System Definition and Analysis Gas-Fired Industrial Advanced Turbine Systems

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TASK 6- SYSTEM DEFINITION AND ANALYSIS

6.0 TASK OBJECTIVE

Define and analyze an engine system based on the gas fuel Advanced Turbine from Task 3.

Using the cycle results of Task 3, a technical effort was started for Task 6 which would establish the definition of the engine flowpath and the key engine component systems.

The key engine systems are:

- Gas Turbine Engine Overall Flowpath
- Booster (Low Pressure Compressor)
- Intercooler
- High Pressure Compressor
- Combustor
- High Pressure Turbine
- Low Pressure Turbine & Materials
- Engine System Packaging
- Power Plant Configurations

As noted earlier in Task 3, the design objective is to use the GE90 engine as the platform for the GE Industrial Advanced Turbine System. This objective sets the bounds for the engine flowpath and component systems.

6.1 ADVANCED TURBINE SYSTEM ENGINE FLOWPATH, BOOSTER AND POWER TURBINE DESIGN

The flowpath for the Advanced Turbine System (ATS) for the Task 6 cycle has been completed and incorporates as much hardware from the GE90 as possible. This unit is an intercooled and turbocooled design which pushes the core compressor to a high exit pressure, but limits high pressure turbine inlet temperature due to NO_x emissions. The intercooler allows the high-pressure compressor to pass increased mass flow of air, which results in a larger power output. This unit uses the GE90 core compressor, the GE90 high pressure turbine, and the GE90 turbine mid frame. The power turbine is a GE90 type turbine, and the booster is a new design. Using General Electric's developed software called Flowpath, which uses engine cycle data and key mechanical design algorithms, an engine cross section was developed for the General Electric's Industrial Advanced Turbine System. The resulting Flowpath developed cross section of the engine is shown in Figure 6.1.1.

Figure 6.1.1. Industrial ATS Flowpath

The booster is a 4 stage design which is sized for a pressure ratio of 3.1 and an inlet corrected flow of 613 lbs/sec with fixed inlet guide vanes. Figure 6.1.2 shows the flowpath cross section of the booster. The booster sized in this fashion will allow operation at 3,000 rpm with fixed IGVs to produce the cycle pressure ratio of 2.4 and cycle inlet corrected flow of 496 lbs/sec. The booster operation at 3,000 rpm is shown on the map given in Figure 6.1.3. The map indicates that the booster is expected to operate at about 90.5% adiabatic efficiency. The booster operation at 3,600 rpm with IGVs closed is shown on the map given in Figure 6.1.4. This map indicates that at 3,600 rpm the booster is expected to operate at about 89.0% adiabatic efficiency. Since the majority of the cycle pressure rise occurs in the core, this should not result in a big loss in simple and combined cycle performance. The performance implied by the maps should be considered as preliminary.

The intercooler produces an increased core mass flow (compared to the GE90), which causes the high pressure turbine to operate at about a 17% larger flow function and at about a 4% slower speed than the GE90. The increased flow function can possibly be accommodated by either designing all new turbine airfoils or by rotation of the existing turbine airfoils. A preliminary look was taken at rotating the first stage vane and blade airfoils open in order to match the cycle flow function. The estimated performance agreed well with the cycle deck. An initial estimate shows that the HPT efficiency increases about 0.8% points if all new airfoils are designed. It is felt that the performance level of the HPT in the cycle deck can be obtained.

The power turbine uses a GE90 type flowpath as can be seen in Figure 6.1.5. The inlet to the power turbine has been sized so that the GE90 mid turbine frame can be used. The power turbine exit flow area has been sized to give a maximum value of annulus area times rotor rpm (AN^2) of 45×10^9 which results in an exit Mach number of 0.56 at the design point. A preliminary trade study was done to determine the number of power turbine stages needed to give high efficiency. It can be seen, in Figure 6.1.6, that only a small insignificant gain in efficiency is possible in going from 7 stages to 8 stages, and that a marginal gain of about 0.35% points occurs in going from a 6 stage to a 7 stage design. It was decided to pick a 6 stage design at this time. However, a more detailed trade study in the future might show that a 7 stage design would make sense from an economic point of view.

The power turbine design point was taken at 3,285 rpm to minimize the increase in loading when the power turbine is operated at 3,000 rpm. The power turbine efficiency at 3,285 rpm was estimated at 94.6%. The off-design power turbine efficiency at 3,600 rpm is estimated to be 94.5%, with a 1% drop in flow function from the design point. There is essentially no loss in performance between the design point and 3,600 rpm. The off-design power turbine efficiency at 3,000 rpm is estimated to be 93.9%, with about a 1% increasing flow function from the design point. The power turbine efficiency can be expected to drop about 0.6% points when operating at 3,000 rpm in comparison to performance at 3,600 rpm. The power turbine was assumed to have an exhaust diffuser with a 2:1 area ratio and a recovery factor of 0.7, which implies either a cold end or hot end drive. A diffuser with a recovery factor of 0.8, which implies only a cold end drive, would result in another 0.7 MW in output. A 15-in. H_2O stack loss in static pressure was assumed to occur after the exhaust diffuser. The power turbine efficiency estimate should be reevaluated later in the detail design process.

Figure 6.1.2. Industrial ATS Booster Stages

Figure 6.1.3. Booster Performance Map (3,000 rpm)

Figure 6.1.4. Booster Performance Map (3,600 rpm)

Figure 6.1.5. Industrial ATS High-Pressure Core and Power Turbine

Figure 6.1.6. Power Turbine Efficiency Change vs. Number of Stages

The cycle deck has assumed a 95% power turbine efficiency at 3,600 rpm, which is 0.5 point higher than the current estimate using a 6 stage turbine. It might be possible to attain the 95.0% level with further optimization and/or adding an additional stage. The current estimate shows that a power turbine with a high level of performance should be attainable.

The net change in simple cycle efficiency between 3,600 rpm and 3,000 rpm is that the 3,000 rpm operation is estimated to be about 0.3% poorer. The performance loss for combined cycle operation is estimated to be only about 0.1% lower. These estimates are based on cycle derivatives for the booster and the power turbine and their estimated change in efficiencies from 3,600 rpm to 3,000 rpm. The small difference in performance between 3,600 rpm and 3,000 rpm are well worth being able to use the same booster hardware and use the same power turbine hardware for both 3,600 rpm and 3,000 rpm operation. The core is the same for both 3,600 rpm and 3,000 rpm. Designing the power turbine at 3,285 rpm and designing the booster at 3,600 rpm at a higher pressure ratio and larger corrected flow than in the cycle resulted in a versatile ATS design.

6.2 INTERCOOLER DESIGN

Two vendors were surveyed about the intercooler design performance. Both vendors, Allied Signal and General Atomics, indicated that the desired performance could be met. The indicated size of the intercooler was estimated to be 9 feet in diameter and 10 feet in length. Figure 6.2.1 shows the performance and shape of the intercooler.

INTERCOOLER

INTERCOOLER DESIGN

PARAMETERS	DESIGN GOALS	VENDOR PROJECTIONS
AMBIENT TEMP(°F)	59	59
ALTITUDE	SEA LEVEL	SEA LEVEL
INLET AIR FLOW(LBS/SEC)	485	485
INLET AIR TEMP(°F)	220	220
INLET PRESSURE(Psia)	35	35
INLET MACH NUMBER	0.15	0.15
WATER INLET TEMP(°F)	84	84
EXIT AIR TEMP(°F)	90	90
AIR SIDE PRESSURE DROP(%)	2	1.6
(FLANGE TO FLANGE)		
WATER FLOW (LBS/SEC)		1251
WATER OUTLET TEMP(°F)		96

Figure 6.2.1. Intercooler Shape and Performance

6.3 ATS DLE COMBUSTOR PRELIMINARY DESIGN

The ATS combustor design is based on the use of Lean Premix Combustion technology developed on the LM6000 DLE engine, which was introduced to field service in December 1994. The ATS combustor utilizes a scaled triple dome configuration of the LM6000 design. The DACRS III premixers developed for the LM6000 are scaled up to the ATS design point.

The preliminary design process included the steps of (1) determining the design point, (2) determining the combustor fuel air ratio needed to satisfy the NO_x goals set in the program, (3) sizing the triple annular combustor in accordance with the design methodology developed on the LM6000 DLE program, (4) sizing the premixers in accordance with above methodology, (5) adjusting the diffuser design, and (6) performing trade studies.

