on channel flows across a plate. At the lower wall, no-slip conditions are specified for velocity, and a zero gradient specified for pressure and temperature. Periodic conditions are specified in the spanwise direction, and symmetry conditions are imposed along the top boundary. At the inlet Dirichlet conditions are specified and include the perturbation quantities. Zero-gradient conditions are imposed along the outflow boundary. From uniform flow initial conditions, the code has correctly modeled the development of steady laminar flow along a plate that is more than 10

inches in length (see Figure 8) for fluids with Reynolds numbers ranging between $Re/L = 100 - 53,000 \text{ inch}^{-1}$. After introducing a broad spectrum of velocity perturbations at the inlet to the channel in the highest Reynolds number flows, we also successfully observed the onset of turbulence. Figure 8 displays the spanwise vorticity as a function of the X (horizontal) and Y (vertical) coordinates late in one such evolution. The vorticity plot, in particular, illustrates how initial low-level perturbations are amplified leading to the evolution of large scales. These large scales, transfer energy to the smaller scales, and further downstream, as the small scales are energized, the laminar flow makes its transition to turbulence. It is however important to note that consistent with our initial objectives we are able to predict the dynamical features of the flow

The simulation depicted in Figure 8 was specifically set up for comparison with the work of Rai and Moin (1991). Gas having a density of 1.293 kg m^{-3} , temperature 273°K , and kinematic viscosity of $1.266 \times 10^{-5} \text{ m}^2 \text{ s}^{-1}$ enters the channel from the right with a streamwise velocity of 26 m s^{-1} . It encounters a thin plate along the bottom face of the channel shortly after entering the channel (at zone 11 out of 256). Here, a 13 inch plate extends to the end of the simulated channel (with 244 grid zones along the plate, the streamwise resolution is $0.0533 \text{ inches/zone}$); gas is permitted to flow freely out of the channel on the left; reflection boundary conditions are imposed at the top channel face (the top and bottom faces are separated by 0.213 inches, so with 64 grid zones the vertical resolution is $0.00333 \text{ inches/zone}$); and in the spanwise direction, periodic boundary conditions are imposed every 0.524 inches (so with 64 grid zones, the Z resolution is $0.00818 \text{ inches/zone}$).

We are now preparing the code for simulations of smooth flow through a rotating channel duct, which can also be compared with previously published simulation as well as

experimental results. This will be followed by simulations of flows through ribbed channels, permitting us to make direct comparisons with our ongoing experimental program, as originally proposed.

Regarding the higher order algorithm, the primary effort during the last quarter of this year has been to test the incompressible DNS code. The pressure Poisson solver in this code, which represented one of the major challenges, has been developed and tested. This equation can be expressed as $\nabla \cdot (\nabla p^{n+1}) = (\nabla \cdot \tilde{u})/\Delta$ where $\tilde{u}$ is the velocity computed from the momentum equation in the previous step. Several approaches to the solution of the pressure Poisson equation were examined. In each case a Fast Fourier Transform (FFT) was applied in the spanwise (z) direction, so that the resulting problem was two-dimensional for each wave number in the Fourier space. Three different solution strategies were investigated to solve the two-dimensional pressure Poisson equation. These are: (1) the ADI difference scheme approach where each second order derivative is made implicit in alternate steps, (2) a combined spectral-difference scheme approach in which a two-dimensional FFT is applied leading to a tridiagonal system in the two-dimensional wave

Figure 9: Poisson Solver Comparisons

number space, and (3) and a direct solution approach with matrix splitting and inversion. Since the coefficient matrix of the pressure Poisson equation can be determined a priori and does not change with each step, the direct solver was found to be computationally most efficient (Figure 9).

The pressure Poisson solver was tested for many different geometry's including the ribbed duct, and the ribbed duct with a vortex generator. Currently the entire 3-dimensional code is being tested for a smooth channel and results will be compared with those of Kim et al. (1987). Calculations beyond a certain Reynolds number and time steps are exhibiting numerical instability, and this problem is currently being investigated.

FUTURE WORK

The effort to achieve the project goals will proceed as outlined previously. The objectives during the next reporting period with regard to the naphthalene part will be (1) to obtain more baseline data in a stationary ribbed duct which will be compared with the published mass transfer data, (2) to obtain stationary data with vortex generators in place, (3) to obtain data in a rotating smooth and ribbed duct, and (4) to obtain rotating data with vortex generators. Spectra measurements will also be continued in parallel. A parametric study will be carried out to determine the optimum geometrical configuration of the rib-vortex-generator pairs for maximum heat transfer enhancement in the rotating duct. Development of the Computational DNS code will be continued, validated, and used to provide further guidance for the experiments as well as to extend the parametric study range.

REFERENCES

Hajek, T. J., Wagner, J. H., Johnson, B. V., Higgins, A. W., and Stauber, G. D., 1991, Effects of Rotation on Coolant Passage Heat Transfer, NASA 4396, Vol 1.

Han, J. C., Park, J. S., Ibrahim, M. Y., 1986, Measurement of Heat Transfer and Pressure Drop in Rectangular Channels with Turbulence Promoters, NASA 4015.

Kim, J., Moin P., and Moser, R., 1987, Turbulence statistics in fully developed channel flow at low Reynolds number, *Journal of Fluid Mechanics*, Vol. 177, p.133.

Myrum, T. A., Qiu, X., and Acharya, S., 1993, "Heat Transfer Enhancement in a Ribbed Duct Using Vortex Generators", *International Journal of Heat Mass Transfer*, accepted.

Rai, M., and Moin, P., 1991, Direct Numerical Simulation of Transition and Turbulence in a Spatially Evolving Boundary Layer, AIAA-91-1607, AIAA 10th Computational Fluid Dynamics Conference, June 1991, Honolulu, HI.

Wagner, J. H., Johnson, B. V., and Kopper, F. C., 1991a, "Heat Transfer in Rotating Serpentine Passages with Smooth Walls", *ASME Journal of Turbomachinery*, Vol. 113, pp. 321-330.

Wagner, J. H., Johnson, B. V., Graziani, R. A., and Yeh, F.C., 1991b, "Heat Transfer in Rotating Serpentine Passages with Trips Normal to the Flow", ASME Paper No. 91-GT-265.

Wagner, J.H., Johnson, B.V. and Hajek, T.J., 1991c, "Heat Transfer in Rotating Passages with Smooth Walls and Radial Outward Flow", ASME Paper No. 89-GT-272 and *ASME J. Turbomachinery*, 113, 42-51.

Wagner, J. H., Johnson, B. V., Steuber, G. D., and Yeh, F. C., 1992, "Heat transfer in rotating serpentine passages with trips skewed to the flow", ASME Paper No. 92-GT-191.

Advanced Turbine Cooling, Heat Transfer, And Aerodynamic Studies

CONTRACT INFORMATION

Contract Number 93-01-SR011

Contractor Texas A&M University
Department of Mechanical Engineering
College Station, TX 77843-3123
Ph: (409) 845-3738
Fax: (409) 862-2418

Head Principal Investigator Prof. Je-Chin Han and Prof. Taher Schobeiri

**Co-Investigators /
Research Students** S. Dutta, S. Ekkad, A.B. Mehendale,
K. Pappu, and P. Chakka

DOE Project Manager Dr. Paul Micheli and Dr. Norman Holcombe

Period of Performance September 1, 1993 to August 31, 1994

Schedule and Milestones

1993-1994 Program Schedule

	S	O	N	D	J	F	M	A	M	J	J	A
Test Plan	-----											
Fabrication		-----										
Testing					-----							
Analysis							-----					

OBJECTIVES

The contractual work is in three parts: **Part I**- Effect of rotation on enhanced cooling passage heat transfer, **Part II**- effect of Thermal Barrier Coating (TBC) spallation on surface heat transfer, and **Part III**- effect of surface roughness and trailing edge ejection on turbine efficiency under unsteady flow conditions. Each section of this paper is divided into three parts to individually accommodate these three items.

Part I

The internal coolant passages of a typical turbine blade are often serpentine and contain rib turbulators. The objective of this study is to determine the effects of rotation on cooling passage heat transfer from a two-pass square duct model. In rotation the heat transfer from the trailing surface is enhanced and the leading surface heat transfer is suppressed for radially outward flow. However, the reverse is true for radially inward flow. The effects of two different rotating model orientations on their leading-trailing surface heat transfer are investigated. The combined effects of rotation model orientation and rib turbulators are also included.

Part II

The objective of this study is to investigate the effects of TBC spallation on surface heat transfer from turbine components. Currently under investigation is the complex problem of spallation on the combustor wall, on the leading edge and pressure and suction sides of the turbine blade. The combustor wall is simulated by a flat surface with 12 different spallation configurations. A transient liquid crystal image technique quantitatively found the detailed heat transfer coefficient distributions within and around the spallation region.

Part III

To date the published studies on turbine aerodynamics have largely concentrated on the measurement of blade surface properties such as the wall shear stress. The few available boundary layer measurements are not comprehensive enough to provide any conclusive evidence for the interpretation of the boundary layer transition and separation process and its direct impact on the profile loss, which is a critical parameter for blade design. The objective of this work is to provide essential detailed unsteady boundary layer information for evaluating the impact of the unsteady wake flow on the blade efficiency. Keeping in mind the vast amount of time involved in boundary layer measurement and the number of parameters needed to be studied, a new test facility has been developed to study efficiently and systematically the influence of periodic unsteady and highly turbulent flow on turbine cascade aerodynamics and heat transfer.

BACKGROUND INFORMATION

Part I

For the case of a multi-pass smooth channel, Wagner et al. (1991) reported that the rotating trailing surface heat transfer coefficient (Nu) for the radial outward flow increased up to 3.5 times the non-rotating fully developed circular pipe values (Nu_0). However, the leading surface heat transfer coefficient decreased to 40% of Nu_0 , but the rotating trailing surface heat transfer coefficient for the radial inward flow was 70% of Nu_0 and the leading surface heat transfer was enhanced to 1.2 Nu_0 . Rib roughened surfaces increase surface heat transfer. Johnson et al. (1994) concluded that the Nu can be increased to 5 Nu_0 by placing the ribs at a 45° angle to the bulk direction of the flow. Johnson et al. (1993) and Parsons et al. (1994) studied the effects of model orientation on rotating coolant passage heat transfer. They found that the model orientation under rotation condition has a significant effect on

coolant passage heat transfer. The present work focuses on the rotation effect on coolant passage heat transfer for two different model orientations with smooth and rib-roughened walls. This work is new and never published before.

Part II

In advanced gas turbine engines, the trend is toward using thermal barrier coating (TBC) on components guiding the host post-combustion gas. With the use of coal derived fuels, spallation of the TBC (coating loss due to thermal stress, erosion, or corrosion) has become a serious problem. Spallation causes an increase in local and downstream heat transfer coefficients. The effects of shape, size, and depth of a spallation on the local and downstream heat transfer coefficients for flow over a flat surface are studied using a transient liquid crystal image method. This work is new and relevant prior publications are not available.

Part III

The improvement of efficiency and performance of turbine and compressor stages has been a key issue in turbomachinery research and development for several decades. Considerable efforts have been made to develop blade profiles with optimum efficiency for a variety of specific stage-load and flow coefficients (λ , ϕ) encountered in turbine engine design technology. The optimization of blade geometry regarding the profile loss/efficiency, incidence, and deviation range by the industry and research institutions was performed using steady flow cascade experiments. Discrepancy in predicted and actual efficiencies was partially due to the unsteady nature prevalent in turbomachinery stage flow neglected in experiments.

To understand the effect of unsteady wake flow on turbine blade aerodynamics, Trost (1975) used a rotating cylinder cascade in front of an annular turbine cascade. The unsteady blade loss

coefficients of three different types of turbine cascades were compared with the steady measurements. Trost (1975) found that the steady data could not be used for turbine rotors, especially where the turbine blades were subjected to laminar separation. Later Schröder et al. (1992) underscored the significance of the influence of the periodic wake impingement on the blade aerodynamics. Referring to a series of experimental investigations, Schröder et al. (1992) concluded that measurements taken on individual airfoils or in cascades without a periodic wake provided no conclusive evidence for the interpretation of the boundary layer transition process. Arndt (1993) illustrated the profound influence of the wake induced rotor-stator and rotor-rotor interaction on the flow through downstream blade rows in a multistage Low-Pressure (LP) turbine. Arndt (1993) showed that the interaction resulted in strongly amplitude-modulated periodic and turbulent velocity fluctuations downstream of each rotor blade row. Hodson et al. (1994) performed experimental investigations on a multistage LP-turbine with surface mounted hot film anemometers. They suggested that the presence of wake influenced the transition in a separated flow region and concluded that a LP-blade design based on steady flow methods might not represent the achievable efficiency.

PROJECT DESCRIPTION

Part I

The experimental set up of Han et al. (1993) is modified to increase the mean rotating radius to better simulate the real engine conditions. A frequency controlled electric motor is the prime mover for the test rig. The shaft rotates at a constant rpm. Air acts as the working coolant. The air enters the rotating rig through a rotary seal at the bottom of the rig. The instrumentation of the rotating test section is coupled to stationary power transformers and a

Figure 1.1. Schematic of the Two-Pass Duct for Model A Orientation

data logger through a 100-channel slip ring unit.

The test apparatus can determine the regionally averaged heat transfer coefficients in the streamwise flow direction of the two-pass duct. The two-pass square duct is divided into 12 2.54 cm long copper sections as shown in Figure 1.1. Thin Teflon strips, machined along the periphery contact surface between copper sections, act as insulation to reduce heat conduction. The cross-section of the square duct is 1.17 cm by 1.17 cm. This geometry results in a hydraulic diameter,

D , of 1.17 cm. The length-to-hydraulic diameter ratio (L/D) of each pass of the test duct is 13 and the mean rotating radius is $55D$. The resistance heaters are cemented between the copper plate sections and the Teflon wall. Each of the nine walls is instrumented with independently variac transformer controlled heaters. The entire heated test section is insulated by Teflon. The copper heater sections are equipped with individual thermocouples. One thermocouple measures the inlet and two thermocouples measure the outlet bulk temperatures. An unheated Teflon entrance duct (not shown in Figure 1.1) has the same cross section as that of the test duct.

The local heat transfer coefficient (h) is calculated from the local net heat flux (q_{net}), the local wall temperature (T_w), and the bulk mean air temperature (T_b) difference as $h = q_{\text{net}} / (T_w - T_b)$. q_{net} is calculated by subtracting heat loss from the power supplied to the heaters. Heat loss is estimated by running a separate test on the same test configuration but without air flow in the duct. Characteristic heat loss lines are obtained for each heater element. T_w is obtained from the thermocouple reading of that heater section while T_b is calculated from the net heat input to the air.

The Reynolds number, Re ($Re = \rho V D / \mu$; ρ =density, V =mean velocity, μ =viscosity), varies in the order as 2500, 5000, 10000, and 25000. The test section rotates at 800 rpm resulting in rotation numbers, Ro ($Ro = \Omega D / V$; Ω =rotation speed), of 0.3, 0.15, 0.075, and 0.03. A higher Ro corresponds to a lower Re . The average wall temperature is maintained at 340K, which results in a density ratio, DR ($DR = 1 - \rho_w / \rho_{\text{in}} = 1 - T_{\text{in}} / T_w$), of 0.13. Note that the Ro and Re are calculated based on the properties of air at the inlet. The heat transfer results are presented as $Nu / Nu_o = hD / K \cdot 1 / (0.023 Re^{0.8} Pr^{0.4})$, where K is the thermal conductivity of air and Pr is the Prandtl number. The uncertainty of Nu / Nu_o is estimated as 8% for $Re > 5000$. However, the uncertainty for $Re = 2500$ can be as high as 20%.

Figure 2.1. Experimental Setup With Transient Liquid Crystal Image Technique

Part II

This work uses the data acquisition technique of thermo-chromatic liquid crystals. The liquid crystal data acquisition system consists of a chart recorder for recording the mainstream temperature and a real time image processing system. Figure 2.1 illustrates the data acquisition setup. The image processing hardware consists of an RGB camera, a monitor and a PC with CFG frame grabber board. The real time image processing software does a frame by frame analysis on the selected digitally scanned image. The time required for the color of each pixel to change to green in the captured image is stored. The time history of color change at each pixel is later analyzed to obtain the local heat transfer coefficient at each corresponding point on the test surface.

Figure 2.2 shows the test plate configuration. The test plate is 1.27 cm thick black plexiglass. The test section is 10.16 cm x 15.24 cm (4 in x 6 in) and 44.5 cm long. A smooth plate and 12 spallation plates are fabricated for testing. Figure 2.2 shows the simulated spallation geometries. Five circular shapes of diameters 1.27

Figure 2.2. Simulation of Spallation Configurations on a Flat Surface

cm (0.5 in), 2.54 cm (1.0 in), 3.81 cm (1.5 in), 5.08 cm (2.0 in), and 6.35 cm (2.5 in) with a depth of 0.51 cm (0.2 in) are tested for the effect of spallation size. Five different shapes: square, diamond, rectangle, vertical ellipse, and horizontal ellipse with a depth of 0.51 cm are tested for the effect of spallation shape. Three depths of 0.254 cm (0.1 in), 0.51 cm (0.2 in), and 0.76 cm (0.3 in) are tested on a circular spallation of diameter 3.81 cm (1.5 in) for the effect of depth. Table 2.1 shows the studied geometries.

A thin coating of encapsulated liquid crystals is sprayed on the test surface. The main flow is heated to a required temperature and the software is activated once the diverted (away from the test section) mainstream reaches the required temperature. The hot mainstream is then brought

Table 2.1. Flat Plate Spallation Geometries

Shape	Size (cm)	Depth (cm)
Circle	Diameter=1.27,2.54,5.08,6.35 Diameter=3.81	0.51 0.254,0.51,0.76
Square	side=3.81	0.51
Diamond	side=3.81	0.51
Rectangle (vertical)	5.08 x 1.91	0.51
Ellipse (vertical)	major axis=5.08 minor axis=1.91	0.51
Ellipse (horizontal)	major axis=5.08 minor axis=1.91	0.51

in to the test section. The time of switching the mainstream to the test section from the diverted path is synchronized with the time of initialization of the image processing software. Flow of hot air starts a transient heating of the cold surface and thermo-chromatic liquid crystals start to respond to the change in temperature by changing color with time. The image processing software records the time of color change at each pixel. Tests are performed at the mainstream Reynolds number Re_x of 10^5 , where x is the distance of the spallation upstream edge from the exit of nozzle. The mainstream turbulence is produced by placing a bi-planar square-bar grid with a 50% porosity at the nozzle exit.

The data analysis is based on the assumption that the flat plate test surface is a 1-D semi-infinite solid material. The basic 1-D transient conduction equation is $k \cdot \partial^2 T / \partial x^2 = \rho c_p \cdot \partial T / \partial t$, where the initial condition is $T = T_i$ at $t = 0$ and the boundary conditions are $-k \cdot \partial T / \partial x = h(T_w - T_m)$ at $x = 0$ and $T = T_i$ as $x \rightarrow \infty$. T_i is the initial temperature of the test surface, T_w is the color change temperature obtained from calibration, and T_m is the mainstream temperature. α , k are thermal diffusivity and conductivity of black plexiglass and t is the time of color change. Solving the one-dimensional transient equation with initial and boundary conditions, an inverse equation for the heat transfer coefficient (h) can be

Figure 3.1. Test Section: 1-Static Pressure Taps, 2-Boundary Layer Probe, 3-Wake Generating Rods, 4-Slots for Streamwise Movement of the Probe

Table 3.1. Specifications of Inlet Flow, Blade, Cascade, and Wake Generator Characteristics

Parameters	Values	Parameters	Values
Inlet velocity	$V_1 = 15 \text{ m/s}$	Inlet turbulence intensity	$Tu = 0.8 \%$
Blade inlet metal angle	$\alpha_1 = 0^\circ$	Blade exit metal angle	$\alpha_2 = 61.8^\circ$
Blade height	$L = 200 \text{ mm}$	Blade spacing	$S_B = 160 \text{ mm}$
Blade chord	$C = 281.8 \text{ mm}$	Blade Re-number	$Re_c = 563600$
Cascade solidity	$\sigma = 1.76$	Cascade flow coefficient	$\phi = 2.14$
Steady flow spacing	$S_R = \infty \text{ mm}$	Ω - parameter steady case	$\Omega = 0.000$
Cluster 1 rod spacing	$S_R = 160 \text{ mm}$	Ω - parameter for cluster 1	$\Omega = 0.822$
Cluster 2 rod spacing	$S_R = 80 \text{ mm}$	Ω - parameter for cluster 2	$\Omega = 1.644$
Cluster 3 rod spacing	$S_R = 40 \text{ mm}$	Ω - parameter for cluster 3	$\Omega = 3.288$
Rod diameter	$D_R = 5 \text{ mm}$	No. of rods in cluster 2	$n_R = 19$
No. of rods in cluster 1	$n_R = 11$	No. of rods in cluster 3	$n_R = 38$

obtained as $(T_w - T_i)/(T_m - T_i) = 1 - \exp(h^2 \alpha t / k^2) \cdot \text{erfc}(h\sqrt{(\alpha t)/k})$. This can be solved to obtain h at each pixel. This methodology is similar to that presented by Ireland and Jones (1985).

Part III

A large scale high subsonic research facility, capable of sequentially generating four different unsteady inlet flow conditions that lead to four different passing frequencies, structures, and freestream turbulence intensities, is developed to study the cascade aerodynamics and heat transfer. A large 110 kW centrifugal fan with a volume flow rate of 15 m^3 supplies air with a possible maximum mean velocity of 100 m/s at the test section inlet. There is a $5 \mu\text{m}$ fiber-glass filter upstream to the fan. The testing facility is designed to give uniform flow at the inlet of the test section. A diffuser located downstream of a straight pipe decelerates the flow before entering

a settling chamber. The settling chamber, made of four 900 mm length sections and $1500 \text{ mm} \times 900 \text{ mm}$ cross section, consists of five screens and one honeycomb flow straightener to control the flow uniformity and turbulence level. Downstream of the settling chamber is a nozzle, with an area ratio of 6.75, which establishes a smooth transition of the flow from the settling chamber to the wake generator. With this tunnel configuration, it is possible to achieve a constant mean fluctuation velocity over a wide velocity range, and a turbulent intensity of $Tu = 0.7\%$ at the exit of nozzle.

The motion of a wake generator (see Figure 3.1) simulates two-dimensional periodic unsteady inlet flow, with a series of cylindrical rods attached to two parallel operating timing belts driven by an electric motor. The belts are 5000-mm long and span five shaft-pulleys arranged around the cascade test section. The diameter (2-10 mm) and number of rods can vary to generate different wake widths and spacings. Wakes from the rods simulate the wakes from the trailing edge of rotor blades. The belt-pulley system is driven by a 10 kW electric motor. A frequency controller controls the belt speed to a maximum of 25 m/s . A fiber-optic sensor monitors the wake passing frequency. The sensor also serves as the triggering mechanism for data transfer and its initialization, which is required for ensemble-averaging.

Three rod clusters with constant diameter and spacing (see Table 3.1) sequentially measure the effect of three different spacings at a single boundary layer point. The data analysis program separates the buffer zones between data clusters and evaluates the data pertaining to each cluster. The cascade test section, located downstream of the wake generator, can integrate up to 7 blades with a height of 200 mm and a maximum chord of 250 mm. For boundary layer investigations, five NASA turbine blades are implemented, whose geometry is described in the NASA report by Schobeiri et al. (1991). One blade is specially manufactured and instrumented with 40 static pressure taps distributed on the suction and

pressure surfaces. This instrumented blade connects to a scanivalve for the measurement of surface pressure distribution. A computer controlled traversing system with feedback from an optical encoder provides accurate probe positioning with a resolution of 0.0025 mm. Boundary layer investigations specifically requires this high resolution where the measurement of the laminar sublayer is of particular interest. A single hot-wire probe measures the boundary layer flow.

A 486-personal computer with a 16 channel, 12-bit analog-digital (A/D) board controls the data acquisition system. Mean velocities and turbulent fluctuations are obtained using a commercial 3-channel (TSI) constant temperature hot-wire anemometer system. The wake generator speed and passing frequency signals of the rods are transmitted by a fiber-optic trigger sensor. The passage signals of the rods are detected by the sensor using silver-coated reflective paint on one of the belts.

To ensure a high level of accuracy, the calibration method and the facility described in John and Schobeiri (1993) was used for all hot-wire calibrations. The instantaneous velocity components are calculated from the temperature compensated instantaneous voltages using the calibration coefficients. The instantaneous velocity components are then decomposed into a time-averaged mean and turbulent fluctuation. A sampling rate of 1000 Hz is used in the steady flow investigation. For unsteady flow, 2048 samples are taken for each of 100 revolutions of the wake generator, rotating at 7 m/s, at each boundary layer position. Variation of the number of samples per revolution and total number of revolutions are performed to determine the optimum settings for convergence of the ensemble-averaging.

RESULTS

Part I

Effect of Rotation for Smooth and Rib-Roughened Duct. Figure 1.2 shows the

Figure 1.2. Schematic of the Secondary Flows Due to Rotation and Rib Orientation in the Two Passes of the Square Duct With Different Model Orientations

conceptual secondary flow vortex formation due to rotation and due to rib orientation for both Model A and Model B. The secondary flow by rotation is mostly on the bulk flow region, whereas the rib effect is expected to be more near the mounted surfaces. Note that the secondary flow in the second pass (II) is influenced by the 180° turn region. For simplicity Figure 1.2 does not include the effects of the turn.

Figures 1.3 and 1.4 show the regionally averaged Nusselt number ratios (Nu/Nu_0) at different duct locations. The Nu/Nu_0 is the ratio of heat transfer with rotation to the smooth fully developed tube data without rotation. Heat transfer results from the leading and trailing walls, as shown in Figure 1.2, are presented. Figure 1.3 is

Figure 1.3. Regionally Averaged Nusselt Numbers at Different Locations of the Square Duct for Model A

for model A and Figure 1.4 is for model B. Inclusion of ribs increase the heat transfer. Rotation enhances heat transfer from the first pass trailing and second pass leading surfaces, which are called unstabilized surfaces due to thin cooler boundary layer and enhancement of turbulence with rotation. Whereas, heat transfer decreases from the first pass leading and second pass trailing surfaces, which are stabilized surfaces due to thick warmer boundary layer with rotation. Higher rotation number shows larger difference between the trailing and leading surface heat transfer results than that for a lower rotation number.

Figure 1.3 shows the heat transfer results for smooth and ribbed walls for model A. Rotating

Figure 1.4. Regionally Averaged Nusselt Numbers at Different Locations of the Square Duct for Model B

results are compared with non-rotating stationary heat transfer results. Re of 5000 and 25000 with Ro of 0.15 and 0.03 results are shown. The rotating and stationary results at lower Ro are not significantly different. The unstabilized surfaces show an increase in Nu/Nu_0 over the stationary results at higher Ro (Ro=0.15). Heat transfer from the stabilized surfaces is lower with rotation than that at stationary case. Note that the ribbed trailing surface of the second pass loses much of the heat transfer capabilities at Ro=0.15, and Nu/Nu_0 from that surface is relatively low.

Figure 1.4 shows the effect of rotation on heat transfer for model orientation B. Note that the effect of 180° turn is not symmetric for leading and trailing walls at this model

Figure 1.5. Effect of Rotation Number on Nusselt Number Ratio at Selected Locations of the Duct

orientation. The stationary results of $Re=2500$ with smooth walls show that the stabilized surfaces (1st pass leading and 2nd pass trailing) have lower Nu/Nu_0 compared to the unstabilized surfaces (1st pass trailing and 2nd pass leading). Heat transfer from unstabilized surfaces increases and that from stabilized surfaces decreases with rotation. The ribbed results show characteristic increase and decrease in Nu/Nu_0 at unstabilized surfaces. These nonuniform Nu distributions are due to the interaction of the secondary flows by rotation and rib orientation.

Effect of Rotation Number and Model Orientation. Figure 1.5 shows the effect of rotation number and model orientation on heat transfer. For smooth surface results the heat

transfer from unstabilized surfaces (1st pass trailing and 2nd pass leading) increase and from stabilized surfaces (1st pass leading and 2nd pass trailing) decrease with increasing rotation number (Ro). Heat transfer from ribbed walls are, as expected, higher than the smooth wall results. Note that the dependence of heat transfer characteristics on Ro from the stabilized surfaces with ribs are different from those with smooth walls. Heat transfer from ribbed stabilized walls increases for smaller Ro and then decreases at higher Ro as opposed to a monotonic decrease in stabilized smooth surfaces. Interaction of rotation with secondary flow from ribs and flow separation near ribs are suspected to be the cause for an increase in heat transfer from the stabilized surfaces at low Ro . Heat transfer from the unstabilized surfaces with ribs behave similar to that for the smooth walls (but with higher Nu/Nu_0 values). That is, heat transfer increases as Ro increases.

The ribbed unstabilized surfaces show a sharp increase in Nu/Nu_0 with an increase in Ro . Whereas the smooth wall results show a gradual increase in Nu/Nu_0 with Ro . The difference in Nu/Nu_0 between stabilized and unstabilized surfaces are more for ribbed walls than those with smooth walls at $Ro > 0.1$. At $Ro=0.3$ the ribbed surfaces of the stabilized surfaces lose their heat transfer effectiveness significantly and become comparable to smooth walls, specially in the second pass. It can be concluded from the results that the rotation enhances heat transfer performance from the unstabilized surfaces (1st pass trailing and 2nd pass leading) and decrease heat transfer from the stabilized surfaces (1st pass leading and 2nd pass trailing). Inclusion of ribs enhance heat transfer. Heat transfer enhancement (Nu/Nu_0) of rotating unstabilized ribbed surfaces are much higher than stationary ribbed or rotating smooth surfaces. However, the heat transfer enhancement of rotating stabilized ribbed surfaces can be lower than its non-rotating values at higher rotation conditions. Model B provides higher heat transfer enhancement (Nu/Nu_0) for 1st pass

Figure 2.3. Effect of Spallation Depth on Heat Transfer Enhancement

trailing and 2nd pass leading ribbed surfaces than that of Model A.

Part II

The mainstream flow is tested for uniformity by measuring velocity profiles parallel and normal to the test plate. The centerline mainstream flow velocity (U_∞) is measured to be 7.4 m/s. The turbulent boundary layer velocity profile is measured at the upstream edge of the spallation location. The boundary layer thickness (δ) is 1.02 cm, the displacement thickness (δ^*) is 0.127 cm, the momentum thickness (θ) is 0.096 cm, and the momentum thickness Reynolds

number ($Re_\theta = U_\infty \theta / \nu$) is 447. The centerline mainstream turbulence intensity at the upstream edge of the spallation is 8.5%.

The results are presented as a ratio of heat transfer coefficients. The heat transfer coefficient with spallation (h) is normalized with heat transfer coefficient on a smooth surface (h_0) to obtain heat transfer enhancement due to the simulated spallation. This heat transfer ratio (h/h_0) is presented for the effect of spallation depth, spallation size, and spallation shape.

Effect of Spallation Depth. The effect of depth (d) is tested for a circular spallation of diameter (D) 3.81 cm. The depths tested are 0.254 cm, 0.51 cm, and 0.76 cm. The corresponding depth-to-momentum thickness (d/θ) ratios are 2.65, 5.3, and 7.92 and the corresponding diameter-to-depth (D/d) ratios are 15, 7.5, and 5. Figure 2.3 presents the heat transfer ratio contours for the three spallation depths. As the depth of the spallation hole increases, the heat transfer enhancement increases downstream of the spallation. Due to the increasing depth, the flow is more turbulent near the downstream leading edge of the spallation hole. Increasing depth increases separation at the upstream leading edge of spallation causing higher heat transfer at the downstream edge inside the spallation hole. Near the upstream edge of the spallation hole, heat transfer is low due to recirculation caused by the separation.

Effect of Spallation Size. The effect of size (D) is tested on a circular spallation of increasing diameters of 1.27 cm, 2.54 cm, 3.81 cm, 5.08 cm, and 6.35 cm. The depth is 0.51 cm for all the spallation sizes. The corresponding diameter-to-depth (D/d) ratios for the spallation sizes are 2.5, 5.0, 7.5, 10, and 12.5. The d/θ value is 5.3 for all the sizes. Figure 2.4 presents the heat transfer ratio contours for different spallation sizes. The effect of the spallation on heat transfer is typically one-diameter downstream of the spallation. With increasing size of spallation, the heat transfer ratio downstream of spallation increases. The heat

Figure 2.4. Effect of Spallation Size on Heat Transfer Enhancement

Figure 2.5. Effect of Spallation Shape on Heat Transfer Enhancement

transfer enhancement immediately downstream of the smallest spallation hole ($D/d=2.5$) is as high as 2.2 and as high as 4.5 for the largest spallation hole ($D/d=12.5$). For small spallation, the mainstream flow is similar to flow over a cavity. The flow trapped inside the spallation is recirculating thus decreasing heat transfer. Flow over large spallation sizes is similar to flow over a back-step followed by a forward-step at the downstream edge of the spallation hole. The mainstream flow separates at the upstream edge and reattaches inside the spallation hole and is tripped by the downstream edge of the spallation. Heat transfer is greatly enhanced at the spallation downstream (after the spallation hole) due to increased turbulence from the tripping of the boundary layer. Heat transfer inside the spallation near the upstream edge is low ($h/h_0=0.8$) due to the recirculation region created by the separation of mainstream flow. Heat transfer inside the spallation is enhanced due to reattachment of the mainstream flow.

Effect of Spallation Shape. The effect of spallation shape is studied for five different shapes other than the circle. These five shapes are a square, diamond, rectangle, a vertical ellipse, and a horizontal ellipse. The depth (d) for each of the five shapes is 0.51 cm. The corresponding depth-to-momentum thickness (d/θ) ratio is 5.3. The heat transfer ratio contours for all the shapes are presented in Fig. 2.5.

The effect of the square spallation hole is similar to a back-step and forward-step combination. Heat transfer enhancement immediately downstream of the spallation hole as high as 3.5. The effect of spallation on heat transfer is felt one-side distance downstream of the spallation hole. Heat transfer within the spallation is also high due to reattachment of the mainstream following separation at the upstream edge of the spallation.

The downstream sides of the diamond spallation trip the mainstream increasing heat transfer along the sides. Heat transfer is lower

near the upstream corner due to separation of the flow along the edges. Heat transfer within the spallation is not as high as that for the square shape.

The rectangular spallation hole is similar to the square shape. The heat transfer enhancement is not very high as the width of the spallation is small in the mainstream flow direction. The flow is similar to flow over a rectangular cavity. The effect of the spallation hole on heat transfer is felt a distance of one-width of the rectangle. Heat transfer enhancement within the spallation is lower than the square and diamond shapes.

Horizontal ellipse spallation produces high heating on the downstream edges. There is a reattachment region within the spallation hole. Heat transfer enhancement extends to a distance of one major axis length downstream of the spallation hole. The mainstream flow is tripped at the downstream edge of the spallation causing lower heat transfer coefficient inside the edge, but higher heat transfer coefficient outside the edge.

Heat transfer enhancement for a vertical ellipse is the lowest compared to the other 4 shapes studied. Mainstream flow is similar to the flow over a elliptical cavity. There is a small reattachment region within the spallation near the downstream edge. Heat transfer enhancement is felt one minor axis length downstream of spallation.

Part III

Figure 3.2 shows the time averaged pressure coefficient (C_p) along the suction and pressure surfaces. This pressure distribution indicates that the flow on both surfaces, except for a short distance around $s/s_0 \approx 0.1$, is subjected to a negative pressure gradient until $s/s_0 \approx 0.25$ is reached. Beyond this point, the pressure gradient on the pressure surface remains negative, while on the suction surface, positive pressure gradient prevails.

The integral parameters such as momentum thickness (δ_2) and shape factor (H_{12}) are of

Figure 3.2 Nondimensional Pressure Distribution as a Function of Streamwise Position at $Re_c = 563600$

particular interest to a turbine designer, since they provide an accurate first estimation about the quality of the designed blade. The ensemble-averaged distributions of the momentum deficiency thickness and shape factor on pressure surface for three Ω -values [$\Omega = (\sigma S_B)/(\phi S_R)$] at $s/s_0 = 0.471$ are presented in Figure 3.3. The period τ represents one wake-passing period that corresponds to the individual Ω -value under investigation. Pronounced time dependent behavior of the momentum thickness is visible. However, the time dependent distribution of the shape factor approaches a constant low value with increasing Ω -value. A similar tendency is observed on the suction surface. Other information regarding the overall evaluation of turbine blade is provided by the time averaged momentum thickness, shown in Figure 3.4 for pressure and suction surfaces. Considering the steady case as the reference case, the unsteady flow operation has caused a shift of the momentum thickness to higher values. This clearly shows that the steady state data cannot be transferred to the unsteady turbine design technology without modifications.

Figures 3.5 and 3.6 show The temporal-spatial distribution of the ensemble - averaged

Figure 3.3. Ensemble-Averaged (a) Momentum Thickness and (b) Shape Factor on Pressure Surface at $s/s_0 = 0.471$

turbulence intensity for a lateral position of $y = 2.5$ mm on both the pressure and suction surfaces for three different Ω -values. As shown in Figures 3.5(a,b), the boundary layer is periodically disturbed by the wakes that cause high turbulence intensity cores and extended becalmed regions. These regions are observed between the leading edge and the streamwise position $s/s_0 = 0.5$. The existence of becalmed regions was discussed in detail by Pfeil et al. (1983) and thoroughly measured by Orth (1992). With increasing Ω value, the extent of becalmed regions diminished rapidly with no becalmed regions present for the

Figure 3.4. Time-Averaged Momentum Thickness Distribution on (a) Pressure and (b) Suction Surfaces

case of $\Omega = 3.288$ (Figure 3.5(c)). These observations together with those from time-averaged contour plots (not presented) reveal that increasing wake-passing frequency, i.e., Ω , shifts the onset of transition towards the leading edge. Similar interpretations can be drawn from Figures 3.6(a,b,c) that show temporal-spatial distribution of ensemble-averaged turbulence intensity on the suction surface.

Figure 3.5. Ensemble-Averaged Turbulence Intensity in Temporal-Spatial Domain on Pressure Surface at $y = 2.5$ mm for (a) $\Omega = 1.644$, and (c) $\Omega = 3.288$

Figure 3.6. Ensemble-Averaged Turbulence Intensity in Temporal-Spatial Domain on Suction Surface at $y = 2.5$ mm for (a) $\Omega = 0.822$, (b) $\Omega = 1.644$, and (c) $\Omega = 3.288$

FUTURE WORK

Part I

This work continues to optimize the combined effects of cooling model orientation and rib turbulator configuration on rotating leading and trailing surface heat transfer enhancement. The work for the second year of the project will include the rotation impingement heat transfer for a two-pass test model.

Part II

The effect of spallation on surface heat transfer enhancement continues for flow over a flat surface and for flow across a cylinder by varying mainstream Reynolds number and turbulence conditions. The results will be correlated to provide a useful data base for turbine designers. The work for the second year will include the effect of spallation on film cooling effectiveness distributions.

Part III

Investigating the combined effects of blade aerodynamics and cooling air ejection in particular the trailing edge ejection on blade efficiency and performance, and determining the optimum geometric and aerodynamic configuration of the ejection holes and the optimum ejection mass flow ratio will be carried out during the second year. The design preparations for the instrumented blades have already been started. In addition to this, farther theoretical evaluation of the unsteady boundary layer measurement is being carried out.

REFERENCES

Part I

Han, J.C., Zhang, Y.M., and Kalkuehler, K., 1993, "Uneven Wall Temperature Effect on Local Heat Transfer in a Rotating Two-Pass Square Channel

with Smooth Walls," *ASME Journal of Heat Transfer*, Vol. 115, No. 4, pp. 912-920.

Johnson, B.V., Wagner, J.H., Steuber, G.D., and Yeh, F.C., 1993, "Heat Transfer in Rotating Serpentine Passages with Selected Model Orientations for Smooth or Skewed Trip Walls," ASME Paper No. 93-GT-305.

Johnson, B.V., Wagner, J.H., Steuber, G.D., and Yeh, F.C., 1994, "Heat Transfer in Rotating Serpentine Passages With Trips Skewed to the Flow," *ASME Journal of Turbomachinery*, Vol. 116, No. 1, pp. 113-123.

Parsons, J.A., Han, J.C., and Zhang, Y.M., 1994, "Effects of Model Orientation and Wall Heating Condition on Local Heat Transfer in a Rotating Two-Pass Square Channel With Rib Turbulators," accepted for publication in the *International Journal of Heat and Mass Transfer*.

Wagner, J.H., Johnson, B.V., and Kooper, F.C., 1991, "Heat Transfer in Rotating Serpentine Passages With Smooth Walls," *ASME Journal of Turbomachinery*, Vol. 113, pp. 321-330.

Part II

Ireland, P.T., and Jones, T.V., 1985, "The Measurement of Local Heat Transfer Coefficients in Blade Cooling Geometries," *AGARD Conference on Heat Transfer and Cooling in Gas Turbines*, cp 390, paper 28, Bergen.

Part III

Arndt, N., 1993, "Blade Row Interaction in a Multistage Low Pressure Turbine," *ASME Journal of Turbomachinery*, Vol. 115, pp. 137-146.

Hodson, H. P., Huntsman, I., Steele, A., 1994, "An Investigation of boundary layer development in Multistage LP-Turbine," *ASME Journal of*

Turbomachinery, Vol. 116, pp. 375-383.

John, J., Schobeiri, T., 1993, "A Simple and Accurate Method for Calibrating X-Probes," Being printed in the ASME Transactions, *Journal of Engineering Fluid Engineering*.

Orth, U., 1992, "Unsteady Boundary Layer Transition in Flow Periodically Disturbed by Wakes," ASME Paper 92-GT-283.

Pfeil, H., Herbst, R., Schröder, T., 1983, "Investigation of the Laminar Turbulent Transition of Boundary Layers Disturbed by Wakes," *ASME Journal of Engineering for Power*, Vol. 105, pp. 130-137.

Schobeiri, M. T., McFarland, E., Yeh, F., 1991, "Aerodynamics and Heat Transfer Investigations on a High Reynolds Number Turbine Cascade," NASA TM-103260.

Schröder, T., Baier, R.D., Broichhausen, K., 1992, "Laminar Flow and Unsteady Phenomena in Boundary layers of Turbomachinery Blades," DGLR-Bericht 92-06, pp. 201-211.

Trost, N., 1975, "Einfluß der Zuströmturbulenz auf die Strömung in Axialgittern," Dissertation D-17, Technische Hochschule Darmstadt, Germany.

Emissions Models for Lean Premixed Turbines

CONTRACT INFORMATION

Cooperative Agreement DE-FC21-92MC29061

Contractor Vanderbilt University
Box 1592, Station B
Nashville, TN 37235-1592
(615) 343-6214
(615) 343-6687 FAX

Head Principal Investigator Professor A.M. Mellor, Vanderbilt University

Co-Investigators/ Research Students Professor P.C. Malte, University of Washington
Professor R.W. Pitz, Vanderbilt University
Dr. R.J. Roby, Hughes Associates, Inc.

T.D. Magruder, Vanderbilt University
J.P. McDonald, Vanderbilt University
S.P. Nandula, Vanderbilt University
D.G. Nicol, University of Washington
P.A. Skaggs, Vanderbilt University
R.C. Steele, University of Washington
J. Tonouchi, University of Washington

DOE Project Manager Dr. Paul Micheli/Dr. Norm Holcombe

Period of Performance September 01, 1993 to August 31, 1996

Program Schedule

Task/	Year 1, 9/1/93-8/31/94				Year 2, 9/1/94-8/31/95				Year 3, 9/1/95-8/31/96			
Subtask	Q1	Q2	Q3	Q4	Q1	Q2	Q3	Q4	Q1	Q2	Q3	Q4
1. Combustor Modeling												
1.1	—	METC Mods	—	METC Tests	—	METC Mods	—	METC Tests	—	METC Mods	—	METC Tests
1.2	Design, predict METC flameholder tests			METC Tests	Design, predict METC tests for Comb A			METC Tests	Design, predict METC tests for Comb B			METC Tests
1.3	Bragg reactor model for lab flame- holders; develop reduced mechanisms				Zonal model for Comb A				Zonal model for Comb B			
2. NO _x and CO Modeling												
2.1	UV Raman/LIPF Tests flame #4			Sandia Tests	UV Raman/LIPF/CO LIF tests, flames #1, 2			Sandia Tests	UV Raman/LIPF/CO LIF tests, flames #3, 5			Sandia Tests
2.2	—	—	—	—	Size, arrange reactors for refined Zonal model				Develop CD model; incorporate into refined Zonal model			

OBJECTIVES

The present study is a three year effort on the application and refinement of engineering models for the prediction of NO_x and CO emissions of lean, premixed (LP) combustion turbines. Although LP combustion turbines were introduced into service in the late 1980s/early 1990s, and are now offered by most of the nation's gas turbine engine manufacturers for either electrical generation or mechanical drive service, research/development is required to reduce the NO_x emissions from the levels of the current technology (i.e., 25 to 40 ppmvd adjusted to 15% O_2) to the goal of under 10 ppmvd NO_x , while maintaining the very low CO emissions inherent in gas turbine engine combustion. The present study is contributing to this goal by refining engineering models for LP combustion, and by applying this modeling.

Two types of engineering models are considered. The first is chemical reactor modeling (CRM), used in the gas turbine industry to understand the formation and emission of NO_x and CO. For example, CRM can be used for screening chemical reactions and kinetic mechanisms, for developing the simplest possible (reduced) mechanism which is adequate for the combustion process being modeled, and for evaluating the dependencies of the NO_x and CO emissions on flame temperature (T_{ad} or T_ϕ), inlet temperature (T_{in}), pressure (p_{in} or P), and residence time (t_{res}). Chemical reactor modeling is relatively easy-to-use, engineering modeling, which can be applied efficiently to study the effects of combustor geometry and operation on the NO_x and CO emissions.

Further development and validation of characteristic time models (CTMs) (Mellor, 1976) for NO_x and CO emissions from dry, low NO_x combustion turbines is another focus of this effort. In a CTM, relatively simple algebraic relations are used to correlate exit plane emissions data. These relations include combustor inlet conditions, combustor geometry, fuel properties, and fuel injector effects. The CTM approach has been found to work well for NO_x and CO from conventionally fired gas turbines (Derr and Mellor, 1990). It has seen limited, but promising,

application to LP NO_x formation with laboratory flameholders (Newbury and Mellor, 1994). In addition to CTMs for NO_x and CO emissions, models for combustion efficiency, lean blowoff (lbo), and spark ignition have been developed and applied. Because the machine cycle design variables are included in the CTM relations, this model can also be used to study parameters affecting NO_x and CO emissions.

PROJECT DESCRIPTION

To meet the objectives outlined above, the three-year program is divided into Task 1, Combustor Modeling, and Task 2, NO_x and CO Modeling, with five subtasks (see the Program Schedule above): 1.1, Experiments at METC; 1.2, Characteristic Time Models; 1.3, Chemical Reactor Modeling; 2.1, Premixed Turbulent Flame Laboratory Measurements; and 2.2, Turbulence-Chemistry Modeling. Due to delays in the completion of the METC facility, the start of Subtask 1.1 experimental work designed in Subtask 1.2 has been delayed until Year 2 of the program. Data have been acquired during Summer 1994 at Sandia National Laboratories, Livermore under Subtask 2.1 but not yet reduced and analyzed. Finally, Subtask 2.2 is scheduled to begin in Year 2 of the program. The progress report which follows is thus taken largely from work conducted under Subtasks 1.2 and 1.3.

BACKGROUND INFORMATION

1. Methane Mechanism

During the first year of this study, a simple CRM, consisting of a perfectly stirred reactor (PSR) followed by a plug flow reactor (PFR), has been used to screen chemical kinetic mechanisms, to develop a reduced mechanism, and to evaluate NO_x emissions as a function of the principal combustion parameters. In order to model methane combustion, several full C_1 - C_2 hydrocarbon oxidation mechanisms with nitrogen chemistry have been considered. These are the mechanisms of Miller and Bowman

(1989), Drake and Blint (1991), and Michaud et al. (1992). The Miller and Bowman mechanism (with pressure corrections added to a few of the reactions) was chosen for the first year of study for the following reasons: (1) this mechanism has been widely used in combustion modeling; (2) the mechanism, when applied through chemical reactor modeling, shows very good agreement to the atmospheric pressure stirred reactor measurements of Steele et al. (1994); and (3) the choice of one established mechanism provides consistency in this research.

2. Nitrogen Chemistry

The nitrogen chemistry of LP combustion is complex. It involves all of the known pathways for NO_x formation, which are the Zeldovich, nitrous oxide, and prompt mechanisms, and it is influenced strongly by the non-equilibrium free radical chemistry.

This situation is much different than that of conventional, diffusion flame gas turbine combustors, in which the NO forms mainly by the thermal NO mechanism. Thermal NO is formed by the Zeldovich mechanism acting in a field of free radicals (i.e., O-atom, H-atom, and OH-radical) which are assumed to be at their local chemical equilibrium concentrations. Conventional diffusion flame combustors provide an opportunity for thermal NO to form in the hot, stoichiometric post-flame zones. A secondary contribution from the prompt mechanism is suspected for these engines. Simple engineering models built upon the thermal NO mechanism have been developed, calibrated against engine data, and used for years to predict NO behavior (e.g., see Derr and Mellor, 1990; Nicol et al., 1992). Some of these models include a prompt NO_x contribution.

Lean premixed combustors have been developed to avoid the hot, stoichiometric zones, and the large amounts of thermal NO_x formed in such zones. In these combustors the relatively small amounts of NO_x which are produced form mainly in and near the flame zones, in regions of non-equilibrium free radical chemistry.

For atmospheric pressure, LP combustion, modeling indicates that the approximate percentages of NO_x formed by the Zeldovich, nitrous oxide, and prompt mechanisms vary from 25:65:10 at 1650 K to 50:35:15 at 1850 K. For flame temperatures greater than about 1800 K, the Zeldovich pathway is favored, whereas for temperatures below 1800 K, the nitrous oxide pathway becomes significant. Modeling also indicates that as the pressure is increased the nitrous oxide mechanism becomes predominant. This occurs because the N_2O -forming step is a three-body reaction. For a nominal case with 14.31 atm, 673 K inlet temperature, and fuel/air equivalence ratio $\phi = 0.5$, the approximate percentages of the Zeldovich, nitrous oxide, and prompt NO_x are 15:84:1.

The high pressure simplifies the NO_x chemistry. The principal reactions forming the NO_x are the following reactions from the nitrous oxide mechanism:

That is, the nitrous oxide (N_2O) is formed by the forward direction of reaction (1), and is destroyed mainly by the reverse direction of reaction (1). (At reduced pressure, additional reactions affect the N_2O destruction.) The NO_x is formed mainly by the attack of the O-atom on the N_2O . The nitrous oxide acts as a reaction intermediate. Product (i.e., emission) levels of the N_2O are small, about 1 to 3 ppmvd.

Further, the small amounts of Zeldovich NO formed can be predicted by the following rate equation:

$$d[\text{NO}]/dt = 2k[\text{N}_2][\text{O}] \quad (3)$$

where k is the rate constant for the first Zeldovich reaction ($\text{N}_2 + \text{O} \rightarrow \text{NO} + \text{N}$).

The prompt mechanism involves attack of CH-radicals on N_2 , leading to HCN and N-atom, which subsequently are converted to NO_x . However, for high pressure, LP combustion, this mechanism appears to be very weak, although new kinetic data might conceivably change this result.

3. Characteristic Time Model

As shown in the Program Schedule above, Year 1 of AGTSR Subtask 1.1 was planned to involve testing (in the Low Emissions Combustor Test Rig at METC) of several simple flameholder geometries. Simple discs, toothed discs, cones, and modified cones, with several sizes of each, were selected, and an inlet condition matrix was designed based on lean blowoff and LP NO_x CTM sensitivity analyses. Initial predictions of CO and NO_x for the METC tests have been made.

The current LP NO_x model that was used for both the sensitivity analysis and the predictions of METC test results is discussed in detail by Newbury and Mellor (1994). Model constants are based on correlation of NO_x emissions for a perforated plate flameholder burning a lean premixed commercial grade propane/air mixture at elevated pressures and inlet temperatures (Roffe and Venkataramani, 1978a). The resulting model equation is:

$$\text{NO}_x \text{EI}|_{\phi < 1} \text{ (g/kg)} = 4.21 \tau_{sl,tno} / \tau_{tno} + 0.15 \pm 0.67 \quad (4)$$

where τ_{tno} is the kinetic time for LP NO_x formation, given by:

$$\tau_{tno} \text{ (ms)} = 10^{-5} (p_{in} / 1 \text{ atm})^{1-n} \exp(39700 / RT_{\phi}) \quad (5)$$

and n is the NO_x pressure exponent, estimated from the methane results of Nicol et al. (1992, 1993), predecessor studies to that discussed in Sections 1 and 2 above. The activation energy, 39700 cal/gmol K, was obtained from Arrhenius plots of the Roffe and Venkataramani (1978a) data (Newbury and Mellor, 1994).

The mixing time, $\tau_{sl,tno}$, follows the equation:

$$\tau_{sl,tno} \text{ (ms)} = l_{tno} / V_{\phi} \quad (6)$$

where l_{tno} is the characteristic length scale for NO_x:

$$l_{tno}^{-1} = d_{eff}^{-1} + l_{pr}^{-1} \quad (7)$$

Here d_{eff} is the diameter of a circular disc of the same area blockage as the flameholder, l_{pr} is the position downstream of the flameholder at which gas samples are taken, and V_{ϕ} is the velocity of the downstream burned gases:

$$V_{\phi} = V_{ref}(T_{\phi}/T_{in}) \quad (8)$$

Reference velocity (V_{ref}) is evaluated with combustor inlet density and flow rate, but at the maximum cross-sectional area in the combustor. The definition of l_{tno} and the adequacy of using V_{ϕ} will be addressed in the following.

An investigation into the proper definition of the characteristic time for NO quenching, $\tau_{sl,tno}$, is under way based on propane fired, LP experiments by Roffe and Venkataramani (1978b). In these experiments inlet conditions were essentially held constant, while flameholder geometry and residence time were varied. The results of this investigation suggest modification of the velocity used to characterize NO quenching. The lean premixed NO_x characteristic time model also has been applied to natural gas fired data from Joshi et al. (1994), with both velocities, and results compared. The modified velocity results in much improved, though still imperfect, correlation of the data.

RESULTS

1. Reduced Mechanism

Full detailed chemical mechanisms for methane-air combustion as given, for example, by Miller and Bowman (1989), involve over 200 reaction steps and 40 chemical species. Due to current computational limitations, the large size of these mechanisms essentially precludes their use in all but the simplest computer modeling, i.e., chemical reactor and laminar one-dimensional flame codes. Therefore, reduced chemical kinetic mechanisms for LP combustion of methane-air with NO_x formation by the nitrous oxide and Zeldovich pathways need to be developed for use with more complex computer models which directly incorporate some aspects of the fluid dynamics associated with the combustion

process. The reduced kinetic mechanism must be able to accurately predict over a range of operating conditions key species concentrations (e.g., CO, CO₂, NO, NO₂, N₂O, N₂, H, OH and O), ignition delay times, and flame speeds.

Chemical reactor modeling (CRM) consisting of arrangements of PSRs and PFRs is the simplest and computationally fastest means of numerically modeling the flame and post-flame zones of combustion. However, due to the zero-dimensional, micro-mixed nature of the PSR used to model the flame zone in CRM and the associated increased levels of temperature and combustion precursor free radicals, CRM is unable to provide information regarding ignition delay times and flame speeds. In order to determine these characteristics of the combustion process, the more complex and hence computationally slower laminar one-dimensional flame model must be used.

The full Miller and Bowman (1989) mechanism is used as the starting point for development of the reduced methane/NO_x mechanism. In the initial step of the development, a sensitivity analysis of the full kinetic mechanism is performed using both laminar one-dimensional modeling of a freely propagating adiabatic flame (i.e., the CHEMKIN code), and chemical reactor modeling consisting of a PSR followed by a PFR. From this analysis, those reaction steps and chemical species are identified which most strongly affect ignition delay time, laminar flame speed and the concentrations of the key species, and conversely, it is likewise determined which reactions steps and species have only minor effects on these combustion parameters. The full mechanism is reduced by systematically eliminating and/or globalizing these minor reactions and species. The Arrhenius rate coefficients of the reduced kinetic mechanism thus obtained are adjusted such that the predicted key species concentrations differ by no more than 1% (relative) and the predicted laminar flame speed differs by no more than 5% (relative) from the corresponding full mechanism values.

The reduction process has been used to develop the reduced kinetic mechanism involving 17 species and 30 reactions for LP

methane-air combustion shown in Table 1. The specific combustion parameters used to develop this reduced mechanism are those of the nominal case given above. Agreement between the full Miller and Bowman (1989) mechanism and the reduced mechanism for the nominal condition using chemical reactor modeling consisting of a PSR followed by a PFR is shown to be exact for all key species. The agreement between full and reduced mechanisms is additionally verified using laminar one-dimensional analysis of an adiabatic freely propagating flame operating under the same conditions. The results of this comparison are shown in Fig. 1. All key species differ by less than 1% (relative) from their full mechanism values and the laminar flame speed differs by less than 3% (relative). Further testing using CRM has shown that this reduced mechanism is valid over the following operating range: ϕ from 0.4 to 0.6, p_{in} from 1 to 30 atm, T_{in} from 298 to 800 K, and PSR t_{res} from blowout up to equilibrium.

Figure 1. Comparison Between the Full Miller-Bowman (1989) Mechanism and the Reduced Mechanism for the Nominal Condition Based on Laminar One-Dimensional Analysis of an Adiabatic Freely Propagating Flame

Table 1. Reduced 17 Species/30 Reaction Mechanism for Lean Premixed Methane–Air Combustion over the Pressure Range of 1 to 30 atm

Reaction Mechanism Rate Coefficients in the Form: $k_f = 10^A T^B \exp(-E_a/RT)$ Units are gmole, cubic centimeters, seconds, Kelvins, and kcalories/gmole					A	B	E _a
1.	H ₂	+ OH	= H ₂ O	+ H	9.068	1.30	3.63
2.	OH	+ OH	= H ₂ O	+ O	8.778	1.30	0.00
3.	H	+ O ₂	+ M = HO ₂	+ M	17.558	-0.72	0.00
Enhanced Third Body Efficiencies H ₂ O = 19.0; CO ₂ = 4.2; H ₂ = 2.9; CO = 2.1; N ₂ = 1.3							
4.	HO ₂	+ H	= OH	+ OH	14.146	0.00	1.07
5.	HO ₂	+ O	= OH	+ O ₂	13.146	0.00	1.07
6.	HO ₂	+ OH	= H ₂ O	+ O ₂	12.875	0.00	0.00
7.	O	+ OH	= O ₂	+ H	14.602	-0.50	0.00
8.	CH ₄	+ OH	= CH ₃	+ H ₂ O	6.204	2.10	2.46
9.	CH ₃	+ H	+ M = CH ₄	+ M			
High Pressure Limit:					16.778	-1.00	0.00
Low Pressure Limit:					26.903	-3.00	0.00
SRI Parameters ¹ : 4.50E-01, 797.0, 979.0							
Enhanced Third Body Efficiencies H ₂ = 2.0; CO = 2.0; CO ₂ = 3.0; H ₂ O = 5.0							
10.	CH ₃	+ O ₂	= CH ₃ O	+ O	19.312	-1.60	29.23
11.	CH ₃ O		+ M = CH ₂ O	+ H	14.000	0.00	25.00
12.	CH ₃	+ HO ₂	= CH ₃ O	+ OH	13.301	0.00	0.00
13.	CH ₃	+ O	= CH ₂ O	+ H	13.903	0.00	0.00
14.	CH ₃	+ OH	= CH ₂	+ H ₂ O	6.875	2.00	5.00
15.	CH ₃	+ OH	+ O ₂ = CH ₂ O	+ HO ₂	13.470	0.00	6.24
16.	CH ₃	+ OH	= CH ₂ O	+ H ₂	13.383	0.00	6.24
17.	CH ₂ O	+ OH	= CO	+ H ₂ O	9.416	1.18	-0.45
18.	CH ₂ O	+ OH	= CO	+ H ₂	8.232	1.18	-0.45
19.	CH ₂	+ O ₂	= CO ₂	+ H	12.204	0.00	1.00
20.	CH ₂	+ O ₂	= CO ₂	+ H ₂	11.839	0.00	0.50
21.	CH ₂	+ O ₂	= CH ₂ O	+ O	13.699	0.00	9.00
22.	CO	+ O ₂	= CO ₂	+ O	13.204	0.00	41.00
23.	CO	+ OH	= CO ₂	+ H	7.179	1.30	-0.76
24.	N ₂ O	+ M	= N ₂	+ O	14.204	0.00	51.60
25.	N ₂ O	+ H	= N ₂	+ OH	13.881	0.00	15.20
26.	N ₂ O	+ O	= NO	+ NO	14.000	0.00	28.20
27.	N ₂ O	+ O	= N ₂	+ O ₂	14.000	0.00	28.20
28.	N ₂ O	+ OH	= N ₂	+ HO ₂	12.301	0.00	10.00
29.	N ₂ O	+ H	+ O ₂ = NO	+ NO	14.743	0.00	29.83
30.	N ₂	+ O	+ O ₂ = O	+ NO	14.250	0.00	76.18

The reverse reaction rates for Reactions 15, 16, 17, 18, 29 and 30 are zero. The forward reaction rates for Reactions 15, 29 and 30 are treated as if they were two-body reactions. For example, $R_{f,15} = k_{f,15} [\text{CH}_3] [\text{OH}]$. The O₂ concentration does not factor into the reaction rate calculation and is used only to preserve atom conservation.

¹Pressure correction parameters in the standard SRI form. See Miller and Bowman (1989), Kee et al. (1989).

2. Effects of Operating Conditions

Chemical reactor modeling has been used to determine the effects of flame temperature, fuel–air equivalence ratio, inlet temperature, pressure, and residence time on NO_x formation in LP combustion of methane. The modeling work has been performed for the following range of conditions: pressure (P): 10, 20, and 30 atm; T_{in} : 600, 700, and 800 K; adiabatic equilibrium flame temperature (T_{ad}): 1539 to 1974 K (this temperature range corresponds to $0.32 \leq \phi \leq 0.65$); overall t_{res} from 2 to 5 ms; and PSR t_{res} of blowout to 0.5 ms. The chemical reactor model is the PSR–followed–by–PFR arrangement. The chemical kinetic mechanism is that of Miller and Bowman (1989).

A summary of the key results of the modeling is given as follows:

(1) The primary variable affecting the NO_x is the flame temperature. A sample of the exponential increase in the NO_x with increasing flame temperature is shown in Fig. 2. Of course, for a given pressure and

Figure 2. Chemical Reactor Modeling Results for CH_4 Fuel. The Reactor Modeling Consists of a PSR+PFR Configuration with the PSR at Blowout and the Total Residence Time at 2.0 ms. The Chemical Kinetic Mechanism of Miller–Bowman (1989) is Used.

inlet temperature the flame temperature increases with increasing fuel–air equivalence ratio. By the convention found in the recent IGTI literature, the NO_x is plotted versus the adiabatic equilibrium temperature rather than as a function of the equivalence ratio.

