Advanced Turbine System (ATS) Program Conceptual Design and Product Development

**Quarterly Report
December 1, 1993 - February 28, 1994**

Work Performed Under Contract No.: DE-AC21-93MC30244

For
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Office of Fossil Energy
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REPORTING PERIOD: 12/1/93 - 2/28/94

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1.0 EXECUTIVE SUMMARY

General Electric Advanced Turbine Systems Program

GE has achieved a leadership position in the worldwide gas turbine industry in both industrial/utility markets and in aircraft engines. This design and manufacturing base plus our close contact with the users provides the technology for creation of the next generation advanced power generation systems for both the industrial and utility industries. GE has been active in the definition of advanced turbine systems for several years. These systems will leverage the technology from the latest developments in the entire GE gas turbine product line. These products will be USA based in engineering and manufacturing and are marketed through the GE Industrial and Power Systems.

Achieving the advanced turbine system goals of 60% efficiency, 8 ppmvd NO_x and 10% electric power cost reduction imposes competing characteristics on the gas turbine system. Two basic technical issues arise from this. The turbine inlet temperature of the gas turbine must increase to achieve both efficiency and cost goals. However, higher temperatures move in the direction of increased NO_x emission. Improved coating and materials technologies along with creative combustor design can result in solutions to achieve the ultimate goal.

GE's view of the market, in conjunction with the industrial and utility objectives requires the development of Advanced Gas Turbine Systems which encompasses two potential products: a new aeroderivative combined cycle system for the industrial market and a combined cycle system for the utility sector that is based on an advanced frame machine.

The GE Advanced Gas Turbine Development program is focused on two specific products.

1. A 70 MW class industrial gas turbine based on the GE90 core technology utilizing an innovative air cooling methodology.
2. A 200 MW class utility gas turbine based on an advanced GE heavy duty machines utilizing advanced cooling and enhancement in component efficiency.

Both of these activities require the identification and resolution of technical issues critical to achieving Advanced Turbine System (ATS) goals. The emphasis for the industrial ATS will be placed upon innovative cycle design and low emission combustion. The emphasis for the utility ATS will be placed upon innovative cycle design and low emission combustion. The emphasis for the utility ATS will be placed on developing a technology base for advanced turbine cooling while utilizing demonstrated and planned improvements in low emissions combustion. Significant overlap in the development programs will allow common technologies to be applied to both products. GE's Industrial and Power Systems is solely responsible for offering GE products for the industrial and utility markets. The GE ATS program will be managed fully by this organization with core engine technology being supplied by GE Aircraft Engines (GEAE) and fundamental studies supporting both product developments being conducted by GE Corporate Research and Development (CRD). GE's worldwide experience in commercialization of these products will ensure that the ATS program can proceed to the marketplace.

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STATUS OF TASK 3 THROUGH TASK 8

Task 3 - Gas Fired ATS Selection

Work continued on GFATS cycle selection. Task 3A (Industrial GFATS) progress is described in the Tab 3 section. Task 3B (Utility GFATS) progress is described in the Tab 4 section.

Task 4 - Conversion to Coal

No progress in this reporting period.

Task 5 - GFATS Market Study

No progress in this reporting period.

Task 6 - System Definition and Analysis

No progress this reporting period.

Task 7 - Integrated Program Plan

No progress this reporting period.

Task 8 - Component Design and Test

Task 8 subtask progress is described in the Tab 5 section.

3.1 CYCLE ANALYSIS

Derivatives of simple and combine cycle performance were determined with respect to gas turbine component performance and cycle parameters, turbocooler circuit airflows, and bottoming cycle performance. The results determine ATS performance sensitivity to thermodynamic inputs and highlight components and parameters critical to achieving project goals.

STUDY VARIABLES

The derivatives were determined using the configuration shown schematically in Figure 1 with the variables indicated. This is the direct drive, turbo-cooled, 1.75 booster pressure ratio, 2700°F firing temperature design. The compressor size and speed are fixed at levels consistent with GE90 sea level static operation at the 88,800 lb. thrust level. All cases were designed to constant inlet and exhaust (stack) pressure losses. LP shaft speed was held at 3,600 RPM.

Derivatives were calculated for emissions, thermal efficiency, power output and high pressure turbine flow function (HPTFF). Emission derivatives reflect values estimated based on flame temperature only. Thermal efficiency and power output derivatives were calculated for both simple and combined cycle systems. Combined cycle derivatives are calculated from a simplified three pressure level non-reheat steam cycle method. Also, high pressure turbine flow function (HPTFF) derivatives were calculated because changes in HPT size are constrained by the need to use GE90 hardware. Baseline values of the independent variables are listed in Table 1.

The derivatives were calculated with respect to independent variables including gas turbine and combined cycle component and cycle thermodynamic parameters, turbocooler circuit airflows, and inter-turbine reheat variables. Affected components and parameters are highlighted in Figure 1. Gas turbine variables include component efficiencies, pressure losses, and parasitic flows. Component efficiencies are defined as adiabatic except the booster which is defined as polytropic. Pressure losses are defined by ratio of component inlet to exit total pressure. Parasitic flows are defined by a fraction of HPC inlet physical flow. In addition, high pressure compressor (HPC) inlet guide vane (IGV) angle, which regulates HPC flow and efficiency, was evaluated. Also, an estimated derivative on steam cycle efficiency was included. Thermodynamic (cycle) variables include booster pressure ratio and firing temperature. These independent variables are defined further and their baseline values are listed in the left columns of Table 2.

Turbocooler circuit flows include burner, HPT nozzle, and HPT stage 1 rotor cooled cooling air and turbocooler turbine discharge air. Turbocooler independent variables are listed along with source, sink, and baseline values in the top rows of Table 4. They are highlighted in Figure 1.

Performance derivatives were also examined for a reheat system which could be placed between the HPT and power turbine. The reheat system independent variables include burner firing temperature and pressure loss and power turbine cooling flow. These variables are listed in Table 5 and highlighted in Figure 1.

RESULTS

The results reflect partial derivatives calculated by varying independent variables separately and resizing the engine system around a fixed HPC flow size. The derivatives reflect a delta of one point (1 point = 0.01) in independent variable as defined in Table 2, col. 2 and Table 4, row 1, except for firing temperature (T41) and HPC IGV (STP25) setting. T41

derivatives are per degree Fahrenheit and STP25 derivatives are per degree of vane angle. With these exceptions, the derivatives are defined by:

- o NOX: Δ emission units per point
- o Combined and simple cycle power output: $\% \Delta$ horsepower per point
- o Combined and simple cycle thermal efficiency: change in points per point
- o HPT flow function: $\% \Delta$ W41R per point

Table 4, listing derivatives with respect to turbocooler flows, contains additional data defining how the turbo-cooling system reacts to turbocooler flow changes. This is because turbocooler parameters must satisfy turbocooler operability requirements. For instance, the heat exchanger cannot function at an effectiveness greater than one and the turbo-cooler turbine exit air pressure must be sufficient to re-enter the flowpath downstream of the HPT. For this reason, the following additional parameters are listed (these are not derivatives):

- o PS42: static pressure downstream of HPT exit (Turbocooler turbine sink pressure)
- o P80: turbocooler turbine discharge manifold pressure (P80 is calculated such that cooled cooling flow is delivered at a manifold pressure of P3)
- o ϵ_{HX} : heat exchanger effectiveness
- o T_{fuel} : temperature of fuel entering combustor

Since the above parameters are not derivatives, the result listed is due to an **increase** of one point in flow fraction. These parameters also are highlighted in Figure 1.

Emissions Index derivatives are defined in Tables 2 and 4. Only firing temperature, cooling flow (cooled and uncooled), and parasitic flow derivatives are nonzero: these are the only variables affecting burner fuel to air ratio. Cooled burner/vane air has a strong impact because this air is the only dilution air between burner and HPT rotor inlet: if T41 is fixed, this air sets the combustor flame temperature, hence emission level. The remaining parasitic flows have a second order effect: if T41 and dilution airflow are constant, dilution airflow becomes a larger fraction of total burner airflow as larger parasitic flows bypass the combustor. Hence, flame temperature increases to maintain T41 and emissions increase.

Combined cycle efficiency derivatives are influenced by a variety of parameters. HPC and LPT efficiencies dominate in component efficiency effects. Pressure loss results are about even regardless of engine location. The parasitic flow results show that the derivative is a function of port location, with high pressure air more costly than low pressure air. Also, the further downstream the discharge port is, the greater the loss. For example, seventh stage balance piston air is more expensive than fourth stage air, and dumping seventh stage air in the stage two HPT rotor is less costly than if it were needed to cool the LPT rotor. Booster pressure ratio and T41 derivatives are small but can have significant influence if large changes occur. Increasing T41 and reducing booster PR have positive effects but the selected values are dependent on other factors such as cost, emissions, and size. The derivative on bottoming (steam) cycle efficiency simply reflects the contribution of the steam cycle to the total system power. That is, the derivative becomes stronger as steam cycle power becomes a larger fraction of total power.