GOALS

The design goals for the combustor include the emissions requirements set by DOE, which are 8 ppm NO_x emissions and 20 ppm CO emissions. Other design goals include the profile and pattern factor, as well as combustion system pressure drop. These are shown in Table 6.3.1. The design intent is to meet the above emissions goals from 60% power to 100% power

TABLE 6.3.1

Emissions Goals

	Initial goal	Ultimate goal
NO _x	<9 ppm	<8 ppm
CO	<20 ppm	<20 ppm

6.3.1 DOUBLE ANNULAR COUNTER-ROTATING SWIRLER (DACRS) PREMIXER DESIGN

The DACRS series of premixers was developed for application to lean premixed combustion systems for the LM series of gas turbines. DACRS stands for Double Annular Counter Rotating Swirler. Figure 6.3.1.1 shows a cross section through a DACRS III premixer. These compact premixers utilize two counter-rotating swirlers followed by a mixing duct. The inner swirler has a centerbody to keep the air/fuel mixture from separating. This centerbody also allowed the injection of a small quantity of air at its tip to help avoid separated flows. The mixing duct is conical, with the smaller diameter at the exit plane. The conical shape of the duct continuously accelerates the mixture, avoiding buildup of boundary layers on the duct surface. These features of the DACRS family of premixers make the design flashback-resistant. The small residence time of these devices (<1 msec) and the avoidance of any recirculation within them ensures that autoignition will not occur within the mixing duct at any operating

conditions. The DACRS-I design utilized radial fuel injection from the centerbody, the DACRS-II design utilized fuel injection through spokes located on the centerbody, and the DACRS III design utilizes hollow outer swirl vanes through which fuel is injected into the air stream.

For the determination of the fuel air ratios at design point, the NO_x emissions obtained by Leonard and Correa were utilized. This assumes that the DACRS-III premixer performance will be improved from the published data of Joshi et.al. (1994) to achieve the level of mixing achieved by Leonard and Correa (1990). Figure 6.3.1.2 shows the NO_x correlations used for these studies.

DESIGN

VELOCITY PROFILES

Figure 6.3.1.1. Double Annular Counter-Rotating Swirler (DACRS)

NOx Emissions from Premixer to be reduced to CRDC Line
Combustor design to be modified to dump heat shield cooling air into
the premixers to reduce flame temperature.

Figure 6.3.1.2. DACRS NOx Emissions vs. Flame Temperature

6.3.2 SIZING STUDY BASED ON LM6000 DLE COMBUSTOR DESIGN

The LM6000 DLE combustor design, shown in Figure 6.3.2.1, utilizes lean premixed combustion from ignition to full power. The original LM6000 diffuser and compressor rearframe (which houses the combustor) were modified to accommodate the larger DLE combustor. The DLE combustor is configured as a triple-dome combustor. Most of the combustor air passes through the premixers, with only a small fraction of air being used for liner cooling at the aft ends and for heat shield cooling in the dome. The heat shield cooling air is introduced into the dome downstream of the primary combustion zone, avoiding dilution of the primary zone in any way, which permits the combustor to operate at lower flame temperatures. The inner dome has 15 premixers; the middle and the outer each have 30. The premixers are designed to be removable to permit ease of replacement and repair. The combustor itself is mounted from the front end, unlike the non-DLE combustor, which is mounted from the aft end. The combustor utilizes a set of igniters on the outer liner. Ignition is achieved by fueling a set of 3 outer dome premixers in line with the igniters, in addition to the middle dome. These 3 premixers are switched off as soon as ignition is detected in the middle dome. The middle dome is the pilot dome that stays lit at all times. As engine power is increased, the inner and the outer domes are lit in sequence. Additional operability is obtained by modulating combustion air flow by bleeding air flow from the engine.

Figure 6.3.2.1. LM6000 DLE Combustor Cross-Section

6.3.3 DESIGN POINT CONDITIONS

The design point conditions used for the preliminary design of the ATS DLE combustor are shown in Table 6.3.3.1.

TABLE 6.3.3.1

ATS DLE Combustor Design Conditions

Parameter	Units	
T2	deg F	59
P3	psia	760
T3	deg F	955
W3	PPS	463
W36	PPS	391
Wf36	pph	47249
LHV of fuel	Btu/lb	19000
W41	PPS	453.24
T41	deg F	2684

6.3.4 COMBUSTOR AIRFLOWS

The air flow diagram and design point cycle conditions are shown in Figure 6.3.4.1. The air flow from the diffuser, which is about 94% of W25, is split into several streams. One of the streams is diverted to the turbocooler, where the air is cooled and compressed further for cooling of the turbine nozzle and combustor. The rest of the air is passed into the combustor. The cooling air returning from the turbocooler is used to cool the turbine nozzle diaphragm, as shown Figure 6.3.6.4, as well as the combustor liners and dome. Details of the preliminary cooling design is described in Section 6.3.6.

Figure 6.3.4.1. ATS DLE Combustor Air Flow

6.3.5 DESIGN DESCRIPTION

The air flow split between the combustion air (burning) and cooling air for the heat shield and liners is determined by cycle conditions. The combustor utilizes the maximum air available to decrease the combustion flame temperatures and thereby minimize NO_x emissions. The three-dome air flow splits were set by assuming that the dome heights were of similar dimensions, and that the dome reference velocity was equal in each dome. This design methodology is similar to that used on the LM6000 DLE combustor design. The inner diameter of the liner is determined by geometric and fit-up requirements, and was set at 25" to give sufficient space for the design of an impingement baffle for the liner. A heat shield thickness of 0.375 in. was assumed for the heat shield in order to calculate the flowpath inner diameter. A dome reference velocity of 40 fps (cold) was assumed, and the outer diameter was calculated for the inner dome, based on the cycle parameters shown in Table 6.3.3.1, and the air flow split, shown in Figure 6.3.4.1. A heat shield thickness of 0.375 in. was again assumed to calculate the inner flow path diameter for the middle dome, and the method outlined above was repeated to calculate the outer diameter of the middle dome. The methodology was repeated to calculate the outer liner diameter. A summary of the dome sizing study is shown in Table 6.3.5.1. Once the dome diameters were obtained, a trade study was done to evaluate the number of premixers that will be required in each dome. A summary of the trade study is shown in Table 6.3.5.2. One of the design considerations is the ease and simplicity of pre-mixer removal from the engine. This consideration, and the requirements for case integrity, increases with the number of premixers. Although beneficial from an emissions point of view (increased fuel sources), this has to be traded with cost (more premixers increase cost) and the integrity of the case (thinner web between pre-mixer windows leads to softer case). For the preliminary design, shown in Figure 6.3.5.1, 36 premixers were chosen for the middle and outer domes and 18 were picked for the inner dome. This is similar to the LM6000 DLE design, as shown in Figure 6.3.2.1, where each dome has, respectively, 30/30/15 premixers.

TABLE 6.3.5.1

Combustor Dome Sizing Study Summary.

INNER DOME DESIGN		
Heat shield thickness	0.009525m	0.375 in
Inner liner radius =	0.327025 m	12.5 in
Dome outer radius	0.400790075 m =	15.779137 in.
Dome height =	0.073765075 m =	2.9041368 in.
MIDDLE DOME DESIGN		
Inner radius of the middle dome	0.410315075 m	16.154137 in.
Dome outer radius	0.492797763 m =	19.401487 in.
Dome height =	0.082482689 m =	3.2473499 in.
OUTER DOME DESIGN		
Inner radius of the outer dome	0.502322763 m	19.77649in
Dome outer radius	0.58215031 m	22.919304 in.
Outer liner radius	0.59167531 m	23.294304 in.
Case radius	0.61707531 m	24.294304 in.

TABLE 6.3.5.2

Summary of Premixer Number Trade Study

Outer Engine Case Circumference =				3.87719 m
Outer Case Circumference Spacing per Number of Premixers				
# of Pre Mixers				
	20	0.19385 m=	7.63228 in.	
	30	0.12924 m =	5.08818 in.	
	36	0.1077 m =	4.24015 in.	
	40	0.09693 m =	3.81614 in,	
	45	0.08616m =	3.39212 in.	
Dome Premixer Spacing				
Dome Location	#of Premixers	Dome Circumferential Spacing		
Inner	18	0.12702 m =5.00109 in.		
	20	0.1 1432 m =4.50098 in.		
	30	0.07621 m =3.00065 in.		
	36	0.06351 m =2.50054 in.		
	40	0.05716 m =2.25049 in.		
	45	0.05081 m =2.00043 in.		
Middle	30	0.09457 m =3.72337 in.		
	36	0.07881 m =3.10281 in.		
	40	0.07093 m =2.79253 in.		
	45	0.06304 m =2.48225 in.		
Outer	30	0.11356 m =4.47109 in.		
	36	0.09463 m =3.72591 in.		
	40	0.08517 m =3.35331 in.		
	45	0.07571 m =2.98072 in.

Figure 6.3.5.1. Preliminary ATS Combustor Layout

Having picked 36/36/18 premixers in the outer, middle, and inner domes the premixer diameters were determined by scaling the DACRS-III premixer design from the LM6000 DLE combustor design. The premixer design parameters are summarized in Table 6.3.5.3. The mixing length of the premixer was reduced from the LM6000 value of 2 in. in order to limit exposure to auto-ignition in the premixers due to the increased severity of the cycle. A dimensional check was performed to make sure that the premixers did not interfere with each other in the combustor and were removable through an appropriately sized window in the case.