(2) The effect of pressure on the NO_x is shown in Fig. 3 and 4. For the leanest condition shown (Fig. 3), the NO_x does not

Figure 3. Chemical Reactor Modeling Results for CH_4 Fuel. The Reactor Modeling Consists of a PSR+PFR Configuration with the PSR at 0.5 ms. Results are Presented for an Adiabatic Equilibrium Temperature of 1685 K and Total Residence Times of 2.0 ms and 5.0 ms. The Miller–Bowman (1989) Chemical Kinetic Mechanism is Used.

continuously increase as the pressure is increased from 1 to 30 atm. The NO_x increases to a maximum and then decreases as the pressure is increased. This behavior occurs in the regime where the NO_x is formed mainly by the nitrous oxide mechanism, and the NO_x concentrations are small. The complex behavior is attributed to the dependency of the O–atom on pressure and to its non–linear role in the nitrous oxide mechanism. For the higher flame temperature cases (Fig. 4), the NO_x increases continuously with pressure, although the

Figure 4. Chemical Reactor Modeling Results for CH₄ Fuel. The Reactor Modeling Consists of a PSR+PFR Configuration with the PSR at 0.5 ms. Results are Presented for an Adiabatic Equilibrium Temperature of 1974 K and Total Residence Times of 2.0 ms and 5.0 ms. The Miller–Bowman (1989) Chemical Kinetic Mechanism is Used.

dependency at the high pressures is not as great as at the low pressures. In these figures, the NO_x dependency on pressure changes fairly dramatically at $P = 7$ to 8 atm. These results suggests two trends: 1) for land-based gas turbine engine pressure levels, the dependency of the NO_x on pressure is weak; and 2) experiments conducted at atmospheric pressure should show significantly reduced NO_x compared to "equivalent" high pressure experiments.

(3) The effect of residence time on the NO_x is shown in Fig. 5. Most of the NO_x forms early in the combustion process, i.e., in the PSR and initial part of the PFR. (The initial plateau of NO_x indicates the PSR zone.) These results indicate that the NO_x forms mainly in the flame zone (which is simulated by the PSR and very early part of the PFR), and is strongly dependent on the non-equilibrium flame zone chemistry. After the initial part of the PFR is passed, the NO_x forms slowly by the thermal NO_x mechanism. These results strongly suggest that significant

Figure 5. Chemical Reactor Modeling Results for CH₄ Fuel with the Fuel–Air Equivalence Ratio at 0.5, the Inlet Mixture Temperature at 700 K, and the Pressure at 20 atm. The Reactor Modeling Consists of a PSR+PFR Configuration with the Miller–Bowman (1989) Chemical Kinetic Mechanism.

reductions in the NO_x might be attained in practical systems by limiting the size of the zones of the highly non-equilibrium free-radical chemistry (see also Section 3 below). The results also suggest, because of the strong sensitivity shown, that differences in the NO_x between different LP combustors might be due to differences in the sizes of the non-equilibrium zones.

Curve fits are given below for the results based on the 0.5 ms PSR, over the flame temperature range of 1675 to 1950 K. These results show a simple exponential dependency of the NO_x on flame temperature, based on a single apparent activation energy. The NO_x is expressed as ppmvd (15% O₂), the temperature as degrees K, and the pressure as atm.

PSR at 0.5 ms + PFR, $t_{res} = 2.0$ ms:

$$NO_x = (2.06E + 7)(P/10)^{0.00212T_{ad}} \exp\left[\frac{-3.9297}{T_{ad} + 0.125(T_{in} - 600)}\right] \quad (9)$$

PSR at 0.5 ms + PFR, $t_{res} = 5.0$ ms:

$$NO_x = (4.70E + 7)(P/10)^{0.037T_{ad}} \exp\left[\frac{-4.3039}{T_{ad} + 0.125(T_{in} - 600)}\right] \quad (10)$$

In Fig. 6 and 7 these curve fits are compared to the original calculations. Results from the original calculations are assigned the symbols in Fig. 6 and 7. The curve fits indicate a relatively weak pressure dependence of the NO_x . At the high temperature limit (1950 K), the NO_x varies as $P^{0.20}$ for the 2.0 ms case and as $P^{0.32}$ for the 5.0 ms case. This trend is expected, because of the greater amount of thermal NO_x formed at the longer residence time. (Thermal NO_x varies as $P^{-0.5}$.) At the low temperature limit (1675 K), the NO_x varies as $P^{-0.38}$ for the 2.0 ms case and as $P^{-0.33}$ for the 5.0 ms case. This suggests a beneficial effect of increasing pressure for NO_x control for very lean (low temperature) conditions.

Figure 6. Comparison of Curve-Fit Expression Eq. (9) with Chemical Reactor Modeling Results. The Curve-Fit Expression is Given for an Adiabatic Equilibrium Temperature Range of 1675 K to 1950 K.

Figure 7. Comparison of Curve-Fit Expression Eq. (10) with Chemical Reactor Modeling Results. The Curve-Fit Expression is Given for an Adiabatic Equilibrium Temperature Range of 1675 K to 1950 K.

The curve fits indicate that an inlet temperature increase from 600 to 800 K effectively increases the flame temperature by 25 degrees. That is, the NO_x forms at a slightly higher temperature.

The curve fits show a relatively small effect of the overall residence time. In order to calculate this, the average case of $P = 20$ atm and $T_{in} = 700$ K is treated. At the high temperature limit (1950 K), the increase in the overall residence time from 2.0 to 5.0 ms increases the NO_x by 25% (relative). At the low temperature limit (1675 K), the increase in the overall residence time from 2.0 to 5.0 ms increases the NO_x by 7% (relative).

The apparent activation energy of NO_x formation indicated by the curve fits is 54.9 kcal/gmol (i.e., $1.987 \times 27608/1000$) for the 2.0 ms case, and 57.5 kcal/gmol for the 5.0 ms case. The trend of increasing apparent activation energy with increasing residence time is expected because of the greater amount of thermal NO_x formed. (The apparent activation of thermal NO_x formation is about 135 kcal/gmol.) The apparent

activation energy indicated by this computational work is close to that of 57.0 kcal/gmol inferred from the laboratory burner data of Leonard and Stegmaier (1993). The data of Leonard and Stegmaier, which were obtained with a "nearly perfect premixer" and a variety of flameholders, for flame temperatures of about 1675 to 1925 K, pressures of 1 to 30 bar, inlet air temperatures of 300 to 800 K, and overall residences times of 2 to 100 ms, can be fit with the following equation:

$$\text{NO}_x \text{ (ppmvd, 15\% O}_2\text{)} = (2.635\text{E}+7)\exp(-28695/T(\text{K})) \quad (11)$$

Curve fits for the results obtained using a blowout PSR are not given. For these cases the NO_x does not show a simple exponential dependency on flame temperature; a single value for the apparent activation energy is not indicated. This behavior is shown in Fig. 8,

Figure 8. Plot of Chemical Reactor Modeling Results, Presented for $\ln(\text{NO}_x)$ as a Function of the Adiabatic Equilibrium Temperature and Inlet Mixture Temperature. The Results are Given for an Adiabatic Equilibrium Temperature Range of 1675 K to 1900 K.

for the blowout PSR/overall 5.0 ms case. At the low temperature limit (1675 K), the apparent activation energy is 34.7 kcal/gmol and the pressure dependency is $P^{0.12}$. At the high temperature limit (1900 K), the apparent activation energy is 83.3 kcal/gmol and the pressure dependency is $P^{0.27}$. The high activation energy is the result of a thin PSR zone, followed by a relatively long PFR zone (with thermal NO_x formation).

3. Effects of Combustor Geometry

Admittedly the use of the PSR—followed-by—PFR configuration is a simplification of the combustion process in LP burners including laboratory flameholders such as the Roffe and Venkataramani (1978a) perforated plate, data from which provided the preliminary LP NO_x CTM (Newbury and Mellor, 1994). However, the CTM has been used for both simplified, axisymmetric laboratory flameholders and production conventional, diffusion flame combustors (Derr and Mellor, 1990). The link between different geometries is the mixing time for NO formation, $\tau_{sl,tno}$, in Eq. (6).

Tuttle et al. (1977), who originated the laboratory—flameholder thermal NO_x CTM, investigated diffusion flames, stabilized on a disc of diameter d_{disc} , where the time—mean stoichiometric contour will fall in the shear layer that separates the inlet air flow from the recirculation zone into which the fuel is sprayed, as illustrated in Fig. 9. Detailed probing of such flames revealed peak NO concentrations at the downstream end of this shear layer, whereas CO and HC peaked on the lean side of the leading edge of the shear layer. These maximum NO concentrations are associated with either peak local formation rates or with convection of NO from upstream reaction zones. In either case, Tuttle et al. (1977) showed that the appropriate length scale for their diffusion flames, with fixed exhaust sampling rake position, is $l_{no} = d_{disc}$, for discs both with and without teeth, most likely because the length of the recirculation zone is proportional to the disc diameter. The velocity in Eq. (6) was evaluated in the burned gases, somewhat

Figure 9. Schematic of Simple Disc-in-Duct Diffusion Flame (after Tuttle et al., 1977)

similar to that defined by Eq. (8). Quenching of the thermal NO_x formation occurs just downstream of the recirculation zone due to mixing of the reaction products with the excess air flowing around the flameholder wake.

In lean premixed combustion, the process is much different from diffusion flames. The highest temperatures occur in the fully burned gases, and the flame location is altered due to the initial mixedness of the reactants. Figure 10 is a diagram of the expected LP flame structure for the disc-in-duct flameholder. The cold mixture begins burning at the outer edge of the shear layer expanding from the disc trailing edge, and the spread angle may be a function of inlet conditions and the turbulent scale and intensity in the approach flow. The flame length depends on this angle and the spacing between the flameholder and the wall, but for constant inlet conditions, will increase linearly with the gap width. Alternatively, the flame area is related to the circumference of the flameholder (termed ignition perimeter by Roffe and Venkataramani, 1978b) as well as the gap width. It is likely that NO_x does not

form in the portions of this shear layer close to the cold wall, but, similar to the diffusion flame, CO will quench near the wall. The inner edge of the shear layer, important for NO formation in the diffusion flame, is associated with turbulent heat transfer to the flame zone from the hot burning gases residing in the recirculation zone.

Roffe and Venkataramani (1978b) measured emissions indices of NO_x , CO, and unburned hydrocarbons (UHC) with an area-averaging, water-cooled gas sampling rake for several flameholders stabilizing LP combustion at nominally 10 atmospheres pressure and 800 K air inlet temperature. The fuel used was commercial grade propane. Six different types of flameholders were tested at two blockage (by area) levels. Emissions data for these twelve flameholders were obtained at reference velocities of 20, 25, and 35 m/s and rake positions 1.25 and 3.75 duct diameters (10 and 30 cm) downstream of the flameholder. The equivalence ratio was varied when possible from approximately 0.3 to 0.8 for each test.

To date, the LP model given by Eq. (4) through (8) is based on Roffe and

Figure 10. Lean Premixed Combustion for a Disc-in-Duct

Venkataramani data gathered with a single point gas sampling probe for one flameholder geometry (Roffe and Venkataramani, 1978a). Combustor inlet pressure, temperature, and equivalence ratio were varied, and thus τ_{tno} was the focus of the model development (Newbury and Mellor, 1994). In Roffe and Venkataramani (1978b), inlet pressure and temperature were the constant parameters, and flameholder shape and size and reference velocity were changed. Thus, these data allow refinement of the defining equations for characteristic length and velocity in $\tau_{\text{sl,tno}}$: either the definitions used previously (see Eq. (6) through (8)) will be verified, or more appropriate choices can be identified. Figure 11 is a comparison of NO_xEI measurements versus $\tau_{\text{sl,tno}}/\tau_{\text{tno}}$ (as defined in Eqs. (4) through (8)) for perforated plate (PP) flameholders in the two Roffe and Venkataramani experiments.

It can be seen in Fig. 11 that the NO_xEI levels for data from Roffe and Venkataramani (1978b) generally lie above the model curve. The 80% blockage perforated plate flameholder in particular would be expected to be very similar to the 78% blockage flameholder from Roffe and Venkataramani (1978a), which agrees with Eq. (4). It is

likely that the probe used to measure NO_xEI in the 1978a experiments, upon which Eq. (4) is based, is not measuring all of the NO_xEI which is produced. As noted previously, the data for Roffe and Venkataramani (1978b) are obtained using an area-averaging rake, which should average the NO_xEI contributions over all regions of the flow.

The LP combustion process is somewhat different from that in Fig. 10 in cases such as the perforated plate and multiple cone flameholders described in Roffe and Venkataramani (1978b) where the outer edges of shear layers from adjacent flameholders meet, as illustrated in Fig. 12. For submerged jets of constant density, the potential core length is 4 to 5 jet diameters, and the mixing layer diameter at the end of the core is about 2.2 jet diameters. However, cold jets issuing into hot fluid (the case here) penetrate farther and mix (and entrain) more slowly than the constant density submerged jet. These effects tend to scale as the square root of the density ratio. Also, in this case, unlike that in Fig. 10, there is no cold wall to limit NO_x production and quench CO oxidation in the flame zone. The important length scale remains the ignition perimeter and/or flameholder gap spacing normal to the flow.

Figure 11. Measured NO_xEI Versus $\tau_{sl,tno}/\tau_{tno}$ for $T_{in} = 800 \text{ K}$, $p_{in} = 10 \text{ atm}$, $V_{ref} = 25 \text{ m/s}$ for Perforated Plate (PP) Flameholders of Indicated Blockages from Roffe and Venkataramani (1978a,b). Rake is at 30 cm for Roffe and Venkataramani (1978b), and Single-Point Probe from 11.4 to 15.3 cm for Roffe and Venkataramani (1978a). The Solid Line is Eq. (4), the Correlation of All the 1978a Data Including Those Shown for the Listed p_{in} and T_{in} .

3.1 Velocity for LP NO_x Emissions

A tabulation of the actual emissions measurements and the corresponding inlet conditions was unavailable in Roffe and Venkataramani (1978b), so the former were digitized off poorly reproduced graphs. Inlet conditions were taken to be the nominal values given in the report. This has given

data with large uncertainties. The NO_x is estimated to be taken from the graphs with $\pm 1.5 \text{ g/kg}$ accuracy, corresponding to about $\pm 15\%$ at the higher NO_xEI levels. It should be noted that the original graphs were semi-logarithmic, so the worst absolute errors are at the highest NO_xEI levels. The equivalence ratio is estimated accurate to ± 0.01 ($\pm 3.3\%$ at the lowest equivalence ratios). Also, for most flameholders at 35 m/s reference velocity and 30 cm rake position, NO_x emissions level off with increasing equivalence ratio, an unexpected and unexplained occurrence.

The equilibrium properties of the burned fuel-air mixture are calculated using the thermochemical code STANJAN (Reynolds, 1987). Roffe and Venkataramani (1978b) give the composition of the commercial grade propane used in their tests to be 90% propane (C_3H_8), 9.2% propene (C_3H_6) and 0.8% other (butane, ethene, ethane, volatile sulfur). For STANJAN calculations, the composition was approximated as 90.1% propane and 9.9% propene (a similar approximation is made in Newbury and Mellor (1994)). The fuel temperature was given as 380 K, the air inlet temperature is 800 K, and the pressure is 10 atmospheres for this experiment.

Adiabatic flame temperature and speed of sound were calculated for equivalence ratios from 0.3 to 0.8. Due to the high velocities encountered in these experiments, the effects of compressible flow were evaluated at the worst case conditions, 80% blockage flameholders at 35 m/s reference velocities, and compared to incompressible calculations. The primary concern is the annulus velocity, or velocity at the trailing edge (lip) of the flameholder, V_{lip} , which for incompressible flow is calculated based on the area ratio. At the conditions evaluated, the compressible flow V_{lip} is 0.4% higher than the incompressible flow value. Thus, it was concluded that incompressible flow equations can be used.

Some of Roffe and Venkataramani's run conditions led to high measured values of COEI and UHCEI, indicating low combustion efficiency and lower actual flame temperatures than those obtained with STANJAN. In order to correct for this, the combustion efficiency, η_c , was estimated based on COEI and UHCEI at each run condition and the

Figure 12. LP Combustion without Cold Wall Abutting Outer Edge of the Shear Layer

actual flame temperature was found and used in calculating $V\phi$ and τ_{tno} . This correction is important because at combustion efficiencies below 95%, there are differences of 50 K or greater between the flame temperature thus computed and $T\phi$. However, it should be emphasized that since the values of COEI and UHCEI were estimated, the combustion efficiency values are only crude estimates used to get a more realistic value of flame temperature at low η_c operating conditions.

3.2 Characteristic Velocity for LP NO_x

It was shown above in Fig. 5 that in lean premixed combustion the majority of the NO_x is formed early in the flame, and thus at the leading edge of the shear layer surrounding the flameholder recirculation zone, while the amount of NO_x formed in the downstream shear layer and plug flow portions (associated with thermal NO_x) is small. This hypothesis can be examined for the data of Roffe and Venkataramani (1978b) by comparing identical flameholders at similar run conditions, with l_{pr} at 10 and 30 cm. It is expected that the NO_x produced at the flameholder will be the same (within the uncertainty of the measurements) and the downstream NO_x will

be small and follow the equation:

$$NO_x EI_{down,i} = m \times \tau_{down,i} / \tau_{tno} \quad (12)$$

where $\tau_{down,i} = (l_{pr,i} - l_{rec}) / V\phi$, l_{rec} is recirculation zone length, and m , the slope, is an as yet undetermined constant for all cases. Data at 10 and 30 cm rake positions and 25 m/s reference velocity which have equivalence ratios within 0.01 of each other are available for multiple cone and perforated plate flameholders (four pairs for the former, and seven for the latter).

The length of the recirculation zone is estimated to be approximately equal to the diameter of the cones (d_{cone}). For single cone flameholders, $l_{rec} \approx d_{cone} \approx l_{pr}$ for 10 cm rake positions. Since the rake might actually be within the recirculation zone for the 10 cm rake positions (causing questionable emissions measurements), the single cone (SC) flameholders were omitted from this analysis. The length of the recirculation zone for the multiple cone (MC) flameholder is on the order of 1.9 cm, the diameter of an individual cone. For the perforated plate flameholders, the recirculation zone length is taken as the average between the largest and smallest solid lengths, 1.13 cm for 70% blockage and 1.05 cm for 80% blockage.

These pairs of data are substituted into

Eq. (13) to solve for $\text{NO}_x\text{EI}_{fh}$ and m in Eq. (12):

$$\text{NO}_x\text{EI}_{\text{meas},i} = \text{NO}_x\text{EI}_{fh} + \text{NO}_x\text{EI}_{\text{down},i} \quad (13)$$

Three of the eleven pairs exhibit higher NO_xEI at the 10 cm rake position than at the 30 cm rake position, suggesting that the differences between the measured NO_xEI at 30 cm and 10 cm for these cases are within the scatter of the data. This is further evidenced by the fact that the calculated slopes range from -3.8 to $+22.0$ g/kg and the calculated $\text{NO}_x\text{EI}_{fh}$ are in most cases close to both the 10 cm and 30 cm data. Figure 13 shows the two extreme slopes, one typical slope, and the amount of uncertainty at each measurement. It can easily be seen from the figure that the differences between NO_x formed at the flameholder (y-intercept of each curve), NO_x measured at the 10 cm rake position, and NO_x measured at the 30 cm rake position are all

Figure 13. Three Sample Pairs of Points. The y-Intercept of the Lines is the Calculated NO_x Formed at the Flameholder.

within the uncertainty of the measurements.

These results lead to the conclusion that most of the NO_x forms in the region close to the flameholder and that the NO_x which forms in the downstream portion can be neglected in most cases. Since thermal NO_x is attributed to the latter region, this finding is consistent with negligible Zeldovich NO formation in lean flames. Therefore, it is unlikely that the velocity of the downstream gases, $V\phi$, represents the turbulent flow where LP NO_x forms. The velocity of the gas at the flameholder lip, V_{lip} , should be a better parameter characterizing LP NO_x formation, and will be used in $\tau_{sl,tno}$ for further calculations. This choice implies that all other geometry effects for the data of Roffe and Venkataramani (1978b) must be accommodated in t_{no} in Eq. (7).

3.3 Application to Natural Gas Premixer

The analysis above suggests that the velocity at the trailing edge of the flameholder (V_{lip}) should be used in the definition of $\tau_{sl,tno}$, instead of the velocity at the trailing edge of the shear layer ($V\phi$). To examine this hypothesis, data from a premixer configuration tested in a sudden expansion burner by Joshi et al. (1994) were analyzed using each velocity in Eq. (6) and (4).

Figures 14 and 15 show the results, using data taken specifically from Fig. 11 of Joshi et al. (1994). Nominal values of p_{in} , T_{in} , and premixer exit velocity reported in Joshi et al. (1994) were used in model calculations. Flame temperature and NO_x values were digitized. Equivalence ratios were estimated based on the digitized flame temperatures, assuming natural gas combustion. The burned gas velocity $V\phi$ was used for the correlation in Fig. 14, and the velocity at the leading edge of the shear layer (V_{lip}), referred to as V_{exit} in Joshi et al. (1994), was utilized for Fig. 15. In Fig. 14 the model does not collapse the data for different inlet conditions onto a single line. Solid lines represent the overall best fit and plus and minus one standard deviation. In Fig. 15 the model correlation is much improved, as evidenced by the statistics in Fig. 14 and 15 (see captions).

Figure 14. Characteristic Time Model Graph of NO_xEI Versus Damköhler Number for NO_x Formation (Based on Burned Gas Velocity). Solid Lines are the Least Squares Fit of the Joshi et al. (1994) CH_4 Data and $\pm$ One Standard Deviation in y -Estimate (0.58 g/kg). The Correlation Coefficient is equal to 0.77. Dashed Lines are Eq. (4) and $\pm$ One Standard Deviation.

The original LP NO_x correlation for commercial grade propane, Eq. (4), is shown in Fig. 14 by the dashed lines; in Fig. 15 the Roffe and Venkataramani (1978a) data were recorrelated using V_{lip} for comparison with the data of Joshi et al. (1994). As for the propane correlation, recommendations from Nicol et al. (1993) for methane were utilized to estimate the pressure exponent for Eq. (5). The error associated with the fuel switch and the uncertainties in length scales for the different flameholders, as well as using nominal, instead of measured, values for p_{in} , T_{in} , ϕ , and the velocities, are thought responsible for the difference between the two best fits for the two sets of data, and for much of the scatter exhibited by the natural gas correlation in Fig. 15. Quality of fuel/air mixedness is also unknown in the experiments. However, the degree of improvement

Figure 15. Characteristic Time Model Graph of NO_xEI Versus Damköhler Number for NO_x Formation (Based on V_{lip}). Solid Lines are the Least Squares Fit of the Joshi et al. (1994) CH_4 Data and $\pm$ One Standard Deviation in y -Estimate (0.38 g/kg). The Correlation Coefficient is Equal to 0.91. Dashed Lines are the New Correlation of the Roffe and Venkataramani (1978a) Data Using V_{lip} and $\pm$ One Standard Deviation.

resulting from the use of V_{lip} suggests that most of the LP NO_x is, in fact, formed near the leading edge of the shear layer, as was suggested in the last section. Work is now in progress to use Eq. (9) and (10) in conjunction with a modification of Eq. (5) to refine the correlation in Fig. 15 of the data from Joshi et al. (1994).

3.4 Characteristic Length Scale for LP NO_x

For the preliminary results with the Joshi et al. (1994) data shown in Fig. 14 and 15, d_{eff} was taken as twice the step height plus the centerbody diameter at the station of the sudden expansion, analogous to the procedure used for the original correlation of the Roffe and Venkataramani (1978a) data

(see Eq. (7)). However, for the following preliminary analysis of the data of Roffe and Venkataramani (1978b), l_{pr} was omitted due to the observation that the NO_x forms in the region close to the flameholder, and that the downstream thermal NO_x is negligible.

Several possible definitions of l_{tno} were tested with the latter data (and will be examined in the experiments at METC). Figure 16 shows the results when the model as defined in Eq. (4) through (8) is used. In this case, l_{tno} is identical to d_{eff} , and it can be seen that this model does not collapse data for flameholders of other geometries.

One alternative procedure is to follow Tuttle et al. (1977) and define the NO_x length scale to be the recirculation zone length, which is proportional to disc diameter. This definition of l_{tno} assumes that NO_x forms at the inner edge of the shear layer (see Fig. 10), which is not believed to be true in LP combustion, but this length scale might be appropriate if NO_x is proportional to another length which scales with d_{disc} .

The distance between the cold wall and the disc is the length scale discussed previously that could be important in LP NO_x formation, because it is associated with the outer edge of the shear layer, as illustrated in Fig. 10. In this third case l_{tno} is defined as:

$$l_{tno} = d_{comb} - d_{disc} \quad (14)$$

where d_{comb} is combustor diameter.

The final length scale to be discussed here is the ignition perimeter, the total perimeter of the flameholder. For multiple cone flameholders (diameter of each d_{cone} ; number N_{cone}) as in Roffe and Venkataramani (1978b), ignition perimeter is:

$$l_{tno} = \pi d_{cone} N_{cone} \quad (15)$$

and for perforated plate flameholders with hole diameter d_{hole} :

$$l_{tno} = \pi d_{hole} N_{hole} \quad (16)$$

This definition has shown the most success in correlating the different types of flameholders used by Roffe and Venkataramani (1978b). Figure 17 shows results for the six flameholders that have been examined with the CTM.

Clarification of the appropriate LP NO_x

Figure 16. NO_xEI Data from Roffe and Venkataramani (1978b) Versus LP Fluid to Kinetic Time Ratio for $V_{ref} = 25 \text{ m/s}$, $l_{pr} = 30 \text{ cm}$, $p_{in} = 10 \text{ atm}$, $T_{in} = 800 \text{ K}$, $\tau_{sl,tno} = d_{eff}/V_{\phi}$.

length scale for these experiments must await the results of the tests at METC. The new data should give conclusive evidence into the proper definition of l_{tno} .

FUTURE WORK

Currently the characteristic length scale is assumed to be a dimension characterizing the size of the flameholding recirculation zone, an assumption based on the successful thermal NO_x CTM for diffusion flame combustors. We will continue examination of the validity of this concept and re-examine the correlation shown in Fig. 17 when including probe/rake

Figure 17. NO_xEI Data from Roffe and Venkataramani (1978b) Versus LP Fluid to Kinetic Time Ratio for $V_{\text{ref}} = 25 \text{ m/s}$, $l_{\text{pr}} = 30 \text{ cm}$, $p_{\text{in}} = 10 \text{ atm}$, $T_{\text{in}} = 800 \text{ K}$, $\tau_{\text{sl,tmo}} = (\text{Ignition Perimeter})/(\text{Lip Velocity})$.

position, l_{pr} , in the characteristic NO quench length. New correlations of data from Joshi et al. (1994) will also be performed using Eq. (9) and (10) so that the best available overall kinetic time for LP NO_x formation in CH_4/air systems is incorporated in the CTM.

A zonal chemical reactor model is being developed based on the Roffe and Venkataramani (1978a) geometry using PSR zones for the shear layer, a recycle zone (for which both a PSR and a PFR will be tried), and a PFR downstream zone. The ratio of the mass flow rate recycled to that of the inlet jet will be varied in this modeling. Preliminary calculations, based on jet theory, indicate

that this ratio is small, in the 0.2 to 0.3 range. The mass flow rate of fuel-air mixture entering the shear layer will be based on the loss of volume of the potential core. Zonal chemical reactor modeling also is being applied to disc flameholders (see Fig. 10), modeled as a shear layer of several PSR or PFR elements, and a large recirculation zone modeled as either a PSR or PFR. The amount of material recirculated is a variable in this model.

The study will also be broadened to involve the effects of macromixing and micromixing on pollutant formation in LP combustion. Macromixing is described by a measure of gross convective recirculation within the combustion device and is characterized by the distribution of fluid residence times within the combustor. Micromixing measures the extent of molecular homogeneity or turbulent small-scale mixing within the combustor and is the necessary requisite for a chemical reaction to take place. Practical combustion devices experience a finite degree of macromixing and micromixing. Therefore, an understanding of the interaction between mixing and chemistry can be gained by models which have the capability to independently vary the degree of macromixing and micromixing.

Three mixing models are used to study the effects of micromixing and macromixing in high-intensity, LP combustion. These models include the macromixed/micromixed PSR model, the variable micromixing/macromixed coalescence-dispersion (C-D) model, as described by Pratt (1975), and the variable macromixing/micromixed Jain and Spalding (1966) (J-S) model. The J-S model uses one parameter to vary the amount of recirculation within the reactor using Jain and Spalding's flow description for the reactor.

The three models are being used to evaluate the LP, stirred reactor experiments of Steele et al. (1994) for finite mixing effects. Once this evaluation is completed, and experience is gained with the models, a fourth model will be developed, which will include together in one code both finite rate macromixing and micromixing. It will be incorporated into the chemical reactor modeling predictions of NO_x and CO for LP combustion turbines.

REFERENCES

- Derr, W.S. and Mellor, A.M. (1990), "Recent developments," pp. 477-550, Design of Modern Turbine Combustors, Academic Press, London.
- Drake, M.C. and Blint, R.J. (1991), "Calculations of NO_x formation pathways in propagating laminar, high pressure premixed methane/air flames," *Combust. Sci. Tech.* **75**, 261-285.
- Jain, V.K. and Spalding, D.B. (1966), "The effects of finite recirculation in a stirred reactor," *Combust. Flame* **10**, 37-43.
- Joshi, N.D., Epstein, M.J., Durlak, S., Marakovits, S. and Sabla, P.E. (1994), "Development of a fuel air premixer for aeroderivative dry low emissions combustors," ASME Paper No. 94-GT-144.
- Kee, R.J., Rupley, F.M. and Miller, J.A. (1989), "Chemkin-II. A Fortran chemical kinetics package for the analysis of gas phase chemical kinetics," Sandia National Laboratories Report No. SAND89-8009B.
- Leonard, G. and Stegmaier, J. (1993), "Development of an aeroderivative gas turbine dry low emissions combustion system," ASME Paper No. 93-GT-288.
- Mellor, A.M. (1976), "Gas turbine engine pollution," *Prog. Energy Combust. Sci.* **1**, 111-113.
- Michaud, S.R., Westmoreland, P.R. and Feitelberg, A.S. (1992), "Chemical mechanisms of NO_x formation for gas turbine conditions," pp. 879-887, Twenty-Fourth Symposium (International) on Combustion, The Combustion Institute, Pittsburgh.
- Miller, J.A. and Bowman, C.T. (1989), "Mechanism and modeling of nitrogen chemistry in combustion," *Prog. Energy Combust. Sci.* **15**, 287-338.
- Newbury, D.M. and Mellor, A.M. (1994), "Characteristic time model correlation of NO_x emissions from lean premixed combustion," Submitted to International Gas Turbine Institute Turbo Expo '95.
- Nicol, D.G., Malte, P.C., Lai, J., Marinov, N.M., Pratt, D.T. and Corr, R.A. (1992), " NO_x sensitivities for gas turbine engines operated on lean-premixed combustion and conventional diffusion flames," ASME Paper No. 92-GT-115.
- Nicol, D.G., Steele, R.C., Marinov, N.M. and Malte, P.C. (1993), "The importance of the nitrous oxide pathway to NO_x in lean-premixed combustion," ASME Paper No. 93-GT-342.
- Pratt, D.T. (1975), "Mixing and chemical reaction in continuous combustion," *Prog. Energy Combust. Sci.* **1**, 75-88.
- Reynolds, W.C. (1987), "STANJAN Version 3.93," Dept. of Mech. Eng., Stanford Univ.
- Roffe, G. and Venkataramani, K.S. (1978a), "Emission measurements for a lean premixed propane/air system at pressures up to 30 atmospheres," NASA CR-159421.
- Roffe, G. and Venkataramani, K.S. (1978b), "Experimental study of the effects of flameholder geometry on emissions and performance of lean premixed combustors," NASA CR-135424.
- Steele, R.C., Malte, P.C., Nicol, D.G. and Kramlich, J.C. (1994), " NO_x and N_2O in lean-premixed jet-stirred flames," paper presented at the Twenty-Fifth Symposium (International) on Combustion, (Combust. Flame, to appear).
- Tuttle, J.H., Colket, M.B., Bilger, R.W. and Mellor, A.M. (1977), "Characteristic times for combustion and pollutant formation in spray combustion," pp. 209-219, Sixteenth Symposium (International) on Combustion, The Combustion Institute, Pittsburgh.

Combustion Modeling in Advanced Gas Turbine Systems

CONTRACT INFORMATION

AGTSR Subcontract No. 93-01-SR014

University Advanced Combustion Engineering Research Center
Brigham Young University
45 CTB, BYU
Provo, UT 84602

Principal Investigator Paul O. Hedman
phone: (801) 378-6238 fax: (801) 378-7799

Co-Investigators L. Douglas Smoot, Project Director;
B. Scott Brewster, Thomas H. Fletcher, and
Stephen K. Kramer, Senior Investigators

SCERDC Managers Daniel P. Fant, Lawrence P. Golan

DOE Project Manager Dr. Paul Micheli/ Dr. Norm Holcombe

Period of Performance September 1, 1993 to August 31, 1997

Schedule and Milestones

FY 94 - 97 Program Schedule

OBJECTIVE

The primary objective of the four year
ATS program at this center is to develop a

comprehensive combustion model for advanced gas turbine combustion systems using primarily natural gas fuel with consideration of coal gasification or biomass fuels.

BACKGROUND INFORMATION

Foundations for this work are well established in the Brigham Young University's (BYU), National Science Foundation (NSF)-sponsored Advanced Combustion Engineering Research Center (ACERC). ACERC's 3-D code, PCGC-3, applies to coal combustion and gasification, as well as to reacting and non-reacting gaseous flows. It includes submodels in turbulent flow, gaseous reactions, solids reactions, radiative and convective heat transfer, NO_x/SO_x formation, and particulate flows. This code has been evaluated by extensive comparisons with measured data and has been applied to many different industrial furnaces and combustors, including gas turbines.

BYU/ACERC has also developed, constructed, and demonstrated a laboratory-scale, gas-turbine combustor (LSGTC) with advanced, laser-based, combustion diagnostic instruments. The LSGTC simulates many of the key combustor characteristics of commercial gas turbines. Use of advanced optical diagnostics permits near-instantaneous, non-intrusive measurement of such quantities as velocity, temperature, flame shape, and species concentrations that are essential for model validation.

Finally, BYU/ACERC has established close relationships with industry. ACERC is guided by industrial interaction and has as a central mission the improvement of the competitive position of the U.S. fossil energy industry.

PROJECT DESCRIPTION

Three tasks have been undertaken to achieve the overall objective of this program. The first task is to improve and validate various submodels needed to accurately predict various aspects of the gas turbine combustion process. Of particular emphasis are submodels for predicting carbon monoxide and nitrogen oxides, particularly at the higher temperatures and pressures proposed for advanced turbine systems. Ongoing improvements in existing submodels such as turbulence, finite-rate chemistry, and pollutant emissions are also being considered for addition to the 3-D code. These submodels will be

validated by data obtained with advanced diagnostic systems from the LSGTC.

The second task is to refine the BYU/ACERC 3-D code (PCGC-3) for application to gas turbine combustors and implement proven submodels into the gas turbine model. Refinements and submodels will also be available for incorporation into other prominent computational fluid dynamics (CFD) platforms such as those from Fluent, a commercial developer of CFD codes.

The third task is to interact extensively with industrial and university organizations with strong gas turbine interest. An ACERC/ATS technical advisory committee (TAC) composed of prominent industrial organizations with specific interest in utility and industrial gas turbine manufacture and application has been formed. The TAC provides direction and support of this research program. Concerns particular to the industry are being addressed, and results will be provided to participating industrial members.

RESULTS

Key accomplishments of the past year are described below.

Task 1 - Submodel Development

The first task is to improve and validate various submodels needed to accurately predict various aspects of the gas turbine combustion process. Work that has been performed on Task 1 includes modeling of the LSGTC with premixed and non-premixed burners using PCGC-2. A new burner was designed and constructed to study turbulent, premixed natural gas/air flames in the LSGTC. Foundational velocity and gas temperature data were obtained. A literature review of NO_x and CO formation submodels was begun. Calculations were performed on reduced mechanisms using a partially stirred reactor model.

1.1 - Code Evaluation. The objective of this subtask is to evaluate the existing PCGC-3 code by comparing model results to local

measurements from the BYU/ACERC laboratory-scale gas turbine combustor (LSGTC). Variables of immediate interest include local gas velocities and temperatures. Future efforts will also include CO/CO₂ concentrations, and NO_x pollutant levels. Key model needs are being identified through this effort.

PCGC-2 calculations were performed of the LSGTC with the premixed burner and with the Pratt & Whitney Task 150-HS burner. The Task 150-HS is a non-premixed burner for which extensive data are available (Hedman et al., 1994, Hedman and Warren, 1994). These calculations were made using the mixture fraction approach (i.e., mixing-limited chemical reactions) in order to understand some of the critical variables in modeling flames in the LSGTC.

Experiments with the premixed burners in the LSGTC show flame structures similar in shape and intensity to those obtained with the non-premixed burners. However, as expected for the mixture-fraction approach, calculations for the premixed burner predicted uniform temperatures in the LSGTC except near the walls where wall heat transfer effects are important. These results demonstrate the inadequacy of the mixture-fraction approach in the prediction of premixed flame behavior.

Code predictions were made of the non-premixed TASK 150-HS burner in order to evaluate other key characteristics of the model. The data set chosen for comparison with model predictions was from a propane/air flame, with an air flow rate of 500 slpm and fuel equivalence ratio (ϕ) of 0.72. The nominal swirl number for this case was 1.41. Experimentally-measured velocity profiles at the mouth of the burner were used as input conditions for the PCGC-2 calculations. These profiles were found to be non-symmetric, mainly due to the presence of the insert jets, which are 15 discrete holes positioned at the outer edge of the burner mouth, and the discrete swirl vanes in the injector. The pressure in these calculations was 1 atmosphere, and wall temperatures were assumed to be 1000 K. Turbulence intensity profiles were assumed to be 10%. The k- ϵ turbulence model was used in conjunction with a clipped Gaussian PDF based on the mixture fraction and the variance of the

mixture fraction, as is customary in current PCGC-2 calculations.

PCGC-2 calculations of axial velocity and temperature in the Task 150 combustor are compared with experimental measurements in Figures 1 and 2. Two main areas of disagreement are apparent: 1) the model does not accurately predict the spread of the flame in the radial direction, and a thinner flame is predicted than observed experimentally; and 2) the axial predicted velocity near the burner mouth increases to ~60 m/s due to the diffusion flame of propane and air, whereas the maximum measured axial velocity is ~25 m/s. There is also disagreement in the temperature of the recirculation zone near the dome jets. These areas of disagreement indicate that the inlet conditions used for the model predictions may not be representative of the combustion case. It is possible that either the cold flow measurements of inlet velocities do not represent the combustion case, or else some premixing occurs that needs to be taken into account.

Despite the areas of discrepancy mentioned above, several features were accurately modeled: 1) the approximate flame height; 2) the maximum flame temperature; and 3) the general flame structure, including the zone of low temperature extending upwards from the burner mouth above the insert jets. It is thought that these areas of agreement are general characteristics of turbulent diffusion flames of this configuration. The calculations show a cold region at the inlet along the centerline (Figure 1), which is characteristic of diffusion flames, but the data show relatively hot temperatures in this same region. This indicates some degree of mixing and reaction of primary air and fuel in the gap between the injector and the burner mouth. More agreement between experiment and modeling may be possible if the 3-D version which includes burner geometry and turbulent reaction chemistry is used.

1.2 - Fundamental Experiments. The objective of this subtask is to conduct key experiments in the LSGTC to provide needed data for PCGC-3 and submodel evaluation. Measurements include: 1) gas temperature using coherent anti-Stokes Raman spectroscopy

Figure 1. CARS measurements and modeling results of gas temperature (500 slpm air flow, $\phi = 0.75$)

Figure 2. LDA Measurements and modeling results of axial velocity (500 slpm air flow, $\phi = 0.72$)

(CARS), 2) velocity components using laser Doppler anemometry (LDA), 3) presence of OH, CH, CO, NO, NO₂ and possibly N₂O using planar, laser-induced fluorescence (PLIF), 4) flame and flow structure using film images of Mie scattering from seed particles illuminated with a sheet of light from a pulsed laser, and 5) flame and flow structure using digitized images from film or video cameras.

For this study, the LSGTC was modified to incorporate a turbulent, premixed, swirling injector with gaseous propane as fuel. Three different swirl block inserts have been prepared which give the injector theoretical swirl numbers of 0.43 (LS), 0.74 (MS), and 1.29 (HS). The combustor has been configured such that the geometry around the injector is nearly axisymmetric with a diameter of 150 mm. Four 60 mm wide flat quartz windows are located on each of the four sides to allow use of the laser diagnostic equipment. This configuration allows complex laser diagnostic measurements to be made in a combustor that is easier to model mathematically but still embodies most of the features of an actual gas turbine combustor. Measurements to date include digitized video images, CARS gas temperature and LDA velocity data.

Flame structures for each of the three swirl levels over a wide range of fuel equivalence ratios have been observed using the digitized video images. Selected flame structures from these video recordings were digitized and filtered to produce the isochromatic contour plots shown in Figure 3.

Both the vortex dimensions and the point of attachment vary significantly with the fuel equivalence ratio (ϕ) and swirl number. At the rich condition of $\phi = 1.5$, none of the burner configurations exhibited any definable structure. Between $\phi = 1.3$ and 1.2 (Figure 3C) the flame in the LS burner changed from an amorphous flame to a tight, well defined vortex with the base of the vortex funnel stabilizing approximately 20 to 30 mm above the injector. The top of the vortex struck the burner wall and was partially deflected downward and doubled over on itself. In addition, a large diffusion flame was attached to the combustor exit as a result of oxygen in the

ambient surroundings diffusing into and reacting with the unburned fuel leaving the combustor. In the range $\phi = 1.2$ to $\phi = 0.7$ (Figures 3A and 3B) the flame shape remained unchanged with only a reduction in visible intensity as the fuel equivalence ratio was reduced from the stoichiometric condition ($\phi = 1.0$) to the leaner conditions ($\phi < 1.0$). The external flame disappeared for $\phi \leq 1.0$. As the fuel flow was reduced further the flame shape again became undefinable until lean blowout occurred at $\phi = 0.57$.

The MS burner began to exhibit a definable structure at a higher fuel equivalence ratio than the LS burner. In the range $\phi = 1.4$ to $\phi = 1.3$ a thick cloud was evident at the top of the vortex but the main body of the vortex could not be seen. Between $\phi = 1.2$ and $\phi = 0.7$ (Figures 3D through 3F) the MS burner showed a flame structure comparable to the flame structure for the LS at the same fuel equivalence ratios, with the major difference being that the base of the vortex funnel stabilized and attached at the injector exit. Below $\phi = 0.7$ the flame structure again appeared as a thick cloud without a definable vortex, with lean blowout occurring at $\phi = 0.52$. As with the LS case, a large external flame was present for $\phi > 1.0$.

The HS burner exhibited some of the most interesting flame shape transitions. Between $\phi = 1.3$ and $\phi = 1.1$, the flame began to take on the form of a thick cloud with a vortex funnel cloud slowly emerging as the fuel equivalence ratio was reduced from 1.3 to 1.1 (Figure 3I). As in the other two cases an external flame was present for $\phi > 1.0$. In the range $\phi = 1.0$ to $\phi = 0.9$ (Figure 3H), the flame had a bimodal structure with the flame shape alternating between a taller, vortex shape and a shorter, broader funnel shape. At $\phi = 0.8$ (Figure 3G) the flame maintained the shorter, broader funnel shape. Below $\phi = 0.7$ the vortex funnel again disappeared with the flame eventually losing all structure until lean blowout at $\phi = 0.51$. Unlike the MS burner, the tip of the vortex funnel for the HS burner was not visible since the base of the vortex was drawn into the injector exit and lost from sight.

LDA data for axial, radial and tangential velocity have been obtained for the HS

A) ATS-LS $\phi = 0.75$

B) ATS-LS $\phi = 1.00$

C) ATS-LS $\phi = 1.25$

D) ATS-MS $\phi = 0.75$

E) ATS-MS $\phi = 1.00$

F) ATS-MS $\phi = 1.25$

G) ATS-HS $\phi = 0.75$

H) ATS-HS $\phi = 1.00$

I) ATS-HS $\phi = 1.25$

Figure 3. Characteristic flame shapes from digitized video image of the ATS burner at various operating conditions

configuration at a $\phi = 0.75$ for an air flow of 500 slpm. The basic flow field in the combustor was derived by interpolation of the velocity data. The flow field is dominated by several recirculation zones. The axial velocity data for both the X and Y coordinate traverses were combined and interpolated to yield the zero velocity contours shown in Figure 4. Each zero velocity contour bisects a recirculation zone.

Figure 5 shows iso-velocity contour plots of the mean axial, radial and tangential velocity measurements. The radial velocity data was limited to ± 30 mm due to the width constraints of the windows. The flame at this operating condition was well attached with the base of the funnel cloud being drawn into the nozzle and out of sight. The sharp axial velocity peak near the injector exit is readily apparent (Figure 5A). It can be seen that the axial velocity is highest at the midpoint between the recirculation zones and then decays outward in both directions as part of the flow moves upward and the other part is drawn down into the region of negative axial velocity near the wall.

Figure 4. Recirculation zones and zero velocity contours in ATS-HS burner $\phi = 0.75$

Note: (N) denotes negative number

Figure 5. Iso-velocity contours for the ATS-HS burner at $\phi = 0.75$ and 500 slpm air flow

Adiabatic Flame Temperature = 1963 K at $f = 0.75$

Figure 6. Mean gas temperature contours for ATS-HS, -MS, and -LS burners at $\phi = 0.75$ and 500 slpm air flow

The radial velocity contours in Figure 5B support the observations about the flow structure made from the axial velocity information. The large recirculation zone in and above the vortex is shown by the region of low negative radial velocity around the 250 mm axial location and the regions of low positive velocity around the central zero velocity contour inside the funnel cloud. The existence of the recirculating eddy in the lower corner of the combustor is also supported by the radial velocity data. The vortex like structure of the flow can be clearly seen from the tangential velocity contours in Figure 5C. The decay of the tangential velocity in the axial direction along the funnel cloud exhibits the same trend as the axial velocity (Figure 5A) in this region. Examination of the velocity contours for the axial, radial and tangential velocities shows that this central region has a negative axial velocity (down draft), low radial velocity and negligible tangential velocity.

CARS gas temperature data have been collected for the LS, MS and HS configurations at a $\phi = 0.75$ and an air flow of 500 slpm. Like the

gas velocity measurements, the CARS temperature measurements were taken at closely spaced radial increments (3 mm) where large gradients in temperature were found and in larger increments (10 mm) where the temperature gradients were relatively shallow. The axial locations for the temperature measurements were identical to the axial locations for the velocity measurements.

The temperature contours shown in Figure 6 are for the ATS-HS, -MS and -LS burners. Despite the same fuel equivalence ratio and air flow rate the effects of the different swirl intensities are clearly evident from the temperature contours. The shape of the funnel cloud is easily distinguished by the temperature contours, particularly in Figures 6B and 6C. The troughs in the contours are due to the central recirculation zone which draws hot product gases into the interior of the vortex.

When the temperature contours for the HS burner are superimposed on the corresponding

axial velocity contours a definite correlation between the temperature field and velocity field becomes evident. Though the temperature data is limited to a radius of 30 mm, the effects of the corner and outside recirculation zones (Figure 4; zones 2,3, and 4) on the temperature field can be seen. Examination of Figure 6C reveals a narrow region of sharply increasing temperature which seems to fold back onto itself. This region, which coincides with the region of sharply decreasing axial velocity in Figure 5A, is shaped by the recirculation of the reacting gases from the cloud at the top of the vortex, which are partially deflected downward and then drawn towards the center of the burner due the two outside recirculation zones (Figure 4; flow path A-C). A similar temperature pattern can be seen in both the LS and MS burners.

The most intense part of the flame from a visual standpoint is found in the cloud at the top of the vortex. This corresponds to the temperature region between approximately 1500 and 1700 K. The peak temperatures for all three burner configurations occurred in the region above the vortex and away from the visible reaction zone. The peak temperatures measured for the ATS-HS, -MS and -LS burners were 1857 K, 1811 K, and 1768 K respectively, with each configuration having a theoretical adiabatic flame temperature of 1963 K at $\phi = 0.75$.

Combination of the CARS temperature data and the LDA velocity data for the HS configuration at $\phi = 0.75$ has yielded local continuity and energy release information for differential elements throughout the flame zone. The calculated heat release has been normalized by the volume of the control volume and is shown in Figure 7. The regions of largest heat release are consistent with the visually observed flame. The location of the cloud at the top of the vortex, which is the most intense visible portion of the flame, is especially prominent with heat release in excess of 200 kJ/m³s.

1.3 - Submodel Improvements. The objective of this subtask is to improve the submodels in the code using available technology and advances from this study, from other ACERC researchers under independent funding, and from the literature, in order to provide increased

Figure 7. Energy release contours (kJ/m³ s) for ATS-HS burner $\phi = 0.75$ at 500 slpm air flow

agreement with the key experiments identified in Subtask 1.2. Specific submodels being considered include finite-rate chemistry, NO_x formation at high pressures in fuel-lean systems, and CO/CO₂ non-equilibrium.

A comprehensive review of the literature related to reduced kinetic mechanisms for modeling lean premixed turbulent combustion was initiated. Based on this review, efforts were directed at: 1) identifying proposed reduced mechanisms for describing combustion of CH₄ and formation of NO_x; 2) evaluating these proposed reduced mechanisms using ideal reactor codes; 3) implementing the best mechanisms into the BYU/ACERC multi-dimensional gas turbine combustion simulation code (PCGC-3); and 4) evaluating the reduced mechanism(s) by comparison with measurements at atmospheric pressure from the BYU laboratory-scale, gas turbine combustor (LSGTC). This subtask is closely coordinated with on-going NO_x research in ACERC, where reburning and N₂O and NO₂

formation are being investigated, and with ATS-sponsored studies at other universities.

Several potential reduced mechanisms were identified through contacts at the International Symposium on Combustion and in the literature. Methods have been developed that derive reduced mechanisms from a full set of reaction schemes (Chen, 1988; Warnatz, 1993; Wang and Frenklach, 1991; Maas and Pope, 1992a,b). Sensitivity analysis is one such method. This analysis is used to identify the rate-limiting steps in a group of parallel reactions. The non-rate limiting reactions are then removed. Chen (1988) has outlined a computerized procedure to develop a set of reduced mechanisms from detailed mechanisms if independent approximate relations are present. These relations are based on steady state or partial equilibrium assumptions and allow the calculation of some species in terms of other known concentrations. For complex flow models, Maas and Pope (1992) have developed a computerized reduction method which requires only the detailed kinetic mechanism and the number of degrees of freedom needed in the reduced scheme.

There are many reduced mechanisms published in literature which have been used to model laminar systems for a wide range of flame parameters. Westbrook and Dryer (1981) derived a two-step mechanism that adequately describes energy release and burned gas compositions but does not represent the intermediate chemistry of methane combustion. To generally represent the intermediate chemistry in methane oxidation, at least four overall reactions are required, such as is done by Jones and Lindstedt (1988). The mechanisms of Hautman et al. (1981) and Seshadri and Peters (1990) cover the widest ranges of pressure and stoichiometry, but may be less accurate for certain conditions.

A partially-stirred reactor (PaSR) code has been used to identify time-step and mixing-frequency dependence. Initial comparisons were made between a 77-step mechanism for methane combustion and a 25-step reduced mechanism using the PaSR code. The partially-stirred reactor (PaSR) is a zero-dimensional idealistic reactor that attempts to account for both mixing and chemical kinetics. In this sense, the PaSR spans

the gap between a perfectly-stirred reactor (PSR), where mixing is instantaneous, and a plug flow reactor (PFR), where no axial mixing occurs. The mixing time or frequency is specified in the PaSR. For very slow mixing times, the PaSR approaches a PFR, while for very fast mixing times, the PaSR predictions approach those of a PSR.

The PaSR can be used with a choice of mixing models, such as the Curl's mixing model (Curl, 1963) and the IEM (Interaction-by-Exchange with the mean) model (Correa, 1993). Both these models express the unmixedness in terms of the "flow-through residence time" (τ_{res}) and the "mixing frequency" (ω_{mix}) (Chen, 1993).

The PaSR model used in this research was obtained from Prof. J.Y. Chen (U. C. Berkeley). This model includes both of the mixing models described above, and can be used in either mode. The PaSR code was used to evaluate two kinetic schemes for methane combustion. The aim of this preliminary study was to evaluate the dependency of the PaSR on 1) the time-step size and 2) the mixing frequency. The IEM mixing model was used for this purpose. For the first set of numerical experiments using the PaSR code, time-step size was varied and the mixing frequency was held constant at 5000 Hz. In the second set of numerical experiments, the time step was held constant at 1×10^{-5} s, and the mixing frequency was varied. Both sets of numerical experiments were conducted with a full mechanism having 77 steps and 27 species (Miller and Bowman, 1989) and a 25-step, 16-species skeletal mechanism (Smooke, 1991).

Figure 8 shows the effects of time-step size and mixing frequency on PaSR calculations of CO mass fraction as a function of residence time. The time step varied here is not the time step of integration of the time-dependent chemical reactions, but is the time step which governs removal of individual reacted "particles" from the reactor. These calculations were performed for a fuel-lean premixed methane-air case simulating a gas turbine combustor ($\phi = 0.38$, $T_{init} = 1200$ K, 30 atm) using the 25-step mechanism of Smooke (1991). It is seen in Figure 8a that for small time-steps, the CO mass fractions converge fairly well, but that the code fails for large time-steps. This shows that the IEM model is sensitive to time-step

Figure 8. PaSR calculations of CO mass fractions vs. time for premixed CH₄/air combustion as a function of (a) time-step size ($\omega = 5000$ Hz), and (b) mixing frequency ($\Delta t = 1 \times 10^{-5}$ s). Calculations performed using the IEM mixing model, a 25-step CH₄/air kinetic mechanism (Smooke, 1991), $T_{\text{init}} = 1200$ K, $\phi = 0.38$, $P = 30$ atm, and 650 fluid particles.

size, and that small time-steps must be used. The set of numerical experiments performed by varying mixing frequency, holding step size constant at 1×10^{-5} s, are shown in Figure 8b. Figure 8b shows that the steady-state value of CO mass fraction converges to a uniform value at high mixing frequency, which is presumably near the limit calculated using a PSR. Similar results were published by Correa (1993).

After performing the demonstration sets of PaSR calculations, the PaSR code was used to compare the two sets of methane-air reaction mechanisms. Calculations were performed using a mixing frequency of 5000 Hz, the IEM mixing model, and a time step of 2×10^{-5} s. Mean concentrations of CH₄ and CO calculated with the PaSR code are shown in Figure 9 as a function of time. Although the steady-state behavior is the most relevant comparison, the two calculations show some time-dependent similarities, such as the initial rise in concentrations followed by a slower decrease that leads to convergence. Also, the rate of convergence to steady-state for both mechanisms is similar.

Additional numerical experiments were performed to show the effects of both time-step size and mixing frequency on the PaSR calculations for each of the two kinetic mechanisms. PaSR calculations of mean temperature and mass fractions of CH₄ and CO are shown in Figure 10 as a function of the time-step size for both mechanisms and in Figure 11 as a function of frequency. In Figure 10 at the small time steps necessary for accurate predictions (10^{-5} s), only small variations in calculated temperature are seen, but differences in steady-state mass fractions of CH₄ and CO were 10% and 30%, respectively. In the calculations versus mixing frequency (Figure 11), the disagreement in the calculated CO between the two mechanisms is larger at the high frequencies (PSR limit) than at the low frequencies (PFR limit). Calculations of CH₄ and temperature show fairly good agreement over a wide range of frequencies. Predictions of radical species in the PaSR as a function of frequency have been shown for the 77-step mechanism (Figure 11). These results were very similar to the results obtained by Correa and Braaten (1993).

Task 2 - Code Development

The objective of Task 2 is to develop an advanced, 3-dimensional gas turbine combustor model from the foundation of ACERC's existing 3-D code, PCGC-3. Task 2 is closely coordinated with an independently funded DOE/METC (DE-AC21-93MC30040) study being jointly conducted by Advanced Fuel Research, Inc. (AFR), and BYU/ACERC, which has a task for conducting gaseous combustion modeling. A developmental plan for the advanced gas turbine combustor model was adopted. The approach is to use Monte Carlo PDF, finite-rate chemistry. Codes from J.Y. Chen (U.C. Berkeley) and NASA were obtained and calculations performed. Further, the feasibility of applying domain composition methods to CFD codes has been investigated and several runs made. The Mississippi State University (MSU) CFD platform has been investigated and work has been performed under independent funding at BYU to incorporate ACERC combustion submodels into the widely used FLUENT CFD code.

The major components in the development of a 3-D gas turbine combustor code are shown in Figure 12. Code development begins with the first generation ACERC 3-D coal and gas combustion code 93-PCGC-3. The NASA pdf submodels are being considered as the basis for an improved pdf submodel in PCGC-3. Combustion and NO_x mechanisms with a table lookup procedure for chemical rates and properties are important elements of the pdf submodel. The combustion mechanism must be a reduced mechanism (five or fewer scalars) capable of predicting phenomena of interest, such as ignition, extinction, and CO conversion. Currently, it is thought that the NO_x mechanism will be implemented separately as a post-processor. The partially-stirred reactor (PaSR) code and a parabolic pdf jet flame code (JETFLAME) (with associated Sandia Chemkin code interface for thermodynamics and reaction rates) obtained from J.Y. Chen (UC Berkeley) are being used to evaluate reduced mechanisms and to implement a table look-up procedure in the 3-D code. It is anticipated the linear eddy model for turbulent mixing being developed by McMurtry (U. of Utah) in ACERC will be implemented. After extensive evaluation with data obtained in the

Figure 9. PaSR calculations of (a) CH_4 and (b) CO mass fraction vs. time for premixed CH_4/air combustion using a 25-step mechanism (Smooke, 1991) and a 77-step mechanism (Miller and Bowman, 1989). Calculations performed using the IEM mixing model, a 25-step CH_4/air kinetic mechanism (Smooke, 1991), $T_{\text{init}} = 1200 \text{ K}$, $\phi = 0.38$, $P = 30 \text{ atm}$, 650 fluid particles, $\Delta t = 2 \times 10^{-5} \text{ s}$, and $\omega = 5000 \text{ Hz}$.

Figure 10. PaSR calculations of (a) Temperature, (b) CH₄, and (c) CO mass fraction vs. time-step size for premixed CH₄/air combustion using a 25-step mechanism (Smooke, 1991) and a 77-step mechanism (Miller and Bowman, 1989). Calculations performed using the IEM mixing model, a 25-step CH₄/air kinetic mechanism (Smooke, 1991), $T_{init} = 1200$ K, $\phi = 0.38$, $P = 30$ atm, 650 fluid particles, and $\omega = 5000$ Hz.

Figure 11. PaSR calculations of (a) Temperature, (b) CH_4 , and (c) CO mass fraction vs. mixing frequency (ω) for premixed CH_4/air combustion using a 25-step mechanism (Smooke, 1991) and a 77-step mechanism (Miller and Bowman, 1989). Calculations performed using the IEM mixing model, a 25-step CH_4/air kinetic mechanism (Smooke, 1991), $T_{\text{init}} = 1200 \text{ K}$, $\phi = 0.38$, $P = 30 \text{ atm}$, 650 fluid particles, and $\Delta t = 2 \times 10^{-5} \text{ s}$.

into PCGC-3. ACERC has undertaken its own program to implement BFC as well as collaborations with Mississippi State University and Fluent, Inc. Mississippi State University has a NSF-funded engineering research center on Computational Field Simulation that has codes with advanced gridding techniques. FLUENT has incorporated BFC and represents an alternative approach for modeling gas turbine combustors using ACERC combustion technology. These projects are independently funded but are closely coordinated with this project.

2.2 - Incorporation of Submodels. The objectives of this subtask are: 1) to incorporate the best-available, proven, advanced code submodels from the broad, on-going ACERC research program, from the literature, from known developers of submodels, and those developed in Task 1, and 2) to perform systematic checks of the code to verify accuracy of code programming, and appropriateness of newly installed improved submodels.

The partially-stirred reactor (PaSR) model was used to investigate the Monte Carlo pdf method for chemistry/turbulence interactions. Calculations were made for conditions typical of LPC in stationary gas turbine combustors (premixed methane in air, 700 K inlet, 0.5~0.6 equivalence ratio, 15 atm, 10-20 ms residence time, and 100-5000 s^{-1} mixing frequency, where mixing frequency is an input parameter to the model). These results were extended to a wider range of frequencies and supplemented with plug-flow and perfectly-stirred reactor calculations to see if the PaSR predictions approach the correct limits at low and high frequencies. Calculations were also performed using a new methane combustion scheme (GRI-Mech 1.1) obtained from SRI International (Smith, 1994). The GRI mechanism produced results similar to those obtained previously with a 77-step scheme based on the work of Miller and Bowman (1989).

The NASA-Lewis probability density function code, RPLUS/LPDF2D (Hsu et al., 1994), which applies to two-dimensional, high-speed flows, was obtained, installed and successfully run on the HP-735 workstation. Some minor code modifications were necessary. The LPDF2D code is a Monte Carlo pdf code

which has been designed as a submodel to easily interface with existing flow codes. It was incorporated as a submodel into the ACERC code PCGC-3 and initial runs performed. Since LPDF2D only works for two dimensions, the submodel is restricted to the two-dimensional, axisymmetric option in PCGC-3. Also, it is restricted to hydrogen-air combustion, which is hardwired into the code.

Task 3 - Industry and AGTSR Interaction

Significant efforts are being made to coordinate this program with the affiliated industrial manufacturers and other participants in the AGTSR program.

3.1 - Industry Coordination. The technical advisory committee (TAC), composed of the manufacturers and users of industrial and power gas turbine systems, have provided specific recommendations to the conduct of the research. The members of the TAC have been contacted by telephone and mail, and have been sent copies of the quarterly reports. A survey was prepared and sent to the members, and responses received and analyzed. Visits have been made by BYU researchers to the various turbine manufacturers. BYU researchers have also participated in several conferences and short courses. Papers were presented on related work at the joint ASME/IGTI conference at The Hague (Hedman et al., 1994) and the recent Combustion Symposium (International) (Hedman and Warren, 1994).

3.2 AGTSR Participant Coordination. An effort has been made to coordinate with other participants in the AGTSR program to determine what effect the advances in each research area might have on the design and performance of the gas turbine combustor. Contact with the other university ATS programs has been initiated and maintained. BYU investigators presented their work at the ATS conference at Nashville and held a meeting with Technical Advisory Committee members and other key experts. Research team members also presented papers at the annual ACERC conference.

In addition to the work being performed under this contract, research at BYU on advanced

gas combustion models is being supported by other funding. The Department of Energy, through the Morgantown Energy Technology Center, is funding a collaborative project by Advanced Fuel Research (AFR) of East Hartford, CT, and BYU (contract number DE-AC21-93MC30040) which focuses specifically on the development of advanced submodels for turbulent, gaseous combustion. This study is referred to in this report as the AFR study. Further, ACERC is supporting several initiatives in adapting PCGC-3 to complex geometries and turbulence that complement work on this project. Other work has also been performed by ACERC researchers that can be applied to this project.