Combined cycle power derivatives are impacted strongly by HPC, LPT, and gooseneck pressure loss with other parameters having less effect. The HPC efficiency is high due to the combined effects of reducing the HPC power requirement and dropping compressor discharge temperature (T3). Hence, these effects both provide more energy to the power turbine (HPT extracts less power) and increase burner temperature rise (T41-T3) as HPC efficiency increases. LPT performance provides for more simple cycle shaft power as LPT efficiency rises. Gooseneck pressure loss affects power output by increasing physical HPC airflow for a fixed

HPC size. Hence, as pressure loss drops, airflow rises, and fuel flow (constant fuel to air ratio) and power output increase. In addition, parasitic flows affect CC power the same way they impact efficiency - high pressure air is more costly than low pressure air. Turbocooler circuit flows impact power in a similar way: any airflow bypassing the burner results in a loss of fuel flow, hence power. Again, T41 and booster PR are influential but subject to other constraints.

The derivatives of simple cycle performance listed in Table 3 follow similar trends to CC derivatives but are stronger. The derivatives are stronger because cycle inefficiencies converted into waste heat are partially recovered in the bottoming cycle which tends to reduce the effect of inefficiency on combined cycle performance. HPC and LPT efficiency have a strong impact on efficiency and power output. Gooseneck pressure loss drives power output and parasitic and cooling flows drive emissions, efficiency, and power. In addition, although T41 and booster PR can have strong influences, these parameters are set by external considerations.

The HPC IGV setting derivatives show that small increases in efficiency are available but at a significant cost of power output. The baseline setting of zero degrees was picked mainly to maximize airflow and thus power.

Derivatives of HPTFF are also shown in Table 3 along with the corresponding influence on HPT efficiency. The idea here is that as flow function (area) deviates from its design value (about 30), HPT efficiency drops as it becomes harder to adapt the GE90 HPT to the ATS. This effect is included in the simple and combined cycle efficiency derivatives.

The results of the reheat study are listed in Table 5. The primary benefit of reheat is it allows a higher gas turbine exit temperature (T5) which increases steam cycle efficiency. However, simple cycle efficiency falls. In addition, the reheat burner requires some sort of pressure drop and a higher power turbine inlet temperature may require a cooled power turbine. For example, a reheat burner pressure drop of 6% requires 115 degrees of temperature rise to maintain constant CC efficiency. Two percent in cooling flow has about the same impact. This means a reheat system would require a critical balance between temperature rise, pressure drop, and cooling flows if it is to have any benefit in system thermal efficiency. Emissions index for a reheat system was not evaluated.

RECOMMENDATIONS

It is important to remember that these derivatives are intended to provide guidance in selecting a final cycle design: they should allow for making small tradeoffs rationally. The derivatives will be accurate for small deviations in the baseline cycle defined in Table 1. However, accuracy will deteriorate with larger deviations.

FIGURE 1: ATS DERIVATIVE STUDY SCHEMATIC AND VARIABLES

BASELINE ATS PRELIMINARY PERFORMANCE

EINOX	6.908
η_{CC} (PTS)	58.4
POWER _{CC} (MW)	97.0
W41R	32.2 (GE90 + 8%)
η_{SC} (PTS)	46.55
POWER _{SC} (MW)	77.3
ϵ_{HX}	0.830
T _{fuel} (°F)	965
PS42 (PSI)	109.1
PT80 (PSI)	178.9
η_{HPT} (PTS)	92.34

Table 1

<u>COMPONENT</u>	<u>BASELINE VALUE</u>	<u>EMISSIONS $\Delta EI/PT$</u>	<u>COMBINE CYCLE $\Delta \eta(PTS)/PT$</u>	<u>POWER $\% \Delta HP/PT$</u>
BOOSTER	$\eta_p = 0.92$	0	+ 0.063	+ 0.367
HPC	$\eta_a = 0.8536$	0	+ 0.250	+ 1.097
HPT	$\eta_a = 0.9234$	0	+ 0.197	+ 0.329
LPT	$\eta_a = 0.9405$	0	+ 0.274	+ 0.462
BC EFFICIENCY	$\eta_{ST} = 0.85$	0	+ 0.160	+ 0.269
GOOSENECK	P/P = 0.982	0	+ 0.115	+ 1.215
BURNER	P/P = 0.9467	0	+ 0.103	+ 0.173
REAR FRAME	P/P = 0.96	0	+ 0.102	+ 0.291
4TH STG BP AIR	W/W25 = 0.0144	+ 0.034	- 0.272	- 1.601
7TH STG BP AIR	W/W25 = 0.0	+ 0.036	- 0.356	- 1.725
HPT STG 2 AIR	W/W25 = 0.0326	+ 0.036	- 0.199	- 1.464
BOOSTER PR	P/P = 1.75	0	- 0.003	+ 0.196
FIRING TEMP.	T41 = 2700°F	+ 0.033°F	+ 0.010PTS/°F	+ 0.126%/°F
HPC IGV SETTING	STP25 = 0	0	+ 0.043PTS/°	- 0.984%/°

Table 2

<u>COMPONENT</u>	<u>BASELINE VALUE</u>	<u>SIMPLE CYCLE $\Delta\eta$(PTS/PT)</u>	<u>SIMPLE CYCLE %ΔHP/PT</u>	<u>HPTFF %ΔW41B/PT</u>	<u>$\Delta\eta_{HPT}$(PTS/PT)</u>
BOOSTER	$\eta_p = 0.92$	+ 0.083	+ 0.440	+ 0.104	- 0.020
HPC	$\eta_a = 0.8536$	+ 0.344	+ 1.424	+ 0.011	- 0.088
HPT	$\eta_a = 0.9234$	+ 0.395	+ 0.850	0	-
LPT	$\eta_a = 0.9405$	+ 0.550	+ 1.183	0	0
BC EFFICIENCY	$\eta_{ST} = 0.85$	0	0	0	0
GOOSENECK	P/P = 0.982	+ 0.203	+ 1.015	0	0
BURNER	P/P = 0.9467	+ 0.208	+ 0.450	- 1.025	0
REAR FRAME	P/P = 0.96	+ 0.206	+ 0.442	0	0
4TH STG BP AIR	W/W25 = 0.0144	- 0.284	- 1.754	- 1.277	+ 0.040
7TH STG BP AIR	W/W25 = 0.0	- 0.447	- 2.078	- 1.260	+ 0.058
HPT STG 2 AIR	W/W25 = 0.0326	- 0.134	- 1.419	- 1.271	+ 0.058
BOOSTER PR	P/P = 1.75	+ 0.004	+ 0.163	+ 0.008	- 0.0016
FIRING TEMP.	T41 = 2700°F	+ 0.0036PTS/°F	+ 0.084%/°F	+ 0.020%/°F	- 0.0036%/°F
HPC IGV SETTING	STP25 = 0	+ 0.035PTS/°	- 0.533%/°	- 0.281%/°	- 0.042%/°

Table 3

<u>PARAMETER</u>	<u>COOLED VANE/BURNER AIR</u>	<u>COOLED STAGE 1HPT ROTOR AIR</u>	<u>ATC TURBINE DISCHARGE AIR</u>
BASELINE W/W25	0.08	0.0252	0.021
SOURCE	ATC COMPRESSOR	ATC COMPRESSOR	ATC TURBINE
SINK	HPT NOZZLE, BURNER	HPT STAGE 1 ROTOR	LPT NOZZLE/DUCT
DERIVATIVES:			
$\Delta E_I/PT$	+ 0.400	+ 0.038	+ 0.037
$\Delta \eta_{CC}(PTS)/PT$	+ 0.009	- 0.280	- 0.255
$\% \Delta HP_{CC}/PT$	- 0.026	- 1.941	- 1.555
$\% \Delta W_{41R}/PT$	- 0.109	- 1.260	- 1.249
$\Delta \eta_{SC}(PTS)/PT$	+ 0.027	- 0.216	- 0.246
$\% \Delta HP_{SC}/PT$	+ 0.016	- 1.934	- 1.655
OTHER			
ϵ_{HX}	0.900	0.911	0.838
$T_{fuel} (^{\circ}F)$	1040	1052	975
PS42 (PSI)	109.2	106.8	107.0
PT80 (PSI)	157.7	157.7	261.7
$\Delta \eta_{HPT}(PTS)/PT$	- 0.001	+ 0.081	+ 0.079

Table 4

<u>PARAMETER</u>	<u>R/H BURNER TEMPERATURE RISE</u>	<u>R/H BURNER PRESSURE RATIO</u>	<u>LPT ROTOR (4TH STAGE HPC) COOLING FLOW</u>
BASELINE (NO REHEAT)	$\Delta T = 0$	$P/P = 1.0$	$W/W_{25} = 0$
DERIVATIVES:			
$\Delta \eta_{CC}/PT$	+ 0.005/°F	+ 0.096	- 0.341
$\% \Delta HP_{CC}/PT$	+ 0.079/°F	- 0.165	- 1.196
$\Delta \eta_{SC}/PT$	- 0.007/°F	+ 0.200	- 0.544
$\% \Delta HP_{SC}/PT$	+ 0.055/°F	+ 0.446	- 1.826

Table 5

3.2 COMBINED CYCLE STUDIES

Using configuration 4, shown in Table 6, combined cycle studies were completed to determine combined cycle efficiency and power output. Two studies were completed with the following attributes:

	Lower Heating Value- Natural Gas	Gas Turbine Exhaust Back Pressure
Study 1	19000 Btu/lbm.	10 in. H ₂ O
Study 2	21515 Btu/lbm.	15 in. H ₂ O

The combined cycle analysis were made using the following assumptions:

- A standard production Rankin steam bottoming cycle with three pressure levels and no reheat.
 - One steam turbine per gas turbine.
 - A condenser pressure of 1.2 in. Hg.
 - A generator loss of 1.2%

The results of the combined cycle studies are shown in Figure 3 compared to the combined cycle goal of 59.2% and the program emission goals. The results show that the emission goal for NO_x can be met, but there needs to be more cycle development to meet the combined cycle efficiency goal.