TABLE 6.3.5.3

Premixer Diameters Scaled From LM6000 DLE Combustor

<u>Dome</u>	<u>Swirler Diameter (inches)</u>	<u>Exit Diameter (inches)</u>	<u>Mixing Length (inches)</u>
Inner	2.62	1.298	1.56
Middle	2.55	1.298	1.56
Outer	2.7	1.4	1.56

The combustor length was set at 7.0 in. in order to generate sufficient residence time at temperature to oxidize CO formed in the primary combustor zone. This combustor length, coupled with the need to pull the premixers forward for disassembly, required the diffuser to be extremely short, so that the overall increase in the diffuser/combustor length could be minimized. Based on the LM6000 DLE design, a diffuser length of 5 in. is carried in this design.

6.3.6 COMBUSTOR COOLING SYSTEM DESIGN

The combustor cooling system is designed to enable the combustor to operate for 50,000 hours (with repair) on natural gas while using a minimum of cooling air consistent with the stated emission goals. The cooling system has been sized for baseload operation at a flame temperature of 2853F and compressor discharge pressure of 760 psia. Cooling design assumptions include uniform flame temperature, negligible liner film cooling benefit from the outer and inner heatshield centerbodies, usage of impingement baffles that employ crossflow reductions features, TBC surface temperature capability to 2400F, and a 100 degree temperature rise in cooling air that passes through the turbine inlet nozzle to the inner liner cooling circuit.

The ATS liners are cooled by a combination of impingement, backside forced convection, and hot side film cooling. Cooling air is supplied from the turbocooler at 300F (baseload condition) and 841.6 psia and delivered to a distribution plenum within the combustor case surrounding the combustor. The liner cooling air is directed to the liners through impingement baffles covering both the outer and inner liners. 1.8%W25 cooling air is impinged to the outer liner and 1.2%W25 to the inner liner. 70% of the spent impingement air (1.3% W25 outer inner and

0.8% W25 inner liner) flows forward between the baffles and liner to cool the forward portion of the liners via forced convection. This cooling air is then dumped into the plenum upstream of the dome and enters the combustor as heatshield cooling air or premixer combustion air. The remaining 30% spent impingement air (0.5% outer liner and 0.4% inner liner) flows to the rear of the liners to cool the aft portion by forced convection before discharging through exit slots into the flowpath, approximately 1.5 inches from the end of the liners. Film cooling from the exit slots cool the last 1.5 in. of the liners. The hotside surface of the liners is protected by a layer of 0.035-in. thick dense microcracked thermal barrier coating (TBC) applied over a 0.007-in. bond coat.

The backside surface of each liner is manufactured with surface enhancement features which multiply the impingement and forced convection heat transfer. The impingement region features a textured surface that effectively multiplies the impingement heat transfer by a factor of 1.5, while the forced convection region is enhanced by turbulators which multiply the effective heat transfer coefficient by a factor of 1.8.

Performance of the liner cooling system at baseload conditions results in maximum liner metal temperatures of less than 1475F and maximum TBC surface temperatures slightly lower than 2300F. The heatshield components protect the dome region of the combustor and effectively divide the combustor into three dome regions with the heatshield centerbodies. The heatshields are mounted to the domeplate and employ the same advanced cooling design as used in the LM6000 DLE. The source of cooling air is a combination of CDP air and the portion of the liner cooling air discharged forward into the compressor discharge plenum. The heatshields require a total of 5.1% W25 cooling air (3% W25 CDP air and 2.1% W25 spent liner cooling air). The heatshield faceplate is cooled by impingement through the domeplate. The spent impingement cooling air from the faceplate flows through the heatshield centerbodies that are cooled by enhanced forced convection. The centerbody enhanced convection is provided by full-length pin fins that span the channel height of the centerbody. Textured surfaces are employed, on the faceplate and up to the pin bank on the centerbody portion of the heatshield, that effectively multiply the heat transfer coefficient by a factor of 1.5. The hotside of the heatshields are protected by a layer of 0.035-in. thick dense microcracked thermal barrier coating (TBC) applied over a 0.005-in. bond coat.

Maximum heatshield metal temperatures are expected to run at approximately 1600F on the faceplate, with a TBC surface temperature less than 2300F, and at 1550F on the centerbodies, with a TBC surface temperature of less than 2300F.

The design goals and assumptions made for the combustor liner cooling are shown in Figures 6.3.6.1 thru 6.3.6.6.

COMBUSTOR COOLING SYSTEM

KEY ASSUMPTIONS:

- T75 = 300.0 F T70 = 955.0 F
P75 = 841.6 PSIA P70 = 752.4 PSIA

- T405= 2853.0 F (T_{flame})
P405= 717.2 PSIA

- NATURAL GAS
- UNIFORM FLAME TEMPERATURE (NO HOT STREAKS)
- NEGLECT LINER FILM COOLING BENEFIT FROM OUTER AND INNER HEATSHIELD CENTERBODIES
- IMPINGEMENT BAFFLES EMPLOY CROSSFLOW REDUCTION FEATURES
- TBC SURFACE TEMPERATURE CAPABILITY TO 2400 F
- COOLING AIR SUPPLIED TO INNER LINER INCREASES 100 F PASSING THROUGH TURBINE INLET NOZZLE

Figure 6.3.6.1. ATS Combustor Cooling System - Assumptions

COMBUSTOR COOLING SYSTEM

LINERS:

- COOLING AIR SUPPLIED BY TURBOCOOLER
- IMPINGEMENT, BACKSIDE FORCED CONVECTION, AND FILM COOLING (AT COMBUSTOR EXIT)
- COOLING SYSTEM FEATURES:
 - ENHANCED BACKSIDE CONVECTION
 - IMPINGED TEXTURED SURFACES
(1.5 X IMPINGEMENT FORCED CONVECTION)
 - TURBULATED PASSAGES
(1.8 X CHANNEL FORCED CONVECTION)
 - THICK TBC (0.035 INCHES DENSE MICROCRACKED)
 - CONVENTIONAL SLOT AT COMBUSTOR EXIT
- 1.8 % W25 OUTER LINER COOLING
 - 1.3 % W25 FLOWS TO DOME
 - 0.5 % W25 FLOWS TO EXIT SLOT
- 1.2 % W25 INNER LINER COOLING
 - 0.8% W25 FLOWS TO DOME
 - 0.4% W25 FLOWS TO EXIT SLOT

Figure 6.3.6.2. ATS Combustor Cooling System - Liners

COMBUSTOR COOLING SYSTEM

HEATSHIELDS:

- THERMAL PROTECTION TO DOME STRUCTURE
- DIVIDES COMBUSTOR INTO THREE DOME REGIONS
- COOLING AIR SUPPLIED FROM COMPRESSOR DISCHARGE PLUS 70% OF TOTAL SPENT LINER COOLING
- COOLING SYSTEM FEATURES:
 - FACEPLATE IMPINGEMENT COOLED
 - TEXTURED SURFACE
(1.5 X IMPINGEMENT FORCED CONVECTION)
 - CENTERBODY FORCED CONVECTION COOLED
 - SPENT IMPINGEMENT AIR FROM FACEPLATE
 - FULL LENGTH PIN FINS
(3.5 X SIMPLE FORCED CONVECTION)
 - THICK TBC (0.035 INCHES DENSE MICROCRACKED)
- 5.1% W25 TOTAL COOLING
 - 3% W25 CDP AIR
 - 2.1% W25 SPENT LINER COOLING AIR

Figure 6.3.6.3. ATS Combustor Cooling System - Heat Shields

COMBUSTOR COOLING SYSTEM

COOLING FLOW DISTRIBUTION :

FLows %W25

Figure 6.3.6.4. ATS Combustor Cooling System - Cooling Flow Distribution

COMBUSTOR COOLING SYSTEM

COMPONENT TEMPERATURES :

Figure 6.3.6.5. ATS Combustor Cooling System - Temperatures

COMBUSTOR COOLING SYSTEM:

SUMMARY:

- LINER IMPINGED WITH TURBOCOOLER AIR
- 70% OF SPENT LINER COOLING AIR MIXES WITH COMPRESSOR DISCHARGE AIR AND INTRODUCED BACK INTO COMBUSTOR
- IMPINGEMENT SYSTEM REQUIRES CROSSFLOW REDUCTION FEATURE
- HEATSHIELDS EMPLOY ADVANCED COOLING USED IN LM6000 DLE APPLICATION

Figure 6.3.6.6. ATS Combustor Cooling System - Summary

6.3.7 ADDITIONAL DEVELOPMENTS NEEDED

Premixers

The current DACRS III premixers will have to be developed further to improve the emissions performance of these devices. Leonard and Correa NOx emissions correlation (fully premixed Methane/air flames) has been used for predicting NOx emissions from the combustor.

