Advanced Turbine Systems Program Conceptual Design and Product Development

**Annual Report
August 1993 - July 1994**

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November 1994

Work Performed Under Contract No.: DE-AC21-93MC30246

For
U.S. Department of Energy
Office of Fossil Energy
Morgantown Energy Technology Center
Morgantown, West Virginia

MASTER

By
Solar Turbines
San Diego, California

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Morgantown Energy Technology Center
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EXECUTIVE SUMMARY

This Yearly Technical Progress Report covers the period August 3, 1993 through July 31, 1994 for Phase II of the Advanced Turbine Systems (ATS) Program by Solar Turbines Incorporated under D.O.E. Contract No. DE-AC421-93MC30246. As allowed by the Contract (Part III, Section J, Attachment B) this report is also intended to fulfill the requirements for a fourth quarterly report.

The objective of Phase II of the ATS Program is to provide the conceptual design and product development plan for an ultra-high efficiency, environmentally superior and cost-competitive industrial gas turbine system to be commercialized in the year 2000.

During the period covered by this report, Solar has completed three of eight program tasks and has submitted topical reports. These three tasks included a Project Plan submission of information required by NEPA, and the selection of a Gas-Fueled Advanced Turbine System (GFATS). In the latest of the three tasks, Solar's Engineering team identified an intercooled and recuperated (ICR) gas turbine as the eventual outcome of D.O.E.'s ATS program coupled with Solar's internal New Product Introduction (NPI) program. This machine, designated "ATS50" will operate at a thermal efficiency (turbine shaft power/fuel LHV) of 50 percent, will emit less than 10 parts per million of NOx and will reduce the cost of electricity by 10 percent. It will also demonstrate levels of reliability, availability, maintainability, and durability (RAMD) equal to or better than those of today's gas turbine systems.

Current activity is concentrated in three of the remaining five tasks -- a Market Study, GFATS System Definition and Analysis, and the Design and Test of Critical Components. The Market Study has identified a strong future demand for an ATS product in electrical power generation as well as in all of the other markets served by Solar. These opportunities lie in multiple sizes (output power) within the range described by D.O.E. for an industrial class ATS.

Based on this Market Study, the GFATS System Definition Task is currently directed toward selection of those technologies gathered into the "ATS50" which are appropriate for field demonstration in ATS Phase IV. The key goal in this selection is to maximize early return on ATS investment -- both Solar's and the D.O.E.'s -- in terms of clean, efficient gas turbine horsepower installed and resulting in lower energy costs and other benefits to the U.S. economy.

Three subtasks are presently underway in the design and test of critical components. Subjects include combustion, recuperators, and an Autothermal fuel Reformer (ATR). Additional subtasks, soon to be brought into the D.O.E./Solar cofunded program, include expanded combustion and recuperator development ceramic materials, and an advanced control system.

As of July 31 1994, a total of 12,032 labors hours have been charged to the contract versus 10,836 planned. Total program cost (including Solar's cost share) stands at \$1,320K versus \$1,096 planned. the current overrun is temporary and reflects accelerated activity following a late start. An additional 5,338 labor hours have been charged to internally funded development activity which is directly ATS-related. This represents approximately \$480,000 of cost sharing in addition to that provided in the current contract.

A total of four briefings to D.O.E.'s COTR and other staff were held during the period covered by this report. They are summarized as follows:

<u>Date</u>	<u>Location</u>	<u>Subject</u>
09/22/93	D.O.E.-METC	Phase II Kickoff, Program and Management Plan
12/07/93	Solar	Discussion of additional Task 8 test work
04/20/94	Solar	Results of Task 3 - System Selection
07/28/94	D.O.E.-METC	System modifications indicated by Market Study

SECTION 1

TASK 1 - PROJECT PLAN

Task 1 was completed and a Topical Report was submitted in December 1993. The plan submitted was structured to follow the contractual Statement of Work (SOW). This format was implemented in order to simplify the presentation of the plan and to show the interaction of the several tasks. Each of these tasks was discussed at length in terms of the work to be performed and the criteria used to define task completion. Cost and Labor Plans were submitted with Task 1 Topical Report as well as the schedule of Phase II work as shown below in Figure 1. Solar's performance to schedule and budget will be discussed in Section 9 of this report.

The Task 1 Topical Report does not include any discussion of additional Task 8 critical component design and test work. These add-ons were suggested by D.O.E. in December of 1993 and a proposal submitted by Solar in January of 1994. Funding for this additional proposal was announced by D.O.E. in July of 1994 and the necessary reviews, modifications, and approval of the January proposal were initiated. Once final signature of the contract modification is obtained, revised labor and cost plans will be prepared and the additional activity added to the schedule in Figure 1.

Figure 1. ATS Phase II Schedule

SECTION 2

TASK 2 - INFORMATION REQUIRED FOR NATIONAL ENVIRONMENTAL POLICY ACT (NEPA)

The Topical Report for Task 2 was submitted in November 1993. This Report covered the original contract scope of work. Solar submitted its proposal for add-on subtasks under Task 8 in January, 1994. Information required for NEPA pertaining to these add-on subtasks was provided to D.O.E. by letters in January and February, 1994 as requested. D.O.E. informed Solar in February that it was authorized to perform Task 8 work (original scope of work), pursuant to Task 2 and NEPA.

Solar has been informed that funding for Task 8 add-ons Subtasks 8.4 through 8.9 has been approved but has not yet received formal approval of the add-ons. Solar will provide a final Task 2 Topical Report, incorporating information regarding the add-ons within 60 days after receiving formal approval.

SECTION 3

TASK 3 - SELECTION OF GFATS CYCLE

Task 3 was completed in April 1994 and the final revision to the Topical Report submitted in July. This task identified an intercooled and recuperated (ICR) gas turbine as the eventual Solar product to result from Solar's internal New Product Introduction (NPI) program combined with D.O.E.'s ATS program. When fired in the region of 2500°F at the optimized pressure ratio, this machine will be capable of operating at 50 percent thermal efficiency (turbine shaft power/fuel LHV). Task 3 activity was centered around determining the sensitivity of cycle efficiency to component performance, cooling air consumption and other parameters. Based upon these studies, a power plant designated "ATS50" evolved which will be consistent with materials and other technologies which can be captured by an industrial, stationary gas turbine system typical of Solar's past and current offerings.

One open issue remained at the writing of the Task 3 Topical Report, that of a one-spool versus a two-spool gas producer section. Additional studies of this problem were made at the beginning of Task 6 - System Definition and Analysis.

3.1 OVERVIEW

The ICR gas turbine system chosen in Phase I can be configured in various ways. The first configuration examined was an intercooled, one-spool gas turbine with a free power turbine (Figure 2). Such an arrangement is particularly attractive for base-load systems when first cost, as a function of the overall life-cycle cost, is very important. A second approach is to use two spools with a free power turbine and to intercool between the two spools (Figure 3). The latter approach provides high efficiency over a wider range of off-design operating conditions than the one-spool configuration. In addition to these two configurations, which were selected as baselines, three other configurations were evaluated. These three were variations of the two-spool baseline configuration, intended to maximize the energy extraction in the first-stage turbine. One of these configurations places the power or free turbine as the first stage; another would locate the power turbine and the low pressure compressor on the same shaft; the third would use a flow split arrangement. This last approach would divide the flow after compression, with one stream going to the power turbine and the other to the gas producer turbines. Two recuperators and two combustors would be necessary to produce an ICR configuration with such a flow split.

All of these approaches were investigated in terms of their ability to meet the ATS50 program goals and also Solar's extended goals (Table 1). The performance of these five configurations was evaluated at both on-design and off-design conditions. The on-design ambient conditions adopted for the analyses were those associated with a sea-level, 59°F day as defined by the International Standards Organization (ISO). Hot day conditions were defined as 100°F, again at sea level. An existing computer code that provided a simplified analytical method was used initially to determine the relative merits of the two general approaches. This analytical methodology involved the use of a generalized compressor map and operating line to determine potential compressor surge problems.

While the above analyses were being performed, a more sophisticated performance code was under development. This Solarj-developed code incorporates both compressor and turbine maps with a cooling model for the turbine section. The cooling model is considered to be extremely important because the air flow required for cooling purposes will largely determine the overall thermal

Figure 2. One-Spool Gas Producer With Free Turbine

Figure 3. Conventional Two-Spool Gas Producer With Free Turbine

Table 1. Advanced Turbine System Goals

Parameter	Bseline ATS50	Expanded Goals
Thermal efficiency	Fifty (50) percent	Sixty (60) percent
Exhaust Emissions Nox	Eight (8) parts per million	Five (5) parts per million
Exhaust Emissions CO & UHC	Fifteen (15) parts per million	Twelve (12) parts per million
Cost of Power (COP)	Ten (10 percent Reduction From Today	Ten (10) percent Reduction From Today
Reliability, Availability, Maintainability & Durability (RAMD)	Equal To or Better Than Today	Equal To or Better Than Today

efficiency. This cooling model determines the approximate air flow for turbine blades and vanes given the hot and cold gas temperatures and the materials of construction together with the desired life at a predetermined stress level. The size and shape of the blades and vanes also affects the air flow requirements. These were approximated for the performance analyses based on a combination of theory and past experience. Blade sizes were determined at one set of conditions and then scaled as a function of the chosen cycle pressure ratio (i.e., higher pressures, smaller blades). The stress rupture life initially selected for the blades was fixed in the model at 60,000 hours. In addition, the blades were assumed to be fabricated from directionally solidified materials such as DS CM247LC. These material assumptions also affect the cooling air flow levels through stress- or oxidation-imposed temperature limits. The nozzle guide vanes and ducting were assumed to be fabricated from ceramic materials and were either not cooled or, in the case of the vanes, were minimally cooled.

A more advanced cooling model was also developed, using Solar funds, to replace that initially used. This model invoked more sophisticated cooling concepts and techniques that allowed the cooling air requirements to be reduced. Single-crystal (CMSX-4) blades coated with thermal barrier materials (TBCs) were assumed for the model. As above, the vanes and ducting were assumed to be manufactured from ceramic materials. Variable life and in particular reduced life for blade design (one overhaul cycle [30,000 hours]) was also included in the analysis. A trade-off between reduced life for the turbine section and the resulting increased maintenance costs, can be considered with the ATS50 because its high thermal efficiency provides significantly reduced fuel costs over the life of the system. However, the frequency of engine change-outs and associated downtime should be taken into account. This will be investigated during the Task 5 Market Study and Task 6 System Definition and Analysis.

3.2 BACKGROUND

In the ATS Phase II Program, Solar has adopted a two-tier set of goals for its ATS product. Four Phase III goals in the following listing were adopted for the near-term ATS50 (year 2000):

- 1) 50 percent thermal efficiency (ATS50),
- 2) Eight parts-per-million by volume (ppmv) NOx,

- 3) 10 percent reduction in the cost of power (COP), and
- 4) Reliability, Availability, Maintainability and Durability (RAMD) meeting (at least) today's standards.

For the far-term, beyond 2005, the efficiency and emissions goals selected were 60 percent thermal efficiency and five ppmv NOx. The same COP and RAMD as targeted for the first-generation ATS50 were retained. The initial size chosen for the ATS analyses was a 12-MWe or 16,000-hp engine. This size fits well with the commercial industrial power generation marketplace. Further study as to the appropriate size for the dispersed power marketplace is continuing as part of the Phase II Task 5 Market Study. The engine efficiencies given in this report are those at the engine design point (full power) at sea-level and a 59°F day (ISO day), unless otherwise stated. The design point efficiencies quoted are sufficiently conservative that they should be close to the "guaranteed efficiencies", taking into account the estimated parasitic losses

The word "risk", where used in this document, is associated with technical risk unless otherwise stated. Risk, as used, refers to the potential inability to reach one or more of the goals described above, if the component or process in question fails to meet its performance criteria. Incremental benefits will be involved (technical or market related) even if the component or process fails to meet the desired performance level.

Although the concept of RAMD as a primary driver for customer acceptance is valid, the issues associated with RAMD are not discussed in detail in this report. RAMD is primarily a function of the engine component and system designs. RAMD is considered in tradeoff studies from the start of the program, however, and are included in this document. The incorporation of features specifically intended to improve RAMD will be addressed in Task 6 System Definition and Analysis. Assessments of the likely overall RAMD for the ATS50 will also be provided as part of the output of Task 6.

A total of five candidate configurations of an ICR were evaluated in Phase II, Task 3 Selection of GFATS, for their potential to meet the ATS50 goals. The enhanced goals were addressed by adding reheat, ultra-low cooling requirement turbine blades, and chemical recuperation in the form of autothermal reforming to the near-term configuration. Steam cooling of the turbine blades will be considered in site-specific market studies. However, the availability and cost of water may prohibit its use in some locations.

The five ICR configurations investigated are detailed in the sections below. The two engine configurations selected for further development from the five examined, include:

- 1) a one-spool gas producer section and a free power turbine and
- 2) a two-spool gas producer section and a free power turbine.

Advantages associated with the one-spool approach include the ease with which 50 percent or greater thermal efficiency at the design point conditions can be achieved. In addition, the simplicity of the one-spool approach translates to low first-cost and potentially low maintenance costs. The ease with which the single-spool ICR can be started is an issue and must be further evaluated. Although the task of achieving an efficiency greater than 50 percent was judged to be more difficult with the two-spool gas producer section, this approach offers considerably better part-load efficiencies. Such part-load efficiencies are particularly important in load-following applications, and will be investigated further.

Because of the market implications of the differing efficiencies for the one- and two-spool approaches, Solar intends to continue to investigate both configurations. The results of the Task 5 Market Study will allow Solar to make a selection that will maximize the commercial applications of the ATS. This decision point will occur toward the end of Task 6 System Definition and Analysis.

Both the one-spool and two-spool engines have the same pressure ratio and turbine rotor inlet temperature and thus will have some common components. These common components allow the pursuit of two different configurations with minimal additional effort.

The technology options in five major areas within the ICR system are being evaluated. These areas include:

- 1) Combustion System
- 2) Gas Producer Turbine Blades
- 3) Gas Producer Turbine Nozzle.
- 4) Recuperator
- 5) Intercooled Compressor System

The risk associated with the application of the Primary Technology Choice, the Back-up Technologies and the High-payoff Technologies varies within each area. Generally, however, the Back-up Technologies diminish the risk and the High Pay-off Technologies increase the risk when compared with the Primary Technical Choices. The technologies for each are listed in Table 2.

Table 2. Component Technology Selection Options

System/Component	Primary Technical Choice	Back-up Technologies*		High Pay-off Technologies**	
		First	Second	First	Second
Combustion	Catalytic	Lean Premixed		Autothermal Reformation	
Gas Producer Turbine Blades	Film-cooled Leading Edge	All Film Cooled		Advanced Uncooled Ceramics	Cooled Metal w/TBCs
Gas Producer Turbine Nozzles	Uncooled Ceramic w/TBC	Cooled Metal w/TBCs	Cooled Ceramic		

* Descending Technical Risk

** Ascending Technical Risk

Risk will be managed in the ATS50 by evaluating, testing and introducing products to the marketplace with incremental technical advancements. Field experience, combined with additional laboratory development, will help assure ATS50 program goals can be met within the required time.

The completion of development of the high pay-off technologies will probably not all come together until after the contractual demonstration system has been delivered. Therefore, Solar's ATS program demonstration system is likely to consist of choices identified in Table 3.

Table 3. Solar ATS Demonstrator Technologies

System/Component	Selection
Combustion	Catalytic
GP Turbine Blades	Advanced Single Crystal with TBC
GP Turbine Nozzles	Advanced Superalloy (1st) / Uncooled Ceramic Stages (2nd)
Primary Surface Recuperator Core Material	Type 347SS
Intercooler	2 Circuit with Water Tower

In the event that demonstrator technologies are not available in the time frame required for the field evaluation systems, the back-up technologies from Table 2 would be selected.

3.3 ICR CONFIGURATIONS EVALUATED

A one-spool ICR with a free power turbine, and a two-spool arrangement, also with a free power turbine, were the two primary configurations evaluated (Figures 2 and 3). In both of these configurations the air flow required for cooling the turbine section plays a dominant role in determining the overall efficiency. In general, the lower the turbine cooling air flow, the higher the cycle efficiency. Because of this need to minimize the turbine section cooling air flows, the ideal configuration would have the smallest possible number of turbine stages, and the greatest temperature drop on the first turbine stage. A large temperature drop over the first-stage turbine translates to lower operating temperatures for the later turbine stages and thus a lower total cooling flow requirement. In order to increase the temperature drop over the first-stage turbine, the level of work extracted has to be increased.