ADVANCE TURBINE SYSTEMS(ATS) PROGRAM

Table 6

ATS ENGINE CYCLES UNDER STUDY

FEATURES	1*	2*	3	4	5	6	7
BOOSTER/DRIVE TURBINE	NONE	YES/LOW PRESSURE TURBINE	YES/LOW PRESSURE TURBINE	YES/LOW PRESSURE TURBINE	YES/LOW PRESSURE TURBINE	YES/LOW PRESSURE TURBINE	YES/HIGH PRESSURE TURBINE
BOOSTER DRIVE TYPE	NONE	DIRECT DRIVE	DIRECT DRIVE	DIRECT DRIVE	DIRECT DRIVE	DIRECT DRIVE	GEARED DRIVE TO 3600 RPM
PRESSURE RATIO OF BOOSTER	0	1.63	1.75	1.75	2.0	1.75	1.75
ENGINE AIRFLOW(LBS/SEC)	223	317	352	352	372	352	352
DRY LOW NO _x COMBUSTOR	YES	YES	YES	YES	YES	YES	YES
TURBOCOOLER	NO	NO	NO	YES	YES	YES	YES
NEW LOW PRESSURE TURBINE	NO-GE90	YES	YES	YES	YES	YES	YES
LOW PRESSURE TURBINE SPEED	2000 RPM	3600 RPM	3600 RPM	3600 RPM	3600 RPM	**2000/3600	3600 RPM
LOAD CONNECTION	GEARBOX	DIRECT DRIVE	DIRECT DRIVE	DIRECT DRIVE	DIRECT DRIVE	DIRECT DRIVE	DIRECT DRIVE
COMPRESSOR DISCHARGE TEMP. (°F)	912	1075	1148	1148	1176	1148	1148
TURBINE INLET TEMP. (°F)	2514	2500	2500,2600, 2700, 2800	2500,2600 2700,2800	2500,2600 2700, 2800	2500,2600 2700, 2800	2500,2600 2700, 2800
POWER OUTPUT MW	43.95	59.6	63.6,69.9, 76.4,82.9	64.4,70.8, 77.3,83.4	66.6,73.5 80.4,87.4	64.4,70.8 77.3,83.4	64.1,70.6 77.0,83.4
SIMPLE CYCLE EFFICIENCY (%)	41.873	44.53	45.0,45.7, 46.2,46.6	45.5,46.1, 46.6,46.9	46.0,46.2, 46.7,47.1	45.5,46.1 46.6,46.9	45.31,45.98 46.5,46.9

ADVANCE TURBINE SYSTEMS(ATS) PROGRAM

Figure 3

3.3 COMBUSTOR AERO DESIGN

The combustor preliminary aero design effort during this reporting period focused in part on continued generation of flowpath layouts that accommodate the challenging performance requirements of the combustion system, yet fit into the available GE90 CDN with a minimum of mechanical modifications. In addition some supporting 3D CFD and chemical kinetics analysis were performed to get a first cut look at the performance.

The current status of the preliminary aero layout has a combustion system, (compressor OGV trailing edge to turbine nozzle leading edge), flowpath that is approximately 1.85 inches longer than the current GE90. Figure 4 provides an illustration of the current combustor flowpath aero layout. The combustor preliminary design features a tri-dome configuration in which there are 60 equally spaced mixer/injector assemblies in the outer two domes and 30 equally spaced mixer/injector assemblies in the inner dome. The lesser number in the inner dome is the result of circumferential spacing constraints. This is double the number of mixer/injector assemblies as featured in the LM6000 DLE combustor. This was done to allow for smaller mixer/injector assemblies, which reduced their length to better fit the overall combustion system into the limited available space of the GE90 CDN. The mixer/injector assemblies were designed using the DCARS mixer design criteria developed for the LM6000 DLE combustor. Analysis show that residence times in the mixer/injector assemblies are on the order of 0.6 to 0.8 msec. This should be low enough to prevent autoignition at all operating conditions. A similar mixer/injector dome configuration as used in the current LM6000 DLE is adopted. A tri-passage combustor inlet diffuser is also featured. This design provides for reduced length for a given area ratio and performance, and allows for better alignment of the compressor discharge flow streams into each of the three combustor domes to improve recovery and flow feed into the mixers.

A fully 3D CFD analysis of the current combustor design was performed using the CONCERT code developed at GE. CONCERT is a state-of-the-art 2D/axisymmetric or fully 3D elliptic turbulent flow code with fuel addition and combustion capability. CONCERT was developed expressly for application to gas turbine engine combustion systems and has been in production use at GE since 1987. CONCERT has been used for providing predictions of combustor exit gas temperature performance, and exhaust emissions including CO, UHC, and NOx. CO and UHC predictions are provided via a simple 2-step reaction/eddy breakup type model while NOx predictions are provided by a Zeldovich mechanism. The predictions of these emissions species are not highly accurate for very lean combustor designs do to the limited chemical kinetics that can be practically used with a fully 3D calculation. CONCERT undergoes continual enhancement to improve predictive accuracy and applicability. For a typical 3D analysis a single periodic section of the combustor is modeled. This assumption allows sufficient grid detail within limited available computer resources. This approach in essence assumes that all periodic sections of the combustor are identical. A grid network of 85x69x33, (193545), grid points was used for this model. Figure 5 presents views of the model grid for the 3 principal planes. Figure 6 shows the flow splits used for the model. Boundary conditions defining the mixer/injector discharge flow characteristics into the combustor were obtained from a CONCERT axisymmetric flow analysis of the mixer/injector. A single case was run representing maximum power operating conditions. The results, (using gaseous methane fuel), are presented in Figures 7 and 8, showing as expected with essentially no cooling flow the exit profile is essentially flat. Predictions of CO and UHC emissions at the max power operating condition, made as part of this modeling analysis, show a CO level of 3.5 ppm and the UHC level at 1.3 ppm.

Predictions of emissions were also made utilizing the CELESTE code. CELESTE is a stirred reactor type model developed at GE. It allows for the constructing of a network of perfectly stirred and/or plug flow reactors to represent the combustor. Because there are no complex

fluid mechanics involved more detailed chemical kinetics can be used than possible with the fully 3D CFD type analysis. In the analysis performed a kinetics mechanism consisting of 154 reactions was used. CELESTE models for the max power and 40% power operating condition were run. Figure 9.1 shows the simplistic network of reactors used. A natural gas representative fuel was used in these analysis. For the max power condition the predicted emissions levels were 5.7 ppm [CO] and 10 ppm [NO_x]. Given the predictive uncertainty of this model the predicted NO_x levels are consistent with the goals for the combustor. Results with this same model at the 40% power operating condition failed to produce a combusted solution. The fuel/air mixture was below the lean extinction limit. This is in agreement with test results of the DCARS mixer/injector assembly supporting the LM6000 DLE development. This outcome suggests the need to stage the combustor to maintain stability at this operating condition. A second CELESTE model was built to represent staging the combustor, the inner dome unfueled while the outer and center domes were fueled. With staging the two fueled domes operate at a fuel/air mixture above the lean extinction limit. Flow splits for this configuration and the CELESTE model of networked reactors is shown in Figure 9.2. The predicted results from this revised model at the 40% power operating condition were 20 ppm for [CO] emissions and 3 ppm for [NO_x] emissions.

WORK PLANNED FOR THE NEXT PERIOD

- 1) Perform a CONCERT3D model run of the 40% power operating condition with fuel staged to only the center and outer domes.
- 2) Refine the CELESTE stirred reactor models, especially the model for 40% power operation, to better reflect the internal flow mixing predicted from the CONCERT3D CFD models of the combustor.
- 3) Look at possibilities for shortening the current design to fit into the existing GE90 length. Initiate studies with compressor aero to look at ways to shorten the diffuser by aerodynamically integrating it with the OGV's. Perform additional analysis to see if length can be taken out of the diffuser dump region and the combustor without compromising emissions and other performance issues.
- 4) Obtain performance predictions at other operating points to initially define fuel staging requirements to optimize emissions performance, stability, and liner life.