Diffuser

The ATS combustor design uses a 5 in. long diffuser. In order to limit diffuser dump losses significant development will have to be undertaken to design a low loss ultra short diffuser. Trades between diffuser losses and combustor/engine lengths will have to be made to optimize the design of the diffuser.

Highly Skewed Part Power Profiles

The LM6000 combustor operates with the middle dome lit at all times. As load on the engine is increased, the staging sequence ignites the inner, and then the outer, as shown in Figure 6.3.7.1. This results in highly skewed temperature profiles entering the turbine nozzle. The nozzle and blade design will have to take the skewed profiles at part power into consideration.

LM6000 DLE COMBUSTOR STAGING

Figure 6.3.7.1. LM6000 DLE Combustor Staging

Combustor Acoustics and Operability

Lean premixed combustors have been shown to have increased dynamic pressures. The ATS combustor is also a lean premixed combustion system and will require on engine development to ensure that dynamic pressures are within design limits. The ATS combustor is designed to operate within a narrow flame temperature window with the use of air flow modulation. The transient dynamics of the turbocooler and the rest of the engine will have to be modeled in the design of the control system for the ATS engine.

Hot Fuel/Coal Gas Capability

The combustor operates at part power with the inner and outer domes switched off. The control system requires separate shut off valves for each of the domes. The design of the fuel system will have to take into consideration the impacts of heated fuels if the cycle design requires it. Additional analysis will be required to establish the effects of burning coal gas in the compact design of the ATS combustor.

Sealing Technology

The premixer to dome joint, the outer/inner liner to turbine nozzle joint, and the igniter/flame detector penetrations have been assumed not to leak, so that all air can be used for combustion to minimize NO_x emissions. To achieve this goal, current sealing concepts may need further development.

6.3.8 CONCLUSIONS

The preliminary design of the ATS combustor has been completed. The lean premixed natural gas fired combustor design utilizes the concept and operating modes proven on the LM6000 DLE program. However technical challenges in premixer performance, sealing, operability/staging effects, diffuser design, and coal gas capability will be addressed in the follow-on program. The detail design process will address operability, system integration, and mechanical design of the combustor.

6.4 HIGH PRESSURE TURBINE COOLING DESIGN

Preliminary turbine cooling designs were developed for the Stage 1 and Stage 2 high-pressure turbine vanes and blades to support the evolution of the Advanced Turbine System cycle and establish the design constraints necessary for a successful detail design.

The ATS turbocooler provides cooling flow to the combustor liners and high pressure turbine vane and bands at reduced temperatures and increased pressures, as compared to traditional CDP (Compressor Discharge Pressure) air-cooled designs. Flow from the turbocooler cools the combustor liner, the HPT vane outer band, and HPT vane as film as well as non-film impingement cooling. The post-impingement spent cooling air is used to cool the combustor inner liner, vane inner band, and forward seal. Figure 6.4.2 indicates the cooling flow from the turbocooler.

Coolant supply to the remaining HPT components is based upon a more traditional configuration: CDP air cools the HPT Stage 1 blade; 7th Stage Bleed cools the HPT Stage 2 blade; and interstage air cools the HPT Stage 2 vane.

In order to achieve a cost effective design, existing GE90 hardware was used where possible. The ATS design flow function increased 19% relative to the GE90's. This will require a rotation of the airfoils in the turbine and may require new airfoils.

The gas temperature profile is based upon the ATS profile and pattern factor, which is very flat compared to other DLE (Dry Low Emission) engines. This effect can be seen in Figure 6.4.4.

Figures 6.4.1 thru 6.4.24 outline the critical design areas that have been analyzed for the high-pressure turbine.

Cycle Conditions & Gas Temperature Profiles

The design cycle conditions for the ATS follow:

ATS Cycle Conditions

T3	(F)	955.4
P3	(psi)	760.0
T41	(F)	2684.3
P4	(psi)	717.7
T42	(F)	1885.0
W25	(pps)	491.9
W41	(pps)	453.2
XN25	(rpm)	9433

Figure 6.4.1. ATS Turbine Design Considerations

Figure 6.4.2. Turbocooler Cooling Flow

Cooling Flow Status

Part	From T/C	From CDP
Combustor Aft Seal Leakages	1.80	
Combustor Outer Liner	1.50	
Combustor Inner Liner	1.50	
HPT Vane Inner Band	0.92	
HPT Vane Outer Band	1.03	
HPT Vane Film	1.38	
HPT Vane Trailing Edge	0.98	
HPT Shroud	<u>0.50</u>	
Total	9.61	9.61
HPT Blade		2.60
Forward Combustor	0.60	
TC Turbine Flow		<u>1.50</u>

**Cycle carries 0.97% W25 more flow for design margin

Gas Temperature Profile Impact of Pattern and Profile Factor

The following is a listing of the pitchline gas temperatures for the HPT Stage 1 & 2 vanes and blades. As a comparison, the gas temperatures of the airfoils are shown with pattern and profile factors similar to the LM6000 Dry Low Emission combustor. As can be seen, the impact of the much lower ATS pattern and profile factors reduces the pitchline gas temperatures approximately 100° F.

	ATS	ATS
	ATS PF	LM6000 DLE PF
Pattern Factor	0.070	0.132
Profile Factor	0.030	0.092
HPT Stage 1 Vane		
T Hot Streak (F)	2966	3082
HPT Stage 1 Blade		
Ttb (F)	2682	2794
HPT Stage 2 Vane		
T Hot Streak (F)	2308	2409
HPT Stage 2 Blade		
Ttb (F)	2114	2209

Figure 6.4.3. ATS Pattern and Profile Factors

ATS HPT Stage 1 Blade Relative Gas Temperature (Tib)

An additional impact of the reduced ATS profile and pattern factors is an increase in the gas temperatures in the hub and tip of the airfoils. This plot indicates the difference of the ATS and LM6000 DLE profiles on the HPT Stage 1 Blade relative gas temperatures (Tib).

Figure 6.4.4. ATS HPT Stage 1 Gas Temperature

HPT Stage 1 Vane

The design goal for the HPT stage 1 vane is 3000 cycles at the ATS design point with 300F intercooler air. The life limiting location of the N6 airfoil was found to be the trailing edge, which is potentially a repairable location. The next life limiting point was the suction side panel. An optimized thermal design is critical to achieving life.

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 includes internal impingement for increased heat transfer. A portion of that flow exhausts through film holes and trailing edge pressure side bleed as in traditional designs, but over half of the flow is captured at the hub of the airfoil and used to cool the combustor inner liner and HPT stage 1 inner bands. The total flow into the HPT vane is 5.73%W25. Of that, 1.38%W25 is used as film at the leading edge and .98%W25 is pressure side bleed cooling the trailing edge. The remaining 3.37%W25 provides a high impingement heat transfer coefficient on the internal surface of the airfoil, then passes through the vane to cool the inner band, combustor inner liner and forward leaf seal. The ability to reuse the spent vane cooling air is only possible due to the increased pressure 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 cooling air pressure supplying the combustor inner liner, which is the vane cooling air exhaust pressure, is a design constraint.

HPT Stage 1 Vane

Thermal barrier coating is important in achieving the life of the stage 1 vane. TBC outer surface temperatures were limited to 2350 °F and TBC bond coat interface temperatures were limited to 1900 °F in order to minimize the risk of the TBC in sintering and increasing the TBC thermal conductivity. Two TBC thickness designs were analyzed to balance the cooling design of the airfoil. Both configurations used 2.36%W25 in film and 3.66%W25 non-film flow. Both also had a life limiting location at the trailing edge with 3000 cycles nominal N6 material properties and 1500 cycles with minimum properties. This is potentially a repairable location. The first configuration had 10 mils of TBC on the leading edge forward, both pressure and suction side, and 15 mils in the aft cavity. The life of the suction side panel for this design was 3700 cycles for minimum N6 material properties. This is not a repairable location. In order to provide a more balanced temperature distribution, the TBC thickness of the second configuration was modified to 10 mils leading edge, and 10 mils pressure side aft and 20 mils suction side aft. This reduces the aft suction side panel temperature and increases the pressure side aft temperatures which increased the suction side panel life to over 10000 cycles for minimum N6 properties.

In conclusion, an HPT Stage 1 Vane design is dependent on a optimum thermal design which includes thermal barrier coating and N6 properties. The airfoil will be limited at the trailing edge and only achieves 1500 cycles versus the 3000 required. However, this is considered a repairable location. The next limiting location is the suction side panel which will achieve the design goal of 3000 cycles.

