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**Advanced Turbine Systems Program
Conceptual Design and Product Development**

**Quarterly Report
February - April 1994**

G. J. Benjamin

June 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

By
Solar Turbines, Incorporated
San Diego, California

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9250 Sky Park Court
San Diego, California 92123**

June 1994

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INTRODUCTION

This Quarterly Technical Progress Report covers the period February 1, 1994 through April 30, 1994, for Phase II of the Advanced Turbine Systems (ATS) Program by Solar Turbines Incorporated, under DOE Contract No. DE-AC21-93MC30246. The objective of this program is to provide the conceptual design and product development plan for an industrial gas turbine system to operate at a thermal efficiency of 50 percent (ATS50) with efficiency enhancements to be added as they become technically viable and acceptable in the marketplace.

During this quarter, Solar's Engine Design Team has further refined both the 1-spool and 2-spool cycle concepts previously identified, to quantify the sensitivity of each to key component efficiencies, cooling air usage and origin, and the location of compressor surge lines. The process of refining the analysis included more detailed component work such as compressor and turbine design. As part of this effort, different speed trade-offs for the low-pressure (LP) and high-pressure (HP) compressor in the 1-spool configuration were examined to find the one yielding the best overall compressor efficiency.

High temperature and creep testing work for the recuperator candidate materials continued. Data from these tests have been used as guidelines for the cycle selection. Creep, yield and proportional limit data have been generated for foil thicknesses between 0.0030 and 0.0050 for Type 347 SS, Inconel 625 and Haynes Alloy 230 materials, and will continue to be generated to ensure the ideal and most appropriate material is selected. The recuperator characterization work is on schedule.

Combustor design work completed this quarter included preliminary layout of a multi-can annular combustor which was integrated into the main engine layout. Testing using the subscale catalytic combustion rig at Solar continued during the quarter. During this test, NOx emissions below 5 ppmv were measured under laboratory conditions.

Integration of the engine concept designs into the full power plant system designs commenced this quarter. The Engine Design and Power Systems Teams shared common databases to achieve the desired level of integration.

1.0 CYCLE AND COOLING WORK

1.1 SENSITIVITY OF CYCLE EFFICIENCY TO COMPONENT CHANGES

The two primary system arrangements (conventional 1-spool and 2-spool gas producers) were evaluated for sensitivity to various cycle parameters (component efficiencies, pressure losses, etc.) governing a practical intercooled, recuperated (ICR) cycle. The results of this study are intended to highlight the areas where performance is most critical as well as to show where improvements from the technology baselines can yield the maximum efficiency improvement. This information will influence later design tradeoffs.

1.1.1 Assumptions and Methodology

The initial assumptions used for previous cycle work still hold. (However, as the cycle, aerodynamic, and heat transfer analyses converge, the input to the final cycle analysis changes slightly.) Two assumptions were changed in this quarter that have an effect on the character of the cycle parameter studies. The changes made were to:

1. the cooling assumptions, and
2. the assumed character of the variation of turbine rotor inlet temperature (TRIT)- and pressure ratio (PR)-dependent variables.

The change to the cooling assumptions was a refinement to the preliminary cooling design model that increased the dependence of the cooling level on the turbine stage pressure ratio. The change to the assumed variation of the TRIT- and PR-dependent variables was to assume a non-linear (second order curve fit) variation, rather than the linear variation previously used. The effects of these changes are discussed below.

At this more refined level of cycle study, more limits must be applied to account for realistic performance and durability of engine components and subsystems. The pressure ratio was limited to 20:1 and below primarily because it would be unrealistic to assume that premixing in the combustion system without autoignition at pressure ratios above 20:1 and at temperatures around 1200°F is achievable in a cost-effective manner.

1.1.2 Discussion

1.1.2.1 Level of Thermal Efficiency

Figures 1 through 3 show overall thermal efficiency versus overall pressure ratio with varying TRIT for different levels of total hot section cooling (taken from HPC discharge) for a conventional 2-spool GP arrangement and Figures 4 through 6 are plots of overall efficiency versus pressure ratio for a 1-spool GP arrangement. In the case of the 2-spool arrangement, compared with the efficiency curves previously generated, the absolute thermal efficiency values are lower than before. This is due primarily to the revised cooling level estimates. Comparing the 2-spool and 1-spool arrangements on absolute thermal efficiency, the 1-spool is more efficient at the same conditions, by amounts varying with pressure ratio. This difference is due mostly to the reduced total hot section cooling level used in the 1-spool, because only a single turbine stage is used to drive both compressors.