FUTURE WORK

The first of four years in the study have just been completed. In order to complete the planned model, much work remains to be completed. In Task 1, predictions of the Task 150 non-premixed combustor will continue, with emphasis on different inlet velocity profiles in an attempt to predict the observed radial flame thickness. Experimental plans include the complete mapping of velocity, temperature, and N_2 , O_2 , CO , and CO_2 species concentrations for various configurations of the ATS premixed natural gas/air burner using LDA and CARS. In addition, PLIF images of OH and CH in the burner will be obtained. As part of the study to evaluate submodels, several reduced mechanisms will be evaluated with the PaSR code and a PSR code. Results of calculations with reduced mechanisms will be compared to those obtained using a full mechanism. The major focus of these calculations will be on predicting NO_x and CO in lean premixed CH_4 /air combustion. The search for additional reduced mechanisms proposed in the literature and by other contractors in the AGTSR program will be continued. Work on Task 2 will focus on the incorporation of smooth-wall boundaries into the code. Various turbulence and turbulence/chemistry interaction submodels will be considered for implementation into PCGC-3 and incorporated if desirable: work will continue on the incorporation and evaluation of the LPDF2D submodel. Coordination with the other ACERC projects will be continued.

ACKNOWLEDGEMENTS

This report was prepared with the support of the U.S. Department of Energy, Morgantown Energy Technology Center, Cooperative Agreement No. DE-FC21-92MC29061. This financial support through the South Carolina Energy Research and Development Center (Mr. Daniel Fant, Program Director, and Dr. Lawrence P. Golan, SCERDC Director) is greatly appreciated. Students who have participated on this project include Steven Cannon, Robert Dawson, Jason Haslam, Hemant Mallampalli, Robert Murray, Craig Phillips, Stephan Schmidt, and David Warren. The services of Michael King and Joy Critchfield, ACERC staff members, are gratefully acknowledged. References are made in this report to work performed under separate funding agreements with the Advanced Combustion Engineering Research Center (with the principal grant from the National Science Foundation's Engineering Centers Division), Brigham Young University, the Air Force, and the DOE/METC with Advanced Fuel Research, Inc.. Their support is appreciated.

REFERENCES

- Chen J.Y., "A General Procedure for Constructing Reduced Reaction Mechanisms with Given Independent Reactions" *Combustion Science and Technology*, 57, pp. 89- 94 (1988).
- Chen, J.Y., "Stochastic Modeling of Partially Stirred Reactors," presented at the Fall Meeting of the Western States Section of the Combustion Institute, Menlo Park, CA, Paper no. 93-071 (October, 1993).
- Correa, S.M. and Braaten, M.E., "Parallel Simulations of Partially-Stirred Methane Combustion," *Combustion and Flame*, 94, 469-486 (1993).
- Correa, S.M., "Turbulence-Chemistry Interactions in the Intermediate Regime of Premixed Combustion," *Combustion and Flame*, vol. 93, pp. 41-60 (1993).

- Curl, R.L., "Dispersed Phase Mixing: 1. Theory and Effects in simple Reactors," *AIChE Journal*, 9, 175-181 (1963).
- Hautman, D.J., Dryer, F.L., Schug, K.P., and Glassman, I., "A Multi-Step Overall Kinetic Mechanism for the Oxidation of Hydrocarbons" *Combustion Science and Technology*, 25, pp. 219-235 (1981).
- Hedman, P.O., Sturgess, G.J., Warren, D.L., Goss, L.P., and Shouse, D.T., "Observations of Flame Behavior from a Practical Fuel Injector Using Gaseous Fuel in a Technology Combustor," presented at the International Gas Turbine and Aerospace Congress and Exposition, The Hague, Netherlands, ASME 94-GT-389 (1994).
- Hedman, P.O., and Warren, D.L., "Turbulent Velocity and Temperature Measurements from a Gas-fueled Technology Combustor with a Practical Fuel Injector," 25th Symposium (International) on Combustion, University of California at Irvine, Irvine, California (July 31-August 5, 1994)
- Hsu, A.T., Raju, M.M., and Norris, A.T., "Application of a PDF Method to Compressible Turbulent Reacting Flows," paper presented at the 32nd Aerospace Sciences Meeting and Exhibit, Reno, NV, Jan. 10 (1994), available from AIAA, Washington, DC, as AIAA 94-0781.
- Jones, W.P. and Lindstedt, R.P., "Global Reaction Schemes for Hydrocarbon Combustion" *Combustion and Flame*, 73, pp. 233-249 (1988).
- Maas, U. and Pope, S.B., "Implementation of Simplified Chemical Kinetics based on Intrinsic Low-Dimensional Manifolds" 24th Symposium (International) on Combustion, The Combustion Institute, Pittsburgh, PA, pp. 103-112 (1992b).
- Maas, U. and Pope, S.B., "Simplifying Chemical Kinetics: Intrinsic Low-Dimensional Manifolds in Compositional Space" *Combustion and Flame*, 88 pp. 239-264 (1992a).
- Miller, J.A. and Bowman, C.T., "Mechanism and Modeling of Nitrogen Chemistry in Combustion," *Progress in Energy and Combustion Science*, 15, 237-338 (1989).
- Seshadri, K. and Peters, N., "The Inner Structure of Methane-Air Flames," *Combustion and Flame*, 81, 96-118 (1990).
- Smith, G.P., SRI International, letter dated 27 April 1994.
- Smooke, M.D., "Formulation of the Premixed and Nonpremixed Test Problems," in *Reduced Kinetic Mechanisms and Asymptotic Approximations for Methane-Air Flames*, M.D. Smooke (Ed), Springer-Verlag, 1-28 (1991).
- Wang, H. and Frenklach, M., "Detailed Reduction of Reaction Mechanisms for Flame Modeling" *Combustion and Flame*, 87 pp. 365-370 (1991).
- Warnatz, J., "Detailed Chemistry of Hydrocarbon Combustion and Its Coupling with Flow Processes" *Turbulence and Molecular Processes in Combustion*, Elsevier Science Publishers B.V., Amsterdam, The Netherlands (1993).
- Westbrook, C.K. and Dryer, F.L., "Simplified Reaction Mechanisms for the Oxidation of Hydrocarbon Fuels in Flames" *Combustion Science and Technology*, 27, pp. 31-43 (1981).

REFERENCES

**LEAN PREMIXED COMBUSTION STABILIZED BY RADIATION FEEDBACK AND
HETEROGENEOUS CATALYSTS**

Contract Number: 93-01-SR007

Contractor: The Regents of the University of California
c/o Sponsored Projects Office
University of California
Berkeley

Principal Investigator: Professor Robert W. Dibble

Co-Principal Investigators: Professor Jyh-Yuan Chen
Professor Robert F. Sawyer

Student Researchers: Tami Bond
C.P. Chou
Brad Edgar
Rajiv Mongia
Ryan Nogouchi

Project Manager Dr. Jack Halow
Dr. Norm Holcomb

Period of Performance: October 1, 1993 to September 30, 1996

LEAN PREMIXED COMBUSTION STABILIZED BY RADIATION FEEDBACK AND HETEROGENEOUS CATALYSTS

Abstract

We report progress in several areas. We have numerically modeled the two stage combustion approach presented by Catalytica, Inc., and the different two stage combustion presented by Toshiba Corporation. Our results agree with published data at a pressure of 1 atmosphere. Our models, when extended to 12 atmospheres show significant effects of increased collision rate and reduced diffusivity. Experimentally, we have ignited methane in a Platinum based catalyst system. For this system, we use a novel approach to 'preburners' by adding hydrogen directly to the flow which brings the catalyst to light-off temperature for methane. On a related task, we have stabilized ultralean combustion ($\phi = 0.3$) by using radiative feedback from a ceramic mesh which is convectively heated by the combustion products. We are exploring the use of thermal-shock insensitive hollow ceramic spheres for both radiation control and catalyst support.

Table of Contents

1. Objectives
2. Background Information
 - 2.1. Need for NO_x Reduction
 - 2.2. The Ultra-Low Emission Combustor
 - 2.3. Catalytic and Noncatalytic Combustion
 - 2.4. Considerations in Catalytic Combustion
3. Results: Catalytic Combustion Modeling

- 3.1. Introduction
- 3.2. Surface chemistry
- 3.3. Gas-phase chemistry
- 3.4. Integration of Surface Chemistry with Gas-Phase Combustion

4. Results: Catalytic Combustion Experiments

- 4.1. Introduction
- 4.2. Catalyst Selection
- 4.3. Reactor Construction
- 4.4. Preliminary Results
- 4.5. Diagnostics

5. Results: Investigation of Ceramic Spheres

- 5.1 Background
- 5.2. Calculated and measured pressure drops

6. Technology Transfer

7. References

1. OBJECTIVES

The objective of the program is to aid industry in the design and implementation of advanced gas turbine combustion schemes that will respond to increasing demands for high efficiency, ultra-low NO_x operation.

Our program is focused primarily on the combustion portion of the gas turbine system. We will consider the benefits of combustion stabilized by catalytic and non-catalytic porous media.

The primary goal of our research is to develop generalized models, that will capture the effects of both gas phase and surface reactions in the combustion chamber. These models will aid industry in designing state of the art combustion systems.

To this end we are considered approaches to ultra-low NO_x combustion based on catalytic and non-catalytic porous materials

2. BACKGROUND INFORMATION

This report discusses the motivation for research and our progress in this first year of research funded by the Department of Energy under the Advanced Turbine Systems project.

In this section, we discuss some current strategies for reducing emissions of oxides of nitrogen and present the idea of achieving such emission reduction with the use of catalytic and noncatalytic porous combustors. Catalytic combustion is addressed first in this report. Section 3 reviews some considerations required in experimentation and modeling of catalytic combustion. In Section 4, we discuss our modeling efforts to date and present some results. Section 5 contains a description of our experimental work and some results.

Section 6 contains results from an investigation of hollow ceramic spheres which may be used in either catalytic or noncatalytic porous burners.

This research is intended to provide better numerical models for the gas turbine industry and the combustion community, and we review our interactions with those sectors in Section 8.

2.1. Need for NO_x Reduction

Gas-turbine based systems are becoming the preferred and dominant approach to gaseous and liquid fossil and biomass fueled electric power generation. As coal gasification progresses, gas turbines will become important in the generation of electricity from coal as well. The repowering of large, existing central station steam power plants

frequently utilizes gas turbines in a topping cycle role, increasing both efficiency and power and reducing emissions. In new, distributed and smaller installations, gas turbines offer the possibility of cogeneration of electricity and heat with increased efficiency and reduced pollutant emissions.

The most important problem facing combustion-based electric power generation is the control of air pollutants. Increasingly stringent emission standards for oxides of nitrogen are forcing the use of technologies, such as selective catalytic reduction, which are expensive, reduce efficiency, and introduce new potential environmental problems.

New source performance standards for stationary gas turbine oxides of nitrogen are 75 to 150 ppm at 15% O₂, depending on size and with some allowance for nitrogen-containing fuels. Regions experiencing severe photochemical smog are promoting the application of Best Available Control Technology (BACT). The South Coast Air Quality Management District has set NO_x BACT emissions at 9 ppm (15% O₂) based on selective catalytic reduction (SCR) technology. BACT for carbon monoxide is set at 10 ppm (15% O₂).

2.2. The Ultra-Low Emission Combustor

In most gas turbine cycles, combustor exit temperatures are constrained to the material limitations of the power turbine. The high temperatures achievable in the flame are well in excess of this temperature limit. Traditional gas-turbine combustion requires that fuel and air mix and burn at the same time. At combustor pressures, the temperature of the combustion products is typically 4400F (~2700K), while the temperature acceptable

to the power turbine is typically less than 2800F (~1800K). Accordingly, the practice has been to cool the hot combustion products from the primary zone with dilution air into the secondary zone of the combustor.

However, in the high temperature primary zone, "thermal" nitric oxide (NO) is formed. In recent years, a combination of air and steam has been injected into the primary and secondary zones of the combustor. The effect, and intent, has been to cool the flame and thus reduce the amount of NO formed. The steam can also be part of a bottoming cycle, improving system efficiency. However, steam injection is limited by the fact that NO decreases are accompanied by unacceptable increases in CO and unburned hydrocarbons UHCs (Malte, 1988).

2.2.1. Dry Low NO_x

Rather than burn at high temperatures, followed by air or steam dilution, one approach is to burn lean by premixing fuel with air and combusting in a homogeneous zone without the need for a significant dilution air component. Because these low NO_x combustors do not rely upon steam injection, they are called Dry Low NO_x (DLN) combustors. In DLN combustors, fuel and air are premixed to a lean condition ($\phi \sim 0.3 - 0.7$) such that the adiabatic flame temperature is low enough that NO_x generation through the thermal route is suppressed. Less air dilution is required, although a certain amount is required to cool various parts of the combustor and power turbine blades.

In DLN, the combustion occurs in the gas phase in the form of lean turbulent premixed flames in combustion chambers. The lean flame is stabilized by recirculation of hot combustion products and then mixing of these products with incoming reactants. Such

devices have done well, with NO_x levels well below 30 ppm (Lovett and Abuaf, 1990). However, in these devices, the coupling between the energy release reactions and the acoustics of the combustor often leads to destructive large pressure fluctuations, known as "dynamics". Also, the combustion instabilities can lead to higher-than-predicted emissions of CO and UHCs (Correa, 1992).

2.3. Catalytic and Noncatalytic Combustion

In this project, we are investigating methods of avoiding the pressure dynamics and achieving even lower levels of pollutant emissions through the use of lean combustion inside high-porosity solid matrices. These matrices suppress both acoustic oscillations and the high temperatures that lead to thermal NO_x production. As discussed below, we intend to explore both catalytic and noncatalytic surfaces with a view to building the best models of combustion reactions in two phases. Such models will prove useful in the design of gas turbine combustors.

2.3.1. Catalytic Combustion

Catalytic combustion has a number of potentially important practical applications. Of special relevance to the present low emission combustors is the effect of a catalytic surface for sustaining combustion at relatively low temperature, thus reducing the opportunity for NO_x formation. Although catalytic combustion was discovered more than 150 years ago, our understanding of surface reactions is far behind our understanding of gas phase reactions. A summary of the state-of-the-art is available in the proceedings of a recent GRI-sponsored International Workshop on Catalytic Combustion held at SRI, Menlo Park, CA, (McCarty, 1993), and the Second International Workshop on Catalytic

Combustion held in Tokyo, Japan (Arai, 1994).

2.3.2. Noncatalytic Combustion

Although most combustion engineers are aware of the possibilities of incorporating catalytic surfaces, the use of noncatalytic porous solids for stabilizing flames has not received as much attention. It has been known for some time that combustion can be supported inside a porous matrix or by a series of stacked wafers of reticulated ceramics or metal. The wafers downstream of the combustion zone will radiate through the combustion zone to the upstream wafers, which then preheat the incoming premixed reactants. Unlike catalytic combustion, the matrix material is chemically "inactive", and it serves primarily as a heat sink in the hot sections and as a heat source in the preheat sections. Some commercial burners using this concept are now available for appliances; the potential use in gas turbine applications is still under exploration.

Examples of industrial burners include the low NO_x noncatalytic radiant surface burners marketed by Alzeta Inc. (Pyrocore), and Eclipse Inc. (Bilayer Ceramic Foams). More recently, lean combustion **inside** of packed beds of high temperature ceramics has demonstrated low NO_x (e.g. Viskanta 1991; Sathe S. B., Peck R. E. and Tong T.W. 1990). The temperature inside of the packed bed can be above the adiabatic flame temperature. Such superadiabatic temperatures are a manifestation of radiant heat transfer from the product end of the bed toward the reactant part of the bed. Issues of special significance to gas turbine combustion are the performance of radiative feedback at elevated pressures, the ability to limit acoustic oscillations, and the durability and thermal-shock resistance of the ceramic materials.

2.4. Considerations in Catalytic Combustion

Catalysts for combustion of natural gas typically consist of either a noble metal (platinum or palladium, as discussed by Dalla Betta (1994)) or a ceramic such as manganese-substituted hexaaluminate (Machida et al., 1994). Each type of material has its advantages and disadvantages. Ceramics are more durable at elevated temperatures, but require a higher temperature to initiate catalytic activity. Noble metals are catalytically active at temperatures as low as 350 °C. However, if exposed to gas-turbine inlet temperatures (about 1300 °C), noble metals will evaporate in a short period of time due to their high vapor pressure at those temperatures.

One solution to alleviate the problem of metal evaporation is to allow only a portion of the combustible mixture to react within a noble-metal catalyst, thus avoiding catalyst exposure to high temperatures. The remainder of the combustion takes place either in the gas phase, as proposed by Dalla Betta et al. at Catalytica (1994) and Furuya et al. at Toshiba (1994), or over ceramic catalysts as proposed by Sadamori et al (1994) at Osaka Gas.

There are two possible methods of keeping a noble-metal catalyst at a moderate temperature, both of which we investigate in this project. The first method is that explored by Dalla Betta et al. (1994) at Catalytica involving the use of the catalyst itself to limit the substrate temperature. The technique takes advantage of the tendency of palladium oxide (PdO) to dissociate and form palladium (Pd) at a moderate temperature (about 800 °C). Because PdO is the more catalytically active species, surface reaction sites become limited when the temperature reaches the

dissociation temperature of PdO. The temperature of the catalyst is thus prevented from rising above that temperature. A test rig used for such experiments is shown in Figure 2-1.

In this scheme, gas-phase combustion in the catalyst section is avoided only because the low temperature and lean mixture do not permit gas-phase thermal ignition. However, if the fuel and air are not well mixed, pockets which are more fuel-rich may autoignite, increasing the temperature and melting the catalyst.

Figure 2-2, also from a report by Catalytica, emphasizes the use of catalyst properties to limit temperature. This combustor consists of three stages: two catalytic stages followed by gas-phase combustion. The properties of the catalyst are different in each catalytic stage and alter the maximum allowable surface temperature.

The second method for limiting catalyst temperatures involves limiting the heat release possible within the catalyst by providing a very lean fuel-air mixture to the catalyst. This method is used at Toshiba by Furuya et al. (1994). In this system, shown in Figure 2-3, additional fuel is injected after the catalyst section and burned in the gas phase to achieve the desired turbine inlet temperature. In the post-catalyst injection system such as proposed by Toshiba, the extremely lean fuel-air mixture within the catalyst prevents excessive temperatures. Because the catalyst surface temperature is not self-limited, a nonuniform mixture can create a hot spot on the catalyst, resulting in thermal stresses. This requires, that the gas mixture be well mixed before entering the catalyst section. After the catalyst, additional fuel is injected into the hot stream of combustion products. The injected fuel will mix with the hot causing the fuel-

product mixture to auto-ignite. *In practice, the fuel may autoignite before it is completely mixed; burning in a non-premixed mode. In the non-premixed mode, local temperatures will be higher than the average final mixture temperature, hence zones of excessive thermal NO production will exist.* This problem may be exacerbated at high pressure and temperature, due to decreased molecular mixing and increased reaction rates.

3. RESULTS: CATALYTIC COMBUSTION MODELING

3.1. Introduction

Many catalytic burners for gas turbine applications consist of several sections: one or two catalyst sections and a homogeneous "burnout" section which may include injection of additional fuel. Numerical modeling of a practical catalytic combustor will include the following physical processes:

- Chemical kinetics on the catalyst surface
- Gas-phase combustion, including the effects of fuel injection, in the post-catalyst region
- Transport effects which describe the interaction between surface and gas-phase chemistry inside the catalyst section
- Radiation Transport

Developing a detailed kinetics model for surface chemistry is one of the principal challenges in modeling catalyst combustion. We are developing detailed mechanisms which will better simulate the fundamental kinetics process. This effort is a necessary step toward building better numerical models.

Modeling of gas-phase combustion processes in the post-catalyst region is important in determining the autoignition delay time, and hence the residence time and combustor length required for complete burnout. Our model calculations with a detailed methane mechanism indicate that the autoignition delay time is reasonably short (<1 ms) at high pressures (10 atm and above).

The gas-phase model is also used to predict whether autoignition and hot spots occur prior to complete mixing when additional fuel is injected into the post-catalyst zone. The results for a high-pressure combustor indicate potential autoignition prior to complete mixing. This problem may limit the range of pressures for which post-catalytic fuel injection is viable. Finally, because NO_x formation occurs primarily in the gas phase, this model also predicts the concentration of NO_x emissions for the catalytic combustor.

Transport of molecules between the catalyst surface and the gas phase is an important aspect in modeling the interaction between the surface and gas-phase reactions. The transport model will predict whether a catalytic process is limited by diffusion to the surface or by kinetics at the surface. Knowledge of the limiting factors in the process will aid in the design of a catalytic system. For example, depending on the activity, the catalyst surface can serve as a flame holder or a flame suppresser by removing radical species from the gas phase.

3.2. Surface Chemistry

Most catalytic combustion experiments reported in the literature use either ceramic supported polycrystalline platinum or ceramic-supported palladium as the catalyst material. Our goal is to model the system of greatest interest to the gas turbine industry:

methane oxidation over a palladium catalyst. We have identified the following salient features of palladium surface reactions:

1. The dissociation of active surface species of palladium oxide into less active palladium crystals at 900 °C (Farrauto et al. 1992; McCarty, Wong and Chang 1993)
2. The absorption of oxygen by palladium, which impairs the catalytic properties of the metal (Campbell, Foyt and White 1977; Ladas, Imbihl and Ertl 1989)
3. Significant hysteresis effects in the temperature-dependent formation and dissociation of PdO (Garbowski et al. 1994; Briot and Primet 1991; Lieske and Volter 1985; Chen and Ruckenstein 1981)
4. The nonlinear dependence of catalytic activity on catalyst loading and metal particle size (Hicks et al. 1990)
5. Metal-support interactions which may alter the catalytic activity (Baldwin and Burch 1990; Baker, Prestidge and McVicker 1984)
6. Variations in reaction sites due to differences in crystalline planes (Wang, Lee and Schmidt 1985; Greenler et al. 1985; Corma, Martin and Perez 1983)

3.3. Gas-phase chemistry

The main function of the catalyst is to serve as a low- NO_x combustor. The catalyst bed converts a portion of the premixed fuel-air inlet stream into combustion products which heat up the gas-phase mixture to a temperature substantially below that of a conventional flame. The hot combustion products leaving the catalyst bed auto-ignite in the second stage of the combustor. This process results in less NO_x than a noncatalytic combustor because maximum gas-phase

temperature is about 1300 °C, at which temperature negligible thermal NO is formed.

Fuel injection into the combustor can take place in either one or two modes: either a single injection at the inlet to the catalyst bed, or the inlet injection followed by a second injection of fuel into the post-catalyst region. We have performed computational studies for both cases with a well-established detailed gas-phase chemical mechanism. To focus on the post-catalyst ignition, we treat the catalytic stage as a "black box". Graphs from these calculations are presented in **Figure 3-1** through **Figure 3-8**.

Figure 3-1 presents the equilibrium temperatures for combustion products versus the equivalence ratio starting with an inlet temperature of 350°C (typical of air exiting the compressor stage of the turbine.) Constrained by a maximum turbine inlet temperature of 1300 °C, we see from the figure that the overall stoichiometry of all fuel and air entering the combustor at all stages must be less than $\Phi = 0.4$.

3.3.1. Single-stage injection mode

In the case of single fuel injection, we are concerned with the autoignition delay time of the partially burned gas-phase mixture after it leaves the catalyst section. Treating the catalyst bed as a device converting a portion of fuel, we computed the temperatures of the gas leaving the catalyst section as a function of the percent conversion of methane within the catalyst bed. The result is depicted in **Figure 3-2**

We model the gas phase combustion as a plug-flow-reactor with a detailed methane mechanism from GRI (Miller & Bowman, 1988), with inlet mixtures corresponding to 70% and 82% fuel conversion inside the catalyst. The gas stream exiting from the

catalyst bed autoignites downstream with a ignition delay time which has a strong dependence on the exit temperature. **Figure 3-3** depicts the dramatic difference in ignition delay between 70% and 82% methane conversion within the catalyst at atmospheric pressure. This suggests that conversion of fuel in the catalyst must reach a certain temperature in order to have complete burn out within the combustor.

Figure 3-4 shows the time histories of NO_x at atmospheric pressure for two different catalytic conversions. Both show the NO concentration level of approximately 0.2 ppm and N_2O of approximately 0.8 ppm. These levels of emission are considerably lower than the conventional gas turbine combustors (~30-50 ppm).

Next, we examine the effect of pressure on the ignition delay. **Figure 3-5** presents the ignition delay time at 10 atmospheres. A comparison with **Figure 3-2** reveals that the ignition delay time is reduced considerably at higher pressure, as expected. **Figure 3-6** shows the NO_x concentration at 10 atmospheres; a comparison with **Figure 3-4** shows that NO formation is nearly unchanged, while much more N_2O is produced.

3.3.2. Two-stage Injection Mode

With the catalyst section operating with a very lean mixture ($f=0.25$), additional fuel must be injected into the post-catalyst region in order to bring the final gas temperature to about 1300 °C. We first model the chemical kinetics of the fuel jet with a modified well-stirred reactor model termed a "well-stirred balloon.". Like a physical balloon, this well-stirred reactor has input but no output, hence that name. In the case the reactor (balloon) initially contains methane only, and the input is hot combustion products ($f=0.25$) entrained

at a constant rate. When the well-stirred balloon reaches an overall equivalence ratio of 0.4, the entrainment process ceases. The subsequent chemical kinetics are modeled by a plug-flow reactor.

Figure 3-7 presents the equivalence ratio of the fuel jet versus time for a model in which the entrainment rate is set so that mixing is complete at 50 ms. Figure 3-8 shows the temperature and NO profiles at ambient and 10 atmosphere pressures. For the atmospheric pressure case, autoignition occurs after mixing has completed, and the maximum gas temperature never exceeds 1300 °C. At 10 atm, the computed temperature history shows that autoignition occurs prior to complete mixing. As a result, the gas phase does experience temperatures higher than 1300 °C and NO production is much higher. NO concentrations are about four to five times greater at the combustor exit at 10 atmospheres than at one atmosphere.

3.4. Integration of Surface Chemistry with Gas-Phase Combustion

The basis of our modeling is the software package CRESLAF (Chemically Reacting Shear Layer Flow) written by Sandia National Laboratories. This program calculates velocity, temperature, and species profiles of chemically reacting flows within two-dimensional channels. The program has both gas phase and surface chemical reactions as well as species transport. We have improved the program to provide an adaptive gridding capability, which places a greater density of computational mesh points in those locations of particularly high sensitivity to small scale phenomena. We have used CRESLAF with a detailed kinetics model for the combustion of hydrogen on a platinum polycrystalline surface.

Future modifications to the CRESLAF architecture may involve its adaptation to three-dimensional channels or packed bed reactor simulations. In the previous sections we have focused on the post catalyst gas phase combustion; where most of the NO_x is formed. We are also developing a model for the catalyst.

4. RESULTS: CATALYTIC COMBUSTION EXPERIMENTS

4.1. Introduction

Our experiments in catalytic combustion are intended to provide understanding of the catalytic process and to provide data for improvement of numerical models. In our experimental work, we are addressing the following issues:

Pressure dependence of the catalytic combustion process. Depending on temperature and pressure, the combustion process may be limited either by mass transport to the catalyst surface or by the kinetics of the reaction on the surface. By understanding the effects of pressure, we will be able to scale results from atmospheric conditions to gas-turbine inlet conditions. In particular, flow in channels is turbulent at gas turbine conditions, so experiments with laminar flows must be avoided or tested very carefully.

Effects of incomplete mixing, both in the fuel-air mixture and in the post-catalyst zone if secondary injection is used. As discussed in Section 2, imperfect mixing can cause catalyst failure in both the catalyst-limited and the post-injection modes, and it can result in higher NO_x emissions in the post-injection mode.

Rate of fuel consumption within the catalyst. In much of the literature to date, noble-metal catalysts, especially palladium, have been modeled as "black boxes" with a given percentage of fuel conversion (e.g. Tsujikawa et al. (1994), Griffin et al. (1994)). The predictions of these models have generally agreed well with experimental NO_x results because most of the NO_x is generated after the catalyst; however, questions regarding low temperature ignition and flow velocity in the channels will be addressed by a model of the catalyst. The goal of our project is to develop a realistic, mechanistic model of catalytic combustion at gas-turbine conditions.

4.2. Catalyst Selection

We have chosen platinum catalysts for our first experiments because of the abundance of experimental data. A successful model of catalytic combustion includes submodels for heat and mass transfer from the channel flow to the substrate surface, gas phase chemistry, surface coverage, and surface chemistry and heat release. The kinetics of methane oxidation over platinum are well-characterized, and a comparison of the model with experimental data will enable us to assess the accuracy of our transfer and gas-phase modeling before introducing the uncertainties in the less well-understood palladium kinetics. Catalytica, Inc. has provided us with two catalysts with different loadings of platinum on cordierite honeycombs.

We are in the process of refining our modeling capabilities for palladium oxide, as discussed in Section 3, and we will perform experiments with those catalysts as the model is developed. Palladium catalysts are of greater interest to the gas turbine industry than platinum catalysts.

4.3. Reactor Construction

In the interest of obtaining temperature and species profiles along the catalyst, we have constructed a reactor in which the catalyst is segmented into wafers. Gaps between each wafer allow sampling access. Several researchers, including Sadamori et al. (1994) and Hayashi, Yamada and Shimodaira (1994), have performed experiments with segmented catalysts. However, neither species concentrations nor visual observations, made possible by the segmentation, are reported in the literature.

Our first reactor, which is round and internally insulated, has been constructed and operated. This reactor should be sufficient for operation at pressures up to 10 atmospheres. A schematic of this reactor is shown in **Figure 4-1**. The air is preheated to about 350 °C with an electric heater, and the fuel is injected and mixed after the preheater so the mixture will not ignite on the heater coils. Before entering the catalyst section, the fuel-air mixture passes through a flow straightener, which also serves to capture heat radiated from the face of the first catalyst.

The reactor section is a pipe containing two layers of insulation, the inner one of which is removable from the reactor and holds the catalyst wafers in place. Ports for thermocouples and suction-probe sampling are located between each catalyst section. We insert the probes radially instead of axially so that we can measure radial temperature profiles and determine uniformity across the catalyst section. A second set of ports will be introduced at 90° to the thermocouple and sampling ports to provide optical access.

During operation of the internally-insulated combustor, we noticed that understanding

both the steady-state and light-off combustion processes required visual observation of the catalyst wafers. We have therefore constructed another reactor of quartz, which contains four catalyst wafers wrapped with high-temperature insulation and held in the quartz tube by a pressure fit. Figure 4-2 shows a schematic of this reactor. The quartz reactor operates at 1 atm and is nearly adiabatic; about 8% of the chemical energy released is lost from the catalyst section.

The preheat for the quartz-reactor system is provided by a small amount of hydrogen gas (about 4% of the total flow rate) combined with the premixed fuel and air. At that flow rate, the hydrogen provides a temperature increase of about 330 °C, corresponding to a preheat temperature of 360 °C. The activity of hydrogen on platinum is very high, and the hydrogen is completely consumed on the first catalyst wafer.

We are modifying the quartz reactor to include a suction-probe sampling port between each wafer. We will also use palladium catalysts and vary the catalyst activity. By performing several runs with the quartz reactor, we will gain familiarity with the behavior of each catalyst section at different equivalence ratios and flow rates. We will then reinstall the catalyst into the insulated reactor and conduct experiments at pressures up to 6 atmospheres.

4.4. Preliminary Results

Table 4-1 shows conditions for two runs with the quartz reactor. The conditions are similar for the two experiments except for the equivalence ratio, which was fairly low ($\phi=0.15$ and $\phi=0.26$). The "preheated" temperature of 360 °C is the temperature which would have been achieved if the hydrogen in the mixture had been completely burned prior to any methane combustion.

Other experiments in our laboratory have shown that this is about the lowest "preheat" temperature at which methane burns on a platinum catalyst.

Temperatures along the catalyst section were measured with thermocouples between the wafers. The gap between each pair of catalyst wafers was 2 cm. Axial temperature profiles for these experiments are shown in Figure 4-3; the thermocouple readings are uncorrected for radiation to or from the catalyst faces or the surroundings.

Temperatures rise initially because of energy release from the reaction. Farther down the catalyst, fuel is consumed and the reaction ceases. The temperature drops due to heat loss by convection and radiation. However, when catalyst wafers which face each other are in thermal equilibrium, the radiative heat loss is small because both are at the same temperature and the view factor is near 1. The greatest drop in temperature occurs across the last catalyst wafer, because one side of that section radiates to the surroundings. Such observations dictate that our catalyst model will require radiation transport.

4.5. Diagnostics

As a method of determining methane concentrations between the catalyst wafers and hence the extent of combustion, we are investigating absorption spectroscopy using an infrared Helium-Neon laser at 3.39 μm . This IR laser line is fortuitously absorbed by methane. We have conducted experiments to determine the dependence of this method on temperature and pressure and will use it to measure methane concentrations when optical access ports are installed in our research combustor.

Table 4-1. Conditions for catalyst experiments

	Run 1	Run 2
Flow rates (L/min)		
Natural gas	2.8	5.1
Air	202.1	207.6
Hydrogen	8.8	8.8
Equivalence ratio for CH ₄ *	0.15	0.26
Flow speed (m/s at 25°C, 1 atm)	1.76	1.82
Inlet temperature (°C)	28	26
Preheated temperature (°C)	360	360
Pressure (atm)	1.0	1.0

* Based on O₂ present after H₂ burn

5. RESULTS: INVESTIGATION OF CERAMIC SPHERES

5.1. Background

Ceramic materials have attractive characteristics at the high temperatures normally encountered in gas turbines. However, available ceramic structures--honeycombs or foams-- are not often used as catalyst supports because of thermal shock problems. Recently, thin-walled spheres which are believed to better withstand severe thermal shocks have been developed. We have investigated some properties of these spheres to determine whether they would be viable as a catalyst support.

The spheres we tested are monosized, thin-walled hollow "Aerospheres" approximately 2.5 mm in diameter. These spheres can be bonded together to form a ceramic foam which has several properties that make it attractive for use as a catalyst support (Cochran, 1993). The foam can be shaped into complex geometries in this bonding process. This foam can withstand severe thermal shock conditions; it has a lower thermal expansion coefficient than a honeycomb constructed of the same material.

If the foam did fail due to thermal shock, the resulting fragile pieces should shatter upon impact with the turbine blades without damaging them. The foam could be used as a high-temperature catalytic support, or the fabrication process may allow construction of the Aerospheres themselves out of a catalytically active ceramic.

5.2. Calculated and measured pressure drops

A major concern in the use of packed spheres in a gas turbine application is the large pressure drop through a packed bed. Of the several correlations developed relating the pressure drop and flow conditions through a packed bed of spheres, the expressions in Equations 1 and 2 are most commonly used for a wide range of Reynolds number ratios (Froment, 1990).

$$f = 6.8 \frac{(1 - \varepsilon)^{1.2}}{\varepsilon^3} \text{Re}^{-0.2} \quad (1)$$

$$\frac{\Delta P}{L} = \frac{fv^2 \rho_g}{d_p} \quad (2)$$

where ε is the void fraction, v is the free stream velocity, ρ_g is the gas density, d_p is the sphere diameter, and Re is a Reynolds number based on the sphere diameter.

We measured the pressure drop across a bed of spheres by passing air through a 2-cm section of packed spheres. Figure 6-1 compares the observed pressure drop to the correlation shown in Equations 1 and 2 with the void fraction varied between 0.44 and 0.48. The experimental data most closely matches the calculated pressure drop for a void fraction of 0.46.

Pressure drops at gas turbine conditions of higher temperature and pressure are obtained by using the correlation given in Equations 1 and 2. Figure 6-2 compares the pressure drop for a packed bed of spheres to that of a 200 cpi (cells per inch) honeycomb support. The pressure drop of the honeycomb support is modeled as both laminar flow and turbulent flow through a series of square channels.

Although the pressure drop for the packed spheres at 10 m/s is higher than is seen in currently operating systems, the pressure drop can be significantly reduced by decreasing the velocity of the gas passing through the packed bed. This can be accomplished by increasing the cross-sectional area of the bed while holding the volumetric flowrate constant.

5.3. Qualitative thermal shock characteristics

In order to study the thermal shock resistance of the Aerospheres, alumina spheres were subjected to several temperature cycles. The spheres were heated to 700 °C with the exhaust of a combustor. After reaching the maximum temperature, the spheres were allowed to cool to 40 °C. After nine temperature cycles, the spheres had no visible thermal shock damage when compared to fresh spheres. We will perform the same test on a foam of hollow spheres to determine if it exhibits the same thermal shock characteristics.

6. TECHNOLOGY TRANSFER

During the course of this year, we have had several opportunities to interact with the gas turbine industry and with the combustion community. Professors Dibble and Chen gave results from our modeling and our research combustors at the DOE contractors' meeting

held in Nashville, Tennessee in February 1994. Results from the radiative-feedback combustor were presented as a work-in-progress poster at the International Combustion Symposium in Irvine, California in August 1994.

Professor Dibble attended the Second International Workshop on Catalytic Combustion in Tokyo, Japan in April of this year. At this meeting, state-of-the-art work was presented by researchers from several countries.

Finally, we have held meetings with several experts in the field of catalytic combustion. In particular, we have made two visits to the facilities at Catalytica, Inc., and Ralph Dalla Betta has provided us with catalysts for our research. We have also met with Jon McCarty at SRI, who has given us advice on the manufacture of catalysts used in combustion.

7. REFERENCES

- Baker, R.T.K., Prestidge, E.B. and McVicker, G.B. (1984). "The interaction of palladium with alumina and titanium oxide supports," *Journal of Catalysis* 89 (1984) 422-32
- Baldwin, T.R. and Burch, R. (1990). "Catalytic combustion of methane over supported palladium catalysts: II. Support and possible morphological effects *Applied Catalysis* 66 (1990) 359-381.
- Briot, P. and Primet, M. (1991). "Catalytic oxidation of methane over palladium supported on alumina: Effect of aging under reactants," *Applied Catalysis* 68, 301-314.
- Campbell, C.T., Foyt, D.C. and White, J.M. (1977). "Oxygen penetration into the bulk of palladium," *Journal of Physical Chemistry* 81, 491-494.

- Chen, J.J. and Ruckenstein, E. (1981). "Spreading and surface tension gradient driven phenomena during heating of alumina-supported palladium crystallites in oxygen," *Journal of Physical Chemistry* **85**, 1606-1612.
- Cochran, J.K., Lee, K.J., and Hwang, T.J., "Ceramic Foams for Radiant Burners," presented at the Gas Research Institute Radiant Structure Meeting, Chicago, Ill, Oct. 13, 1993.
- Corma, A., Martin, M.A. and Perez, J. (1983), "Determination of crystallographic planes on the surface of supported metallic crystallites," *Journal of the Chemical Society, Chemical Communications*, 1512-3
- Correa, Sanjay M. (1992). "Carbon Monoxide Emissions in Lean Premixed Combustion," *Journal of Propulsion and Power*, **8**, 1144-51.
- Dalla Betta, R.A., Schlatter, J., Chow, M., Yee, D.K., and Shoji, T. (1994). "Catalytic Combustion Technology to Achieve Ultra Low NO_x Emissions: Catalyst Design and Performance Characteristics," *Proceedings of the International Workshop on Catalytic Combustion*, Tokyo, Japan (Hiromichi Arai, editor).
- Farrauto, R.J., Hobson, M.C., Kennelly, T. and Waterman, E.M. (1992). "Catalytic Chemistry of Supported Palladium for Combustion of Methane," *Applied Catalysis* **81**, 227-237.
- Froment, G.F. and Bischoff, K.B., *Chemical Reactor Analysis and Design*, John Wiley & Sons, New York, 1990.
- Furuya, T., Sasaki, K., Hanakata, Y., Mituyasu, K., Yamada, M., Tsuchiya, T. and Furuse, Y. (1994). "Development of a Hybrid Catalytic Combustor for a 1300 °C Class Gas Turbine," *Proceedings of the International Workshop on Catalytic Combustion*, Tokyo, Japan (Hiromichi Arai, editor).
- Garbowski, E., Feumi-Jantou, C., Mouaddib, N. and Primet, M. (1994), "Catalytic combustion of methane over palladium supported on alumina catalysts: Evidence for reconstruction of particles," *Applied Catalysis* **109**, 277-291.
- Greenler, R.G., Burch, K.D., Kretzschmar, K., Klausner, R., Bradshaw, A.M. and Hayden, B.E. (1985), "Stepped single-crystal surfaces as models for small catalyst particles," *Surface Science* **152/153**, 338-345.
- Griffin, T., Weisenstein, W., Schlegel, A., Buser, S., Benz, P., Bockhorn, H., and Mauss, F. (1994). "Investigation of the NO_x Advantage of Catalytic Combustion," *Proceedings of the International Workshop on Catalytic Combustion*, Tokyo, Japan (Hiromichi Arai, editor).
- Hayashi, S., Yamada, H. and Shimodaira, K. (1994). "High-Pressure Reaction Characteristics of a Catalytic Combustor for Gas Turbines," *Proceedings of the International Workshop on Catalytic Combustion*, Tokyo, Japan (Hiromichi Arai, editor).
- Hicks, R.F., Qi, H., Young, M.L. and Lee, R.G. (1990). "Structure sensitivity of methane oxidation over platinum and palladium," *Journal of Catalysis* **122** (1990) 280-294.
- Ladas, S., Imbihl, R. and Ertl, G. (1989). "Kinetic oscillations during the catalytic CO

oxidation on Pd(110): The role of subsurface oxygen," *Surface Science* **219**, 88-106.

Lieske, H. and Volter, J. (1985). "Pd redispersion by spreading of PdO in O₂ treated Pd/Al₂O₃," *Journal of Physical Chemistry* **89**, 1841-1842.

Lovett, Jeffrey A. and Abuaf, Nesim (1992) "Emissions and Stability Characteristics of Flameholders of Lean-Premixed Combustion," ASME paper 92-GT-120

Machida, M., Sato, A., Kijima, T., Inoue, H., Eguchi, K. and Arai, H. (1994). "Catalytic Properties and Surface Modification of Hexaaluminate Microcrystals," *Proceedings of the International Workshop on Catalytic Combustion*, Tokyo, Japan (Hiromichi Arai, editor).

Malte, P.C. (1988). Perspectives on NO_x Formation and Control for Gas Turbine Engines. Presentation for Sandia Turbulent Combustion Group at General Electric Research and Development Center.

McCarty, J.G., Wong, V.L. and Chang, Y.-F. (1993). "Kinetics and mechanism of catalytic combustion of methane by supported palladium oxide," Combustion Institute/Western States, Fall 1993.

Miller, J.A. and Bowman, C.T. (1989). "Mechanism and Modeling of Nitrogen Chemistry in Combustion," *Progress in Energy and Combustion Science*, **15**, 287-338.

Sadamori, H., Tanioka, T. and Matsuhisa, T. (1994). "Development of a High Temperature Combustion Catalyst System and Prototype Catalytic Combustor Turbine Test Results," *Proceedings of the International Workshop*

on Catalytic Combustion, Tokyo, Japan (Hiromichi Arai, editor).

Sathe S. B., Peck R. E. and Tong T.W. (1990). "A numerical analysis of heat transfer in porous radiant burners." *Int. J. Heat Transfer*, **33**, 1331-1338.

Tsujikawa, Y., Fujii, S., Sadamori, H., Ito, S., and Katsura, S. (1994). "Numerical Simulation of 2D Flow of Catalytic Combustor," *Proceedings of the International Workshop on Catalytic Combustion*, Tokyo, Japan (Hiromichi Arai, editor).

Viskanta, R. (1991). "Enhancement of Heat Transfer in Industrial Combustion Systems: Problems and Future Challenges," *ASME/JSME Thermal Engineering Proceedings*, **5**, 161-173.

Wang, T., Lee, C. and Schmidt, L.D. (1985). "Shape and orientation of supported Pd particles," *Surface Science* **163**, 181-197.
Weinberg, F.J. (1986) *Advanced Combustion Methods*, Academic Press, London & New York.

Younis, L. B. and Viskanta, R. (1993). "Experimental determination of the volumetric heat transfer coefficient between stream of air and ceramic foam", *International Journal of Heat and Mass Transfer*, **36**, 1425-1434.

List of Figures

- Figure 2-1. Catalyst-limited temperature scheme
- Figure 2-2. Three stages of combustion in a catalyst-limited temperature scheme.
- Figure 2-3. Post-catalyst injection scheme
- Figure 3-1. Equilibrium temperature versus equivalence ratio for CH_4 combustion
- Figure 3-2. Catalyst Exit Temperature for methane combustion
- Figure 3-3 Gas temperature in the burnout section for the methane-air combustion
- Figure 3-4. Pollutant species in the burnout section for two percent conversions
- Figure 3-5. Gas temperature in the burnout phase of a high-pressure combustor
- Figure 3-6. Pollutant concentrations in the burnout phase of a high-pressure combustor
- Figure 3-7. Equivalence ratio versus time for injection into the post-catalyst region
- Figure 3-8a. Temperature profiles for a post-catalyst injection model
- Figure 3-8b. Pollutant profiles for a post-catalyst injection model.
- Figure 4-1. Internally-insulated catalyst reactor.
- Figure 4-2. Quartz catalyst reactor.
- Figure 5-3. Temperatures as a function of distance along catalyst.
- Figure 5-1. Calculated and measured pressure drops for packed spheres
- Figure 5-2. Pressure drop per unit length for a packed bed of spheres

Figure 2-1. Catalyst-limited temperature scheme (from Dalla Betta et al., 1994)

Figure 2-2. Three stages of combustion in a catalyst-limited temperature scheme.

Figure 2-3. Post-catalyst injection scheme (from Furuya et al., 1994). Note the three different fuel injection zones

Equilibrium Temperature vs. Equivalence Ratio for CH₄ Combustion

Figure 3-1. Equilibrium temperature versus equivalence ratio for CH₄ combustion, with an inlet temperature of 350 °C
Catalyst Exit Temperature: CH₄-Air Combustion

Figure 3-2. Catalyst Exit Temperature for methane combustion

Gas Temperature in the Burn-Out Section: CH₄-Air Combustion

Figure 3-3 Gas temperature in the burnout section for the methane-air combustion

Pollutant Species Concentration in the Burn-out Section: CH₄-Air Combustion

Pollutant Species Concentration in the Burn-out Section: CH₄-Air Combustion

Figure 3-4. Pollutant species in the gas-phase burnout section for two catalyst-percent conversion in the catalyst

Figure 3-5. Gas temperature in the burnout phase of a high-pressure combustor.

Figure 3-6. Pollutant concentrations in the burnout phase of a high-pressure combustor

Figure 3-7. Equivalence ratio versus time for injection into the post-catalyst region

Well-Mixed-Balloon Model: CH₄ into burned premixed products at $\phi=0.25$

Figure 3-8a. Temperature profiles for a post-catalyst injection model.

Well-Mixed-Balloon Model: CH₄ into burned premixed products at $\phi=0.25$

Figure 3-8b. Pollutant profiles for a post-catalyst injection model.

Figure 4-1. Internally-insulated catalyst reactor.

Figure 4-2. Quartz catalyst reactor.

Figure 4-3. Temperatures as a function of distance along catalyst.

Figure 5-1. Calculated pressure drops at various void fractions and measured pressure drops

Figure 5-2. Pressure drop per unit length for a packed bed of spheres and 200 cpi honeycomb

NO_x Abatement in Advanced Gas Turbines

by

P.E. Sojka, J.P. Gore, N.M. Laurendeau
M.S. Klassen, L-K. Tseng, Y.R. Sivathanu
A. Kelkar, D. Thomsen, J. Bryjak
Thermal Sciences and Propulsion Center
School of Mechanical Engineering
Purdue University
West Lafayette, IN 47907-1003

November 1994

INTRODUCTION

Current government emission standards have mandated that advanced gas turbines have low dry-NO_x levels (< 10 ppm @ 15% O₂). However economic competitiveness requires that the efficiency of industrial turbines increase to over 60%. Higher efficiency requires operation at elevated pressures and temperatures; hence, the goal of lower NO_x emissions will require partially or fully premixed combustion [1].

Of critical importance is a simplified high-pressure NO_x model capable of predicting pollutant formation in practical gas turbine engines. To achieve this goal, a thorough understanding of the chemical kinetics involved in the production of NO in high-pressure natural gas-fired systems is needed. Since flames in practical gas turbine engines depend strongly on mixing, the interaction of turbulent fluid mechanics and reaction chemistry is as important as the chemistry itself. An understanding of NO chemistry mandates accurate, but compact, reaction schemes which in turn require *in-situ* measurements of NO concentration. Efficient modeling of the complex turbulence-chemistry interactions requires a new approach to solving the governing equations if engines are to be designed in a timely fashion. Combining the compact reaction schemes and efficient turbulence-chemistry reaction method with a flow solver provides an improved method for engine designers to use when evaluating NO_x production in the next generation of engines. Obviously, the predictions based on this method must be compared to benchmark data to ensure code accuracy.

The three contributions listed in the preceding paragraph, a compact reaction scheme, and efficient method of describing turbulence-chemistry interactions, and comparison of predictions based on these contributions to experimental data, will result from the successful completion of this program. Each is described in detail in the following sections.

High-pressure, high-temperature NO_x Chemistry
M. S. Klassen, D. D. Thomsen, and N. M. Laurendeau
Flame Diagnostics Laboratory
School of Mechanical Engineering
Purdue University
West Lafayette, IN 47907-1288

NO is produced through three main reaction mechanisms: (1) the Zeldovich, or thermal-NO mechanism, (2) the N₂O-intermediate mechanism, and (3) the prompt-NO mechanism. The amount of NO formed through each of these mechanisms depends on the temperature, pressure and equivalence ratio of the flame.

For low-temperature flames, the amount of NO produced via the Zeldovich mechanism is small, but for high-temperature flames, such as those found in high-pressure gas turbines, the thermal NO mechanism becomes the dominant pathway for NO production [6]. The N₂O-intermediate mechanism is very significant in lean, premixed combustion. The third reaction path for NO formation is the prompt-NO mechanism. This mechanism tends to produce much more NO under moderately fuel-rich conditions than under fuel-lean conditions. However, prompt-NO production is significantly curtailed for highly fuel-rich flames.

Miller and Bowman [4] performed a thorough chemical kinetics modeling study to determine the relative importance of the various reaction paths. They found that, in general, the NO production reactions need to be coupled with those reactions involving oxidation of the fuel. This procedure is necessary because of the presence of O and OH radicals in the thermal-NO and N₂O mechanism and of CH in the prompt-NO mechanism. Their work also suggests that the rapid drop-off of prompt-NO production in rich flames correlates with a rapid decrease in the HCN concentration. This would correspondingly indicate a lack of N atoms being produced through reaction (8).

Corr et al. [6] recommend that all three mechanisms be considered for lean, premixed combustion. This recommendation is further substantiated by the work of Drake et al. [2] who found that for high-pressure, low-temperature, lean, premixed C₂H₆/O₂/N₂ flames, all three mechanisms (with the thermal-NO and N₂O-intermediate mechanisms enhanced by super-equilibrium O-atom concentrations) contribute significantly to flamefront NO production. Unlike nonpremixed flames, which are dominated by thermal-NO production, the lower temperature of non-stoichiometric premixed flames tends to enhance the influence of the prompt and N₂O mechanisms. Since advanced gas turbines are expected to most closely approximate lean premixed systems [1], all three reaction mechanisms and their pressure scaling must be investigated.

Experimental Apparatus/LIF Methodology/Kinetic Modeling

The laser system and optical layout used in performing the LIF measurements of NO are described elsewhere [3]; thus, only a brief summary will be presented here. The excitation of NO is achieved through use of the Q₂(26.5) line in the $\gamma(0,0)$ band. The Q₂(26.5) line was chosen because (1) the Boltzmann fraction is insensitive to temperature fluctuations over the range of temperatures of our flames, and (2) other species do not interfere spectrally with this NO absorption line [3]. The excitation wavelength is generated by employing the second harmonic ($\lambda = 532$ nm) of a Nd:YAG laser to pump a dye laser, which produces laser radiation at 572 nm. The output of the dye laser is frequency doubled in a Wavelength Extender, and the doubled-dye beam is mixed with the first harmonic of the Nd:YAG laser, producing a 2 mJ/pulse at 225.5 nm.

After leaving the laser system, the beam is directed towards a 2.5-cm diameter, water-cooled, sintered-bronze flat-flame burner. The burner is located inside the high-pressure combustion facility described by Carter et al. [7]. After exiting the vessel, the beam passes through a fused silica plate which directs a portion of the beam towards a UV-sensitive photodiode. This photodiode is employed to monitor the beam energy, which is required for normalization of the fluorescence signal.

For fluorescence detection, we make use of an optical port perpendicular to the laser entrance and exit ports. The fluorescence is focused on the entrance slit of a 1/2-m monochromator. The detector is an RCA 1P28B photomultiplier tube specially wired for temporal resolution of the fluorescence signal [8]. The broadband fluorescence signal is detected at 236 nm, corresponding to 3 nm of the $\gamma(0,1)$ band of NO. To account for pressure shifting of the Q₂(26.5) absorption line, the laser and collection optics are tuned for the maximum NO fluorescence signal at each pressure. A 500-ps window at the peak of the fluorescence pulse is sampled using a fast sampler. The image of the entrance slit over the burner is 80 $\mu\text{m} \times 6.67$ mm. Each data point is averaged over 600 laser shots.

Several of the experimental flames in this study were investigated through computer modeling. The modeling of the chemical kinetics was performed using the Sandia, steady, laminar, one-dimensional, premixed flame code [9]. The mechanism used as the chemical kinetics input into the computer model is based on the comprehensive mechanism assembled by Glarborg et al. [10] as modified by Drake and Blint [5], referred to as GMKDB. Drake and Blint [5] adopted most of the reaction mechanism from Glarborg et al. [10]; however, they made a few modifications. These include the introduction of pressure dependency into four unimolecular reactions, the addition of a C₃H₈ reaction mechanism, and the introduction of rate parameters for $\text{CH} + \text{N}_2 \rightarrow \text{HCN} + \text{N}$ based on measurements in a high temperature shock tube [11]. The rate parameters for the unimolecular reactions at pressures of 3.05, 6.1, and 9.15 atm are

given by Drake et al. [2]. The rate parameters for 11.9 atm were obtained from Drake and Blint [12].

Results and Discussion

Using the experimental apparatus described previously, quantitative LIF measurements of NO were performed in flat, laminar, $\text{CH}_4/\text{O}_2/\text{N}_2$ flames at pressures ranging from 1 to 14.6 atm. Similar work has been done in $\text{C}_2\text{H}_6/\text{O}_2/\text{N}_2$ flames for pressures up to 14.6 atm [13]. The temperatures in the post-flame region of these flames, as measured with radiation-corrected Pt-Pt/10%-Rh thermocouples, ranged from 1650 to 1950 K (precision ± 30 K, accuracy ± 75 K). Flames with dilution ratios ($V_{\text{N}_2}/V_{\text{O}_2}$) of 3.1 and 2.2 were investigated. A reduction of the dilution ratio to 2.2 was required to obtain stable flames at $\phi < 0.6$. The total flow rates were held constant at each pressure, and were as follows: 3.50 slpm (1 atm), 6.20 slpm (3.05 atm), 9.1 slpm (6.1 atm), 10.95 slpm (9.15 atm), 12.75 slpm (11.9 atm) and 14.5 slpm (14.6 atm).

Figure 1 presents the measurements of NO number density as a function of equivalence ratio at a pressure of 1 atm. The dilution ratio for these flames was 3.1. Each data point in Figure 1 was obtained at 7 mm above the burner surface, which is well into the post-flame zone. The NO number densities increase fairly gradually as the equivalence ratio increases until the peak NO concentration is attained ($\phi = 1.4$) after which the amount of NO drops quite rapidly. Reisel et al. [3] found a similar trend in NO number density for $\text{C}_2\text{H}_6/\text{O}_2/\text{N}_2$ flames at atmospheric pressure. Figure 1 also includes the temperature for the various flames measured at 7 mm above the burner surface. The bi-modal shape of the temperature profile is a result of the movement of the flame front relative to the burner surface as the flame stoichiometry changes [14]. The flame is closest to the burner surface for the stoichiometric and slightly rich flames, but moves farther away from the burner as the equivalence ratio departs from these conditions. Similar temperature profiles, both in shape and temperature range, were found at the higher pressures investigated in this study.

Figure 1 also includes predictions of NO concentration using the GMKDB mechanism. In previous work [13] in high-pressure flames, this mechanism was found to give better quantitative predictions of NO concentration than the mechanism presented by Miller and Bowman [4]. Nevertheless, the GMKDB mechanism tends to under predict the NO concentrations in lean flames and to over predict the peak NO concentration found in rich flames. The predictions shown in Fig. 1 utilized the temperature profile generated by the energy solution of the GMKDB mechanism.

Figure 2 presents measurements of NO concentration as a function of equivalence ratio at pressures from 1 to 14.6 atm. The dilution ratio of all the flames was 3.1. A shift in the position of peak NO towards leaner equivalence ratios with increasing pressure is evident. This behavior

is qualitatively similar to that found by Reisel and Laurendeau [13] in $C_2H_6/O_2/N_2$ flames, though the number density of NO is higher in the present flames. The larger amount of NO is probably due to the higher temperatures found in the CH_4 flames. Reisel and Laurendeau [13] attributed the shift in peak NO with increasing pressure to the increased importance of CH production resulting via $CH_2 + OH \rightarrow CH + H_2O$ near stoichiometric conditions. At higher pressures (11.9 and 14.6 atm), the NO number density does not drop off as rapidly after reaching its peak (as compared to lower pressures), but levels off slightly before falling. This result may be due to enhanced thermal NO production at the higher temperatures associated with peak NO production near stoichiometric conditions at the higher pressures.

Predictions of NO production in the high-pressure flames (3.05 to 11.9 atm; 3.1 dilution ratio) are presented in Figure 3. These predictions utilized the temperature profile generated by the energy solution for the GMKDB mechanism. Significant over-prediction of peak NO number density is found when compared with the measurements presented in Figure 2. The shift in the position of peak NO towards leaner equivalence ratios found in the experiments is consistent with the predictions. This feature was also captured very well by the GMKDB mechanism in previous work [13]. The modeling also captures the plateau region associated with peak NO production in the 11.9 atm flames.

Leonard and Correa [3] found that NO_x emissions showed little pressure dependence in lean, premixed turbulent flames, both for experiments and computations. Correa and Smooke [15] also found that NO_x (on a ppm basis) was insensitive to pressure in their computations for lean, premixed laminar CH_4 /air flames. Figure 4 presents detailed measurements of NO number density in lean flames at pressures from 1 to 14.6 atm. The NO number density has been normalized by pressure so as to remove the direct effect of pressure on number density. Though detailed measurements of NO were made in lean flames with dilution ratios of 3.1 and 2.2, only the results from flames with a dilution ratio of 2.2 are shown for clarity. These measurements indicate that in the flat, laminar flames studied here, there is a substantial increase in NO concentration with increasing pressure. This is most evident for the NO measurements in flames at pressures greater than 3.05 atm.

Conclusions

Quantitative measurements of NO number density have been made in high-pressure, laminar, flat $CH_4/O_2/N_2$ flames. Qualitatively similar results to those found previously in $C_2H_6/O_2/N_2$ flames were obtained, though the amount of NO produced is larger in methane flames, probably due to higher temperatures. Modeling of these flames using the GMKDB mechanism demonstrates the sensitivity of the NO predictions to temperature and the need for accurate measurements of temperature profiles in these flames. Detailed measurements in lean

flames indicate that post-flame NO concentrations are affected by pressure for pressures greater than 3.05 atm.

References

1. S.M. Correa, *Combust. Sci. and Tech.* 87, 329(1992).
2. M.C. Drake, J.W. Ratcliffe, R.J. Blint, C.D. Carter, and N.M. Laurendeau, *Twenty-third Symposium (International) on Combustion*. The Combustion Institute, Pittsburgh, PA, 387 (1990).
3. J.R. Reisel, C.D. Carter, N.M. Laurendeau and M.C. Drake, *Combust. Sci. and Tech.* 91, 271(1993).
4. J.A. Miller and C.T. Bowman, *Prog. Energy Combust. Sci.* 15, 287 (1989).
5. M.C. Drake and R.J. Blint, *Combust. Sci. and Tech.* 75, 261 (1991).
6. R.A. Corr, P.C. Malte, and N.M. Marinov, *ASME Journal of Engineering for Gas Turbines and Power* 114, 425 (1992).
7. C.D. Carter, G.B. King, and N.M. Laurendeau, *Rev. Sci. Instrum.* 60, 2606 (1989).
8. J.M. Harris, F.E. Lytle, and T.C. McCain, *Anal. Chem.* 48, 2095 (1976).
9. R.J. Kee, J.F. Grcar, M.D. Smooke, and J.D. Miller, A FORTRAN program for modeling steady laminar one-dimensional premixed flames. *Sandia Report SAND85-8240* (1985).
10. P. Glarborg, J.A. Miller, and R.J. Kee, *Combust. Flame* 65, 177 (1986).
11. A.J. Dean, D.F. Davidson, R.K. Hanson, and C.T. Bowman, Development and application of CH laser absorption diagnostic for shock tube kinetic studies. Western States Section/Combustion Institute, Paper 88-91 (1988).
12. M.C. Drake and R.J. Blint, Personal Communication (1991).
13. J.R. Reisel and N.M. Laurendeau, Laser-induced Fluorescence Measurements and Modeling of Nitric Oxide Formation in High-Pressure Flames, *Combust. Sci. Tech.*, in press (1994).
14. C.R. Ferguson and J.C. Keck, *Combust. Flame* 34, 85 (1979).
15. S.M. Correa and M.D. Smooke, *Twenty-third Symposium (International) on Combustion*, The Combustion Institute, Pittsburgh, PA, 289 (1990).

Figure 1. Comparison of LIF-measured NO concentrations and predictions in atmospheric $\text{CH}_4/\text{O}_2/\text{N}_2$ flames. The flames have a dilution ratio of 3.1. Temperature measurements are obtained with radiation corrected Pt-Pt/10% Rh thermocouples. The measurements were taken 7 mm above the burner surface.

Figure 2. LIF-measured NO concentrations in $\text{CH}_4/\text{O}_2/\text{N}_2$ flames at pressures ranging from 1 to 14.6 atm. The dilution ratio of the flames was 3.1. Measurements in the flames at pressures greater than 1 atm were made at 3 mm above the burner surface.

Figure 3. Predictions of NO number density in CH₄/O₂/N₂ flames at pressures ranging from 1 to 11.9 atm. The dilution ratio of the flames was 3.1. The predictions were made using the GMKDB reaction mechanism with the temperature profile generated by the energy solution.

Figure 4 Detailed LIF-measured NO concentrations in lean flames at pressures ranging from 1 to 14.6 atm. The flames have a dilution ratio of 2.2.