HPT Stage 1 Vane

Hot Streak Temperature	(F)	2966
Coolant Temperature	(F)	300
Pcoolant/Pgas		1.10
TBC Thickness	(mils)	10/15
Wc	(%W25)	6.02
	Film	2.36
	Non Film	3.66
Life	(cycles)	3000
TBC Outer Temperature	(F)	2350
Bond Coat Temperature	(F)	1700

Figure 6.4.7. ATS HPT Stage 1 Vane - Design Temperature Values

ATS Stage 1 Vane Estimated Leakage Losses

	ATS (%W25)
Band Spline Seals	
Inner	0.27
Outer	0.28
Outer Aft Leaf Seal	0.35
Inner Mount	0.25
Forward Leaf Seals	
Inner	0.95
Outer	0.85
Vane (Airfoil)	
Forward & Aft Circuit Film	1.38
Trailing Edge Film	0.98
Bands	
Inner	0.4
Outer	<u>0.4</u>
Total	6.11

Figure 6.4.8. ATS HPT Stage 1 Vane Estimated Leakage Losses

Figure 6.4.9. ATS HPT Stage 1 Vane Cooling and Leakage Flows

T BULK= 1516
 SEGMENT= 5
 RADIUS= 17.75113

Figure 6.4.10. ATS HPT Stage 1 Vane - Metal Temperatures

HPT Stage 1 Vane Thermal & Stress Analysis

Objective: Determine whether a viable HPT Stage 1 Vane airfoil design was achievable using the 300 F intercooler air.

Studies: GE90 HPT Stage 1 Vane (N5) 3D Model was used to run Ansys thermal & stress analysis.

	1	2
TBC Thickness	10 mils LE fwd/ 15 mils TE aft	10 mils LE fwd/10 mils ps aft/20 mils ss aft
Cooling Flow	2.36% W25 Film; 3.66% W25 NonFilm	2.36% W25 Film; 3.66% W25 NonFilm

Results: ☐ Life limiting location occurred at trailing edge:

N5: 1400 cycles nom; 600 cycles min

N6: 3000 cycles nom; 1500 cycles min (scaled)

☐ Suction side panel:

Iter 1: N5: 1400 cycles min

N6: 3700 cycles min

Iter 2: Reduce aft suction side panel temperature; Increase pressure side aft temp

N5: 4000 cycles min

N6: > 10000 cycles min

HPT Stage 1 Blade

The design goal for the HPT stage 1 blade is 25000 hours at the ATS design point.. Due to the flat temperature profile factor from the combustor, the airfoil 10% span is the life limiting location for this design. Ten mils of thermal barrier coating on the blade is required to achieve life.

The ATS design cycle conditions for the HPT stage 1 blade are shown in Figure 6.32. The GE90 HPT stage 2 blade cooling circuit will be maintained for the ATS with N6 (versus N5) material properties. The circuit consists of a cold bridge leading edge, five pass serpentine and cold bridge trailing edge. The cooling flow has been reduced to 2.66% W25 with 0.84% in circuit 1, 0.82% in circuit 2 and 1.0% in circuit 3. Chambers 2-8 include the angled turbulators for enhanced heat transfer.

The coolant supply to the ATS HPT stage 1 blade is the same as found in the GE90. Compressor discharge air supplies the blade circuit as shown in Figure 6.33.

As mentioned above, the 10% span is the blade life limiting location due to the gas temperature profile. Typical metal and thermal barrier coating temperatures are shown in Figure 6.35.

HPT Stage 1 Blade Pitchline

Gas Temperature (Ttb)	(F)	2682
Coolant Temperature	(F)	972
Pcoolant/Pgas		1.10
TBC Thickness	(mils)	5-10
Wc	(%W25)	2.66

Life & Bulk Temperature Requirements

Span		10%*	50%
Life	(hours)	25000	25000
Tbulk	(F)	1527	1613

***10% span becomes life limiting location of airfoil due to temperature profile**

Figure 6.4.13. ATS HPT Stage 1 Blade Design Conditions

Figure 6.4.14. ATS HPT Stage 1 Blade Cross-Section

COOLING SUPPLY CIRCUIT

Figure 6.4.15. ATS HPT Cooling Supply Circuit

Figure 6.4.16. ATS HPT Stage 1 Blade - Metal Temperatures

T BULK = 1540
 SEGMENT = 5
 RADIUS = 17.69954

HPT Stage 2 Vane

The design goal for the HPT stage 2 vane is 3000 cycles at the ATS design point. The GE90 HPT stage 2 vane cooling design will be maintained for the ATS with N6 material properties. The design also includes four to ten mils of thermal barrier coating on the vane to achieve life.

The ATS design cycle conditions for the HPT stage 2 vane are shown in Figure 6.37. The cooling flow is approximately the same as the GE90's at 2.66% W25 using interstage air. The one concern area in the design is a potential problem of negative backflow margin or local gas ingestion at the pressure side film hole location. This occurs due to the increase in flow function of the turbine; gas pressures have increased at the stage 2 vane. To address this, a higher coolant supply pressure will be required.

ATS HPT Stage 2 Vane

Hot Streak Temperature	(F)	2409
Coolant Temperature	(F)	719
Pcoolant/Pgas		1.01*
P3	(psi)	760.0
Ps Vane 2 Source	(psi)	349.2
Pt Gas Vane 2 Fwd	(psi)	353.1
Ps Vane 2 Fwd ID	(psi)	334.2
Ps Vane 2 Aft ID	(psi)	228.7
Ps Vane 2 Aft Pitch	(psi)	247.1
TBC Thickness	(mils)	4-10
Wc	(%W25)	2.23
Life (Required)	(cycles)	3000**

*Gas ingestion pressure side film holes.
holes to achieve life.

** Requires positive backflow margin in pressure side film

Figure 6.4.18. ATS HPT Stage 2 Vane - Temperatures

HPT Stage 2 Blade

The design goal for the HPT stage 2 blade is 25000 hours at the ATS design point. Again, due to the flat temperature profile factor from the combustor, the 10% span of the airfoil is the life limiting location for this design. Ten mils of thermal barrier coating on the blade is required to achieve life.

The ATS design cycle conditions for the HPT stage 2 blade are shown in Figure 6.39. The GE90 HPT stage 2 blade cooling circuit will be maintained for the ATS with N6 (versus N5) material properties. The circuit consists of a single pass leading edge, five pass aft flowing midchord serpentine, and single pass trailing edge. The cooling flow has been increased to 0.86% W25 with 0.2% in circuit 1 0.34% in circuit 2 and 0.32% in circuit 3. All seven chambers include the angled turbulators for enhanced heat transfer.

The coolant supply to the ATS HPT stage 2 blade is the same as found in the GE90. Seventh stage air supplies the blade circuit as shown in Figure 6.40.

As mentioned above, 10% span is the blade life limiting location due to the gas temperature profile. Typical metal and thermal barrier coating temperatures are shown in Figure 6.42

HPT Stage 2 Blade Pitchline

Gas Temperature (Ttb)	(F)	2114
Coolant Temperature	(F)	916
Pcoolant/Pgas		1.43
TBC Thickness	(mils)	10
Wc	(%W25)	0.86

Life & Bulk Temperature ATS Requirements

Span		10%*	50%
Life	(hours)	25000	25000
Tbulk	(F)	1464	1566

***10% span becomes life limiting location of airfoil due to temperature profile**

Figure 6.4.20. ATS HPT Stage 2 Blade - Temperatures

Figure 6.4.21. ATS HPT Stage 2 Blade - Cross-Section

.56% BLADE (ATS)

CHARGEABLE FLOW :

.10% LEAKAGE

7TH STAGE BLEED

COOL 7TH STAGE AIR BATHES DISK BORES

Figure 6.4.22. ATS HPT Stage 2 Blade - Coolant Supply

Figure 6.4.23. ATS HPT Stage 2 Blade - Metal Temperatures

Conclusions

- The 19% increase in ATS design flow function relative to the GE90's will require a rotation of the airfoils in the turbine, and may require new airfoils.
- Due to flat combustor profile and pattern factors, the endwall, hub and tip cooling are critical.
- The stage 1 and stage 2 blade life limiting locations are at the 10-20% span for the ATS design.
- HPT stage 1 vane(N6) achieves life cooled with turbocooler air assuming the trailing edge is repairable. An optimized thermal design is critical to achieving life.
- HPT stage 1 blade achieves rupture life with 10 mils of thermal barrier coating(TBC).
- HPT stage 2 vane indicates there will be gas ingestion with the current interstage coolant supply.
- A higher coolant supply pressure will be required.
- HPT stage 2 blade achieves rupture life with 10 mils of TBC and increased coolant flow.
- Thermal Barrier Coating is critical to being able to meet the ATS design goals.

Final Design Studies Needed

- Evaluate the low engine power settings on gas temperature profiles.
- Include TBC spallation studies for final design acceptance.
- Optimize film and TBC thickness in thermal designs.
- Determine an alternative coolant supply for the HPT stage 2 vane.