CURRENT COMBUSTOR PRELIMINARY AERO LAYOUT

Figure 4

SIDE VIEW PLANE

END VIEW OF MODEL INLET PLANE

CONCERT3D MODEL GRID (85x69x33)

Figure 5

COMBUSTOR MODEL FLOW DISTRIBUTION (%Wcombustor)

MODEL INLET VELOCITY BOUNDARY CONDITIONS

CONCERT3D MODEL FLOW/BOUNDARY CONDITIONS

Figure 6

CONCERT-3D
3dgrd.grd
run1.plt

CALCULATED VELOCITY VECTORS IN PLANE INLINE WITH MIXERS

CONCERT-3D
3dgrd.grd
run1.plt

CALCULATED GAS TEMPERATURES (R) IN PLANE INLINE WITH MIXERS

CONCERT-3D
3dgrd.grd
run1.plt

CALCULATED NON-DIMENSIONAL GAS TEMPERATURES AT EXIT PLANE

Figure 7

Figure 8

CALCULATED CIRCUMFERENTIALLY AVERAGED EXIT PROFILES

Figure 9.1

CELESTE STIRRED REACTOR MODEL FOR MAX POWER

Figure 9.2

CELESTE STIRRED REACTOR MODEL FOR 40% POWER

3.4 HIGH PRESSURE TURBINE HEAT TRANSFER

Preliminary turbine cooling designs for the high pressure turbine vane and blade were developed in the previous reporting period to support the evolution of the ATS cycle and establish the design constraints necessary for a successful detailed design. In this reporting period, technical challenges were addressed pertaining to the thermal barrier coating on the HPT Stage 1 vane and cooling flow estimates were made for the HPT Stage 2 vane.

The ATS Turbocooler provides cooling flow to the combustor liners and high pressure turbine airfoils and bands at reduced temperatures and increased pressures as compared to traditional CDP air cooled designs. Flow from the turbocooler cools the combustor outer liner, vane outer band, and HPT shroud individually. It also cools the HPT vane and the spent cooling air from there is used to cool the combustor inner liner, vane inner band, HPT stage 1 blade, and forward combustor seal. A preliminary estimate of the cooling flow breakdown is as follows:

	<u>Wc (%W25)</u>
Combustor Inner Liner	1.5*
Combustor Outer Liner	1.5
HPT Vane Inner Band & Leakage	1.0*
HPT Vane Outer Band & Leakage	1.5
HPT Vane Film	3.7*
HPT Shroud	0.5
HPT Stage 1 Blade	2.2*
Forward Combustor	0.4*
Total	12.3

Note: All flows noted with a * pass through the HPT vane: 3.67%W25 is film and 5.1%W25 is non-film which is captured and used to cool the combustor inner liner, vane inner band, HPT stage 1 blade, and forward combustor seal.

HPT Stage 1 Vane

Cooling flow for the high pressure turbine vane is supplied from the turbocooler at much lower temperatures and higher pressures than the conventional vane coolant supply from CDP air. The HPT vane cooling design as proposed in the previous reporting period used only trailing edge film in order to minimize the cooling flow and film requirements to achieve life. The cooling circuit included two impingement inserts for increased heat transfer but rather than exhausting through film holes as in traditional designs, the post impingement flow is captured at the hub of the airfoil and used to cool the combustor inner liner and HPT blade. The ability to reuse the spent vane cooling air is only possible due to the increased pressures and lower cooling temperatures supplied by the turbocooler. Typical impingement/film cooled vane designs are limited by backflow margins ($P_{coolant}/P_{gas}$) at the leading edge of the vane to prevent gas ingestion. This is not a restriction for the ATS design; only the cooling air pressure supplying the combustor inner liner, which is the vane cooling air exhaust pressure, is a design constraint.

The TBC surface temperatures proposed for the ATS vane were much higher than our previous experience on either aircraft or military and industrial applications. There was a concern that the TBC would sinter and that the TBC thermal conductivity would increase at these temperatures. Therefore, based upon the recommendations of our material technologists, TBC outer surface temperatures were limited to 2350°F and TBC bond coat interface

temperatures were limited to 1920°F. These design constraints dictated that the vane design should be modified to include additional film cooling. Only 10 mils of TBC is required to achieve these temperature limits, however, which also reduces the technical challenge of thicker TBC. Five mils of TBC is not atypical on today's airfoils although up to ten mils have been tested for limited periods.

A comparison of the ATS vane versus the GE90 vane follows:

	ATS	GE90
Hot Streak Temperature (°F)	3075	3441
Coolant Temperature (°F)	719	1250
Pcoolant/Pgas	1.10	1.013
TBC Thickness (mils)	10	5/10
Wc (%W25)	8.77	7.94
Film (%W25)	3.67	7.94
NonFilm (%W25)	5.10	0.0
Life (cycles)	10000	

The non-film cooling flow which cools the ATS vane impinges on the airfoil and goes on to cool the combustor inner liner and HPT blade; of the remaining film flow. 1.12%W25 film flow exhausts through the trailing edge slots and 2.55%W25 is used as pressure and suction side film. Figure 10 indicates the ATS film geometry and flow distribution. Preliminary metal and TBC temperatures are shown in Figure 11. Additional concepts are still under consideration for further iterations on the cooling design of the vane.

HPT Stage 2 Vane

The coolant supply for the ATS HPT Stage 2 Vane is supplied from the exhaust from the tTurbocooler turbine. Cooling air is furnished at temperature and pressure levels similar to the GE90 design conditions which is fed with 7th stage bleed as shown below.

	ATS	GE90
P3 Design Point	588.2	577.0
Ps Vane 2 Source	299.1	268.5
Gas Temperature (°F)	2488	2463
Coolant Temperature (°F)	948	960
TBC Thickness (mils)	4/7/10	4/7/10
Wc (%W25)	2.50	1.97
Forward	1.13	0.80
Aft	0.15	0.14
Trailing Edge	1.22	1.03
Life (cycles)	10000	

The life limiting location of the ATS HPT Stage 2 vane is the trailing edge. In order to achieve the ATS life requirements, the trailing edge temperatures will have to be reduced ~100° F below GE90 temperatures. This can be accomplished with increased pressure side film and trailing edge flow. A comparison of the GE90 and ATS flows are shown in Figure 12. Figure 13 indicates the TBC distribution and the life limiting temperatures for the GE90 and ATS.

Technology/Testing Needs

Vane Impingement with Crossflow Test:

The HPT vane cooling design relies upon impingement air providing enhanced coolant side heat transfer coefficients. Rather than film cooling the airfoil, the flow is captured at the hub of the airfoil and reused as combustor liner or HPT blade cooling. Intercepting the flow implies that some of the benefit of higher impingement heat transfer coefficients will be reduced due to cross-flow. The vane design will be dependent upon optimization of the internal cooling concept.

Three configurations are to be tested for vane impingement with crossflow - a baseline and two variations. Work authorization for work to begin at Texas A&M is pending approval in order for testing to begin on this nine month testing program.

Technology Challenges

TBC provides a technical challenge in many arenas: spallation of the TBC is not fully known; temperatures through the TBC as well as the maximum temperature of the TBC are areas of concern for the ATS life requirements.

Future Work

Iterations on the HPT vane and blade cooling designs will continue as further information on the life restrictions and the cycle conditions are developed. The HPT Stage 2 Blade cooling flow definition will be completed.

ATS HPT Stage 1 Vane - 10 mils TBC - Film Cooled

COOLING FLOWS

Trailing Edge Film	= 1.12 % W25
Airfoil Film	= 2.55 % W25
Non-Film	= 5.10 % W25
Wc Total	= 8.77 % W25

T BULK= 1597
 SEGMENT= 5
 RADIUS= 17.67885

Figure 10

Outer TBC Temperature
Outer Metal Temperature
Inner Metal Temperature

Outer TBC Temperature
Outer Metal Temperature
Inner Metal Temperature

```
T BULK=      1597
SEGMENT=      5
RADIUS=     17.67885
```

Figure 11

GE90 STAGE 2 VANE FLOWS

ATS STAGE 2 VANE FLOWS

Figure 12

HPT STAGE 2 VANE METAL TEMPERATURE LIMITING LOCATION & TBC DISTRIBUTION

**ATS: T_{max} = 1929F @ airfoil trailing edge
10000 cycles**

TBC Thickness Distribution

Figure 13

3.5 TURBOCOOLER SYSTEM

An internal preliminary design of the Turbocooler (Turbocooler) system has been completed, using as a base the Allied Signal compressor and turbine maps provided for another application.

The current size accommodates an increased primary cooling flow requirement of 12.35% W25, up from the 10.52% Proposal value, due to TBC temperature limits.

A preliminary layout of the system was used to estimate the system pressure losses is shown in Figure 14. Pressure distribution in the system at full power is shown in Figure 15.