For both configurations, the cooling levels, et cetera, were calculated using the same durability criteria and material assumptions. These are:

- 30,000 hours minimum life at design point,
- single crystal blade materials,
- ceramic turbine nozzles, and
- 1500°F rim temperature capability turbine disks.

The first two of these assumptions are clearly conservative for a program such as ATS. However, the assumptions are justified on the grounds that the levels of technology assumed are either at hand or will be available in the near term, and thus performance estimates can be readily made for preliminary design purposes. The technology growth will lie in reducing the total cooling level, as well as improving component efficiencies, to meet the thermal efficiency goal (see section 1.1.2.2 below).

1.1.2.2 Character of Thermal Efficiency Curves

Figures 1 through 3 show single thermal efficiency optima in contrast to those shown in previous curves. This is due to the revised cooling model which tends to show an adverse effect on thermal efficiency of the lower pressure ratio cycles. The increase in thermal efficiency with pressure ratio

Figure 1. Solar GFATS 2-Spool Cycle Baseline Cooling

Figure 2. Solar GFATS 2-Spool Cycle 90% Baseline Cooling

Figure 3. Solar GFATS 2-Spool Cycle, 80% Baseline Cooling

Figure 4. Solar GFATS 1-Spool Cycle, Baseline Cooling

Figure 5. Solar GFATS 1-Spool Cycle, 90% Baseline Cooling

Figure 6. Solar GFATS 1-Spool Cycle, 80% Baseline Cooling

is reduced from that previously shown. This is primarily due to the non-linear curve fit used to express the variation of the dependent cycle parameters. The non-linear curve fit is believed to be a truer representation of reality than the linear fit previously used.

1.1.2.3 Effect of Total Cooling Level

As expected, reduction of the overall hot section cooling level has a pronounced effect on the engine thermal efficiency level of both the 2-spool (Figures 1, 2, and 3) and the 1-spool (Figures 4, 5, and 6), with the sensitivity level varying between 0.04 and 0.06 points/percent total cooling. In the graphs shown, only the 1-spool arrangement appears capable of achieving the goal of 50 percent thermal efficiency, simply through cooling reductions to what is deemed to be a level achievable by 2000 AD (Figure 3). However, gains in other areas could allow the 2-spool arrangement to achieve 50 percent thermal efficiency, see below.

Reduction of overall hot section cooling level does not have any pronounced effect on the character of the efficiency versus pressure ratio curves.

1.1.2.4 Sensitivity Studies

Sensitivity studies were run to identify which component performances are most critical to the engine thermal efficiency, and which ones will give the highest efficiency payback for improvement. Table 1 shows the change in the 2-spool engine thermal efficiency for a given (favorable) change in component performance. The components that were chosen for study were the compressors, intercooler, recuperator, combustor, turbines, and ducting. Table 2 is the equivalent table for the 1-spool arrangement. Where there was a change in sensitivity to component performance with change in pressure ratio or turbine rotor inlet temperature, the sensitivities are shown separately. These tables show that both the 2-spool and the 1-spool arrangements have very similar efficiency sensitivities to the same component variations, with sensitivity variation more dependent upon pressure ratio variation than engine configuration. Although the cycle is sensitive enough to all of the components considered for meaningful gains to be made on each one, it is clear from the tables that the ICR cycle is most sensitive to the recuperator effectiveness and power turbine efficiency.

1.1.2.5 Effect of Cooling Flow Bleed Location on Cycle Efficiency

A study was made to determine the effects on the engine performance of taking the high-pressure cooling flow from aft of the recuperator cold side instead of from the HPC discharge, and the low-pressure cooling flow for the last two power turbine stages from the LPC discharge.

The increased flow required for cooling of hot section airfoils using the hotter air (recuperator cold side discharge) was calculated. This would require that the blade cooling scheme change from a predominantly convection cooled arrangement, with shower head cooling on the leading edge of the hotter turbine blades, to complete film cooling. Also, disk cooling and buffering flows were recalculated. This was done using a mixture of air from HPC discharge and recuperator discharge in order to obtain a flow temperature of 850°F by balancing the mass flows of the mixed air. The effect of the change in cooling air bleed location on flow temperatures at various cooled locations is shown along with total cooling air bleed used in Table 3. Cycle performance was calculated for both the 1 and 2-spool engines using the cooling levels calculated.