Turbulence Chemistry Interactions
J. P. Gore, Y. R. Sivathanu, and A. S. Kelkar
Thermal Sciences and Propulsion Center
School of Mechanical Engineering
Purdue University
West Lafayette, IN 47907-1003

Introduction

Lean premixed combustion shows the greatest promise for achieving ultra-low NO_x emissions in gas turbine engines (Correa, 1992). Computer codes for efficient design and preliminary development of such combustors require a model for the treatment of interactions between turbulence and chemistry. Probability Density Function (PDF) transport methods represent one of the most promising approaches to the treatment of turbulence chemistry interactions (Pope, 1991a). Traditionally, Monte Carlo simulations have been used for the solution of the PDF transport equations. The Monte Carlo simulations require relatively long computational time due to the large number of parcels required for convergence. Furthermore, the solutions contain noise produced by the random number generator which forms the basis of selection of initial conditions, boundary conditions, and unclosed cross-correlations between variables for each parcel in the simulation.

Motivated by these observations, development of a technique based on the use of discrete probability functions (DPF) which shows the promise for significant reductions in computer time required for the simulation of the PDF transport equations with substantially lower noise was initiated during the project period. The technique has been applied to: (1) NO formation in a diffusion flamelet (Gore and Sivathanu, 1993), (2) scalar mixing in a turbulence layer (Sivathanu and Gore, 1994a, and Sivathanu and Gore 1994b), and (3) mixing and low temperature reactions in a turbulence layer (Sivathanu and Gore 1994b). In the case of the reacting and non-reacting turbulence layer problems, a Monte Carlo simulation was also carried out and compared with the DPF simulation. The DPF solutions eliminated the significant statistical noise observed in the Monte Carlo simulations and yielded a factor of two advantage over the Monte Carlo simulations.

The model problems involving NO formation in a diffusion flamelet and scalar mixing and reaction in a turbulence layer were selected in order to develop the methodology using existing data regarding turbulent mixing and reactions. Present work is focused on extending the methodology to NO formation in lean premixed flames. A configuration involving pilot stabilized lean premixed turbulent jet flames in a co-flow of air is selected for this extension. Measurements of PDFs of axial and radial velocities and temperatures are being obtained using laser Doppler velocimetry (LDV) and thin filament pyrometry (TFP). The NO formation rates and concentration PDFs will be predicted using the DPF model. The performance of the model will

be evaluated by comparison of predicted mean NO concentrations in the exhaust stream with measurements. Future work may involve in situ measurements of NO concentration PDFs in turbulent premixed flames using laser saturated fluorescence techniques developed in Task 1 of this project.

The DPF model and its comparison with Monte Carlo simulations are presented in the references discussed above. The experimental arrangement and sample measurements of PDFs of velocity and temperature are discussed in the following. The discussion ends with a description of work planned for the second year of the project.

Experimental Methods

Figure 1 shows a schematic of the pilot-stabilized turbulent premixed flame burner. Metered quantities of methane/air mixture is admitted through the central 15 mm diameter tube. A honeycomb section at the exit of the tube is utilized to provide well-defined initial conditions for the flow. A thin annular ring flame of hydrogen burning in a diffusion flame mode with a relatively small flow rate (pilot flame heat release rate is less than 10% of the total) is utilized for the stabilization of the premixed flames. A co-flow of air admitted into a 100 mm annulus is utilized to guard the flames from surrounding disturbances.

The results discussed in the following are for a 2 kW flame with a burner exit Reynolds number of 4500. The equivalence ratio for the premixed flame was selected to be 0.8.

Velocities in the axial direction were measured using a single component forward scattered laser Doppler velocimeter (LDV) shown schematically in Fig. 2. A reverse cyclone seeder with 0.5 micron Al_2O_3 particles was utilized to seed the inlet flow in quantities sufficient to yield data rates at approximately 20 kHz. Power spectral densities of the resulting velocity signals were examined to ensure that this data rate was sufficient to resolve the turbulent fluctuations down to the noise level of the instrument. Data were collected at a sampling frequency of 10 kHz.

Transient temperatures in the flame were measured using a thin filament pyrometry instrument (Vilimpoc and Goss, 1988) shown schematically in Fig. 3. A 20 micron diameter fiber made out of silicon carbide was mounted across the premixed flame at various heights above the burner. The light emitted by this fiber was collected by a CaF_2 lens and focused on to a liquid nitrogen cooled InSb detector. The measured emission intensities were converted to temperatures using calibrations obtained with a premixed laminar flame. The long infrared sensitivity of the detector allows measurement of temperatures as low as 500 K and the durability of the fiber and the high dynamic range of the data collection instruments allows measurement of temperatures as high as 2400 K. The estimated response time of the system is 0.3 ms allowing resolution of frequencies up to 2.5 kHz which were observed to be adequate for the present flame.

Results and Discussion

Figure 4 shows an axial profile and two radial profiles of mean and root mean square (RMS) axial velocity. Radial profiles at a total of 10 axial stations in the flame were obtained and two representative locations are shown in Fig. 4 for brevity. A total of 200,000 data points at each individual location were collected and stored on disks for statistical analysis.

The axial profile of mean velocity shows a small acceleration of the flow at the burner exit possibly due to the combined effects of heat release and the pilot flame. After the initial acceleration, the mean centerline axial velocity remains almost constant over the length of the flame. The axial velocity begins to decrease at distances greater than 18 cm due to mixing with the co-flowing air. The turbulence intensity is high (30%) near the exit due to the effects of the honeycomb. It decreases to very small values at the centerline as increase in temperature causes increase in viscosity and leads to rapid dissipation of the fluctuations. The centerline turbulence intensity increases to 30% beyond an axial distance of 18 mm due to the generation by shear induced mixing with the co-flowing air.

Radial profiles of axial velocity at $x=0.3$ cm and 15 cm are also shown in Fig. 4. The data at 0.3 cm show the effects of the interacting wakes resulting from the honeycomb flow straightener. The turbulence intensities are high across the width of the flame. These data will serve as the initial conditions for numerical simulations of the flame. A sample radial profile of axial velocity at $x=15$ cm is also shown in Fig. 4. The data show a region of constant axial velocity near the center followed by a region of decreasing temperature in the high shear region. The RMS velocity fluctuations increase continuously with increasing radial distance. The radial velocity profiles show that an axisymmetric flow field particularly at locations away from the burner exit is obtained.

Probability density functions of axial velocity at three sample radial positions in the premixed flame are shown in Fig. 5. The PDF at the centerline is narrow and symmetric around the mean value. The PDF at $r=4$ mm shows a slightly lower mean value than that at the centerline and a long tail towards the lower velocities possibly due to intermittence. The PDF at $r=10$ mm is not symmetric and shows a broad range of axial velocities between 1 m/s and 9 m/s. The quality of the present data appears to be adequate for providing statistically converged and relatively noise-free estimates of velocity PDFs.

Measurements of PDF of temperature at $x=15$ cm and the same three radial locations are shown in Fig. 6. The data show that the mean temperature near the axis is 1120K and the range of temperatures with high probabilities is rather narrow with a long tail with relatively low probability towards the highest temperature range. At $r=4$ mm where the RMS velocity fluctuations are over 15% and the mean axial velocity is relatively high (7 m/s), the temperature PDF shows a very broad distribution between 2100 K (adiabatic flame temperature) and 750K

with the mean occurring at 1280 K. The RMS temperature fluctuations approach 25% at this location. At $r=10$ mm, the peak temperatures never rise beyond 1400 K indicating that the reaction zone does not traverse this region and the mean temperature is approximately 880 K.

The axial velocity PDFs shown in Fig. 5 together with the PDFs of radial velocity determine the residence time available for processes such as molecular diffusion and chemical reaction. The temperature PDFs have a strong influence on the chemical reaction rates. The temperature PDFs also affect the thermophysical property fluctuations. The DPF calculations will use the measurements such as those shown in Figs. 5 and 6 to calculate the NO production. This procedure avoids the uncertainties of the turbulent momentum and energy transport calculations from affecting the evaluation of the interactions between NO chemistry and turbulence.

References

- Correa S. M., 1992, "A Review of NO_x Formation Under Gas Turbine Conditions," *Combust. Sci. and Tech.* Vol. 87, pp. 329-362.
- Gore J. P. and Sivathanu Y. R., 1993, "A Discrete Probability Function (DPF) Method for the Treatment of Turbulence and NO Chemistry Interactions," *AGTSR Contractors Quarterly Report for the Period 9/93-12/93*, Thermal Sciences and Propulsion Center, Purdue University, West Lafayette, IN 47907.
- Gore J. P. and Sivathanu Y. R., 1994a, "Discrete Probability Function Method for A Turbulence Mixing Layer," *AGTSR Contractors Annual Report for the Period 9/93-8/94: Appendix C*, Thermal Sciences and Propulsion Center, Purdue University, West Lafayette, IN 47907.
- Pope S. B., 1990, "Computations of Turbulent Combustion: Progress and Challenges," *Proceedings of the Twenty-Third Symposium (Int.) on Combustion*, The Combustion Institute, Pittsburgh, PA, pp. 591-612.
- Vilimpoc V. and Goss L. P., 1988, "SiC Based Thin-Filament Pyrometry: Theory and Thermal Properties," *Proceedings of the Twenty-Second Symposium (Int.) on Combustion*, The Combustion Institute, Pittsburgh, PA, pp. 1907-1914.
- Sivathanu Y. R. and Gore J. P., 1994a, "Statistical Simulations of Passive Scalar Mixing in a Turbulence Layer Using a Novel Diffusion Model," *AGTSR Contractors Annual Report for the Period 9/93-8/94: Appendix B*, Thermal Sciences and Propulsion Center, Purdue University, West Lafayette, IN 47907.

Figure 1: A Sketch of the Turbulent Pilot Stabilized Premixed Burner.

LASER DOPPLER VELOCIMETRY ARRANGEMENT

Figure 2: A Schematic Diagram of the Laser Doppler Velocimetry (LDV) System.

THIN FILAMENT PYROMETRY SETUP

Figure 3: A Schematic Diagram of the Thin Filament Pyrometry System.

VELOCITY PROFILES

$Q = 2 \text{ kW}$, $d = 15 \text{ mm}$, $D = 100 \text{ mm}$

CH_4 , $Re = 4500$, $\phi = 0.8$

$m_{\text{coflow}} = 0.5 \text{ g/s}$

Data	Velocity
●	Mean
▼	RMS

Figure 4: Mean and RMS Axial Velocities in as a Function of Axial Distance in a Premixed Pilot Stabilized Turbulent Jet Flame.

VELOCITY PDFs

2 kW, $d = 15$ mm, $D = 100$ mm
 CH_4 , $Re = 4500$, $x = 150$ mm
 $m_{\text{coflow}} = 0.5$ g/s, $\phi = 0.8$

Figure 5: Probability Density Functions of Axial Velocity in a Premixed Pilot Stabilized Turbulent Jet Flame.

TEMPERATURE PDFs

2 kW, $d = 15$ mm, $D = 100$ mm
 CH_4 , $Re = 4500$, $x = 150$ mm
 $m_{\text{coflow}} = 0.5$ g/s, $\phi = 0.8$

Figure 6: Probability Density Functions of Temperature in a Premixed Pilot Stabilized Turbulent Jet Flame

Analog Gas Turbine Experiments and Modeling

L- K. Tseng, J.J. Bryjak, and P.E. Sojka
Thermal Sciences and Propulsion Center
School of Mechanical Engineering
Purdue University
West Lafayette, IN 47907-1003

Numerical Modeling

The first goal of this portion of the study is to incorporate the compact chemical kinetics model and turbulence-chemistry interaction model into a commercially available flow solver in order to predict NO_x levels in an axisymmetric sudden dump combustor. Central to this comparison is a demonstration of the flow solver's ability to accurately simulate the cold and reacting flows produced by a sudden dump combustor.

Our initial efforts at code assessment were performed using experimental data obtained from Gould [1987] for an axisymmetric sudden expansion combustor operating on propane. Their equivalence ratio of 0.5 and inlet velocity of 22 m/s are well matched to the range of values we will consider in this program. In addition, our inlet/combustor area ratio of 9 is reasonably close to Gould's value of 4. Gould presents axial and radial velocity profiles for both cold and reacting flows, as well as temperature profiles for the reacting flow case. In addition, their work includes well characterized initial conditions.

Two aspects of flow solver performance have been considered to date: the influence of turbulence models and the effect of inlet boundary conditions. The influence of turbulence model is considered first, followed by the impact of inlet boundary conditions. Both cold flow and reacting flow cases are discussed.

A criteria must be set to gauge the performance of one turbulence model relative to another. Speziale and Thangam note that a codes ability to accurately reproduce the reattachment point is a standard method for gauging performance. Unfortunately, Gould's experimental data does extend to the wall so a comparison with reattachment point cannot be made.

An alternate method to gauging performance is to compare a model's ability to reproduce experimental velocity results. The radial velocity profiles are best suited for this purpose, since the smaller magnitude radial flow emphasizes model differences. We have adopted that method in this study.

Simulations were run using three separate turbulence models to determine benefits of one model over another. The standard k- ϵ model, RNG k- ϵ model, and Reynolds Stress Model (RSM) were chosen because of their widespread use. Various inlet conditions were considered.

Results for the standard k- ϵ model are shown in Figures 1 and 2. The axial flow

predictions in Figure 1 demonstrate acceptable agreement with best agreement being obtained nearest the dump plane. Figure 2 indicates that the radial flow predictions could be improved, especially near the dump plane where recirculation is strongest.

The RNG k- ϵ and RSM models are reported to better represent the fundamental flow physics when recirculation is present than the standard k- ϵ model. They should therefore produce results in better agreement with experimental data.

Figure 1 indicates a slight, but noticeable improvement of RNG k- ϵ axial velocities over standard k- ϵ values, except near the wall at intermediate x/H . As expected, agreement between RNG k- ϵ radial velocities and experiment is superior to that provided by standard k- ϵ predictions, especially near the dump plane where recirculation is most pronounced. Little difference between RNG and standard k- ϵ predictions is observed at locations far downstream.

Figure 1 also contains RSM axial velocity predictions. Interestingly enough, results obtained using this model showed a substantial difference in the position of the reattachment point, it being located much closer to the dump plane than for the other models. Recall, however that Gould's data does not provide reattachment point information so the accuracy of this quantity could not be checked. Otherwise, the RSM predictions appeared to be similar to the standard and RNG k- ϵ model predictions, except near the wall at intermediate locations and near the centerline at axial locations far downstream.

Three different types of boundary conditions were used in the cold flow case. For cold flow, mean velocities and turbulence quantities (k and ϵ) are the only required boundary inputs. Two possible types of input profiles are flat profiles and piece wise linear profiles. Flat profiles can be either "top hat", or "two-step." The latter looks like a two layer cake. A piece wise linear profile allows input quantities to vary across the inlet boundary. The flow solver then linearly interpolates between points to find values at intermediate radial positions.

Experimental values obtained by Gould at an axial distance 7.62 mm downstream of the dump plane were used for the piece-wise linear inlet boundary conditions. Ideally, experimental values would be available at the dump plane proper.

Figure 3 illustrates the importance of using experimentally measured inlet boundary conditions. It contains radial velocity results based on standard k- ϵ , RNG k- ϵ , and RSM turbulence models for a "top hat" inlet velocity profile, a "two-step" inlet velocity profile, and a piece wise profile generated from velocity data obtained by Gould at their first axial station. Note that when using the "top hat" profile, or the "two-step" profile suggested by Gould, no significant improvement was found for RSM or RNG k- ϵ over standard k- ϵ model. However, when piece-wise linear boundary conditions were entered for the turbulent energy and dissipation, then a notable improvement was found in the RNG k- ϵ performance. The radial velocity profiles had the most notable improvement.

Figures 4 and 5 illustrate how variations in turbulence model parameters influence velocity predictions. The standard k-ε model was employed here with the lowest turbulent characteristic length set to one step height. Figure 4 shows that varying characteristic length by a factor of 100 has little impact on axial velocity predictions, until the flow has moved 12 step heights downstream of the dump plane. Figure 5 indicates that varying characteristic length over this same range has only a minor effect on the radial velocity profile, regardless of axial position.

Our finding that predictions based on the standard k-ε model are insensitive to characteristic length is in contradiction to Hwang et al.'s findings. It is speculated by CC Hwang et al that 'inlet conditions for k and ε may play as crucial a role in achieving predicted accuracy as turbulence modeling details'. Hwang found that his solution was very sensitive to even small changes in the inlet conditions.

Preliminary reacting flow predictions are presented in Figures 6 through 8. The standard k-ε model was again employed with the turbulent characteristic length set equal to one step height. Magnussen and Hjerager's [1976] modification of Spalding's [1970] turbulent chemistry model is included, along with a one step reaction for propane oxidation

$$RR = 1e15 e^{-100/RT} [C_3H_8][O_2]$$

Reacting flow axial velocity predictions are presented in Figure 6. The model is able to qualitatively predict axial velocity profiles for all but the furthest downstream distance. Quantitative agreement between model results and experimental data is reasonable near the dump plane, but worsens as the flow moves downstream.

Reacting flow radial velocity predictions are presented in Figure 7. Model predictions are qualitatively correct at larger axial distances, but over predict radial velocities near the edge of the flame zone.

Reacting flow temperature profiles are presented in Figure 8. Qualitative and quantitative agreement exceeds that for either velocity profile. In addition, quantitative agreement improves with downstream distance.

Improved agreement between model predictions and reacting flow data may be obtained in one of two ways: estimation of heat transfer from Gould's flow or increasing the sophistication of the one-step propane oxidation scheme employed here. Both will be considered in future work.

Experimental Measurements

The second goal of this portion of the study is to obtain benchmark data for comparison with code predictions. To that end, a high-pressure (up to 15 atm), high-temperature (up to 375 C at the dump plane) has been designed and fabricated and is now operational. The combustor is

an axisymmetric sudden dump with a 2.5 cm diameter inlet feeding a 7.5 cm diameter chamber. It is fed with premixed methane/air. A photograph of the combustor in operation is provided as Figure 9.

As a first step in building our benchmark data base, we have been NO obtained using a quartz walled version of our high-pressure, high-temperature combustor. Data was obtained using a conventional gas sampling train consisting of a 6.3 mm od, 1.5 mm orifice water cooled probe, heated sampling line, a cold trap, and a Thermo-Environmental Instruments Model 10 chemiluminescence analyzer. The analyze zero and span were adjusted using bottled calibration gases supplied by either Matheson or Airco.

Figure 10 presents preliminary experimental results of NO emissions level versus equivalence ratio for premixed methane-air combustion at three values of inlet temperature. Of particular interest is data in the equivalence ratio range below 0.8, where NO levels are 25 ppm. The NO emissions level increases slightly with temperature over this region, as expected.

Figure 10 also presents NO emissions level data at equivalence ratios above 0.8. As expected, emissions peak near stoichiometric and fall off again for increasingly rich mixtures. Note that the peak occurs at a much lower equivalence ratio value than that of the flat flame burner work described above. We are currently attempting to determine why this discrepancy exists.

Conclusions

Comparison of flow solver predictions with experimental data is underway. The RNG k- ϵ model provides best agreement with previously published cold flow data. Model results are affected only slightly by choice of turbulent characteristic length. Model results for the reacting flow case do not exhibit the same level of agreement as their cold flow counterparts. The lack of agreement is attributed to a combination of heat loss from the experimental facility and the one-step kinetic scheme employed in our simulations.

Preliminary NO data has been obtained using a quartz-walled version of our high-pressure, high-temperature combustor. NO emissions levels are less than 25 ppm for equivalence ratios of 0.8 and below. A peak in the NO versus equivalence ratio curve is observed, but at an equivalence ratio below that of the flat flame data discussed above. We are currently attempting to explain this discrepancy.

References

- Gould, R.D., "Turbulence Characteristics of an Axisymmetric Reacting Flow," Ph.D. Thesis, Purdue (1987).
- Magnussen, B.F. and Hjertager, B.H., "On Mathematical Models of Turbulent Combustion with

Special Emphasis on Soot Formation and Combustion," pp. 719-727 in 16th Symp (Int'l) on Combustion (The Combustion Institute: Pittsburgh, 1977).

Spalding, D.B., "Mixing and Chemical Reaction in Steady Confined Turbulent Flames," pp. 649-657 in 13th Symp (Int'l) on Combustion (The Combustion Institute: Pittsburgh, 1971).

Speziale, C.G. and Thangam, S., "Analysis of an RNG Turbulence Model for Separated Flows," Intl J Engng Sci, 30(10) 1379-1388 (1992).

Figure 1. Comparison of axial velocity profiles for Gould's cold flow case.

Figure 2. Comparison of radial velocity profiles for Gould's cold flow case.

Figure 3. Influence of inlet boundary conditions on radial velocity profiles for Gould's cold flow case.

- experiment
- △ TI=10%, CI=0.0381m
- ▽ Gould's Profile
- ◇ Piecewise linear profile

Figure 4. Influence of turbulent characteristic length on axial velocity profiles for Gould's cold flow case.

Figure 5. Influence of turbulent characteristic length on radial velocity profiles for Gould's cold flow case.

- experiment
- △ CL=.0381m; TI=10%
- CL=.381m; TI=10%
- Gould's Profile

Figure 6. Comparison of axial velocity profiles for Gould's hot flow case.

Figure 7. Comparison of radial velocity profiles for Gould's hot flow case.

Figure 8. Comparison of temperature profiles for Gould's cold flow case.

Figure 9. High-pressure, high-temperature combustion in operation.

Figure 10. NO versus equivalence ratio for three values of inlet temperature.

ACTUATORS FOR COMBUSTION CONTROL--NEW APPROACHES AND MATERIALS

CONTRACT INFORMATION

Contract Number	93-02-SR013
Contractor	Virginia Tech Mechanical Engineering Department Blacksburg VA 24061-0238 703-231-4459 Fax 703-231-9100
Principle Investigator	Uri Vandsburger
Co-Investigators	Seshu B. Desu, MSE, Virginia Tech Larry A. Roe, ME, University of Arkansas
Graduate Research Assistants	S. D. LePera, ME, Virginia Tech Y. Yuan, ME, Virginia Tech Y. Song, MSE, Virginia Tech (2 funded by contract)
DOE Project Manager	Dr. Paul Michel/Dr. Norm Holcombe
Period of Performance	16 August 1993 to 15 August 1996

SCHEDULE AND MILESTONES

Table 1. FY 94 - FY 96 Program Schedule

Task	Month	0	3	6	9	12	15	18	21	24	27	30	33	36
Reacting Flows Actuation Methods Actuator Specifications														
Real Time Flow Control-Non Reacting														
Development of Low Energy Actuators														
Flow Actuation Methods for Noisy Flows														
Development of High Tempera- ture Actuators														
Development of Controlled React- ing Flows Apparatus														
Optical Sensing Technology														
Development of High Deflection Relaxor Materials														
Test New Actuation Methods														
Test New Actuator Materials														
Novel Actuators														
Test Integrated Actuators for Combustion Systems														
Failure Mechanisms of Actuators														

OBJECTIVES

The work presented herein addresses the possibility of controlling the performance of a combustor through direct control of the flowfield structure. The specific objectives are:

- attain control over performance of the combustor, i.e., power and efficiency, and emission levels,
- develop methods based on the physics of the flowfield, and
- develop practical actuators for combustors with the least energy consumption.

To achieve these goals the following have to be realized and demonstrated:

- develop new actuator materials for high temperature operation, and with high deflection,
- develop new actuation modes for shear flows,
- prove the effectiveness of the methodologies and materials in nonreacting, noisy, and reacting environments with shear flows.

BACKGROUND AND MOTIVATION

Emissions from simple-cycle and combined-cycle stationary gas turbines are the primary factor driving both the regulatory environment and the economic realities of new power generating systems. Small deficiencies in efficiency can be dealt with by adjustments of cost-of-electricity to the end-user, but failure to meet environmental regulations can result in a cessation of operation. Because of recently implemented regulations allowing the marketing of pollution credits, simply meeting current or anticipated emission standards is no longer sufficient for the most profitable operation. Since post-combustion gas cleaning has not been shown to be a particularly attractive option, the system component which requires continued development for the reduction of emissions is the combustor.

In the combustor, mixing is the critical process controlling emissions, regardless of the design philosophy. Even small variations in the degree of mixing of reactants and products can increase emissions. In lean premixed designs and the ABB EV geometry, enhanced mixing of the fuel and air is the primary means of emissions reduction. In rich-quench-lean designs, not only is proper mixing in the reaction zones critical, but rapid mixing in the quench section is essential to minimize the residence time requisite for NO_x production. Continued improvements to mixing methods can be expected to further reduce emissions. The mixing process inside the combustor is governed by the characteristics of the specific shear flow, e.g., jets, planar or curved mixing layers. Improved mixing can only be achieved through active manipulation of the shear flow characteristics, which in turn requires detailed knowledge of the physics of these flows.

New understanding of turbulent shear flows gained in the 1970's and 1980's replaced the statistically random concept of turbulence with one containing organized large-scale motions with

continuously evolving smaller scales. This suggests the development of systems permitting control over the spatial, temporal, and spectral evolution of chemically reacting flows in combustion systems. Control of flows in gas turbine combustors must address three major issues: stability of operation (combustion), efficiency of operation, and pollutant emissions. Most studies to date have addressed the topic of combustion instabilities on a fundamental level, i.e., understanding the basic mechanisms involved. Very few flow control studies have considered the reduction of emissions in industrial power generation systems. With the exception of fuel management, the studies did not offer any methodologies based on fundamental fluid mechanics or ones that could be developed into practical technologies.

In current designs, shear layer and mixing characteristics evolve as a result of the establishment of fuel and air flows in fixed combustor geometries. Good emissions performance is obtained under design conditions only if the geometry has been properly optimized during the design and development stages. Off-design performance necessarily suffers, and optimization of the mixing characteristics in response to the actual operating parameters of an in-service engine is not possible.

The drive toward cleaner, more efficient, and stably operating plants is pushing the commonly-used methodologies beyond their limits. As a result, designers of combustion devices from both the power generation equipment industry and aerospace industry have come to the realization that simultaneous achievement of fuel economy, stable combustion and compliance with environmental standards will require active control of the combustor performance. This implies the inclusion of active (dynamic) elements, adequate real-time sensing systems, and state-of-the-art adaptive control logic.

To assist the industry in achieving these goals it is necessary to study and develop methodologies for active combustion control, a process that must address the following aspects:

- a. achievement of an improved understanding of the physics of vortex-dominated flows,
- b. the development of strategies for control of reacting flows to achieve a desired distribution of energy release density in the combustor, thus maximizing efficiency and minimizing emissions,
- c. the development of practical actuators,
- d. the development of real-time, in-situ, field sensors,
- e. the development of a practical and realizable advanced control methodology for such distributed, nonlinear systems, and
- f. the development of new sensors and actuators and their integration into the combustion and control systems.

The work reported here is aimed at advancing the application of adaptive controls to combustion systems to achieve optimized performance under lean-burn conditions at varying loads. In contrast to the majority of the research conducted to date, this is to be accomplished by controlled excitation of the shear flows within the combustor.

Areas of potential use for this technology in a real gas turbine combustor are shown schematically in the Figure 1.

Before any closed loop control system can be utilized for mixing control in gas turbine combustors, it will be necessary to develop actuators and sensors which can fulfill their function over prolonged periods in the harsh environment of a gas turbine combustor. In addition, a clear understanding of the physics of fluids is necessary to train the smart controller.

Figure 1 Conceptual application of active shear flow control (sensors and actuators) in a gas turbine combustor.

As a consequence of the above, the present work focuses on the following aspects:

- a. development of flow control methods for combustor-like flows
- b. development of new actuator materials
- c. development of different actuator geometries for combustor use,
- c. optimization of sensing methodologies,
- e. testing the new actuators in combustion systems and comparing their performance with existing, yet impractical, technologies, and
- r. explore the possibilities for fabrication of integrated actuator-sensor units for combustor use.

The most common actuators used in studies of shear flows to date have been acoustic sources, e.g. loud speakers. These are clearly impractical. In the present effort piezoelectric materials are used for as part of the actuator structure.

Piezoelectric actuators are widely used because of their compact size, low cost, low power consumption, and fast response times. By the application of an electric field, these actuators

produce a shape change utilizing the converse piezoelectric effect $\epsilon_3 = d_{33}E_3$, where ϵ_3 , d_{33} , and E_3 represent mechanical strain, piezoelectric coefficient and electric field, respectively. Actuators for typical shear flows in a combustor will have to meet, on average, the requirements of:

- a. provide displacements in the range of 0.1 to 1.0 mm,
- b. survive at high temperatures (up to 1300K), and
- c. provide reliable and repeatable performance over the life time.

Use of piezoelectric materials with high d_{33} coefficients will result in high displacements. Materials such as BaTiO_3 and $\text{Pb}(\text{Z}_{1-x}\text{Ti}_x)\text{O}_3$ (PZT) show d_{33} coefficients of the order of 150 to 550 Pc/N , whereas relaxor materials such as $\text{Pb}(\text{Mg}_{1/3}\text{Nb}_{2/3}))_3$ (PST), and $\text{Pb}(\text{Fe}_{2/3}\text{W}_{1/3}\text{O}_3$ (PFW) have larger coefficients in the 2000-5000 Pc/N range. However, the maximum operating temperature of these relaxor materials is limited to around 400K.

One part of this effort concentrates on the relaxor materials due to their high d_{33} values. In addition, these materials are truly adaptive in that they sense a change in its environment and respond by producing temporal and spatial changes in their actuator functions. *In reality, the strain of a relaxor is proportional to square of the electric field for small fields and saturates at larger fields.* Due to this nonlinear behavior their response can be controlled by external bias forces. The nonlinear properties of these electrical ceramics are often associated with nano-scale fluctuations in the composition and diffuse phase transitions. Due to the fluctuations in compositions, these materials exhibit large d_{33} values, electro-optic and acousto-optic effects.

Because of the complex nature of the interrelationships between processing parameters,

structure and properties, these materials were insufficiently investigated. A novel processing technique, *Molten Salt Synthesis*, for fabricating these materials at high compositional and phase purity has been developed at Virginia Tech. This new technique will allow the study of these interrelationships in greater detail such that a firm basic foundation can be established for the design and development of reliable actuators.

The second part of this effort seeks to develop high temperature materials such as $\text{Sr}_2\text{Nb}_2\text{O}_7$ and $\text{La}_2\text{Ti}_2\text{O}_7$ for the high temperature actuator applications. This part of the effort includes developing processes for obtaining dense ceramics and films, fabricating, and characterizing the actuators based on the two forms.

These high temperature materials have very low d_{33} values in the range of 10-25 pC/N . In order to obtain high displacements using these materials strain magnification mechanisms have to be applied. Strain magnification can be achieved through actuator designs such as monomorphs and Moonie actuators. In the case of Moonie actuators, the end caps on the transducer act to make contributions from thickness and planar distortions to add, rather than subtract, resulting in high displacements.

Since actuator reliability is the most critical issue in actuator development, the effort also addresses the issues of fracture toughness by grain size control, failure prediction by extending existing unified models, and accelerated life tests.

PROJECT DESCRIPTION AND RESULTS

To realize the goals listed before a coordinated dual area approach is undertaken, parallel work is executed in the fields of flow and combustion control and new ceramic materials for actuators, the interaction is shown schematically in the Figure 2.

Figure 2 Ceramic Actuators for Improved Combustor Performance

MIXING AND COMBUSTION CONTROL

Background

In the present effort control over the performance of the combustor is to be achieved through active control of the flowfield structure and the mixing between initially unmixed fluids, e.g., fuel and oxidizer. These fluid mechanical scenarios fall into the category of free shear flows. Control of shear flows (which in reality is only their excitation) has been the subject of concentrated attention for over a decade. Work was done on planar mixing layers and jets. The fundamentals of planar mixing layer characteristics and its excitation can be found in the works of Ho and Huerre [1], Oster and Wygnanski [2], Fiedler and Mensing [3] and Dimotakis [4]. Three dimensional evolution of planar mixing layers was also investigated, see Lasherns and Choi [5] and Nygaard and Glezer [6].

Jets constitute another form of shear flows. In contrast to planar mixing layers jets contain an additional length scale, the diameter, which gives them additional and unique instability modes. The hydrodynamic stability characteristics of jets have been studied quite thoroughly, especially over the last decade. While many authors contributed to the advances made, the most relevant recent work to the current effort are those of Cohen and Wygnanski [7], Long and Peterson [8] and Corke and Kusek [9]. Before further details of jet flow control are discussed it

has to be mentioned that two methodologies are used to actuate/excite jets, known as passive and active. Passive control involves the use of orifices or nozzles of a non-circular cross section. It has been shown that these can have a dramatic effect on the jet flowfield as reported for example by Gutmark et al. [10,11] and Schadow et al. [12,13]. The advantage of the passive method is that no energy input is required. At the same time, however, these are passive means and therefore do not enable continuous optimization.

The alternative is active control of the jet evolution through introduction of perturbations into the jet shear layer. The perturbation can be small with the frequency as the parameter. Larger perturbation can also be used. Another type of excitation involves spatial, instead of frequency, modes and was shown to enable very good control over the evolution of jets in the near field, see Cohen and Wygnanski [7] and Long and Peterson [8]. The actuators in almost all these studies were acoustic sources, e.g., loud speakers. Studies investigating active control of jets exhibited two weaknesses. First, control was only achieved in the near field $X/D < 10$. Second, the actuators used are impractical for "real-world" systems, especially combustors. Therefore, the two key points that needed to be addressed in order to make active flow control a solution for combustor performance control are:

- a. attainment of control over the jet evolution well into the far field, and
- b. development of practical actuators.

The first issue has been addressed in a study preceding and parallel to the one reported herein. The focus on that study has been the development of new excitation modes for jets which will enable control over the whole flowfield. The first stage of that study was performed on a triangular nozzle with piezoceramic actuators mounted on the flat sides. The jets were highly turbulent with intensities up to 9% which made their excitation

difficult.

The jets were excited at various spatial modes. New fractional and counter rotating modes were developed. Isovelocity contours of half the local centerline velocity, taken from Ding and Vandsburger [14], are shown in Figure 3. Further analysis revealed the advantage of this mode in enabling control over the flowfield and resulting in dramatic expansion in the far field. Shown in Figure 4 are cross sections of the jet at $X/D = 30$ for various excitation modes, from Vandsburger and Ding [15]. Examination of the entrainment rates of ambient fluid by the jet revealed a threefold increase for the new fractional mode excitation compared with the unexcited jet and, less than twofold for the simple $m=1$ mode. In a followup study the same modes were investigated on a round nozzle, more realistic for practical devices. New real time control schemes were developed, which enable control over the jet cross section in time and space. All this technology is intended to be transferred to the field of reacting flows under the ATS program.

Flow and Combustion Control Experimental Facilities

Two experimental setups were constructed for the present project. For the work on combustion, reacting flows control, a jet in co-flow rig was designed fabricated and assembled. The apparatus is of the non-circular active jet in co-flow type. This apparatus, shown in Figure 5, enables the evaluation of excitation modes for mixing enhancement of initially non-premixed combustible gases. The data acquired is optical in nature, including Schlieren, emission, and laser light scattering images of the flows, natural and excited. Several light sources and CCD detectors are available. Phase-locked data acquisition is possible, enabling the reconstruction of 3-D field images. A system setup is shown in Figure 6,

and an arrangement of the optical diagnostics in Figure 7.

The second aspect of the effort presently executed examines the effectiveness of the spatial modes developed for jets issuing into stagnant air, in a noisy co-flow environment. For that purpose, a two-stream planar mixing layer wind tunnel was converted to a jet in a co-flow. The jet issues out of an active square nozzle, with piezoceramic actuators mounted on the flat sides. A schematic is shown in Figure 8. The facility enables the imposition of an acoustic field inside the test duct, through the use of loud speakers. A system setup, including optical diagnostics is shown in Figure 9.

Results

The actuation modes of jets which were developed earlier, Ding and Vandsburger [14], were examined on the triangular jet apparatus. Two types of experiments were performed. Nonreacting flows where CO_2 was ejected from the nozzle, and reacting flows using C_3H_8 as the nozzle fluid. A stream of air supplied by a compressor constituted the co-flow, whenever used. The jet issued into still room air.

The effect of the excitation modes on the jet evolution were examined using Schlieren photography. A phase-locked Schlieren system, with a Xenon flash lamp (10 μsec pulse) and an unintensified CCD camera, was synchronized with a PC486. Phase locking was accomplished by triggering the flashlamp power supply by the PC which also sent the signals to the piezoceramic actuators. The non-reacting jets had a Reynolds number of 12,600 based on the nozzle hydraulic diameter. Shown in Figure 10 are an unexcited jet in part a, and the jet excited at $m = \pm 0.5$ mode. Larger farfield expansion and asymmetric evolution is visible in 10b,c.

In the reacting flows two Reynolds

Figure 3 IsovLOCITY surface $u(x,t)/u(x,t)_{\max}=0.5$. Excited triangular nozzle, $f=80$ Hz. All data for 0° in excitation cycle. m =azimuthal mode number; $m=0$ axisymmetric mode.

Figure 4 Iso-velocity contours, $u(x)/U_{x(max)} = 0.5$ of a triangular jet excited with different modes, $X/D = 30$.

Figure 5 Triangular reacting jet experimental apparatus

numbers were tested. First a low Reynolds number 12,600 (same as the cold flows) was examined. To reduce the flame heat release the fuel stream was diluted with CO_2 . An unexcited jet and one excited with $m=0$ mode are shown in Figure 11 a and b, respectively. A large expansion and hot gas envelope is visible for the excited jet. High Reynolds number, $\text{Re} = 55,700$, results are shown in Figure 12a,b,c. The effects of the excitation are clearly visible even at this higher Reynolds number. A visual photograph of an unexcited and excited flame at $m = \underline{\quad}$ are shown in Figure 13. A noticeable shortening and thickening of the flame is observable.

After these preliminary results the behavior of the piezoceramic driven actuators was investigated. Using laser Doppler Vibrometry the phase and amplitude of the actuators relative to the excitation signal were measured. The magnitude

of the response was found to deviate only 5% from linear, while the phase lag varied only by ± 1.5 degrees over the whole range of excitation magnitudes. All three actuators responded similarly, implying that corrections to the phase and amplitude will have negligible effect on the results obtained.

As the flow rate of the jet increases and ambient noise levels become high, it is expected that energy input requirements into the actuators will rise too. Flow excitation can be deemed a useful method only if the energy requirements of the flow actuators are small compared with the fluid energy. As a consequence an effort was launched to develop flow actuators which will combine self-amplified excitation with accurate phase control by piezoceramic actuators. The first generation of a working device is being tested during the writing of this document. Data is also being acquired from the ducted jet facility,

Figure 6 Triangular reacting jet system schematic

Figure 7 Optical diagnostic layout for Schlieren and planar measurements in reacting jet

Figure 8 Schematic of ducted jet wind tunnel.

Figure 9 Test section configured for optical and acoustical measurements

Figure 10. a) Unexcited CO₂ jet orientation A, b) Excited CO₂ jet, mode +/- .5, orientation A, c) Excited CO₂ jet, mode +/- .5, orientation B.

Figure 11. a) Unexcited $C_3H_8/CO_2/Air$ jet with flame stabilized above triangle jet, b) Excited $C_3H_8/CO_2/Air$ jet with flame, mode 0.

a)

b)

c)

Figure 12. a) Unexcited C_3H_8 /Air flame, b) Excited C_3H_8 /Air flame, mode 0, c) Excited C_3H_8 /Air flame, mode +/- 0.5.

Figure 13 Burning propane jet, left unexcited, right excited. Center, triangular jet apparatus

see Figure 6, with preliminary data to be presented during the meeting.

SYNTHESIS OF NEW PIEZOELECTRIC ACTUATOR MATERIALS

Introduction

The $A_2B_2O_7$ type ($La_2Ti_2O_7$ and $Sr_2Nb_2O_7$) material is known to belong to the family of ferroelectrics with a perovskite-layer structure (PLS). Single crystals of $La_2Ti_2O_7$ and $Sr_2Nb_2O_7$ exhibit excellent piezoelectric and electro-optic properties, Kimura et al. [16], Nanamatsu et al. [17] and show an extremely high Curie temperature ($T_c=1500C$), Nanamatsu et al [18]. It has been reported that $La_2Ti_2O_7$ and $Sr_2Nb_2O_7$ have two phase transition temperatures, Nanamatsu et al [18]. At ferroelectric phase transition temperature, both $La_2Ti_2O_7$ and $Sr_2Nb_2O_7$ transform from orthorhombic space group $Cmc2_1$ to symmetry $Cmcm$, Ishizagaya et al [20,21]. The structure of $A_2B_2O_7$ type was characterized by corner-shared BO_6 octahedra and 12-coordinated A cations within the perovskite-like slabs, which are

connected by A cations at their boundaries.

The $La_2Ti_2O_7$ and $Sr_2Nb_2O_7$ materials were chosen to satisfy the requirement of high temperature actuators, which is to maintain their standard performance level at high reaction flow temperature. Due to their high Curie temperature, the $A_2B_2O_7$ ceramics are promising for high temperature piezoelectric and electrooptic devices.

The $A_2B_2O_7$ type ($La_2Ti_2O_7$ and $Sr_2Nb_2O_7$) and solid solution $Sr_2(Ta_{1-x}Nb_x)_2O_7$ powders were prepared using three different fabrication methods: (1) conventional method (CON method), Marzullo and Bunting [22], MacChesney [23], (2) Molten salt method (MSS method), Arendt et al [24], and (3) Chemical precipitation method (CCP method), Potdar et al [25]. The increase of grain-orientation value was obtained through CCP method, while the decrease of sintering temperature was obtained through MSS method. The microstructures and electrical properties were observed and investigated. For thin films, $La_2Ti_2O_7$ and $Sr_2Nb_2O_7$ solutions were synthesized and deposited on substrate using spin-coating method. Then, the single phase

Figure 14 X-ray diffraction patterns for sintered $\text{La}_2\text{Ti}_2\text{O}_7$ samples prepared by molten salt method [a] and conventional method [b]

formations and their X-ray patterns were observed.

Experimental Procedure

The $\text{A}_2\text{B}_2\text{O}_7$ type $\text{La}_2\text{Ti}_2\text{O}_7$ and $\text{Sr}_2\text{Nb}_2\text{O}_7$ powders were prepared using three different methods; conventional method (CON), molten salt synthesis method (MSS), chemical coprecipitation method (CCP). For $\text{La}_2\text{Ti}_2\text{O}_7$ and $\text{Sr}_2\text{Nb}_2\text{O}_7$ thin film, metallorganic deposition method (MOD) was used.

Powder Synthesis

(A) Conventional Method (mixed oxide): Reagent-grade (purity 99.9%) component oxides in 1:2 mole ratio were mixed for 5 hrs in a crucible including alumina media and alcohol. The resulting slurry was dried at 150°C and calcined at various temperatures for 3 hrs to obtain the single phase formation.

Figure 15 X-ray diffraction patterns for sintered $\text{Sr}_2\text{Nb}_2\text{O}_7$ samples prepared by chemical coprecipitation method [a], molten salt method [b] and conventional method [c]

(B) Molten Salt Method (flux method): Reagent-grade component oxides in 1:2 mole ratio and salt of 50 mol% NaCl and 50 mol% KCl were mixed for 3 hrs in alcohol and alumina media. The weight of the salt constitutes 50 wt% of the total reaction mixture. The mixed slurry was dried at 150°C and calcined at various temperatures ranging from 700°C to 1200°C. The salt within calcined powders was washed repeatedly by deionized water and centrifugation until the specific conductance of the decanted liquid fell below 3S which was measured using a conductance meter.

Figure 16 Density and grain-orientation dependence on sintering temperature for $\text{Sr}_2\text{Nb}_2\text{O}_7$ sample prepared by chemical coprecipitation method.

(C) Chemical Coprecipitation

Method: The component precursors in equal mole ratio and oxalic acid were used for preparing the $\text{A}_2\text{B}_2\text{O}_7$ type powders. The precursors were added dropwise into oxalic acid and stirred until a clear solution was observed. After obtaining the completely mixed clear solution, ammonium hydroxide was added until the fine particles were precipitated. The precipitated solutions were carefully washed using deionized water, and then dried at 150°C. The resulting fine precipitate was calcined at a temperatures of between 600°C and 1300°C for 3 hrs. The resulting powders were pressed into pellets by applying 1000 psi/cm², and then the pellets were sintered at various temperatures. The XRD patterns were used for phase identification of various powders and sintered bodies. The microstructures of both powders and sintered

pellets were investigated using a scanning electron microscopy (SEM). Densities were estimated by the Archimedes method, and plotted as a function of sintering temperatures. Electrical properties such as dielectric loss ($\tan^{-1}$), capacitance, and resistivity, were measured at high frequency using a Q-meter.

Thin film preparation

Metallorganic Deposition Method: The starting compounds were dissolved in equal mole ratio in the solvent. The solution was stirred on hot plate to adjust the concentration. The stirred solution was filtered to 0.45μm particle size and then using the spin-coating method, five droplets of the filtered solution were deposited onto the substrate and spun at a velocity of 2000rpm/min for 40secs. The resulting substrate with a newly

Figure 17 X-ray diffraction patterns for $\text{Sr}_2(\text{Ta}_{1-x}\text{Nb}_x)_2\text{O}_7$ samples ($x = 0.0, 0.8, 1.0$) sintered at 1400°C at different times during the process samples were prepared by (a) conventional method (CON) and (b) chemical coprecipitation method (CCP).

Figure 18 Relative permittivity at Curie temperature and grain orientation vs. composition x for $\text{Sr}_2(\text{Ta}_{1-x}\text{Nb}_x)_2\text{O}_7$ samples

Figure 19 SEM micrographs for $\text{Sr}_2(\text{Ta}_{1-x}\text{Nb}_x)_2\text{O}_7$ samples sintered at 1500°C by conventional method ($x = 0.0 - 1.0$)

formed layer of thin film was placed on hot plate at 200C to evaporate the solvent. This process was repeated four times, the number of repetitions depending on the desired film thickness. The thin films were baked at 450C for 3 hrs under oxygen atmosphere, and then annealed at different temperatures.

Results

$\text{La}_2\text{Ti}_2\text{O}_7$ and $\text{Sr}_2\text{Nb}_2\text{O}_7$ ceramics

A single phase of $\text{La}_2\text{Ti}_2\text{O}_7$ and $\text{Sr}_2\text{Nb}_2\text{O}_7$ powders was formed at a lower temperature using MSS and CCP method compared to CON method. The particle sizes of CCP-derived powders were smaller and finer than that obtained from other

two methods. Figures 14 and 15 illustrate XRD patterns for sintered $\text{La}_2\text{Ti}_2\text{O}_7$ and $\text{Sr}_2\text{Nb}_2\text{O}_7$ sample respectively prepared by various methods. The theoretical densities of $\text{La}_2\text{Ti}_2\text{O}_7$ and $\text{Sr}_2\text{Nb}_2\text{O}_7$ reached above 97% through MSS and CCP method for bulk materials, and a very large value of grain orientation was obtained by the CCP method without hot-pressing. The grain-orientation factor increased steeply with increasing sintering temperature and this grain-orientation contributed to the improvement of electrical property. Figure 16 shows the increase of grain-orientation as a function of sintering temperature. Electrical properties such as dielectric constant and dielectric loss were better in CCP method than in CON or MSS methods. The resulting characteristics of $\text{La}_2\text{Ti}_2\text{O}_7$ and $\text{Sr}_2\text{Nb}_2\text{O}_7$ prepared by various

Table 1. Typical characteristics of $\text{La}_2\text{Ti}_2\text{O}_7$ and $\text{Sr}_2\text{Nb}_2\text{O}_7$ prepared by various methods

	$\text{La}_2\text{Ti}_2\text{O}_7$			$\text{Sr}_2\text{Nb}_2\text{O}_7$		
	CON	MSS	CCP	CON	MSS	CCP
Calcination temperature for single phase (°C) Intermediate phase	1000	900 LaTiO_3 (at 700°C)	•	900 *	850	750 $\text{Sr}_5\text{Ti}_4\text{O}_{15}$
Maximum value of density (%)	94	97	•	93	97	97
Maximum value of grain-orientation factor	•	•	•	0.09	0.16	0.54
Dielectric constant (at 100 Hz)	55	63	•	51	60	75
Dielectric loss	0.00 2	0.001	•	0.008	0.00 4	0.003

*In case of conventional method, no definite intermediate phase was identified.

fabrication methods are summarized in table 1.

The molecular level mixing of the precursors in CCP method led to a homogeneous

mixture of ultra-fine particles which contributed to extra advantages of CCP method. The advantages of CCP method over CON and MSS

Methods are to obtain finer and more uniform particles, synthesize powder at lower temperature, obtain large grain-orientation without hot-pressing, and improve electrical properties.

$\text{Sr}_2(\text{Ta}_{1-x}\text{Nb}_x)_2\text{O}_7$ ceramics

Effects of composition x on sintering behavior and dielectric properties of $\text{Sr}_2(\text{Ta}_{1-x}\text{Nb}_x)_2\text{O}_7$ was investigated. Theoretically density, grain orientation and grain size should increase as a function of composition x . Shown in Figure 17 are XRD patterns for $\text{Sr}_2(\text{Ta}_{1-x}\text{Nb}_x)_2\text{O}_7$ samples prepared by CON and

CCP methods, which were sintered at 1400C, at different times during the process. Note that $\langle 0k0 \rangle$ peak grew as the amount of composition x and sintering time increased. The large value of grain orientation was obtained through CCP method. The dependence of grain-orientation and relative permittivity on composition x is plotted in Figure 18. SEM pictures shown in Figure 19 confirmed the increase of grain size as composition x increased. Curie temperature and dielectric constants at Curie temperature increased with X , but dielectric constants at room temperature stay almost constant except $x=0.0$. The summary of sintered $\text{Sr}_2(\text{Ta}_{1-x}\text{Nb}_x)_2\text{O}_7$ samples is shown in Table 2.

Table 2. Characteristics of $\text{Sr}_2(\text{Ta}_{1-x}\text{Nb}_x)_2\text{O}_7$ Ceramics sintered at 1500C for 3h

	$x = 0.0$ $\text{Sr}_2\text{Ta}_2\text{O}_7$	$x = 0.2$	$x = 0.4$	$x = 0.6$	$x = 0.8$	$x = 1.0$ $\text{Sr}_2\text{Nb}_2\text{O}_7$
Curie Temp. ($^{\circ}\text{C}$)	-107	410	735	1000	1160	1342
[19]		170	315	410	1105	1415
em (at Curie temp.)	76	59	58	60	59	59
er (at room temp)						
Theoretical density (g/cm^3)	7.02					5.15
Density (g/cm^3)	6.15	5.84	5.64	5.53	5.19	4.85
(% density)	(87.6)					(94.2)
Grain orientation	0.01	0.03	0.06	0.22	0.29	0.34

FUTURE WORK

Flow and Combustion Control

During the second year work will be concentrated in the following areas:

- excitation of shear flows in noisy environments
- development and validation of low energy

- actuators for practical combustors
- testing of new high temperature piezo electric materials in a reacting flow
- devising of a useful, and practical, sensing system
- assembly of a combustor system for testing of control of ducted, reacting shear flows.

The combustor facility will be assembled

at the University of Arkansas.

New Piezoelectric Materials

The work will focus mostly on thin films, which will be applied to actuator structures. Specifically, the properties of the thin films will be characterized.

Testing the possible application of thin films made out of the new $A_2B_2O_7$ type materials will be undertaken. Application of the new materials in reacting flows and evaluation of their failure mechanisms will take place. Work will start on development of large deflection, relaxor type, actuators.

REFERENCES

1. Ho, C.-M. and P. Huerre. 1984. Perturbed Free Shear Layers. *J. Fluid Mech.* Vol. 16, pp. 365-424.
2. Oster, D., and I. Wygnanski. 1982. The Forced Mixing Layer Between Parallel Streams. *J. Fluid Mech.* Vol. 123,, pp. 91-130.
3. Fiedler, H. E. and P. Mensing. 1985. The Plane Turbulent Shear Layer with Periodic Excitation. *J. Fluid Mech.* Vol. 150, pp. 281-309.
4. Dimotakis, P.E. 1991. Turbulent Free Shear Layer Mixing and Combustion in High Speed Flight Propulsion Systems. In *Progress in Aeronautics and Astronautics*, Vol. 137.
5. Lasheras, J. C., and E. Choi. 1988. Three Dimensional Instability of Plane Shear Layer: An Experimental Study of the Formation and Evolution of Streamwise Vortices. *J. Fluid Mech.* Vol. 189, pp. 53-86.
6. Nygaard, K. J., and A. Glezer. 1991. Evolution of Streamwise Vortices and Generation of Small-Scale Motion in a Plane Mixing Layer. *J. Fluid Mech.* Vol. 231, pp. 257-301.
7. Cohen, J., and I. Wygnanski. 1987. The Evolution of Instabilities in the Axisymmetric Jet. (Part I. The Linear Growth of Disturbances Near the Nozzle, and Part 2. The Flow Resulting from Interaction Between Two Waves). *J. Fluid Mech.* Vol. 176, pp. 191-235.
8. Petersen, R. A., and T. A. Long. 1992. Controlled Interaction in a Forced Axisymmetric Jet (Part 1. The Distortion of the Mean Flow, and Part 2. The Modulation of Broad-Band Turbulence). *J. Fluid Mech.* Vol. 235, pp. 37-72.
9. Corke, T. C., and S. M. Kusek. 1993. Resonance in Axisymmetric Jets with Controlled Helical-Mode Input. *JFM* 249, pp. 307-336.
10. Gutmark, E., K. C. Schadow, T. P. Parr, D. M. Hanson-Parr, and K. J. Wilson. 1989. Flow Characteristics of Orifice and Tapered Jets. *Phys. of Fluids.* Vol. 30, No. 11, pp. 3448-3454.
11. Gutmark, E., and K. C. Schadow. 1989. Non Circular Jets in Combustion Systems. *Experiments in Fluids.* Vol. 7, pp. 248-258.
12. Schadow, K. C., E. Gutmark, D. M. Parr, and K. J. Wilson. 1988. Selective Control of Flow Coherence in Triangular Jets. *Experiments in Fluids.* Vol. 6, pp. 129-135.
13. Schadow, K. C., E. Gutmark, K. J. Wilson, and R. A. Smith. 1990. Multi-Step Dump Combustor Design to Reduce Combustion Instabilities. *J. Prop. and Power.* Vol. 6, No. 4, pp. 407-411.

14. Ding, C. and U. Vandsburger. 1992. Control of Vertical Flows, Part I: Excited Jets. in Recent Advances in Active Control of Sound and Vibration, Technomic Press.
15. Vandsburger, U., and C. Ding. 1994. The Spatial Modulation of a Forced Triangular Jet. Experiments in Fluids, in press.
16. M. Kimura, S. Nanamatsu, K. Doi, S. Matsushita, and M. Takahashi. 1972. Electrooptic and Piezoelectric properties of $\text{La}_2\text{Ti}_2\text{O}_7$ Single Crystal. Jpn. J. Appl. Phys., 11, 904.
17. M. Nanamatsu, M. Kimura, K. Doi and M. Takashi. 1971. Ferroelectric Properties of Single Crystal. J. Phys. Soc. Japan, 30, pp. 300-30.
18. S. Nanamatsu, M. Kimura, K. Doi, S. Matsushita, and N. Yamada. 1974. A New Ferroelectric: $\text{La}_2\text{Ti}_2\text{O}_7$. Ferroelectrics, 8, 511-13.
19. Nanamatsu, S., M. Kimura, and T. Kawamura. 1975. Crystallographic and Dielectric Properties of Ferroelectric $\text{A}_2\text{B}_2\text{O}_7$ (A=Sr, B=Ta,Nb) Crystals and Their Solid Solutions. J. Phys. Soc. Japan, 38(3), pp. 817-24.
20. Ishizawa N., F. Marumo, T. Kawamura and M. Kimura. 1975. The Crystal Structure of $\text{Sr}_2\text{Nb}_2\text{O}_7$, a Compound with Perovskite Type Slabs. Acta Crystallogr. B31, pp. 1912-15.
21. N. Ishizawa, F. Marumo, S. Iwai, M. Kimura, and T. Kawamura. 1982. Compounds with perovskite Slabs V. A High-temperature Modification of $\text{La}_2\text{Ti}_2\text{O}_7$. Acta Crystallogr., B38, 368-72.
22. S. Marzullo and E. N. Bunting 1958. Dielectric Properties of Titania or Tin Oxide Containing Varying Proportions of Rare-Earth Oxides. J. Am. Ceram. Soc., 41, 40-41.
23. J. B. MacChesney and H. A. Sauer. 1962. The System $\text{La}_2\text{O}_3\text{TiO}_2$: Phase Equilibria and Electrical Properties. J. Am. Ceram. Soc., 45, 416-22.
24. R. H. Arendt, J. H. Rosolowski, and J. W. Szymaszek. 1979, Lead Zirconate Titanate Ceramics from Molten Salt Solvent Synthesized Powders. Mater. Res. Bull., 14, 703-709.
25. Potdar, H. S., P. Singh, S. B. Deshpande, P. D. Godbole and S. K. Date. 1990. Low-temperature synthesis of ultrafine barium titanate (BaTiO_3) using organometallic barium and titanium precursors. Materials Letters, 10(3), pp.112-17.

Poster Session

New Research Projects

CONTRACT INFORMATION

Contract Number

Clemson U. Research Foundation/94-01-SR021/DOE

Contractor

University of Minnesota

Department of Mechanical Engineering

111 Church St. S. E.

Minneapolis, MN 55455

Phone: 612-625-5831

FAX: 612-624-1398

Head P.I.

R. J. Goldstein

CO-Investigators

S. V. Patankar, T. W. Simon,

E. R. G. Eckert, and H. H. Cho

Period of Performance

July 1, 1994 to June 30, 1997

Schedule and Milestones

FY 1994 - 1997

	FIRST YEAR	SECOND YEAR	THIRD YEAR
	Acquisition of equipment and construction and qualification of test facility	Development of turbulent heat flux measurements	Single row of shaped holes (A.R. @ 2.5)
Flow Measurement	Implementation of 3-D flow measurements	Effect of high T.I. and lateral injection angle ($\beta=30^\circ$ & 60°)	Effect of curvature
	Single row of holes ($\beta=45^\circ$, T.I. @ 0.5%, and M=1, 2, and 4)	Double rows of holes	
Surface Measurement	Acquisition of equipment and construction and qualification of test facility	Single row Effect of high T.I. and lateral injection angle ($\beta=30^\circ$ & 60°)	Effect of curvature (Single/double rows)
	Single row of holes ($\beta=45^\circ$, T.I. @ 0.5%, and M=1~4)	Double rows ($\beta=45^\circ$ and T.I. @ 0.5%)	Single row of shaped holes (A.R. @ 2.5)
Numerical study	Compound angle injection Effect of blowing rate	Effect of pressure gradient and inlet turbulence	Parametric studies Consideration of advanced turbulence models

EXPERIMENTAL AND COMPUTATIONAL STUDIES OF FILM COOLING WITH COMPOUND ANGLE INJECTION

by

T. W. Simon, R. J. Goldstein, S. V. Patankar, E. R. G. Eckert, and H. H. Cho

Heat Transfer Laboratory, Department of Mechanical Engineering
University of Minnesota, 111 Church Street SE
Minneapolis, MN 55455

OBJECTIVES

The thermal efficiency of gas turbine systems depends largely on the turbine inlet temperature. Recent decades have seen a steady rise in the inlet temperature and a resulting reduction in fuel consumption. At the same time, it has been necessary to employ intensive cooling of the hot components. Among various cooling methods, film cooling has become a standard method for the cooling of the turbine blades and combustion chamber walls. The University of Minnesota program is a combined experimental and computational study of various film-cooling configurations.

Whereas a large number of parameters influence the film cooling process, this research focuses on compound angle injection through single row and double rows of holes with high free stream turbulence, streamwise pressure gradients, and curvature. The advantages of compound angle injection have only recently been understood; this project should further this understanding.

Approaches to be applied include: (1) a new measurement system that extends the mass/heat transfer analogy to obtain both local film cooling and local mass (heat) transfer results in a single system, (2) direct measurement of three-dimensional turbulent transport in a highly-disturbed flow, (3) the use of compound angle and shaped holes to optimize film cooling performance, and (4) an exploration of anisotropic corrections to turbulence modeling of film cooling jets.

The outcome of this research will be threefold. First, it will provide fundamental scientific information in the form of detailed measurements, computational results, and turbulence model validation. Second, the results of the experiments and computations will be presented in a generalized form so that they will be directly usable by design engineers in industry. And finally, the experimental and computational activities will be used to train graduate students.

BACKGROUND INFORMATION

Numerous papers have been written over the past 40 years on the performance of film cooling schemes. Many of these have emanated from the Heat Transfer Laboratory and have been instrumental in the steady rise in turbine inlet temperature over that same time. The most common tests were for slot injection or for flows with in-line injection, although a few have shown that a compound-angle injection scheme promises to provide a more uniform coverage of film cooling protection. One early study, conducted in the Heat Transfer Laboratory, showed that for high blowing rates, the compound angle injection scheme allows higher blowing rates without lift-off of the cooling film (Goldstein et al., 1969). This is supported by recent computations by Sathiyamurthy and Patankar (1990).

Whereas streamwise injection is the more common mode of film cooling, lateral injection has been found to give a more uniform film-cooling effectiveness. The reason lies in the cross-

stream velocity patterns that are produced. The streamwise injection tends to lift the coolant jet away from the wall and, in turn, brings the mainstream hot fluid in contact with the wall; on the other hand, lateral injection produces a continuous blanket of cold fluid on the blade surface. This can be seen in Fig. 1, taken from Sathyamurthy and Patankar (1990), where the cross-stream velocity vectors and temperature distributions are shown for the two types of injection. Figure 2 shows the temperature distribution on the blade surface for different angles of lateral injection. The 90-degree lateral injection can be seen to produce a fairly uniform cooling effect over the entire surface.

The anisotropy of turbulence is recognized as a major consideration in the correct prediction

of film-cooling, especially for the case of streamwise injection. The use of an isotropic turbulence model does not give sufficient lateral spreading of the film-cooling jet. This is shown in Fig. 3 taken from Sathyamurthy and Patankar (1992), where the cooling effectiveness values are shown for the isotropic and anisotropic turbulence models. Figure 4 shows comparison of predictions with the measurements of Kadotani (1975). The isotropic model over predicts the effectiveness along the centerline of the hole and under predicts it along the line midway between two holes. A proper anisotropic correction to the model is able to give a satisfactory agreement with experimental data.

For handling complex geometries, Patankar and co-workers have developed

Figure 1. Secondary Flow Patterns for Streamwise and Lateral Injection (from Sathyamurthy and Patankar, 1990).

calculation methods employing general curvilinear coordinates (Karki and Patankar, 1988a, 1988b). These techniques can be used for an accurate solution of the flow in and around the injection holes of different geometries.

Recent experimental studies of film cooling have indicated that free-stream turbulence is an important parameter in determining film cooling behavior (Jumper et al. 1991). This important test was with in-line injection, however, not with compound angle injection. Mayle and Camarata (1975) compared a superposition model for effectiveness with compound-angle injection data and found disappointing results. Colladay and Russell (1976) performed a flow visualization study with compound-angle injection into a 2% free-stream turbulence intensity primary flow.