Current pressure drops are higher than Proposal values for both the turbocooler turbine and turbocooler compressor circuits. Effects of the pressure drop estimates on the system include an increase in the required turbocooler compressor pressure ratio from 1.2 to 1.29, and an attendant increase in the fuel sink utilization to maintain Turbocooler primary coolant supply temperature of 700°F. The available air side pressure drop in the heat exchanger was reduced to partially compensate the increased extraction and feed losses.

In the current design the Turbocooler turbine exhaust is directed to and provides cooling for the engine HPT second stage vane, replacing the HPC (high pressure compressor) interstage coolant for this purpose. The resultant increase in Turbocooler turbine exit pressure (relative to the proposed system) reduces the available head, requiring an increase in extraction for the purpose of driving the Turbocooler compressor. There is more air available for downstream (secondary) cooling, but it will necessarily be hotter than the HPC interstage source. The increased primary flow and pressure requirements noted above, and the increase in pressure loss in the Turbocooler turbine circuit also contribute to an increase in Turbocooler turbine extraction levels.

Meeting the primary cooling requirements (HPT vane and blade, burner liner and HPC shaft) at the baseline required-to-run fuel flow results in a fuel temperature at the burner of 1049°F. (Column A below). Fuel temperature at the burner can be maintained at 1000°F only by reducing the primary coolant 0.75% W25 (B), or increasing its supply temperature by 36°F (C). Column D provides a comparison with the original proposal system.

	<u>A</u>	<u>B</u>	<u>C</u>	<u>D</u>
Primary coolant (%W25)	12.3	11.6	12.3	10.5
	5	0	5	
Supply temperature (°F) @ HPT stg. 1 vane	700	700	736	700
Fuel temp at burner (°F)	1049	1001	1000	893
HX volume (cuft.)	1.78	1.4	1.32	0.8
Turbocooler wheel dia. (in.)	6.18	6.	6.23	6.34
Turbine exit temp (°F)	1003	1003	1003	735
Turbocooler wheel dia. (in.)	5.3	5.1	5.25	11.1
Turbocompressor speed	4684	4859	4722	3765
	0	0	0	0
Turbocooler pressure ratio	1.29	1.29	1.29	1.19
Turbocooler pressure ratio	1.58	1.58	1.58	4.35

ESTIMATED PRESSURE DISTRIBUTION
IN ATS TURBOCOOLER
AT FULL POWER

Figure 14

Turbocooler

3.6 LOW PRESSURE TURBINE DESIGN

MECHANICAL DESIGN

The ATS power turbine is a five stage design that drives both the low pressure booster and the power output shaft. The mechanical design is based on the GE90 LPT philosophy with the intention to utilize the maximum amount of GE90 hardware. The turbine cross-section is shown in Figure 16.

The power turbine uses the GE90 flowpath with new vanes and blades. The GE90 LPT secondary flows and temperatures were used for the preliminary estimate of the ATS turbine flows and temperatures which were used for component sizing. The disks have been sized for ATS operating conditions (3600rpm) and 100,000 hours creep life.

The turbine mid frame and the rear frame have been adopted for the GE90 design with minimum modification. To ensure the use of the GE90 LPT case, the axial distance between the two frames has not changed. Instead, spacers have been added to the flowpath because the ATS turbine has 1 fewer turbine stage than the GE90.

The bearing sump and rotor thrust balance design is similar to the GE90 design with the exception that an additional bearing has been incorporated to support the new power output shaft.

TURBINE AIRFOIL MATERIAL SELECTION

For initial assessment of the material requirements for the LPT Blades, the stress and temperature regime for each stage was established and compared to the 0.2% average creep life capability for a series of candidate turbine airfoil alloys. The goal life for LPT blades is 50,000 hours minimum, a period of time for which materials data does not exist for most of these alloys; this required extrapolation and estimation to project the long time materials capability. Based on prior experience, the physical metallurgy of these materials, and the temperature range of the estimate, we expect these estimates to be accurate.

The 0.2% creep requirement is a standard, but conservative, criteria first step design assessment which, if met, provides a product component with substantial deformation margin (is not deformation limited) which usually lasts the product lifetime or is refurbished due to long time environmental effects. If the criteria is not met, then the designer moves to another step and conducts further analytical work to calculate detailed creep deformation for the airfoil and assess this result against configurational requirements; tuning the airfoil configuration to reduce stress in limiting areas may be part of this step in the process. Typically, for a machine this size, total radial creep of 20 mils for stage 1 to 30 mils for stage 2 represent acceptable limits. This report covers only the initial step in this life assessment process.

LPT Blades - Temperature Study Regime and Stress Requirements

Stage	Stress (Ksi)	X - Denotes Temperature Regime									
		180 °F	170 °F	1600 °F	150 °F	140 °F	1300 °F	1200 °F	1100 °F	1000 °F	900 °F
1	17	X	X	X							
2	30		X	X	X						
3	37			X	X	X					
4	39					X	X	X			
5	43								X	X	X

LPT Blades Temperatures (°F) for 2700°F T41 and 2600°F T41

<u>Stage</u>	<u>Stress(Ksi)</u>	<u>2700°F T41</u>	<u>2600°F T41</u>
1	17	1700	1600
2	30	1560	1450
3	37	1420	1320
4	39	1230	1140
5	43	1090	990

A range of temperatures was studied to correlate with the various ATS cycles under study and display the influence of varying temperature and stress on component life. Material capability in terms of 0.2% creep life was assessed for cast nickel based turbine blade alloys used by GE. These alloys are listed below:

<u>Alloy</u>	<u>Form</u>
R77	Conventionally Cast
R80	Conventionally Cast
In738	Conventionally Cast
GTD111	Conventionally Cast
R125	Conventionally Cast
R108	Directionally Solidified - Multi Grain
R142	Directionally Solidified - Multi Grain
RN4	Directionally Solidified - Single Crystal
RN5	Directionally Solidified - Single Crystal
RN6	Directionally Solidified - Single Crystal

Average 0.2% creep lives were calculated for each alloy for the temperature regime (3x100°F increments) and stress condition of each LPT Blade Stage, plotted as bar charts and compared with the 50,000 hour life goal for these components. The following four charts present the results of the analysis for Stages 1 through 4, respectively. The Stage 5 analysis indicated lives greater than 1,000,000 hours for all the alloys considered for temperatures of 1100°F and lower and no chart is presented for this situation.

**ATS GE LM9000 Stage 1 LPT Blade
Study - 0.2% Creep Life at 17 Ksi**

**ATS GE LM9000 Stage 2 LPT Blade
Study-0.2% Creep Life at 30 Ksi**

Review of the chart for the Stage 1 LPT blade indicates at 1800°F none of the alloys meet the 50,000 hour 0.2% creep criteria. At 1700°F, RN6 comes close with 42,000 hour capability and at 1600°F both RN5 and RN6 exceed the requirement. The chart for Stage 2 shows that, although the temperature regime is lower, the increased stress level causes a similar situation as stage 1 with no alloys meeting the criteria at both 1700°F and 1600°F, but RN5 and RN6 satisfy needs at 1500°F.

**ATS GE LM9000 Stage 3 LPT Blade
Study - 0.2% Creep Life at 37 Ksi**

**ATS LM9000 Stage 4 LPT Blade Study -
0.2% Creep Life at 39 Ksi**

Review of the chart for the Stage 3 stress again shows the deformation free criteria not to be met by any alloy until the temperature is around 1500°F where RN6 would be fully satisfactory. For Stage 4, conventional cast alloys should be capable of meeting the deformation free criteria with the temperature below 1300°F at 2700°F T41. Stage 5 is not creep limited with any of the alloys.

The overall conclusions are:

- 1. LPT Stages 1, 2, 3 are creep deformation limited and will require DS/Single crystal alloys**
- 2. LPT Stages 4&5 material capability requirements can be satisfied with conventionally cast materials.**

4.0 Work Planned For Next Reporting Period.

Engineering effort under Task 3 will be completed.

ADVANCE TURBINE SYSTEMS(ATS) PROGRAM

GE90 TURBINE CASE

DISK MATERIAL

STAGES 1 TO 5 — R88DT
see I-718

SPACERS

MODIFIED GE90
TURBINE REAR FRAME

GE90 TURBINE MID FRAME

GE90 SHAFT

INCO 718 POWER SHAFT

Figure 16

LM9000 POWER TURBINE

4.0 TECHNICAL PROGRESS SUMMARY

12/1/1993 - 2/28/1994

TASK 3B - Selection of a Utility Natural Gas Fired Advanced Turbine System (GFATS)

The preliminary selection of the gas fired advanced turbine system is a combined gas and steam turbine cycle system adapted from the system that was developed by GE and reported in EPRI Report No. 2620-1. The gas turbine is of single shaft configuration with high firing temperature and modest pressure ratio with selected steam cooled components and dry low NO_x combustion system. A cross-section of the gas turbine is shown on Figure 1. The steam cycle is a two pressure, reheat Rankine steam cycle with unfired heat recovery steam generator (HRSG). The steam cycle is integrated with the gas turbine cooling steam system to maximize the utilization of the energy removed from the gas turbine components for production of power.