Table 4 compares the design point thermal efficiencies of the 1- and 2-spool cycles with the baseline efficiencies. The 2-spool engine accrues a 2.09 points increase in thermal efficiency by changing from all HPC discharge cooling bleed to a mixture of HPC and recuperator discharge bleed and the

Table 1. ATS Conventional 2-Spool - Sensitivity of Thermal Efficiency

Compressor Polytropic Efficiency Change (Δ) - %				
Cycle Parameters (PR, Firing Temp, Bleed Factor)	LPC		HPC	
	$\Delta=(-0.50)$	$\Delta=(0.50)$	$\Delta=(-0.50)$	$\Delta=(0.50)$
12:1, 2450°F, 1.0	46.74	46.92	46.71	46.95
	-0.09	0.09	-0.12	0.12
12:1, 2450°F, 0.9	47.39	47.57	47.36	47.60
	-0.09	0.09	-0.12	0.12
12:1, 2450°F, 0.8	48.04	48.22	48.01	48.25
	-0.09	0.09	-0.12	0.12
16:1, 2500°F, 1.0	47.79	48.00	47.77	48.02
	-0.10	0.10	-0.13	0.13
16:1, 2500°F, 0.9	48.38	48.59	48.36	48.61
	-0.10	0.10	-0.13	0.13
16:1, 2500°F, 0.8	48.98	49.18	48.95	49.20
	-0.10	0.10	-0.13	0.13
20:1, 2500°F, 1.0	48.01	48.25	47.99	48.27
	-0.12	0.12	-0.14	0.14
20:1, 2500°F, 0.9	48.55	48.78	48.53	48.81
	-0.12	0.12	-0.14	0.14
20:1, 2500°F, 0.8	49.09	49.32	49.07	49.34
	-0.12	0.12	-0.14	0.14

Intercooler Effectiveness Change (Δ) - %						
Cycle Parameters (PR, Firing Temp, Bleed Factor)	$\Delta=-20$ (65)	$\Delta=-10$ (75)	$\Delta=-5$ (80)	$\Delta=-1$ (84)	$\Delta=1$ (86)	$\Delta=5$ (90)
12:1, 2450°F, 1.0	45.81	46.32	46.58	46.78	46.88	47.08
	-1.02	-0.51	-0.25	-0.05	0.05	0.25
12:1, 2450°F, 0.9	46.45	46.97	47.22	47.43	47.53	47.73
	-1.03	-0.51	-0.26	-0.05	0.05	0.26
12:1, 2450°F, 0.8	47.09	47.61	47.87	48.08	48.18	48.39
	-1.04	-0.52	-0.26	-0.05	0.05	0.26
16:1, 2500°F, 1.0	46.61	47.26	47.58	47.86	47.99	48.21
	-1.29	-0.64	-0.32	-0.03	0.10	0.32
16:1, 2500°F, 0.9	47.19	47.84	48.17	48.42	48.55	48.81
	-1.30	-0.64	-0.32	-0.06	0.06	0.32
16:1, 2500°F, 0.8	47.77	48.43	48.75	49.01	49.14	49.40
	-1.31	-0.65	-0.32	-0.06	0.06	0.32
20:1, 2500°F, 1.0	46.56	47.35	47.74	48.06	48.21	48.52
	-1.57	-0.78	-0.39	-0.08	0.08	0.39
20:1, 2500°F, 0.9	47.09	47.88	48.28	48.59	48.75	49.06
	-1.58	-0.79	-0.39	-0.08	0.08	0.39
20:1, 2500°F, 0.8	47.61	48.41	48.81	49.12	49.28	49.60
	-1.59	-0.79	-0.39	-0.08	0.08	0.39

Table 1. ATS Conventional 2-Spool - Sensitivity of Thermal Efficiency (Contd)

Recuperator Effectiveness Change (Δ) - %						
Cycle Parameters (PR, Firing Temp, Bleed Factor)	$\Delta=-10$ (80)	$\Delta=-5$ (85)	$\Delta=-2$ (88)	$\Delta=-1$ (89)	$\Delta=1$ (91)	$\Delta=2$ (92)
12:1, 2450°F, 1.0	44.53	45.65	46.35	46.59	47.08	47.32
	-2.30	-1.18	-0.48	-0.24	0.24	0.49
12:1, 2450°F, 0.9	45.07	46.24	46.98	47.23	47.74	47.99
	-2.40	-1.24	-0.50	-0.25	0.26	0.52
12:1, 2450°F, 0.8	45.62	46.84	47.60	47.87	48.40	48.67
	-2.51	-1.29	-0.53	-0.26	0.27	0.54
16:1, 2500°F, 1.0	45.98	46.92	47.50	47.70	48.10	48.30
	-1.91	-0.98	-0.40	-0.20	0.20	0.40
16:1, 2500°F, 0.9	46.49	47.47	48.07	48.28	48.70	48.91
	-1.99	-1.02	-0.41	-0.21	0.21	0.42
16:1, 2500°F, 0.8	47.00	48.01	48.65	48.86	49.30	49.52
	-2.08	-1.06	-0.43	-0.22	0.22	0.44
20:1, 2500°F, 1.0	46.53	47.31	47.80	47.97	48.30	48.47
	-1.61	-0.82	-0.33	-0.17	0.17	0.34
20:1, 2500°F, 0.9	47.00	47.82	48.32	48.50	48.84	49.02
	-1.67	-0.85	-0.35	-0.17	0.17	0.35
20:1, 2500°F, 0.8	47.46	48.32	48.84	49.02	49.39	49.57
	-1.74	-0.89	-0.36	-0.18	0.18	0.36