They noticed that compound angle injection generated a single, tightly-wound vortex filament which stays attached to the wall, even at higher blowing rates. Though the vortex helps keep the flow attached to the wall, it potentially augments the heat transfer rate. The effect of that vortex in a highly-disturbed primary flow remains undocumented. Kim et al. (1979) measured the flow field in a full-coverage film cooling study with compound-angle injection. Their measurements were limited to mean velocity profiles upstream of the holes. They noted an improvement over the first five rows of holes, followed downstream by a worsened film cooling effectiveness as a result of compound angle injection. Crawford et al. (1980) computed the Kim et al. flow, noting satisfactory performance.

Figure 2. Temperature Distribution on the Blade Surface for Different Angles of Lateral Injection (from Sathyamurthy and Patankar, 1990).

They did this with a STANCOOL version that was "tuned" to these results, however, and, thus, could not be considered a "prediction". The nearest study to the present study is one by Jubran and Brown (1985). In this study, profiles of streamwise turbulence were measured in addition to surface temperature profiles. The film cooling flow was injected at a compound angle and there was some variation of free-stream turbulence and streamwise acceleration. It was noticed that high free-stream disturbance reduced the film cooling effectiveness at low blowing rates, but increased it at high blowing rates. A case was run at 14% free-stream turbulence intensity and no acceleration and another was run with acceleration at $K = \nu/U^2 (dU/dx) = 1.1 \times 10^{-6}$ and a 3% free-stream turbulence intensity. This is a valuable case. It would be more useful, however, if the acceleration and the turbulence were

simultaneously representative of the engine environment and if more information were taken from the flow, such as the Reynolds stress tensor components. Jubran followed this study with a prediction of the results using the program PHOENICS and a $k-\epsilon$ model using an anisotropic correction to the viscosity. At low blowing rates, the model was successful, but it was not at higher rates. The author suggests that the inability to accurately compute the flow is due to the non-equilibrium turbulence situation of the real flow. Tafti and Yavuzkurt (1990) incorporated 3-D effects via an "entrainment fraction" model. The technique was successful at low blowing rates; less so at higher blowing rates. Honami, et al. (1992) probed a film cooling flow which was at 30° to the surface and 90° to the main flow. In addition to surface contours, they measured profiles of u' and t' (the instantaneous

Isotropic turbulence model

Anisotropic turbulence model

Figure 3. Cooling Effectiveness Distributions Predicted by the Isotropic and anisotropic Turbulence Models (from Sathyamurthy and Patankar, 1992).

value of temperature relative to the mean) and the correlation of the two. With these measurements, they were able to identify the effects of the vortex formed by the injection on the streamwise turbulent heat transport. Three-wire velocity measurements are proposed herein.

PROJECT DESCRIPTION

A coordinated experimental and computational program is underway which is designed to enhance the gas turbine industry's

methodology for designing optimum film cooling schemes. Measurements of film cooling effectiveness, surface heat transfer coefficients, and flow field mean and turbulence quantities are used for evaluating and developing computation models. One result will be a recommended computational scheme and the detailed data base which supports it. The program addresses modern cooling geometries with compound angle injection and shaped holes. Important effects to be included are free-stream turbulence, streamwise pressure gradient, and surface curvature. Accompanying the numerical model development will be perhaps

Figure 4. Comparison of Measured and Predicted Values of Film-Cooling Effectiveness (from Sathyamurthy and Patankar, 1992).

the most complete description available of compound angle film cooling flow physics. It will include detailed measurements of selected cases and a discussion of sensitivity to various parameters founded on evidence from computational and experimental perturbation runs.

RESULTS

Presently, the program has been underway for three months. Preparation of the test facilities, instrumentation, and computational programs is underway. For instance, a new style of turbulence generator, fashioned from the work of Ames and Moffat (1990) is being incorporated into an existing facility and three-wire, hot-wire anemometer techniques are being readied for measurement in the mixing zone downstream of film cooling injection. Also, film cooling test section design is underway.

FUTURE WORK

Flow Measurements

Complementary to surface measurements and in support of the turbulence model development and qualification of the numerical predictions, turbulence measurements will be taken in the flow approaching the film cooling region and at several stations downstream of the film cooling injection zone. An experimental flow is being established which has a compound angle injection of film cooling flow from one or two row(s) of holes which merges with a high-disturbance, primary flow field. Streamwise pressure gradients and surface curvature which match those of the film cooling zones of modern gas turbine engines will be imposed. This flow then simulates elements of the real gas turbine airfoil boundary layer and film coolant flows which are considered to be of major importance and which are not captured well by the design models.

The flow will be established in a recirculation-type wind tunnel. Prior to arrival to the film cooling zone, the primary air will pass through a turbulence generator similar to one of two presently used in similar studies in the Heat Transfer Laboratory. Both primary and secondary air streams will be at the same temperature; thus, the density ratio will be unity. Velocity

measurements will be taken with triple-wire, hot-wire anemometry. Such measurements will allow documenting the mean flow field and the six components of the Reynolds stress tensor. Turbulence kinetic energy needed for $k-\epsilon$ modeling will be documented. The individual components of the Reynolds stress tensor will also allow documentation of the anisotropy of the flow, a characteristic that cannot be completely captured by the $k-\epsilon$ model. This measure of anisotropy will indicate possible weaknesses in the model if it is found that poor predictions of the surface effectiveness, heat transfer coefficient, or other important computed values are found where the measured anisotropy of the turbulence is strong. Such an indication may lead to implementation of turbulence modeling that incorporates such anisotropy; examples would be the Reynolds stress model and the algebraic stress model. Frequently, the performance of a model is judged only by comparison to surface measurements. This indicates where the model fails but provides little support toward revision of the model.

Surface Measurements

A variation of the mass transfer analogy by which one can obtain effectiveness and heat transfer parameters for film cooling was described by Cho and Goldstein (1993). The surface of a blade is covered by a layer of naphthalene and its sublimation rate provides the required information. This method has a number of advantages; one experimental setup can produce effectiveness and heat transfer with great detail, the effect of variable properties can be studied, and mass transfer experiments exclude conduction in the solid wall. The experiments will utilize this technique. The flat plate with injection holes is covered with a thin layer of naphthalene and the local sublimation rate of the naphthalene is measured after the test plate has been exposed to the main and secondary airflows in the wind tunnel. In this way, the mass flow rate into the boundary layer is known for the given boundary conditions which, through the analogy, can be converted to the heat flow rate for the analogous thermal boundary conditions of the film cooling process.

To verify, the naphthalene sublimation results for film cooling effectiveness will be

checked against measurements from the foreign gas sampling method, which is presently used, on the same test apparatus. The naphthalene test plate with an injection hole will be replaced with a metal plate with an injection hole and sampling taps near the injection hole. Helium will be used as a tracer gas to produce mixture density ratios near unity.

Experiments will be on a flat plate and a curved wall (turbine blade) for compound angle injection. Although film cooling is now quite common on gas turbine components, few data are available in the archival literature, especially on heat transfer and flow behavior downstream of film cooling holes, with compound angle orientations and with curved surfaces. The results obtained will be compared with information on effectiveness and heat transfer available in the open literature. This gives also the possibility to investigate the influence of the boundary conditions, since constant heat flux was usually prescribed in previous measurements of effectiveness, whereas the present method is analogous to constant wall temperature conditions.

Several compound angle orientations of film cooling holes will be investigated in search of improvement in film cooling efficiency. A shaped-hole geometry will also be evaluated.

There has been no detailed study which deals with the aspects of hole-to-hole spacing, mainstream turbulence intensity, wall curvature, and an expanding hole exit area (shaped or diffusion hole). Sensitivity to these features are being investigated to determine their effects on the heat (mass) transfer coefficient and on the film cooling effectiveness. This is essential for determining the scaling parameters which govern film cooling.

Spanwise hole-to-hole spacing is important with regard to coalescence of flows from neighboring jet streams and to coolant coverage performance along the test surface. At a large spacing, there is no interaction of injectant fluid from adjacent holes until far downstream. At a small spacing, coalescence of neighboring jets often delays lift-off from the surface, especially when two rows of holes are employed.

The effect of an elevated mainstream turbulence intensity on heat transfer coefficient and film cooling effectiveness will be studied

experimentally under compound angle injection on a flat plate.

This compound angle injection study will be expanded to curved surfaces for simulating a gas turbine blade. The Heat Transfer Laboratory has investigated film cooling effectiveness using the foreign gas injection method through one/two rows of film cooling holes on convex/concave surfaces and turbine blades (Schwarz and Goldstein, 1988 and Ito, Goldstein and Eckert, 1978). Results show that the blade surface curvature influences the film cooling effectiveness particularly in the vicinity of the injection holes, with higher effectiveness on the convex surface, lower on the concave surface, as compared to a flat plate case. Near the endwalls on the suction surface of the blade profile, Goldstein and Chen (1985) showed that the film cooling jets are swept away from the surface by the passage vortex, resulting in low levels of film cooling effectiveness. On the pressure surface, the film cooling was unaffected by nearness to the endwall. Thus, the effect of blade wall curvature is very significant on the film cooling performance.

Finally, the effects of (diffusion) holes on film cooling performance will be assessed. The purpose of a shaped hole is to reduce the penetration of an injected jet into the mainstream. This can occur because the momentum at the hole exit is lower while a small metering hole section provides pressure drop. Using a shaped hole with axial injection, the film cooling effectiveness was considerably higher than with a straight hole, especially near the injection holes, as shown in Fig. 5 (Goldstein, Eckert and Burggraf, 1974). This is caused by the jet staying close to the wall, rather than penetrating into the mainstream. If the shaped hole is combined with the compound angle injection, high and uniform film cooling effectiveness, approaching a two-dimensional (slot) film cooling effectiveness, may be obtained on the whole area downstream of the injection jets. The reduced effective blowing rate at the hole exit will increase the effectiveness near the holes and will enhance coalescence of adjacent jets with compound angle injection which, in turn, will increase the effectiveness in the region downstream of the injection holes. Two types of shaped holes are proposed in the present study. One will be a simple conical enlargement, which

is easy to manufacture and is similar to the holes obtained with a laser cutting process. The second one, a rather irregular shape, will include a straight section at the downstream edge of the hole exit. This straight edge, perpendicular to the mainstream, will induce a Coanda effect so that the injected jets will spread out on the surface, rather than penetrate into the mainstream, even at high blowing rates. Ideally, the larger area ratio of the hole enlargement will produce lower effective blowing rate at the hole exit, resulting in better cooling performance.

These surface measurements will be tied closely to flowfield measurements and numerical analysis. The detailed results, in addition, will be used to validate the numerical scheme and the turbulence model.

Proposed operating ranges of parameters

Blowing rate	$1 < M < 4$
Mainstream turb. intensity	$0.3\% < \text{T.I.} < 15\%$
Inclined angles from surface	$\phi = 30^\circ$
Lateral angles	$\beta = 30^\circ, 45^\circ, \text{ and } 60^\circ$
Hole area ratio, shaped holes	A.R. @ 2.5

Computation

With the advances in computational fluid dynamics, it is now possible to calculate three-dimensional flow fields and the associated heat transfer. Turbulence models have been developed for the prediction of complex turbulent flows. In this research, computational analysis of film cooling will be performed. Film cooling will be considered on the blade surfaces, on turbine end walls, and on combustion chamber walls. Various configurations of film-cooling holes will be studied, including lateral injection and full-coverage cooling. The effect of streamwise acceleration, inlet turbulence, and wall curvature will be investigated.

The computational work are coordinated with the corresponding experimental work. Wherever possible, computational results are compared with experimental data to validate the numerical scheme and the turbulence model. Computations will be used to provide guidance for the experimental work to decide the flow parameters, measurement locations, and the quantities to be measured. Once the computational model is validated, it will be used to supplement

the experimental data and to perform parametric studies over a wide range of parameters. The results of the computational analysis will be presented in a generalized form so that they would be directly usable in industry. The governing parameters will be expressed in suitable dimensionless form and their influence will be documented in detail.

Extensive computational investigation will be made of the various aspects of film cooling. The three-dimensional flow will be calculated along with turbulence parameters and temperature distributions. The turbulence will be handled by a two-equation turbulence model, which provides the distribution of the turbulence kinetic energy and its rate of dissipation.

For the film-cooling situation, the geometrical parameters to be considered will include the hole spacing, number of rows, and the angle of injection. For each external flow situation, a number of blowing rates and inlet turbulence levels will be used. The film cooling for turbine end walls will require the calculation of a rather complex flow pattern. Wherever appropriate, the effect of curvature of the surface will be investigated. Full-coverage film cooling will be computed by increasing the number of rows of holes.

Figure 5. Centerline Film Cooling Effectiveness as a Function of Blowing Rate for Injection of Air Through Straight Round Holes and Shaped Holes, $S=3D$ (from Goldstein et al., 1974).

The computational activity will be conducted in close coordination with the experimental work. Extensive comparisons will be made between the numerical results and experimental data. This information will be used to refine the mathematical model and also to provide guidance for the experimental work.

The comprehensive prediction program for film cooling will establish the shortcomings as well as the validity of the mathematical models used. It is possible that factors such as the anisotropy of turbulence become important in some film-cooling scenarios. More advanced turbulence models will be investigated to accommodate these influences.

REFERENCES

- Ames and Moffat, 1990, *Heat Transfer with High Intensity, Large Scale Turbulence: The Flat Plate Turbulent Boundary Layer and the Cylindrical Stagnation Point*, HMT-44, Stanford University Dept. of Mech. Engr. Report.
- Chen, P.H. and Goldstein, R.J., 1992, "Convective Transport Phenomena on the Suction Surface of a Turbine Blade Including the Influence of Secondary Flows Near the Endwall," *J. of Turbomachinery*, vol. 114, pp 776-787.
- Cho, H.H. and Goldstein, R.J., 1993, "Heat (Mass) Transfer and Film Cooling Effectiveness with Injection through Discrete Holes - Part I: within Holes and on the Back Surface," *ASME Paper 93-WA/HT-58*.
- Cho, H.H. and Goldstein, R.J., 1993, "Heat (Mass) Transfer and Film Cooling Effectiveness with Injection through Discrete Holes - Part II: on the Exposed Surface," *ASME Paper 93-WA/HT-59*.
- Colladay, R.S. and Russel, L.M., 1976, "Streakline Flow Visualization of Discrete-hole Film Cooling with Normal, Slanted, and Compound Angle Injection," *NASA TN D-8248*.
- Crawford, M.E., Kays, W.M., and Moffat, R.J., 1980, "Full-Coverage Film Cooling, Part II: Heat Transfer Data and Numerical Simulation," *J. Engineering for Power*, Vol. 102, pp. 1006-1012.
- Goldstein, R.J. and Chen, H.P., 1985, "Film Cooling on a Gas Turbine Blade Near the Endwall," *J. of Eng. for Gas Turb. and Power*, vol. 107, pp 117-122.
- Goldstein, R.J., Eckert, E.R.G., and Burggraf, F., 1974, "Effects of Hole Geometry and Density on Three-Dimensional Film Cooling," *Int. J. H. M. Transfer*, vol. 17, pp 595-607.
- Goldstein, R.J., Eckert, E.R.G., Eriksen, V.L., and Ramsey, J.W., 1969, "Film Cooling Following Injection Through Inclined Circular Tubes," *NASA CR-72612*.
- Honami, J., Shizawa, T., and Uchiyama, A., 1992, "Behaviors of the Laterally Injected Jet in Film Cooling: Measurements of Surface Temperature and Velocity/Temperature Field Within the Jet," *ASME Paper 92-GT-180*.
- Ito, S., Goldstein, R.J., and Eckert, E.R.G., 1978, "Film Cooling of a Gas Turbine Blade," *J. of Eng. for Power*, vol. 100, pp 476-481.
- Jubran, B. and A. Brown, 1985, "Film Cooling from Two Rows of Holes Inclined in the Streamwise and Spanwise Directions," *J. Turbomachinery*, Vol. 107, pp. 84-91.
- Jumper, G.W., Elrod, W.C., and Rivir, R.B., 1991, "Film Cooling Effectiveness in High-turbulence Flow," *J. Turbomachinery*, Vol. 113, no. 3, pp. 479-483.
- Kadotami, K., 1975, *Effects of Mainstream Variables on Heated and Unheated Jets Issuing from a Row of Holes*, Ph.D. Thesis, University of Minnesota.
- Karki, K. C., and S. V. Patankar, 1988a, "Calculation Procedure for Viscous Incompressible Flows in Complex Geometries," *Numerical Heat Transfer*, Vol. 14, pp. 295-307.
- Karki, K. C., and S. V. Patankar, 1988b, "Solution of Some Two-Dimensional Incompressible Flow Problems Using a Curvilinear Coordinate System Based Calculation Procedure," *Numerical Heat Transfer*, Vol. 14, pp. 309-321.
- Kim, H.K., Moffat, R.J., and Kays, W.M., 1979, "Heat Transfer to a Full-Coverage, Film-

Cooled Surface with Compound-Angle Hole Injection," *NASA CR-3103*.

Mayle, R.E. and Camarata, R.J., 1975, "Multihole Cooling Film Effectiveness and Heat Transfer," *J. Heat Transfer*, Vol. 97, pp. 534-538.

Sathyamurthy, P. S., and S. V. Patankar, 1992, "Film Cooling Studies Using a Three-Dimensional Parabolic Procedure," Minnesota Supercomputer Institute Research Report UMSI 92/144.

Sathyamurthy, P. and Patankar, S.V., 1990, "Prediction of Film Cooling with Lateral Injection," *Heat Transfer in Turbulent Flows*, ASME-HTD, Vol. 138, pp. 61-70.

Schwarz, S.G. and Goldstein, R.J., 1988, "The Two-Dimensional Behavior of Film Cooling Jets on Concave Surfaces," *ASME Paper 88-GT-161*.

Tafti, D.K. and Yavuzkurt, S., 1990, "Prediction of Heat Transfer Characteristics for Discrete Hole Film Cooling for Turbine Blade Applications," *Journal of Turbomachinery*, Vol. 112, pp. 504-511.

Improving Aerodynamics of the Intercooler Flow Path For the Development of High Efficiency Gas Turbines

CONTRACT INFORMATION

Contract Number 94-01-SR029

Contractor University of Oklahoma
Office of Research Administration
1000 Asp Avenue, Suite 314
Norman, OK 73019-0430
Phone: (405) 325-4757
Fax: (405) 325-6029

Principal Investigators Dr. Ajay K. Agrawal
Dr. S. R. Gollahalli

DOE Project Manager Dr. Paul Micheli/Dr. Norm Holcombe

Period of Performance July 1, 1994 to June 30, 1996

Schedule and Milestone

FY 94-95 Program Schedule

	J	A	S	O	N	D	J	F	M	A	M	J
Scaling Criterion/Flow System												
Design and Analysis												
Fabrication of the Test Model												
Flow Experiments												

OBJECTIVES

Cooling of air between compressor stages (or intercooling) in advanced gas turbines must occur without adversely affecting the compressor aerodynamics, while incurring minimum pressure losses. The primary objective of this research is to provide experimental data and analytical tools to help in the design of an effective intercooler flow path. Such information is essential to gas turbine manufacturers for

achieving a higher system efficiency through improved aerodynamics of its components.

Using scaling analysis, the results of this study could be applied to both industrial and utility gas turbines. Although the primary thrust is on the intercooler system, the information obtained could be applicable to other similar systems such as a recuperator and/or an aftercooler.

BACKGROUND INFORMATION

Intercooling entails cooling of air between the low pressure (LP) and high pressure (HP) compressor sections of the gas turbine. Cooler air entering the HP compressor decreases the overall compression work. Intercooling also lowers temperature at the HP compressor discharge, thus allowing for more effective use of air as a coolant in the hot gas flow path.

Recent studies on cycle analysis by gas turbine manufacturers have shown that intercooling is critical to achieving high efficiency gas turbines for power generation [1-4]. A Westinghouse team led by Bannister [1] analyzed several combined cycle plants to accomplish Advanced Turbine Systems (ATS) goals for utility scale gas turbines. Intercooling was common to all advanced cycles studied by Bannister et al. [1]. A system study by the General Electric Company [2] also identified intercooling as a necessary cycle modification to accomplish the ATS goals. Similar conclusions were drawn by Solar Turbine [3] and Allison Gas Turbine [4] in their feasibility studies of industrial gas turbine cycles.

Overall, there appears to be a general consensus on adopting intercooling in advanced gas turbines of different capacities. This conclusion is reinforced by the US Navy's intercooled-recuperative (ICR) gas turbine engine program for propulsion of ships [5-8] and by the Collaborative Advanced Gas Turbine (CAGT) program sponsored by a private consortium with a focus on developing advanced aeroderivative gas turbines [9].

In an intercooler system, the air exiting the LP compressor must be decelerated to provide the necessary residence time in the heat exchanger. The cooler air must subsequently be accelerated towards the inlet of the HP compressor. The heat exchanger could cause circumferential nonuniformities in the flow, which must be isolated from the compressors because of the adverse impact on the system

performance and efficiency.

Pressure losses in the intercooler flow path would contribute directly to power losses which must be minimized in advanced gas turbines. Depending upon a particular configuration, these losses may reach 2-6% of the compressor discharge pressure. A one-percent loss in the total pressure amounts to a loss of approximately one-third percentage point in the overall thermal efficiency. Although pressure losses can not be eliminated completely, an aerodynamic design of the intercooler flow path could have a significant impact on the overall gas turbine system efficiency. The importance of the intercooler pressure loss was also recognized in a sensitivity analysis of the Navy's ICR engine by Crisalli and Parker [6] who found that the intercooler pressure drop had a strong influence on specific fuel consumption (or thermal efficiency) while the intercooler heat transfer effectiveness had a weak influence.

Thus, the implementation of intercooling requires modifications to the gas turbine engine including (i) a diffuser for decelerating airflow at the LP compressor discharge, (ii) a heat exchanger for cooling this air, and (iii) a nozzle for accelerating the cooler air to the inlet of the HP compressor.

PROJECT DESCRIPTION

This project involves a study of flow through nonrotating parts between the LP and HP compressors of intercooled gas turbines. The research will provide quantitative data and analytical tools to help in the design of an effective intercooler flow path. The following is a description of the experimental and theoretical components of the project.

Experimental

Flow experiments will be conducted with

scale models of the intercooler flow path. This approach is a natural choice because it leads to a simpler test-section while satisfying the prototype requirement that the intercooler remain isolated from the compressors. The scale model will correctly simulate flow in the prototype if the velocity profile at its inlet match those at the LP compressor discharge. This matching will be achieved by inducing controlled air flow through the test model by a suction type wind tunnel. The following are the details of the test model, wind tunnel, instrumentation, and data acquisition system.

Test Model. Figure 1 shows a gas turbine engine configuration for positioning an intercooler between the compressors. The intercooler flow path includes a discharge diffuser, a heat exchanger, and an inlet nozzle. The discharge diffuser isolates the LP compressor from the intercooler by directing airflow away from the turbine axis. This diffuser also recovers the kinetic energy while decelerating the compressor air, thereby minimizing the flow discharge losses. Similarly, the inlet nozzle isolates the HP compressor from the intercooler while minimizing the inlet flow losses by gradually accelerating the cooler air.

Figure 1. Schematic of the Intercooler Flow Path

The intercooler could be located on-axis (as shown in Figure 1) or off-axis of the gas turbine. An on-axis intercooler could be nearly axisymmetric and connect directly to the discharge diffuser and the inlet nozzle. However, such an intercooler is difficult to position and repair, and it limits access to the gas turbine. Alternatively, an off-axis intercooler is easy to install, but it requires interconnecting flow passages which are asymmetric.

Both on-axis and off-axis configurations will be considered in this study. The on-axis arrangement will be used to establish baseline information such as the diffuser and nozzle geometries. Because the emphasis of this study is on flow nonuniformities and pressure losses in the intercooler flow path, the experiments will be conducted at cold flow conditions. Thus, the intercooler will be simulated by honeycombs and screens to induce the desired levels of flow resistance and circumferential nonuniformities.

The test model is modular, thus, allowing easy assembly and modification. The overall length of the test model is expected to be about 0.9 m. The test model will be instrumented with wall pressure taps, and it will have access ports for different probes and transparent windows for optical flow visualization.

Wind Tunnel. An open-circuit wind tunnel with a suction fan was modified to provide the desired airflow through the test model. A schematic of this low turbulence wind tunnel and a photograph of the facility are shown, respectively, in Figures 2 and 3. The ambient air entering the wind tunnel passes through a honeycomb, a set of screens, and a 9:1 area ratio contraction section which guides the airflow to a 1.98 m long annular flow developing section upstream of the test model. The inner pipe (0.352 m outer diameter) of this section is machined

Figure 2. Schematic of the Wind Tunnel

Figure 3. A View of the Experimental Facility

from aluminum. A propeller nose cone is attached to the upstream end of the inner pipe for gradually transitioning airflow into the annulus. The outer pipe (0.454 m inside diameter) consists of a fiberglass section followed by a 0.76 m long machined plexiglas section. The inner pipe is positioned inside the outer pipe by sets of 3 airfoil shaped struts at two axial locations. These struts transfer weight of the aluminum pipe to external supports, and also allow precise adjustment of spacing between the two pipes.

The contraction and flow developing sections are mounted on a support system which traverses axially on a rail to ensure accurate alignment during assembly or disassembly. The desired velocity profile at the test section inlet can be obtained by strategically placing rings of varying width and height in the flow developing section.

A 0.46 m long annular section simulating the inlet of the HP compressor is located downstream of the test model. Flow nonuniformities caused by intercooler can be observed in this section which is made of a plexiglas outer pipe and an aluminum inner pipe. The spacing between the two pipes can be adjusted in a manner similar to that in the upstream flow developing section. A propeller nose cone is attached to the downstream end of the inner pipe for gradually transitioning airflow from the annulus to a conical diffuser.

The air from the conical diffuser flows through the inlet guide vanes, and a suction fan followed by a vertical diffuser which discharges the air to outside of the laboratory. The inlet guide vanes to the suction fan are connected to a variable speed motor for regulating airflow through the wind tunnel. The motor can be integrated to the data acquisition and control system, thereby allowing airflow rate regulation by a microcomputer. The suction fan is run at a constant RPM by a 30kW, 3-phase electric motor.

Instrumentation and Data Acquisition.

The point measurements of static pressure, total pressure, and velocity are necessary throughout the test model to characterize the flow field and to determine frictional losses. The primary measurements are obtained by wall pressure taps, pitot-static probes, and Kiel probes. A two-channel, constant temperature hot-wire anemometer (TSI Model IFA-100/300) will be used to measure velocities in regions where the pitot-static probes can not be aligned with the flow streamlines. Thermocouples are used to monitor temperature in the flow system.

Stepper-motor-driven traverse systems have been designed to mount and move various probes. Because of the compact size, several of these traverse systems can be attached directly to the test model or the wind tunnel. Thus, the pressure and velocity profiles at several circumferential and/or axial locations can be obtained simultaneously.

A computer controlled scanning system (Scanivalve Model WSS-5/24) is used to scan 4 pressure input ports at a time from a total of up to 96 ports. Four pressure transducers with a range of ± 0.70 m of water head (or 1 psig) are used to measure pressure outputs from the scanning system. The pressure sensors are calibrated against a U-tube manometer. The data acquisition and control operations are performed on a microcomputer with a 60 MHz pentium processor, and hardware and software from the Strawberry Tree Inc. An underlying feature of the instrumentation is that different operations such as controlling the airflow rate, probe traversing, scanning pressure channels, measuring pressure, temperature, and velocities, and storing data on hard disk are all integrated and automated within the same interactive software environment.

Theoretical

The theoretical part of the project is to first

develop configurations representing important and essential features of the intercooler flow path. Subsequently, the analysis could be used to interpret the experimental data and to develop design guidelines.

The major components of the intercooler flow path are (i) a discharge diffuser, (ii) a heat exchanger, and (iii) an intake nozzle. Because of the complex geometry and flow behavior, the design of these components is not amenable to simple analytical procedures. The existing experimental data base on annular diffusers and nozzles, especially those with curved surfaces, is also extremely limited. In this project, computational fluid dynamics (CFD) is used for detailed modeling and analysis to overcome the current drawbacks.

Wall geometries of the diffuser and the nozzle are obtained using the analysis by Agrawal and Yang [10]. In this procedure, fully elliptic Navier-Stokes equations for turbulent flow are solved on a body-fitted coordinate system. The computations are done repeatedly until the geometry providing the desired flow behavior is achieved through trial and error. Additional computations will be done on the resulting configurations to determine the effects of upstream distortions in the velocity profiles. Both 2- and 3-dimensional calculations are involved because the circumferential distortions may result in a 3-D flow field even when the geometry is axisymmetric. Computations are done on a Sun Sparkstation Model 10/40 using a modified version of the general purpose CFD software PHOENICS.

RESULTS

Scaling Criterion/Flow System

A scaling analysis was performed prior to the design of the experiments. The scaling factor was determined from the annular passage width at the inlet of the test model which was set at

5.08 cm to allow reasonable measurement accuracy. The dimensions of the test model correspond to about one-eighth scale for a utility size turbine and to one-fourth scale for an industrial size turbine. Unlike the prototype, the test model uses air at the atmospheric pressure and temperature. The average velocity in the test model is maintained the same as that in the prototype. The model Reynolds number based on the annular passage width is 220,000 which is significantly lower than that in the prototype. Still the model Reynolds number is sufficiently large that its effects on flow behavior are deemed minimal. The influence of Reynolds number on flow behavior, if any, would be quantified from experiments at different model Reynolds numbers. Table 1 summarizes the primary technical data for a utility gas turbine, an industrial gas turbine, and the cold flow test model.

Preliminary measurements indicate that the airflow rate in the wind tunnel exceeds 15 m³/s (31,800 scfm) when the test model and the annular flow sections are not connected. With the test model in place, the wind tunnel is expected to provide flow rates up to 3.5 m³/s (7,400 scfm) at a pressure loss of 0.65 m (or 25.5") of water head.

Modeling and Analysis

Computations were done for straight wall annular diffusers and nozzles with area ratios of 6 and 10 for a length to annular width ratio of 6. The flow turbulence was modeled by a standard two-equation model. The axial velocity profiles at the inlet of the diffuser was specified as fully developed and that at the inlet of the nozzle was specified as uniform. A constant reference pressure was specified at the exits of the diffuser and the nozzle. Computations were done with 20 grids in the radial direction and 60 grids in the axial direction.

Table 1. Primary Technical Data

Parameter	Utility Scale	Indus. Scale	Test Model
Scale	1.0	0.5	0.125
Pressure, bar	3.0	3.0	1.0
Temperature, K	410	410	300
Density, kg/m ³	1.8	1.8	1.2
Axial Velocity, m/s	80	80	80
Reynolds Number	1.4e6	0.7e6	0.2e6
Mass Flow Rate, kg/s	400	100	4.2
Mach Number	0.2	0.2	0.23

Figures 4 and 5, respectively, show velocity vectors in the diffuser and the nozzle for an area ratio of 6. These results indicate that the flow remains attached throughout the passages. Computations for an area ratio of 10 show flow separation in both the diffuser and the nozzle. A larger area ratio diffuser is desirable to lower the air velocity in the heat exchanger, and hence, to improve the heat transfer effectiveness. These preliminary results indicate the need to optimize wall geometries of the diffuser and the nozzle for avoiding flow separation at high area ratios.

FUTURE WORK

The designs of the diffuser and nozzle sections will be refined using modeling and analysis. Representative on-axis and off-axis test models will be designed and fabricated. The flow experiments will be conducted for a range of Reynolds numbers. Finally, the data will be analyzed in conjunction with the CFD analysis to

develop guidelines for designing the intercooler flow path.

REFERENCES

1. Bannister, R.L. , Little, D.A. , and Wiant, B.C. , 1993, "Development of Advanced Gas Turbine Systems," *Proc. of the Joint Contractors Meeting*, D.W. Geiling (ed.), DOE/METC-93/6132, pp. 3-15.
2. Cook, C.S. and Nourse, J.G. , 1993, "GE Power Generation Technology for Advanced Gas Turbines," *Proc. of the Joint Contractors Meeting*, D.W. Geiling (ed.), DOE/METC-93/6132, pp. 16-25.
3. LeCren, R.T. and White, D.J. , 1993, "Gas Fired Advanced Turbine System," *Proc. of the Joint Contractors Meeting*, D.W. Geiling, (ed.), DOE/METC-93/6132, pp. 26-32.
4. Wilkes, C. , Mukavetz, D.W. , Knickerbocker, T.K. , and Ali, S.A. , 1993, "Advanced Turbine Systems Program," *Proc. of the Joint Contractors Meeting*, D.W. Geiling (ed.), DOE/METC-93/6132, pp. 33-41.
5. Jeffs, E. , 1992, "U.S. Navy Chose ICR Gas Turbine for New Propulsion System," *Turbomachinery International*, vol. 33, no. 7, pp. 45-48.
6. Crisalli, A.J. , and Parker, M.L. , 1993, "Overview of the WR-21 Intercooled Recuperated Gas Turbine Engine System: A Modern Engine for a Modern Fleet," *ASME Paper 93-GT-231*.
7. Shepard, S.B. , Bowen, T.L. , and Chiprich, J.H. , 1994, "Design and Development of the WR-21 Intercooled Recuperated (ICR) Marine Gas Turbine," *ASME Paper 94-GT-79*.
8. Hawkins, W.J. , Mathieson, D. , Bruce, C.J. , and Socoloski, P. , 1994, "System Development Test Program for the WR-21 Intercooled Recuperated (ICR) Gas

Figure 4. Velocity Vectors in the Diffuser with an Area Ratio of 6

Figure 5. Velocity Vectors in the Nozzle with an Area Ratio of 6

Turbine Engine System," *ASME Paper 94-GT-186*.

9. Cohn, A. , 1993, "Collaborative Advanced Gas Turbine Program," *Proc. of the Joint Contractors Meeting*, D.W. Geiling (ed.), DOE/METC-93/6132, pp. 103-108.
- 10 Agrawal, A.K. and Yang, T.T. , 1991, "Viscous Flow Computations for Compressor/Combustor Diffuser Design to allow Air Extraction for IGCC Systems," *ASME Paper 91- GT-62*.

Examining the Role of Turbulence on Internal Heat Transfer

CONTRACT INFORMATION

Contract Number	94-01-SR025
Contractor	University of Wyoming Department of Mechanical Engineering P.O. Box 3295 Laramie, WY 82071-3295
Principle Investigator	Paul A. Dellenback
Co-Principle Investigator	William R. Lindberg
Research Assistant	Roger W. Radomsky
DOE Project Manager	Paul Michele / Norm Holcombe
Period of Performance	June 1, 1994 to May 31, 1995
Schedule and Milestones	

FY94-95 Program Schedule

J J A S O N D J F M A M

Testing; including LDA measurements in simulated turbine blade cooling passages and model combustors

X - - - - - X

Build data base through literature search

X - - - - - X

Examine physical basis for developing correlations having 'Tu dependence

X-----X

Final reports and interaction with industrial partners

X - -X

OBJECTIVES

The relationship between convective heat transfer and highly turbulent internal flows is extremely important to several gas turbine engine applications, including flows in combustors and internal cooling passages in turbine blades and vanes. In spite of many recent investigations of internal turbine blade cooling passages, there continues to be an incomplete understanding of the problem because the role of turbulent fluctuations is not directly accounted for.

The present investigation is examining the hypothesis that there may be a geometry-independent relationship between convective heat transfer coefficients and turbulence levels in highly turbulent internal flows. Such relationships could have at least three important ramifications on design of flow passages in gas turbine engines:

- (1) Heat transfer correlations could be developed where none currently exist due to an unmanageable number of geometrical parameters.
- (2) Existing empirical correlations could be greatly simplified if they explicitly included the effects of turbulent fluctuations because the influence of geometric parameters could be largely or totally eliminated.
- (3) Existing internal flow correlations could be rendered more accurate as the important influence of turbulence is accounted for.

Computational fluid dynamics can, or soon will be capable of predicting turbulence levels in ribbed passage flows. The relationships sought in the present study will help to complete the design problem by allowing a direct computation of heat transfer coefficients, as opposed to requiring solution of the turbulent energy equation with speculative approximations for the turbulent Prandtl number. Insofar as numerous flow passages are designed specifically to increase convective heat transfer by increasing turbulence

levels within the passages, a methodical examination of the role of turbulence on convective heat transfer for internal flows appears long overdue.

BACKGROUND INFORMATION

Turbulent heat transfer solutions or correlations for internal flows typically show the Nusselt number to be some function of Reynolds number, Prandtl number and flow geometry. Fluctuating velocity components have traditionally not been included in empirical correlations in an explicit way, in spite of the fact that they are clearly an important parameter in turbulent flows, and in some cases the dominant influence on convective heat transfer¹⁻³. In many problems the flow geometry can be so complex as to be unmanageable, as for example in the ribbed cooling passages of turbine blades where at least eight separate geometrical factors influence the magnitude of local convective heat transfer coefficients. Since the purpose of ribs in turbine blade passages (and other ducts) is to generate turbulence, then the focus of attention should be on the turbulence levels themselves and not on the myriad of geometric variables that interact in complex ways to generate the turbulence.

Numerous investigations in the last two decades have examined the impact of free stream turbulence on convective heat transfer in boundary layer flows. Recent papers by Maciejewski and Moffat¹⁻³ contain several important contributions to an understanding of the interrelationship between turbulence and convective heat transfer for external flows. One contribution of their work was a new, simpler, and perhaps quite general correlation for the heat transfer coefficient in terms of an RMS velocity fluctuation that characterizes the local flow field. An additional conclusion of their study was that turbulence length scales were not important in characterizing the heat transfer in the presence of high free stream turbulence.

Finally, Maciejewski and Moffat concluded that the classical Reynolds analogy does not hold, and suggest that momentum and heat are transported by different mechanisms, at least for the highly turbulent flows upon which their investigation focused attention.

The Maciejewski and Moffat correlation fit their own data (consisting of boundary layer flow over a flat constant temperature surface), as well as the data from experiments by MacArthur⁴ (heat transfer associated with a circular wall jet passing over a flat surface) and Vogel⁵ (flow and heat transfer over a backward facing step). Because the data from the three diverse experiments were well correlated by a single expression, Maciejewski and Moffat speculated that a universal correlation with no dependence on flow geometry could be developed to include all highly turbulent external flows.

Finally, it is noteworthy that the Maciejewski and Moffat correlation extends to relatively low levels of turbulence intensity (Tu) on the order of 15%, so that the correlation is applicable over a wide range of turbulence levels. In fact, their results suggest that Tu is the dominant variable upon which the heat transfer coefficient depends for Tu greater than 15%, which is perhaps a surprisingly low threshold.

PROJECT DESCRIPTION

The internal cooling passages typical of high-temperature turbine blades and vanes are often constructed with rib turbulators that are intended to increase turbulence and thus heat transfer coefficients within the passages. At present, design of these passages is usually governed by a vast geometry-dependent array of experimental data. In virtually all the past experimental research (that ultimately guides passage design) no effort to characterize turbulence intensities has been made, in spite of the fact that the principal purpose of the ribs is to

enhance heat transfer by generating high turbulence levels. Useful correlations derived from these experiments are rare because no one series of experiments can possibly address the numerous geometrical variables in the problem, and because when the effects due to turbulent fluctuations are not considered, the experiments fail to capture the essential physics. In fact, any correlations that might be developed without regard to incorporating the effects of turbulence level are likely to fail, just as boundary layer correlations failed for many years when the effect of high turbulence levels was ignored. However, it may be that the heat transfer coefficients for these, and other, highly turbulent internal flows could be easily and accurately predicted in a simple way based on a knowledge, calculation, estimate, or measure of turbulence characteristics associated with the flow field.

The present authors have experience with various confined swirling flows⁶⁻¹⁰, including axial swirling flow through a sudden expansion, coaxial jet mixing with inner jet swirled, and counterswirled coaxial jet mixing. These problems have represented a progression of ever-more sophisticated dump combustor flow field models. In the two coaxial jets experiments, the focus of the investigation was on the convective heat transfer aspects, and beyond documentation of the inlet flow, no attempt was made to undertake tedious LDA measurements throughout the mixing region. At the time, measurements in the mixing region seemed to be of little value since several other investigations¹¹⁻¹⁴ have reported exhaustive velocity data for similar flows, so that there appeared to be little to learn about mean flow behavior.

The present research will provide critical measurements which will support development of relationships for highly turbulent flow environments that are directly relevant to gas turbine engine design. Detailed velocity measurements will be made in two test rigs for which the authors already have extensive local

heat transfer data, including; (1) flows in ribbed passages of rectangular cross section that model internal turbine blade cooling passages, and (2) counterswirled coaxial jet mixing of two confined jets that model flows in combustors. The velocity measurements will be made with a two-component laser Doppler anemometer. The flow loops, test sections, and optical instrumentation are all in place, tested, and available for immediate use. To compliment our internally generated data, we will continue to search the technical literature for supporting velocity and heat transfer data.

The data gathered will then be incorporated into relationships that include the influence of turbulence as well as mean flow characteristics like Prandtl, Reynolds, and swirl numbers.

RESULTS

At the present time, the authors have turbulence and heat transfer data for three different internal flow geometries relevant to the present investigation;

- (1) Data for swirled and unswirled water flow through a sudden axisymmetric expansion were acquired in a prior investigation⁶⁻⁷. The data were measured for three different Reynolds numbers and a wide variety of swirl numbers. For any one Reynolds and swirl number, there are several data points corresponding to various axial positions that include wall-bounded recirculation regions and reattached, developing flow in the tube.
- (2) Water data for mixing of confined coaxial jets, with the inner jet swirled, were also accumulated in a prior investigation⁸⁻¹⁰. Data are available at several different axial locations for each of three different values of annular flow rate.
- (3) Data for air flows through rectangular ducts with various turbulator rib configurations

have been acquired as a part of the present investigation, again at various axial locations between the ribs.

The influence of turbulence on the convective heat transfer was never fully resolved in the two swirling flow experiments described above, so that when they appeared, the Maciejewski and Moffat papers were a welcome starting point into the possible mechanisms at work in highly turbulent heat transfer. Maciejewski and Moffat developed an argument for, and subsequently employed, a modified Stanton number (St'), where the usual mean velocity was replaced by a streamwise fluctuating velocity (u'). The modified Stanton number was found to correlate well with the largest turbulence intensity measured in the wall affected region, but there appears to be some ambiguity about how the mean velocity should be characterized for various flows. Nevertheless, data from three different external flow experiments examined by Maciejewski and Moffat were well fit by their correlation to within $\pm 15\%$.

Thus, the Maciejewski and Moffat correlation would seem a reasonable starting place for examining the internal flow water data generated by the present authors. Figure 1 shows both water and air data from the three experiments listed above plotted against the Maciejewski and Moffat correlation. The correlation does not fit the present data particularly well, suggesting that the physical arguments presented by Maciejewski and Moffat require some modification for the present data, and perhaps for internal flows in general.

One very important observation made by Maciejewski and Moffat was that the heat transfer coefficient depended linearly on the maximum streamwise fluctuating velocity (u'_{max}) in the near wall region. The present authors began by examining the relationship between these two parameters in their internal flow data, and found the relationship to be sporadic and far from linear. It should be noted that all velocity measurements

Figure 1. A composite of the author's sudden expansion, coaxial jet, and ribbed channel data plotted using the Maciejewski and Moffat Correlation

Figure 2. The relationship between the heat transfer coefficient and u'_{avg} for the authors' sudden expansion data

made by the present authors were with a laser Doppler anemometer (LDA) that, characteristic of the instrument, had limited capability for detailed measurements very near the wall. To this point it has been assumed that not only is near-wall turbulence important to the heat transfer problem, but so is the level of turbulent kinetic energy throughout the duct important to determining the heat transfer coefficients. Thus, in the present context of internal flows, u'_{max} has been taken to be the largest streamwise fluctuation in the cross sectional profile of the duct flow. The importance of u'_{max} in correlating the data is not entirely intuitive to the present authors so that u'_{avg} or u'_{min} were viewed as possible alternative scales.

In fact, for the existing water data a linear relationship does exist between the heat transfer coefficient and u'_{avg} . This linear dependence is shown in Figure 2, where a single linear curve is drawn through each sudden expansion data set. We see that the linear fits for the data in each case have Reynolds number-dependent slopes, and that for any one Reynolds number (e.g., $Re=100,000$), there is some apparent dependence on swirl number. We have yet to determine how much of the heat transfer enhancement due to swirl is

attributable to increased mean velocity and how much is due to the higher turbulence levels that result from the increased shearing associated with larger mean velocities. The difficulty lies in knowing how other fluctuating velocity components interact with u' in the heat transfer problem, and this is a primary reason to expand the data base with new swirling flow data. It should be noted that the heat transfer coefficients in the downstream tube were larger than could be satisfactorily explained by the combination of heat transfer enhancement due to; (1) separated flow, and (2) swirl of the flow. Thus, while Figure 2 offers encouragement, it also shows that u'_{avg} is not by itself a completely satisfactory parameter for correlating the data.

Figure 3 is similar to Figure 2 in showing that there is some nominal correlation between the heat transfer coefficient and the average streamwise fluctuation for a variety of ribbed passage flow geometries. Once again, there is significant shearing associated with the shear layers downstream of the turbulators so that u'_{avg} will apparently be insufficient by itself for correlating the data. Nonetheless, Figures 2 and 3 must be viewed as encouraging to the overall goals

Figure 3. The relationship between the heat transfer coefficient and u'_{avg} for the authors' ribbed passage data

of the present project in light of the relatively simple relationships that they exhibit between the heat transfer coefficient and streamwise fluctuating velocity.

It should be emphasized that this program is not a "brute-force" scheme for determining how to correlate the data base, but rather the authors view a rational, physics-oriented approach to data reduction as essential to any substantial contribution. Results of the investigation should be an extremely useful tool to the community of turbine engine designers concerned with heat transfer problems. For example, the optimal design of turbine blade cooling passages will no longer be one of sorting through numerous geometrical parameters. Rather, it will be apparent how the heat transfer depends on mean flow quantities and turbulence levels. Design choices can then be made through a rational decision process and augmenting turbulence levels can be treated as a separate issue. These same conclusions pertain to computations of combustor flows. In addition, the tradeoffs between optimizing mixing of air/fuel mixtures and minimizing heat transfer to liner walls will be more

easily resolved with the quantitative tools that will be generated by this program.

REFERENCES

- 1) Maciejewski, P.K. and R.J. Moffat, 1989, "Effects of Very High Turbulence on Heat Transfer," Seventh Symposium on Turbulent Shear Flows, Stanford University, pp. 20.3.1-20.3.5.
- 2) Maciejewski, P.K. and R.J. Moffat, 1989, "Heat Transfer with Very High Free Stream Turbulence", NASA NAG 3-522 Report no. HMT-42.
- 3) Maciejewski, P.K. and R.J. Moffat, 1992, "Heat Transfer with Very High Free Stream Turbulence: Part I - Experimental Data," and "Part II - Analysis of Results," *J. of Heat Transfer*, **114**, pp. 827-833 and 834-839.
- 4) MacArthur, C.D., 1986, "Fluid Dynamics and Heat Transfer of the Circular Tangential Wall Jet," Ph.D. Theses, University of Dayton, Dayton, OH.
- 5) Vogel, J.C., 1984, "Heat Transfer and Fluid Mechanics Measurements in the Turbulent Reattaching Flow Behind a Backward-Facing Step", Ph.D. Thesis, Stanford University, Stanford, CA.
- 6) Dellenback, P.A., D.E. Metzger, and G.P. Neitzel, "Heat Transfer to turbulent Swirling Flow Through a Sudden Axisymmetric Expansion," *ASME J. of Heat Transfer*, **109**, 1987, pp. 613-620.
- 7) Dellenback, P.A., D.E. Metzger, and G.P. Neitzel, "Measurements in Turbulent Swirling Flow Through an Abrupt Axisymmetric Expansion," *AIAA J.*, **26**, 1987, pp. 669-677.
- 8) Sanger, J.L. and P.A. Dellenback, 1994, "Heat Transfer in Counterswirled Coaxial Jet Mixing," submitted to *ASME J. of Turbomachinery*.

- 9) Dellenback, P.A., J.L. Sanger and D.E. Metzger, 1994, "Heat Transfer in Confined Coaxial Jet Mixing with Swirled Inner Jet," accepted for publication in *J. Heat Transfer*.
- 10) Memar, H., J.P. Holman, and P.A. Dellenback, 1993, "The Effect of a Swirled Annular Jet on Convective Heat Transfer in Confined Coaxial Jet Mixing," *Int. J. Heat Mass Transfer*, 36, no. 16, pp. 3921-3930.
- 11) Gouldin, F.C., J.S. Depsky, and S.L. Lee, 1985, "Velocity Field Characteristics of a Swirling Flow Combustor", *AIAA J.*, 23, pp. 95-102.
- 12) Vu, B.T. and F.C. Gouldin, 1982, "Flow Measurements in a Model Swirl Combustor", *AIAA J.*, 20, pp.642-651.
- 13) Ramos, J.I. and H.T. Somer, 1985, "Swirling Flow in a Research Combustor", *AIAA J.*, 23, pp. 241-248.
- 14) Khodadadi, J.M. and N.S. Vlachos, 1989, "Experimental and Numerical Study of Confined Coaxial Turbulent Jets", *AIAA J.*, 27, pp. 532-541.

Measurements of Surface Temperature and Heat Transfer Using a UV-Induced Phosphor Fluorescence Thermal Imaging System

CONTRACT INFORMATION

Contract Number DE-FC21-92MC29061

Contractor Carnegie Mellon University
Department of Mechanical Engineering
Schenley Park
Pittsburgh, PA 15213 - 3890
Phone: (412) 268-3658
Fax: (412) 268-3348

Principal Investigator Dr. Mingking K. Chyu

Co-PI Dr. Shi-Chune Yao

DOE Project Manager Dr. Paul Micheli / Dr. Norm Holcombe

Period of Performance September 1, 1994 to October 31, 1994

FY 94 Program Schedule

	S	O	N	D	J	F	M	A	M	J	J	A
Phosphor Evaluation												
TGP Imaging System Design, Fabrication & Calibration												
Cascade Fabrication												
Film Cooling Test												

OBJECTIVES

This newly initiated research program for Advanced Turbine System has two major objectives. The first objective targets the development of a novel, non-intrusive imaging system that will immensely advance the current thermal measurement technology for both engine diagnostics and turbine cooling research. The second objective is to apply the measurement technique developed to systematically investigate some of the well recognized, though unresolved, issues in film cooling.

BACKGROUND INFORMATION

To achieve the efficiency goals, an advanced turbine system will need to operate at sustained turbine firing temperatures of more than 1400 C. High temperatures and transient temperature response of engine components can produce significant thermal deterioration. The life of highly stressed components decreases significantly with increasing metal temperatures. There are many complex and interrelated mechanisms affect the heat transfer from the hot gas to the turbine blades. They include secondary flows, unsteady wakes, boundary layer transition, freestream turbulence, surface curvature, flow separation, shock, injectant coolant flow, etc. Knowledge of the detailed heat transfer distribution is required to design turbines for thousands of hours for continuous operation without failure. In order to predict the service life of the engine, it is essential to have both local temperature levels of the hot gas and local temperature gradients from the hot gas to the components.

The conventional method for temperature measurement in heat transfer research relies on the use of electrically-resistive sensors such as thermocouples. While such

devices occur good accuracy, their usage for surface temperature measurement does present certain difficulties. Owing to the discrete nature of temperature measurement, thermocouples may be ill suited for the study of local convective heat transfer in complex flow fields such as those that occur in an axially cooled rotor-stator disk cavity. In such experiments it may not be feasible to instrument the surface of the rotor or stator with the large number of thermocouples necessary to provide an accurate mapping of local variations in heat transfer. Finally, installation of thermocouples on rotating surfaces such as a turbine disk present severe technical difficulties. Great care must be exercised in thermocouple installation to ensure that the thermocouples or their leads do not affect the flow, and thus the heat transfer, within the disk cavity. An additional concern in turbine instrumentation is the need to use slip ring assemblies, which permit intermittent contact between rotating thermocouple leads and stationary data acquisition system leads. The movement of these contacts introduces noise in the output signal. More importantly, however, such assemblies are not suitable for use at the very high rotation rates associated with modern turbomachinery. Consequently, it is not possible to obtain temperature data at normal operating conditions. Rather, nominal heat transfer characteristics must be inferred from lower speed tests.

With recent advances in digital video technology, non-intrusive optical temperature measurement techniques such as infrared imaging or liquid crystal thermography have become attractive alternatives to thermocouple instrumentation. Each of these techniques, however, present certain limitations that prevent their general use. While infrared imaging is a viable technique for use in high temperature environments, room temperature measurement is difficult due to the inability to accurately screen

background radiation. The chief disadvantage of liquid crystal thermography is the limited temperature measurement range, which is from -25 C to 250 C. Furthermore, because this technique relies on the measurement of reflected light, results are sensitive to both illumination intensity and viewing angle.

The primary effort for the present study is to develop a thermal measurement system that overcomes the limitations aforementioned. The technique proposed is a thermal imaging system based on the temperature-sensitive properties of rare-earth ion doped phosphors (i.e., thermographic phosphors or TGP). With the choice of an appropriate thermographic phosphor, this technique can be used over a broad temperature range, from cryogenic temperatures to the high temperatures associated with combustion.

PROJECT DESCRIPTION

This paper describes a TGP thermal imaging system presently available at Carnegie Mellon University. Although the system needs further improvement for future tasks in the proposed ATS project, it is capable of accurate, two-dimensional surface temperature measurement. This system exploits the temperature sensitivity of both the fluorescence intensity and lifetime of the 512 nm emission triplet of europium-doped lanthanum oxysulfide ($\text{La}_2\text{O}_2\text{S:Eu}^{+3}$). A pulsed laser is used to excite a phosphor-coated surface, and the resulting fluorescence of the 512 nm emission, along with that of the relatively temperature-independent 620 nm emission line, is acquired. The ratio of the intensities of the 512- and 620-nm emissions is then correlated with temperature.

The TGP thermal imaging system is shown schematically in Figure 1. Key elements

of the system include: 1) the laser excitation hardware, 2) the phosphor temperature sensor, 3) the image collection optics and 4) the imaging subsystem. The function of each of these components or subsystems are briefly discussed below.

Ultraviolet excitation of the phosphor coating of the test surface is provided by the tripled output (355 nm) of a Nd:YAG pulsed laser. The maximum 355-nm pulse energy output with a 5 to 6 ns pulse duration is approximately 75 mJ. The phosphor coating on test specimens consists of a 1.0 mole percent europium-doped lanthanum oxysulfide formulation having a mean particle size of 7 microns. The test surfaces are prepared using a settling technique commonly employed in the construction of cathode ray tubes. This method provides a 25- μm thick phosphor coating that under uniform UV excitation exhibits a fluorescence uniformity better than ± 5 percent. Since this type of coating is extremely fragile and can be damaged by incidental contact, a thin transparent potassium silicate overcoat is applied. Tests to assess the influence of the protective overcoat on the fluorescence properties of the phosphor reveal there to be no perceptible change due to the presence of the transparent overcoat.

The fluorescent image of the test surface is collected by a 150 mm enlarger lens and split into two equal length optical paths that are focused side-by-side onto the image detector. One optical path passes through a 510-nm narrow bandpass filter, while the other is directed through a 620-nm bandpass filter and a neutral density filter. The neutral density filter is required to reduce the intensity of the 620-nm fluorescence signal to a level comparable to that of the 510-nm signal, because the intensity and lifetime of the 620-nm emission is considerably greater than that of the temperature sensitive 510-nm emission. Precise focus of the two

Figure 1. UV-Induced Fluorescence Thermal Imaging System

Figure 2. Phosphor Calibration

images on the detector is accommodated by a bellows assembly that permits movement of the enlarger lens to vary its focal plane.

The major components of the imaging subsystem are an image-intensified CCD detector, a gate pulse generator, a detector controller and a microcomputer for overall system control. Concurrent with laser excitation, the detector simultaneously records images of the surface at the emission-line wavelengths of interest. The timing and duration of the image acquisition are controlled by the gate pulse generator, which issues a square wave pulse slightly in advance of laser firing. This pulse, which enables the image intensifier, essentially serves as an electronic shutter. The duration of the pulse is variable and defines the fluorescence integration period. To prevent exposure of the detector during read out following image acquisition, the controller disables the pulse generator until the image data has been read and stored to disk. Finally, a 80486 processor-based computer allows display of real time or stored video images and is used in image post-processing to calculate local emission intensity ratio values that are used in conjunction with the phosphor calibration to determine local surface temperatures.

RESULTS

As an initial evaluation of the capabilities of the TGP thermal imaging system, tests were conducted to develop a calibration curve for the $\text{La}_2\text{O}_2\text{S}:\text{Eu}^{+3}$ phosphor and to assess the steady-state temperature measurement accuracy and spatial resolution. Results of these experiments are discussed below.

Thermographic Phosphor Calibration

Phosphor calibration is conducted using a 25-mm square by 12.7-mm thick phosphor-coated copper test block whose temperature is precisely controlled by a forced circulation isothermal bath. The bath temperature is controlled by an internal thermostat that cycles a heater on and off to maintain circulating water temperature to within 0.1 C of the setpoint temperature. Whenever the bath temperature is increased during a calibration test, approximately five minutes are allowed for the test block to reach a stable, steady-state temperature before image data are collected.

To establish a calibration curve, fluorescent image data are gathered for eight temperatures in the range of 18.9 C to 60.0 C. Prior to collection of image data, however, real time images of the test specimen are employed to adjust the laser power. Once the laser power is adjusted, 10 image frames, each representing the integrated fluorescence intensity over 20 laser pulses, are acquired at each temperature. The intensifier gate, or signal integration period, for each laser pulse is 40 ms. Following data acquisition, the value of the 510-to-620 nm intensity ratio for each frame is calculated using an 11 by 11 pixel array, which is equivalent to a 1.6 mm² area on the test block surface. The average values of this ratio for the 10 frames collected at each temperature are presented in Table 1, along with pertinent statistical data. For an individual image frame, the maximum deviation of the calculated intensity ratio from the averaged value is equivalent to a temperature error of 0.5 C, while the deviation at lower temperatures is about one-half of this amount. As has been noted, this decrease in accuracy may be traced to the decrease in the SNR of the 510-nm intensity as its value decreases with temperature.

Table 1. $\text{La}_2\text{O}_2\text{S}:\text{Eu}^{+3}$ Calibration Data

Temperature (C)	Average Ratio	Std Deviation of Ratio	Greatest Deviation from Average Ratio
18.9	1.3320	0.0151 (0.21 C)	0.0181 (0.26 C)
25.3	0.9282	0.0129 (0.23 C)	0.0202 (0.37 C)
30.1	0.6915	0.0061 (0.14 C)	0.0087 (0.21 C)
35.4	0.5045	0.0035 (0.12 C)	0.0073 (0.24 C)
40.1	0.3841	0.0030 (0.14 C)	0.0053 (0.25 C)
45.0	0.2945	0.0049 (0.33 C)	0.0077 (0.52 C)
50.1	0.2316	0.0024 (0.23 C)	0.0041 (0.39 C)
60.0	0.1490	0.0027 (0.48 C)	0.0030 (0.53 C)

A series of sensitivity tests were performed to determine how slight variations in key operating parameters such as the excitation level, intensifier gain and exposure period may affect the measured emission intensity ratio. None of these parameters was found to have a significant effect on the measured emission intensity ratio. In fact, data from these independent calibrations are used to define the $\text{La}_2\text{O}_2\text{S}:\text{Eu}^{+3}$ calibration curve shown in Figure 2.

Measurement Accuracy and Spatial Resolution

The measurement accuracy and spatial resolution tests are conducted using the same setup employed in the phosphor calibration. After the obligatory system warm up, but before heat up of the circulating isothermal bath, a room-temperature reference fluorescent image of the test specimen is acquired, and the surface temperature associated with this image is recorded for later use in raw image data processing. The thermostatic control of the circulating isothermal bath is then used to raise

the water temperature to 30.2 C. When a steady-state bath temperature is reached, 10 frames of fluorescent image data are collected and stored.

The raw image data are processed to yield emission intensity ratios over a 40 by 6 pixel area, which is equivalent to a surface area of 6.6 mm by 1.0 mm. For each pixel in this area, the frame-averaged intensity ratio for both sets of image data are calculated. The ratio from the data obtained at the "unknown" temperature (i.e., 30.2 C) is then divided by the corresponding ratio from the reference image to obtain a corrected ratio. With a knowledge of the reference temperature, corrected ratios can be used in conjunction with the calibration curve to determine temperature. That is, the corrected ratio is multiplied by the value of the emission ratio at the reference temperature from the calibration curve to yield an intensity ratio for which the curve may be directly used to determine temperature. This "correction" process is employed to account for slight variations in intensity ratio data that may be caused by differences in excitation level between the calibration and the experiment.

Table 2. Measurement Accuracy as a Function of Measurement Area

Measurement Area (pixel dimensions)	Maximum Error (C)	Average Error (C)
1 x 1	1.37	0.09
2 x 2	1.19	0.21
3 x 3	0.96	0.10
4 x 4	0.70	0.20
5 x 5	0.64	0.12
6 x 6	0.52	0.10

To ascertain the spatial resolution capabilities of the system, local temperatures in the measurement area are determined using different sized calculational areas, ranging from a single pixel to a 6 by 6 pixel area. While the temperature error decreases as the calculational area is increased, a minimum error of 0.1 C is approached for an area comprised of 200 pixels. For the present setup, this area corresponds to a square area on the test surface with a dimension of approximately 2.3 mm.

The temperature errors for the different calculational areas employed in determining local temperature are given in Table 2. The average error reported in the table is the average error for all individual points along the vertical dimension of the measurement area. For example, for a calculational consisting of a single pixel, there are 40 points in the total measurement area at which discrete surface temperatures are calculated. From the results in the table, it is readily apparent that the tradeoff involved with increased spatial resolution is a decrease in the measurement accuracy. To ensure a measurement accuracy of ± 0.5 C, these results indicate the minimum spatial resolution to be 1.0

mm. This does not, however, imply that the level of error associated with greater spatial resolution is unacceptable.

FUTURE WORK

This study demonstrates the efficacy of the LIF thermal imaging system for two-dimensional steady-state temperature measurement. The calibration of the phosphor, which is relatively immune to slight variations in excitation level, camera intensifier gain and exposure period, reveals the capability to achieve temperature measurement accuracy on the order of ± 0.5 C a spatial resolution of 1.0 mm. These results are based on surface imaging at a distance of approximately 1 m from the test surface. Future work will be focused on further improvement of imaging system performance, phosphor selection for high temperature applications, and phosphor-surface bonding techniques.