6.5 ATS TURBOCOOLER SYSTEM

6.5.1 PURPOSE/FUNCTION

The purpose of the turbocooler subsystem is to enhance coolant air extracted at the engine compressor discharge and deployed to cool the HP Stage 1 vane and combustor liner. Enhancement is achieved by cooling and pressurizing the cooling air. Enhancement permits significant reduction in cooling flow and virtual elimination of film cooling of the combustor liner. The latter results in reduced emissions at reduced flame temperature. Fuel heating provides an additional cycle efficiency benefit.

The resulting effect of cooling and compression of the coolant air is a turbocooler exit (engine re-entry) temperature 660 degrees below engine extraction temperature levels. Cooling of this air is achieved by rejecting heat to metered natural gas fuel. The cooling flow requirement and fuel flow available dictate exit fuel temperatures approaching the flammability temperature, well above the fuel temperatures of conventional thermal management systems. Because of the consequent issues of fouling and fire safety, the heat exchange system is probably the principal challenge in ATS development. Enabling technologies are being explored concurrently and are discussed in subsequent sections.

Pressurization of the coolant air is achieved in a compressor operating at a pressure ratio of 1.27. This compensates for losses associated with extraction, cooling, and transport of flow in the system, and provides sufficient pressure upon re-entry to permit convective cooling of the vane and impingement (instead of film) cooling of the combustor liner.

Specifically, the cooling requirements influencing the turbocooler design include: 10.6% W25 flow required to cool the vane and combustor liner, vane leakage flow delivered at a pressure 7.2% above compressor discharge pressure (P3), and at a temperature of 300F at the engine manifold for engine full-power operation.

Additional air extraction needed for this cooling enhancement include:

1. W25 extraction at CDP (not included in the flow requirement stated above) to drive the precooled compressor by expansion in a turbine; this air is available at the engine LP turbine for cooling.
2. W25 (included in the coolant flow requirement stated above) is employed, after cooling and compression, to cool the turbomachine air bearing.

6.5.2 CONFIGURATION

The turbocooler system consists of a turbocompressor section and a heat exchange section. The turbocompressor consists of a single-stage axial compressor and a single-stage radial inflow turbine mounted on a shaft supported by an air bearing; the bearing is cooled by reverse Pitot extraction at the air cycle compressor diffuser exit. Turbocompressors similar to these have been used in aircraft environmental control systems for decades; the principal differences between their application in environment control system (ECS) packages and their use as turbocoolers for the ATS are the temperature exposure of the turbine and the fact that the

turbine and compressor are not aerodynamically coupled in the ATS application. For this turbocooler application with precooled compressor and intercooled cycle, the compressor disc is fabricated from aluminum and the turbine disc from Inconel 718. Precooling the compressor air minimizes the air extraction required to drive the turbine, provides a source of air capable (without further cooling) of cooling the foil bearings, and allows use of conventional lightweight materials for the compressor blades and disc.

The heat exchangers are tube and shell type multipassed on the shell side. The shell side fluid is directed between passes by means of segmented baffles which also provide lateral support for the tubes. Specifically, the fuel heat sink system consists of two heat exchangers with an intermediate water loop. Heat from the air is transferred to water circulating in a closed loop; the water rejects heat to the fuel in the second heat exchanger. A motor-driven circulating pump is located at the water exit of the fuel heat exchanger, and the water circuit pressure level is maintained (assuring required energy exchanges) by means of a pressure amplifier referenced to engine CDP. All fuel wetted surfaces, in the heat exchanger as well as downstream, in the fuel delivery system are coated to retard deposition of gum and formation of coke.

6.5.3 CONCEPTUAL DESIGN

The turbocooler system boundary conditions applied to the design of the system are shown schematically in Figure 6.5.3.1. These conditions apply for the intercooled, turbocooled ATS cycle operating at max power. Fuel inlet temperature is assumed to be 77F, and the cycle intercooler water discharge temperature is assumed to be 200F.

Compressor flow requirements were established by reference to the engine and air bearing cooling requirements; the flow requirement above was increased by 15% to provide contingent margin. Compressor operating pressure ratio was determined by accounting for turbocooler system losses, extraction loss, and required supply pressure at the combustor vane/ high pressure vane manifold. The system loss determination at maximum power is summarized in Figure 6.5.3.2.

The supply pressure requirement is predicated upon providing pressure adequate for convective cooling of the vane and combustor liner. The higher supply pressure, on the other hand, increases any existing leakage from the vane outer band.

Compression heat compensation (representing 11% of the cooling load) depends on the operating pressure ratio, inlet temperature, and compression efficiency. The Allied Signal compressor has a peak adiabatic efficiency of 78%. Compressor inlet temperature is low by virtue of precooling, and the compressor pressure ratio is dictated by turbocooler system losses and required reentry value. The compressor performance map is presented in Figure 6.5.3.3. The motive flow extraction, 2.2% W25, is sufficient for pressurizing the coolant (including 15% margin in coolant flow levels) and is based on turbine expansion power available with exhaust to the low pressure vane manifold, given the turbine circuit pressure losses of Figure 6.5.3.2. The turbine adiabatic efficiency is 81%. Bearing and windage losses are accounted for in an assumed shaft efficiency of 96%. A representation of the turbine performance is provided in Figures 6.5.3.2 and 6.5.3.5. A summary of the turbocooler design conditions is shown in Figure 6.5.3.6, and a cross section of the turbocooler is presented in Figure 6.5.3.7.

The fuel sink system design represents Allied Signal's response to the GEAE Statement of Work, dated August 3, 1994. Then, as now, consensus dictates achievement of fire safety by means of complete separation of the fuel and air during heat exchange, the heat energy being transported from air to fuel via an intermediate fluid. Water was selected as the intermediary because of its chemical stability in the operating temperature interval. The Allied Signal design was further influenced by GEAE's preference for relatively modest water circuit operating pressures. The water loop operating pressure chosen was 1600 psia, subcritical, yet higher than the expected fuel system pressure, assuring a positive leakage path in the event of tube failure. Water is evaporated in the air heat exchanger and condensed in the fuel heat exchanger. Compatibility of the temperature profiles in terms of pinch margins limits the available fuel sink and necessitates supplementary cooling of the air to achieve 300F re-entry temperature. Supplementary cooling is achieved in a separate heat exchanger, in which a portion of the engine intercooler water discharge is employed as a heat sink.

The resultant Allied Signal heat exchangers were larger and heavier than expected. Direct fuel-to-air cores sized internally were on the order of 1 ft³ in volume. The two heat exchangers comprising the fuel sink indirect heat transfer system were nearly 27 ft³ in volume, significantly larger than would be expected in going from a direct transfer system to an indirect transfer system. Furthermore, the tubes carrying fuel exceeded the aspect or spacing ratio felt acceptable for coke barrier coating (CBC) application. The issues of heat exchanger size and configuration are discussed in the next section. The original and new Allied Signal heat exchanger designs are presented in Figure 6.5.3.8.

6.5.4. CHALLENGES

6.5.4.1. Hot Fuel

In the Advanced Turbine System, fuel is heated to temperatures well above those in current thermal management systems (300F). Questions of operational safety and fouling of the heat exchanger have been a concern above, and will be addressed more fully below. Fouling is also an issue which must be addressed for the lines downstream of the heat exchanger, such as combustor, the staging valves, the fuel manifold, and injectors. Combustor valve operability at high fuel temperature is a serious concern, since these valves are designed to maintain combustor operation in a narrow band between blowout and burnout.

6.5.4.2 Fire Safety

Preventing a fire in the event of failure of the heat exchanger tubes was a principal concern in the design of the heat exchangers. Separation of fuel and air is felt to provide the best assurance of fire safety. Double-walled tube designs, where leaking fuel is vented away from hot air, are complex and expensive and result in significant reductions in heat exchange capability by imposition of a low conductive vent path between the fluids, requiring large cores to meet the heat transfer requirement.

Barrier passages in plate-fin heat exchangers are somewhat more producible, the barrier pass representing just another layer in the fin stack, but the GEAE perception is that the ATS system operating pressures seriously compromise the structural integrity (and, therefore, intrinsic safety) of a plate-fin heat exchanger. Again, the barrier passage even with highly conductive fins insulates against efficient heat transfer between the two fluids. Large cores result from increased surface required to compensate for the reduced heat transfer conductance and to accommodate the intervening barrier passages between *each* active pass.

Intermediate transfer systems provide for positive separation between the potentially reactive fluids, but the intermediate fluid must be selected carefully. Certain forms of Dowtherm remain stable to 700F, but their deterioration near this temperature can produce reactants as lethal as fuel, relative to fire safety. Heat pipes operating at these temperatures employ liquid metals (potassium, sodium) which are extremely unstable when exposed to either fluid. Water was selected as the intermediary, because it has none of these problems. Subcritical systems will, however, involve phase change, and supercritical systems are characterized by very high water heat capacities, either of which limit utilization of the available fuel heat sink enabled by coke barrier coating.