In the two-spool system each of the two separate compressors is driven by a dedicated turbine section. Each of these turbine sections has a minimum of one row or stage. Two stages may be used in order to maintain the desired high turbine efficiencies. With the work split among four turbine stages the temperature drop over any one stage is low. This observation led to a number of variants of the two-spool configuration, all of which were intended to maximize the work extraction in the first turbine stage. The variant that initially appeared to present the greatest potential for increased work extraction was one in which the power turbine was positioned as the first turbine stage (Figure 4). By definition, 50 percent of the energy in the hot gas stream would be extracted by the power (or free) turbine when the overall thermal efficiency of the engine is 50 percent. Thus this turbine section potentially could provide the greatest temperature drop and, if positioned as the first turbine stage, could reduce the total cooling flow requirements. However, when this arrangement was investigated, it was found that the poor speed match between the second turbine section and the high-pressure compressor would force the use of a gearbox between the two sections. Variable nozzle guide vanes would also be needed in front of both the power turbine and the second gas

Figure 4. Two-Spool Gas Producer With Power Extraction From the High Pressure Turbine

producer turbine stages. This added complexity and the cooling needs of the variable nozzle guide vanes would reduce the efficiency to a level approximately the same as that for the conventional two-spool configuration. Because the additional complexity would add cost and higher maintenance requirements with no significant efficiency advantage, this "power turbine first" approach was abandoned. A similar two-spool arrangement where the low-pressure turbine drove both the low-pressure compressor and the power shaft was investigated (Figure 5). This configuration was abandoned for similar reasons.

Other arrangements were also evaluated including split systems where two combustors and two recuperators could be used to optimize the temperatures at each turbine section. One of these variants is depicted in Figure 6 for illustrative purposes. This system did not provide any significant improvements over the more traditional arrangements, primarily due to the high temperatures imposed on the power turbine. The cooling air flow required for the power turbine exceeded that needed for the baseline configuration turbines. Thus the two primary configurations that were initially considered and used as baselines were the only ones judged likely to succeed in meeting all of the ATS50 program goals.

3.4 ICR COMPONENTS

All of the ICR systems that have been evaluated, including both the one- and two-spool arrangements, have essentially the same components. Each component is described below, in the order that it is encountered by the air passing through the engine. Additional information concerning materials of construction and material selection logic is presented in Section 4.

Figure 5. Two-Spool Gas Generator With Power Extraction From the High Pressure Turbine

Figure 6. Two-Spool Split Flow Configuration

3.4.1 Low Pressure Compressor

The compressor segment in all configurations is divided into two sections, a low-pressure section and a high-pressure section. In the one-spool arrangement both sections are located on the same shaft, while in the two-spool configuration they are driven separately. The low-pressure (LP) compressor for both the one- and two-spool configurations is an axial-flow design, generally with three to five stages (approximately 3:1-to-3.7:1) for a total engine pressure ratio between 12:1 and 16:1. Variable inlet guide vanes and variable second-stage vanes are typically used in this low-pressure section to allow both starting and low-power operation to be accomplished efficiently. These vane angles can be adjusted to match the required flow vectors at the lower rotational speeds associated with part-power conditions and starting. The compressor operating conditions are moved away from surge through the use of such variable geometry. Advanced "controlled diffusion airfoil" design techniques will be used in developing the blading for these compressors. This will provide the very high polytropic efficiencies needed by the ATS to meet the overall engine performance goals. Long chord blade designs will be used and these will minimize the number of blades required and provide a more robust compressor system.

In the one-spool arrangement, the exit of this low-pressure section will have a valve to allow air to be bled from the system into the engine exhaust for starting and possibly part-power conditions. Variable geometry can help, but by itself will be insufficient to prevent surge without air bleed for all conditions. Based upon preliminary analysis, the blades, vanes, disks and casings of the one-spool compressor section will most likely be constructed of standard materials currently used in industrial gas turbines (IGT).

The two-spool has few problems associated with the operation of the low-pressure compressor. Generally both the low- and high-pressure spools stabilize at some speed that is near optimum for the turbine and compressor combination.

Current industrial gas turbine materials of construction for the two-spool LP compressor will most likely be ferrous alloys for the blades, disks, vanes and casings. There is little corrosion potential at the inlet to the low-pressure compressor; however, at the inlet to the high-pressure compressor, the air is likely to be saturated with water. The variable inlet guide vanes and the first blade rows of the high-pressure compressor will be coated with materials that resist both corrosion and erosion.

3.4.2 Intercooler

The air leaving the low-pressure compressor flows into the intercooler through a transition duct that will be designed to have the lowest possible pressure loss. Turning vanes may be used in this duct to reduce the pressure losses. The intercooler cools the air leaving the LPC before entering the HPC, and lowers the temperature of the air leaving the HPC. The lower HPC outlet air temperature allows more energy to be extracted from the exhaust stream per unit heat transfer surface area. A considerable increase in thermal efficiency is realized when high-pressure-ratio cycles use intercooling in combination with recuperation.

There are two principal heat exchange configurations for an intercooler. These are generally referred to as direct heat exchange and indirect heat exchange. There is a considerable difference between the two types both in terms of performance and in the size of the equipment. The direct-transfer type is a simple heat exchanger in which the two fluids are separated by the heat transfer surface. Generally this type of heat exchanger when used as an intercooler is an air-to-air unit. The indirect-transfer system is simply two direct-transfer units coupled together by a pumped heat transfer medium. Intercoolers on gas turbines are usually of the indirect-transfer type and use liquid coolants

such as water/glycol mixtures as the heat transfer medium. These coolants remove the energy from the compressed air in the first heat exchanger and reject the heat absorbed to the ambient atmosphere or to some other heat sink (water or air) through the second heat exchanger. The energy rejected to the ambient can be recovered in a bottoming cycle, if desired. Solar has selected the indirect-type heat exchanger because of its ability to reduce the size of the two individual heat exchange devices and the possible use of a large variety of heat sinks. This ability allows site-specific application of the ICR system.

In some hot and humid environments, water may condense within the intercooler and form small droplets in the air stream as the air exits the intercooler. The intercooler will be designed to minimize this occurrence, however, conditions may exist at some locations where water droplets are present in the air stream leaving the intercooler.

3.4.3 High-Pressure Compressor

Following the intercooler is the high pressure compressor (HPC). This is likely to be constructed as an axial system (see discussion below for comparison to a radial design system). For an overall engine pressure ratio of between 12:1 and 16:1, the axial HPC section usually has four-to-six stages. The number of stages depends upon the design pressure ratio as well as the efficiency desired for the compression process. On the one-spool arrangement, the high-pressure compressor may have variable inlet guide vanes (VIGVs) to aid in achieving high part-load efficiencies and also an air bleed system at the exit. At the outlet of the high pressure compressor is a transition section that allows the air flowing in the compressor exit annulus to transfer into one or two cylindrical ducts, which convey the air to the recuperator.

Because the air leaving the axial high-pressure compressor has to be turned (through the transition duct) and fed to the recuperator in either one or two circular-sectioned ducts, there will be pressure losses. To minimize the pressure losses, the high-pressure compressor may be constructed as a radial- or centrifugal-type of compressor, rather than the axial type. Although the axial-type compressor generally offers higher compression efficiency, the air-turning process required with an axial compressor can induce high losses that are not found with a radial compressor configuration. In the radial compressor, the air is turned as part of the compression process. The normally lower efficiency of the radial system may actually turn out to be higher than the equivalent axial system when the axial turning losses are included. The decision on whether to use a radial high pressure compressor will be made during Task 6 System Definition and Analysis.

The materials used in the construction of the high-pressure compressor will be similar to those used in the low-pressure compressor, i.e., standard IGT materials. In both the one- and two-spool versions, the first stages of the HP compressor may be fabricated from nickel-based alloys to minimize erosion and corrosion from water and condensates that can be carried into the compressor from the intercooler.

3.4.4 Recuperator

The air leaving the high-pressure compressor is ducted to a recuperator where it is heated before it enters the combustor. The recuperator is a heat exchanger that is used to transfer waste thermal energy from the engine exhaust gases to the high pressure inlet air. There are many types of heat exchangers that can be employed as gas turbine recuperators, including shell-and-tube, brazed plate-fin and primary surface recuperator (PSR). From the various types of heat exchangers, Solar has selected a compact surface, all welded, primary surface type of recuperator that offers

significant advantages both in small size and high effectiveness when compared to the other systems. The PSR offers the combined advantages of high effectiveness, low weight and volume, low maintenance and high reliability. Another beneficial feature of the PSR is its noise reduction characteristics. Experience has shown that it can replace standard exhaust silencers.

The recuperator to be used in the Solar ATS50 has a calculated effectiveness of 90 percent. This effectiveness value is based upon the measured performance of over 50 units operating in the field. This effectiveness can be improved by adding more air cells, provided that the additional size is not detrimental. Increasing the effectiveness results in increased size and cost of the recuperator.

The pressure losses associated with the recuperator are modest; with a two percent loss on the air side and a four percent loss on the hot gas side. These values do not include the losses from the compressor to the recuperator nor the losses encountered between the power turbine and the recuperator inlet. The pressure losses of the recuperator can be decreased; however, the size of the unit increases in proportion to the decrease in pressure loss.

The materials of construction for the recuperator are covered in Section 4.

3.4.5 Low Emissions Combustor

A catalytic combustion system has been chosen as the best approach for the ATS50. The low-temperature catalytic oxidation of fuels is considered to be the best technique available to meet the ATS50 goal of eight ppmv NO_x. This approach, much like most low emission systems, relies on premixing of the air and fuel before it enters the combustor (Figure 7).

Figure 7. Catalytic Combustor Design for ATS

Although the combustor inlet conditions affect NOx production in complex ways, it is the probability of autoignition and flashback during premixing that is most influenced by the operating conditions. Therefore, premixing considerations tend to favor selecting lower pressures and temperatures at the combustor inlet. Higher combustion system inlet pressures and temperatures increase developmental risk, because complete fuel/air mixing must be accomplished more quickly to avoid autoignition and flashback.

The work performed to date by Solar on subscale models (at pressures up to 14 atmospheres) has demonstrated that catalytic combustion can provide exceptionally low NOx levels. The system is expected to operate at the higher pressures of the ATS without any significant problems. These tests have indicated that the oxidation catalysts have to be tailored to the particular fuel and specific operating conditions. Once the optimum catalyst has been identified for the ATS environment (methane), Solar is confident that emission levels of NOx lower than the ATS50 program goal of eight ppmv can be achieved.

A typical catalytic oxidation reactor is fabricated using a ceramic or metallic honeycomb monolith having a large number of parallel, square-sectioned passages. These passages or channels are coated with a platinum-based material that catalyzes the reaction of the fuel and air. The catalysis process essentially involves the acceleration of the intrinsic reaction rate between the fuel and oxygen. This allows operation at fuel-air ratios that are below the lean extinction limit for conventional combustion systems. One major benefit of the catalytic system is that it uses no water as part of the process.

As a means of risk reduction, an ultra-lean-premixed combustion system will be developed for ATS in parallel with the catalytic system. The ultra-lean-premixed approach will extend the technology used in low-emissions gas turbine combustors just now entering the marketplace. Several of these systems, including Solar's SoLoNOx™ combustion systems, have shown NOx emissions capabilities below 10 ppmv at discrete operating points. Optimization of the fuel/air premixing process, reduction of CO formation near the combustor liner wall through the use of ceramic combustors, and better combustion control through variable combustor geometry, will all improve the low emissions capabilities of the lean-premixed approach. Technical challenges include:

- 1) premixing without autoignition at high combustor inlet temperatures,
- 2) lean combustion without unacceptable combustor pressure fluctuations, and
- 3) achieving sufficiently low pattern factors with reduced amounts of dilution air.

All of these areas will be addressed in the planned Phase III development work to ensure the availability of a low-emissions combustion system for the ATS.

Another approach, called autothermal reformation (ATR), involves the use of chemical recuperation to reform a steam/fuel mixture. In this way, about 16 percent of the energy in the engine exhaust, is converted into fuel (chemical) energy. Chemical recuperation will be employed along with future efficiency enhancements as they become technically viable and acceptable in the marketplace.

3.4.6 Turbine Section

The design of all turbine stages has been optimized for aerodynamic performance, while maintaining mechanical integrity. Both configurations reflect work and flow coefficients which produce optimum efficiency, while accounting for inter-turbine duct losses. The performance of the turbine stages

includes parabolic work and loss distributions and three-dimensional aerodynamics. The final design of the turbines will reflect a concurrent engineering effort among aerodynamics, heat transfer, mechanical design, and manufacturing disciplines, with the goal being optimum cycle efficiency at industrial turbine system life requirements.

3.4.6.1 Gas Producer Turbine Section

The gas producer turbine section consist of one or more turbine stages which drive the compression section of the engine. The gas producer is a single-stage, high-pressure turbine (HPT) and single-stage, low-pressure turbine (LPT) in the case of the two-spool configuration, and a single-stage HPT for the one-spool configuration. A turbine stage is made up of stationary and rotating hardware. The stationary hardware includes nozzle guide vanes and gas-path walls. The blades, disk, and related seals comprise the rotating hardware of a turbine stage.

All gas producer vanes and stationary gas-path hardware will be uncooled ceramic material. By uncooled it is meant that the vanes will not be internally cooled, however, compressor bleed flow will be available for the purpose of cooling ceramic-to-metal interfaces and to reduce thermally-induced stresses related to the gas-path temperature profile. Internally-cooled ceramic and film-cooled metal vanes are considered lower risk back-ups to the uncooled ceramic.

At this point, the gas producer turbine blades will be cooled, directionally solidified single crystal. Technologies developed in the CSGT program will be phased into the ATS50 leading to cooled ceramic blades. However, in an effort to control risk and satisfy market introduction timing, Solar has elected to begin with cooled metallic blades.

A dual-property disk will be used as the mounting platform for the gas producer turbine blades. The dual-property disk, which consists of a superalloy outer rim hot isostatically pressed (HIP) bonded to a more conventional material inner disk, allows for operating at rim temperatures of approximately 1500°F, while maintaining life requirements. The disk will still require cooling; however, it will be at a reduced level. The higher allowable rim temperature is necessary due to the flat radial profile of the catalytic combustion system.

3.4.6.2 Power Turbine Section

The power turbine consists of three stages on a free shaft. The first stage has an uncooled, ceramic variable vane and a cooled, directionally solidified single crystal blade. The variable vane allows for maximizing system efficiency at part-load and non-standard day ambient conditions. Stages two and three are uncooled metallic vanes and blades. The power turbine of the one- and two-spool configurations differ only by the level of cooling required in the first stage.

3.5 CONFIGURATION AND CYCLE SELECTION

This section of this document defines the process of selecting the engine configuration and cycle for the ATS program. In this section, the engine configuration is the number and arrangement (mechanical linkage and flow path) of components. The cycle is the combination of design parameters such as overall compressor pressure ratio (PR), turbine rotor inlet temperature (TRIT), etc., for a given configuration. This configuration and cycle selection process is designed to identify the configuration and cycle which most closely meets the stated program goals for the ATS 50 (Table 3).

Though several different engine configurations were investigated, the only configurations which looked promising from an overall efficiency, operability, and cost standpoint were the one-spool and two-spool gas producer, intercooled, and recuperated (ICR) engines. Reheat combustors and autothermal reforming were not considered at this stage of the analysis, as these can be added to any of the ATS 50 engines to obtain the ATS 60 engine.

The first step in determining which cycles most closely met the program goals for these one-spool and two-spool configurations was to perform cycle optimization studies. In these studies TRIT, PR, and compressor work split were varied to determine which combinations gave the best overall efficiency. The compressor work split is the combination of LPC and HPC pressure ratio giving the overall compressor pressure ratio. Figures 8 and 9 show plots of efficiency vs. PR at lines of constant TRIT for both the one-spool and two-spool configurations of the engines.

In addition to the basic design point optimization, sensitivities of some of the cycles to changes in component efficiency, cooling, pressure losses, etc., were determined. Table 4 shows the results of these sensitivities.

Figure 8. Overall Thermal Efficiency vs. PR at Lines of Constant TRIT - Two-Spool Configuration. The optimum efficiency is shown to be between Rc of 18:1 to 20:1, with a TRIT of 2500° F.