The selected GFATS is projected to reduce generation cost from that of current technology combined cycle generation system by increased thermal efficiency and reduced installed cost. The improved performance and installed cost and operational characteristics are presented on Figure 2.

The DOE ATS goal of lower heating value thermal efficiency of 60% is indicated to be a feasibility with this cycle concept. Improvements that potentially can be incorporated in the GFATS to achieve this goal are:

- Increased gas turbine firing temperature
- Fuel heating
- Kalina bottoming cycle

Combined cycle thermal efficiency improvement that is achievable with increased gas turbine firing temperature is illustrated by Figure 3 which presents performance projected from the work reported in EPRI Report No. RP2620-1.

**UTILITY GAS FIRED ADVANCED TURBINE SYSTEM
GAS TURBINE CROSS-SECTION**

Figure 1

GFATS COMPARISON WITH CURRENT GENERATION EQUIPMENT

	<u>GFATS</u>	<u>CURRENT</u>
EFFICIENCY	58.5 **	55
RELIABILITY	*	*
OPERATIONAL AND MAINTENANCE	*	*
INSTALLED COST (\$/KW)	-\$20	BASE

* DESIGN PRACTICES AND MAINTENANCE INTERVALS ARE EQUIVALENT

** PROJECTED FROM EPRI CONFIGURATION

Figure 2

COMBINED CYCLE THERMAL EFFICIENCY VARIATION WITH GAS TURBINE OUTPUT

Projected From EPRI Study Configuration

Figure 3

QUARTERLY PROGRESS REPORT BY SUBTASK

Task 8.1 Particulate Flow Deposition

Objective

The primary objective of this task is to characterize the particulate generated in an operating gas turbine combined cycle (GTCC) power plant whose configuration approximates that proposed for an Advanced Turbine System (ATS) power plant. In addition, the task is to evaluate the use of full-flow filtering to reduce the stream particulate loads. General Electric had already negotiated an agreement with the candidate power plant, and piping and a filter unit had already been installed at the power plant site, and major elements of the data acquisition system had been purchased, all on GE funds, prior to the initiation of this Task 8.1. The remaining work involves completing assembly and checkout of the system, and then conducting measurements to meet the above-mentioned objectives.

Technical Progress

No work was done or charged on this contract activity through December 31, 1993, as the scope of work of Tasks 8.1 and 8.2 were readjusted to ATS priorities and requested. However, the GE-supplied test section was completed at the powerplant (on GE funding

Two visits were made to the plant site to make final piping recommendations, plan data acquisition system connections, and specify hydrotest, blow-out (cleaning), and first sets of trials. An updated Piping and Instrumentation Diagram (P&ID) was transmitted to the GE Operation and Maintenance organization (GE O&M) and contractor representatives. Among recent additions are a pressure relief valve at the request of the O&M manager, and the elimination of overhead insulation to allow some cooling of the flow stream. The estimated entry conditions will be 600psig, 850F, about 150F higher temperature than desired, but it was decided to correct filter drop for temperature rather than chance developing condensate with a condenser. It was also learned that the customized, 1" diameter screen filter apparatus (ca. 0.3 lb/min) worked at 1000 psi, 1000F without leakage using annealed nickel "O" rings, though flange bolts seized. The bolting arrangement has been modified, and this filter apparatus is being installed on-line once again at the plant sample panel to get an early measure of the 2 micron filter behaviour before the larger cartridge system goes on-line.

First results of sample filtration through 47mm disk screen filters were obtained. The filter used was a standard Millipore (R) filter holder modified to accept annealed nickel "O"-rings to replace the rubber rings usually employed. Both 2 and 50 micron screen filter disks were exposed to 1050psig, 950F coolant for about 8 hours at gas velocity loadings over 5 times those expected in commercial practice. Scanning electron microscope analyses (SEM's) showed that oxide chips approximately 5 microns thick by 20 micron diameter dominated the material on the 50 micron screen. No oxide chips built up a cake on the screen; only those chips embedded in the wire mesh were observed. However, there was debris in the plastic filter Petri dish, indicating that the material is non-sticky and simply falls off the filter. This was confirmed by the SEM's of the 2 micron screen which showed almost no material on the filter at all since it could not embed itself.

In January, GE operations personnel requested the installation of a full-flow coolant strainer at the point of readmission to the low pressure turbine, and of a pressure relief valve in the return line in case the supply system is accidentally opened with the return valve closed. Those fittings have been received and will be installed after completion of a plant shutdown on March 14. Then the process of commissioning the filter system can begin.

The computer control and data acquisition system has been assembled and programming is about 60% complete. It should be ready for first checkout trials in March.

Task 8.2 Particle Centrifugal Sedimentation

Objective

The primary objective of this task is to determine the settling characteristics of particles in a cooling stream from an operating gas turbine combined cycle (GTCC) power plant when that stream is ducted through a passage experiencing the G-loads expected in a simulated bucket channel specimen representative of designs proposed for an Advanced Gas Turbine (ATS) engine. GE has identified a target power plant at which to site the experiment. GE had completed a proprietary computational code that was proposed to be useful in predicting particle trajectories prior to Task 8.2 start.

Technical Progress

No work was done or charged on this contract activity through December 31, 1993, as the scope of work of Tasks 8.1 and 8.2 were readjusted to ATS priorities. However, preliminary discussions were held with the candidate vendor of the rotating rig to help specify operating conditions that would both give useful results and yet utilize as much available technology as possible for this novel test apparatus.

The test specimen has been designed after consultation with the GE industrial and Power Systems (GE/IPS) 2nd stage blade engineer and GE/IPS project contact to assure good simulation of projected operating conditions. The specimens at each end of the rotating arm will be identical. Flow simulation with a 3D CFD code has shown potential for simulating the projected full flow (0.17 lb/sec) at this critical first pass turn reasonably well. The specimen size (4"x3"x1") and weight (2.5 lb) has been communicated to the design engineers at the centrifuge vendor (MTI) who are on schedule for a conceptual rig design providing the requisite 15,600G's of exposure. They currently estimate a 50 hp driver with a unit that is skid-mounted.

The conceptual design of the rotary centrifugal system has been completed. The system is shown schematically in Figure 1. The system is designed to rotate a 2 lb flow specimen payload at each end of a two arm rotor at flows of 0.17lb/sec of coolant at 600F and 700F at conditions to give 15,600G's of acceleration (24" radius at 4800rpm) at the ATS simulated bucket tip. The apparatus is to be mounted with the axis horizontal on a skid, including the missile-containment housing, motor drive, lube oil system, and housing vacuum pump. Issues of possible large coolant leakage at the two rotor seals and concerns about the method of constraining the flow specimens were raised, but new approaches have been identified that appear to address these issues satisfactorily.

Figure 1 Rotary Centrifuge Schematic

Task 8.3 TBC Mechanical Test and Analysis

Objective

Development of thermal barrier coatings (TBC's) with improved life and reliability will require a comprehensive understanding of mechanisms of degradation that occur during gas turbine service. There are two complimentary objectives in this task. The first is to develop and confirm methods to measure and predict TBC stress states as a function of thermal and mechanical strains. This capability is fundamental to all quantitative TBC design and life prediction methodologies. The second objective is the development of a practical, versatile laboratory-scale thermal gradient exposure facility that is capable of simulating the extreme thermal conditions anticipated for TBC's in an advanced gas turbine.

Technical Progress

Development of thermal barrier coatings (TBC's) with improved life and reliability will require a comprehensive understanding of mechanisms of degradation that occur during gas turbine service. Delamination mechanisms are especially critical because they are capable of catastrophically removing the thermal protection provided by the TBC. Analyses of many examples of coating loss have shown that delamination crack growth often occurs at low temperatures due to high in-plane residual stresses in the coating. There is also evidence that damage may occur at high temperature due to thermal stress caused by high thermal gradients and/or rapid thermal transients.

Methods are available and numerous measurements have been made in this laboratory of the residual stress states of TBC's at room temperature. Comparable methods and measurements, however, are not available for thermal conditions representative of advanced gas turbine operation which include high temperatures, large thermal gradients and high, externally-imposed states of strain. In this task calculations will be made of the stress state of coated substrates in isothermal, gradient and transient conditions. In addition, the actual stress state will be measured versus temperature in isothermal conditions. Comparisons of calculated versus measured stresses will be used to assess the ability to predict stress states in actual turbine operation.

Also, in order to directly evaluate the durability of TBC's in steep thermal gradients and under transient conditions, an improved thermal gradient exposure facility will be developed and constructed. The facility will be capable of exposing TBC coated specimens to thermal conditions representative of advanced gas turbine components and will be demonstrated on coated substrates.

The following two sub-sections describe the progress and plans related to the two objectives of this task.

Analysis and Measurement of Coating Stress

The accurate prediction of TBC stress states requires advances in both stress analysis techniques and measurement of important physical properties. This subtask includes effort directed towards both needs.