6.5.4.3 Coke Barrier Coating Technology

At tube temperatures to 650F, fuel reactions occur near the tube surface, producing long chain molecules which may attach loosely to the tube walls in a random skeletal structure. At higher temperatures, these polymers tend to fragment, but metal ions in the wall diffuse to the surface and react with sulfides in the fuel to form coke, which attaches firmly to the tubewall, obstructing flow and reducing heat transfer capability of the unit. GEAE is, independently exploring means of retarding or eliminating these reactions through coating of the fuel wetted surfaces (creating a barrier to the coking reaction and a catalyst to promote fragmentation of the resin molecules). Tests have been conducted on liquid fuel systems (resistance heated single tubes and convectively heated full tube bundles with hot air flowing across the tubes) and natural gas (in heated single tubes) to determine the effectiveness of the barrier coating.

The natural gas tests were conducted in Schenectady by flowing the fuel inside coated tubes heated to 1000F. After 1,000 hrs. of exposure to these conditions, the tubes showed no coke deposition. Testing of a full tube and shell heat exchanger with liquid fuel flowing inside the tubes and hot air flowing across the tubes was conducted in Evendale last year. These units survived 150 hrs of testing without coke deposition at fuel (exit) temperatures of 850F. Increasing the fuel flow through the system resulted in full coking of all units within 50 hrs. Certain conclusions drawn from this testing (relative significance of residence time and tube temperature in resin formation and deposition; resin deposits may be shed more readily outside the tube than inside) will be the subject of further testing (Phase 2) and repetition of the Phase 1 at a later date. Optimism on the effectiveness of the use of the CBC with natural gas is well founded in the success of the single-tube tests.

6.5.4.4. Heat Exchanger Size

The heat exchangers proposed by Allied Signal in response to GEAE SOW are probably typical of configurations intended for industrial use. These units use relatively large, thick tubes (0.25 in. OD) in coarse spacing arrangements. The estimated surface density for the feedwater to air unit is $110 \text{ ft}^2/\text{ft}^3$. Surface densities in compact heat exchangers approach $300 \text{ ft}^2/\text{ft}^3$, limited by tube bundle assembly issues.

Studies done within GEAE (without full Allied Signal concurrence to date) suggest significant economies available with respect to volume and weight by using smaller tubes (0.125 in. OD) in more dense arrangements (30 mils between tube surfaces). These studies were based on flowing fuel over the tubes (rather than inside). This was suggested intuitively by the CBC test in 1994 (following the SOW issuance) and will be the subject of Phase 2 tests later. The GEAE study conclusions follow.

Disk and donut heat exchanger configurations appear to give the greatest reduction in size and weight for the current ATS design conditions. This configuration appeals intuitively to the favorable effect of symmetrical shell temperature distributions on stress.

Liquid water intermediary systems provide for the most efficient heat transfer (relative to multiphase subcritical or steam intermediary systems), but operate above critical pressure (3200 psia), requiring that water flow inside the tubes for containment. This is consistent with GEAE's current bias toward external fuel configurations. Minimum core size for the liquid water (supercritical) units is achieved by reducing the system water flow (increasing the water temperature change to approach a constant temperature difference across the tubewall). This tends to reduce the frontal area of the shell; considerations limiting contraction of the shell frontal area include limiting air velocities, (1) over the tubes to prevent vibration, and (2) in the turning paths between passes to limit pressure drop. A typical air-to-water unit is shown in Figure 6.5.3.8.

It should be further noted that, as the water loop design maximum temperature is increased beyond the critical value (702F), proportionately more of the heat transfer to the *water occurs in the supercritical realm* with relatively little change in temperature (high, variable heat capacity) tending to bias the water temperature profile into interference with the air temperature profile. This is analogous to, but not as severe as, the effect of the boiling and condensing regimes in subcritical systems on limiting available temperature variation in the outer (principal) fluids.

Power requirements for circulating the intermediate fluid where the water exists as liquid at some point in the cycle are low; a steam intermediary can probably be operated at pressure levels of 100 psia, but because of the relatively large pressure drops and low density, requires significant power for circulation of the vapor. The principal reason for rejecting low pressure intermediary systems, however, is the adverse leakage path, permitting transport of fuel into the air heat exchanger.

6.6 TURBOCOMPRESSOR

ATS Turbocooler power requirements of 1,300 HP dictate the need for larger load capacity foil bearings. The bearings dictate the design rotational speed (40,000 rpm at the max power operating condition), which results in some compromise to turbine performance.

The high load bearings proposed for this application have been recently developed by Allied Signal, so their use at this time involves some risk in the absence of performance history. Current production experience is relative to the F-22 and 737X.

6.7 FAULT DETECTION AND CONTROL

The turbocooler provides a critical function in its cooling, pressurization and maintenance of vane and burner coolant flow levels. Any circumstance (fault) interrupting the operation of and/or degrading the expected performance of the system must be detectable and motivate shutdown of the plant; simply bypassing a defective turbocooler will not provide adequate cooling for the hot parts at the rated condition. Reversion to an internal cooling system requires expensive contingent design of alternative cooling paths and probably involves flow and dilution levels which increase emissions well above acceptable levels.

Loss of cooling in the heat exchanger resulting from loss of pressure or charge in the water circuit, fouling of the tubes (including coking of the fuel tubes), or failure of the water pump compromises the structural integrity of the vane and combustor liners (coolant temperature increases and turbocooler flow output is reduced by tendency to spool down) and also threatens the turbocooler compressor.

Although similar to a standard aircraft ECS pack, an engine cooling turbocooler differs in ways that will necessitate a thorough exploration of failure modes and consequences and provision for detection and correction. The former is a bootstrap air cycle with aerodynamic coupling between compressor and turbine, which results in operation in a narrow band on the compressor map; the equipment is well matched over a wide range of operating conditions. The turbocooler, with independent streams to compressor and turbine, is susceptible to overspeed or stall under circumstances which might not affect an ECS pack as seriously.

No attempt has been made in this phase of the study to address failure detection and controllability of the turbocooler system, apart from identifying location of check valves to prevent inadvertent engine extraction via the supply manifolds in the event of a duct failure event.

The larger issues mentioned here will be addressed in the next phase of the ATS study.

DOE ATS Turbocooler Cycle Conditions

Figure 6.5.3.1. ATS Turbocooler Cycle Conditions

Figure 6.5.3.2. Estimated Pressure Distribution in ATS Turbocooler System

Figure 6.5.3.3. ATS Turbocooler Compressor Performance Map

N TURBINE ROTATIONAL SPEED, RPM
 P_{IN} TURBINE INLET AIR TOTAL PRESSURE, PSIA
 P_{OUT} TURBINE OUTLET AIR TOTAL PRESSURE, PSIA
 PR TURBINE TOTAL PRESSURE RATIO = P_{IN}/P_{OUT}
 T_{IN} TURBINE INLET AIR TOTAL TEMPERATURE, °R
 T_{OUT} TURBINE DISCHARGE AIR TOTAL TEMPERATURE, °R (DAR)
 A_N TURBINE EFFECTIVE NOZZLE AREA = 1.06 SQ. IN.
 $\Delta T_i = T_{IN} (Y/Y+1)$ WHERE $Y = PR^{0.283} - 1$

$$\eta_t = 100 \left[\frac{T_{IN} - T_{OUT}}{\Delta T_i} \right]$$

Figure 6.5.3.4. ATS Turbocooler Turbine Performance

N TURBINE ROTATIONAL SPEED, RPM
P_{IN} TURBINE INLET AIR TOTAL PRESSURE, PSIA
P_{OUT} TURBINE OUTLET AIR TOTAL PRESSURE, PSIA
PR TURBINE TOTAL PRESSURE RATIO = P_{IN}/P_{OUT}
T_{IN} TURBINE INLET AIR TOTAL TEMPERATURE, °R
W TURBINE INLET AIRFLOW RATE, LB/MIN
A_N TURBINE EFFECTIVE NOZZLE AREA = 1.06 SQ. IN.
F_F TURBINE NOZZLE FLOW FACTOR

$$F_F = \frac{W \sqrt{T_{IN}}}{(31.8)(P_{IN})(A_N)}$$

Figure 6.5.3.5. ATS Turbocooler Turbine Performance (cont'd)

Axial Compressor:

- Corrected Flow = 97 lb/min
- Pressure Ratio = 1.266
- Corrected Speed = 34500 (60%)
- Blade Tip Diameter = 6 in.
- Stall Margin = 50%

Radial Inflow Turbine:

- Corrected Flow = 0.566 lb/min
- Pressure Ratio = 3.6
- Corrected Speed = 34500 (60%)
- Blade Tip Diameter = 9.3 in.
- Tip Speed = 1625 ft/sec
- $U/Co = 0.71$

Air Bearings:

- 1300 Horse Power
- 3 inch Journal
- Operating Speed = 40000 RPM
- Operating DN = 3,200,000 mm-RPM
- High Load Bearings in limited production on F-22 & 737X

Figure 6.5.3.6. ATS Turbocooler Design Conditions Summary

Figure 6.5.3.7. Turbocooler Cross-Section

Fire Safe Heat Exchangers Schematics

Figure 6.5.3.8. ATS Turbocooler Heat Exchanger Schematic

6.8 ATS LOW PRESSURE POWER TURBINE

The ATS power turbine is a six-stage design that drives both the low-pressure compressor or booster and the power output shaft. The mechanical design is based on the GE90 low-pressure turbine philosophy. This design utilizes the GE90 type turbine mid frame and case. The power turbine cross section is shown in Figure 6.8.1.