Figure 9. Overall Thermal Efficiency vs. PR at Lines of Constant TRIT One-Spool Configuration. The optimum efficiency is shown to be at an Rc of 16:1, with a TRIT of 2500° F.

3.6 CRITICAL COMPONENTS AND TECHNOLOGIES

There are four components that are considered to be technologically significant to the ATS50;

- 1) the high temperature GP turbine blades,
- 2) the recuperator,
- 3) the intercooler, and
- 4) the low emissions combustor.

Each of the first three components utilizes heat transfer technologies of one form or another, and each is critical to meeting the 50 percent thermal efficiency goal. The catalytic combustion technology selected has neither been demonstrated in a full-size system nor at the full system pressure. These are important issues and will be resolved through experimental testing.

3.6.1 Cooled Turbine Blades

Extensive consideration was given to the method of cooling the ATS50 gas producer turbine and power turbine first-stage blades. Due to the high TRIT, the bulk of the overall engine cooling air flow was required to maintain the blades at acceptable temperature levels. It was also essential that the amount of cooling air utilized for the cooled blades was consistent with meeting the thermal performance goals of the ATS50.

Table 4. Sensitivity of Overall Efficiency to Changes in Component Efficiency, Pressure Losses, Etc.

Sensitivity to Changes in Component Efficiencies, Pressure Losses, Etc.		Changes in Overall Thermal Efficiency					
Cycle Parameter	Change	Two-Spool Cycles (PR/TRIT) (%)			One-Spool Cycles (PR/TRIT) (%)		
		20:1/2500	16:1/2500	12:1/2450	20:1/2550	16:1/2500	12:1/2500
LPC Efficiency	+0.5	+0.12	+0.10	+0.90	+0.11	0.11	0.09
	-0.5	-0.12	-0.10	-0.09	-0.12	-0.11	-0.09
HPC Efficiency	+0.5	+0.14	+0.13	+0.12	+0.14	+0.13	+0.12
	-0.5	-0.14	-0.13	-0.12	-0.14	-0.14	-0.12
Intercooler Effectiveness	+1.0	+0.08	+0.06	+0.05	+0.08	+0.07	+0.05
	-1.0	-0.08	-0.06	-0.05	-0.08	-0.07	-0.05
Recuperator Effectiveness	+1.0	+0.17	+0.20	+0.24	+0.18	+0.22	+0.27
	-1.0	-0.17	-0.20	-0.24	-0.18	-0.22	-0.27
Combustor Efficiency	+0.5	+0.24	+0.24	+0.27	+0.27	+0.24	+0.24
	-0.5	-0.24	-0.24	+0.27	+0.27	-0.24	-0.24
HPT Efficiency	+0.5	+0.06	+0.05	+0.05	+0.11	+0.11	+0.10
	-0.5	-0.06	-0.05	-0.05	-0.11	-0.11	-0.10
LPT Efficiency	+0.5	+0.04	+0.04	+0.03	N/A	N/A	N/A
	-0.5	-0.05	-0.04	-0.03	N/A	N/A	N/A
Power Turbine Efficiency	+0.5	+0.17	+0.17	+0.17	+0.17	+0.17	+0.17
	-0.5	-0.17	-0.17	-0.17	-0.17	-0.17	-0.17
Typical Pressure Loss	+0.5	+0.07	+0.08	+0.09	+0.07	+0.08	+0.09
	-0.5	-0.07	-0.08	-0.09	-0.07	-0.08	-0.09
Reductions in Total Cooling Flow	-10.00	+0.54	+0.59	+0.65	+0.60	+0.54	+0.61
	-20.0	+1.07	+1.18	+1.30	+1.20	+1.08	+1.22

Two different cooling arrangements were considered. The first of these was ceramic blades and the second was metallic single crystal alloy blades. The ceramic configuration resulted in the lowest cooling flow budget, however, there is a significantly higher degree of risk associated with using ceramics in rotating components. This design approach will be carried as a back-up until ceramic technology, being developed at Solar (under the Ceramic Stationary Gas Turbine D.O.E. program) and elsewhere, can be confidently applied to high temperature rotating parts. On the basis of potentially higher durability and lower risk, the proposed ATS50 gas producer and power turbine first stage will consist of metallic blades.

The GP turbine blades have a combined film, impingement, and convection cooling circuit typical of that used in aircraft jet engines. The most significant feature of the GP blade cooling system is the

"shower head" film cooling holes around the leading edge. This method of cooling was required due to the catalytic combustor flat profile and the high relative gas temperature.

Based upon the results of the forthcoming heat transfer tests at Solar, the blade cooling configurations will be optimized in an effort to reduce cooling flows and increase thermal efficiency. The current blade cooling designs have sufficient flexibility built into them, that major redesigns are not likely.

3.6.2 Recuperator

The recuperator selected for the ATS50 is the all primary surface heat exchange system that Solar has produced for several years. The ATS50 primary surface recuperator (PSR) will use appropriate materials required to meet system specifications. As temperatures and pressures increase, material properties must be increased. This may mean the use of some alloys with higher temperature capability than the standard Type 347 Stainless Steel, such as Inconel 625, or combinations of various materials.

3.6.3 Intercooler

The use of an intercooler positioned between two conventional compressors provides near isothermal operation and substantially reduces the energy requirements for compression. The intercooler will be of the indirect type that employs a water/glycol mixture as a heat exchange medium. The heat exchanger located between the two compressors, is linked by a water/glycol loop to a second heat exchanger. This second heat exchanger transfers the heat in the water mixture to the ambient air or to some other heat sink such as a body of water. This combination of two heat exchangers that ultimately transfers heat from the low-pressure compressor discharge to ambient usually has effectiveness limits typically between 0.7 and 0.85. While, at first glance, the intercooler effectiveness may be critical to maximum cycle efficiency, the return is highly dependent upon the cycle pressure ratio. For our studies, a one percent change in intercooler effectiveness translates to an overall cycle efficiency change of 0.075 percent. The overall effect of a 70-to-85 percent intercooler effectiveness change would increase the cycle efficiency by only 1.125 percent.

Reducing the heat rejection sink temperature can improve the intercooler effectiveness. If the sink temperature can be lowered below ambient the effectiveness can be increased. A cooling tower using evaporative cooling can reduce the water temperature to within five degrees of the wet bulb temperature. At ISO conditions the dry bulb temperature is 59°F and the wet bulb temperature is 52°F. At these conditions, the cooling tower can maintain the water temperature at or below the ambient dry bulb temperature. However, approximately 40 gallons-per-minute of make-up water would be needed for an ATS cooling tower.

3.6.4 Catalytic Combustion System

The catalytic combustion system is the preferred method of meeting the ATS50 NOx goals. Because water injection is not required, lower pattern factors can be achieved, and the likelihood of stable combustion over the entire system operating range is high. Risk with catalytic combustion is associated mostly with the integration of this technology into the ATS, and the life of the catalytic reactor at ATS conditions. Despite these concerns, the catalytic combustion system offers the lowest NOx levels (as low as two parts-per-million by volume under laboratory conditions) of any alternative. The planned experimental investigations into the behavior of the catalytic system at pressures typical of ATS50 operation (16:1) will provide design guidelines to address these

technology and integration issues. In parallel, work will be performed with the ultra-lean-premixed approach to ensure that there is an alternative path to commercialization.

3.6.5 Other Systems

Although a reheat combustor is not considered to be a critical component for the ATS50 it will be crucial for the future generation ATS products as they become acceptable in the marketplace. Similarly the potential use of an autothermal reformer to chemically recuperate the ATS and boost efficiency levels will also be critical in attaining efficiencies greater than 50 percent. The component issues with these two add-on systems are being addressed experimentally in Task 8. The reheat combustor requires the development of new high-temperature materials of construction before it becomes practical for operation in an ATS environment. These new materials will be used to construct uncooled combustor walls that will have to operate at temperatures between 2800-and-3000°F. If these material developments are successful, then the use of a reheat combustor as part of the ATS50 will be reevaluated at the end of Phase II.

One of the critical material requirements of the autothermal reformer, is the development of a sulfur-tolerant catalyst. Sulfur is known to poison all the catalysts presently in industrial use for the commercial steam/methane reformation processes. Sulfur-tolerant catalysts will have to be developed because there is no control over sulfur content in natural gas. Removal of the sulfur compounds raises a solid waste disposal issue and thus is preferably avoided. Sulfur tolerance will provide the life needed by the autothermal reformer to be considered suitable for integration with the ATS50. The ongoing experimental evaluation of reformer catalysts (Task 8 Design and Test of Critical Components) is addressing the sulfur tolerance problem and is demonstrating excellent results. Combustor integration and operation problems are also foreseen due to the high-hydrogen-content secondary fuel that is produced. Safety concerns with hydrogen containment and flashback need to be solved. Once the material issues have been resolved, then the problems of operation and integration will be addressed.

SECTION 4

TASK 4 - CONVERSION TO A COAL-FUELED ADVANCED TURBINE SYSTEM (CFATS)

Originally scheduled to begin approximately two months after the start of Task 3, the actual start of detailed Task 4 work requires that somewhat accurate definition of combustor inlet conditions be provided. Such definition was not available during the process of optimizing the ATS50 power plan until near the end of Task 3. As the power plant definition transitioned from Task 3 to Task 6, the Market Study of Task 5 brought new input as to the probable characteristics of a marketable ATS cycle. Since the intent of later phases of D.O.E.'s ATS program is to build and demonstrate gas fueled power plants, there is no downstream impact of delayed performance of Task 4.

The goal of Task 4 is to study the conversion of the Gas-Fueled Advanced Turbine System (GFATS) to a Coal-Fueled Advanced Turbine System (CFATS). This conversion capability would allow the choice of natural gas or coal, based upon availability, cost, or other considerations. As stated in the proposal, the approaches to this conversion range all the way from the utilization of coal gas to direct firing of coal either as a dry powder or as a slurry with water or oil.

A direct-fired CFATS will be based upon previous work done on a D.O.E.-METC contract to develop a coal-fired version of the Solar Centaur 50 generator set. This contract funded development of a 1/10th scale system for this engine and the scale-up of that system to a full sized system intended for use on the engine. Rig tests of the full size system were conducted at near engine operating conditions and indicated the technical feasibility of continuing with a planned engine demonstration. Price differences between coal and natural gas indicated that further activity was not justified at this time.

SECTION 5.0

TASK 5 - MARKET STUDY

The Market Study for the ATS program was formally launched in February 1994. Two separate firms whose background lent themselves to the task, were called upon to deliver independent market studies, analysis evaluations and recommendations to Solar.

5.1 APPROACH

Solar chose the format of the study to be as follows.

- **Define the Gas Fired Advanced Turbine System (GFATS)**

Solar defined the proposed technical, environmental, and economic characteristics so that it could be taken to the marketplace to test the proposed system size and cycle against the market requirements. The initial concept was a 12 MW, intercooled/recuperated cycle (ICR) at 50 percent thermal efficiency.

- **Define the Systems Applications**

Solar studied the potential markets that the ATS system could participate in, and then identified the applications in those markets that, when screened technically, would be candidates for the Advanced Turbine system proposed, or as we determined through field research an ATS size and cycle that optimized commercialization.

The key applications recognized are as follows.

- In the Power Generation industries Solar sees the traditional applications of peaking, intermediate and baseload, and new applications of distributed generation and decentralized reserve. In the Gas & Oil Pipeline and Storage industries, the traditional applications of gas gathering, transmission, and oil field water/steam injection for enhanced recovery are foreseen. Expanded applications of increased regional storage as requirements of natural gas usage increases. In the markets defined as "Other Power User Industries" the traditional applications of mechanical drive and cogeneration are included. New/expanded applications will be in areas of dedicated baseload and/or load following.

- **Characterize Competing Systems**

Recognizing that there is more than one way to meet market needs, Solar requested that the market researchers construct both a technical and economic profile of power generation systems which would compete with the ATS system in the technically suitable market applications. We endeavored to make sure that the technical, economic, institutional, and markets analyses would be such that all competing systems would be viewed on the same basis for each market application.

- 3) competitive installed system costs in \$/kW, and
- 4) lowest unit production cost as calculated as Levelized Annual Cost (LAC).

The marketplace survey indicated a forecast of use as follows: For applications by the power generation industry, and in particular by the investor-owned utilities (I.O.U.), a smaller system will see placement at or near the load center. As load growth in North America is relatively low when compared with the rest of the world, load shifting will direct the present utilities to augment their generation without the expense of transmission and distribution (T&D) upgrades. As well, the current I.O.U. thinking is such that they will not want to repower central power plants to increase generation capacity and the ATS offers an attractive, cost effective opportunity while keeping a satisfied customer base. For application by electric power generators, and those who require mechanical drive, the larger industrial ATS product has shown significant opportunity. Cogeneration and "inside the fence/onsite" electricity generation by small, medium, as well as large manufacturing entities holds great promise.

The market research indicated a reduced emphasis on the importance of ultra-high efficiency, i.e., 45 percent or greater. The market has expressed a reluctance to accept "leapfrog" technology, and prefers a lower efficiency (approximately 45 percent thermal efficiency) over the promise of high efficiency coupled with complex cycles or extremes of pressure ratio and firing temperature. The following figure (Figure 10) illustrates buying criteria as described by the marketplace.

Solar believes that over time, the marketplace will begin to accept technology advancement. Solar is committed to advancing the technology as market conditions permit.

Market data have indicated a higher buying criteria for "permitability" of the product than efficiency. Lower emissions by the ATS product will ease the permitting process, improve commissioning times, and overall, reduce the cost of applying the product to its intended use. Pollution prevention through dry low NOx technology is vastly preferred over pollution abatement products such as selective catalytic reduction. The marketplace perceives that pollution abatement products and their associated high cost, negatively drive the economics of a project.

Figure 10. Key Buying Criteria - Domestic Industrial Power Generation

SECTION 6

TASK 6 - SYSTEM DEFINITION AND ANALYSIS

As planned in Task 1, Task 6 will complete the preliminary design of the GFATS selected in Task 3. This includes a physical definition of a core ICR gas turbine as well as its integration into a complete ATS generator set. During the execution of Task 6, the ATS50 concept will be modified as required by:

- Closer definition of market requirements as generated by the Market Study of Task 5 and
- Any program constraints imposed by D.O.E.'s Solicitation for Cooperative Agreement Proposal (SCAP) for ATS Phases III and IV when it arrives during the course of Task 6.

At the close of Task 6, the Topical Report will include definition of both the ATS50 and, if different, the ATS offered in response to the Phase III-IV SCAP.

6.1 CYCLE WORK

6.1.1 Start and Idle Analysis for One-spool Configuration

A study was performed to determine if the one-spool version of the ATS engine can be started with a reasonable power input requirement and to determine idle conditions.

The study was made by setting up an off-design model of the ATS one-spool cycle selected, using the compressor and GP turbine characteristics previously developed. The flow/pressure ratio characteristics for the power turbine (at low pressure ratios) were taken from an existing power turbine.

This model was then used to establish the no-load operating point, which is approximately the same as idle, and which is very close to the self-sustaining point. For this idle point, the lowest speed at which the recuperator exhaust inlet temperature was not over the limit (estimated to be about 1300° F) and the LP compressor was out of surge was determined. This point included surge bleed required to maintain at least seven percent surge margin throughout a start (fixed area bleed valve), LPC variable geometry (all stages) fully closed, and power turbine variable geometry fully open.

Once the idle point was established, points were run which modeled the engine being driven by a starting motor (no combustor fuel flow), at spool speeds below the idle point. These points were used to determine the power required to start the engine. Although thorough transient modeling is still not available at this time, steady state points were run simulating a starter motor providing power to the engine from 30 percent speed up to the idle speed. The points were run with as much fuel flow to the burner as possible without surging the compressors, and without violating the recuperator inlet temperature limits.

The engine can be started and run at idle, if the description of the engine components used in the study (compressor and turbine efficiency-airflow-pressure ratio-speed characteristics, compressor surge lines, etc.) are correct. However, accurate predictions of performance at the lower speeds and

pressure ratios involved are very difficult and often subject to error. Before actual rig tests of the components, it is difficult to know whether the critical parameters will change in a way to help or hurt the ability to start the engine and run it at idle.

The idle point for the engine was shown to be at 67 percent LPC speed, with 9.1 lbm/sec bleed flow to avoid surge, LPC variable geometry fully closed, power turbine variable geometry fully open. The starting power required to get the engine to the idle condition is about 200 hp. This is not significantly different from cranking power required by current gas turbines of projected ATS50 size.