Burner Efficiency Change (Δ) - %		
Cycle Parameters (PR, Firing Temp, Bleed Factor)	$\Delta=-0.50$	$\Delta=0.50$
12:1, 2450°F, 1.0	46.59	47.07
	-0.24	0.24
12:1, 2450°F, 0.9	47.24	47.72
	-0.24	0.24
12:1, 2450°F, 0.8	47.88	48.38
	-0.25	0.25
16:1, 2500°F, 1.0	47.65	48.14
	-0.24	0.24
16:1, 2500°F, 0.9	48.24	48.24
	-0.25	-0.25
16:1, 2500°F, 0.8	48.83	49.33
	-0.25	0.25
20:1, 2500°F, 1.0	47.89	48.38
	-0.24	0.24
20:1, 2500°F, 0.9	48.42	48.92
	-0.25	0.25
20:1, 2500°F, 0.8	48.95	49.45
	-0.25	0.25

Table 2. ATS Conventional 1-Spool - Sensitivity of Thermal Efficiency

Cycle Parameters (PR, Firing Temp, Bleed Factor)	Turbine Efficiency Change (Δ) - %					
	HPT		LPT		PT	
	$\Delta=-0.50$	$\Delta=0.50$	$\Delta=-0.50$	$\Delta=0.50$	$\Delta=-0.50$	$\Delta=0.50$
12:1, 2450°F, 1.0	46.78	46.88	46.80	46.87	46.66	47.00
	-0.05	0.05	-0.03	0.03	-0.17	0.17
12:1, 2450°F, 0.9	47.43	47.53	47.45	47.51	47.31	47.65
	-0.05	0.05	-0.03	0.03	-0.17	0.17
12:1, 2450°F, 0.8	48.08	48.18	48.10	48.16	47.96	48.30
	-0.05	0.05	-0.03	0.03	-0.17	0.17
16:1, 2500°F, 1.0	47.84	47.95	47.86	47.93	47.72	48.07
	-0.05	0.05	-0.04	0.04	-0.17	0.17
16:1, 2500°F, 0.9	48.43	48.54	48.45	48.52	48.31	48.66
	-0.05	0.05	-0.04	0.04	-0.17	0.17
16:1, 2500°F, 0.8	49.03	49.13	49.04	49.11	48.91	49.25
	-0.05	0.05	-0.04	0.04	-0.17	0.17
20:1, 2500°F, 1.0	48.08	48.19	48.09	48.18	47.96	48.30
	-0.06	0.06	-0.05	0.04	-0.17	0.17
20:1, 2500°F, 0.9	48.61	48.72	48.62	48.71	48.50	48.84
	-0.06	0.05	-0.04	0.04	-0.17	0.17
20:1, 2500°F, 0.8	49.15	49.26	49.16	49.25	49.03	49.37
	-0.05	0.05	-0.04	0.04	-0.17	0.17

Pressure Loss Variation (Δ) - %		
Cycle Parameters (PR, Firing Temp, Bleed Factor)	$\Delta=-0.50$	$\Delta=0.50$
12:1, 2450°F, 1.0	46.92	46.75
	0.09	-0.09
12:1, 2450°F, 0.9	47.56	47.39
	0.09	-0.09
12:1, 2450°F, 0.8	48.21	48.05
	0.08	-0.08
16:1, 2500°F, 1.0	47.97	47.82
	0.08	-0.08
16:1, 2500°F, 0.9	48.56	48.41
	0.08	-0.08
16:1, 2500°F, 0.8	49.15	49.00
	0.08	-0.08
20:1, 2500°F, 1.0	48.20	48.06
	0.07	-0.07
20:1, 2500°F, 0.9	48.74	48.60
	0.07	-0.07
20:1, 2500°F, 0.8	49.27	49.13
	0.07	-0.07

Table 2. ATS Conventional 1-Spool - Sensitivity of Thermal Efficiency (Contd)