To characterize the residual stresses developed during the coating process, a three-layer model is being developed and will be used to allow analysis of changes in stress state with

changes in temperature. The model includes the substrate, the top coat and an intermediate layer, which could represent a bondcoat or a gradient layer with varying properties.

Modeling of the actual coating process is quite difficult and requires accurate characterization of many complex events that occur during coating. Considering the unknowns and the required property-temperature data needed for such a model, we are modelling the simpler problem of changes in stress with changes in temperature and strain. The problem is challenging, however, due to the highly non-linear "elastic" properties that are characteristic of TBC's.

Several key physical properties of an advanced air plasma sprayed (APS) TBC will be measured that are needed to execute the stress analysis. These properties include: coefficient of thermal expansion and elastic modulus of the substrate, bond coat and TBC as a function of temperature, and non-linear characteristics of the TBC elastic modulus as a function of stress. In addition, the initial stress state of the TBC will be needed.

The conventional physical properties will be measured using conventional techniques. The initial residual stress state will be evaluated using differential curvature techniques during specimen thinning. This technique will be extended through use of bimaterial dilatometry to measure residual stress versus temperature. The non-linear elastic properties of the TBC are clearly the most challenging of the property measurements and require development of new measurement techniques. Techniques will be evaluated that use both direct stress-strain measurements and measurements of modulus versus temperature in laminated specimens where the TBC will change stress state with temperature.

Using the measured properties and room temperature residual stresses, the stress state for selected thermal conditions will be calculated. The thermal conditions will include isothermal high temperature, steady state thermal gradients, and thermal transients.

The calculated stress state for isothermal heating will be compared to measurements of stress versus temperature that will be measured using variations of differential curvature techniques. Comparisons of calculated and measured stress states will then be used to assess the quality of the analytical techniques developed and applied in this task.

Computer based methods have been developed for evaluating residual stresses resulting from differences in the thermal and mechanical properties of the different materials over the range of "stress-free" temperatures. Fortran programs have been written for three cases:

1. TBC_CYL : Cylindrical geometry
2. TBC_SPH : Spherical bending of a plate
3. TBC_INP : In-plane extension of a plate

All three models include the substrate, bond coat and ceramic layers. Stress-free temperatures may be different for the bond coat and the top coat. Temperature dependent properties are included in a library of materials data.

In order to begin development of analytical techniques to incorporate non-linear elastic behavior, an initial estimate of the non-linear characteristics was needed. This initial estimate has been derived from a series of 15 measurements of residual stress and in-plane elastic modulus that were collected prior to this contract. The 15 measurements were taken on a

series of 15 different TBC's with markedly different microstructures and properties. The 15 points were fit to a parabolic relationship of stress versus modulus. Finally, this relationship was used to determine the non-linear stress-strain curve consistent with the parabolic modulus versus stress. This behavior will be used in TBC stress analyses until better relationships become available.

In anticipation of the non-linear stress analyses that will be necessary for TBC's, the non-linear material models used in commercial stress analysis codes ANSYS and ABAQUS have been reviewed. ABAQUS was found to have more capability in non-linear analysis than ANSYS. Some of the available models in these two codes are:

ANSYS:

- **Nonlinear Elastic Model:** Tension and compression behavior symmetric. The stress-strain curve is given pointwise.

ABAQUS:

- **Hypoelastic Nonlinear Model:** A general nonlinear elasticity model for small elastic strains.
- **Hyperelastic Nonlinear Model:** A nonlinear elasticity model used for large elastic strains. It is used mostly for modeling plastics, solid propellants and rubber-like materials. The tension and compression behaviors can be different.
- **Ramberg-Osgood Plasticity Model:** Uses deformation theory of plasticity, which is different than the normal plasticity models based on flow theory of plasticity. It can be used approximately to model a nonlinear elastic case if no unloading is assumed. Tension and compression behavior are symmetric.
- **Tension-Cutoff Linear Elasticity Model:** Used with linear elasticity model but tensile behavior is modified such that tension stresses cannot be generated. Compression behavior is linear.
- **Concrete Model:** Cracking is assumed to modify the material response. The tension and compression response can be nonsymmetric. This model is especially useful for describing materials with weak tension and strong compression strengths.
- **User-Defined Material Behavior:** The user can define his own constitutive model in a very general way. This feature is very powerful and general, but is not a routine exercise and requires considerable effort on the part of the user.

In addition to the above models, both ABAQUS and ANSYS have other nonlinear options for plasticity, creep etc.

A vertical tube furnace was configured for measurement of flexural resonance properties of TBC specimens versus temperature up to at least 700 C. The resonance properties will be used to determine the Young's modulus of yttria stabilized zirconia (YSZ) TBC's both as laminated bodies (YSZ, bondcoat and substrate) and as free-standing unconstrained bodies (YSZ removed from bondcoat and substrate). The resonant flexural frequencies will be measured by suspending rectangular beam specimens from support wires at the node positions for the fundamental flexural resonance. The specimen will be excited by impacting the center of the beam with a small zirconia sphere in a manner similar to that described in the upcoming

ASTM standard C-1259-94 entitled Elastic Modulus by Impulse Excitation of Vibration. The resulting resonant vibrations will be sensed with an acoustic microphone, also similar to that described in the ASTM standard. Figure 2 is a schematic of the test setup.

The primary challenge associated with the high temperature measurements is sensing of the resonant vibrations. The measurement requires a waveguide to transmit the resonant signal from the hot specimen to an external (room temperature) microphone. Several different microphones and waveguide configurations have been evaluated to maximize sensitivity. The highest sensitivity was found with a vertical waveguide of fused-silica tubing with one end in close proximity to the bottom midspan of the specimen and the other end outside the furnace in close proximity to a Model 4165 Bruel & Kjaer condenser microphone. This is the configuration that will be used in upcoming high temperature resonant frequency measurements.

The most difficult resonant measurements to be made are on free-standing TBC specimens due to the very high damping characteristic of these highly microcracked microstructures. Resonance is difficult to detect on these specimens, even at room temperature where the microphone can be placed in close proximity to the specimen. When the optimum waveguide and microphone configuration described above was used, it was found that resonance could not be reliably detected in the furnace. This complication was anticipated as a possibility due to prior experience in room temperature measurements of free-standing specimens of TBC. The alternative plan will use dense zirconia to characterize the intrinsic change in modulus of zirconia with increasing temperature. The lower damping of dense zirconia should pose no problems in signal detection with the current microphone arrangement.

Thermal Gradient Exposure Facility

In order to evaluate TBC performance, a laboratory scale Thermal Gradient Exposure (TGE) facility is being developed and constructed. The objective of the TGE facility is to closely simulate the extreme thermal conditions anticipated in an advanced gas turbine TBC.

Initial effort in this task considered a design that utilized an array of IR Quartz Halogen lamps with parabolic reflectors and a specially designed substrate cooling manifold. The lamp assembly and controllers have been designed and are under construction. Associated facilities and building services are being installed. During this reporting period, however, work on the IR lamp facility and associated services has been suspended. This decision was made in light of new information concerning the high optical transparency of YSZ and because of the difficulty in identifying a radiant energy absorption coating that would not interfere with the properties of the TBC under test.

An expanded program has been initiated to investigate alternate energy sources. Various heat sources have been considered including carbon dioxide lasers and electron beam guns (e-beam). The e-beam offers many attractive features. Therefore, current work is concentrating on that approach for generating the high heat fluxes required in this test facility. The particular e-beam configuration that is being examined is an "atmosphere-capable" design wherein the e-beam is brought out into the atmosphere through a series of differentially pumped stages. Such devices are commercially available, and typically are rated at approximately 25 kilowatts of beam power. The atmospheric e-beam appears to fulfill all the major requirements of the energy source in the gradient facility, i.e. flux densities of greater

Figure 2 Young's Modulus Measurement Apparatus

than 300 watts/ cm², operation in an oxidizing environment, flux density uniformity, and ability to heat near-flat samples of greater than 1" x 1". The major questions that surround the use of the e-beam are the depth of beam penetration and the degree of damage to the YSZ structure caused by electron interactions.

Off-site experiments with an atmospheric capable e-beam were carried out to investigate the interaction of the e-beam with YSZ. Samples were exposed to moderate fluxes of the e-beam with no back-side cooling. The specimen temperature was increased to as high as 1200 C as measured by pyrometers and embedded thermocouples. The exposed specimens were then returned to CRD for evaluation of possible damage. The samples exhibited no discoloration of the type expected from creation of point defects in the crystal structure. The specimens are undergoing x-ray diffraction studies to determine phase content and metallographic preparation for inspection of the microstructure.

A finned copper heat sink structure has been fabricated and a TBC coupon was attached to the heat sink to allow the next stage of E-beam experiments to proceed. Much higher fluxes of electrons will be used in conjunction with backside cooling to evaluate the practicality of the E-beam heat source in thermal gradient tests and to evaluate any possible E-beam damage to the TBC at high fluxes.