The parameters which are most critical to starting and running the engine at idle are the power turbine flow capacity at low speed, fully open variable geometry nozzles, and the LPC surge line at low speed, fully closed inlet guide vanes and stators.

The effect of changes in these parameters can be shown by maintaining the same idle speed, and looking at the changes in recuperator exhaust side inlet temperature. For example, if the LPC surge line is higher than predicted, then less surge bleed is required to keep the compressor out of surge, and the recuperator exhaust inlet temperature is reduced. Conversely, if the line is lower, than more bleed is required to keep the compressor out of surge, which raises the exhaust inlet temperature.

The same type of effect is seen in changes to the power turbine flow capacity at low speed and low pressure ratio. If the flow capacity is less than predicted, then the exhaust temperature goes up. If it is more than predicted, then the exhaust temperature goes down.

Table 5 shows the magnitude of these effects.

Table 5. Sensitivity of Recuperator Exhaust Inlet Temperature to Power Turbine and Compressor Parameters

Parameter	Percent Change In Parameter	Change In Recuperator Exhaust Inlet Temp
Power turbine flow capacity	-10%	74° F over temp.
	-5%	35° F over temp.
	+5%	32° F under temp.
	+10%	54° F under temp.
Surge bleed required	-20%	107° F under temp.
	-10%	56° F under temp.
	+10%	57° F over temp.
	+20%	113° F over temp.

The risk from changes in power turbine airflow characteristics are substantial. The increased recuperator temperatures could be handled by increasing the idle speed (it takes an increase in LPC percent reference speed of approximately 3 percent to make up 75° F in overtemperature), and/or the power turbine pressure ratio at idle (which could result in an overspeed of the power turbine at no load). Likewise, a change in the surge line in the low speed area could require an increase in bleed, with the same required increases in speed and/or power turbine pressure ratio at idle. As mentioned before, the extent of these problems will not be fully known until complete descriptions of the power turbine flow-pressure ratio characteristics and LPC surge lines are determined by rig testing of these components.

The risks of changes in the characteristics of the LPC surge line and power turbine flow-pressure ratio were also analyzed at starting points (points at speeds below idle with a starter motor providing power to the HP spool. No table is included, however, as changes in the surge margin (changing the surge bleed flow required) are already shown in Table 5 (above) for the idle point, and the very low speed starting points turned out to be somewhat less sensitive to the turbine flow capacity changes.

6.1.2 Comparison of one-spool and two-spool Engines

Due to the potential rotodynamic difficulties involved in concentric shafts of small diameters, the HPC was redesigned to allow passage for a large diameter GPT shaft. Although this did not necessarily mean a change in HPC design speed, the HPC design speed was changed to be the same as the LPC. This involved both compressor and turbine aero design work. The reason for this change was to allow for the possibility of a two-spool design that could be converted to a one-spool machine as the compromise in efficiency of the original design is small.

Configuration and final design point parameters for the two concepts are shown in Table 6.

Table 6. Final One-spool and Two-spool Concepts

	One-spool	Two-spool
Configuration	Intercooled compressor, single stage GP, 3 stage PT	Intercooled compressor, 2 stage GP, 3 stage PT
Spool Speeds - rpm	LP/HP: 17210 PT: 10800	LP: 14280 HP: 14280 PT: 10800
TRIT - °F	2500	2400
Pressure Ratio	16:1	16:1
Thermal Efficiency - %	50.01	49.99

The information shown in Table 7 was used to review the practical implications of each of the two concepts.

The information in Table 7 shows that the two-spool has many advantages over the one-spool design. However, until the compressor is in development test, it will not be possible to evaluate the starting requirement. Review of the primary advantages of the two-spool cycle over the one-spool shows that they are due to its two-stage GPT. Further, because the two-spool machine's aerothermal design allows it to be a one-spool equivalent at the design point, it follows that a one-spool machine, with a two-stage GPT, would be the best choice in a one-spool machine. Initially the primary reason for pursuit of a one-spool concept was to reduce total hot section cooling through the use of only one stage to drive the compressors. It turns out that when the global effect is considered however, a two-stage GP is the better choice.

It therefore follows that both the one-spool and the two-spool engines will have thermodynamically identical flowpaths and airfoils at the design point.

Table 7. Design Considerations for one-spool vs two-spool

	Two-spool		One-spool	
Design Point Efficiency - %	49.99		50.01	
Optimum TRIT - °F	2400		2500	
Recup. Inlet Temp, °F	1101 (100 khrs life in 0.005 in SS 347 foil) ~985 @ 2200° F TRIT		1219 (Must be IN 625 or better material) ~1030 @ 2200° F TRIT	
Efficiency Sensitivity to TRIT	TRIT °F	Efficiency Change, pts	TRIT Change	Efficiency Change pts.
	2500	+0.2	2400	-0.5
	2300	-0.9	2300	-1.6
	2200	-2.4	2200	-3.1
Variable Nozzle Temp. °F	1763(Possible in uncooled CMSX-4)		1917 (Cooled metal, or ceramic)	
Compressor Geometry Comments	IGV only		IGV (Possibly "flapper" type) plus 4 stages in LPC	
Combustor and Hot Section Comments	2400° F firing temp. would allow margin for dilution air to be used to shape exit profile.		No margin for shaping profile.	
Derivatives	Recup. derivative engine would have better performance than one-spool due to more lightly loaded GPT stages.			
Mechanical Design Challenges	Rotordynamics more difficult.		"Flapper" IGV high temperature variable nozzle.	

The ICR engine cross-section is shown in Figure 11.

6.1.3 Recuperated Cycle Work

During July, studies of recuperated cycles were undertaken based on the component and cycle characteristics, material capabilities and other factors that were identified for the ATS50 power plant defined in Task 3. Recuperated cycles which incorporate ATS technologies are currently viewed as commercial predecessors to an ATS50. Early indications are that such power plants are capable of demonstrating thermal efficiency well in excess of 40 percent while invoking a reasonable level of technical and commercial risk.

Figure 11. ICR Engine Cross Section

6.2 COMPONENT WORK

6.2.1 Engine Compressors

The ATS50 engine compressors were adjusted to allow the HPC and LPC to rotate at the same speed at the design point. The HP compressor flowpath was also adjusted at the same time to allow for passage of a LP shaft large enough to run sub-critical at the design speed. Design parameters for the finalized version are shown in Table 8.

Table 8. Final ICR Engine Compressor Design Parameters

	LP Compressor	HP Compressor
Rotational Speed - rpm	14,280	14,280
Corrected Airflow - lbm/sec	63.44	21.209
Pressure Ratio	3.24:1	5.10:1
Adiabatic Efficiency - %	90.4	89.57

6.2.2 Intercooler

An intercooler extracts heat from the air exiting the low pressure compressor, thereby lowering the temperature of the air and increasing its density. This significantly reduces the work required to further compress the air in the HPC.

Intercooling can be accomplished in at least three conventional methods. These are known as direct air-to-air intercooling, direct water-to-air and indirect liquid-coupled heat exchange. In an air-to-air intercooler, a heat exchanger uses ambient air as a heat exchange fluid to directly cool the partially compressed air from the LPC. The relatively low density of air from the LPC and the low density of the ambient air require additional heat transfer surface area. The resultant heat exchangers are relatively large and pressure losses can be significant. However, a direct heat exchange system can be used to get high heat transfer effectiveness.

A direct water-to-air system would reduce the size of the heat exchanger significantly over the direct air-to-air. This sort of system, however, requires a large body of water upon which to draw the coolant. Therefore, local regulations and availability of such a water source put this system into question as a universal solution.

A liquid-coupled, indirect heat exchanger has certain advantages. The use of water as a primary coolant reduces the size of the primary or hot-side heat exchanger. However, a second, cold-side heat exchanger is required to remove the extracted heat from the water coolant so that it can be recirculated. Therefore, a fan is required to remove that heat in the cold-side heat exchanger. Using this approach, it is difficult to get the high effectivenesses of the direct approaches. However, a cooling tower can be used, instead of a fan, to successfully attain the 85 to 90 percent effectivenesses.

Cooling towers were examined to determine relative size, cost and operating data. Two vendors were contacted. Both vendors came to similar conclusions, in that cooling towers are quite efficient

for removing heat from a hot water stream. In an intercooler using water as a coolant, this heat must be removed before the water is recirculated. A cooling tower is used to evaporate a small portion of the water stream by inducing or forcing a draft of air. The evaporative cooling effect makes it possible to lower the temperature of the water to within five degrees of the so-called wet-bulb temperature. The wet-bulb temperature is less than the ambient by an amount that depends on the humidity content of the air. The less humid the air, the lower the wet-bulb temperature. In any event, the temperature of the water coolant can be lowered significantly and this effect can result in a fairly high intercooler effectiveness.

At ISO conditions, where the temperature is 59°F and the relative humidity is 60 percent, the wet-bulb temperature is 52°F. Therefore, an approach to the wet-bulb of 5-to-10 degrees will effectively lower the water temperature to approximately ambient.

Cooling towers can generally be characterized as requiring some significant amount of space and consuming some power, but are of relatively minor cost. Cooling tower size and cost data were obtained from two vendors and are shown in Tables 9 and 10.

Table 9. Cooling Tower Cost and Operating Data, Vendor 1¹

Factor	8°F approach to wet-bulb	5°F approach to wet-bulb
Cost - \$	17,000-to-24,000	23,000-to-27,000
Fan horsepower	32	80
Height - ft	13	10
Footprint, LxW, ft	22x12	18x16

¹Two towers are required for the 5°F approach. Data provided from Baltimore Aircoil Company, Inc., Baltimore, MD.

Table 10. Cooling Tower Cost and Operating Data, Vendor 2¹

Factor	Value
Cost - \$	28,321
Approach to wet-bulb temperature - °F	8
Nominal cell dimensions, LxW, ft	12 x 18
Overall tower height, ft	16-ft 11.5-in
Makeup water - gpm	25.7
Fan horsepower	36.48

¹Tower Tech, Inc, Chickasha, OK

Since cooling towers employ evaporative cooling, water losses are incurred. The water evaporation rate is equivalent to about 20 gpm for both cases above. It is also necessary that some water is bled from the unit at a rate equal to the evaporation rate to maintain a low impurity content. Therefore, make-up requirements are about 40 gpm for the units described above.

It becomes obvious that a number of devices can be used in the cold-side exchanger. For instance, closed circuit liquid-to-air radiators could be used in applications where the cooling tower's requirement for make-up water might be objectionable, such as in arid environments. Each device will have its own characteristic cost, space claim and parasitic power requirement and its own measure of support for ATS thermal efficiency goals. Both commercial and technical input will be applied to the requisite tradeoff studies.

A cursory examination of absorption chilling was performed. The concept here was to use exhaust waste heat from the engine to drive a refrigeration cycle. In this scenario, the device would be employed to chill the water from the intercooler to sub-ambient temperatures, thereby markedly increasing the effectiveness of the intercooler. Such a device, however, would unduly complicate the system by adding considerable hardware including a significantly larger cooling tower than would otherwise be used, a steam generator, as well as the absorption chilling hardware. The Trane Corporation was contacted to do a rough estimate of size and cost. Approximately 1300 lbs of refrigeration would be required, with a chiller size of 25-ft long, 8-ft high, weighing 61,000 lbs and would cost \$350,000. While an absorption chiller might provide slightly higher intercooler performance, it does so at a much increased cost, size and system complexity. It was not investigated further.

Under certain operating conditions, water condensation will occur at the intercooler exit. Increasing effectiveness and higher humidity conditions result in more condensation. The deleterious effects include impingement and possible damage to the first-stage HPC. After entering the HPC, the compression results in a rapid temperature increase that would vaporize any moisture past the first or second stage. Blade coating might be an acceptable solution. Otherwise, the water would need to be removed with little pressure drop, or the intercooler would have to be operated at a lower effectiveness.

Mist eliminators were examined as possible droplet coalescers. Two companies, ACS Industries, Inc. of Houston, Texas and Koch Engineering Company of Wichita, Kansas were contacted for further information. Lower pressure drop devices are available without problem but are fairly large and do not integrate cleanly or easily with a gas turbine, as they are fairly large devices. For example, a 5-8 ft. diameter mist eliminator vessel would provide a near complete water drop removal at a 0.2-in H_2O pressure loss. Smaller coalescers would result in a higher pressure drop.

An intercooler needs to be compact and efficient in order to accomplish the desired ends of neatly fitting into an overall package and minimizing pressure drop. Various compact heat exchange surfaces, described in the literature, were examined as potential heat exchange surfaces. An early design using surface 11.32-737-SR, shown in Figure 12, resulted in a heat exchanger of roughly 10-sq. ft. of frontal area, and only 1.5 -ft in length. The flattened tubes would carry water as a heat transfer fluid. Such a device would deliver 85 percent effectiveness.

6.2.3 Turbines

Both the one-spool and two-spool core engine configurations from the initial cycle studies have been studied. The two-spool was the original proposal cycle. The one-spool was introduced to improve cycle efficiency by eliminating a gas producer turbine stage, reducing the cooling flow required. Both

Figure 12. Compact Heat Exchanger Surface 11.32-737-SR

cycles were studied even after initial cycle selection chose the one-spool engine, because of concerns over reduced off-design performance of the one-spool engine caused by the presence of the intercooler.

As the program progressed, a decision was made to use Solar's new state-of-the art compressor known as "ACE" on both the one-spool and the two-spool engines. The two-spool engine has always had the first few stages of the ACE compressor as the LPC. However, the speed requirements of the 1-spool had excluded the use of any portion of the ACE compressor on that engine.

To accommodate the use of the ACE compressor on the one-spool, a new gas producer turbine was designed at the lower ACE speed requirements. At the lower ACE speeds, the new gas producer turbine became so large that the cycle efficiency suffered substantially due to the increased disk purge and cooling requirements on this larger disk. This single-stage turbine already had a lower efficiency than a two-stage turbine at this same design point, due to its high stage work. This lower efficiency, combined with the increase in cooling for the large disk resulted in lower overall cycle efficiency with the single-stage turbine, when compared with a two-stage turbine. Therefore, it was decided that a two-stage gas producer turbine would be used on the one-spool engine.

By going to a two-stage gas producer turbine, the one-spool and the two-spool engines became thermodynamically identical at the design point. Therefore, design point efficiency was no longer a discriminator between the two cycles, and other issues became important in trying to decide whether to build a one-spool or two-spool version of the engine. The two-spool was considered to be mechanically more difficult because of the hollow disk on the first stage of the turbine, and the concentric shaft arrangement. The one-spool had questions as to the off-design performance, and whether it could be started.

The end result was to carry both the one-spool and two-spool configurations. By making the design speeds on each of the two-spool gas producer spools the same as the one-spool design speed, the turbine systems for the two engines became aerodynamically identical. Therefore, a single design could be applied to either engine.

The progression of the two configurations is presented in Table 11 which includes physical speed, turbine size, cooling flow, and cycle efficiency, illustrates how constraints on speed drove the configuration. As mentioned previously, the constraints on speed were mechanical as well as programmatic.

Table 11. Turbine Configurations

		N rpm	Hub Radius Inches	Pr :1	Efficiency %	% Cooling	% Cycle Efficiency
Two-Spool Original	HP	19500	8.12	1.83	87.7	18.69	49.4
	LP	14938	8.12	1.5	88.0		
	PT	10080	9.78	5.08	89.6		
Two-Spool Both Spools at One-Spool Speed	HP	14600	9.04	1.89	86.0	18.69	49.2
	LP	14600	9.04	1.49	91.2		
	PT	10080	9.8	4.97	90.5		
One-Spool Original	GP	17210	9.8	2.75	85.7	13.93	49.5
	PT	10080	9.8	5.02	89.4		
One-Spool ACE Compressor	GP	14980	11.4	2.97	83.4	19.38	48.1
	PT	10080	9.8	4.66	89.4		
One-Spool Two-Stage GP	Stage 1	14600	9.04	1.89	86.0	18.69	49.2
	Stage 2	14600	9.04	1.49	91.2		
	PT	10080	9.8	4.97	90.5		

The turbine is a two-stage GP with an inter-turbine duct and a free, three-stage power turbine. The GP has a cylindrical tip for better clearance control. The first stage has a cylindrical tip and the second is conical inward. The inter-turbine duct has been set at the optimum length to minimize loss. The power turbine consists of a conical tip at an angle of 23° and a cylindrical hub. At this point no work has been done on the diffuser. However, the recovery was assumed to be 0.5 for all performance predictions.