ADVANCED MULTISTAGE TURBINE BLADE AERODYNAMICS, PERFORMANCE, COOLING & HEAT TRANSFER

CONTRACT INFORMATION

Contract Number 94-01-SR019

Contractor School of Mechanical Engineering
Purdue University
West Lafayette, Indiana 47907
Fax: 317-494-0530

Head PI Sanford Fleeter

Co PI Patrick B. Lawless

DOE Project Manager Dr. Paul Micheli / Dr. Norm Holcombe

Period of Performance August 15, 1994 to August 14, 1997

Schedule and Milestones

	Year 1	Year 2	Year 3
Advanced Airfoils			
Baseline Experiments			
Advanced Airfoil Experiments			

OBJECTIVES

The overall objective of this experimental research program is to improve the design capability for high temperature turbines by providing a thorough and detailed understanding and data base of the turbine flow field and its effect on heat transfer. In particular, the objective is to experimentally investigate the fundamental three-dimensional and unsteady flow and heat transfer phenomena which control the performance of advanced turbines. Specific experimental research objectives include an investigation to quantify the potential aerodynamic performance improvements associated with advanced design concept turbine blading designed with CFD codes. This will be accomplished in conjunction with the Allison Engine Company. In particular, advanced design turbine airfoils will be designed by the Allison

Engine Company specifically for these experiments. In addition, this research program will investigate and quantify the detailed three-dimensional and unsteady flow through the first stage, both stator and rotor, and second stage vane row of the multi-stage research turbine. The program will also investigate and quantify the detailed unsteady heat transfer due to vane-wake interaction.

BACKGROUND INFORMATION

The gas turbine has the potential for power production at the highest possible efficiency. The challenge is to ensure that gas turbines operate at the optimum efficiency so as to use the least fuel and produce minimum emissions. A key component to meeting this challenge is the turbine.

Turbine performance, both aerodynamics and heat transfer, is one of the barrier advanced gas turbine development technologies. This is a result of the complex, highly three-dimensional and unsteady flow phenomena in the turbine.

Improved turbine aerodynamic performance has been achieved with three-dimensional highly-loaded airfoil designs, accomplished utilizing Euler or Navier-Stokes Computational Fluid Dynamics (CFD) codes. These design codes consider steady flow through isolated blade rows. Thus they do not account for unsteady flow effects. However, unsteady flow effects have a significant impact on performance. Also, CFD codes predict the complete flow field. The experimental verification of these codes has traditionally been accomplished with point data - not corresponding plane field measurements. Thus, although advanced CFD predictions of the highly complex and three-dimensional turbine flow fields are available, corresponding data are not.

To improve the design capability for high temperature turbines, a detailed understanding of the highly unsteady and three-dimensional flow through multi-stage turbines is necessary. Thus, unique data are required which quantify the unsteady three-dimensional flow through multi-stage turbine blade rows, including the effect of the film coolant flow. This requires experiments in appropriate research facilities in which complete flow field data, not only point measurements, are obtained and analyzed.

Also, as design CFD codes do not account for unsteady flow effects, the next logical challenge and the current thrust in CFD code development is multiple-stage analyses that account for the interactions between neighboring blade rows. Again, to verify and or direct the development of these advanced codes, appropriate experimental data, i.e., complete three-dimensional unsteady flow field data, not point measurements, in appropriate research facilities are needed. This will lead to experimentally verified advanced CFD multi-stage codes which can be used in the design process.

PROJECT DESCRIPTION

This project is directed at obtaining fundamental data, both steady and unsteady, to quantify the fundamental turbine flow and heat

transfer phenomena inherent in high performance, high efficiency, multi-stage turbines.

A key requirement for these experiments is the facility itself. Experiments directed at the quantitative study of turbine aerodynamics, performance and heat transfer require that both the unsteady flow and the fundamental interactions of the unsteady flow with the inherently three-dimensional turbine flow field be experimentally simulated. Thus, the only valid facility for the experimental investigation of turbine aerodynamic performance and heat transfer is a research turbine facility. Thus, the experimental research will be performed in the Purdue Turbine Research Facility (Figure 1), a two stage, low speed turbine which produces the essential aspects of the steady and unsteady flow fields inherent in high speed multistage turbines. The current 50% reaction blading with A3K7 series airfoils will be used as a baseline for both the performance and heat transfer studies. The Allison Engine Company will design advanced concept turbine airfoils with one row of film cooling holes specifically for this program.

The unsteady three-dimensional flow through the multi-stage research turbine will be accomplished by means of Particle Image Velocimetry (PIV), a relatively new measurement technique which provides instantaneous whole field high resolution velocity field data. Taking advantage of the high frequency response of thin film sensors, the blade surface unsteady heat transfer measurements will be made using constant-temperature anemometry and platinum hot film surface sensors.

This series of experiments will improve the design capability for high temperature turbines by providing a thorough and detailed understanding and data base of the turbine flow field and its effect on heat transfer. With these data, the designer can increase turbine aerodynamic performance and optimize turbine cooling flow schemes.

FUTURE WORK

This series of experiments will improve the design capability for high temperature turbines by providing a thorough and detailed understanding and data base of the turbine flow field and its effect on heat transfer. In particular, the objective is to experimentally investigate and quantify the fundamental three-dimensional and unsteady flow

and heat transfer phenomena which control the performance of advanced turbines. With these data, the designer can increase turbine aerodynamic performance and optimize turbine cooling flow schemes.

Advanced Airfoil Design, Fabrication & Instrumentation

Design, fabricate and instrument advanced concept turbine blading with provisions for one row of film cooling holes. This will be accomplished in conjunction with the Allison Engine Company, with Purdue University responsible for the fabrication and instrumentation of the airfoils. Also, optical access to the test section of the research turbine will be provided.

Baseline Experiments

A series of experiments will be performed in the baseline turbine facility to investigate and quantify the detailed three-dimensional unsteady aerodynamics and heat transfer through the first stage, both stator and rotor, and second stage vane row of the baseline multi-stage research turbine. Conventional steady aerodynamic performance, i.e., blade surface static pressure taps, and advanced instrumentation including PIV and high frequency thin film sensors will be utilized.

Advanced Airfoil Experiments

A series of experiments will be performed utilizing the advanced concept airfoils designed and fabricated by Allison to investigate and quantify the three-dimensional unsteady aerodynamics and heat transfer.

Figure 1. Research turbine facility schematic

STEAM AS TURBINE BLADE COOLANT PHASE I: DEVELOPMENT OF THE EXPERIMENTAL FACILITY

CONTRACT INFORMATION

Contract Number 94-01-SR026

Contractor A231EB, Turbomachinery Lab
Mechanical Engineering Dept.
Michigan State University,
E. Lansing, MI 48824-1226
Phone: 517-432 1834; Fax: 517-353 1750

Principal-Investigator Abraham Engeda

Co-Investigator John R. Lloyd

DOE Project Manager Dr. Paul Micheli/ Dr. Norm Holcombe

Period of Performance July 1, 1994 - June 30, 1995

Schedule and Milestones July 1, 1994 - June 30, 1995

	S	O	N	D	J	F	M	A	M	J	J	A
Test plan	_____											
Fabric.	_____											
Testing	_____											
Analysis	_____											

OBLLECTIVES

From the various available blade cooling techniques, internal liquid cooling has the highest cooling potential and appears to be the best choice for very high temperature applications in stationary power generation gas turbines. However, using a liquid as a coolant, like water for example, has the disadvantage that either vaporization must be allowed to take place or that operation with very high pressures related to the water temperatures must be allowed.

Using dry steam as a coolant has sparked great research interest because steam has a higher thermal conductivity than air (meaning higher heat transfer coefficient) and steam has a higher specific heat. Both of these qualities qualify steam as a superior coolant to air. Jenkins and Rice /1/ proposed steam as a possible coolant. Arsenieve et al /2/ actually tested steam as a coolant on a gas turbine and produced promising results.

Heat transfer research into blade cooling requires both analytical and experimental work in order to provide, in addition to the heat transfer knowledge, the boundary conditions needed for structural predictions like: thermal fatigue, creep progression, and chemical deterioration. This current work, running in three phases, aims:

• In Phase I - Internal Cooling:

a) Develop

- To develop the major experimental facility and the associated instrumentation on smaller test rigs.

b) Establish

- To establish base line data for the internal coolant capabilities of steam and air on the basis of heating tests (air-air and steam-air) on a simple flat blade geometry.

c) Compare

- To compare the coolant properties of air and steam on the basis of the base line test.

d) Evaluate

- To evaluate existing analytical tools for conduction and convective heat transfer and properties of steam.

BACKGROUND INFORMATION

In high temperature gas turbines, film cooling is more or less the standard method of cooling. Even though a vast amount of research has been conducted in this area in the past thirty years, the designer of film-cooled-blades still does not possess sufficient data for correlation.

In gas turbine technology, turbine reliability is a major design criterion. Therefore, the thermodynamically superior cooling method, transpiration cooling, which still faces many technical drawbacks, has still not been fully applied to a production turbine. However, it continues to be a subject of intense research interest.

One of the major disadvantages of film and transpiration cooling is their harsh effect on the aerodynamic performance of the

turbine. According to Grives /3/, the net turbine efficiency decreases steadily as turbine entry temperature increases.

In the last few years the interest in internal cooling has been strongly renewed. Among the several factors supporting interest in internal cooling, two recent events are:

- the development of thermal barrier coatings, and
- new manufacturing techniques, like diffusion bonding.

One great set back in blade cooling heat transfer is the lack of an organized set of experimental data, prediction methods, and information on coolants. This is mainly due to the fact that much information on the subject is of a commercial and proprietary nature and hence difficult to obtain. Despite this there have been attempts in the past to compile and organize existing knowledge. For example:

- Smith /4/ reviewed early work conducted in the 1940s
- Goldstein /5/ did an excellent review of works on film cooling of flat plates up to 1970
- Hay /6/ also conducted a review of film cooling research
- Bayley and Martin /7/ reviewed the development of liquid cooling technology

PROJECT DESCRIPTION

The current work will consist of experimental part mainly and a small amount of analytical part. Basically, a flat blade in a stream of air will be internally heated by superheated steam. The heat

transfer from the steam to the air will be experimentally studied. The analytical part will deal with the prediction, correlation, and general quantification of the heat transfer process.

Measured Quantities

At the air side, the variation of the static pressure and local heat transfer coefficient around the chosen profile subject to a stream of air flowing parallel to its major axis will be studied over a range of Reynolds number, and steam conditions. One way of dealing with the heat flux through the test surface will be to treat the steam flow as uniform and simply account for the enthalpy change and steam flow rate. The enthalpy change can be determined from temperature and pressure measurements of the steam at inlet and outlet manifold of the test blade. It will be interesting here to verify whether the bulk enthalpy variation along the test section is linear and compatible with uniform heat flux assumptions. Another approach will be to try to obtain the steam heat transfer coefficient inside the steam passage based on the steam flow, pressure, temperature, inner channel geometry, and thermal and transport properties.

The major quantities that will be measured will be:

• Temperatures

- T_u , free stream air temperature
- T_w , air side wall temperature distribution
- $T_{st, i-o}$, temperature of steam in and out

- $T_{a, i-o}$, "coolant" air temperature of steam in and out

• Pressures

- p , static pressure distribution of test surface
- $p_{st, i-o}$, steam pressure, in and out
- $p_{a, i-o}$, "coolant" air pressure, in and out

• Velocity and flow rates

- u_u , free stream air velocity distribution
- turbulence intensity before test section
- "coolant" steam and air flow rates
- main stream air flow rate.

In the analysis it will be essential to consider the significance of these measured quantities in light of their deviation from actual gas turbine values, steadiness and accuracy, and factors that they depend on.

Derived Quantities

The experimental part of the current research is basically a convective heat transfer problem, which is a boundary-layer phenomenon. This requires information in the first place about the Reynolds number and the Mach number of the fluid, in order to be able to determine the flow configuration with heat transfer. Additionally, the temperature and pressure distributions over the test surface and knowledge of the variation of the fluid properties between the test surface and the fluid temperatures are required. To consider the variation of the fluid property across the boundary layer,

knowledge of ratios of fluid to test surface temperatures, viscosity, density, specific heat, thermal conductivity and Prandtl number are required. Major derived quantities that will be required will be:

- h_a , heat transfer coefficient of air, local and average
- h_{st} , heat transfer coefficient of steam, local and average
- turbulence intensity of the air
- Re_a , the Reynolds number of air, based on test section
- temperature ratio
- Ma , the air Mach number
- q , heat flux from "coolant" steam
- Δh_{st} , enthalpy fall of steam

Temperature Distribution in the Metal

One of the main tasks will be to identify and select an analytical tool for predicting the temperature distribution in the metal that separates the air/steam and air/air. Among the early methods for predicting temperature distribution in a blade, the most common one is that of Ainley /8/, which is based on a shape factor defined on the basis of blade chord C , total perimeter of coolant passage S_c , and total area of cooling passages in one blade A_c , expressed as:

$$Z = \frac{\left[\frac{S_c}{C} \right]^{1.2}}{A_c / C^2}$$

The shape factor was related proportionally to the ratio of the products of heat transfer coefficients, α_c , and perimeter, S_c , of coolant to hot gas, $\alpha_g S_g$. Such simplified approaches are limited to "single pass" cooling systems, have a

limited reduction of blade temperature, but are good enough to identify the limitations of the heat flux rate that could be absorbed by internal convection.

Currently a number of analytical methods exist to predict temperature distribution within a blade, as described by Dunn et al /9/. An intensive evaluation of available and related codes will be carried out.

In the current work, one or more analytical tools will be selected, modified if necessary in order to:

- study and quantify the air-air and steam-air heat transfer,
- compute steam properties,
- predict steam cooling from air cooling data, and
- correlate the two modes of cooling.

Convective Heat transfer Air Side

If a conductive wall is either heated or cooled externally by a gas flowing past it, the boundary condition is referred to the temperature profile of the thermal boundary layer, whose range covers the area from the wall ($y=0$) where the gas enthalpy, h_w , and temperature, T_w , are defined on the basis of the nature of the material (insulating/conducting). The upper limit is taken to be where the total enthalpy is equal to that of the free stream flow.

For the case of an adiabatic wall, the temperature is defined as the adiabatic surface temperature, T_{aw} , and the corresponding enthalpy as h_{aw} . A recovery factor is then defined as:

$$r = \frac{h_{aw} - h_u}{h_{tu} - h_u} = \frac{T_{aw} - T_u}{T_{tu} - T_u}$$

where subscripts t and u denote total and free stream. The ratio of the wall temperature to the free stream total temperature can be expressed in terms of the free stream Mach number as.

$$\frac{T_{aw}}{T_u} = 1 + \frac{\gamma - 1}{2} r M_u^2$$

For the case of a conductive wall, the heat flux from the gas to the wall is given as:

$$q_w = - \left(k \frac{\partial T}{\partial y} \right) = - \left(\frac{k}{c_p} \frac{\partial h}{\partial y} \right) = \alpha_u (T_{aw} - T_w)$$

Using dimensional analysis it can be shown that the heat flux rate depends only on:

- curvilinear coordinate, measured along a stream line from the leading edge
- free stream velocity, u_u
- driving temperature potential, $T_{aw} - T_w$
- fluid properties: viscosity, density, conductivity and specific heat.

These variables will be expressed in dimensionless form in terms of the Nusselt, Reynolds and Prandtl numbers:

$$Nu(x) = \frac{q_w x}{k(T_{aw} - T_w)} = \frac{\alpha_u x}{k}$$

$$Re(x) = \frac{u_u x \rho_u}{\mu}$$

$$Pr = \frac{\mu c_p}{k}$$

Thus the heat flux in dimensionless form is:

$$Nu(x) = f[Re(x), Pr]$$

The heat flux rate is known to be closely proportional to the driving temperature potential, and can be expressed as:

$$q_w = \alpha(x)(T_{aw} - T_w)$$

Another important dimensionless parameter used for correlating is the Stanton number,

$$S_t(x) = \frac{q_w(x)}{\rho_a u_a c_p (T_{aw} - T_w)} = \frac{Nu(x)}{Re(x) Pr}$$

Steam Flow

In dealing with the steam side heat transfer, an easy solution is to adopt an existing software like that developed by NIST.

Another approach is to analyze the heat transfer strictly from the point of view of the steam passage only. This requires the development of a computational procedure to accurately calculate the pressure drop of steam flow in the coolant passage on the basis of its temperature, pressure and steam-flow-passage geometry, and the determination of the steam heat transfer coefficient inside the steam passage, based on:

- steam flow rate
- steam pressure
- steam temperature
- steam thermal and transport properties: viscosity, density, conductivity and specific heat.

A starting point for such an analysis to determine the steam heat transfer coefficient can be any of the common correlation relationships.

Experimental Facility

The major test facility can be run in a blow down or suction mode. The air supply for the test section is generated by a centrifugal compressor, which is powered by a 300 HP frequency controlled motor. Flow can therefore be adjusted finely, both by throttling and by speed control.

- The mass-rate will be measured using an ASME square-edged orifice plate.
- The local heat flux will be determined from a two dimensional temperature distribution, measured using liquid crystal and infrared methods. The temperature distribution in the blade will be calculated from the wall temperatures, using a finite element analyses.
- Two 48-port scanivalves will be used with an associated controller and decoder for static and total pressure measurements.
- An HP-3852A 40-port data acquisition unit will be used to scan wall and stream temperature distributions.
- Both the scanivalve and HP-3852 A are controlled by a PC.

An HP-35665A Signal Analyzer will be used to measure and analyse unsteady effects.

- Mean and turbulent velocity measurements will be made using a two channel high frequency anemometer, which has an automatic temperature correction for velocity and built-in signal conditioning. In addition, it makes localized temperature measurements. Its operation is fully automated via Lab Windows- based software.

Figures 1 and 2 show the measurement and the general air loop systems. Figure 3 shows the test section schematically. By opening either valve A or B the test section can be run in a blow down or suction mode. The test section basically consists of three major parts:

1. Inlet Device

In a blow down mode, pressured air from the compressor approaches the test section axially and then is turned smoothly in the radial direction. As the flow is turned radially it passes through a section which can be fitted either by a "turbulence" regulator or by a vortex generator.

2. Test Section

The test section is variable:

- Its height, defined by the spacing between two circular plates, can be varied by changing the spacers.
- Internal, both plates have changeable discs, where different types of blades or test passage arrangement can be fixed.
- Each plate has three window positions (arranged at 120°), which can be arranged correspondingly for

a see through, or at an offset position.

- The test section can be fitted with:

- parallel straight walls
- an accelerating passage with straight walls
- a diffusing passage with straight walls
- parallel curved walls
- an accelerating passage with curved walls
- a diffusing passage with curved walls.

3. Exit Device

In the blow down mode, the radially outer section is the exit device. This section diffuses the flow smoothly to the room.

The first test blade will be a simple flat blade. The coolant passages will be drilled in the model so that steam or air flows at right angle to the main air stream. Insulated manifolds will be fixed on the model at the inlet and the outlet. The temperature of the steam or air entering and leaving the model will be measured at these manifolds using appropriate thermocouples.

On the air side the velocities at different stream wise location over the test model will be determined:

- out of total pressure measurements up stream of the test model and static pressure taps on the model, and
- using thermal anemometry.

The test section side walls will be transparent to allow visual access during axial and span wise traversing of probes, especially during boundary layer traversing.

The wall surface temperature will be obtained using:

- thermocouples embedded in the test model,
- liquid crystal, and
- infrared methods.

RESULTS

Current work progress can be summarized as:

- testing of main stream aerodynamics and "coolant" flow instrumentation on a smaller test apparatus,
- development of surface temperature measurement on another smaller test apparatus, and
- development of the major test section.

FUTURE WORK

Future plans of the current work will include:

- In Phase II - Internal Cooling:

a) Examine

- Examine the mechanical, hydrodynamical, and geometrical limitations and modifications that may arise, if the current blade coolant (air) of ground based gas turbines is replaced by steam.

b) Investigate

- Repeat the studies of Phase I, which was a heating test, by actual air-air and steam-air cooling tests.

c) Predict and Compare

- Predict the steam coolant characteristics from the air-air test and to compare with the steam-air test.

- In Phase III - Conclusive efforts:

- To be defined on the basis of Phase I and Phase II

REFERENCES

- 1) P. E. Jenkins and I. G. Rice: Blading Heat Transfer Considerations in a Reheat-Gas Turbine Combined Cycle, ASME Paper 81-GT-155, 1981
- 2) L. V. Arsenieva, U. G. Korsov, E. A. Khodak, G. A. Romakhova, and G. V. Bazhenov: Combined Cycle Plant Based on High-Temperature Steam Cooled Gas Turbine, 1991 ASME Coegn-Turbo, IGTI-Vol. 6
- 3) E. Le Grives: Advanced Topics in Turbomachinery Technology, Concept ETI, ISBN 0-933283-02-4
- 4) A. G. Smith: Heat Flow in the Gas Turbine, Proc. Inst. Mech. Engr., Vol. 159, 1948
- 5) R. J. Goldstein: Film Cooling, Advances in Heat Transfer, Vol. 7, Academic Press, 1971
- 6) N. Hay: Discrete Hole Film Cooling of Turbine Blades, 5th ISABE, Bangalore, 1981
- 7) F. J. Bayley and W. J. Priddy: Effects of Free Stream Turbulence Intensity and Frequency on Heat Transfer to Turbine Blading, Trans. ASME, J. Engr. Powr., 103, 1981
- 8) D. G. Ainley: Internal Air Cooling for Turbine Blades - A General Design Survey, Aeronautical Research Council, R&M No. 3013, 1955
- 9) M. G. Dunn and F. J. Stoddard, Measurement of Heat Transfer Rate to a Gas Turbine Stator, ASME Paper 78-GT-113, 1978

Figure 1. The major data acquisition system

Figure 2. The major test facility with 300 HP centrifugal compressor

Figure 3. The multi purpose radial cascade test section

Combustion Chemical Vapor Deposition Coatings for Thermal Barrier Coating Systems

CONTRACT INFORMATION

Contract Number	94-01-SR027
University	Georgia Institute of Technology School of Materials Science and Engineering Atlanta, GA 30332-0245 (404) 894-6762 (404) 853-9140 FAX
Head P.I.	W. Brent Carter
Co-investigators/ research students	Janet M. Hampikian (Co-P.I.)/ G.W. Book, Michelle Hendrick and T. A. Polley
DOE Project Manager	Dr. Paul Micheli and Dr. Norm Holcombe
Period of Performance	July 1, 1994 - June 30, 1997

Schedule and Milestones

FY 95 - FY 97 Program Schedule*

	Program Month											
	0	3	9	12	15	18	21	24	27	30	33	36
CCVD of YSZ Overlayers	<hr/>											
CCVD of YSZ Interlayers	<hr/>											
CCVD of Alumina Overlayers	<hr/>											
CCVD of Alumina Interlayers	<hr/>											
CCVD of Silica Interlayers	<hr/>											
CCVD of Ceria Overlayers	<hr/>											
CCVD of Ceria Interlayers	<hr/>											
CCVD of Spinel	<hr/>											

* Georgia Tech's Fiscal Year 1995 began 1 July 1994.

OBJECTIVES

The principal technical objective of this research program is to increase the service life of the TBC's used in the combustor and turbine regions of gas turbines. The strategies for achieving this are discussed below, and include using a new coating technology, combustion chemical vapor deposition (CCVD).

A secondary technical objective in tasks involving the CCVD deposition of overlayers (see below) will be to increase the aerodynamic efficiency of TBC coated parts in the hot section of gas turbines by improving the surface finish of the TBC's.

In addition, this program has as an objective the education of materials science and engineering graduates trained in the materials technology of advanced gas turbines.

BACKGROUND INFORMATION

Thermal Barrier Coatings (TBC's)

Yttria-stabilized zirconia coatings are widely used for the thermal protection of high-temperature components in gas turbine engines. For highest efficiencies, these engines are operated at as high a temperature as possible. Therefore, in addition to providing thermal protection to the underlying metal, the ceramic coatings must also serve as chemical barriers, to protect the underlying metal against oxidation and corrosion. The zirconia-based ceramics are appropriate for gas turbine use because of their low thermal conductivity and good thermal shock properties, however, they are relatively poor chemical barriers for the following reasons. First, depending on the method of deposition (e.g. plasma spray), the zirconia may have a highly porous and microcracked structure. This structure, while providing strain tolerance to the ceramic and lowering its thermal conductivity,¹ is very

permeable to atmospheric gases and liquids, resulting in oxidation of the bond coat and spallation of the coating. Another reason that zirconia coatings are poor barriers in terms of oxidation protection is that being ionic conductors, they are transparent to oxygen at elevated temperatures; ionic transportation occurs through the lattice by reverse movement of oxygen ion vacancies.

Therefore, in order to protect the underlying alloy from catastrophic oxidation, a bond coat is applied prior to deposition of the TBC. This bond coat is of general composition MCrAlY, where M is usually Ni, Co, or NiCo. The bond coat provides protection to the alloy beneath it by forming a protective oxide, such as alumina. However, once the bond coat is depleted of unoxidized aluminum, spinels can form, and failure of the coating system can occur.² As a consequence, industrial turbine engine parts require maintenance, which includes costly re-coating of engine parts, after some predetermined service time, often in the range of 10,000-30,000 hours.

Industrial TBC's are generally produced by either air plasma spray (APS) or electron beam physical vapor deposition (EB-PVD). These techniques are described below.

Air plasma spray thermal barrier coatings utilize rough bond coats to provide a mechanical interlocking or mechanical bond with the ceramic top coat, and relatively large powder sizes to introduce porosity which further reduces the thermal conductivity within the ceramic top coat. Both of these features, combined with the plasma deposition process itself, tend to produce very rough surface finishes for plasma sprayed TBC's. This rough, splat-like structure must then undergo surface finishing operations in order to take full advantage of their thermal barrier properties. Surfaces must typically be smoothed to improve airflow and erosion properties, and densification and/or overlayers are used to improve erosion and corrosion properties.

Electron beam physical vapor deposition thermal barrier coatings rely on a chemical bond with the substrate and therefore, they do not require rough bond coats to promote adhesion. EB-PVD TBC's typically exhibit a columnar structure, with extensive porosity between the columnar grains. The surface roughness of EB-PVD TBC's results from the morphology of the tips of the columnar grains, which are shaped like pyramids. These features result in a much smoother surface for EB-PVD TBC's than their plasma sprayed counterparts. Never-the-less, EB-PVD TBC surfaces are finished to provide optimized airflow and to improve corrosion and erosion resistance.

The two primary deposition techniques, air plasma spray and EB-PVD, therefore typically require surface finishing operations following coating. Alternative surface layers are often applied for this purpose,³ but these overlayers are typically applied by the same APS or EB-PVD processes, and thus suffer from many of the same inadequacies. A technique which could allow for a dense, thin outer finishing layer is thus highly desirable. Laser sealing of the outer bond coat has been investigated, but has limitations which result from the extensive cracking of the melted and rapidly cooled "sealed" layer,⁴ which is, of necessity, the same material as the TBC (usually yttria stabilized zirconia). A technique which could impart a dense, high quality, strain tolerant outer ceramic layer could therefore significantly improve TBC's. Conventional chemical vapor deposition is such a technique, but for reasons including the high capital investment and the necessity to place parts within a reaction vessel, it is not practical as a finishing step for parts. For this reason, an alternative and innovative approach to coating surface enhancement is proposed to enhance coating lifetimes. This new coating technique is entitled *combustion chemical vapor deposition* (CCVD). The remainder of this section describes this recently invented technique.

Combustion Chemical Vapor Deposition

Combustion chemical vapor deposition (CCVD) is a flame assisted CVD method that can be performed in the open atmosphere.⁵ It is a non-line-of-sight process that has produced epitaxial coatings. Coatings of the following oxides have been successfully produced with CCVD: yttria stabilized zirconia,⁶ amorphous silica,⁷ yttrium iron garnet, barium titanate, $\text{YBa}_2\text{Cu}_3\text{O}_x$,⁸ and Y_2BaCuO_x . Metallic platinum and silver have also been deposited with CCVD.

Two major variants of CCVD have been developed: liquid fuel CCVD and gaseous fuel CCVD. In liquid fuel CCVD, chemical precursors of the coating material are dissolved in a flammable organic solvent. This solution is then nebulized and burned in air. A CVD coating is deposited onto a substrate placed near the end of the resulting flame. The fuel is a significant source of carbon when performing liquid fuel CCVD with an organic solvent.

Gaseous fuel CCVD utilizes a combustible gas as fuel. The film precursors are dissolved in water and the resulting solution is either nebulized and sprayed directly into the flame or nebulized into the gas supply upstream of the burner nozzle. The primary source of carbon is the atmosphere when carbonless film precursors, gases, and substrates are used.

CCVD bears some similarity to the deposition of diamond films with the use of an oxygen-acetylene torch, a technique called flame deposition.^{9,10} In this process, elemental rather than compound coatings are deposited with the use of a premixed flame. Both fuel and oxidant are supplied to the nozzle as gases.

In both flame spraying and plasma spraying, high temperatures are used to melt coating material. The molten material splats against the substrates being coating and rapidly solidifies. These techniques are not CVD processes.

The ability to deposit high quality CVD coatings using inexpensive precursors, e.g., nitrates, without the use of a furnace or vacuum

chamber could significantly affect the economics of coating deposition for some applications. CCVD has the potential to coat large, assembled parts that would be difficult to place in a coating chamber. Coating thickness could be varied over the part being coated, with some areas not being coated, something that is difficult if not impossible to achieve with conventional CVD. This could be done by controlling the dwell time of the flame over the various regions of the part. Compositionally-graded coatings are also easily produced via CCVD by varying the composition of the precursor solutions during deposition.

Combustion CVD is more economical than conventional CVD due to less costly and more environmentally friendly precursors and coating materials. It does not require the extensive capital investment in equipment and controls, and it produces coatings that are more practical and possibly superior to conventional CVD. Deposition rates are comparable to conventional CVD, however, which make it impractical to produce 125-2500 micron thick TBC's solely via the CCVD process.

PROJECT DESCRIPTION

The research is being undertaken as several tasks. Each task involves the development and evaluation of a particular thermal barrier coating system. CCVD will be used in combination with air plasma spray 8% YSZ TBC's (with vacuum plasma sprayed bond coat¹¹) to produce modified CCVD coating systems on Rene 80¹² substrates. These coating systems will be extensively characterized and tested using state-of-the-art TBC baselines as controls.

Each task is comprised of several subtasks such as coating deposition, characterization, furnace cycle testing, erosion testing, and corrosion testing. General Electric Power Generation in Schenectady, NY will be subcontracted to perform several of the subtasks including supplying the substrates, applying bond

coats and APS YSZ, and performing furnace cycle testing, corrosion and erosion testing. The various tasks are briefly described below.

Tasks

Task I - Development and Evaluation of CCVD 8% YSZ Overlayer on Air Plasma Sprayed YSZ TBC. Georgia Tech has significant experience depositing YSZ using CCVD.⁶ The optimum experimental conditions to achieve high quality YSZ layers have already been established. Therefore, an initial task of this program is the evaluation of the improvement in surface finish and erosion properties which results from CCVD-applied YSZ overlayers.

Task II - Development and Evaluation of CCVD 8% YSZ Interlayer Between Bond Coat and Overlying Air Plasma Sprayed YSZ TBC. A CCVD deposited interlayer of dense YSZ between the bond coat and the plasma sprayed TBC may serve to improve the adhesion of the TBC during service by providing this failure-prone region with a stronger, more crack resistant morphology. Strain tolerance would be provided by the subsequent APS YSZ top coat, and the dense YSZ interlayer should improve mechanical bonding with the bond coat and substrate.

Task III - Development and Evaluation of CCVD Al₂O₃ Overlayer on Air Plasma Sprayed YSZ TBC. Alumina overlayers should impede the penetration of oxygen into the TBC and improve the corrosion and erosion properties of the coating system. Alumina is generally regarded as one of the best protective oxides, as diffusion through it is relatively slow in comparison with most other oxides, and because it is thermodynamically stable.¹³ A dense smooth surface layer should also improve the surface finish.

Task IV - Development and Evaluation of CCVD Al_2O_3 Interlayer Between Bond Coat and Overlying Air Plasma Sprayed YSZ TBC.

Alumina deposited as an interlayer between the bond coat and YSZ shows promise in improving the overall TBC lifetime. By placing alumina in this layer, subsequent bond coat oxidation will be greatly diminished, which will result in less strain within the overlying ceramic, and consequently reduce TBC spallation.

Task V - Development and Evaluation of CCVD SiO_2 Interlayer Between Bond Coat and Overlying Air Plasma Sprayed YSZ TBC.

Georgia Tech has experience with the CCVD of amorphous SiO_2 .⁷ (Also see RESULTS below.) Silica has been demonstrated to significantly inhibit oxidation.^{6,14} In our investigation of the oxidation protection afforded by CCVD-applied amorphous silica coatings on Ni-20Cr, we have shown a decrease in overall oxidation kinetics by over an order of magnitude. By being placed beneath the plasma sprayed TBC, it should serve as a barrier to oxygen penetration and reduce oxidation of the bond coat, without being heated to temperatures which could cause recrystallization and decreased oxidation resistance.

Task VI - Development and Evaluation of CCVD CeO_2 Overlayer on Air Plasma Sprayed YSZ TBC.

Ceria overlayers should improve the corrosion resistance of the TBC's in heavily contaminated fuel environments.^{15,16} The higher thermal coefficient of expansion of ceria as compared to YSZ implies that it will not "pin" the surface of the TBC and thus degrade the thermal cycling performance. A dense smooth surface layer should also improve the surface finish.

Task VII - Development and Evaluation of CCVD CeO_2 Interlayer Between Bond Coat and Overlying Air Plasma Sprayed YSZ TBC.

With a coefficient of thermal expansion intermediate between that of the metallic substrate and the YSZ TBC, a ceria interlayer should improve the

thermal cycling performance of the TBC. In addition, it may serve to improve the overall oxidation resistance of the TBC.

Task VIII - Development and Evaluation of CCVD Spinel Layers. Note: The deposition of spinel will first be investigated for feasibility. It is anticipated that these may prove difficult to deposit, and therefore one or the other system will be explored, depending on which yields the best initial results.

a. $\text{MgO}\cdot\text{Al}_2\text{O}_3$ spinel overlayer on air plasma sprayed YSZ TBC. $\text{MgO}\cdot\text{Al}_2\text{O}_3$ spinel overlayers should have improved corrosion resistance as compared to YSZ in heavily contaminated fuel environments and possibly in marine or coastal environments. This spinel should have a higher coefficient of thermal expansion than YSZ and therefore should not "pin" the surface of the TBC and degrade its thermal cycling performance. A dense smooth surface layer should also improve the surface finish.

b. $\text{NiO}\cdot\text{Al}_2\text{O}_3$ spinel interlayer between bond coat and overlying air plasma sprayed YSZ TBC. Ni-based bond coats are protected from oxidation via the formation of Al_2O_3 . Upon depletion of Al in the bond coat, spinels form, which lead to spallation of the TBC's (due to a volume change upon spinel formation). The deposition of a dense spinel interlayer may provide additional strain tolerance to the TBC system by negating the effects of the volume change during oxidation - in effect, by allowing the TBC to accommodate the spinel growth stresses before the TBC is even deposited. This improved strain tolerance could lead to increased TBC lifetimes.

RESULTS

Preliminary results have been obtained for CCVD deposited YSZ, alumina, and silica on trial substrates. These are briefly described below.

Yttria Stabilized Zirconia

Yttria stabilized zirconia has been deposited using both liquid and gaseous fuel CCVD onto single crystal alumina substrates. The coatings are dense and epitaxial.

The liquid fuel CCVD depositions involved the dissolution of 2-ethylhexanoates (2-EH) of Zr and Y in toluene and the combustion of an atomized mist of the resulting solution. Figure 1 illustrates the liquid fuel CCVD system. Deposition temperatures are in the neighborhood of 1,200°C as measured on a thermocouple held near the substrate surface facing the flame.

Figure 2 is a typical scanning electron micrograph of a fracture cross section of a liquid fuel CCVD YSZ coating. Figure 3 is an x-ray diffraction pattern of a liquid fuel CCVD coating produced using a 7.5 mole percent Y 2-EH - 92.5 mole percent Zr 2-EH - toluene solution. The phases present are consistent with the phase diagram and the known behavior of this system assuming that the coating composition is similar to that of the precursor solution.

In an effort to produce a more stable flame with a potentially larger, more uniform coating volume, gaseous fuel CCVD (see below) was performed with a premixed burner. This burner utilizes a Bunsen burner (figure 4), into the base of which is injected hydrogen gas (the fuel) and a pneumatically nebulized mist of toluene containing dissolved 2-ethylhexanoates of Zr and Y. Air (the oxidant) is entrained in the normal manner of a Bunsen burner.

The toluene burns, of course, making this process actually a hybrid between liquid and gaseous fuel CCVD. It should be possible to deposit using gaseous fuel only by dissolving the coating precursors in a non-flammable solvent such as water.

Alumina

Initial depositions using liquid fuel CCVD have been performed in attempts to deposit alumina. Deposition temperatures were in the 1,100-1,200°C range as measured on a thermocouple held near the substrate surface facing the flame. Preliminary x-ray diffraction and XPS data indicate that the coatings are alumina. The form of this material has not been determined.

Silica

Silica has been deposited using liquid fuel CCVD onto 20Cr-Ni substrates. Tetraethyl orthosilicate dissolved in ethanol was the working solution. The deposition temperature was 750±75°C as measured on a thermocouple held near the substrate surface facing the flame.

Scanning electron microscopy data indicate that the coatings are nodular and completely cover the substrates. Transmission electron microscopy results indicate that the silica coatings are amorphous. The oxidation resistance of a coating was investigated by comparing the mass gains of coated and uncoated substrates during oxidation in air at 1,000°C. The results are presented in figure 5.

FUTURE WORK

A subcontracting agreement between Georgia Tech and General Electric Power Generation will be negotiated as quickly as possible. When this is in place, overlayer coatings of YSZ will be applied to TBC coated substrates obtained from GEPC. These will then be characterized and tested as described above.

Work will continue to optimize the CCVD deposition of alumina with a goal of producing amorphous coatings. Work is under way to further develop the gaseous fuel CCVD process

using a premixed flame, with the aim of reducing deposition temperatures and increasing the deposition zone size.

REFERENCES

1. R.A. Miller, NASA Tech. Memo. 87288, National Aeronautics and Space Administration, Washington, D.C.
2. E.Y. Lee, R.R. Biederman, and R.D. Sisson, "Modeling the Lifetimes of Thermal Barrier Coatings Based on the Beta-NiAl Phase Life in the Bond Coat." Proceedings of the 1990 Coatings for Advanced Heat Engines Workshop, Aug. 6-9, 1990, Castine, Maine, sponsored by U.S. DoE, pp. II-85 to II-91.
3. R.A. Miller, "Method of Applying a Thermal Barrier Coating System to a Substrate," U.S. Patent #5,169,674, issued 12/08/92.
4. A. Pettitbon, D. Guignot, U. Fischer, J.M. Guillemot, "Laser Surface Treatment of Ceramic Coatings" Mat. Sci. and Eng. A121 (1989) pp. 545-548.
5. A.T. Hunt, W.B. Carter, and J.K. Cochran, "Combustion Chemical Vapor Deposition: A Novel Thin Film Deposition Technique," Applied Physics Letters, **63**, pp. 266-268, (1993).
6. A.T. Hunt, Ph.D. Dissertation: "Combustion Chemical Vapor Deposition from Liquid Organic Solutions" Georgia Institute of Technology, 1993.
7. J.M. Hampikian, A.T. Hunt and W.B. Carter, "Combustion Chemical Vapor Deposition of Silica on Ni-20Cr," TMS Proceeding, Oct. 17-21, 1993, Pittsburgh, PA.
8. W.B. Carter, G.W. Book, and A.T. Hunt, "Combustion Chemical Vapor Deposition of $\text{YBa}_2\text{Cu}_3\text{O}_x$," *Applied Superconductivity*, Vol. 1, ed. H. C. Freyhardt (DMG Informationsgesellschaft-Verlag, Oberursel, Germany, 1993), pp. 391-394.
9. Y. Hirose and N. Kondo, Jpn. Appl. Phys. Soc., 35th Spring Meeting Ext. Absts., p. 434 (29 March 1988).
10. M. Murakawa, S. Takeuchi, and Y. Hirose, "An Experiment in Large Area Diamond Coating Using a Combustion Flame Torch in its Traversing Mode," Surf. and Coatings Technol., **43-44** (1-3), 22, (1990).
11. Bond coat composition: NiCrAlY = Ni-22Cr-10Al-1Y (in wt.%).
12. Rene 80 composition: Ni-9.5Co-14.0Cr-3.0Al-5.0Ti-4.0W-4.0 Mo-0.17C-0.015B-0.06Zr (in wt.%).
13. D.P. Whittle and J. Stringer, "Improvement in Properties: Additives in Oxidation Resistance," Phil. Trans. R. Soc. Lond. **A295**, 309-329 (1980).
14. M.J. Bennett, M.R. Houlton, D.A. Moore, A.I. Foster, and M.A.M. Swidzinski, "Development of a Chemical Vapor Deposition Silica Coating for U.K. Advanced Gas-Cooled Nuclear Reactor Fuel Pins," Nuclear Technology, **66**, pp. 518-522 (1984).
15. R.L. Jones, "The Development of Hot Corrosion Resistant Zirconia Thermal Barrier Coatings." Proceedings of the 1990 Coatings for Advanced Heat Engines Workshop, Aug. 6-9, 1990, Castine, Maine, sponsored by U.S. DoE, pp. II-67 to II-76.
16. B.A. Nagaraj and D.J. Wortman, "Burner Rig Evaluation of Ceramic Coatings with Vanadium-Contaminated Fuels." Transactions of ASME, Vol. 112, Oct. 1990, pp. 536-542.

Figure 1. Schematic illustration of a liquid fuel CCVD system using oxygen as an oxidant and argon as a diluent gas.

Figure 2. SEM micrograph of fracture cross section of YSZ coating on a-axis sapphire substrate.

Figure 3. X-ray diffraction pattern of YSZ deposited using liquid fuel CCVD onto a-axis sapphire. Toluene solvent contained 7.5 mol % Y 2-EH & 92.5 mol % Zr 2-EH.

Figure 4. Schematic illustration of a premixed, gaseous fuel CCVD burner incorporating a Bunsen burner.

Figure 5. Mass gain vs. time of CCVD silica-coated and uncoated Ni-20Cr oxidized in air at 1,000 degrees C. Samples were heated at 30 degrees C per minute and attained 1,000 degrees C after about 0.55 hours.

Compatibility of Gas Turbine Materials with Steam Cooling

CONTRACT INFORMATION

Contract Number 94-01-SR022

University University of Central Florida
4000 Central Florida Blvd.
Orlando, FL 32816
Phone : (407)-823-5777
Fax : (407)-823-0208

Principal Investigator Dr. Vimal H. Desai

Research Students Mr. Yakin Patel and Mr. Dnyanesh Tamboli

DOE Project Manager Dr. Norm Holcombe

Period of Performance July, 94 to October, 94

Schedule and Milestones July, 94 to June, 95

BACKGROUND INFORMATION

Gas Turbine Efficiency

Currently, the turbines are designed with Turbine Inlet Temperature (TIT) of 2300° F while the metal temperatures are maintained around 1700° F by using air cooling. For the Advanced Turbine System (ATS) requirements of TIT as high as 2700°F, improved materials, coatings, and advances in cooling and design are warranted. Development of advanced materials with better capabilities specifically for land base applications are time consuming and may not be available by ATS time frame or may prove costly for first generation ATS gas turbines. Therefore improvement in the cooling system of hot components, which can take place in relatively shorter time frame, is important. One way to improve cooling efficiency is to use better cooling agent. Steam as an alternate cooling agent offers many attractive advantages. These can be summarized as under;

- *Its specific heat is twice that of air

- *Steam has lower viscosity than air

- *Steam being a better coolant, it would be possible to reduce size of the cooling lines.

- *The close tip clearance of both compressors and turbine blades would be easier to maintain with efficient steam cooling.¹

- *In Combined Cycle Plants(CCPs), using a bottoming steam turbine, steam can be extracted from steam turbine, used as a coolant and then readmitted to the steam turbine at a higher temperature to obtain additional steam turbine work before being condensed. Thus, a closed system is possible whereby no steam is lost to atmosphere. This leads to maximization of efficiency.²

- *In a closed system, expansion of hot gas in the gas turbine blade path would be improved due to the elimination of mixing

losses of hot gases and ejected cooling air.³

Problems Anticipated in Steam Cooling

Closed system steam cooling can lead to several materials related problems that warrant attention prior to its introduction in ATS. There are four different kinds of problems expected in steam cooling of gas turbines.

1. The most deleterious effect steam can have on the gas turbine materials is the hot corrosion aided by deposition of impurity salts from the steam. The solubility of salts in steam generally increase with increasing temperature and increasing pressure of the steam. As the steam flows through cooling passages, the steam pressure will be dropping which may lead to salt deposition from steam. The salt deposits can lead to enhanced corrosion and oxidation problems. In gas turbine engines, there are many cases reported of hot corrosion due by Na_2SO_4 deposit. In the steam cooling, the problem is severe because of presence of chloride ions. Chloride ions are known to increase propensity for protective oxide films for cracking and spalling. It also causes aluminum and chromium to be rapidly removed from the alloys⁴.

2. Oxidation of metal in steam environment is usually associated with the liberation of hydrogen. The possible embrittlement due to this hydrogen may pose a problem in steam cooling of hot gas components. Intergranular stress corrosion Cracking (IGSCC) in steam environment has been reported for IN-600 alloys⁵.

3. Steam may carry with itself some solid salt particles and some moisture which can cause considerable erosion-corrosion damage in cooling channels. Severe damages

have been reported in steam exposed components of power plants because of erosion-corrosion. Highly localized material erosion which can go up to several millimeters deep can often lead to wall rupture within short time⁶. The erosion-corrosion resistance can be improved by improving the plasticity of oxide layers⁷.

4. In the closed system, steam will be made to pass through cooling passages of vanes and blades. In the so called open closed system, vanes are cooled by steam in a closed system and blades are cooled by air. This latter system was recommended by Arsen'ev et al⁸ as the one to adopt initially. Furthermore to avoid complexity of arranging steam cooling channels, it has been proposed that vane's trailing edge be allowed to discharge some steam for better cooling of this most critical section. The possibility of leakage of steam carry over to the other components can't be overlooked even in the closed system of steam cooling. In this case, one will have to worry about lube-oil systems as some reports indicate that they suffered from adverse effect due to steam injection, a practice followed to lower NO_x level.⁹

OBJECTIVES

Information on steam/materials interaction is mainly available for materials relevant to boilers and heat exchanger tubes. The effects of steam on superalloys are little known. For the continuous reliable operation of gas turbines this information is very vital, so that alternatives can be considered and allowances can be made.

The objectives of this research are;

1. To study how steam affects corrosion and oxidation behavior of blade, vane and transi-

tion piece materials.

2. To study the effect of common impurities in steam on the hot corrosion behavior of the alloys and make a preliminary assessment of its impact on the reliable operation of gas turbines.

3. To study the effect of steam/air intermittent cooling on oxidation and corrosion behavior of superalloys, and

4. To investigate the effect of hydrogen on the integrity of superalloys.

PROJECT DESCRIPTION

The Test Environment

Three different environments have been identified for this proposed work simply classified as "aggressive", "mild" and "clean". These refer to the quality of steam utilized. An aggressive steam environment is utilized in this accelerated test program to determine whether there is a cause for concern regarding long term effect of steam on gas turbine materials. Previous experience with the steam turbine indicate that concentration mechanisms such as deposition from superheated steam, evaporation, drying, concentration in oxides, and concentration in liquid because of a temperature gradient lead to most turbine-corrosion problems. Although the steam conditions in its cooling pathway in a gas turbine may be different from the steam in a steam turbine, occurrence of such deposits in gas turbine can not be ruled out and hence the "aggressive" steam environment is a suitable realistic accelerated test environment. The "aggressive" and "mild" test environment contain primary contaminants such as NaCl and Na₂SO₄ in water.

The Test Materials

Two nickel based and one cobalt based superalloys are being utilized for studying the effect of steam cooling. The Nickel based materials selected are IN 738 and IN 617. IN 738 is a choice material for turbine blade construction, IN 617 is the material used in transition piece. The cobalt based X-45 superalloy is a common material for vanes. The nominal composition of these alloys are given in Table I.

Table I Composition of the Alloys (wt.%)

	IN617	IN738	X-45
C	0.07	0.11	0.25
Mn	-	-	1.0
Si	0.14	-	-
B	-	0.01	0.01
Al	1.0	3.4	-
Ti	0.3	3.4	-
Fe	-	-	2.0
Cr	22	16.0	25.5
Mo	9.0	1.7	-
Co	12.5	8.5	Bal.
Ni	Bal.	Bal.	10.5

It has been proposed by Arsen'ev et al⁶ that taking into account the unavoidable complexity of the design with closed circuit steam cooling of elements in a gas turbine flow section, it is advisable firstly to examine the use of this kind of cooling for the vanes alone, retaining air cooling for the moving blades. Thus X-45 is the most

likely candidate to see steam cooling, followed by IN738. The possibility of steam leakage from the closed loop system or combination with steam injection into combustor will lead to the transition piece getting exposed to steam. Choice of both Ni as well as Co based alloys for testing will indicate inherent differences in their behavior when exposed to steam at high temperatures, if any.

RESULTS AND CURRENT STATUS

The Specimen Geometry and Setup

The specimens were designed in cylindrical form with a central bore of 1/8" diameter and 1.5" long. The superalloys used were very difficult to machine because of their very high strength and toughness. Hence, Electro Discharge Machining (EDM) was used to get the specimens in cylindrical form. Later on, CNC machines were used to machine central bore and put threads on these specimens. Each incoming line of steam is assembled with three specimens, representing each type of test materials. The specimens are connected with threaded IN-600 couplings. The use of ceramic couplings, in order to electrically isolate each specimen, was ruled out because of the large mismatch in thermal expansion properties between ceramics and the superalloys which may lead to cracking of the couplings. In any case, the current due to galvanic couplings would be minimal because of poor conductivity of steam and because the potential difference in galvanic series between these materials is not significant. Two such assemblies have been constructed, one for exposure study and another for EIS (Electrochemical Impedance Spectroscopy) study.

Steam Generation Unit: Design and Fabrication

Two steam generating units were fabricated in-house to produce three different types of steam. The second unit is used as a stand by while the first one is under blow-down and maintenance. The steam is superheated before entering the specimens. The constructed setup is shown in fig. 1. The line diagram of the set up is given in fig. 2. A diaphragm metering pump feeds water into the steam generator. The steam generator consist of three 1/2" OD pipe of IN-600. These pipes are housed inside a 5000 Watts cylindrical radiative furnace at equidistance. As indicated above three different type of steam is produced in each of these IN-600 pipe. Once steam comes out of the steam generator, it is distributed as indicated below and passed through the specimen assembly.

1. Aggressive Steam:
 1. Exposure tests specimens
 2. EIS test specimens
2. Mild Steam:
 1. Exposure test specimens
 2. EIS test specimens
3. Pure Steam:
 1. Exposure test specimens
 2. EIS test specimens

The steam pressure is kept slightly above atmospheric pressure to ensure passage through the specimen. The steam velocity through the sample is maintained about 2 ft/sec. The specimen temperature is maintained suitable to simulate the actual operating conditions.

The salt solubility in steam is too low to differentiate between mild and aggressive steam. Therefore, a separate set up was prepared to get aggressive steam. In this

case, in place of superheated steam, saturated steam is utilized. Saturated steam contains substantial moisture which can dissolve considerable amount of impurity salts. This steam when it gets in contact with hot surface, gets superheated, loses moisture and deposits salts on the surface.

Initial microstructure Evaluation

The initial microstructures of these alloys were recorded for comparisons following exposure studies and also to serve as guide if a second batch of materials is procured for future studies.

Figure 3 shows micrographs of X-45 alloy taken at two different magnifications. The microstructure is typical of a cast cobalt alloy with a high Cr:C ratio. It shows carbide particles in both chunky and fine precipitate form at the grain boundaries. In X-45 most of the carbides formed are of the type $M_{23}C_6$. The chunky carbide particles form while being cooled down during the casting. The lath-like pseudoeutectic structure at the interstices of the dendrite arms is generated by the last liquid to freeze. The fine carbide particles, surrounding the chunky carbides may have resulted from the exposure at high temperature which causes dissolution and re-precipitation of the carbides as very fine particles.

Figure 4 shows the microstructure of IN-738. The original freezing pattern of this cast alloy is clearly visible. Dendrite arms are far apart. Different shadings of dendrite arms are due to inhomogeneities during casting.

Figure 5 shows the microstructure of wrought IN-617 alloy. This alloy was obtained in the form of rolled plate. The

Fig. 1 **Experimental Set-Up**

Fig. 2 **Schematic of Experimental Set-up**

(a)50X

(b) 200X

Fig. 3

Microstructure of X-45: It shows chunky carbide particles surrounded by fine carbide precipitates at the dendritic boundaries. It also shows carbides of lath-like morphology, formed by last freezing liquid during casting.

Fig. 4 **Microstructure of IN-738 (50 X):** It shows large dendrites. The different shadings of the dendrites are due to the compositional inhomogeneities during casting.

Fig. 5 **Microstructure of IN-617 (200X):** It shows a number of deformation twins characteristic of wrought alloys. Since, this alloy is basically a solid solution strengthened alloy, the extent of precipitation is small.

alloy is basically a solid solution strengthened alloy. The microstructure consists mainly of matrix and MC type carbides and a small amount of γ' precipitates. The carbide particles seem to be aligned in rolling direction. A large number of deformation twins, characteristic of wrought superalloys are also visible.

FUTURE WORK

The task for the first three month period, namely literature search, materials procurement, experimental design, fabrication and setup has been accomplished. Steam exposure tests have just begun. The result of the first EIS study is expected shortly and the results of two month exposure study in aggressive steam environment would be available shortly thereafter. The exposure times for mild and pure steam environments are four and six months respectively. Following exposure, the material damage due to steam will be microscopically evaluated. Following paragraphs describe the techniques to be used in this research.

Electrochemical Tests

Electrochemical methods give quantitative measurement of corrosion rate as a function of exposure time, temperature and other environmental parameters. The Electrochemical Impedance Spectroscopy (EIS), a non-destructive technique, is being used for the study. The technique can be employed for ex-situ or in-situ investigation. In this project ex-situ study will be conducted. The EIS, apart from being a non-destructive electro-chemical technique, has the advantage of giving a variety of important system parameters such as polarization resistance, double layer

capacitance and phase shift in a single experiment. These parameters give the information about corrosion rate, mechanism of corrosion, oxide film thickness, film defects and the film integrity. Each specimen will be subjected to the EIS test at the end of every two weeks and the progress of corrosion/oxidation and the effects on the surface oxide will be studied. The first of such EIS tests is due shortly.

Post Exposure Microscopic Evaluations

The specimens exposed to various steam environments will be sectioned and subjected to metallographic evaluation followed by detailed scanning electron microscopic observations for detection and quantification of any localized corrosion, oxidation, cracking, erosion-wear etc. The form of corrosion or damage, in general will be identified. Scanning Auger Microscopy (SAM) and Energy Dispersive Spectroscopy (EDS) will be utilized to isolate the species or contaminant responsible for the damage and understand the mechanism of attack. If required, Transmission Electron microscopy (TEM) of thin sections will be carried out to detect microscopic damage, identify phases, corrosion products and diffusion kinetics. In general, detailed metallography will be performed on polished radial cross-sections to assess microstructural damage of each specimen. Where appropriate, depth profile elemental concentration will be obtained using the SAM's sputter gun. Specimens that are undergoing electrochemical evaluation will be observed at the end of their exposure period under metallograph and SEM and evaluated by EDS as well as SAM in a manner similar to the specimens undergoing exposure study.

REFERENCES

1. Rice, I.G. "Steam Cooled Gas turbine Casings, Struts, and disks in a Reheat Gas Turbine Combined Cycle: Part I-Compressor and Combustor", Transactions of ASME, Journal of Engineering for Power, Vol. 105, October 1983, pp 844-8503.
2. Rice, I.G. "Steam Cooled Gas turbine Casings, Struts, and disks in a Reheat Gas Turbine Combined Cycle: Part II-Gas Generator Turbine and Power Turbine", Transactions of ASME, Journal of Engineering for Power, Vol. 105, October 1983, pp 851-858
3. Jericha, H. and Hoeller F. "Combined Cycle Enhancement", Transactions of ASME, Journal of Engineering for Gas Turbine and Power", Vol.113, April 1991, pp 198-202
4. Pettit, F. S. and Giggins, C. S. "Hot Corrosion" in Superalloys II, Eds. C.T.Sims, N.S.Stoloff and W.C. Hagel, Wiley, New York, 1987, chap.12
5. Shen, C. H. and Shewmon, P. G., "A mechanism for hydrogen induced stress corrosion cracking in alloy 600", Metallurgical Transactions, Vol. 21A, May 1990, pp 1261-1271
6. Kunze, E. and Nowak, J. "Erosion corrosion damage in power plants", Werkst.Korros, Vol 33, May 1982, pp 262-273
7. Stephenson, D.J. Nicholis, J.R. and Hannock P. " Erosion of gas turbine blade materials by solid sea salt", Corrosion Science, Vol. 25, October 1985, pp 1181-1192
8. Arsen'ev, L.V. Korsov, Yu. G., Khodak, E.A. and Romankhova, G.A. "A Highly efficient combined cycle plant with steam-cooled gas turbine", Thermal Engineering, Vol. 37, March 1990, pp111-114
9. Bretz, E..A. "Examine full impact of injecting steam into gas turbine systems", Power, June 1989, pp 53-59

CONTRACT INFORMATION

Project Title: The role of reactant unmixedness, strain rate and length scale on performance and emissions in a combustion system.

Contract No.: 94-01-SR020

Contractor: University of California, Irvine
 UCI Combustion Laboratory
 Irvine, CA 92717-3550
 Ph: (714) 824-5950
 Fax: (714) 824-7423

Principal Investigator: G. Scott Samuelson

Co-Investigator: John C. LaRue

Research Students: Suresh Vilayanur, Darrell Guillaume

DOE Project Manager: Paul Micheli

Period of Performance: June 1, 1994 to September 30, 1994

Schedule and Milestones:

Name	Q3 '94				Q4 '94			Q1 '95			Q2 '95			Q3 '95	
	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug
FIRST YEAR															
Easily Controlled Premixer															
Model Premixer															
Model Combustor															
Non Reacting Tests															
Reacting Tests															
Premixed Combustor Operation															
DIAGNOSTICS															
SURVEY															
ANALYSIS															
Experimental Review															
Paper Review															

OBJECTIVES

Lean premixed combustion provides a means to reduce pollutant formation and increase combustion efficiency (Sattelmayer et. al., 1990). However, fuel-air mixing is rarely uniform in space and time. This nonuniformity in concentration will lead to relative increases in pollutant formation and decreases in combustion efficiency. The nonuniformity of the concentration at the exit of the premixer has been defined by Lyons (1981) as the "unmixedness." Although turbulence properties such as length scales and strain rate are known to effect unmixedness, the exact relationship is unknown. Evaluating this relationship and the effect of unmixedness in premixed combustion on pollutant formation and combustion efficiency are an important part of the overall goal of US Department of Energy's Advanced Turbine Systems (ATS) program and are among the goals of the program described herein. The information obtained from ATS is intended to help to develop and commercialize gas turbines which have (1) a wide range of operation/stability, (2) a minimal amount of pollutant formation, and (3) high combustion efficiency. Specifically, with regard to pollutants, the goals are to reduce the NO_x emissions by at least 10%, obtain less than 20 PPM of both CO and UHC, and increase the combustion efficiency by 5%.

The contributions to the program which the University of California (Irvine) Combustion Lab (UCICL) will provide are: (1) establish the relationship of inlet unmixedness, length scales, and mean strain rate to performance, (2) determine the optimal levels of inlet unmixedness, length scales, and mean strain rates to maximize combustor performance, and (3) identify efficient premixing methods for achieving the necessary inlet conditions.

To understand unmixedness, the initial part of the program addresses the definition and measurement of unmixedness. The heart of the program is anchored on three experiments -- one to study premixing strategies; a second controlled study to address the effect of unmixedness, turbulent length scale, and turbulent strain rate on mixing and pollutant formation; and a third to study the effect of premixing strategies on combustor performance.

BACKGROUND

Oxides of nitrogen (NO_x) refer to nitric oxide (NO) and nitrogen dioxide (NO_2). Air quality is significantly impacted by NO_x emissions since they directly or indirectly affect concentrations of nitrogen dioxide (NO_2), inhalable particulate (PM10) and photochemical oxidant (including ozone and peroxyacylnitrates). Furthermore, the NO_2 and PM10 formed from the emission of NO_x decrease visibility and lead to nitric acid deposition. Concerns have been expressed about the potential adverse health effects of other nitrogen compounds, such as nitrosamines, and nitrous oxide (N_2O).

Several strategies are available for reducing NO_x emissions from gas turbines. Most notable are selective catalytic reduction (SCR), exhaust gas recirculation (EGR), steam injection, and retrofitting with low- NO_x combustors (LNC). The key to these low- NO_x combustors is either a combination of staged combustion air and staged fuel or partially premixing the fuel and air. The level or degree of premixing required to effectively manage a low- NO_x combustor is still to be determined. Specifically, the local fuel-air unmixedness inside the premixer and inside the combustor will affect the combustor's performance and, therefore, the effectiveness.

In a practical gas turbine, the fuel-air mixing is rarely uniform in time or space. Consequently, large deviations (i.e., fluctuations) about the mean local temperature can occur as packets of various mixtures of fuel and air pass through a point in space inside the combustor. These variations are not only observable as fluctuations in the flame luminosity; they also play a significant role in the generation of NO_x . Decreasing the amplitude of the temperature has been shown in simulations to have a significant effect on NO_x production. Understanding how to control the local turbulence to optimize the local unmixedness will aid in the design of more effective and less bulky fuel-air premixers.

Unmixedness is a quantity which is used to indicate, in a statistical sense, the degree of mixing of fuel and air at the molecular level. In a given system in which two or more streams of miscible fluids are mixing, a specific level of molecular unmixedness (grain size of fuel-and-rich parcels of fluid) depends on initial length scales, initial distribution of fuel-rich and air-rich parcels of fluid, the mean strain rate field, and the characteristics of the turbulent field (e.g., the intensity, the large structure, the degree of anisotropy, etc.). Two means are most frequently used to determine the statistical measure of unmixedness. Lyons (1981) introduces one which also requires the definition of the area averaged equivalence ratio, $\bar{\Phi}$, across the profile of the flow which in her case is a pipe of radius r . Thus:

$$\bar{\Phi} = \frac{\int_0^{r_{\max}} \Phi_1 r dr}{\int_0^{r_{\max}} r dr}$$

where, Φ_1 is the equivalence ratio at a particular radial location. Using this value for the area averaged equivalence ratio, an area averaged measure of unmixedness, S , can be defined as the

area averaged standard deviation. This standard deviation can be expressed as follows:

$$S = \left(\frac{\int_0^{r_{\max}} (\Phi_1 - \bar{\Phi})^2 r dr}{\int_0^{r_{\max}} r dr} \right)^{\frac{1}{2}}$$

The second measure of unmixedness (cited by Fric, 1992) is developed by considering temporal fluctuations in the fuel concentration. The level of unmixedness, U , at a particular position in the flow can be defined as a function of the mean fuel concentration, $\bar{C}$, at a location and the variance fluctuation about the mean, c^2 , as:

$$U = \frac{\overline{c^2}}{\bar{C}(1 - \bar{C})}$$

This equation provides a statistical measure of the unmixedness due to temporal fluctuations while the preceding equation provides a measure of the unmixedness due to spatial variation. Both of these equations are representative of the unmixedness in a statistical sense. However, neither of these equations indicate the flow structure that leads to a particular value of unmixedness. For example, the same unmixedness could be obtained with a small number of large fuel- and air- rich eddies or a large number of smaller fuel- and air- rich eddies. Thus a complete determination of the unmixedness requires not only a statistical measure of the unmixedness but also a structural measure of the unmixedness, i.e., a measure of the statistical distribution of the length scales of fuel rich and air rich regions.