The GE90 LPT secondary flows and temperatures were used for the preliminary estimate of ATS turbine flows and temperatures. These temperatures were used for hardware sizing and are shown in Figure 6.8.2. The hardware has been sized at the ATS operating conditions to meet the following requirements:

- Blades: 50,000 Hours, 0.2% Creep Life
- Disks: 100,000 Hours, 0.2% creep life and 25% burst margin
- Shaft and Spline: Low Cycle Fatigue Life of 50,000 Cycles at Max. Torque

Results are summarized in Tables 6.8.1, 6.8.2, and 6.8.3. Figure 6.8.3 shows the recommended materials for this turbine. Due to the high low-pressure turbine inlet temperature and 50,000 hours, 0.2% creep life requirements, Stages 1 and 2 N6 blades will require cooling to reduce the metal temperature to 1600F. Preliminary estimates indicate that a chargeable cooling air flow of approximately 7% W25 would be needed to obtain a bulk blade temperature of 1600 F for the two stages.

To eliminate power turbine blade cooling and the associated complexity, ceramic blades were also evaluated for this application. Due to high rim temperatures, no change in disk material was feasible, but due to lighter ceramic blades, a substantial reduction in disk weight (30% to 50%) is possible. Results are summarized in Tables 6.8.4 and 6.8.5.

The turbine mid frame and the rear frame have been adopted from the GE90 design. The forward power turbine bearing and sump design is similar to the GE90 design, whereas the aft sump is based on the proven LM6000 design. For thrust balance, a magnetic thrust compensator is being proposed.

ATS POWER TURBINE

Figure 6.8.1. ATS Power Turbine

ATS LPT COOLING FLOW TEMPERATURE PRELIMINARY ESTIMATE

Case #1023

T3 = 960°F

P3 = 760 psia

T48 = 1900°F

P48 = 194 psia

T_E 27 = 480°F

PE 27 = 156 psia

T48 = 1882°F

Figure 6.8.2. ATS LPT Cooling Flow - Preliminary Estimate

LPT BLADE SIZING

LIFE REQUIREMENTS — 50,000 HOURS, .2% CREEP				
STAGE	TREL (°F)	MATERIAL	σ MAX. (KSI)	REMARKS
1	1802	N6	20	Cooled * to 1600°F
2	1655	N6	22	Cooled * to 1600°F
3	1486	N4	26	
4	1300	R125	26	
5	1124	R125	38	
6	955	R125	37	

ATS LPT Blade Sizing

TABLE 6.8.1.

* NO PRIOR GE EXPERIENCE (COOLED SHROUDED BLADES) WOULD REQUIRE DEVELOPMENT OF TECHNOLOGY.

LPT DISK SIZING

LIFE REQUIREMENTS — 100,000 HOURS, .2% CREEP						
STAGE	σ (KSI) / T°E			MATERIAL	BURST MARGIN	LCF LIFE CYCLES
	POST	L. RIM	BORE			
1	21. / 1202	20.0 / 1202	146 / 664	R88DT	25%	50K
2	26. / 1243	17.0 / 1243	157 / 697	"Groh" #6 I718		50K
3	26. / 1269	25.0 / 1269	157 / 655			50K
4	29. / 1178	41.0 / 1178	111 / 676			25K
5	27. / 1030	27.0 / 1030	115 / 578			25K
6	27. / 969	8.0 / 969	111 / 621			25K

TABLE 6.8.2

ATS LPT Disk Sizing

POST KT $\approx$ 1.6, L. RIM KT $\approx$ 1.8

R88DT .2% CREEP @ 1270°F (100K HOURS) — $\sigma_{\text{LIMIT}} = 39$ KSI

"GROH" #6 I718 .2% CREEP @ 1165°F (100K HOURS) — $\sigma_{\text{LIMIT}} = 45$ KSI

TABLE 6.8.3

ATS LPT Aft Shaft and Spline Sizing

AFT SHAFT AND SPLINE SIZING

SPEED — 3000 RPM

MAX TORQUE — 315,220 FT. LB.

ADDERS: OVERSHOOT - 5%
 GROWTH - 10%
 VIBRATION - 15%

DESIGN TORQUE 418,690 FT. LB.
SHAFT LCF LIFE 50,000 CYCLES

MATERIAL: MARAGE 250 AFT SHAFT
 INCO 718 LPT SHAFT

LPT DISK SIZING (CERAMIC BLADES)

LIFE REQUIREMENTS — 100,000 HOURS, .2% CREEP							DISK Δ W (%)
STAGE	σ (KSI) / T°E			MATERIAL	BURST MARGIN	LCF LIFE CYCLES	
	POST	L. RIM	BORE				
1	9. / 1202	17.0 / 1202	138 / 664	R88DT	25%	50K	— 35
2	12. / 1243	11.0 / 1243	155/ 697	"Groh" #6 I718		50K	— 43
3	12. / 1269	21.0 / 1269	159 / 655			50K	— 43
4	13. / 1178	37.0 / 1178	117 / 676			25K	— 44
5	15. / 1030	27.0 / 1030	114 / 578			25K	— 45
6	13. / 969	5.0 / 969	114 / 621			25K	— 49

POST KT $\approx$ 1.6, L. RIM KT $\approx$ 1.8

R88DT .2% CREEP @ 1270°F (100K HOURS) — σ_{LIMIT} = 39 KSI

"GROH" #6 I718 .2% CREEP @ 1165°F (100K HOURS) — σ_{LIMIT} = 45 KSI

ATS LPT Disk Sizing - Ceramic Blades

TABLE 6.8.4

LPT BLADE SIZING

LIFE REQUIREMENTS — 50,000 HOURS, .2% CREEP				
STAGE	TREL (°F)	MATERIAL	σ MAX. (KSI)	REMARKS
1	1802	MI SiC/SiC	8.0	At 1800°F, for 50K Hours .2% Creep Life, $\sigma_{\max} \approx 13.0$ KSI
2	1655		10.0	
3	1486		10.0	
4	1300		12.0	
5	1124		14.0	
6	955		14.0	

TABLE 6.8.5

ATS LPT Blade Sizing

6.9 ENGINE SYSTEM AND PACKAGING

Using the resulting engine flowpath, as shown in Figure 6.1.1, an engine layout was completed. This engine layout and identification of major engine systems is shown in Figure 6.9.1.

The engine system shown in Figure 6.9.1 was also incorporated in an engine packaging study, and the resulting layout of the engine packing module is shown in Figure 6.9.2.

6.10 PLANT LAYOUT

The resulting engine system module was integrated in plant layout arrangements for a combined cycle system and a simple cycle system. Figures 6.10.1 and 6.10.2 show the combined cycle plant layout, and Figures 6.10.3 and 6.10.4 show the simple cycle layout.

Advanced Turbine System Industrial Engine

Figure 6.9.1. ATS Industrial Engine Cross-Section

Figure 6.9.2. ATS Packaging Layout

Figure 6.10.1. Industrial ATS Combined Cycle - Top View

Figure 6.10.2. Industrial ATS - Combined Cycle - Side View

NO	EQUIPMENT IDENTIFICATION LIST
1	GENERATOR
2	GENERATOR COOLING SYSTEM
3	OVERHEAD MATERIALS CRANE
4	STEAM TURBINE
5	GAS TURBINE
6	GAS TURBINE GENERATOR
7	EXHAUST
8	STATION AUXILIARY TRANSFORMER
9	NON-SECTORED BUS DUCT
10	NON-SECTORED BUS DUCT
11	NON-SECTORED BUS DUCT
12	NON-SECTORED BUS DUCT
13	NON-SECTORED BUS DUCT
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100	NON-SECTORED BUS DUCT

SIMPLE CYCLE---SIDE VIEW

Figure 6.10.4. Industrial ATS - Simple Cycle - Side View