The GP consists of a moderately-loaded first stage and a lightly-loaded second. The work split of the turbine is set by the optimum split for the compressors in the two-spool version of the machine. This is necessary to keep the two configurations aerodynamically the same.

In the case of the two-spool engine, changing the work split to optimize for the turbine has a detrimental effect on cycle performance. The performance gain due to the intercooler is very dependent on the location of the intercooler relative to the work split of the compressors. Therefore, in the two-spool version of the engine, the turbine work split is set by the compressor work split.

The work split for the one-spool turbine is independent of the compressor split. Therefore, it can be optimized. However, initial studies show only marginal gains from this optimization.

Table 12 contains the pertinent performance information for the final configuration. The performance numbers quoted in Table 12, reflect current technology. The performance will be upgraded for year 2000 technology at a later date.

Table 12. Final Turbine Configuration

		N rpm	W/T/P	Pr :1	Efficiency	$\Delta H/U^2$	Cz/U
GP	Stage 1	14280	14.94	1.93	0.873	1.44	0.35
	Stage 2	14280	29.34	1.48	0.915	0.86	0.44
PT	Stage 1	10080	43.66	1.52	0.906	1.22	0.41
	Stage 2	10080	64.85	1.66	0.917	1.23	0.43
	Stage 3	10080	101.96	1.76	0.927	1.13	0.51
				GP (t-t) PT (t-ds)	0.895 0.906		

It should be pointed out that, because of concerns about the cooling delivery system, the cooling flow temperatures changed between Table 11 and Table 12. The change in cooling assumptions produced the decrease in cycle efficiency of 0.5 points.

The first stage of the GP turbine suffered in efficiency because of the mismatch in work split and the speed requirement. Close coupling of the first and second stage of the GP required the first stage to have a relatively high work coefficient and the second to be relatively low. The work coefficient on the first stage of 1.44 is high but under the recommended 1.5. A high work coefficient indicates increased turning losses. The flow coefficient is low at 0.35, which will reduce losses related to through flow velocities. The resulting first stage efficiency is 87.3 percent. The performance of the first stage is based on an nonshrouded blade at a minimum clearance of 0.02 inches.

The second stage of the GP benefited from the work split and speed requirements. The work coefficient of the second stage is 0.87 which indicates low turning losses. The lower limit of the work coefficient is set by the turning at the hub of the blade. When the turning is very low, it is difficult to obtain a converging channel. At 0.44, the flow coefficient of the second stage is higher than that of the first stage. This value, however, is still low. The uncooled total-to-total efficiency of the second stage is 91.5 percent based on all non-shrouded blade at a running clearance of 0.02 inches.

The inter-turbine duct has been designed for optimum performance. The area ratio (AR) of the duct is 1.36. For optimum performance at this area ratio, the nondimensional diffuser length (length divided by the difference in radius at the inlet to the duct ($L/\Delta R$)) - is 1.5, establishing the length of this duct. The pressure loss for this duct is estimated to be 1.1 percent.

The speed of the power turbine was set to satisfy several constraints. These include:

- 1) matching the engine airflow
- 2) doing the work required of this turbine.
- 3) limiting root stress of the limiting blade (related to annulus area times speed squared)
- 4) maximizing diffuser performance (related to exit Mach number, lower being better)
- 5) minimizing the inter-turbine duct loss
- 6) maintaining reasonable work coefficients.

The third blade is the mechanically limiting component of the power turbine. The area of the third blade is set for diffuser performance. Minimizing the inter-turbine duct loss and having reasonable work coefficients set the radial location of the turbine.

The work split of the three-stage power turbine was set at 31, 34, and 35 percent. The logic was to do more work in the later, more efficient stages. The efficiency of the power turbine is 90.6 percent, total to diffuser static. The performance of the power turbine is based on shrouded blades at a minimum clearance of 0.02 inches and a diffuser recovery of 0.5. The individual stage performance is provided in Table 12.

The non-rotating components of the turbines feature ceramic and metallic hardware. The GP turbine will have ceramic vanes, endwalls, and tip shoes. There are low levels of cooling flow available to these ceramics (see Figure 13 for the purpose of reducing thermal gradient stresses and to cool ceramic to metal interfaces. The first vane of the power turbine is also ceramic, and is variable. The inter-turbine duct will be either ceramic or metallic with a thermal barrier coating (TBC). Vanes two and three of the power turbine and their related stationary hardware will be metallic.

The rotating components will be metallic. The GP will feature cooled single crystal directionally solidified blades. The cooling of the blades will be a combination of impingement and film including shower head film at the leading edge. The first blade of the power turbine will also be a cooled blade and will use the same super alloy as the GP. The remaining two blades of the power turbine will be uncooled and will be made of a less expensive alloy. The last two blades and possibly the first blade of the power turbine will be shrouded airfoils.

A dual-property disk will be used as the mounting platform for the gas producer turbine blades. The dual-property disk, which consists of a superalloy outer rim HIP bonded to a more conventional material disk, allows for operating at rim temperatures of approximately 1500° F while maintaining life requirements. The disk will still require cooling. However, the cooling will be at a reduced level. The higher allowable rim temperature is necessary due to the flat radial profile of the catalytic combustion system.

6.3 MECHANICAL DESIGN

Efforts have been focused on the assessment of turbine hot section rotating component designs for the two-spool ICR ATS. These efforts have been worked in parallel with the cycle evaluation to aid and guide the aerodynamicists as to the cycle impacts on the mechanical design. Mechanical concerns addressed include; blade root bulk stress for cooled and uncooled options, and disk stresses both radial and tangential for conventionally bladed disks, each with and without central bores. Evaluations have been made using typical material physical properties (density, modulus, etc) representing the cooled metal blade approach. Work has also been performed to identify the dual-property disk design requirements. Efforts to select appropriate rolling element "antifriction" bearings has begun. Additionally, a set of compressor diffusers and an exhaust diffuser have been sized, each exhibiting acceptable performance.

Each of the turbine blade rows have been modeled for bulk root stress and show good potential for meeting a required service life of 30,000 hours. One exception is the first-stage HP blade, which requires a 25 degree reduction in bulk temperature or a three ksi reduction in stress, to meet the 30,000 hour TBO goal. Each of the preliminary turbine disks exhibit peak bore and average tangential stress levels within the range of current and developing (dual property) disk material technology. Preliminary bearing selection indicates that a substantial reduction in net thrust load is

Configuration	111	2433.83 ft	18:01 Wg	04:59
Two stage GP	Cooling after recuperator			

[illegible]

required if rolling element bearings are to be considered. For the current estimation of thrust load, fluid film tilt pad bearings should exhibit acceptable service life.

6.3.1 Blade and Disk Preliminary Design Tool

To evaluate the myriad of turbine design options, a generic preliminary design tool has been compiled. The system has been developed to address blade P/A bulk stresses for varied airfoil configurations, and to apply these loads as disk rim pulls for evaluation of uniform thickness disk stress equations. The disk stress results from the closed form solutions provide a bound of the upper limit of radial and tangential stress. True production disk designs are rigorously contoured which can show reductions in peak stresses as much as 30 percent compared to the uniform thickness solution.

The preliminary blade root stress calculations utilize much more than generic AN^2 calculations with blade material density effects included. The generic solution utilizes several aero design parameters extracted from the aero meanline analysis. These parameters, mean blade chord, true chord, blade height, blade count, and rotor speed are married with Solar data on appropriate blade taper ratios for both chord and thickness, and platform/attachment practices to produce blade centrifugal loads and mechanical stresses. The results from the blade stress evaluation are then applied to the preliminary disk analysis. The disk evaluations unite the blade loading information with the desired disk configuration options which results in local radial, local tangential, and area averaged tangential stresses. The peak stress locations can then be recontoured (departing from the uniform thickness assumption) resulting in more robust disk configurations.

6.3.2 Two Spool ICR Evaluation

The engine mechanical analysis evaluated blade and disk stresses for many HP and LP and power turbines for both two-spool gas producer and single-spool gas producer configurations. Cycle trade-off studies identified the two-spool ICR as the overall best configuration. This configuration allows for the HP and LP rotors to operate at the same rotational speed at design point. This feature would allow for a single-spool gas producer design to evolve using the same aerodynamics, when single-spool issues of starting and surge margin are resolved. The details of the final turbine design are presented in Figure 13. The content of the final data sheet had evolved through many iterations involving the cycle deck, turbine design, cooling design, and the mechanical evaluation.

Basic assumptions included:

- Nozzles are to be minimally cooled ceramic
- Blades are to be cooled metal where necessary
- Blade material to exhibit minimum properties of CMSX-4 SC
- Cooling air supply to be mix of recup. exit and PCD
- Airfoil taper ratio 0.7 for film cooled option
- Airfoil taper ratio 0.5 for impingement/conv. design

- 0.7 taper ratio last stage shrouded PT blade
- Airfoil life $\geq 30,000$ hrs.

The basic assumptions listed above allowed the turbine aero designer to evaluate preliminary designs. The initial flowpath designs were evaluated to estimate bulk airfoil stress levels due to centrifugal loads. The stress level was then used to determine the allowable bulk metal temperature for 30,000 hrs of stress rupture life. Table 13 similarly shows allowable temperatures for given stress levels resulting in 30,000 hrs of life.

Table 13. Stress and Allowable Temperature, 30,000 Hours Minimum Stress Rupture Life

Stress (ksi)	Allowable Temperature (° F)	Stress (ksi)	Allowable Temperature (° F)	Stress (ksi)	Allowable Temperature (° F)
10	1930	30	1590	50	1486
11	1905	31	1585	51	1482
12	1876	32	1578	52	1478
13	1843	33	1572	53	1474
14	1811	34	1565	54	1470
15	1783	35	1559	55	1466
16	1758	36	1553	56	1463
17	1737	37	1548	57	1459
18	1717	38	1542	58	1456
19	1701	39	1537	59	1451
20	1688	40	1532	60	1449
21	1675	41	1527	61	1445
22	1664	42	1522	62	1442
23	1652	43	1516	63	1438
24	1644	44	1512	64	1435
25	1632	45	1507	65	1432
26	1623	46	1503	66	1429
27	1615	47	1498	67	1425
28	1607	48	1494	68	1422
29	1599	49	1490	69	1420

The cooling model was then adjusted to accommodate the necessary blade metal temperatures. The updated cooling flows and turbine component designs were fed back into the cycle deck. Iterations continued until relative convergence was reached.

6.3.2.1 Blade Stress

The turbine designs were fed into the mechanical preliminary design tool described above. The speed and radius trade-off was worked between the mechanical and aero designers to arrive at good aerodynamic designs which have sound mechanical capability. The final iteration blade stress results are summarized in Table 14.

Table 14. Turbine Blade Root Stresses, Temperatures, Stress Rupture Life

Turbine Stage	Root Stress (ksi)	Airfoil Allowable Bulk Metal Temperature (°F)	Airfoil Bulk Metal Temperature (°F)	Minimum S.R. Life (hours)	Typical S.R. Life (hours)
HPT	32.7	1574	1597	16,200	62,100
LPT	37.4	1545	1549	27,500	97,200
PT1	34.3	1562	1563	29,400	115,300
PT2	45.1	1505	1475	72,100	267,000
PT3	65.2	1431	1249	12E ⁶	64 E ⁶

The first-stage high-pressure turbine blade shows a minimum component stress rupture life of 16,200 hours. This life was calculated using the 80 percent allowable stress criteria described below. Initially the HPT blade was envisioned to have a bulk average root stress of 29 ksi assuming an airfoil taper ratio of 0.5. The allowable metal temperature of 1597°F was set to meet the 30,000 hour life requirement. Cooling levels were calculated and this stress/temperature combination was incorporated into the cycle. The final mechanical evaluation saw it necessary to modify the taper ratio to 0.7 since the HPT blade is to be film-cooled which will typically have heavier internal passage configurations. The change in taper ratio resulted in the 3 ksi stress increase and hence the reduction in stress rupture life. The design will require either a few ksi reduction in stress or a 25° F reduction in bulk temperature (or some compromise).

The second-stage power turbine blade shows a 72,100 hr minimum life when evaluated using single crystal material properties. This life clearly shows potential to down grading the material to a directionally solidified (DS) or even an Equiax cast material.

Similarly, the last-stage power turbine blade operating at a bulk metal temperature near 1250°F will not require either single crystal or DS configurations. This blade should not pose any problems in terms of stress rupture or creep, but rather will require investigation into high cycle fatigue due to it's long length and mass.

The design criteria selected utilizes 80 percent of typical stress rupture data when minus three sigma statistical data is not available. The 80 percent stress allowable leads to substantial conservatism in terms of total life. For example, the first-stage high-pressure turbine blade shows an average bulk root stress of 33 ksi. Dividing 33 ksi by 0.8 indicates that the nominal curve LMP corresponding to 41.25 ksi should be used. Given 30,000 hours of required life and an LMP of 49.75e3 the allowable metal temp would be 1573°F. To evaluate the nominal life, we must hold the temp to 1573°F but utilize the nominal curve LMP corresponding to the original 33 ksi. The nominal curve LMP for 33 ksi is 50.86e3, which indicates the nominal life to be 104,000 hours. In this example, lowering the allowable stress level to 80 percent of nominal, leads to life calculations reduced to 28.8 percent of nominal life.

The reduction in stress essentially shifts the Larson-Miller parameter curve downward. This downward shifting may be deemed as inappropriate because the curves generated form data gathered at constant stress levels. The scatter of data at one stress level is not necessarily related to the scatter at another stress level. Should we choose to update the design criteria, a possibly better scheme would shift the LMP curve to the left. This shift would be addressing the scatter in LMP of many samples all at the same stress level. Another approach, is to place the conservatism on the allowable temperature, either a percentage of temperature or a constant temperature

differential (ΔT). This approach is similar to lowering the LMP value since they are both linearly related to life. Best of all, stress rupture testing should be conducted with enough samples at each stress level such that good confidence in minimum properties can be obtained, eliminating the need for estimating minimum properties.

6.2.3.2 Disk Stress

Table 15 shows the disk stress results for the high-pressure, low-pressure, and power turbines. Figures 14 through 18 display the tangential and radial stress as a function of disk radius for each of the turbine stages.

Table 15. Turbine Disks, Uniform Thickness Disk Stresses

Turbine Stage	Bore Tangential Stress (ksi)	Bore Radial Stress (ksi)	Average Tangential Stress (ksi)	Bore Configuration
HPT	132	0	67.4	Hollow
LPT	56.0	56.0	48.3	Solid
PT Stage 1	39.3	39.3	34.2	Solid
PT Stage 2	41.6	41.6	36.4	Solid
PT Stage 3	48.4	48.4	43.0	Solid

Figure 14. High Pressure Turbine Disk Stress

Figure 15. Low Pressure Turbine Disk Stress

Figure 16. Stage 1 Power Turbine Disk Stress

Figure 17. Stage 2 Power Turbine Disk Stress

Figure 18. Stage 3 Power Turbine Disk Stress

The HP turbine disk is clearly the most highly loaded. The disk design has a hollow bore which provides access for the LP turbine shaft. The peak tangential stress of 132 ksi is well within the range of typical disk material capabilities. Several Solar engines currently run hollow bore designs exhibiting higher peak stress levels than shown for the ATS ICR, and have each proven durable in operation. The LP turbine disk having a solid bore indicates much lower stresses while spinning at the same rotational speed and having heavier rim loading. All of the power turbine disks have solid-bore configurations, creating low peak stress levels. The main challenge for the turbine disks is dealing with the increased thermal load at the rim. The two-spool ICR will have increased rim temperatures up to 200° F hotter, as compared to current Solar disk designs. The increased rim temperature will require new materials for the HP turbine disk rim which exhibits better creep and rupture properties than current wrought alloys. The LP turbine disk may require the same approach, although the application of a single material wrought disk is still possible. The power turbine disks will not see the same severity of the increased thermal rim loading. The first- and second-stage power turbine disks mirror conventional Solar designs and should not pose any difficulty in detail design. Lastly, the third-stage power turbine disk will require efforts to optimize the firtree attachment. The first four stages have blade counts that will allow for adaptation of current broach configurations. The last stage power turbine will require a new slender broach configuration to attach the 64 blades to the 18 inch diameter disk.

6.3.3 Dual Property Disks

The requirement to operate turbine disks at higher rim temperatures also having higher stress levels due to blade pull requires an upgrade of material properties for creep, stress rupture and possible low cycle fatigue strength.