The structural measure of unmixedness can be obtained by first, forming indicator functions from the concentration signal and second determining the distribution of the length scales of fuel-rich and air-rich fluid parcels from those indicator

functions. (Indicator functions are signals that have either zero or unity values depending on whether the concentration signals are, respectively, above or below a selected threshold.

In order to illustrate the formation of indicator functions for the fuel- and air- rich parcels of fluid, consider the distribution of eddies shown in the top part of Fig. 1. The fuel- and air- rich parcels of fluid are indicated respectively, by the letter "F" and "A". If this distribution of fluid parcels were convected by a concentration sensor with adequate spatial and temporal resolution, the time resolved concentration signal, shown in Fig.1, would be obtained. That signal would have high and low values corresponding to the fuel- and air- rich fluid parcels. The indicator function, $I_{c_{f,t}}$, which has unity values for the fuel-rich parcels could be formed by setting the threshold $c_{f,t}$, to slightly below the concentration of the fuel-rich parcels. The duration of the unity portion of the indicator function could be obtained using appropriate software. Using a threshold, $c_{a,t}$, which is chosen to be slightly above the concentration of air parcels, the indicator function, $I_{c_{a,t}}$, would be formed. The duration of the zero values of this indicator function would correspond to the length of the air-rich parcels. The statistical variation of length scales of fuel- and air- rich parcels can be determined from the indicator function in a straight forward manner.

PROJECT DESCRIPTION

Although lean premixed combustion does lead to reduced production of NO_x , and increased efficiency of gas turbine combustors, the minimal unmixedness required to achieve adequate combustor performance, or the point at which further mixing is ineffective are unknown. Thus, a relationship that describes combustor emissions and efficiency as a function of unmixedness is required. Specifically, since turbulence properties

such as length scales, mean strain rate and other turbulent properties are known to effect unmixedness, the goals of this study are to determine this relationship and then use the results of this study to optimize the performance of gas turbine combustors.

Mixing devices and turbulence measurement techniques will be used to obtain spatial and temporal unmixedness, as defined in the background section, for a wide range of initial fuel and air length scales and mean strain rates and intensities. With unmixedness quantified with respect to the turbulence properties, these properties can be used within premixed systems to control unmixedness. The data obtained from experiments within these reacting systems will be used to determine a relationship between unmixedness and combustor performance (efficiency, pollutant formation, stability, lean blow-out).

APPROACH

The program has three objectives. The first objective is to establish the relationship of inlet unmixedness, length scales, and mean strain rate to combustor performance. The second objective is to determine the optimal levels of inlet unmixedness, length scales, and mean strain rates that maximize combustor performance. The third is to identify efficient premixing methods for achieving the necessary inlet conditions. These three objectives are directly related.

Task 1 - Experimental Design and Development

The objective of this task is to identify the facility needs of the project and to design experiments that best meet these needs. This section contains a description and purpose of the experiments. In addition, alterations to existing facilities are described. This task is subdivided into Premixing

Experiments, a Controlled Reactor Experiment, and Combustor Experiments.

Premixing Experiments

Two premixing experiments have been developed to assess two simple premixing devices. The first is a configuration that allows controlled mixing tests to be performed on a mixing cone as shown in Fig. 2. The second is an experiment in which the relationship between premixer length and unmixedness can be investigated. Specifically, the facility allows the length, and therefore time, to be changed between the point of scalar injection to the point of measurement (Fig. 3). Studies will be performed that determine the effects of this length parameter on unmixedness.

Controlled Reactor Experiment

A simple reactor which permits easy and independent control of length scales, strain rate, and unmixedness has been designed. It can operate both non-reacting and reacting. As such, it is an effective "bridging" experiment between the premixer and combustor experiments. A similar configuration has been described by Kostiuk, et. al. (1993). However, these studies did not involve variation of the unmixedness length scale. This configuration can be seen in Fig. 4 and is labeled "Controlled Reactor." The strength of the Controlled Reactor lies in that all of the turbulence properties of the flow can be controlled independently. Table 1 lists this reactor and a model gas turbine combustor which has the essential features of a practical combustor which will be discussed below. The table discusses the ability of each to control each necessary parameter. For the Controlled Reactor, the air length scales are controlled by the grid spacing, the gas length scales are controlled by the jet grid hole diameter, the unmixedness is controlled by the distance between the grid and the exit of the tube, and the mean strain rate is controlled by the

distance between opposing jets. Once this experiment has been used in a non-reacting flow to determine length the scales-mixing relationship, it will be used to explore the relationship between length scales/unmixedness/strain rate and pollutant formation.

Combustor Experiments

The combustor design is fundamentally an axisymmetric can combustor. This configuration was chosen since it lends itself to easy implementation of most diagnostic techniques, control of unmixedness/length scales/strain rate, and has the essential features of a practical gas turbine combustor. Combustor #1 controls many of the turbulence properties that the non-reacting study has deemed important (Fig. 5). Gas length scales are controlled by altering the diameter of the gas injection tubes. Air length scales are controlled with a upstream grid. Mean strain rate is controlled by changing the swirl vanes while the degree of unmixedness is controlled by altering the distance between gas injection and reaction zone.

Task 2 - Experimental Studies

To date, preliminary Premixer Experiments have been conducted.

One of the most obvious goals of the grid injection system is to produce equal injection at each node point. Unfortunately, the very nature of the design leads to different lengths of tubing prior to gas exit at various positions (Fig. 6). This variable length leads to variable headloss. At equal pressure input, the flow output across the grid will vary. Although individual valves and regulators at each node point would solve the problem, a more practical solution exists. As shown in Fig. 7, a configuration that has equal tube lengths at each node is used to obtain uniform injection velocity. The size of the storage chamber compared to the tube diameter easily allows for the uneven length

at the injection end. Fig. 8 shows the velocity at the grid output for traditional grid injection and equal tube length injection. There it can be seen that the velocity at the exit of each injection tube is nearly equal. For this reason, the system shown in Fig. 7 will be used for the premixer in Combustor #1.

The heart of the any premixing device is the premixer configuration. A simple mixing cone has been chosen for the preliminary premixing experiments. This cone can be configured in two ways. Either the input ports in the cone are tangential to the cone periphery causing swirl, or the inlet ports are radial towards the center causing impingement. Both are known means of enhancing mixing though they have a large pressure drop and will not be used in subsequent tests. Preliminary experiments have been performed to investigate mixing in the cone geometry in both cases. Temporal and spatial unmixedness downstream of these two simple premixers have been determined. Heated and ambient air were used as the working fluids and spatially and temporally resolved temperature measurements are used to determine spatial and temporal unmixedness. The optimal configuration of the cone has not been determined. For this reason, tests with geometric variations in the cone configuration will be performed.

Task 3 - Data Analysis

Thermocouples and cold-wire probes have been used. Baseline measurements shown in Fig. 8 were obtain with no cone. As expected, there was variation in the mean temperature. With the cone in place, variation in the mean temperature was reduced to 1/7 of the value with no cone present. Further, Fig. 9 shows that the radial temperature distribution is flattened at the cone exit and has a negligible change with downstream distance. To compliment these spatial experiments, cold wire measurements were obtained. As expected, the

preliminary data show that the root mean square (RMS) of the temperature was much lower with the cone. Thus, consistent with the reduction in spatial unmixedness, the temporal unmixedness is reduced.

Task 4 - Industrial Interaction

The first stage in the industrial interaction is the generation of a needs survey. The propose of this questionnaire is three fold. The first goal is to determine practical issues or concerns that the industry has regarding premixing. The second is to determine the general specifications (e.g., geometry, energy release) for a research combustor. The final goal is to determine the most useful form of the experimental data to aid CFD studies. The types of experiments that are most useful and the immediate needs of the finite element audience will be identified

FUTURE WORK

Below is a brief description of the plans for the immediate future plans for the program.

Task 1 - Experimental Design and Development

Although the simple test fixtures of Figs. 3 and 4 will continue to yield preliminary data relative to the unmixedness relationship, the Controlled Study Reactor (Fig. 2) is needed so that the relationship between unmixedness, combustor efficiency, and combustion emissions can be determined. Hence the next step is to begin construction of this reactor.

Since fabrication of the model gas turbine combustor's (Combustor #1) premixing components are nearing completion, initial studies of the relationships between unmixedness, pollutant formation and efficiency will soon

commence in Combustor #1. After some initial tests, the combustor will continue to be modified to allow better control of the unmixedness.

Task 2 - Experimental Studies

The Premixing Experiments will continue using temperature as a scalar indicator and thermocouples/cold-wires as a means to assess premixer performance. A PLIF system will be developed for use in both reacting and non-reacting flows for the measurement of unmixedness. Finally, emissions measurement equipment will be used to evaluate the pollutant formation (NO_x , CO, etc.) in the premixed model gas turbine combustor (Combustor #1).

Task 3 - Data Analysis

Further analysis of the data obtained using the cone premixer will be performed to determine length scales. The length scales and unmixedness at the exit of the model gas turbine combustor will be measured. Finally, an inter-comparison will be made between PLIF measurements and those obtained using cold-wire probes.

Task 4 - Industrial Interaction

The major next goal of this task is the distribution of the survey and the start of the analysis of its findings.

ACKNOWLEDGMENTS

This report was prepared with the support of the US Department of Energy, Morgantown Energy Technology Center. Contract number 94-01-

SR020. This financial support through the South Carolina Energy Research and Development Center (Daniel Fant, Research Manager, and Lawrence P. Golan, SCERDC Director) is greatly appreciated. Students who have participated on this project include Graham W. Clark and Keita Enjoji. This project is also funded in part by the Southern California Gas Company.

REFERENCES

Fric, T. F., Effect of Fuel-Air Unmixedness on NO_x Emissions, AIAA Paper No. 92-3345, (1992).

Kostiuk, L. W., Bray, K. N. C., and Cheng, R. K., Experimental Study of Premixed Turbulent Combustion in Opposed Streams. Part I -- Non-reacting Flow Field, Combustion and Flame, Vol. 92, pp. 377-395 (1993).

Kostiuk, L. W., Bray, K. N. C., and Cheng, R. K., Experimental Study of Premixed Turbulent Combustion in Opposed Steams. Part II -- Reacting Flow Field and Extinction, Vol. 92, pp. 396-409 (1993).

Lyons, V. J., Effect of Fuel-Air-Ratio Nonuniformity on Emissions of Nitrogen Oxides, NASA Technical Paper No. 1798, (1981).

Sattelmayer, Th., Felchlin, M. P., Haumann, J., Hellat, J., and Styner, D., Second Generation Low-Emission Combustor for ABB Gas Turbines: Burner Development and Test at Atmospheric Pressure, AIAA Paper No. 90-GT-162 (1990).

Fig 1. Formation of Indicator Function.

Fig 2. (Mixing Experiment #1) Schematic showing the test fixture designed to measure and analyze the mixing performance of the various mixer configurations required for Combustor #1.

Fig 3. (Mixing Experiment #2) Schematic showing the test fixture designed to measure and analyze the relationship between unmixedness and distance from reaction zone required for Combustor #1.

Fig 4. (Controlled Reactor) Schematic of a research combustor that permits independent and controlled variation of the length scales of the two gases and the mean strain rate. Length scales are adjusted by varying the grid spacing, the mean shear stress is adjusted by changing the distance between opposing jets, and the degree of unmixedness is adjusted by altering the position of the grids within the jet.

Fig 5. (Combustor #1) Schematic of fully controllable practical premixed combustor. Length scales are set by the diameter of the gas inlet tubes and an upstream air grid, mean shear stress is adjusted by changing the swirl angle, and the degree of unmixedness is changed by changing the distance between gas tube outlet and reaction zone.

Fig 6. Schematic of test fixture used to measure velocity output of single-tube, multiple port, gas injector. Configuration simulates a simple injection system that utilizes a grid as a supply source for the gas.

Fig 7. Schematic of test fixture used to measure velocity output of multiple-tube, single port, gas injector. Configuration simulates an injection system that allows equal flow velocity at every node location.

Fig. 8 Velocity versus hole position of multi-tube injection system and single-tube injection system.

Fig. 9 Normalized temperature at cone exit and one diameter downstream from cone exit with and without cone.

Table 1. Means of control of turbulence properties for various combustors.

	Strain Rate Scale	Unmixedness	Air Length/Time Scale	Gas Length/Time	Notes
Controlled Reactor	Change L	Change M	Vary Mesh Spacing of Grid	Vary Mesh Spacing of Grid	Comparison Data from Previous Studies
Combustor #1	Change Vane Gap	Change Distance from Exit of Gas Injection Tube		Change the Number of Tubes	Promising Research Combustor

Advanced Concepts for High Efficiency and Low NO_x Gas Turbine Combustor Development

CONTRACT INFORMATION

Contract Number 94-01-SR028

Contractor University of Maryland
Department of Mechanical Engineering
College Park, Maryland 20742
(301) 405-5276
Fax: (301) 314-9477

Principal Investigator Professor Ashwani K. Gupta*

Co-Investigators Professor Adeboyejo A. Oni[†]
Professor Mark J. Lewis[§]

Research Student Martin Zhu[‡]

DOE Project Managers Dr. Paul Micheli
Dr. Norm Holcombe

Period of Performance August 15, 1994 - August 14, 1995

ABSTRACT

This research program is aimed at developing an understanding of techniques for improved fuel-air mixing, with the specific focus of developing advanced concepts that will allow combustor operation near the stoichiometric limit, while at the same time minimizing thermal NO_x and enhancing efficiency. Various mixing geometries will be investigated both experimentally and

computationally, to seek a thorough understanding of fuel/air mixing mechanisms that incorporate high levels of straining, yet produce good flame stability and low emissions.

This project will be conducted in collaboration both Pratt&Whitney, East Hartford, Connecticut, and the General Electric Research and Development Center in Schenectady, New York. Both will provide guidance and technical support throughout the program, and will be seeking to apply results of this program to practical combustor development.

The ultimate product will be an understanding of the most desirable combustor geometries, air swirl, and fuel nozzle injector geometries to provide high combustion efficiency with low associated NO_x emissions.

*Professor, Dept. of Mechanical Engineering, University of Maryland

[†]Associate Professor, Morgan State University

[§]Associate Professor, Dept. of Aerospace Engineering, University of Maryland

[‡]Graduate Research Assistant, Dept. of Aerospace Engineering, University of Maryland

Although this work will be directed towards using natural gas and propane gas, as the fuel, subsequent investigations may examine alternative fuels and other hydrocarbon fuels.

BACKGROUND INFORMATION

An important goal for the improvement of gas turbine power generation technology, using natural gas fuel, is the increasing of overall system efficiency while at the same time minimizing deleterious emissions such as NO_x . The achievement of higher combustion efficiency has generally been accomplished by driving the operating pressure and temperature to higher values, which promotes the formation of thermal NO_x via the reaction of atmospheric nitrogen and oxygen. As such, there is a difficult tradeoff between efficiency enhancement and emissions reduction, and the development of gas turbines with both high efficiency and low NO_x poses an important challenge.

Studies carried out at the University of Maryland¹ and elsewhere reveal that combustor geometry, fuel nozzle injector design, swirl strength, radial distribution of swirl, and fuel-air distribution have strong influence on the combustion efficiency and intensity of NO_x emission levels. In addition, introducing the reactants into a combustor with a swirling motion is known to exert a significant effect on the resulting combustion characteristics in terms of heat release rate, combustion efficiency, and flame stability. Most gas turbines utilize swirl for these characteristics.

It is also known that combustion with swirl can have a marked effect on the nitric oxide (NO) and nitrogen dioxide (NO_2) produced, both in terms of the total quantities and the ratio between the two gases^{2,3}. Previous data from swirl-stabilized flames reveals evidence of the destruction of thermal NO in the internal toroidal recirculation zone, despite the high temperatures associated with this zone. The recirculation zones can be either fuel-rich or fuel-lean, and, in both cases, NO destruction has been observed.

Swirl-generated and bluff-body recirculation is an important feature of gas turbine engine combustors. It is important to determine and quantify parameters which affect efficiency and the extent of NO-reduction, and to relate the net production of NO/ NO_x to conditions existing in the recirculation zone. These in turn must be linked to the design parameters of the circulation-producing device. In addition, it is important to develop an improved chemical understanding of the dominant mechanisms that exist within the recirculation zone(s). The role of thermal stratification in fuel-rich and fuel-lean swirling zones also deserves examination. In addition, the role of suddenly-strained thin fuel shear layers, fluid dynamic development, turbulent mixing, reaction rate and fluid entrainment at higher overall equivalence ratios (close to stoichiometric) requires study.

All of these quantities affect flame stability, combustion, and emission characteristics. Therefore, questions to be addressed here are: (i) how does a reacting shear layer develop, (ii) what does sudden straining of the layer do to this development, combustion intensity, efficiency, and flame stability, and (iii) what is the most effective strategy to strain the layer? Answers to these questions are relevant for the swirling recirculation zones and other previous concepts used, such as air staging, fuel staging, and the individual simple fuel injectors required.

PROJECT DESCRIPTION

This research program has the following objectives:

- Examine the role of thermal stratification in fuel-rich and fuel-lean swirling flows with special reference to internal recirculation.
- Determine the extent of apparent NO-reduction effects in the recirculation zone and relate the net NO/ NO_x production to the chemical and thermal conditions that exist in the recirculation zone. These can be related to the flow dynamics and swirler design that produced

the internal toroidal recirculation zone. It is, for example, possible that the presence of certain species yields higher reduction of NO, depending on the mechanism pathways. Attempts will be made to provide an improved understanding of the chemistry that exists in the recirculation zone. This will then provide better insight for the purposes of developing high efficiency and low NO_x combustors in addition to providing a data-base for the modeling of such flows.

- Study the effect of a suddenly-strained shear layer (as obtained by swirl to the flow or via the use of other nozzle exit geometries) on the fuel/air mixing, flame stability, combustion efficiency and the emission of chemical species including NO_x, CO, and unburned hydrocarbons (UHC).

A double concentric swirler will be used to systematically examine the effect of swirl and nozzle injector geometry on the subsequent flowfield (including recirculation zone size and strength) and combustion and emission characteristics. Parallel computational studies will be carried out on this geometry.

The practical experimental configuration will use an available variable geometry combustor with optical accessibility. Work will focus on an examination of the fuel and air staging, and subsequent role of thermal stratification in both fuel-rich and fuel-lean swirling flows, with special reference to internal recirculation. Efforts will be directed towards determining the extent of associated NO_x reduction in the recirculation zone, relating net production of NO_x to chemical and thermal conditions that exist in that region. Both experimental measurements and computational analysis will be employed to characterize this flowfield.

For mixing geometries, several candidate devices will be studied, incorporating inert gas injection for swirl inducement, as well as a variety of strain-inducing concepts such as boundary layer trips, vortex generators, trailing edges with a bluff body or simple chutes. The effect of a resulting suddenly-strained shear layer on the fuel/air mixing process, flame

stability, overall combustion efficiency, and generation of undesirable species will be examined. These effects will be correlated to the various flowfield variables, including pressure, temperature, density, and local velocities, using computational solutions matched to the chosen geometries.

FUTURE WORK

The first phase of this work will begin with the design and fabrication of an optically-accessible swirl combustor. This will be a double concentric design, and will be used to examine the detailed flow and reaction structure under conditions of isothermal air flow. In parallel to that effort, instrumentation will be developed which will include flow visualization and both intrusive and non-intrusive laser diagnostic techniques. Experiments will determine the mean velocity and turbulence characteristics of the flow, which will be matched to computational results. Gases of various densities will also be used to conduct thermal stratification experiments in the combustor, along with zero heat release reaction rate studies. Combustion studies will follow upon completion of the isothermal mixing studies.

An important goal will be to use the computational results to understand and guide the experimental results, and thus reduce the required test matrix. The General Aerodynamic Simulation Program (GASP) and Fluent codes will both be available for this effort. Fluent is an especially promising code because of its incorporation of detailed chemistry, and its previously demonstrated effectiveness in calculating properties in combusting systems.

A subsequent phase of this program will focus on the most promising fuel shear layer straining devices, which will be examined under combustion conditions using either natural gas or propane as the fuel. Experiments will determine the mean flow velocities and turbulence characteristics in these fuel shear layer straining devices. Diagnostic techniques developed for the combustor in the first phase of this effort will be applied to this work. Thin-film

pyrometry, two-color optical pyrometry, and electronically-compensated high-frequency thermometer measurements will be used as well. These techniques will provide thermal mapping of the flames.

Measurements will determine local concentrations of NO and NO_x in the flames, as well as species concentrations of CO, CO₂, O₂, UHC, HCN, and NH₃ in the flames. As in the first phase, computational modeling will be used to provide further understanding of these experiments, and to reduce the parametric variations studied. This will require the refinement of chemically-reacting models in the computational solutions.

A third and final phase of this program will incorporate the existing Variable Geometry Swirl Combustor (VGC) in the Combustion Laboratory at the University of Maryland in studies of the distribution of species concentrations in the most promising flames. Advanced fuel injection configurations that have shown the most potential for high combustion efficiency with low overall NO_x emissions will be used in this effort. An emphasis will be placed on the examination of the upstream portion of the flame, near the fuel nozzle exit, where mixing processes will have a strong influence on flame stability, combustion characteristics, and on the pollutant emissions. The results of the first two phases will be applied to this study.

Specific activities in this portion of the effort will begin by providing the desired optical access and installation of instrumentation in the Variable Geometry Swirl Combustor. Once this is accomplished, detailed species concentrations will be measured within selected flames. As in the earlier work, measurements will be made of NO and NO_x, as well as CO, CO₂, O₂, UHC, HCN, and NH₃ in each of these flames. Detailed studies will examine the effect of radial swirl distribution and the various fuel shear layer straining devices on NO_x production, flame stability, and combustion characteristics. As before, computational efforts will support this work.

REFERENCES

¹ Gupta, A.K. and Taha, M.R., "The Effect of Nozzle Geometry on the structure of Flames and NO_x Emission," AIAA Paper 92-0766, presented at the AIAA 30th Aerospace Sciences Meeting, Reno, Nevada January 1992.

²Ramavajjala, M., and Gupta, A.K. "Swirl Effects on NO and NO₂ Emissions in a Variable Geometry Combustor," AIAA Paper 91-0643, presented at the 29th AIAA Aerospace Sciences Meeting, Reno, Nevada, January 1991.

³Gupta, A.K. and Ramavajjala, M., "The Role of Swirl on NO and NO₂ Emission and Combustion Characteristics," Proc. of the Tenth International Symposium on Air Breathing Engines, Nottingham, U.K., Sept. 1-6, 1991, pp. 258-266.

Appendices

Agenda

Advanced Turbine Systems Annual Program Review
Crystal Gateway Marriott
Arlington, Virginia
November 9-11, 1994

Agenda (Continued)

Day 1 - Wednesday, November 9, 1994 (Continued)

Session 2: Program Element Overviews - Grand Ballroom Salon D/E

Session Chair: Charles M. Zeh, METC

1:30 pm	Gas Turbine Power Generation Evolution
Advances for the Future	<i>James C. Corman</i>
General Electric
1:50 pm	ATS Program Element Overview--
Allison Engine Company	<i>Jimmy L. Lunsford</i>
Allison Engine Company
2:10 pm	Overview of Westinghouse's Advanced Turbine
Systems Program	<i>Ronald L. Bannister</i>
Westinghouse Electric Corp.
2:30 pm	Gas Fired Advanced Turbine System	<i>David A. Rohy</i>
Solar Turbines Incorporated
2:50 pm	Break	
3:05 pm	METC Combustion Research Facility	<i>Jack Halow</i>
METC
3:25 pm	A Collaborative Vision: The Advanced Gas
Turbine Systems Research Program	<i>Daniel Fant</i>
South Carolina Energy R&D Center
3:45 pm	Materials/Manufacturing Support Element for
the Advanced Turbine Systems Program	<i>Michael Karnitz</i>
Oak Ridge National Laboratories (ORNL)
4:00 pm	Break	
4:20 pm	Panel Discussion: Distributed Power	Moderator: <i>Keith Davidson</i>
On-Site Energy
	Panel Members	
	<i>Eric Wong</i>
California Energy Commission	
	<i>David Eppinger</i>
Fluor Daniel, Incorporated	
	<i>Dan Whitney</i>
Sacramento Municipal Utility District	
	<i>Malcolm Jacobson</i>
Enron Emerging Technologies	
5:30 pm	Adjourn	
5:30 pm	Reception - Grand Ballroom Salon B
Sponsored by the Gas Turbine Association	

Advanced Turbine Systems Annual Program Review
Crystal Gateway Marriott
Arlington, Virginia
November 9-11, 1994

Agenda (Continued)

Day 2 - Thursday, November 10, 1994

Session 3: ATS Program Technical Review - Grand Ballroom Salon D/E

Session Chairs: Steve Waslo, DOE, Chicago
Ed Parsons, METC

9:00 am	Phase 2 Development Projects Advanced Turbine System Program	<i>Ihor S. Diakunchak</i>
Westinghouse Electric Corporation
9:25 am	Technologies for Next Generation Power Systems	<i>Harold E. Miller</i>
General Electric
9:50 am	Gas Fired Advanced Turbine System	<i>Karl Karstensen</i>
Solar Turbines Incorporated
10:15 am	Break	
10:30 am	ATS Turbine Modification for Coal and Biomass Fuels	<i>Richard Wenglarz</i>
Allison Engine Company
10:55 am	Ceramic Retrofit Project	<i>Mark van Roode</i>
Solar Turbines Incorporated

Session 4: ATS Project Related Activities - Salon D/E

Session Chairs: Leland E. Paulson, METC
Dr. Michael A. Karnitz, ORNL

11:20 am	Coatings for High Temperature Structural Materials - NMAB Study	<i>Robert V. Hillery</i>
General Electric Aircraft Engines
11:45 am	Long Term Testing	<i>Matt Ferber</i>
ORNL
12:10 pm	Luncheon - Arlington Ballroom Salon V	
1:30 pm	Dry Ultra-Low Emissions Combustion System for Allison's Industrial Engines	<i>Mohan Razdan</i>
Allison Engine Company
1:55 pm	Investigation of Combustion Instability in Lean-Premix Combustion	<i>George A. Richards</i>
METC
2:20 pm	High Performance Steam Developments	<i>Tom Duffy</i>
Solar Turbines Incorporated
2:45 pm	CFCCs for Turbine Applications	<i>Rick Lowden</i>
ORNL
3:10 pm	Turbine Airfoil Manufacturing Technology-- Land Based Turbine Casting Initiative	<i>Michael Karnitz</i>
ORNL
3:35 pm	Break	

Advanced Turbine Systems Annual Program Review
Crystal Gateway Marriott
Arlington, Virginia
November 9-11, 1994

Agenda (Continued)

Day 2 - Thursday, November 10, 1994 (Continued)

Session 5: University/Industry Consortium Project Review - Grand Ballroom Salon D/E
Session Chair: Lawrence P. Golan, South Carolina Energy Research and Development Center

3:55 pm Materials Session

Functionally Gradient Materials for Thermal *Arnold R. Marder*
Barrier Coatings in Advanced Gas Turbine Systems Lehigh University

Life Prediction of Advanced Materials for Gas *Sam Y. Zamrik*
Turbine Application Penn State University

Reduction of Turbine Endwall Total Pressure Loss *Ronald S. LaFleur*
and Heat Transfer Using the Iceformation Clarkson University
Design Method

4:55 pm Heat Transfer Session

Vortex-Generator Induced Enhanced Heat *Sumanta Acharya*
Transfer in Gas Turbine Coolant Channels Louisiana State University

Advanced Turbine Cooling, Heat Transfer, and *Je-Chin Han*
Aerodynamics Studies Texas A&M University

5:45 pm Adjourn for the Day

Advanced Turbine Systems Annual Program Review
Crystal Gateway Marriott
Arlington, Virginia
November 9-11, 1994

Agenda (Continued)

Day 3 - Friday, November 11, 1994

Session 5: University/Industry Consortium Project Review (Continued) - Arlington Ballroom Salon I

Session Chair: Lawrence P. Golan, South Carolina Energy Research and Development Center

9:00 am	Combustion Session	
	Emissions Models for Lean Premixed Turbines	<i>Arthur M. Mellor</i> Vanderbilt University
	Combustion Modeling in Advanced Gas Turbine Systems	<i>Paul O. Hedman</i> Brigham Young University
	Lean Premixed Combustion Stabilized by Radiation Feedback and Heterogeneous Catalysts	<i>Robert W. Dibble</i> University of California, Berkeley
	NO _x Reduction in Advanced Gas Turbines	<i>Paul E. Sojka</i> Purdue University
	Actuators for Combustion Control: New Approaches and Material	<i>Uri Vandsburger</i> Virginia Polytechnic Institute
10:45 am	Break	
11:00 am	Panel Discussion: AGTSR Education	Moderator: <i>Daniel Fant</i> South Carolina Energy R&D Center
	Panel Members:	
	Undergraduate Research	<i>Ting Wang</i> Clemson University
	Undergraduate Research	<i>Terry W. Simon</i> University of Minnesota
	Advanced Gas Turbine Systems Research	<i>Ajay K. Agrawal</i> University of Oklahoma
	Summer Fellowships	<i>Ronald LaFleur</i> Clarkson University
12:00 pm	Luncheon - Arlington Ballroom Salon V	

Advanced Turbine Systems Annual Program Review
Crystal Gateway Marriott
Arlington, Virginia
November 9-11, 1994

Agenda (Continued)

Day 3 - Friday, November 11, 1994 (Continued)

Session 5: University/Industry Consortium Project Review (Continued) - Arlington Ballroom Salon I
Session Chair: Lawrence P. Golan, South Carolina Energy Research and Development Center

1:00 pm Poster Session - New Research Topics

Aerodynamics Group

Advanced Design Methodology Using Combined Computational/Experimental Techniques for Gas Turbine Film Cooling	<i>James H. Leylek and James A. Liburdy</i> Clemson University
---	---

Experimental and Computational Studies of Film Cooling with Compound Angle Injection	<i>Terry W. Simon</i> University of Minnesota
---	--

Improving Aerodynamics of the Intercooler Flow Path for the Development of High Efficiency Gas Turbines	<i>Ajay K. Agrawal</i> The University of Oklahoma
---	--

Examining the Role of Turbulence on Internal Heat Transfer	<i>Paul A. Dellenback</i> University of Wyoming
---	--

Heat Transfer Group

Rotational Effects on Turbine Blade Cooling	<i>Gerald R. Guenette</i> Massachusetts Institute of Technology
---	--

Measurements of Surface Temperature and Heat Transfer Using a UV-Induced Phosphor Fluorescence Thermal Imaging System	<i>Mingking K. Chyu</i> Carnegie Melon University
---	--

Advanced Multistage Turbine Blade Aerodynamics, Performance, Cooling & Heat Transfer	<i>Sanford Fleeter and Patrick Lawless</i> Purdue University
--	---

Steam as Turbine Blade Coolant	<i>Abraham Engeda and John Lloyd</i> Michigan State University
--------------------------------	---

Materials Group

Combustion Chemical Vapor Deposited Coatings for Thermal Barrier Coating Systems	<i>W. Brent Carter</i> Georgia Institute of Technology
---	---

Compatibility of Gas Turbine Materials with Steam Cooling	<i>Vimal Desai</i> University of Central Florida
--	---

Advanced Turbine Systems Annual Program Review
Crystal Gateway Marriott
Arlington, Virginia
November 9-11, 1994

Agenda (Continued)

Day 3 - Friday, November 11, 1994 (Continued)

Session 5: University/Industry Consortium Project Review (Continued) - Arlington Ballroom Salon I
Session Chair: Lawrence P. Golan, South Carolina Energy Research and Development Center

1:00 pm **Poster Session - New Research Topics** (Continued)

Combustion Group

The Role of Unmixedness, Strain Rate and
Length Scale on Performance and Emissions

Scott Samuelson and John LaRue
UCI Combustion Laboratory
University of California, Irvine

Advanced Concepts for High Efficiency and
Low NO_x Gas Turbine Combustor
Development

Ashwani K. Gupta
University of Maryland

Manifold Methods for Methane Combustion

Stephen Pope
Cornell University

3:00 pm **Meeting Adjourns**

**Advanced Turbine Systems
Annual Program Review
Arlington, Virginia
November 9-11, 1994**

ATTENDEES

Mr. Jeffrey Abboud
Director of Government Affairs
US Advanced Ceramics Assn.
1600 Wilson Blvd., Ste. 1008
Arlington, VA 22209
(703) 812-8740
FAX: (703) 812-8743

Dr. Sy Ali
Director, Industrial Engine Tech
Allison Engine Company
PO Box 420, MS W16
Indianapolis, IN 46206
(317) 230-6864
FAX: (317) 230-5600

Mr. Roy Allen
Consulting Engineer
155 Kare Fre Blvd.
West Union, SC 29696
(803) 638-8575
FAX: (803) 638-8575

Mr. Robert Allinson
Director
Aerospace Alloys Inc.
12 East Gate Road
Long Valley, NJ 07853
(908) 879-7916
FAX: (908) 879-7897

Mr. Charles Alsup
Project Manager
METC
P.O. Box 880, MS0-DO1
3610 Collins Ferry Road
Morgantown, WV 26507-0880
(304) 285-5432
FAX: (304) 285-4403

Dr. Richard Anderson
Oak Ridge National Lab
PO Box 2008
I&C Division, MS 6007
Oak Ridge, TN 37831-6007
(615) 574-5706
FAX: (615) 574-4058

Dr. Don Anson
Research Leader
Battelle
505 King Avenue
Columbus, OH 43201
(614) 424-5823
FAX: (614) 424-3321

Mr. Juan Araujo
Graduate Research Assistant
Clarkson University
c/9 Ron LaFleur
Box 5725, Dept. MAE
Potsdam, NY 13699-5725
(315) 268-3887
FAX: (315) 268-6438

Dr. David Archer
Adjunct Professor
Carnegie Mellon University
114 Kentzer Road
Pittsburgh, PA 15237-2816
(412) 268-6808
FAX: (412) 268-3348

Mr. Patrick Arthur
Engineer
Antares Group, Inc.
8240 Professional Place, Ste. 207
Landover, MD 20875
(301) 731-1900
FAX: (301) 731-1904

Mr. Aziz Azimi
Energetics, Incorporated
7164 Gateway Drive
Columbia, MD 21046
(410) 290-0370

Dr. Richard Bajura
Director
NRCCE
PO Box 6064/Evansdale Drive
Morgantown, WV 26506-6064
(304) 293-2867
FAX: (304) 293-3749

Ms. Rita Bajura
Division Director
METC
P.O. Box 880, MS0-DO1
3610 Collins Ferry Road
Morgantown, WV 26507-0880
(304) 285-4109
FAX: (304) 285-4292

Mr. Peter Baldwin
Vice President Business Dev.
North. Research & Eng. Corp.
39 Olympia Avenue
Woburn, MA 01801-2073
(617) 937-3610
FAX: (617) 935-9052

Dr. Katayun Barmak
Professor
Lehigh University
Dept. of Matls. Science & Engineering
5 E. Packer Avenue
Bethlehem, PA 18015
(610) 758-4218
FAX: (610) 758-4244

M. Arun Basu
Senior Research Engineer
Amoco
Amoco Research Center
MS E-1-F, Bldg. 300
Naperville, IL 60566

Mr. Paul Bautista
Product Manager, Power Generation
Gas Research Institute
8600 West Bryn Mawr Ave.
Chicago, IL 60631
(312) 399-5460
FAX: (312) 394-8170

Mr. Charles Benson
Manager, Combustion Technology Unit
Arthur D. Little, Inc.
Acorn Park
Cambridge, MA 02140
(617) 498-5898
FAX: (617) 498-7007

Dr. Thomas Bernecki
Research Scientist
BIRL
1801 Maple Ave.
Evanston, IL 60201
(708) 491-2448
FAX: (708) 463-1022

Mr. Frank Bevc
Manager, Emerging Tech.
Westinghouse Electric Corp.
4400 Ala Faya Trail, MC 381
Orlando, FL 32826-2399
(407) 281-3393
FAX: (407) 281-5014

Mr. Ben Bingham
Gas Turbine Assoc.

Ms. Jennifer Bingham
Gas Turbine Assoc.

Mr. Donald Blomquist
Chief, Automated Production Tech. Div.
NIST
Rt. 270 & Clopper Road
Bldg. 233/Sound, Room B-108
Gaithersburg, MD 20899
(301) 975-6600
FAX: (301) 869-3536

Mr. Daniel Boss
Research Scientist
BIRL
1801 Maple Ave.
Evanston, IL 60201
(708) 491-3373
FAX: (708) 467-1022

Mr. Michael Brands
Executive Engineer
Cummins Engine Co.
Mail Code 20320, Box # 3005
Columbus, IN 47201
(812) 377-5797
FAX: (812) 377-5535

Mr. Richard Brent
Marketing Manager, ATS
Solar Turbines, Inc.
PO Box 85376
San Diego, CA 92186-5376
(619) 544-5663
FAX: (619) 544-2878

Dr. Scott Brewster
Research Associate
Brigham Young University
45 CTB
Provo, UT 84602
(801) 378-6240
FAX: (801) 378-3831

Mr. Virgil Bullington
Staff Engineer
Martin Marietta
PO Box 2008, Bldg. 4515
Oak Ridge, TN 37831-6066
(615) 574-6098
FAX: (615) 574-6098

Mr. Warren Bunker
Program Manager
Department of Energy
FE-231, 1000 Independence Ave. S.W.
Washington, DC 20585
(301) 903-2824
FAX: (301) 903-2406

Mr. R. D. Buxton
Power Generation Manager
Rolls-Royce, Inc.
11911 Freedom Drive
Reston, VA 22090
(703) 318-9008
FAX: (703) 318-9009

Mr. James Carr
Materials Engineer
Department of Energy
1000 Independence Avenue, S.W., FE-72GTN
Washington, DC 20585
(301) 903-6519
FAX: (301) 903-8350

Mr. Peter Carroll
VP, Govt. Relations
Solar Turbines, Inc.
818 Connecticut Ave., N.W., Ste. 600
Washington, DC 20006-2702
(202) 293-4327
FAX: (202) 293-4336

Dr. Kent Casleton
Research Chemist
METC
P.O. Box 880, M/S N05
Morgantown, WV 26505
(304) 285-4573
FAX: (304) 285-4403

Dr. M. R. Cates
Oak Ridge National Lab
PO Box 2008
K-1220, MS-7280
Oak Ridge, TN 37831-6007
(615) 574-8056
FAX: (615) 576-0279

Dr. Baki Cetegen
Associate Professor
University of Connecticut
ME Dept.
Storrs, CT 06269-3139
(203) 486-2966
FAX: (203) 486-5088

Mr. Tom Chance
General Electric Co.
Bldg. 53, Rm. 425
One River Road
Schenectady, NY 12345
(518) 385-2968
FAX: (518) 385-4314

Dr. Jyh-Yuan Chen
Associate Professor
University of California
6163 Etcheverry Hall
Berkeley, CA 94720
(510) 642-3286
FAX: (510) 642-6163

Dr. Ndaona Chokani
Associate Professor of Aerospace Eng.
North Carolina State Univ.
Department of Mech. & Aerospace Eng.
Box 7910
Raleigh, NC 27695-7910
(919) 515-5667
FAX: (919) 515-7968

Mr. Joe Cleveland
Mgr. - Operations Engineering
Department of Energy
ETEC, PO Box 7930
Canoga Park, CA 91304
(818) 586-6128

Dr. Arthur Cohn
Technical Manager
EPRI
PO Box 10412
Palo Alto, CA 94303
(415) 855-2525
FAX: (415) 855-2954

Mr. Arthur Cox
Manager-Adv. Matls./Process Engineering
Pratt & Whitney
MS 114-41, 400 Main Street
East Hartford, CT 06108
(407) 796-6560
FAX: (407) 796-7454

Mr. Ron Cox
ABB Power Generation Inc.
1200 Willis Road
Richmond, VA 23237
(804) 743-4042

Mr. Kevin Craig
Senior Chemical Process Eng.
NREL
1617 Cole Blvd.
Golden, CO 80401
(303) 275-2931
FAX: (303) 275-2905

Mr. Phil Craig
Program Manager
Dupont Lanxide Composites
1300 Marrows Road, PO Box 6077
Newark, NJ 19714-6077
(302) 456-6577
FAX: (302) 456-6480

Mr. Michael Cybulsky
Sr. Consulting Engineer
ABB Combustion Engineering
1000 Prospect Hill Road
CEP: 900 7-0301
Windsor, CT 06095-0500
(203) 285-3717
FAX: (203) 285-2513

Mr. Peter D'Alessandro
President
HSDE Inc.
14900 Woodham Drive, Ste. A-130
Houston, TX 77073
(713) 821-8448
FAX: (713) 821-8442

Mr. Ralph Dalla Betta
CATALYTICA
430 Ferguson Drive
Mountain View, CA 94043
(415) 960-3000
FAX: (415) 960-0127

Dr. Brian Damiano
Oak Ridge National Lab
PO Box 2008, Bldg. 3500, MS 6010
Oak Ridge, TN 37831-6010
(615) 574-5541
FAX: (615) 574-4058

M. Thong Dang
Associate Professor
Syracuse University
149 Link Hall
Syracuse, NY 13244
(315) 443-4311
FAX: (315) 443-9099

Mr. Joseph Darguzas
Project Manager
Sargent & Lundy
55 E. Monroe
Chicago, IL 60603
(312) 269-6902
FAX: (312) 269-3681

Mr. Frank Davis
Cleveland Office Manager
AlliedSignal Aerospace
7550 Lucerne Drive, Ste. 203
Middleburg Heights, OH 44130-6503
(216) 826-0330
FAX: (216) 826-0333

Dr. J. Day
Sr. Development Associate
Corning Incorporated RD&E
SP-DV-2-1
Corning, NY 14831
(607) 974-3823
FAX: (607) 974-2103

Dr. William Day
Director, Gas Turbine Eng./New Prod. Dev
Turbo Power & Marine Systems
PO Box 611, Aircraft Road
Middletown, CT 06457
(203) 343-2006
FAX: (203) 343-2268

Mr. Richard De Corso
Mechanical Engineer
1075 West Martha's Court
Knoxville, MD 21758-9120
(301) 834-9871

Mr. Mario DeCorso
Vice President
Power Tech Associates, Inc.
925 Baltimore Pike
PO Box 521
Concordville, PA 19331
(610) 358-1800
FAX: (610) 358-1802

Mr. Elia Demetri
Senior Project Engineer
North. Research & Eng. Corp.
39 Olympia Avenue
Woburn, MA 01801-2073
(617) 937-4624
FAX: (617) 935-9052

Mr. Victor Der
Office Director
Department of Energy
19901 Germantown Road, FE-73
Germantown, MD 20585
(301) 903-2700
FAX: (301) 903-2713

Mr. Clark Dohner
Project Manager
EPRI
3412 Hillview Ave., PO Box 10412
Palo Alto, CA 94303
(408) 446-0623
FAX: (415) 855-8759

Mr. Kenneth Dreitlein
Manager, Business Development
United Tech. Research Ctr.
411 Silver Lane - MS 24
East Hartford, CT 06108
(203) 727-7116
FAX: (203) 727-2160

Dr. David Duhl
Pratt & Whitney
MS 114-41, 400 Main Street
East Hartford, CT 06108
(203) 565-3428
FAX: (203) 565-5635

Dr. Michael Dunn
Vice President, Research Fellow
Calspan Advanced Tech. Ctr.
PO Box 400
Buffalo, NY 14225
(716) 631-6747
FAX: (716) 631-6701

Mr. Chris Ebel
Business Development Manager
Carborundum Company
Niagara Falls Technology Center
PO Box 832
Niagara Falls, NY 14302
(716) 278-2049
FAX: (716) 278-2068

Mr. Jack Eisenhower
Energetics, Incorporated
7164 Gateway Drive
Columbia, MD 21046
(410) 290-0370

Dr. William Ellingson
Senior Scientist
Argonne National Lab
9700 South Cass Ave.
Argonne, IL 60439
(708) 252-5068
FAX: (708) 252-4798

Mr. John Eustis
Department of Energy - Retired
28027 Playtor Road
Easton, MD 20585
(410) 822-2684
FAX: (410) 820-0127

Mr. Lanny Feder
Asst. Director, Prog. Development
Institute of Gas Technology
1825 K St., N.W., Ste. 503
Washington, DC 20006
(202) 785-3511
FAX: (202) 223-4625

Dr. Raymond Fessler
Director
BIRL
1801 Maple Ave.
Evanston, IL 60201
(708) 491-7600
FAX: (708) 491-4486

Mr. L. Robert Flais
Principal Engineer
Babcock & Wilcox Co.
R&D Division, 1562 Beeson Street
Alliance, OH 44601
(216) 829-7296
FAX: (216) 823-0639

Mr. Dick Fleming
CATALYTICA
430 Ferguson Drive
Mountain View, CA 94043
(415) 960-3000
FAX: (415) 960-0127

Mr. John Ford
President
Ford Associates
4382 Ampudia Street
San Diego, CA 92103
(619) 295-9324
FAX: (619) 295-9324

Dr. Steven Freedman
Executive Technology Manager
Gas Research Institute
8600 West Bryn Mawr Ave.
Chicago, IL 60631
(312) 599-8390
FAX: (312) 399-8170

Mr. Doug Freitag
Consultant
Dupont Lanxide Composites
17 Rocky Glen Ct.
Brookeville, MD 20833-6077
(301) 570-3821
FAX: (301) 570-3836

Dr. Albert Fresco
Market Development Manager
Dupont Lanxide Composites
1300 Marrows Road, PO Box 6077
Newark, DE 19714-6077
(302) 456-6241
FAX: (302) 456-6480

Mr. Robert Fuller
Manager, Marketing
Westinghouse Sci. & Tech. Ctr.
1310 Beulah Road
Pittsburgh, PA 15235
(412) 256-2426
FAX: (412) 256-2789

Mr. Stephen Gates
Director, ATS Programs
Solar Turbines, Inc.
2200 Pacific Coast Highway
San Diego, CA 92138
(619) 575-7463

Mr. Donald Geiling
Project Manager
METC
P.O. Box 880, MS0-D6
3610 Collins Ferry Road
Morgantown, WV 26507-0880
(304) 285-4784
FAX: (304) 285-4403

Mr. Maurice Gell
Professor In Residence
University of Connecticut
U-136, Dept. of Metallurgy
Storrs, CT 06269
(203) 486-3514

Dr. Randall Gemmen
Mechanical Engineer
METC
P.O. Box 880, MS N05
Morgantown, WV 26505
(304) 285-4536
FAX: (304) 285-4403

Mr. Madhav Ghate
Division Director
METC
P.O. Box 880, MS0-DO1
3610 Collins Ferry Road
Morgantown, WV 26507-0880
(304) 285-
FAX: (304) 285-4403

Mr. Robert Giglio
Sr. Development Engineer
Foster Wheeler Dev. Corp.
12 Peach Tree Hill Rd.
Livingston, NJ 07039
(201) 535-2458
FAX: (201) 535-2242

M. S. R. Gollahalli
Aerospace & Mechanical Engineer
University of Oklahoma
865 Asp., Rm. 212
Norman, OK 73019-0601
(405) 325-5011
FAX: (405) 325-1088

Dr. J. P. Gore
Professor
Purdue University
School of Mech. Engr.
West LaFayette, IN 47906
(317) 494-1452
FAX: (317) 494-0530

Dr. Heinz Gorges
President
VINETA, Inc.
3705 Sleepy Hollow Road
Falls Church, VA 22041
(703) 841-7252
FAX: (703) 841-7252

Mr. George Graves
Senior Research Ceramist
Univ. of Dayton Resrch. Inst.
300 College Park
Dayton, OH 45469-0172
(513) 229-4415
FAX: (513) 229-3433

Mr. Edward Gray
President
Antares Group, Inc.
8240 Professional Place, Ste. 207
Landover, MD 20875
(301) 731-1900
FAX: (301) 731-1904

Mr. Robert Gross
Project Manager
METC
P.O. Box 880, MS0-DO1
3610 Collins Ferry Road
Morgantown, WV 26507-0880
(304) 285-5426
FAX: (304) 285-4403

Ms. Valerie Grubb
Engineering Exhibits Coordinator
Allison Engine Company
PO Box 420, Speed Code S7B
Indianapolis, IN 46206-0420
(317) 230-2217
FAX: (317) 230-2987

Mr. John Gruver
Executive Consultant
Westinghouse Electric Corp.
1801 K Street, N.W.
Washington, DC 20006
(202) 835-2350
FAX: (202) 835-2379

Mr. David Hadacek
Principal Engineer, ATS
Solar Turbines, Inc.
2200 Pacific Highway
San Diego, CA 92186
(619) 544-5672

Mr. Mark Hadley
Mechanical Engineer
METC
P.O. Box 880
Morgantown, WV 26505
(304) 285-4929
FAX: (304) 285-4403

Mr. Barry Haest
Westinghouse Electric Corp.
4400 Ala Faya Trail, MC 381
Orlando, FL 32826-2399
(407) 281-3270

Mr. Michael Halter
Research Engineer
Clarkson University
c/o Ronald LaFleur
Box 5725, Dept. MAE
Potsdam, NY 13699-5725
(315) 268-3887
FAX: (315) 268-6438

Mr. John Hanigofsky
Senior Materials Scientist
TA&T, Inc.
133 Defense Highway, Ste. 212
Annapolis, MD 21401
(301) 261-8373
FAX: (410) 224-4678

Dr. Prith Harasgama
Chief-Heat Transfer
ABB Power Generation Ltd.
Gas Turbine Dept., KWRB
Baden, CH-5401
(+41)-56-49184
FAX: (+41)-56-753797

Mr. Martin Harmer
Director, Materials Research Ctr.
Lehigh University
Whitaker Lab, 5 E. Packer Avenue
Bethlehem, PA 18015
(610) 758-4227
FAX: (610) 758-4244

Mr. Robert Harrison
Consultant
Oak Ridge National Lab
414 Waterbury Circle
Cincinnati, OH 45231-6064
(513) 771-3158
FAX: (513) 771-3158 (call first)

Mr. George Hay
CAGT Program
42 Sunrise Lane
Lafayette, CA 94549
(510) 947-2910
FAX: (510) 866-5318

Dr. Bruce Hedman
Senior Program Manager
Pacific Northwest Lab
901 D Street S.W., Ste. 900
Washington, DC 20024
(202) 646-7808
FAX: (202) 646-5020

Mr. Jack Hill
Turbine Metallurgical Engineer
A. Finkl & Sons
PO Box 4279
Greenville, SC 29608
(803) 235-4109
FAX: (803) 271-7997

Dr. Don Hindman
Principal Engineer
Babcock & Wilcox
Box 11165, LRC, MS 76
Lynchburg, VA 24506
(804) 522-5825
FAX: (804) 522-6196

Mr. E. Hoffman
Manager, Industrial Conservation Tech.
Department of Energy
1 Bethel Valley Road
Bldg. 45156, MS 6065
Oak Ridge, TN 37831-6065
(304) 291-4114

Mr. Robert Holcomb
Oak Ridge National Lab
PO Box 2009
MS 8088, Bldg. 9108
Oak Ridge, TN 37831-6007
(615) 574-0273
FAX: (615) 574-2102

Dr. Norman Holcombe
Project Manager
METC
P.O. Box 880, MS0-DO1
3610 Collins Ferry Road
Morgantown, WV 26507-0880
(304) 285-4829
FAX: (304) 285-4403

Dr. William Hong
Research Staff Member
Inst. for Defense Analyses
1801 N. Beauregard St.
Alexandria, VA 22311
(703) 578-2826
FAX: (703) 578-2877

Dr. Diane Hooie
Project Manager
METC
P.O. Box 880, MS0-DO1
Morgantown, WV 26507-0880
(304) 285-4524
FAX: (304) 285-4403

Mr. Mike Horner
Manager, Adv. Products
GE/M&I
5122 One Neumann Way
Cincinnati, OH 45215-6301
(513) 552-5550
FAX: (513) 552-5558

Mr. Michael Hsu
President
ZTEK Corporation
460 Totten Pond Road, Rm. 600
Waltham, MA 02154
(617) 890-5665
FAX: (617) 890-3731

Mr. Richard Hune
Marketing Representative
Westech Gear Corp.
117 N. Palmer
Houston, TX 77003
(713) 224-8911
FAX: (713) 224-8584

Mr. George Husman
Vice President, Engineering Div.
Southern Research Institute
757 Tom Martin Drive
Birmingham, AL 35211
(205) 581-2977
FAX: (205) 581-2414

Mr. Michael Izsak
Engineer
Burns and Roe Svcs. Corp.
PO Box 18288
Pittsburgh, PA 15236
(412) 892-4190
FAX: (412) 892-4736

Mr. Michael Janus
Mechanical Engineer
METC
P.O. Box 880, MS N05
Morgantown, WV 26505
(304) 285-4912
FAX: (304) 291-4403

Ms. Cindy Jones
General manager
Precision Engine Contr. Corp.
11661 Sorrento Valley Road
San Diego, CA 92121
(619) 792-3242
FAX: (619) 792-3200

Mr. Michael Jones
Manager, Power Generation Mktg.
Solar Turbines, Inc.
PO Box 85376
San Diego, CA 92180
(619) 544-5740
FAX: (619) 544-2633

Dr. Larry Joseph
Department of Energy
FE-231, 1000 Independence Ave. S.W.
Washington, DC 20585
(615) 574-6098

Dr. Tony Kaiser
V.P. Power Plant Laboratories
ABB Combustion Engineering
1000 Prospect Hill Road
Windsor, CT 06095
(203) 285-5081
FAX: (203) 285-2565

Mr. Peter Kamber
Consulting Engineer
ABB Power Plant Labs
1000 Prospect Hill Road
Windsor, CT 06095
(203) 285-5534

Dr. Jayanta Kapat
Visiting Assistant Professor
Clemson University
Department of Mechanical Engineering
Box 340921
Clemson, SC 29634-0921
(803) 656-5643
FAX: (803) 656-4435

Dr. John Kibler
Director of Engineering
Materials Sciences Corp.
500 Office Center Drive, Ste. 250
Ft. Washington, PA 19034
(215) 542-8400
FAX: (215) 542-8401

Mr. Thomas King
Mechanical Engineer
Department of Energy
1000 Independence Ave. S.W.
OIT - Rm. 5G-067, EE-221
Washington, DC 20585
(202) 586-9663
FAX: (202) 586-7114

Mr. Thomas Kircher
Sermatech International
155 S.Limerick Rd.
Limerick, PA 19468
(610) 948-5100
FAX: (610) 948-0811

Mr. David Knowles
Senior Program Analyst
MELE Associates
7428 West More Road
Rockville, MD 20850
(301) 309-8442

Dr. Klod Kokini
Professor
Purdue University
1288 Mechanical Engineering Bldg.
West LaFayette, IN 47907-1288
(317) 494-5727
FAX: (317) 494-0539

Mr. Gil Kraemer
Research Engineer
Precision Combustion, Inc.
25 Science Park
New Haven, CT 06511
(203) 786-5215
FAX: (203) 786-5214

Dr. John LaRue
Professor
Univ. of CA, Irvine
UCI Combustion Lab
Irvine, CA 92717-3550
(714) 661-4437
FAX: (714) 856-8585

Mr. Ming-Chia Lai
Associate Professor
Wayne State University
Mechanical Engineering Dept.
Detroit, MI 48202
(313) 577-3893
FAX: (313) 577-8789

Dr. Lee Langston
Professor
University of Connecticut
ME Dept., U-139
191 Auditorium Road
Storrs, CT 06269-3139
(203) 486-4884
FAX: (203) 486-5088

Mr. John Latcovich
Manager, GT Machines
ABB Power Generation Inc.
1200 Willis Road
Richmond, VA
(804) 743-4042
FAX: (804) 743-4018

Mr. Normand Laurendeau
Professor of Mechanical Engineering
Purdue University
School of Mech. Engr.
West LaFayette, IN 47906
(317) 494-2713
FAX: (317) 494-0539

Mr. Abbie Layne
Project Manager
METC
P.O. Box 880, MS0-D06
Morgantown, WV 26505
(304) 291-4603
FAX: (304) 291-4469

Dr. Woo Lee
Research Staff Member
Oak Ridge National Lab
PO Box 2008, MS 6063
Oak Ridge, TN 37831-6063
(615) 576-2894
FAX: (615) 574-6918

Dr. Gerald Leverant
Director
Southwest Research Institute
6220 Culebra Rd.
San Antonio, TX 78230
(210) 522-2041
FAX: (210) 522-5122

Dr. Stanley Levine
Chief, Ceramics Branch
NASA Lewis Research Ctr.
21000 Brookpark Road, MS 106/5
Cleveland, OH 44135
(216) 433-3276
FAX: (216) 433-5544

Dr. Mark Lewis
Associate Professor
University of Maryland
College Park, MD 20742
(301) 405-1133
FAX: (301) 314-9001

Mr. Charles Logan
Director, Business Development
B. P. Chemicals (HITCO)
1600 W. 135th St.
Gardena, CA 90249
(310) 970-5784
FAX: (310) 516-5786

Mr. Bill Long
Technical Advisor
Babcock & Wilcox
Box 11165, LRC, MS 76
Lynchburg, VA 24506
(804) 522-5716
FAX: (804) 522-6196

Dr. Daniel Maloney
General Engineer
METC
P.O. Box 880, MS N05
Morgantown, WV 26505
(304) 285-4629
FAX: (304) 285-4403

Ms. Amal Mansour
CEO & Chairman of the Board
MTCI
PO Box 21
Columbia, MD 21045
(410) 730-0445
FAX: (410) 997-6980

Dr. Momtaz Mansour
President
MTCI
PO Box 21
Columbia, MD 21045
(410) 730-0445
FAX: (410) 997-6980

Mr. Thomas Marciniak
Manager, Ind. & Utility Technology
Argonne National Lab
9700 South Cass Ave.
Bldg. 362, Rm. C205
Argonne, IL 60439
(708) 252-5860
FAX: (708) 252-9728

Mr. Richard Marloff
Manager, Engineering Systems
Westinghouse Sci. & Tech. Ctr.
1310 Beulah Road
Pittsburgh, PA 15235
(412) 256-1308
FAX: (412) 256-2444

Mr. Joe Mavec
Program Manager - CFCC
Department of Energy
9800 S. Cass
Argonne, IL 60439
(708) 252-2323

Mr. David McDeed
Mechanical Thermal Engineer
Siemens Power Corp.
1040 S. 70th Street
Milwaukee, WI 53214
(414) 475-3181
FAX: (414) 475-4117

Mr. Jon McDonald
Dept. of Mechanical Engineering
Vanderbilt University
Box 1592, Station B
Nashville, TN 37235-1592
(615) 322-3512
FAX: (615) 343-6687

Mr. Jim McEvers
Senior Staff Engineer
Oak Ridge National Lab
PO Box 2008, MS 6003
Oak Ridge, TN 37831-6003
(615) 574-5733
FAX: (615) 574-1249

Mr. Michael Merlo
Manager of Tech. Planning & Dev.
Southern CA Edison Co.
PO Box 800, 2244 Walnut Grove Ave.
Rosemead, CA 91770
(818) 302-9510
FAX: (818) 302-9663

Dr. Nathan Messersmith
Assistant Professor
Purdue University
School of Aeronautics & Astronautics
1282 Grissom Hall
West LaFayette, IN 47907-1282
(317) 494-4782
FAX: (317) 494-0307

Dr. William Messner
Manager
Consolidated Edison
4 Irving Place, Rm. 1428
New York, NY 10003
(212) 460-2753
FAX: (212) 529-8511

Mr. Harry Morehead
Mgr., New Program Dev.
Westinghouse Electric Corp.
4400 Ala Faya Trail, MC 381
Orlando, FL 32826-2399
(407) 281-3322
FAX: (407) 281-5014

Mr. Dale Mukavetz
ATS Program Manager
Allison Engine Company
PO Box 420, S/C T12A
Indianapolis, IN 46206
(317) 230-2575
FAX: (317) 230-3691

Mr. James Mummert
Manager, Sales and Marketing
AlliedSignal Fluid Systems
PO Box 22200
Tempe, AZ 85285-2200
(602) 893-4532
FAX: (602) 893-5289

Dr. S. N. B. Murthy
Professor
Purdue University
1003 Chaffee Hall
West LaFayette, IN 47907-1003
(317) 494-1509
FAX: (317) 494-0530

Mr. Tom Naslund
Wyman Gordon Invest. Castings
839 Poquonnock Rd., PO Box 999
Groton, CT 06340-0999
(203) 445-7421
FAX: (203) 449-6835

Dr. Ken Natesan
Senior Scientist
Argonne National Lab
9700 South Cass Ave.
Argonne, IL 60439
(708) 252-5103
FAX: (708) 252-3604

Mr. Gary Neuner
Business Development Manager
Westinghouse Electric Corp.
401 East Hendy Ave., PO Box 3499
Sunnyvale, CA 94088-3499
(408) 735-2712
FAX: (408) 735-4419

Mr. Don Newburry
Research Associate
METC
P.O. Box 880
3610 Collins Ferry Road
Morgantown, WV 26507-0880
(304) 285-4174
FAX: (304) 285-4403

Dr. Dimitris Nikitopoulos
Associate Professor
Louisiana State University
Mechanical Engineering
CEBA 2513B
Baton Rouge, LA 70803
(504) 388-5903
FAX: (504) 388-5924

Dr. Nirm Nirmalan
Staff Research Scientist
Allison Engine Company
PO Box 420, MS W16
Indianapolis, IN 46206
(317) 230-3052
FAX: (317) 230-5600

Mr. Paul Norton
Lead Design Engr. ATS
Solar Turbines, Inc.
2200 Pacific Highway
San Diego, CA 92128
(619) 544-5207
FAX: (619) 544-2682

Dr. Thomas Norton
Mechanical Engineer
METC
P.O. Box 880, MS N05
Morgantown, WV 26505
(304) 285-4617
FAX: (304) 285-4403

Mr. Armand Onesto
Program Manager
Department of Energy
ETEC, PO Box 7930
Canoga Park, CA 91309
(818) 586-5524
FAX: (818) 586-5118

Dr. Adeboyejo Oni
Morgan State University
School of Engineering
220 Mitchell Bldg.
Baltimore, MD 21239
(410) 319-3106
FAX: (410) 319-3982

Mr. Mark Ortner
Project Manager
Department of Energy
10 Rosecrest
Alexandria, VA 22301
(703) 683-4491
FAX: (202) 586-7714

Mr. Nick Otter
Director of Technology
European Gas Turbines
Cambridge Road, Whetstone
Leicester, LE8 6LH
(533) 750750
FAX: (533) 750768

Mr. George Padgett
Manager, Advanced Design Engineering
Solar Turbines, Inc.
PO Box 85376
San Diego, CA 92186-5371
(619) 544-5708
FAX: (619) 544-2682

Mr. Robert Pafe
Budget Analyst
Department of Energy
Office of CFO CR147
1000 Independence Ave., S.W.
Washington, DC 20585
(202) 586-4026
FAX: (202) 586-4504