Task 8.4 Advanced Seal Technology

Objective

Advanced turbine designs will produce more power with fewer stages and will be more efficient than the existing designs. This requires much higher operating temperature and higher pressures across the stages to be sealed. From the combustor on through the turbine stages, the entire system has to be efficiently sealed. This Task will identify critical sealing applications that potentially can provide 1-3% improvement in overall turbine efficiency by reducing leakages and reducing maintenance cost resulting from wear and related failure.

Technical Progress

Leakage that occurs at the Combustor-Turbine interface (ie., at the Transition-Piece to First-Stage-Nozzle interface) has a major influence on the turbine's efficiency and on NO_x production. Similarly, leakages in the gas-path, such as between the nozzle and the shroud, would lower the efficiency of a turbine. Due to the very high temperatures of operation at the above identified locations, seals have to be configured to suit the materials and their unique manufacturing requirements. For example, if single-crystal materials are used, there are only a few ways in which these components could be machined, and any seal used in conjunction with them will have to be suitably designed around this limitation. We have defined sealing requirements for the following interfaces:

1. Transition-Piece to Stage I nozzle comprising inner, outer and side and lip seals for curved and straight interfaces.. The design requirements include:
 - Expected leakage of less than 1% total
 - Only radial slots are permitted on the Stage I nozzle
 - Stage I nozzle casting should be common for all design concepts being evaluated;
 - Ease of assembly;

- Transition Piece end on which the sealing occurs must have room for cooling (impingement) holes that do not interfere with the sealing.
2. Nozzle Shroud sealing: Design requirements include:
- Expected leakage, less than 1%.
 - Seals must be able to fit within available gap and overlap.
 - Seals must accommodate the expected axial and radial growth.

A GE-funded Vibratory Seal Test Rig has been designed.. This test rig is capable of testing a variety of seal designs under varied pressure and vibratory load conditions: This rig comprises a test chamber divided (by a diaphragm) into two sub-chambers; the test seal is used to isolate one from the other. One sub-chamber is fed with high-pressure air; the pressure drop across the test seal is monitored in the other sub-chamber

Test seals (of standard 12 inches length) are designed so as to fit on to jaws that can be moved relative to each other. These jaws are mounted on vibrators that provide the excitation. Different kinds of seals can be tested in this rig by suitably designing the jaws and the seals, for easy mounting on the diaphragm.. Figure 3. is a schematic of the rig.

The vibratory seal test rig assembly has been completed. Preliminary pressure tests showed some leaks at junctions between adjoining components which were corrected by remachining for better fits and improved seals between components.

The test rig has been hydrostatically pressure tested for safety. Benchmarking tests have been conducted on the rig under various inlet-chamber pressures without seals, measuring the corresponding outlet-chamber pressures for different flow conditions. This was necessary to determine what constriction the dividing diaphragm provides before evaluating the performance of the test seals.

Task 8.5 Enhanced Impingement Heat Transfer

Objective

Tests will be performed to evaluate a new concept for backside impingement cooling aimed at lessening the adverse effect of crossflow on air jet thermal dilution and to determine the upper limits for jet heat transfer at higher air supply pressures.

For long impingement channels the spent jet impingement air represented as crossflow mixes and raises the temperature of the cold air jet before it strikes the hot wall.

Technical Progress

Texas A&M University has been selected to perform this task. A contract will be placed with Texas A&M once subcontract approval has been granted by DOE.

Task 8.6 Rotating Heat Transfer

Objective

Prediction of gas turbine blade life requires sufficient accuracy in the prediction of both the local hot gas side and coolant side heat transfer coefficients present at the relevant blade surfaces. While a considerable data base exists for the hot gas side coefficients, that for the

Figure 3 Seal Test Rig

rotating blade coolant passages is very limited. Only recently have measurements in rotating simulated blade passages become available that cover the conditions of interest to aircraft gas turbine blades. At the conditions present in power gas turbines being designed today excessive extrapolation of the existing data base is needed

An effort is presently underway at CRD that will provide the required heat transfer data base over the range of dimensionless parameters.

The rotating test facility installed at CRD in 1993 is now obtaining data in smooth rectangular full scale blade cooling passages. Electrical heat addition is employed on each of the few passage walls which are instrumented with thermocouples to determine the local heat transfer coefficients. The coolant pressure is varied over a broad range to achieve the required ranges of Reynolds number and Rotation number.

Technical Progress

Design and drafting for the new turbulated test section were carried out during December 1993 for installation in the Rotational Heat Transfer Facility shown in Figure 4. In January the high speed experiments, using the installed smooth test section, were completed and the data reduced. To prepare for the turbulated test heaters were added and bracing was installed in the rotating arm enclosure. The latter was found necessary during the high speed tests completed to-date to improve experimental accuracy at high rotational speeds.

The smooth wall rectangular duct was determined to have minimum lead side heat transfer coefficients that are 30% less than those obtained in a square duct. There is also a Reynolds number effect in the rectangular duct that has not been detected in the lower Reynolds number data published by other workers.

The construction of the new test section is scheduled for completion during the next reporting period after which its installation will begin. The enclosure modifications will be tested to determine their effect on reducing heat losses and decreasing vibration at high speed. After these tests the smooth wall test section will be dismantled.

Task 8.7 Turbine Inlet Nozzle Heat Transfer

Objective

Program objectives include the investigation and determination of the external heat transfer coefficient distribution for the Stage 1 Nozzle airfoil of the Advanced Machine, as well as the validation of computational heat transfer predictions. The program will include the characterization of external heat transfer due to the effects of inlet flow preparation including turbulence intensity level and swirl, airfoil surface roughness, and Reynolds number.

The test apparatus will have as its core a 1/2-scale linear airfoil cascade designed to model the appropriate nondimensional turbine nozzle parameters, including inlet flow Reynolds number, nozzle pressure ratio, and airfoil Mach number distribution. The cascade will be operated at an inlet total pressure of 5 atm, and temperature of 150 F. The cascade will consist of five airfoils, instrumented to obtain aerodynamic and heat transfer information. The central airfoil will be a thin-walled stainless steel unit with imbedded thermocouples, employing a thin-foil surface heater to determine local surface heat transfer coefficients. The airfoil surface will be variously coated with alumina particles to provide well characterized roughness distributions.

Figure 4 Rotational Heat Transfer Test Rig

The flow entering the cascade section will be conditioned to model the turbulence intensity level and swirl, which would be obtained with a Dry Low NO_x combustor under cold flow conditions, including the actual swirler geometries and development sections. In addition, a range of axial turbulence intensities without swirl will be available through the use of perforated plates at the cascade inlet. The effects of flow Reynolds number variation will also be investigated.

Technical Progress

The technical progress for this period includes both design, drafting, and procurement tasks funded by GE, as well as fabrication and installation planning tasks funded under the present contract. The facility is shown schematically in Figure 5.

The design of the cascade and associated hardware has been completed. This includes the airfoil cascade proper, all upstream flow preparation pieces, and all instrumentation. Drawings required for fabrication have also been completed, with only minor revisions expected as fabrication proceeds. All major rig hardware and materials have been purchased, including the flow preparation pressure vessel, flanges, piping, flow control valves, monitoring instruments, and cascade stainless steel and aluminum. The cast stainless steel premix swirlers of the Dry Low NO_x design have been purchased.

Cascade fabrication and instrumentation is well underway. All airfoils have been wire-EDM machined to the appropriate sizes. The thin-walled airfoil has been machined and delivered to the vendor for installation of the thermocouples. Cascade side plates and flanges are complete. Some delays have occurred due to machine malfunctions and machine shop work loading. The schedule does allow for such inevitabilities though, so that current progress will still meet the desired test start date.

The premix swirlers, cast as half-scale of the Dry Low NO_x geometry, were received and installed in the head piece. All components dealing with the upstream flow preparation prior to the cascade are about 90% complete as a group.

The pressure vessel to contain the upstream flow components was prepared for installation, with all penetrations and instrumentation ports welded on. Installation of piping, orifice station, and control valves was begun.

Routing of cabling and instrumentation leads from the test rig to the control room was also started. The data acquisition system is well along in preparation. The computer is operational with the installed software for database and graphical interface functions. Programming is in progress to allow the receipt, conversion, and display of test facility data.

A cascade test plan has been drawn up, summarizing the necessary test sequence and ranges of test conditions to be performed. The cascade will be pressure tested and any necessary support or fixture modifications made. The cascade will then be re-assembled with the static pressure measuring airfoils installed, the 1/2-scale combustor system components installed in the flow preparation vessel, and all relevant instrumentation ported to the computer system. Flow testing of the cascade will begin with measurement of the Mach number distributions in the two central flow passages. Other flow tests will include measurement of the inlet temperature and pressure profiles, as well as the inlet and exit turbulence intensity levels of the mainstream. Flow testing will continue in this period with variations to inlet turbulence

Figure 5 Turbine Inlet Nozzle Heat Transfer Cascade Facility

OPERATING POINT
 MASS FLOW = 16.7 PPS
 INLET T = 70°F

intensity levels obtained by the use of different perforated plates. The upstream component configuration will also be altered in additional tests to provide a more generic swirl and turbulence generation means.