Current Solar disk configurations, isotropic property single material wrought disks, have design limitations in creep or stress rupture at or near the disk rim, and exhibit limitations in low cycle fatigue at the disk bore or centerline. Dead rim temperatures of ~1100° F currently exist with post average stress levels of 40 to 50 ksi containing peak concentrated stresses of 80+ ksi. This rim condition is coupled with a bore stress near 90 ksi for solid bore disks, and 150+ ksi tangential stress for hollow bore configurations. The hollow bore configurations challenge the LCF allowables in thermal transient cycles, where the solid bore configurations have comfortable margins.

Increasing the disk rim allowable temperatures will allow blades and their attachments to run hotter, requiring less cooling flow, and resulting in greater cycle efficiencies. Boosting rim temperatures by, 200° F would certainly require better creep and S.R. properties in both the blade attachments and disk posts. Further, the 200° F rim temperature increase would heighten the severity of the disk bore's thermal transient condition.

Dual property disks have been proposed with cast MAR-M-247 rims coupled with forged Astroloy (powder) or Udimet 720 bore materials. This combination would allow for higher steady state rim temperatures but could not allow the bores to run any hotter than the peak bore temperatures of current Solar disks. Current cooling delivery schemes need substantial modification to fully exploit the advantages of dual property disk designs.

Today's first-stage disks receive cooling air via a preswirl impinging 800-850° F compressor discharge air onto the disk forward rimseal. The cooling air, matching the tangential disk velocity, passes through an array of axial holes in the rotating rimseal delivering appropriate cooling flow for the disk posts and turbine blades. The required turbine blade cooling flow, passing first by the disk posts maintains the disk live rim at reasonably cool levels, where radial conduction inward (from the

disk posts) toward the disk bore, and disk cavity surface convection, does not lead to excessive bore temperatures.

It is important to note that the total heat flux due to conduction from the blade platform/ post area to the disk rim is absorbed by the cooling air delivered from the preswirler. By controlling the steady state disk temperature profile solely with this rim cooling air, the disk live rim steady state temperature can be no hotter than the allowable bore temperature. Under steady state operation thermal conduction will drive the disk temperature to a near uniform temperature matching that of the disk at the heat sink. Further, it should be recognized that the thermal gradient between the flowpath heat source and the cooling air sink is sustained over a short length. The entire gradient exists between the platform and the disk rimseal area at steady state condition. It is known from Fourier's law governing thermal conduction, that the heat flux due to thermal conduction is linearly proportional to the thermal gradient and inversely proportional to the length over which the heat is conducting. This indicates that if the bore is maintained at the same temperature by cooling air delivered closer to the bore, the heat flux would be reduced and less cooling air would be required.

The main difference between these two cooling approaches is that in the conventional high radius cooling scheme, the cooling air drives the maximum disk web temperature to a controlled level "shutting off" the heat flux from conducting inward toward the disk bore. This forces the balance of the disk to run at or slightly below the cooled live rim temperature.

With the introduction of cooling at lower radii, the cooling air heat sink will govern the coolest disk temperature. The total heat flux conducted through the disk will be reduced, but will result in a thermal gradient being sustained during steady state operation.

6.3.4 Proposed ATS Disk Cooling Schemes

Utilizing the Solar standard configuration using 1050-to-1100°F recuperated compressor discharge air would provide the means to operate rims substantially hotter while keeping blade temperatures within reason. The disks only having rim cooling would force disk bore steady state temperatures above 1100°F greatly reducing the low cycle fatigue (LCF) life for our current forged bore materials.

A cooling scheme having both a mix of compressor discharge and recuperator air below 850°F coupled with exclusive recuperator exit air at 1000°F could provide for potential dual property disk material applications. The recuperated compressor discharge air could be delivered to the disk rim using the standard preswirler/rim seal approach, while the 850°F mixed air would need be impinged at or above the dual material bond line to provide a thermal sink for the rim thermal conduction. This cooling scheme would certainly move the worst case transient thermal stress from the bore region radial outward into the disk web or possibly to the disk rim. Provisions to discharge the cooler disk bore cooling/buffering air back into the main gas stream would be required. One option would require opening the lower forward rimseal lab seal operating clearances providing a path for all expended disk bore cooling air to be mixed the blade cooling air supply. Balancing the cooling flow levels under transient seal clearance would certainly prove to be challenging. A second approach, similar to the first, entails recuperated compressor discharge air (1100°F) to be delivered through a standard rim seal, and would likewise have lower temperature Pcd cooling air providing disk bore cooling and disk buffering. The bore cooling air would be expended through either disk through holes transferring the cooling air to the disk aft face below the aft rimseal, or could be expended through the disk bore (if hollow) and allowed to wash up the aft disk face buffering the stage 1/stage 2 disk cavity.

Alternative cooling schemes have been suggested to incorporate a rotating disk cover where blade cooling air would be delivered under the cover, near the bore. The disk pumping action would provide a desirable increase in cooling air pressure needed for new blade film cooling schemes. The cooling air delivered at a low radius would travel between the disk cover and the disk face washing the disk surface while force be pressurized to increased levels.

This configuration has limited possibilities for the ICR engine configuration where the main cooling air source is intended to be exclusively from the recuperator exit. The 1100° F air would be too hot for current disk bore materials to survive. Thus, for the two spool ICR configuration, we have proposed to mix compressor discharge air with recuperator exit air. The overall thermal efficiency would decline somewhat, because some of the core flow bypasses the recuperator. This has proven to be a reasonable compromise allowing for simplified cooling delivery schemes, combined with the pressure increase needed for HPT blade film cooling.

6.4 POWER PLANT INTEGRATION

The Packaging Team is nearing completion of a solid 3-D graphic representation of an ATS50 power plant, see Figure 19. The graphic lacks only skid enclosure and systems packages for a complete picture. The most noticeable feature of the power plant is a vertically mounted recuperator which provides a number of packaging advantages with few negatives.

The Packaging Team has also reviewed typical turbine-driven power system parasitic losses (see Figure 20). Schedules of potential reductions to package-related parasitic losses have been compiled. Feasible parasitic load reductions that have already been identified are reductions in mechanical losses of 10 percent and of continuous parasitic electrical loads of 54 percent.

Markups of system schematics have been completed showing the elimination of a number of components, contributing to improved RAMD and reduced first cost.

Figure 19. System Package

Figure 20. Typical Industrial Power Generation System Energy Distribution

SECTION 7.0

TASK 7 - INTEGRATED PROGRAM PLAN

The detailed planning work on Task 7 has been paced by the completion of Tasks 5 and 6 (Market Studies and System Definition and Analysis) and has not begun. However, development of a Phase III program strategy has been initiated based upon preliminary data from Tasks 5 and 6.

This preliminary program strategy has allowed for initiation of general program planning in the critical areas of manpower requirements, material costs, facility identification, strategic supplier evaluation, and overall scheduling. Upon completion of Tasks 5 and 6 and our "program strategy" is firmed up, concentrated efforts to define the detail of the research, development and demonstration program plan to commercialize the GFATS will commence.

SECTION 8

TASK 8 - DESIGN AND TEST OF CRITICAL COMPONENTS

The purpose of Task 8 is to perform advanced design and test work on technologies which are critical to the success of Solar's ATS. Starting such work in Phase II of the program will assure that sufficient progress has been made so that timely application of these technologies can be made in Phase III. This timing will also provide for early discovery of any technological barriers that may prevent application of the intended technology. Should such occur, timely transition to a backup technology can then be made.

As originally awarded, Solar's Phase II contract specified three subtasks within Task 8:

<u>Subtask Number</u>	<u>Title</u>
8.1	Low Pressure Drop Recuperator
8.2	Low Pattern Factory Combustor
8.3	Autothermal Fuel Reformer (ATR)

In December of 1993, discussions with the COTR led to Solar's preparation of a proposal for six additional subtasks:

<u>Subtask Number</u>	<u>Title</u>
8.4	High Temperature (Dual Property) Turbine Discs
8.5	Low Emissions Combustion
8.5	Total Plant Controls (Man/Machine Interface)
8.7	Recuperator Materials
8.8	Reheat Combustor Materials
8.9	Composite Ceramic Airfoils

Funding for these additional tasks did not become available until July of 1994, at which time Solar initiated the requisite review and update of costs. This and corresponding D.O.E. reviews are expected to be completed such that award of the revised contract could occur in August.

8.1 LOW PRESSURE DROP RECUPERATOR (Subtask 8.1)

Solar has at its disposal, a steady-state performance rig located at the Caterpillar Technical Center outside of Peoria, Illinois. This rig is pictured in Figure 21. These tests are required for accurate full-scale performance predictions. In order to establish the performance of an air cell assembly, which is the modular component of Solar's PSR, a group of full size cells are built into a test core. The number of cells in the core should be the maximum that the rig can accommodate for the flow conditions required.

Figure 21. Steady-State Performance Rig

Solar has recommissioned this rig at Caterpillar in order to provide an accurate prediction of recuperator performance. Prior problems with this rig have resulted in a heat balance deviation greater than three percent. A thermal mixer was recently designed, build and added to the rig as well as adequate measurement instrumentation to improve this heat balance measurement.

Yet another important prediction is the recuperator's part load performance and response to transients. A simple transient response test has been designed to measure the response of the recuperator, primarily its heat transfer effectiveness, to a sudden reduction in load. Data obtained from all of these tests will be integrated with ongoing cycle studies in order to more completely map the system performance under all expected operating conditions.

The Fanning Friction Factor (F) and the Colburn Modulus (J) are two parameters that relate frictional factors (pressure loss) and heat transfer. A small transient test rig is available at the Caterpillar Technical Center for acquiring f and j data for a particular compact heat exchange surface. This rig is pictured in Figure 22. Small sample cores are built of the air side flow geometry and separate cores are made of the gas side geometry. The Solar hicapacity air cell will be tested in this rig to determine the f and j data. These data are instrumental in the design of an advanced recuperator. An error band of about 13 percent has been commonly accepted as "the best you can do" for f and j tests. However, recent literature challenges that point and rig modifications are underway to minimize the uncertainty to approximately two to three percent if successful. Preliminary tests run on previously run sample geometries indicate the rig was functioning as expected before the modifications were started.

Figure 22. Transient test Rig (f and j rig)

8.2 LOW PATTERN FACTOR COMBUSTOR (Subtask 8.2)

A subscale catalytic combustor test rig is being used to evaluate different premixer and catalyst bed designs at conditions anticipated in the ATS gas turbine. The rig includes a preheat combustor, premixer, catalyst bed, and a post catalyst combustor. An illustration of the rig is included in Figure 23. This rig has been developed internally at Solar and has been described in detail¹. The rig accommodates a five inch diameter catalyst bed with a four inch diameter flow path. The preheat combustor is not coupled to the premixer but is located in the air supply pipe more than 15 diameters upstream of the premixer inlet. The premixer is a multi-point venturi design that includes six venturi's around one central venturi. At the mouth of each venturi is a fuel delivery tube with four fuel orifices near the tip. The post catalyst combustor is an adiabatic volume that contains ceramic flame stabilizer bars to allow sufficient residence time to complete the combustion reactions. Located within the post-catalyst combustor is an auxiliary fuel injector which is used in the light-off of the catalyst.

Figure 23. Catalytic Combustor Test Rig

the goal is to minimize variations in fuel concentration temperature profile and velocity profile across the face of the bed. The temperature profile has been adjusted to 50°F through design of the premixer and keeping sufficient distance between the preheat combustor and the premixer. Velocity profile has not been measured, but is estimated to be flat because of the pressure losses taken across the multi-point venturi premixer and catalyst bed. The fuel concentration is the most critical parameter to control. The multi-point venturi design allows tailoring of the fuel concentration across the bed inlet by adjusting the fuel flow into each of the seven venturis. The fuel profile is measured with seven, single-point emission probes installed directly upstream of the catalyst bed. A standard hydrocarbon analyzer is used to measure the fuel concentration while injecting natural gas through the premixer at typical pressures and temperatures of combustion tests. With this equipment the fuel concentration of the premixer described above was optimized to within three percent. The fuel flow was adjusted to each of the venturis by increasing the fuel orifices.

Combustion testing of the catalyst beds is summarized in Table 16. As the table indicates, testing with these beds has been required to improve catalyst bed activity sufficiently to maintain stable catalytic combustion. Stable catalytic combustion is achieved when the auxiliary burner

¹Cowell, L. H. and Larkin, M.P. "Development of a Catalytic Combustor for Industrial Gas Turbines," ASME 94-GT-254.

Table 16. Summary of Testing of Catalytic Combustor in Subscale Rig

Test No.	Set-up	Results	Comments
70 to 72	Bed Design A, (4 pc, 5.0 in. long). Inlet plate on premixer included in 70 and 72. 60 fps, 115 psig 800 to 960° F inlet, no preheat combustor	Temperature rise across bed below 100° F. Stable catalytic combustion not achieved.	Low bed activity.
73	Bed Design A New Piece 1. 60 fps, 775 to 980° F inlet, 4 vol%, 118 psig, no preheat combustor.	Temperature rise across bed of 150 to 250° F. Stable catalytic combustion not achieved.	Low bed activity, improved with different first piece.
74	Same bed and operating conditions as Test 73. Preheat combustor operated to elevate inlet temperature from 900 to 1120° F.	Temperature rise across bed of 500° F. Stable catalytic combustion achieved without auxiliary burner.	Increased activity with inlet temperature increases. Catalytic combustion not very stable. Bed activity still low.
75	Bed design B, (5 piece, 6.5 in. 60 fps, 115 psig, 4 vol%, 800 to 1120° F inlet. With and without preheat combustor.	Temperature rise of 225° F. Stable catalytic combustion not maintained.	Increased bed length did not increase activity.
76 and 77	Bed design B, new first piece. 60 fps, 35 psig, 3.5 to 4 vol%, 900 to 1025° F, with preheat combustor.	Temperature rise of 300 to 400° F. Stable catalytic combustion maintained for 30 minutes. High CO and UHC emissions. Catalyst bed cooler in center.	Bed activity improved. cool catalyst center indicates greater fuel concentration required in center.

downstream of the bed can be extinguished and combustion is maintained. As described previously in the footnote, testing has been on-going with Engelhard-supplied beds for the past two years. The reduced activity reported in these test results from a change in the operating parameters for ATS conditions, primarily an increase in the face velocity. The face velocity has been increased from 40-to-50 fps to reduce the likelihood of combustion occurring within the premixer upstream of the catalyst bed. Combustion upstream of the catalyst bed has been a problem and by increasing the catalyst inlet temperature as is the case in a recuperated engine, the problem can be more pronounced.

Tests 76 and 77 have successfully demonstrated stable catalytic combustion for an extended period. These tests have been completed at lower combustor pressures to repeat tests in Solar's rig that have been completed by Engelhard. The data from the two facilities compare favorably. High CO and UHC emissions (greater than 1000 ppm) indicate uneven combustion occurring within the bed. The cause is suspected to be a non-ideal fuel distribution. This is being investigated. NOx emissions remain high (5-to-15 ppm) as long as the preheat combustor is required to boost catalyst inlet temperatures. The preheat combustor has been required for rig testing because of heat lost in the facility air heater and will not be required for the ATS engine. These tests were concluded with a controlled shutdown after operating the catalytic combustor for 30 minutes.

Future activity will include optimization of the fuel distribution and catalyst bed activity to accommodate the higher activity near the wall. Tests will be conducted with fuel flow in the center premixer venturi approximately five percent higher than the average fuel concentration. Beds of longer length will be tested as required to optimize performance at all conditions anticipated in the ATS.

8.3 AUTOTHERMAL FUEL REFORMER (ATR) (Subtask 8.3)

The steam-fuel reforming converts natural gas and steam to a high-hydrogen-content fuel. The reforming process requires heat to produce the steam-methane reactions. To accomplish the reforming reactions, parts of the parent fuel have to be burned to raise the temperature to approximately 1250-to-1500°F. This approach of burning parent fuel partially to supply reaction energy is known as the autothermal fuel reforming. The high-hydrogen-content fuel has a wider range of combustion stability than natural gas. Hydrogen is lighter and more reactive than the original fuel and it mixes better with the combustor air. The hydrogen in the combustible stream enhances the flame stability and speeds the burn rate through the combustion zone. It can be burned at much leaner (low flame temperature) conditions, producing lower NOx emissions. The ignition energy of the hydrogen-rich gas is much less than that of natural gas, a major factor in making low-temperature catalytic combustion possible. The steam reformation of fuel utilizes the heat removed from the intercooling and the exhaust. Recovering the waste thermal energy in the form of steam and then converting it to hydrogen which chemically recuperate a gas turbine engine enhances the system efficiency.