Mr. Al Pallotta
Mgr., Combustion Turbine Materials
Westinghouse Electric Corp.
4400 Ala Faya Trail, MC 381
Orlando, FL 32826-2399
(407) 281-3223
FAX: (407) 281-2334

Dr. Joseph Panzarino
Director of Government Programs
Saint/Gobain/Norton
Goddard Road
Northboro, MA 01532
(508) 351-7820
FAX: (508) 351-7760

Mr. Yakin Patel
Graduate Student
University of Central FL
4000 Central Florida Blvd.
Orlando, FL 32816
(407) 823-5650
FAX: (407) 823-0208

Mr. Henry Price
Senior Engineering Analyst
NREL
1617 Cole Blvd.
Denver, CO 80401
(303) 275-3637
FAX: (303) 275-3601

Mr. Jeffrey Price
Senior Ceramic Engineer
Solar Turbines, Inc.
2200 Pacific Hwy., PO Box 85376
San Diego, CA 92186-5376
(619) 544-5538
FAX: (619) 544-2830

Mr. James Prochaska
Director of North Am. Sales
Stewart & Stevenson
2707 N. Loop West
Houston, TX 77008
(713) 868-7654
FAX: (713) 868-7697

Dr. Ishwar Puri
Associate Professor
Univ. of Illinois at Chicago
Department of Mech. Eng. - M/C 251
842 W. Taylor St., Rm. 2039
Chicago, IL 60607-7022
(312) 413-7560
FAX: (312) 413-0447

Mr. James Quinn
Office of Industrial Technologies
Department of Energy
100 Independence Ave., S.W.
EE-20, Rm. 64-025
Washington, DC 20585
(202) 586-5725
FAX: (202) 586-2176

Mr. Jim Reed
Energetics, Incorporated
7164 Gateway Drive
Columbia, MD 21046
(410) 290-0370

Mr. Michael Reitz
Budget Analyst
Department of Energy
1000 Independence Ave., S.W.
Washington, DC 20002
(202) 586-0690
FAX: (202) 586-4504

Mr. Ivan Rice
Consulting Engineer
PO Box 233
Spring, TX 77383
(713) 353-5040
FAX: (713) 353-5040

Mr. Tom Richards
Manager - New Technology
Caterpillar, Inc.
Technical Center, Bldg. A
PO Box 1856
Peoria, IL 61656
(309) 578-8597

Mr. Mike Roberts
Institute of Gas Technology
1825 K St., N.W., Ste. 503
Washington, DC 20006
(202) 785-3511

Mr. Paul Rodebaugh
Supervising Engr.
Florida Power & Light
PO Box 14000
Juno Beach, FL 33408-0420
(407) 691-2620
FAX: (407) 691-2695

Dr. Larry Roe
University of Arkansas
Mechanical Engineering Dept.
MEEG 204
Fayetteville, AR 72701
(501) 575-3750
FAX: (501) 575-6982

Dr. Glenn Romanoski
Staff Development Engineer
Oak Ridge National Lab
PO Box 2008
Bldg. 4508, MS-6088
Oak Ridge, CA 37831-6064
(615) 574-4838
FAX: (615) 576-8424

Dr. Ramendra Roy
Professor of Mechanical Engineering
Arizona State University
Dept. of Mechanical & Aerospace Eng.
Tempe, AZ 85287-6106
(602) 965-1482
FAX: (602) 965-1384

Mr. Sims Roy
Environmental Engineer
US EPA
Combustion Group, MD-13
Research Triangle Park, NC 27711
(919) 541-5263
FAX: (919) 541-4028

Dr. Purush Sahoo
Sermatech International
155 S.Limerick Rd.
Limerick, PA 19468
(610) 948-5100
FAX: (610) 948-0811

Mr. Louis Salvador
Associate Director
METC
Office of Product Technology Mgmt.
PO Box 880
Morgantown, WV 26507-0880
(304) 285-4147
FAX: (304) 285-4292

Dr. A. T. Sanders
Team Leader, Mechanical Technology
European Gas Turbines
PO Box 1
Lincoln, LN6 7AA
(+44) 522 583222
FAX: (+44) 522 583594

Dr. M. Taher Schobeiri
Associate Professor
Texas A&M University
Mechanical Engineering Dept.
College Station, TX 77843-3123
(409) 845-0819
FAX: (409) 845-3081

Mr. Robert Schulz
Program Manager
Department of Energy
EE-34, Forrestal Bldg.
Washington, DC 20585
(202) 586-8051
FAX: (202) 586-1600

Mr. Andrew Scriven
Research and Engineering Dept.
National Power (UK)
National Power Place, White Hill Way
Trigonos, Windmill Hill Business Park
Swindon, Wiltshire SN5 6PB
(044) 0793 896260
FAX: (044) 0793 896251

Mr. Shane Sedell
Engineer
RTT
1819 Sycamore Valley
Reston, VA 22090
(703) 482-1864
FAX: (703) 790-5736

Mr. Sam Shepard
Senior Engineer for R&D
Naval Sea Systems Command
Code 03X3C
Washington, DC 20362
(703) 602-3615
FAX: (703) 602-9512

Ms. Shellie Simler
Manager, Marketing
Advanced Materials Division
5945 Wellington Road
Gainesville, VA 22065
(703) 754-5736
FAX: (703) 754-5024

Dr. J. P. Singh
Scientist
Argonne National Lab
9700 South Cass Ave., Bldg. 212
Argonne, IL 60439
(708) 252-5123
FAX: (708) 252-3604

Mr. Kurt Sisson
Director, Office of Waste Reduction Tech
US DOE
11000 Independence Avenue
Washington, DC 20585
(202) 586-0139
FAX: (202) 586-3237

Dr. Edward Skolnik
Senior Scientist
Energetics, Incorporated
7164 Gateway Drive
Columbia, MD 21046
(410) 290-0370

Dr. Kenneth Smith
Chief Engineer, Combustion
Solar Turbines, Inc.
PO Box 85376
San Diego, CA 92119
(619) 544-5539
FAX: (619) 544-5500

Mr. Leslie Southall
Mgr. CT Engine Programs
Westinghouse Electric Corp.
450 Timber Ridge
Longwood, FL 32779
(407) 281-2960

Mr. Steven Sperling
Undergraduate Research Fellow
Clarkson University
c/9 Ronald LaFleur
Box 5725, Dept. MAE
Potsdam, NY 13699-5725
(315) 268-3887
FAX: (315) 268-6438

Mr. Robert Steele
PHD Candidate
University of Washington
Mechanical Engineering, FU-10
Seattle, WA 98195
(206) 685-3424
FAX: (206) 685-8047

Mr. David Stinton
Staff Scientist
Oak Ridge National Lab
PO Box 2008
Bldg. 4515, MS-6063
Oak Ridge, TN 37831-6063
(615) 574-4556
FAX: (615) 574-6918

Mr. Ted Strickland
Development Engineer
Precision Combustion, Inc.
25 Science Park
New Haven, CT 06511
(203) 786-5215
FAX: (203) 786-5214

Dr. Willard Sutton
Principal Scientist
United Tech. Research Ctr.
411 Silver Lane - MS 24
East Hartford, CT 06108
(203) 727-7639
FAX: (203) 727-7879

Ms. Denise Swink
Acting Deputy Assistant Secretary
OIT
US Dept of Energy
Washington, DC 20585
(202) 586-9232

Mr. Stephan Sylvan
Environmental Protection Specialist
EPA
401 M Street S.W.
Washington, DC 20460
(202) 233-9307
FAX: (202) 233-9575

Dr. Widen Tabakoff
Professor
University of Cincinnati
Dept. of Aerospace Engines
Mail Drop 70
Cincinnati, OH 45221
(513) 556-3226
FAX: (513) 556-5038

Mr. Rick Tetzloff
Black & Veatch
8400 Ward Parkway
Kansas City, MO 64114
(913) 339-2375

Dr. A. B Thakker
Senior Manager
Rolls Royce, Inc.
2849 Paces Ferry Road
Atlanta, GA 30339
(404) 436-7900
FAX: (404) 436-8570

Dr. John Tuzson
John Tuzson & Assoc.
1220 Maple Ave.
Evanston, IL 60202
(708) 864-2551

Mr. Robert Tyson
Associate Director, RD&D Programs
American Gas Association
1515 Wilson Blvd.
Arlington, VA 22209-2470
(703) 841-8572
FAX: (703) 841-8677

Mr. S. Van Der Linden
Vice President
ABB Power Generation
1160 Livingston Ave.
North Brunswick, NJ 08092
(908) 932-6319
FAX: (908) 932-6463

Mr. Jim VanLaar
Project Manager
Sargent & Lundy
55 E. Monroe
Chicago, IL 60603
(312) 269-2790
FAX: (312) 269-3681

Mr. Rick Vosburgh
President, CEO
Monitoring Technology Corp.
2731 Prosperity Avenue
Fairfax, VA 22031
(703) 698-5520
FAX: (703) 698-5539

Ms. Cecilia Wasterlid
Reporter - Technical Outlook
Embassy of Sweden
1501 M Street N.W.
Washington, DC 20005
(202) 467-2672
FAX: (202) 467-2678

Mr. Harvey Weisenfeld
Energetics, Incorporated
7164 Gateway Drive
Columbia, MD 21046
(410) 290-0370

Mr. James Wessel
Dir., Fed. Business Dev.
Dow Corning Corporation
1800 M St., NW, #325 South
Washington, DC 20036
(202) 785-0585
FAX: (202) 785-0421

Mr. Pat West
Project Engineer
US Generating
7500 Old Georgetown Road
Bethesda, MD 20814
(301) 718-6755
FAX: (301) 718-6917

Mr. Michael Wherley
Mechanical Engineer
SAIC
1710 Goodridge Drive, MS 2-7-12
McLean, VA 22102
(703) 734-5501
FAX: (703) 790-5744

Mr. Laurence Wilken
Business Manager, ATS
Solar Turbines, Inc.
PO Box 85376
San Diego, CA 92816-5376
(619) 544-5650
FAX: (619) 544-2878

Mr. John Wills
Director, Business Development
Rolls-Royce, Inc.
IPG Program
11911 Freedom Drive
Reston, VA 22090
(703) 318-9008
FAX: (703) 318-9009

Mr. Patrick Wing
Undergraduate Research Fellow
Clarkson University
c/o Ronald LaFleur
Box 5725, Dept. MAE
Potsdam, NY 13699-5725
(315) 268-3887
FAX: (315) 268-6438

Dr. I. G. Wright
Research Leader
Battelle
ASIT, Electric Power Research Institute
3412 Hillview Ave.
Palo Alto, CA 94303
(415) 855-2363
FAX: (415) 855-2287

Dr. Suxing Wu
Lehigh University
MRC, 5 Lehigh Univ., Whitaker Lab
Bethlehem, PA 18015
(610) 758-4216
FAX: (610) 758-4244

Mr. S. C. Yao
Professor
Carnegie Mellon University
Department of Mechanical Engineering
Pittsburgh, PA 15213
(412) 268-2508

Dr. Rich Yetter
Research Scientist
Princeton University
Dept. of Mech. & Aerospace Eng.
O-312 Equad
Princeton, NJ 08544
(609) 258-2947
FAX: (609) 258-1939

Dr. M. Joseph Yip
Mechanical Engineer
METC
P.O. Box 880
Morgantown, WV 26505
(304) 285-5434

Mr. Cherif Youssef
Industrial Manager
Southern California Gas
PO Box 3249
Los Angeles, CA 90051-1249
(213) 244-5325
FAX: (213) 486-5725

Mr. Martin Zhu
Graduate Student
University of Maryland
Department of Mechanical Engineering
College Park, MD 20742
(301) 405-1154

**Advanced Turbine Systems
Annual Program Review
Arlington, Virginia
November 9-11, 1994**

SPEAKERS

Mr. Sumanta Acharya
Louisiana State University
Mechanical Engineering
Baton Rouge, LA 70803
(504) 388-5809
FAX: (504) 388-5924/5990

Mr. Ajay Agrawal
Aerospace & Mechanical Engineer
University of Oklahoma
Norman, OK 73019-0601
(405) 325-1754
FAX: (405) 325-1088

Mr. Ronald Bannister
Manager, Gasification & Adv. Cycles
Westinghouse Electric Corp.
4400 Ala Faya Trail, MC 381
Orlando, FL 32826-2399
(407) 281-3270
FAX: (407) 281-5014

Mr. Thomas Bechtel
Director
METC
P.O. Box 880, MS0-DO1
3610 Collins Ferry Road
Morgantown, WV 26507-0880
(304) 285-4511
FAX: (304) 285-4292

Dr. Brent Carter
Assistant Professor
Georgia Tech
School of Materials Science & Engr.
Atlanta, GA 30332-0245
(404) 894-6762
FAX: (404) 853-9140

Mr. Mingking Chyu
Carnegie Mellon University
Mechanical Engineering
Pittsburgh, PA 15213-3890
(412) 268-3658
FAX: (412) 268-3348

Mr. James Corman
General Manager
GE Co. Advanced Tech.
1 River Road
Bldg. 53, Rm. 417
Schenectady, NY 12345
(518) 385-7779
FAX: (518) 385-0404

Mr. John Cuttica
Gas Research Institute
8600 West Bryn Mawr Ave.
Chicago, IL 60631
(312) 399-5460

Mr. Keith Davidson
Vice President, Project Development
On-Site Energy Corp.
701 Palomar Airport Rd., Ste. 200
Carlsbad, CA 92002
(619) 931-2400
FAX: (619) 931-2405

Dr. Paul Dellenback
Associate Professor
University of Wyoming
Mechanical Engineering
PO Box 3295
Laramie, WY 82071-3295
(307) 766-2946
FAX: (307) 766-2695

Dr. Vimal Desai
University of Central Florida
4000 Central Florida Blvd.
Orlando, FL 32816
(407) 823-5777
FAX: (407) 823-0208

Mr. Ihor Diackunchak
Advisory Engineer
Westinghouse Electric Corp.
4400 Ala Faya Trail, MC 381
Orlando, FL 32826-2399
(407) 281-5115
FAX: (407) 281-5633

Dr. Robert Dibble
Professor
Univ. of CA at Berkeley
6159 Etcheverry Hall
Dept. of Mechanical Engineering
Berkeley, CA 94270
(510) 642-4901
FAX: (510) 642-1850

Mr. Thomas Duffy
Manager, High Performance Steam
Solar Turbines, Inc.
PO Box 85376
San Diego, CA 92186-5376
(619) 595-7429
FAX: (619) 544-5528

Dr. Abraham Engeda
Assistant Professor
Michigan State University
Mechanical Engineering Department
A231 Engineering Building
East Lansing, MI 48824-1226
(517) 432-1834
FAX: (517) 353-1750

Mr. David Eppinger
Fluor Daniel
3333 Michelson Drive
Irvine, CA 92730
(714) 975-2103
FAX: (714) 975-3294

Ms. Christine Ervin
Assistant Secretary
U.S. Department of Energy
Energy Efficiency and Renewable Energy
1000 Independence Ave., S.W.
Washington, DC 20585
(202) 586-9220
FAX: (202) 586-9260

Dr. Dan Fant
Research Manager
South Carolina Energy R&D Center
386-2 College Avenue
Clemson, SC 29634-5180
(803) 656-2267
FAX: (803) 656-0142

Dr. Mattison Ferber
Group Leader, Mech. Prop. User Center
Oak Ridge National Lab
PO Box 2008, Bldg. 4515, MS-6064
Oak Ridge, TN 37831-6064
(615) 576-0818
FAX: (615) 576-6098

Ms. Patricia Godley
Assistant Secretary
U.S. Department of Energy
Fossil Energy
100 Independence Ave. S.W.
Washington, DC 20585
(301) 903-2824

Dr. Lawrence Golan
Director
SC Carolina Energy Center
386-2 College Avenue
Clemson University
Clemson, SC 29634-5180
(803) 656-2267
FAX: (803) 656-0142

Dr. Gerald Guenette
Principal Research Engineer
Massachusetts Inst. of Tech.
77 Mass Ave., Rm. 31-214
Gas Turbine Lab
Cambridge, MA 02139
(617) 253-3764
FAX: (617) 258-6093

Dr. Ashwani Gupta
Professor
University of Maryland
Department of Mechanical Engineering
College Park, MD 20742
(301) 405-5276
FAX: (301) 314-9477

Dr. John Halow
Director, Combustion/Cleanup Proj. Div.
Department of Energy
PO Box 880
Morgantown, WV 26505
(304) 291-4114
FAX: (304) 285-4403

Dr. Je-Chin Han
Professor
Texas A&M University
Mechanical Engineering Dept.
College Station, TX 77843-3123
(409) 845-3738
FAX: (409) 862-2418

Dr. Paul Hedman
Brigham Young University
ACERC, Chem. E. Dept.
Provo, UT 84602
(801) 378-6238
FAX: (801) 378-7799

Dr. Robert Hillery
Manager, Airfoil Materials
GE Aircraft Engines
1, Neumann Way, Mail Drop H85
Cincinnati, OH 45215
(513) 243-5580
FAX: (513) 243-6511

Mr. Malcolm Jacobson
Enron Emerging Technologies

Dr. Michael Karnitz
Program Manager
Oak Ridge National Lab
PO Box 2008, MS-6065
Oak Ridge, CA 37831-6065
(615) 574-5150
FAX: (615) 574-6098

Mr. Karl Karstensen
Program Manager
Solar Turbines, Inc.
PO Box 85376
San Diego, CA 92186-5376
(619) 544-5490
FAX: (619) 544-2878

Mr. William Koop
Wright Patterson AFB
1950 5th
Wright Patterson AFB, OH 45433-7251
(513) 255-8211
FAX: (513) 255-4100

Dr. Ronald LaFleur
Associate Professor
Clarkson University
Box 5725, Dept. MAE
Potsdam, NY 13699-5725
(315) 268-3823
FAX: (315) 268-6438

Dr. Patrick Lawless
Assistant Professor
Purdue University
School of Mech. Engr.
West LaFayette, IN 47906
(317) 494-7856
FAX: (317) 494-0530

Dr. James Leylek
Clemson University
Mechanical Engineering
313-B Riggs Hall
Clemson, SC 29634
(803) 656-5640
FAX: (803) 656-4435

Dr. James Liburdy
Clemson University
Dept. of Mechanical Engineering
Clemson, SC 29634
(803) 656-3295
FAX: (803) 656-4435

Dr. John Lloyd
Distinguished Professor
Michigan State University
Mechanical Engineering Department
A231 Engineering Building
East Lansing, MI 48824-1226
(517) 432-1834

Mr. Richard Lowden
Oak Ridge National Lab
PO Box 2008, MS 6087, Bldg. 4508
Oak Ridge, TN 37831-6087
(615) 576-2769
FAX: (615) 576-8424

Mr. Jimmy Lunsford
Vice President, Industrial Engines
Allison Engine Company
PO Box 420, MS U8
Indianapolis, IN 46206
(317) 230-6864
FAX: (317) 230-3348

Dr. Arnold Marder
Professor
Lehigh University
Whitaker Lab
Dept. of Matls. Science and Engineering
Bethlehem, PA 18017
(610) 758-4197
FAX: (610) 758-4244

Ms. Jan McFarland
Office of Air and Radiation
Environmental Protection Agency
401 M Street S.W.
Washington, DC 20460
(202) 260-2090

Dr. A. Mellor
Centennial Professor of Mech. Engr.
Vanderbilt University
Box 1592, Station B
Nashville, TN 37235-1592
(615) 343-6214
FAX: (616) 343-6687

Mr. Harold Miller
General Electric Corp.
One River Road
Bldg. 53, Room 425
Schenectady, NY 12345
(518) 385-1660
FAX: (518) 385-0404

Mr. Dennis Moran
American Gas Association
1515 Wilson Blvd.
Arlington, VA 22209-2470
(703) 841-8572

Mr. William Parks
Program Manager, CE-222
Department of Energy
1000 Independence Ave., S.W.
Washington, DC 20585
(202) 586-2093
FAX: (202) 586-7114

Mr. Edward Parsons
Chief, Power Gen. Tech. Branch
METC
P.O. Box 880, MS0-DO6
Morgantown, WV 26505
(304) 285-4533
FAX: (304) 285-4403

Mr. Leland Paulson
METC
P.O. Box 880
Morgantown, WV 26507
(304) 285-4074

Dr. Stephen Pope
Professor
Cornell University
Upson Hall
Ithaca, NY 14853
(607) 255-4314
FAX: (607) 255-1222

Dr. Mohan Razdan
Chief of Combustor R&D
Allison Engine Company
PO Box 420, Speed Code T14
Indianapolis, IN 46206-0420
(317) 230-6404
FAX: (317) 230-3691

Dr. George Richards
Principal Investigator
METC
PO Box 880
Morgantown, WV 26505
(304) 291-4458
FAX: (304) 285-4403

Mr. David Rohy
Director Advanced Turbine Systems
Solar Turbines, Inc.
PO Box 85376
San Diego, CA 92186-5376
(619) 544-5078
FAX: (619) 544-2682

Dr. Scott Samuelsen
Professor
Univ. of CA, Irvine
UCI Combustion Lab
Irvine, CA 92717-3550
(714) 856-5468
FAX: (714) 856-7423

Dr. Terrence Simon
Professor
University of Minnesota
Mechanical Engineering Department
111 Church Street
Minneapolis, MN 55455-0111
(612) 625-5831
FAX: (612) 624-1398

Dr. Paul Sojka
Associate Professor
Purdue University
School of Mech. Engr.
West LaFayette, IN 47907-1003
(317) 494-1536
FAX: (317) 494-1530

Dr. Thomas Tom
Director, Advanced Technology
Howmet Corporation
1500 South Warner St.
Whitehall, MI 49461
(616) 894-7897
FAX: (619) 894-7826

Mr. George Touchton
Target Manager
Electric Power Research Inst.
3412 Hillview Avenue
Palo Alto, CA 94304
(415) 855-8935
FAX: (415) 855-8759

Dr. Uri Vandsburger
Virginia Tech
Mechanical Engineering Dept.
Blacksburg, VA 24061-0238
(703) 231-4459
FAX: (703) 231-9100

Dr. Ting Wang
Professor
Clemson University
Department of Mechanical Engineering
Box 340921
Clemson, SC 29634-0921
(803) 656-5630
FAX: (803) 656-4435

Mr. Steve Waslo
U.S. Department of Energy
9800 S. Cass Avenue
Argonne, IL 60439
(708) 252-2143
FAX: (708) 252-8649

Dr. Richard Wenglarz
Senior Staff Research Engineer
Allison Engine Company
PO Box 420, S/C T14
Indianapolis, IN 46206
(317) 230-2185
FAX: (317) 230-3691

Mr. Daniel Whitney
Staff Consulting Engineer
Sacramento Munic. Util. Dept.
PO Box 15830
Sacramento, CA 95852-1830
(916) 732-5351
FAX: (916) 732-6423

Mr. Eric Wong
Senior Policy Advisor
California Energy Commission
1516 Ninth Street, MS #32
Sacramento, CA 95814
(916) 654-3992
FAX: (916) 654-4420

Dr. Sam Zamrik
Professor of Engr. Mech.
Penn State University
121 Hammond Bldg.
University Park, PA 16802
(814) 865-5241
FAX: (814) 865-5241

Mr. Charles Zeh
Product Manager, Advanced Heat Engines
METC
P.O. Box 880, MS0-DO1
3610 Collins Ferry Road
Morgantown, WV 26507-0880
(304) 285-4265
FAX: (304) 285-4403

Dr. Mark van Roode
CSGT Project Manager
Solar Turbines, Inc.
PO Box 85376
San Diego, CA 92186-5376
(619) 544-5549
FAX: (619) 544-2830

DISTRIBUTION

Jeff Abboud
Director, Gas Turbine Association
U.S. Advanced Ceramics Association
1600 Wilson Boulevard, Suite 1008
Arlington, VA 22209

Charles Alsup
METC
P.O. Box 880, MS0 DO1
3610 Collins Ferry Road
Morgantown, WV 26507-0880

Mr. Ronald Bannister
Manager, Gasification & Adv. Cycles
Westinghouse Electric Corp.
4400 Ala Faya Trail, MC 381
Orlando, FL 32826-2399

Dr. Katayun Barmak
Professor
Lehigh University
Dept. of Matls. Science & Engineering
5 E. Packer Avenue
Bethlehem, PA 18015

M. Arun Basu
Senior Research Engineer, Amoco
Amoco Research Center
MS E-1-F, Bldg. 300
Naperville, IL 60566

Mr. Paul Bautista
Product Manager, Power Generation
Gas Research Institute
8600 West Bryn Mawr Ave Avenue
Chicago, IL 60631

Mr. Charles Benson
Manager, Combustion Technology Unit
Arthur D. Little, Inc.
Acorn Park
Cambridge, MA 02140

Dr. Thomas Bernecki
Research Scientist
BIRL
1801 Maple Avenue
Evanston, IL 60201

Mr. Ralph Dalla Betta
CATALYTICA
430 Ferguson Drive
Mountain View, CA 94043

Mr. Frank Bevc
Manager, Emerging Tech.
Westinghouse Electric Corp.
4400 Ala Faya Trail, MC 381
Orlando, FL 32826-2399

Stan Blazewicz
U.S. Department of Energy
EE-232 Forrestal
1000 Independence Avenue SW
Washington, DC 20585

Mr. Donald Blomquist
Chief, Automated Production Tech. Div.
NIST
Rt. 270 & Clopper Road
Bldg. 233/Sound, Room B-108
Gaithersburg, MD 20899

Mr. Daniel Boss
Research Scientist
BIRL
1801 Maple Avenue
Evanston, IL 60201

Mr. Michael Brands
Executive Engineer
Cummins Engine Co.
Mail Code 20320, Box # 3005
Columbus, IN 47201

Mr. Richard Brent
Marketing Manager, ATS
Solar Turbines, Inc.
P.O. Box 85376
San Diego, CA 92186-5376

Dr. Scott Brewster
Research Associate
Brigham Young University
45 CTB
Provo, UT 84602

Mr. Warren Bunker
Program Manager
Department of Energy
FE-231, 1000 Independence Avenue, S.W.
Washington, DC 20585

Mr. R. D. Buxton
Power Generation Manager
Rolls-Royce, Inc.
11911 Freedom Drive
Reston, VA 22090

Mr. Brian Cahill
Hitachi of America
50 Prospect Avenue
Tarrytown, NY 10591

Mr. James Carr
Materials Engineer
Department of Energy
1000 Independence Avenue, S.W., FE-
Washington, DC 20585

Mr. Peter Carroll
VP, Govt. Relations
Solar Turbines, Inc.
818 Connecticut Avenue, N.W., Ste. 60
Washington, DC 20006-2702

Dr. Brent Carter
Assistant Professor
Georgia Tech
School of Materials Science & Engr.
Atlanta, GA 30332-0245

Dr. Kent Casleton
Research Chemist
METC
P.O. Box 880, M/S N05
Morgantown, WV 26505

Dr. M. R. Cates
Oak Ridge National Lab
P.O. Box 2008
K-1220, MS-7280
Oak Ridge, TN 37831-6007

Dr. Baki Cetegen
Associate Professor
University of Connecticut
ME Dept.
Storrs, CT 06269-3139

Mr. Tom Chance
General Electric Co.
Bldg. 53, Rm. 425
One River Road
Schenectady, NY 12345

Dr. Jyh-Yuan Chen
Associate Professor
University of California
6163 Etcheverry Hall
Berkeley, CA 94720

Dr. Ndaona Chokani
Associate Professor of Aerospace Eng.
North Carolina State Univ
Department of Mech. & Aerospace
EngBox 7910
Raleigh, NC 27695-7910

Mr. Mingking Chyu
Carnegie Mellon University
Mechanical Engineering
Pittsburgh, PA 15213-3890

Mr. Joe Cleveland
Mgr.-Operations Engineering
Department of Energy
ETEC, P.O. Box 7930
Canoga Park, CA 91304

Dr. Arthur Cohn
Technical Manager
EPRI
P.O. Box 10413
Palo Alto, CA 94303

Mr. James Corman
General Manager
GE Co. Advanced Tech
1 Rivers Road
Bldg. 53, Rm. 417
Schenectady, NY 12345

Mr. Richard De Corso
Mechanical Engineer
1075 West Martha's Court
Knoxville, MD 21758-9120

Dr. Arthur Cox
Manager-Adv. Matls./Process Engineering
Pratt & Whitney
MS 114-14, 400 Main Street
East Hartford, CT 06108

Mr. Ron Cox
ABB Power Generation Inc.
1200 Willis Road
Richmond, VA 23237

Mr. Kevin Craig
Senior Chemical Process Eng.
NREL
1617 Cole Blvd.
Golden, CO 80401

Mr. Phil Craig
Program Manager
Dupont Lanxide Composites
1300 Marrows Road, P.O. Box 6077
Newark, NJ 19714-6077

Mr. John Cuttica
Gas Research Institute
8600 West Bryn Mawr Avenue
Chicago, IL 60631

Mr. Michael Cybulsky
Sr. Consulting Engineer
ABB Combustion Engineering
1000 Prospect Hill Road
CEP: 900 7-0301
Windsor, CT 06095-0500

Dr. Brian Damiano
Oak Ridge National Lab
P.O. Box 2008, Bldg. 3500, MS 6010
Oak Ridge, TN 37831-6010

M. Thong Dang
Associate Professor
Syracuse University
149 Link Hall
Syracuse, NY 13244

Mr. Joseph Darguzas
Project Manager
Sargent & Lundy
55 E. Monroe
Chicago, IL 60603

Mr. Keith Davidson
Vice President, Project Development
On-Site Energy Corp.
701 Palomar Airport Road, Ste. 200
Carlsbad, CA 92002

Mr. Frank Davis
Cleveland Office Manager
AlliedSignal Aerospace
7550 Lucerne Drive, Ste. 203
Middleburg Heights, OH 44130-6503

Dr. William Day
Director, Gas Turbine Eng./New Prod.
Dev
Turbo Power & Marine Systems
P.O. Box 611, Aircraft Road
Middletown, CT 06457

Mr. Mario DeCorso
Vice President
Power Tech Associates, Inc.
925 Baltimore Pike
P.O. Box 521
Concordville, PA 19331

Dr. Paul Dellenback
Associate Professor
University of Wyoming
Mechanical Engineering
P.O. Box 3295
Laramie, WY 82071-3295

Mr. Elia Demetri
Senior Project Engineer
North Research & Eng. Corp.
39 Olympia Avenue
Woburn, MA 01801-2073

Mr. Victor Der
Office Director
Department of Energy
19901 Germantown Road, FE-73
Germantown, MD 20585

Dr. Vimal Desai
University of Central Florida
4000 Central Florida Boulevard
Orlando, FL 32816

Mr. Ihor Diackunchak
Advisory Engineer
Westinghouse Electric Corp.
4400 Ala Faya Trail, MC 381
Orlando, FL 32826-2399

Dr. Robert Dibble
Professor
University of CA at Berkeley
6159 Etcheverry Hall
Dept. of Mechanical Engineering
Berkeley, CA 94270

Sara Dillich
U.S. Department of Energy
EE-232 Forrestal
1000 Independence Avenue SW
Washington, DC 20585

Mr. Thomas Bechtel, Director
METC
P.O. Box 880, MS0-D01
3610 Collins Ferry Road
Morgantown, WV 26507-0880

Mr. Clark Dohner
Project Manager
ERPI
3412 Hillview Avenue, P.O. Box 10412
Palo Alto, CA 94303

Mr. Kenneth Dreitlein
Manager, Business Development
United Tech. Research Ctr.
411 Silver Lane-MS 24
East Hartford, CT 06108

Mr. Thomas Duffy
Manager, High Performance Steam
Solar Turbines, Inc.
P.O. Box 85736
San Diego, CA 92186-5376

Dr. David Duhl
Pratt & Whitney
MS 114-41, 400 Main Street
East Hartford, CT 06108

Dr. Michael Dunn
Vice President, Research Fellow
Calspan Advanced Tech. Ctr.
P.O. Box 400
Buffalo, NY 14225

Mr. Péter D'Alessandro
President
HSDE Inc.
14900 Woodham Drive, Ste. A-130
Houston, TX 77073

Mr. Chris Ebel
Business Development Manager
Carborundum Company
Niagara Falls Technology Center
P.O. Box 832
Niagara Falls, NY 14302

Mr. Jack Eisenhower
Energetics, Incorporated
7164 Gateway Drive
Columbia, MD 21046

Dr. William Ellingson
Senior Scientist
Argonne National Lab
9700 South Cass Avenue
Argonne, IL 60439

Dr. Abraham Engeda
Assistant Professor
Michigan State University
Mechanical Engineering Department
A231 Engineering Building
East Lansing, MI 48824-1226

Mr. David Eppinger
Fluor Daniel

Ms. Christine Ervin
Assistant Secretary
Department of Energy
Energy Efficiency and Renewable Energy
Washington, DC 20585

Mr. John Eustis
Department of Energy—Retired
28027 Playtor Road
Easton, MD 20585

Dr. Dan Fant
Research Manager
South Carolina Energy R&D Center
386-2 College Avenue
Clemson, SC 29634-5180

Mr. Lanny Feder
Asst. Director, Prog. Development
Institute of Gas Technology
1825 K Street, N.W., Ste. 503
Washington, DC 20006

Dr. Mattison Ferber
Group Leader, Mech. Prop. User Center
Oak Ridge National Lab
P.O. Box 2008
Bldg. 4515, MS-6064
Oak Ridge, TN 37831-6064

Dr. Raymond Fessler
Director
BIRL
1801 Maple Avenue
Evanston, IL 60201

Mr. L. Robert Flais
Principal Engineer
Babcock & Wilcox CO
R&D Division, 1562 Beeson Street
Alliance, OH 44601

Mr. Dick Fleming
CATALYTICA
430 Ferguson Drive
Mountain View, CA 94043

Mr. John Ford
President
Ford Associates
4382 Ampudia Street
San Diego, CA 92103

Dr. Steven Freedman
Executive Technology Manager
Gas Research Institute
8600 West Bryn Mawr Avenue
Chicago, IL 60631

Mr. Doug Freitag
Consultant
Dupont Lanxide Composites
17 Rocky Glen Court
Brookeville, MD 20833-6077

Dr. Albert Fresco
Market Development Manager
Dupont Lanxide Composites
1300 Marrows Road, P.O. Box 6077
Newark, DE 19714-6077

Mr. Robert Fuller
Manager, Marketing
Westinghouse Sci. & Tech. Ctr.
1310 Beulah Road
Pittsburgh, PA 15235

Mr. Stephen Gates
Director, ATS Programs
Solar Turbines, Inc.
2200 Pacific Coast Highway
San Diego, CA 92138

Mr. Donald Geiling
Project Manager
METC
P.O. Box 880, MS0-D6
3610 Collins Ferry Road
Morgantown, WV 26507-0880

Mr. Maurice Gell
Professor In Residence
University of Connecticut
U-136, Dept. of Metallurgy
Storrs, CT 06269

Dr. Randall Gemmen
Mechanical Engineer
METC
P.O. Box 880, MS N05
Morgantown, WV 26505

Mr. Madhav Ghatge
Division Director
METC
P.O. Box 880, MS0-0880
3610 Collins Ferry Road
Morgantown, WV 26507-0880

Mr. Robert Giglio
Sr. Development Engineer
Foster Wheeler Dev. Corp.
12 Peach Tree Hill Road
Livingston, NJ 07039

Ms. Patricia Godley
Assistant Secretary
Department of Energy
Fossil Energy
Washington, DC 20585

Dr. Lawrence Golan
Director
SC Carolina Energy Center
386-2 College Avenue
Clemson University
Clemson, SC 29634-5180

M. S. R. Gollahalli
Aerospace & Mechanical Engineer
University of Oklahoma
865 Asp., Rm. 212
Norman, OK 73019-0601

Dr. J. P. Gore
Professor
Purdue University
School of Mech. Engr.
West Lafayette, IN 47906

Dr. Heinz Gorges
President
VINETA, Inc.
3705 Sleepy Hollow Road
Falls Church, VA 22041

Mr. George Graves
Senior research Ceramist
Univ. Of Dayton Resrch. Inst.
300 College Park
Dayton, OH 45469-0172

Mr. Edward Gray
President
Antares Group, Inc.
8240 Professional Place, Ste. 207
Landover, MD 20875

Mr. Robert Gross
Project Manager
METC
P.O. Box 880, MS0-DO1
3610 Collins Ferry Road
Morgantown, WV 26507-0880

Ms. Varlerie Grubb
Engineering Exhibits Coordinator
Allison Engine Company
P.O. Box 420, Speed Code S7B
Indianapolis, IN 46206-0420

Mr. John Gruver
Executive Consultant
Westinghouse Electric Corp.
1801 K Street, N.W.
Washington, DC 20006

Dr. Gerald Guenette
Principal Research Engineer
Massachusetts Inst. of Tech.
77 Mass Avenue, Rm. 31-214
Gas Turbine Lab
Cambridge, MA 02139

Dr. Ashwani Gupta
Professor
University of Maryland
Department of Mechanical Engineering
College Park, MD 20742

Mr. David Hadacek
Principal Engineer, ATS
Solar Turbines, Inc.
2200 Pacific Highway
San Diego, CA 92186

Mr. Mark Hadley
Mechanical Engineer
METC
P.O. Box 880
Morgantown, WV 26505

Mr. Barry Haest
Westinghouse Electric Corp.
4400 Ala Faya Trail, MC 381
Orlando, FL 32826-2399

Dr. John Halow
Director, Combustion/Cleanup Proj. Div.
Department of Energy
P.O. Box 880
Morgantown, WV 26505

Mr. Michael Halter
Research Engineer
Clarkson University
c/o Ronald LaFleur
Box 5725, Dept. MAE
Potsdam, NY 13699-5725

Dr. Je-Chin Han
Professor
Texas A&M University
College Station, TX 77843-3123

Mr. John Hanigofsky
Senior Materials Scientist
TA&T, Inc
133 Defense Highway, Ste. 212
Annapolis, MD 21401

Dr. Prith Harasgama
Chief-Heat Transfer
ABB Power Generation Ltd.
Gas Turbine Dept., KWRB
Baden, CH-5401

Mr. Martin Harmer
Director, Materials Research Ctr.
Lehigh University
Whitaker Lab, 5 E. Packer Avenue
Bethlehem, PA 18015

Mr. Robert Harrison
Consultant
414 Waterbury Circle
Cincinnati, OH 45231-6064

Mr. George Hay
CAGT Program
42 Sunrise Lane
Lafayette, CA 94549

Dr. Bruce Hedman
Senior Program Manager
Pacific Northwest Lab
901 D Street S.W., Ste. 900
Washington, DC 20024

Dr. Paul Hedman
Brigham Young University
ACERC, Chem. E. Dept.
Provo, UT 84602

Mr. Jack Hill
Turbine Metallurgical Engineer
A. Finkl & Sons
P.O. Box 4279
Greenville, SC 29608

Dr. Robert Hillery
Manager, Airfoil Materials
GE Aircraft Engines
1, Neumann Way, Mail Drop H85
Cincinnati, OH 45215

Dr. Don Hindman
Principal Engineer
Babcock & Wilcox
Box 11165, LRC, MS 76
Lynchburg, VA 24506

Pat Hoffman
U.S. Department of Energy
EE-232 Forrestal
1000 Independence Avenue SW
Washington, DC 20585

Mr. E. Hoffman
Manager, Industrial Conservation Tech.
Department of Energy
1 Bethel Valley Road
Bldg. 45156, MS 6065
Oak Ridge, TN 37831-6065

Mr. Robert Holcomb
Oak Ridge National Lab
P.O. Box 2009
MS 8088, Bldg. 9108
Oak Ridge, TN 37831-6007

Dr. Norman Holcombe
Project Manager
METC
P.O. Box 880, MS0-DO1
3610 Collins Ferry Road
Morgantown, WV 26507-0880

Dr. William Hong
Research Staff Member
Inst. for Defense Analyses
1801 N. Beauregard Street
Alexandria, VA 22311

Dr. Diane Hooie
Project Manager
METC
P.O. Box 880, MS0-DO1
Morgantown, WV 26507-0880

Dr. John Kibler
Director of Engineering
Materials Sciences Corp.
500 Office Center Drive, Ste. 250
Ft. Washington, PA 19034.

Mr. Thomas Kircher
Sermatech International
155 S. Limerick Road
Limerick, PA 19468

Mr. David Knowles
Senior Program Analyst
MELE Associates
7428 West More Road
Rockville, MD 20850

Dr. Klod Kokini
Professor
Purdue University
1288 Mechanical Engineering Bldg.
West Lafayette, IN 47907-1288

Mr. William Koop
Wright Patterson AFB
1950 5th
Wright Patterson AFB, OH 45433-7251

Mr. Gil Kraemer
Research Engineer
Precision Combustion, Inc
25 Science Park
New Haven, CT 06511

Dr. Ronald LaFleur
Associate Professor
Clarkson University
Box 5725, Dept. MAE
Potsdam, NY 13699-5725

Mr. Ming-Chia Lai
Associate Professor
Wayne State University
Mechanical Engineering Dept.
Detroit, MI 48202

Dr. Lee Langston
Professor
University of Connecticut
ME Dept., U-139
191 Auditorium Road
Storrs, CT 06269-3139

Mr. John Latcovich
Manager, GT Machines
ABB Power Generation Inc.
1200 Willis Road
Richmond, VA 23209

Mr. Normand Laurendeau
Professor of Mechanical Engineering
Purdue University
School of Mech. Engr.
West of Lafayette, IN 47906

Dr. Patrick Lawless
Assistant Professor
Purdue University
School of Mech. Engr.
West LaFayette, IN 47906

Mr. Abbie Layne
Project Manager
METC
P.O. Box 880, MS0-D06
Morgantown, WV 26505

Dr. Woo Lee
Research Staff Member
Oak Ridge National Lab
P.O. Box 2008, MS 6063
Oak Ridge, TN 37831-6063

Dr. Gerald Leverant
Director
Southwest Research Institute
6220 Culebra Road
San Antonio, TX 78230

Mr. Mike Horner
Manager, Adv. Products
GE/M&I
5122 One Neumann Way
Cincinnati, OH 45215-6301

Mr. Michael Hsu
President
ZTEK Corporation
460 Totten Pond Road, Rm. 600
Waltham, MA 02154

Mr. Richard Hune
Marketing Representative
Westech Gear Corp.
117 N. Palmer
Houston, TX 77003

Mr. George Husman
Vice President, Engineering Div.
Southern Research Institute
757 Tom Martin Drive
Birmingham, AL 35211

Mr. Michael Izsak
Engineer
Burns and Roe Svcs. Corp.
P.O. Box 18288
Pittsburgh, PA 15236

Mr. Malcolm Jacobson
Enron Emerging Technologies

Mr. Michael Janus
Mechanical Engineer
METC
P.O. Box 880, MS N05
Morgantown, WV 26505

Ms. Cindy Jones
General Manager
Precision Engine Contr. Corp.
11661 Sorrento Valley Road
San Diego, CA 92121

Mr. Michael Jones
Manager, Power Generation Mktg.
Solar Turbines, Inc.
P.O. Box 85376
San Diego, CA 92180

Dr. Larry Joseph
Department of Energy
FE-231, 1000 Independence Avenue S.W.
Washington, DC 20585

Dr. Tony Kaiser
V.P. Power Plant Laboratories
ABB Combustion Engineering
1000 Prospect Hill Road
Windsor, CT 06095

Mr. Peter Kamber
Consulting Engineer
ABB Power Plant Labs
1000 Prospect Hill Road
Windsor, CT 06095

Dr. Jayanta Kapt
Visiting Assistant Professor
Clemson University
Department of Mechanical Engineering
Box 340921
Clemson, SC 29634-0921

Dr. Michael Karnitz
Program Manager
Oak Ridge National Lab
P.O. Box 2008, MS-6065
Oak Ridge, TN 37831-6065

Mr. Karl Karstensen
Program Manager
Solar Turbines, Inc.
P.O. Box 85376
San Diego, CA 92186-5376

Dr. Stanley Levine
Chief, Ceramics Branch
NASA Lewis Research Ctr.
21000 Brookpark Road, MS 106/5
Cleveland, OH 44135

Dr. Mark Lewis
Associate Professor
University of Maryland
College Park, MD 20742

Dr. James Leylek
Clemson University
Mechanical Engineering
313-B Riggs Hall
Clemson, SC 29634

Dr. James Liburdy
Clemson University
Dept. of mechanical /engineering
Clemson, SC 29634

Mr. S. Van Der Linden
Vice President
ABB Power Generation
1160 Livingston Avenue
North Brunswick, NJ 08092

Dr. John Lloyd
Distinguished Professor
Michigan State University
Mechanical Engineering
A231 Engineering Building
East Lansing, MI 48824-1226

Mr. Charles Logan
Director, Business Development
B. P. Chemicals (HITCO)
1600 W. 135th Street
Gardena, CA 90249

Mr. Bill Long
Technical Advisor
Babcock & Wilcox
Box 11165, LRC, MS 76
Lynchburg, VA 24506

Mr. Richard Lowden
Oak Ridge National Lab
P.O. Box 2008, MS 6087, Bldg. 4508
Oak Ridge, TN 37831-6087

Mr. Jimmy Lunsford
Vice President, Industrial Engines
Allison Engine Company
P.O. Box 420, MS U8
Indianapolis, IN 46206

Dr. Daniel Maloney
General Engineer
METC
P.O. Box 880, MS N05
Morgantown, WV 26505

Dr. Momtaz Mansour
President
MTCI
P.O. Box 21
Columbia, MD 21045

Mr. Thomas Marciniak
Manager, Ind. & Utility Technology
Argonne National Lab
9700 South Cass Ave.
Bldg. 362, Rm. C205
Argonne, IL 60439

Dr. Arnold Marder
Professor
Lehigh University
Whitaker Lab
Dept. of Matls. Science and Engineering
Bethlehem, PA 18017

Mr. Richard Marloff
Manager, Engineering System
Westinghouse Sci. & Tech. Ctr.
1310 Beulah Road
Pittsburgh, PA 15235

Mr. Joe Mavec
Program Manager-CFCC
Department of Energy
9800 S. Cass
Argonne, IL 60439

Mr. David McDeed
Mechanical Thermal engineer
Siemens Power Corp.
1040 S. 70th Street
Milwaukee, WI 53214

Mr. John McDonald
Dept. of Mechanical Engineering
Vanderbilt University
Box 1592, Station B
Nashville, TN 37235-1592

Mr. Jim McEvers
Senior Staff Engineer
Oak Ridge National Lab
P.O. Box 2008, MS 6003
Oak Ridge, TN 37831-6003

Ms. Jan McFarland
Office of Air and Radiation
Environmental Protection Agency
401 M Street S.W.
Washington, DC 20460

Dr. A. Mellor
Centennial Professor of Mech. Engr.
Box 1592, Station B
Nashville, TN 37235-1592

Mr. Michael Merlo
Manager of Tech. Planning & Dev.
Southern CA Edison Co.
P.O. Box 800, 2244 Walnut Grove Avenue
Rosemead, CA 91770

Dr. Nathan Messersmith
Assistant Professor
Purdue University
School of Aeronautics & Astronautics
1282 Grissom Hall
West LaFayette, IN 47907-1282

Dr. William Messner
Manager
Consolidated Edison
4 Irving Place, Rm. 1428
New York, NY 10003

Mr. Harold Miller
General Electric Corp.
One River Road
Bldg. 53, Room 425
Schenectady, NY 12345

Mr. Dennis Moran
American Gas Association
1515 Wilson Boulevard
Arlington, VA 22209-2470

Mr. Harry Morehead
Mgr., New Program Dev.
Westinghouse Electric Corp.
4400 Ala Faya Trail, MC 381
Orlando, FL 32826-2399

Mr. Dale Mukavetz
ATS Program Manager
Allison Engine Company
P.O. Box 420, S/C T12A
Indianapolis, IN 46206

Mr. James Mummert
Manager, Sales and Marketing
Allied Signal Fluid Systems
P.O. Box 22200
Tempe, AZ 85285-2200

Dr. S. N. B. Murthy
Professor
Purdue University
1003 Chaffee Hall
West LaFayette, IN 47907-1003

Mr. Tom Naslund
Wyman Gordon Invest. Castings
839 Poquonnock Road, P.O. Box 999
Groton, CT 06340-0999

Dr. Ken Natesan
Senior Scientist
Argonne National Lab
9700 South Cass Avenue
Argonne, IL 60439

Mr. Gary Neuner
Business Development Manager
Westinghouse Electric Corp.
401 East Hendy Avenue, P.O. Box 3499
Sunnyvale, CA 94088-3499

Mr. Don Newburry
Research Associate
METC
P.O. Box 880
3610 Collins Ferry Road
Morgantown, WV 26507-0880

Dr. Dimitris Nikitopoulos
Associate Professor
Louisiana State University
Mechanical Engineering
CEBA 251 2513B
Baton Rouge, LA 70803

Dr. Nirm Nirmalan
Staff Research Scientist
Allison Engine Company
P.O. Box 420, MS W16
Indianapolis, IN 46206

Mr. Paul Norton
Lead Design Engr. ATS
Solar Turbines, Inc.
2200 Pacific Highway
San Diego, CA 92128

Mr. Thomas Norton
Mechanical Engineer
METC
P.O. Box 880, MS N05
Morgantown, WV 26505

Mr. Armand Onesto
Program Manager
Department of Energy
ETEC, P.O. Box 7930
Canoga Park, CA 91309

Dr. Adeboyejo Oni
Morgan State University
School of Engineering
220 Mitchell Bldg.
Baltimore, MD 21239

Mr. Nick Otter
Director of Technology
European Gas Turbines
Cambridge Road, Whetstone
Leicester, LE8 6LH

Mr. George Padgett
Manager, Advanced Design Engineering
Solar Turbines, Inc.
P.O. Box 85376
San Diego, CA 92186-5371

Mr. Robert Pafe
Budget Analyst
Department of Energy
Office of CFO CR147
1000 Independence Avenue, S.W.
Washington, DC 20585

Mr. Al Pallotta
Mgr. Combustion Turbine Materials
Westinghouse Electric Corp.
4400 Ala Faya Trail, MC 381
Orlando, FL 32826-2399

Dr. Joseph Panzarino
Director of Government Programs
Saint/Gobain/Norton
Goddard Road
Northboro, MA 05132

Mr. William Parks
Department of Energy
1000 Independence Avenue, S.W.
CE-222, 5F/035 FORS
Washington, DC 20585

Mr. Edward Parsons
Chief, Power Gen. Tech. Branch
METC
P.O. Box 880, MS0-DO6
Morgantown, WV 26505

Mr. Yakin Patel
Graduate Student
University of Central FL
4000 Central Florida Boulevard
Orlando, FL 32816

Mr. Leland Paulson
METC
P.O. Box 880
Morgantown, WV 26507

Dr. Stephen Pope
Professor
Cornell University
Upson Hall
Ithaca, NY 14853

Mr. Henry Price
Senior Engineering Analyst
NREL
1617 Cole Boulevard
Denver, CO 80401

Mr. Jeffery Price
Senior Ceramic Engineer
Solar Turbines, Inc.
2200 Pacific Highway, P.O. Box 85376
San Diego, CA 92186-5376

Mr. James Prochaska
Director of North Am. Sales
Stewart & Stevenson
2707 N. Loop West
Houston, TX 77008

Dr. Ishwar Puri
Associate Professor
Univ. of Illinois at Chicago
Department of Mech. Eng.-M/C 251
842 W. Taylor Street, Rm. 20339
Chicago, IL 60607-7022

Mr. James Quinn
Office of Industrial Technologies
Department of Energy
100 Independence Avenue, S.W.
EE-20, Rm. 64-025
Washington, DC 20585

Mary Rawlins
Department of Energy
Oak Ridge Office of Operations
Building 4500N, MS 6269
P.O. Box 2008
Oak Ridge, TN 37831-6269

Dr. Mohan Razdan
Chief of Combustor R&D
Allison Engine Company
P.O. Box 420, Speed Code T14
Indianapolis, IN 46206-0420

Mr. Jim Reed
Energetics, Incorporated
7164 Gateway Drive
Columbia, MD 21046

Mr. Michael Reitz
Budget Analyst
Department of Energy
1000 Independence Avenue, S.W.
Washington, DC 20002

Mr. Ivan Rice
Consulting Engineer
P.O. Box 233
Spring, TX 77383

Dr. George Richards
Principal Investigator
Department of Energy
METC
P.O. Box 880
Morgantown, WV 26505

Mr. Tom Richards
Manager-New Technology
Caterpillar, Inc.
P.O. Box 1856
Peoria, IL 61656

Mr. Mike Roberts
Institute of Gas Technology
1825 K Street, N.W., Ste. 503
Washington, DC 20006

Mr. Paul Rodebaugh
Supervising Engr.
Florida Power & Light
P.O. Box 14000
Juno Beach, FL 33408-0420

Dr. Larry Roe
University of Arkansas
Mechanical Engineering Dept.
MEEG 204
Fayetteville, AR 72701

Mr. David Rohy
Director Advanced Turbine Systems
Solar Turbines, Inc.
P.O. Box 85376
San Diego, CA 92186-5376

Dr. Glenn Romanoski
Staff Development Engineer
Oak Ridge National Lab
P.O. 2008
Bldg. 4508, MS-6088
Oak Ridge, TN 37831-6064

Mr. Sims Roy
Environmental Engineer
US EPA
Combustion Group, MD-13
Research Triangle Pa, NC 27711

Dr. Ramendra Roy
Professor of Mechanical Engineering
Arizona State University
Dept. of Mechanical & Aerospace Eng.
Tempe, AZ 85287-6106

Dr. Purush Sahoo
Sermatech International
155 S. Limerick Road
Limerick, PA 19468

Mr. Louis Salvador
Associate Director
METC
Office Of Product Technology Mgmt.
P.O. Box 880
Morgantown, WV 26507-0880

Dr. Scott Samuelson
Professor
Univ. of CA, Irvine
UCI Combustion Lab
Irvine, CA 92717-3550

Dr. A. T. Sanders
Team Leader, Mechanical Technology
European Gas Turbines
P.O. Box 1
Lincoln, LN6 7AA

Dr. M. Taher Schobeiri
Associate Professor
Texas A&M University
Mechanical Engineering Dept.
College Station, TX 77843-3123

Mr. Robert Schultz
Program Manager
Department of Energy
EE-34, Forrestal Bldg.
Washington, DC 20585

Mr. Andrew Scriven
Research and Engineering Dept.
National Power (UK)
National Power Place, White Hill WA
Trigonos, Windmill Hill Business PA
Swindon, Wiltshire SN5 6PB

Mr. Shane Sedell
Engineer
RTT
1819 Sycamore Valley
Reston, VA 22090

Dr. Vijay K. Sethi
Senior Staff Research Engineer
365 North 9th Street
P.O. Box 3395, University Station
Laramie, WY 82071-3395

Mr. Sam Shepard
Senior Engineer for R&D
Naval Sea Systems Command
Code 03X3C
Washington, DC 20362

Dr. Terrence Simon
Professor
University of Minnesota
Mechanical Engineering Department
111 Church Street
Minneapolis, MN 55455-0111

Dr. J. P. Singh
Scientist
Argonne National Lab
9700 South Cass Avenue, Bldg. 212
Argonne, IL 60439

Dr. Edward Skolnik
Senior Scientist
Energetics, Incorporated
7164 Gateway Drive
Columbia, MD 21046

Dr. Kenneth Smith
Chief Engineer, Combustion
Solar Turbines, Inc.
P.O. Box 85376
San Diego, CA 92119

Dr. Paul Sojka
Associate Professor
Purdue University
School of Mech. Engr.
West Lafayette, IN 47907-1003

Mr. Leslie Southall
Mgr. CT Engine Programs
Westinghouse Electric Corp.
450 Timber Ridge
Longwood, FL 32779

Mr. Steven Sperling
Undergraduate Research Fellow
Clarkson University
c/9 Ronald LaFleur
Box 5725, Dept. MAE
Potsdam, NY 13699-5725

Dr. J. Day, Sr.
Development Associate
Corning Incorporated RD&E
SP-DV-2-1
Coring, NY 14831

Mr. Robert Steele
PHD Candidate
University of Washington
Mechanical Engineering, FU-10
Seattle, WA 98195

Mr. David Stinton
Staff Scientist
Oak Ridge National Lab
P.O. Box 2008
Bldg. 4515, MS-6063
Oak Ridge, TN 38731-6063

Mr. Ted Strickland
Development Engineer
Precision Combustion, Inc.
25 Science Park
New Haven, CT 06511

Dr. Willard Sutton
Principal Scientist
United Tech. Research Ctr.
411 Silver Lane-MS 24
East Hartford, CT 06108

Mr. Stephan Sylvan
Environmental Protection Specialist
EPA
401 M Street S.W.
Washington, DC 20460

Dr. Widen Tabakoff
Professor
University of Cincinnati
Dept. of Aerospace Engines
Mail Drop 70
Cincinnati, OH 45221

Mr. Rick Tetzloff
Black & Veatch
8400 Ward Parkway
Kansas City, MO 64114

Dr. A. B. Thakker
Senior Manager
Rolls Royce, Inc.
2849 Paces Ferry Road
Atlanta, GA 30339

Dr. Thomas Tom
Director, Advanced Technology
Howmet Corporation
1500 South Warner Street
Whitehall, MI 49461

Mr. George Touchton
Target Manager
Electric Power Research Inst.
3412 Hillview Avenue
Palo Alto, CA 94304

Dr. John Tuzson
John Tuzson & Assoc.
1220 Maple Avenue
Evanston, IL 60202

Mr. Robert Tyson
Associate Director, RD&D Programs
American Gas Association
1515 Wilson Boulevard
Arlington, VA 22209-2470

Dr. Uri Vandsburger
Virginia Tech
Mechanical Engineering Dept.
Blacksburg, VA 24061-0238

Mr. Jim VanLaar
Project Manager
Sargent & Lundy
55 E. Monroe
Chicago, IL 60603

Dr. Mark VanRoode
CSGT Project Manager
Solar Turbines, Inc.
P.O. Box 85376
San Diego, CA 92186-5376

Mr. Robert Steele
PHD Candidate
University of Washington
Mechanical Engineering, FU-10
Seattle, WA 98195

Mr. David Stinton
Staff Scientist
Oak Ridge National Lab
P.O. Box 2008
Bldg. 4515, MS-6063
Oak Ridge, TN 38731-6063

Mr. Ted Strickland
Development Engineer
Precision Combustion, Inc.
25 Science Park
New Haven, CT 06511

Dr. Willard Sutton
Principal Scientist
United Tech. Research Ctr.
411 Silver Lane-MS 24
East Hartford, CT 06108

Mr. Stephan Sylvan
Environmental Protection Specialist
EPA
401 M Street S.W.
Washington, DC 20460

Dr. Widen Tabakoff
Professor
University of Cincinnati
Dept. of Aerospace Engines
Mail Drop 70
Cincinnati, OH 45221

Mr. Rick Tetzloff
Black & Veatch
8400 Ward Parkway
Kansas City, MO 64114

Dr. A. B. Thakker
Senior Manager
Rolls Royce, Inc.
2849 Paces Ferry Road
Atlanta, GA 30339

Dr. Thomas Tom
Director, Advanced Technology
Howmet Corporation
1500 South Warner Street
Whitehall, MI 49461

Mr. George Touchton
Target Manager
Electric Power Research Inst.
3412 Hillview Avenue
Palo Alto, CA 94304

Dr. John Tuzson
John Tuzson & Assoc.
1220 Maple Avenue
Evanston, IL 60202

Mr. Robert Tyson
Associate Director, RD&D Programs
American Gas Association
1515 Wilson Boulevard
Arlington, VA 22209-2470

Dr. Uri Vandsburger
Virginia Tech
Mechanical Engineering Dept.
Blacksburg, VA 24061-0238

Mr. Jim VanLaar
Project Manager
Sargent & Lundy
55 E. Monroe
Chicago, IL 60603

Dr. Mark VanRoode
CSGT Project Manager
Solar Turbines, Inc.
P.O. Box 85376
San Diego, CA 92186-5376

Mr. Rick Vosburgh
President, CEO
Monitoring Technology Corp.
2731 Prosperity Avenue
Fairfax, VA 22031

Dr. Ting Wang
Professor
Clemson University
Department of Mechanical Engineering
Box 340921
Clemson, SC 29634-0921

Mr. Steve Waslo
Department of Energy
9800 S. Cass Avenue
Argonne, IL 60439

Ms. Cecilia Wasterlid
Reporter-Technical Outlook
Embassy of Sweden
1501 M Street N.W.
Washington, DC 20005

Mr. Harvey Weisenfeld
Energetics, Incorporated
7416 Gateway Drive
Columbia, MD 21046

Dr. Richard Wenglarz
Senior Staff Research Engineer
Allison Engine Company
P.O. Box 420, S/C T14
Indianapolis, IN 46206

Mr. James Wessel
Dir., Fed. Business Dev.
Dow Corning Corporation
1800 M Street, N.W. #325 South
Washington, DC 20036

Mr. Pat West
Project Engineer
U.S. Generating
7500 Old Georgetown Road
Bethesda, MD 20814

Mr. Michael Wherley
Mechanical engineer
SAIC
1710 Goodridge Drive, MS 2-7-12
McLean, VA 22102

Dr. Arvel White
1912 MacArthur Street
Rancho Palos Verde, CA 90275

Mr. Daniel Whitney
Staff Consulting Engineer
Sacramento Munic. Util. Dept.
P.O. Box 15830
Sacramento, CA 95852-1830

Mr. Laurence Wilken
Business Manager, ATS
Solar Turbines, Inc.
P.O. Box 85376
San Diego, CA 92816-5376

Mr. John Wills
Director, Business Development
Rolls-Royce, Inc.
IPG Program
11911 Freedom Drive
Reston, VA 22090

Mr. Patrick Wing
Undergraduate Research Fellow
Clarkson University
c/o Ronald LaFleur
Box 5725, Dept. MAE
Potsdam, NY 13699-5725

Mr. Eric Wong
Senior Policy Advisor
California Energy Commission
1516 Ninth Street, MS #32
Sacramento, CA 95814

Ian Wright
Oak Ridge National Laboratories
Building 4500S, MS 6157
P.O. Box 2008
Oak Ridge, TN 37831-61572

Dr. Suxing Wu
Lehigh University
MRC, 5 Lehigh Univ., Whitaker Lab
Bethlehem, PA 18015

Mr. S. C. Yao
Professor
Carnegie Mellon University
Department of Mechanical Engineering
Pittsburgh, PA 15213

Dr. Rich Yetter
Research Scientist
Princeton University
Dept. of Mech. & Aerospace Eng.
O-312 Equad
Princeton, NJ 08544

Dr. M. Joseph Yip
Mechanical Engineer
METC
P.O. Box 880
Morgantown, WV 26505

Mr. Cherif Youssef
Industrial Manager
Southern California Gas
P.O. Box 3249
Los Angeles, CA 90051-1249

Dr. Sam Zamrik
Professor of Engr. Mech.
Penn State University
121 Hammond Bldg.
University Park, PA 16802

Mr. Charles Zeh
Product Manager, Advanced Heat Engines
METC
P.O. Box 880, MS0-DO1
3610 Collins Ferry Road
Morgantown, WV 26507-0880

Mr. Martin Zhu
Graduate Student
University of Maryland
Department of Mechanical Engineering
College Park, MD 20742