8.3.1 Reforming Catalyst/Parameter Screening

An effective fuel reforming catalyst requires the maintenance of a desired activity over its lifetime. Sulfur poisoning, catalyst migration, and substrate durability are encountered as probably contributing to the problems of catalyst life. Sulfur in natural gas is known to poison the commercial steam/methane reformation catalyst. The selection of a sulfur-tolerant catalyst is one of the critical material requirements for the processing of autothermal reforming.

8.3.1.1 Reforming Operation Conditions

The operating conditions of inlet temperature and steam/methane ratio are varied to optimize the catalyst activity. In the tests, a bench scale tube reactor is used to screen the reforming catalyst. The bench reactor (Figure 24) contains an one-inch diameter of stainless steel tube heated with an electric furnace. The reactor furnace has 36 inches length with three temperature control zones where the temperature in each zone can be controlled and programmed at specified rates. The screening tests have been run using one-inch diameter of monolith or foam which formulated with reforming catalysts to determine the catalyst activity and select an optimal operating condition. Preheated natural gas and steam are mixed before entering the catalytic conversion zone. The fuel used in the test was pipeline natural gas, containing about one ppm of hydrogen sulfide (H₂S), as supplied by the local utility through the distribution network. Typical compositions are:

Component	Composition (mol %)
Methane	92.79
Ethane	4.16
Propane	0.84
Butane	0.18
Pentane	0.04
C6+	0.04
Carbon Dioxide	0.44
Nitrogen	1.51
Hydrogen Sulfide	1. ppm

During a run, the product gas samples are taken at various time intervals. Once collected, the samples are analyzed for hydrogen and other gas components with a fast, highly sensitive gas analyzer (Microsensor Technology Inc., Model M200). The analyzer has been calibrated with standard gases (H_2 , CH_4 , CO , CO_2 , etc.) in the range of the measurements being made. Several catalyst candidates and operating conditions are prescreened to select a primary catalyst and operating conditions for the autothermal reforming rig testing.

Figure 24 shows the test results on producing hydrogen in the product stream as functions of temperature and steam to carbon ratio (S/C, carbon in natural gas). Hydrogen concentration increases remarkably with temperature below 1300°F but beyond this point the advantage of higher temperature is less pronounced. The equilibria of the reactions involved in steam reformation depend largely on the operating temperature and steam to carbon ratio. The percent reforming increases as the temperature and the steam to carbon ratio are increased.

Figure 23. The Tube Reactor for Reforming Catalyst Screening Tests

Figure 24. Test Results of Hydrogen Produced as Functions of Temperature and Steam to Carbon Ratio (R)

The equilibria of the reactions between steam and natural gas (methane) depend also strongly on the operation pressure of the reactor. The reactions of steam-natural gas reforming can be depicted by the following equations:

Using mole fractions or partial pressures, the relationship between the pressure, gas composition, and equilibrium constant at any temperature can be expressed by the equation for the reaction (1):

$$\frac{(P_{\text{H}_2})^3 (P_{\text{CO}})}{P_{\text{CH}_4} (P_{\text{H}_2\text{O}})} = \frac{(X_{\text{H}_2})^3 (X_{\text{CO}}) (P_r)^2}{(X_{\text{CH}_4}) (X_{\text{H}_2\text{O}})}$$

where

- K = equilibrium constant of the reaction
- P = partial pressure of a component in the gas stream
- X = mole fraction of a component in the gas stream
- P_T = total pressure of gas in atmospheres

From the expression for K₁, the quantity of hydrogen produced at equilibrium varies with the pressure. Since another equilibrium also exists from the reaction (2), the yield of hydrogen from the steam and natural gas must also satisfy the equation:

$$K_2 = \frac{(X_{CO_2}) (X_{H_2})}{(X_{CO}) (X_{H_2O})}$$

From the above expression, pressure does not affect the value for equilibrium constant (K₂) of reaction (2).

The maximum possible reforming of methane to hydrogen is limited by equilibrium at given conditions of temperature, pressure and steam to carbon ratio. Conversion efficiency (expressed as hydrogen concentration in product gases) as functions of the temperature and pressure is shown in Figure 25. At higher operating temperatures the reformation reactions are less dependant on pressure.

Figure 25. The Reforming Efficiency (expressed as hydrogen concentration in product gas) as Function of temperature and Pressure

Autothermal reformation burns some of the methane fuel for raising the temperature to accelerate the reactions and to gain a higher level of reforming (more hydrogen production).

The water usage sometime limits the applicability of steam reforming in the gas turbine engine. Figure 26 shows the natural gas reformation producing near 50 percent hydrogen in the product stream without steam injection. Although producing high hydrogen is desirable, a satisfactory amount of hydrogen generated in the reformed product could be more important on the improvement of flame stability. If the reformer would be run without or with a low steam rate, a reforming catalyst needs to be developed to overcome the carbon built-up problem.

8.3.1.2 Catalyst Screening

Catalysts suitable for use in reforming must meet the dual requirements of high catalytic activity and sulfur poison tolerance. The level of catalytic activity dictates the hydrogen yield in the product stream. Candidate catalysts must meet these requirements with strong thermal stability. Two basic groups of reforming catalysts, nickel- and platinum-basis, have been selected for the preliminary testing of the reforming activity. The tests are conducted at 1500° F and steam to carbon ratio of 1.34 under atmospheric pressure. Table 17 lists the catalyst type as well as the normalized product distribution of the test results on a dry gas basis. These experimental data given in Table 17 were carried out over a period of 4-to-10 hours. All catalysts resulted in producing about 70 percent hydrogen in the product stream.

Figure 26. The Product Distribution of Fuel Reformation Without Steam Injection

Table 17. Catalyst Type and Test Product Distribution

Catalyst Supplier	DYCAT		Engelhard			Protech		Met-Pro	
Type Content Bed Dimension	Foam Ni + La on Alumina 1 in. dia x 5 in.		Monolith (200 CPI) Pt + Rh on Cordierite 1 in. dia x 6 in.			Monolith (400 CPI) Ni on Cordierite 1 in. dia x 6 in.		Monolith (300 CPI) Pt + Pd on Cordierite 1 in. dia x 4 in.	
Testing Time	60 min.	210 min.	60 min.	240 min.	600 min.	60 min.	240 min.	250 min.	270 min.
Composition									
H ₂	73.0	69.9	70.9	7.19	69.3	71.2	70.7	71.6	72.2
CO	24.0	23.7	19.6	18.1	15.9	18.0	18.1	15.1	14.6
CH ₄	1.0	4.8	3.2	3.9	7.4	4.9	5.4	3.9	3.9
CO ₂	2.0	1.6	6.3	6.1	7.4	5.9	5.8	9.4	9.3
Heating Value (Btu/lb) (LHV, dry basis)	11,716	12,365	10,606	11,177	11,323	11,407	11,479	10,147	10,318

Figures 27 through 29 plot the product concentrations as a function of testing time for individually tested catalysts. Reactant conversion and product distribution remained about constant during the 4-to-10 hours testing for both the nickel and platinum catalysts. Differences in activity of the nickel and platinum catalysts have been observed on the CO and CO₂ yields. The nickel-basis with lanthanum promoted catalyst (DYCAT foam) converts natural gas to more CO instead of CO₂ and yields a higher heating value of product gas than other catalysts. Analyses of the used catalysts by electronic diffraction X-ray (EDX) show no significant difference between the original and the used catalyst (Figure 30). Several decolorized spots on used catalysts were observed and found later to be due to contaminants of potassium in the water used to generate steam for the test.

Catalyst poisoning is of primary concern on its useful life of autothermal reforming. Sulfur is known to poison the catalysts used for the commercial reformation processes. A zinc oxide-based absorbent can effectively remove all of the sulfur compounds likely to be found in the hot vaporized fuel stream. An absorbent bed could be added in the vaporized fuel delivery line or it could be integrated into the catalytic reformer. In addition, sulfur poisoning is largely reversible. The bulk of the sulfur can be removed from the catalyst by steaming. The ongoing experiments are currently testing the sulfur tolerance by doping 200 ppm of H₂S into the fuel. Regeneration of used catalysts by steaming, as well as study of the effect of a dual oxide absorbent on controlling sulfur poisoning, are underway. Further testing to investigate sulfur poison over longer test time is scheduled.

8.3.2 Rig Design and Testing

The autothermal reforming system can be integrated with gas turbine cycles, using the intercooler to raise the steam and the engine exhaust to heat the steam temperature. The energy usually lost in the cooling process would be recovered and returned to the engine in the form of chemical energy. The low exhaust temperature would allow a considerable portion of the water present in the gas stream to be easily condensed. This would improve the efficiency and provide the desirable emissions. It is anticipated that all of the water needed for autothermal reforming will be available from the exhaust condensate. In addition, the steam produced from the combustion of hydrogen would increase the power available from the cycle when compared to the combustion of the parent fuel. This is equivalent to injecting steam into the turbine section at the combustor exit temperature.

Figure 27. the Product Distribution of Reforming Test with DYCAT Foam Catalyst

Figure 28. The Product Distribution of Reforming Tests With Englehard Monolithic Catalysts

Figure 29. The Product Distribution of Reforming Test With Prototech Monolith Catalyst

However, before the efficiency and emission advantages of fuel reformation can be realized in existing gas turbine systems, a number of catalyst and system design criteria must be addressed. These include catalyst life and poisoning, structural strength, product stream ignition, and flame stability and flashback. Concepts for integration of reformer with combustor and gas turbine system components also need to be developed.

8.3.2.1 Rig Design

Autothermal reforming rig tests will be conducted with a subscale, staged, rich-lean combustor consisting of the reformer and the selected catalyst. The detailed drawing of rig design is shown in Figure 31. The design features a fuel injection premixer, catalytic reformer, and dilution lean combustor. The injector design goal is to achieve effective premixing of steam and natural gas. The drawing also indicates a pilot combustor that partially oxidizes the fuel and supplies heat required for preheating the steam-fuel mixture to desirable reforming temperature. The pilot combustor is operated at rich conditions with an equivalence ratio around 2.3. The exhaust leaving the pilot combustor enters a mixing section where the combustion products contact with the incoming steam and fuel. The steam and fuel would be preheated prior to the mixing section. The mixed gases are constrained to pass through a catalytic reactor constructed as a multi-channel monolith where the high-hydrogen-content secondary fuel is generated. The hydrogen produced from the reforming creates energy from non-combustible steam and increases the thermal efficiency.

Figure 30. EDX Analyses of the Original and Used DYCAT Foam Catalysts

Figure 31. Reformer Rig Design Drawing

The monolith materials of construction will be either ceramic or metal. The active catalytic materials are deposited on the inner surface of the monolith channels thus exposing the reacting gases to a large catalytic surface area. A rapid quench zone consisting of an uncoated metal honeycomb is also included as indicated in Figure 31. After the quenching zone, sufficient air is brought angularly to mix rapidly with the reforming product which leads to autoignite. The rapid mixing is crucial to prevent local hot spots and ensure low NO_x emissions. The equivalence ratio in the post combustor is designed to be around 0.25:1. The pilot combustor and post combustor are all refractory-lined to reduce any heat loss. Thermal Ceramics (Augusta, GA) is being selected to cast these ceramic linings. The fabrication of the test rig is being initiated. A composed picture of the reforming test rig with its housing configuration is illustrated in Figure 32.

Figure 32. A Composed Picture of the Reforming Test Rig With Its Housing Configuration

8.3.2.2 Rig Testing

Combustion integration and operation problems are foreseen due to the high-hydrogen-content secondary fuel that is produced. Rig tests would be designed to resolve some of these problems. Tests will be conducted using heated pressure air and natural gas with steam and/or water injection to assess the likelihood of intercooler heat and exhaust wasteheat recovering. A primary catalyst and the best operating condition would be employed for the rig testing based on the test results and the selection from the previous tube reactor testing. The effect of pressure on combustor emission and efficiency will be quantified. The performance of the reformer/ combustor test will be evaluated as the combustion efficiency, emission levels, pressure drop, operating range, and flame temperature and its stability. All of the exhaust emissions will be monitored with the on-line instrumentation.

Safety concerns with hydrogen leakage and flashback will be detected during the tests. Once the material issues for handling the hydrogen-rich gas are resolved, the problems of operation and integration will be addressed. Non-equilibrium operation using a rapid air quench at the end of the reformer for low NO_x emissions will also be investigated.

Other aspects of adapting the CFC to the ATS50 such as scaling and injector sizing are not considered to be major problems based on previous experience.

As discussed in the proposal, another approach to the CFATS is to gasify the coal. Gasified coal could be burned in a conventional gas turbine combustor with minimal modifications. In fact the catalyst required to attain desired NO_x levels on natural gas might not be required for coal gas. The main tradeoffs between direct fired coal and gasified coal are:

1. The costs of the ancillary equipment; for direct-fired coal, the grinders, slurring equipment and ash removal system and for coal gas, the gasifier and compression systems.
2. For direct-fired coal, the material problems discussed above versus the possible deletion of the catalyst for coal gas.

Since the cycle parameters are now available, this task can now proceed.

8.4 DUAL-PROPERTY TURBINE DISK (Subtask 8.4)

One method of increasing the efficiency of future gas turbines is to reduce the cooling air requirements of the hot section components. This approach requires selection of materials which can perform continuously at higher operating temperatures. For example, application of polycrystalline ceramic, or advanced composite, turbine blade materials would allow substantially reduced, or eliminated, cooling flows.

However, in order to obtain maximum benefit from this approach, it is essential that turbine disk materials be developed which can also operate with reduced cooling requirements. This is the goal of the proposed Subtask addition 8.4, Dual-Property Turbine Disk Development. Reducing the cooling flow to the turbine disk not only improves the overall engine efficiency, but has the secondary advantage of improving the radial temperature profile to the nozzles. Reducing the radial temperature gradient across the nozzle dramatically lowers the thermal stress imposed on the nozzle material. Excessive thermal stress could make the application of ceramic nozzles impractical from a design standpoint. Therefore, development of turbine disk materials which can operate at higher rim temperatures is considered an enabling technology for effective utilization of ceramic nozzles.

Operation at rim temperatures significantly higher than that allowed by high strength wrought alloys necessitates the use of a material with creep properties equivalent to the cast superalloys typically used in turbine blades. The entire disk could conceivably be made of such a blade alloy, however, it is well established that these cast superalloys lack the tensile strength and LCF properties required for a disk hub. The desirable qualities of disk hubs are associated with tough, high-yield-strength materials typically obtained from fine-grain-size wrought forgings, or HIP+forged powder compactions. The diverse property requirements of the rim and hub areas of very high temperature turbine disks can be best achieved by utilizing a dual-material disk approach. This solution has been successfully demonstrated in the past by several fabrication methods, in both radial wheels for small auxiliary power units (APU's) and in larger diameter axial rotors. In the ATS Phase II Task 8 effort, the attachment of the two materials will be accomplished via a diffusion bonding technique developed at Solar. This process has proven highly successful in bonding wrought hubs to cast shrouded blade rings in the manufacture of large diameter centrifugal impellers for Solar's gas compressors.

Development of the dual-property disk technology is considered critical to meeting Solar's ultimate goals in the ATS program. The technology may also have application in Solar's current product line which can benefit from the ability to reduce cooling flows to the turbine. The goal will be to transfer the technology to other products for early technology "wins". Solar has committed some initial

internal funds toward the concept in advance of the Task 8 contract award. Two subscale specimens are in the process of being fabricated for this purpose (see Figure 33). These prototypes are being used to develop the appropriate cleaning procedures prior to HIP bonding the assembly. Metallographic analysis of the bond-line integrity will be performed after HIPing, and then initial heat treat developments will be performed. This will provide valuable information to direct a quick-start in the funded program.

Figure 33. Photograph Showing the Prototype Assembly to Be Used for Development of Cleaning Procedures and Hip Parameters for the Dual-Property Disk. The rim is fabricated from a cast blade alloy and the rim is a high-strength forged superalloy.

8.5 LOW EMISSION COMBUSTOR (Subtask 8.5)

The catalytic combustor design being developed at the subscale level as described in Section 8.2 will be tested in a single-can, full-scale combustor rig. The combustor diameter will be increased to approximately 10 inches. The system will be representative of the full-scale engine design including a premixer, catalyst bed, post-catalyst combustor and an integral part-load injector situated along the centerline of the cylindrical combustor. The combustor will be operated over the full range of engine operating points including simulated start-up, idle, low load, and catalytic mode over 50-to-100 percent simulated engine load. System performance will be characterized through measurements of emissions, pressure loss, catalyst bed temperatures, combustor exit temperature profiles, and pattern factor to the extent possible. Catalyst beds will be supplied by Engelhard Corporation based on their designs being developed during subscale testing. This effort is an important next step in advancing catalytic combustion to engine readiness.

8.6 ADVANCED MAN-MACHINE INTERFACE (Subtask 8.6)

Solar believes that improvements in gas turbine control systems are required in order to fully support ATS goals. Such a control system requires that a Man-Machine Interface (MMI) operate within a multi-tasking, multi-user environment. In this subtask, Solar will design and demonstrate such a system to include advanced human interface features, advanced networking capability the employment of expert systems, and low-cost, long-range communications capability. The systems will provide for easy customization to customer's unique requirements as well as for simple in-the-field reconfiguration.

8.7 RECUPERATOR MATERIALS (Subtask 8.7)

Solar's Primary Surface Recuperator (PSR) is a compact heat exchanger that delivers high effectiveness with a low pressure drop in a compact volume. This design is readily adapted to engines of most any size, however, cycle conditions do dictate that material selection. To date, type 347 stainless steel has been the material of choice for nearly all applications due to its physical properties, fabricability and relatively low cost. Other materials can be used should a particular engine cycle demand it.

The intercooled and recuperated cycles that have been studied are characterized by higher pressure ratios and possible higher exhaust gas temperatures. For these advanced cycles, materials of high creep strength, high yield strength and oxidation resistance are required, and for some cycles, more expensive alloys, such as Inconel 625 or Haynes 230, may be needed..

The primary failure mechanism for the PSR has been creep. Unfortunately, there has been a general lack of validated creep data for the thin foils that are the primary component of Solar's PSR. Therefore, a systematic study was undertaken to fully define the operating lines of various thin foils, including SS-347, Inconel 625 and Haynes 230.

8.8 REHEAT COMBUSTOR MATERIALS (Subtask 8.8)

The work proposed in this effort will be a first step in identifying ceramic materials that have potential for use in an ATS reheat combustor. Through a literature survey, manufacturers' data, and experience in other applications, up to three candidate materials will be selected for evaluation. Potential candidate materials for this application are shown in Table 18. Small (8 inch diameter) can combustors will be fabricated and tested at representative reheat combustor conditions.

Table 18. Candidate Ceramic Materials for the Reheat Combustor Application

Material Name	Type	Supplier
B-30	BZP	LoTEC
Altra KVS 16	high alumina fiber reinforced composite	International Thermoproducts
System 2.2 High Temperature Composite	Glass-Ceramic Fiber reinforced composite	AEA Technology

Testing will employ an existing combustion rig that will require relatively minor modifications. Through testing, the short term durability (50 hours) of the candidate materials will be examined at the extreme reheat combustor conditions. The tests will identify materials that have the potential to provide the durability required in a production gas turbine. In addition, coupon samples will be subjected to long term thermal exposures to determine oxidation behavior, structural stability, and residual strength. Results will be documented in a topical report at the conclusion of the effort.

8.9 ADVANCED CERAMIC MATERIALS (Subtask 8.9)

This subtask, contracted to Super-Temp Division of B.F. Goodrich Aerospace Division, will explore the use of advanced ceramic materials in high temperature components such as combustor liners and turbine airfoils. Materials considered are both carbon-reinforced silicon carbide (SiC) or silicon carbide-reinforced silicon carbide (SiC/SiC). Goals include the achievement of 30,000 hours of design life accurate life prediction and cost-effective fabrication in production quantities.

SECTION 9

PROGRAM SCHEDULE AND BUDGET

9.1 SCHEDULE

Completion of the first three Phase II tasks took place approximately six weeks late to the schedule shown in Section 1 of this report. This is due primarily to time required for key personnel to complete key prior assignments in order to become available for full time ATS work. This delay is not expected to be reflected in late completion of any of the remaining tasks. One exception is the execution and completion of Task 4 - Conversion to Coal. As noted in Section 4 of this report, start of this task depends on complete definition of combustor operating conditions and no other tasks depend upon its completion in order to start.

9.2 BUDGET

As of July 31, 1994, a total of 12,032 labor hours have been charged to the contract versus 10,836 planned. This is reflected in a total program cost (including Solar's contractual cost share) to date of \$1,320 K versus \$1,096 planned. This apparent overrun is temporary and was caused by the assignment of additional manpower in order to bring the program back on schedule. At this point, Solar's Phase II ATS work is expected to be completed within the labor and cost levels currently budgeted plus those associated with the Task 8 additions.

In addition to charges to the contract, an additional 5,338 labor hours were charged to an internally funded project directly related to ATS. This represents cost sharing of approximately \$480,000 in addition to that provided for in the contract.

APPENDIX 1

GLOSSARY

ACE	<u>Advanced Component Efficiency</u> . A Solar-funded multi-disciplinary, advanced technology program. It currently includes programs on compressors, turbines, cooling, and ducting.
ATS	<u>Advanced Turbine System</u> . A gas turbine-based energy conversion system as defined To the U. S. Congress by the U. S. Department of Energy (DOE). An ATS is distinguished from current gas turbines by markedly increased thermal efficiency, reliability, availability and maintainability and by decreased exhaust emissions and cost of power produced.
ATS50	Solar's designation for an ATS designed to meet the contract goal of 50 percent thermal efficiency.
ATS60	Solar's designation for an ATS designed to meet a "stretch" goal of 60 percent thermal efficiency.
BASELOAD	Typically describes a specific number of operating hours the power generation system is operating. In this case it represents (on average) 6500 hours.
BZP	<u>Barium Zirconium Phosphate</u> . A family of materials whose composition can be tailored to result in low, or zero, coefficient of thermal expansion. It is resistant to thermal shock, and is stable to 1200°C.
CFATS	<u>Coal-Fired Advanced Turbine System</u> . An advanced turbine system modified to operate on coal or coal derived fuel.
CFCC	<u>Continuous Fiber-Reinforced Ceramic Composite</u> . A high temperature composite material which consists of a ceramic matrix reinforced with fabric, tape, or rows of a high modulus fiber. CFCC's are typically not as flaw sensitive as monolithic ceramics, and can be fabricated into large parts such as combustion chambers.
CHORD	The axial length of an airfoil measured along a line drawn tangent to the leading- and trailing-edge radii. Chord is the characteristic dimension used in scaling of airfoils.
CSGT	<u>Ceramic Stationary Gas Turbine</u> . A DOE-funded program contracted to Solar Turbines Incorporated. The program goal is to demonstrate improved performance of a production gas turbine by retrofitting key components with ceramic counterparts for a 4000-hour field test.
EDX	<u>Electron Diffraction Xray</u> .

EFFECTIVENESS	(ϵ_R for recuperator, ϵ_i for intercooler). A characteristic of a heat exchanger -- the ratio of the amount of heat transferred to the total heat available in the hot fluid between its entry temperature and the entry temperature of the cold fluid.
EFFICIENCY, COMPRESSOR	(η_c). Ratio of the work imparted to the air flow to the total work input. Unless otherwise specified, "adiabatic" efficiency (subscript. - ad) is implied, i.e., flow conditions at both inlet and outlet are in terms of absolute, or total, pressure and temperature. "Polytropic" efficiency (subscript. - p) is the theoretical efficiency of a very large number of identical low pressure-ratio stages and essentially normalizes pressure ratio for comparison of differing compressors.
EFFICIENCY, THERMAL	(η_{th}). Also termed cycle efficiency. The ratio of energy input to energy output from a gas turbine. Unless specifically designated otherwise, thermal efficiency of an ATS will be defined on the basis of net power output from the turbine shaft.
EFFICIENCY, TURBINE	(η_t). Ratio of the work produced by a turbine to the work it extracts from the gas flow. When applied to a GP turbine, total-to-total efficiency is implied, this definition being based on total or absolute inlet and exit conditions. When applied to a power turbine, total-to-static efficiency is implied, thus accounting for static pressure recovery as a result of kinetic energy conversion in the exhaust diffuser.
FGM	Functionally Graded Material. A multi-layer material or coating system with each layer graded to provide different properties at the surface of the component relative to the core or substrate. An illustrative example would be a turbine tip-shroud with a top coating tailored for abrasability, an intermediate layer designed for oxidation resistance, and a substrate tailored for high strength.
GAS PRODUCER	(Abbr. GP) An element, or module, of a gas turbine which is required to provide a hot gas stream capable of propelling a power output turbine or other energy conversion device. As a minimum it consists of a compressor, a combustor, a turbine and associated shafting. Each of these components, in turn, may be of any of several types and may consist of one or more stages.
GPT	Gas Producer Turbine. May include HP and LP turbines.
GFATS	Gas-Fired Advanced Turbine System. An advanced turbine system which uses natural gas for fuel.
GP	See GAS PRODUCER.
HCF	High Cycle Fatigue. High cycle fatigue is characterized as the fatigue mechanism where vibratory loads are superimposed onto a steady state loading condition. The typical example is turbine blade under aero excitation.

The turbine blade exhibits a significant steady state stress level due to the centrifugal load, where vibratory stresses are induced by high frequency pressure perturbations in the gas stream. Vibratory stress levels are typically within the elastic range of the materials. Failures typically occur due to excitation at the component's natural frequency, setting up a resonant condition. This failure mode is associated with large numbers (10^5 - 10^7) of low stress amplitude cycles.

- HIP** Hot Isostatic Pressing. A process for simultaneously heating a component, or powder metallurgy compact while subjecting it to isostatic pressure using an inert gas atmosphere. The temperature and pressure area are high enough to produce densification of the materials.
- HP** High Pressure. This prefix is used to designate an individual section of a series pair of aerodynamic components operating in the high pressure regime of the two. HPC designates the high pressure compressor and HPT designates the high pressure turbine of an HP, or high pressure, spool.
- INTERCOOLER** A heat exchanger placed between stages of compression in a gas turbine in order to remove heat from the compression air thereby reducing the temperature of air going to and through the succeeding stage(s).
- I.O.U.** Invester Owned Utility. There are approximately 260 investor owned utilities in the United States. They are differentiated by the fact that they are owned through a stock ownership plan and are usually traded on the open market. they are more tightly regulated in their service area by regulatory bodies like public utility commissions than the Municipal utilities or the cooperative utilities.
- IPP** Independent Power Producer. A non-regulated producer of electric energy. Currently the majority of these IPPs are larger power generation stations much like the Central Power Station that is owned by the Investor Owned Utilities.
- LAC** Levelized Annual Cost. The name of the financial model used by the utilities to determine the cost of the capital equipment of the usable life of the equipment. Takes into consideration present worth arithmetic. Sometimes interchanged with book value analysis/life cycle analysis.
- LCF** Low Cycle Fatigue. The fatigue mechanism typically associated with large thermal or mechanical loading. The cycles are typically a transient loading condition, such as start-up and shut-down of an engine. Low cycle fatigue is characterized by allowable pseudo elastic strain range for a material whereby a fatigue crack will develop after a specified number of cycles. typically LCF is considered cyclic loading of the base load leading to plastic deformation. Cycles are on the order of less than 10,000 to 15,000 cycles.

LP	Low Pressure. This prefix is used to designate the individual of a series pair of aerodynamic components operating in the lower pressure regime of the two. LPC designates the low pressure compressor and LPT designates the low pressure turbine of an LP, or low pressure, spool.
MCrAlY	A designation for an overlay coating applied to superalloy turbine components for increased oxidation and corrosion resistance. "McrAlY" is derived from the elemental constituents of the coating i.e., Cr = Chromium, Al = Aluminum, Y = Yttrium M = Ni Fe or Co (depending on substrate).
NEPA	National Environmental Policy Act.
PATTERN FACTOR	(Abbr. PF). A parameter defining the range of temperatures extant in the exit gas flow from a gas turbine combustor. $PF = \frac{T_{MAX\ EX} - T_{AVG\ EX}}{T_{AVG\ EX} - T_{AVG\ INLET}}$ <p>All temperatures are absolute.</p>
PF	See PATTERN FACTOR.
PITCH	The circumferential distance between two adjacent airfoils in an axial turbomachinery annulus. It is equal to 2 times pi times the radius from the annulus centerline to the subject airfoil divided by the number of airfoils in the annulus.
POWER TURBINE	(Abbr. PT). A turbine stage in a gas turbine whose work output is used solely to drive the load with none being used by the compressor. In a 2-shaft gas turbine, the power turbine is fixed to a separate shaft from that of the GP turbine. It is normally -- but not required to be -- the last, or lowest pressure, turbine stage.
PR	Pressure Ratio. For a compressor or compressor stage, the ratio of outlet to inlet total pressure. For a turbine or turbine stage, the ratio of inlet to outlet total pressure.
PRIS	Particle Rejection Impact Separator. A device located between the primary and secondary zones of a rich-lean coal fired combustor which removes particulates from the gas stream.
PROFILE	The distribution of gas temperature within a single radial plane of a gas turbine combustor exit or other downstream annulus. The temperature profile represents the environment experienced by a turbine blade as it rotates through a temperature <u>pattern</u> thus averaging out circumferential variations. A "Profile Factor" can be defined using the Pattern Factor equation but confining the temperature data to a single radial plane.

PSR	Primary Surface Recuperator. Solar's proprietary design in which all heat exchange surfaces are "primary," i.e., transferring heat in their transverse direction. (A "secondary" surface, such as a fin, transfers heat in a lateral direction).
PT	See POWER TURBINE.
RECUPERATOR	A heat exchanger placed in the exhaust gas stream of a gas turbine in order to extract heat from the exhaust gases and return it to the cycle by heating the combustor inlet air. The term "recuperator" designates a static device as opposed to a "regenerator" which is a periodic device.
SHROUDED	A type of turbine blade design. A "shrouded" turbine features a short circumferential shroud segment attached to each blade tip. Together, these segments make up a full circular, rotating shroud. The outside surface of this shroud normally carries one or more circumferential seal elements. Conversely, a "non-shrouded" turbine is characterized by open blade tips rotating in close proximity to a stationary shroud.
SPOOL	The compression and expansion element of a gas producer, consisting of a compressor, a turbine and associated shafting. The turbine is designed so as to provide the amount of work required for the compression process plus any mechanical losses. A gas producer includes one or more spools and will be designated as 1-spool, 2-spool, etc.
SR	Stress Rupture. Stress rupture is the failure mechanism where gross rupture occurs due to time dependent deformation caused by steady stress at elevated temperatures. The mechanism at intermediate stages of deformation is called creep and is measured in terms of plastic strain. Stress rupture is usually associated with the Larson-Miller Parameter (LMP) which correlates the relationship between service life in hours and service temperature at a constant stress level. For a given material, Larson-Miller Parameter curves are developed which depict the LMP as a function of steady stress level.
STAGE WORK FACTOR	The ratio of work actually performed by a flow of cooling air within a turbine stage, to that possible if the flow were added to the hot gas stream.
STALL	A condition arising in one or more stages of an axial compressor in which the design pressure-velocity-rotational speed relationship is disrupted such that separation of the air flow from the blade contour occurs.
STANDBY	Use as a descriptor of a power generation station or unit that is only drawn upon when there is a demand. It currently is thought of as an emergency back-up unit in case the primary system fails, or when the primary unit cannot met the load demand. Sometimes called an "on-demand" system.

SURGE	The condition occurring when a compressor is operated below its minimum stable air flow rate for a given rotational speed. Surge is characterized by cyclic backflow of air in the flowpath accompanied by violent pressure fluctuations. During surge all stages of the compressor will be in a stall condition.
TBC	Thermal Barrier Coating. A thermally insulating ceramic coating (ZrO_2) applied to superalloy combustor and airfoil components. The insulative properties of the coating allow increased turbine inlet temperatures while maintaining acceptable metal substrate temperatures.
T&D	Transmission and Distribution. The system used to get the power (or fuel) from the point of origin to the consumer.
TRIT	Turbine Rotor Inlet Temperature. The average gas temperature entering the rotor annulus of the first turbine stage in a gas turbine. It is less than the average combustor exit temperature due to the mixing of spent cooling air flow from upstream sources, primarily the first stage nozzle vanes.
TSSC	Two-Stage Slagging Combustor.
ZWEIFEL COEFFICIENT	A measure of turbine blade loading based on pitch/chord ratio and gas turning angle as independent variables.