Compressor Polytopic Efficiency Change (Δ) - %				
Cycle Parameters (PR, Firing Temp, Bleed Factor)	LPC		HPC	
	$\Delta=-0.50$	$\Delta=0.50$	$\Delta=-0.50$	$\Delta=0.50$
12:1, 2500°F, 1.0	46.89	47.07	46.86	47.10
	-0.09	0.09	-0.12	0.12
12:1, 2500°F, 0.9	47.49	47.68	47.47	47.70
	-0.10	0.09	-0.12	0.12
12:1, 2500°F, 0.8	48.11	48.30	48.08	48.32
	-0.10	0.09	-0.12	0.12
16:1, 2500°F, 1.0	47.53	47.75	47.51	47.78
	-0.11	0.11	-0.14	0.13
16:1, 2500°F, 0.9	48.06	48.28	48.04	48.30
	-0.11	0.11	-0.13	0.13
16:1, 2500°F, 0.8	48.61	48.83	48.59	48.85
	-0.11	0.11	-0.13	0.13
20:1, 2550°F, 1.0	47.00	47.23	46.98	47.26
	-0.12	0.11	-0.14	0.14
20:1, 2550°F, 0.9	47.61	47.84	47.58	47.86
	-0.12	0.11	-0.14	0.14
20:1, 2550°F, 0.8	48.21	48.43	48.18	48.46
	-0.11	0.11	-0.14	0.14

Intercooler Effectiveness Change (Δ) - %						
Cycle Parameters (PR, Firing Temp, Bleed Factor)	$\Delta=-20$ (65)	$\Delta=-10$ (75)	$\Delta=-5$ (80)	$\Delta=-1$ (84)	$\Delta=1$ (86)	$\Delta=5$ (90)
12:1, 2500°F, 1.0	45.91	46.45	46.72	46.93	47.03	47.25
	-1.07	-0.53	-0.27	-0.05	0.05	0.26
12:1, 2500°F, 0.9	46.51	47.05	47.32	47.53	47.64	47.85
	-1.08	-0.54	-0.27	-0.05	0.05	0.27
12:1, 2500°F, 0.8	47.12	47.66	47.93	48.15	48.26	48.47
	-1.09	-0.54	-0.27	-0.05	0.05	0.27
16:1, 2500°F, 1.0	46.24	46.95	47.30	47.57	47.71	47.99
	-1.40	-0.69	-0.35	-0.07	0.07	0.35
16:1, 2500°F, 0.9	46.77	47.48	47.83	48.11	48.24	48.52
	-1.40	-0.70	-0.35	-0.07	0.07	0.35
16:1, 2500°F, 0.8	47.31	48.02	48.37	48.65	48.79	49.07
	-1.41	-0.70	-0.35	-0.07	0.07	0.35
20:1, 2550°F, 1.0	45.51	46.32	46.72	47.04	47.20	47.52
	-1.61	-0.80	-0.40	-0.08	0.08	0.40
20:1, 2550°F, 0.9	46.11	46.92	47.32	47.64	47.80	48.12
	-1.61	-0.80	-0.40	-0.08	0.08	0.40
20:1, 2550°F, 0.8	46.70	47.52	47.92	48.24	48.40	48.72
	-1.62	-0.80	-0.40	-0.08	0.08	0.40

Table 2. ATS Conventional 1-Spool - Sensitivity of Thermal Efficiency (Cont'd)

Recuperator Effectiveness Change (Δ) - %						
Cycle Parameters (PR, Firing Temp, Bleed Factor)	-10 (80)	-5 (85)	-2 (88)	-1 (89)	1 (91)	2 (92)
12:1, 2500°F, 1.0	44.43	45.67	46.45	46.71	47.26	47.53
	-2.55	-1.32	-0.54	-0.27	0.27	0.55
12:1, 2500°F, 0.9	44.92	46.21	47.03	47.30	47.87	48.16
	-2.67	-1.38	-0.56	-0.28	0.29	0.58
12:1, 2500°F, 0.8	45.44	46.77	47.62	47.91	48.50	48.80
	-2.77	-1.43	-0.58	-0.29	0.30	0.60
16:1, 2500°F, 1.0	45.54	46.56	47.20	47.42	47.86	48.09
	-2.10	-1.08	-0.44	-0.22	0.22	0.45
16:1, 2500°F, 0.9	45.99	47.05	47.72	47.95	48.41	48.64
	-2.19	-1.12	-0.46	-0.23	0.23	0.47
16:1, 2500°F, 0.8	46.46	47.56	48.25	48.48	48.96	49.20
	-2.26	-1.16	-0.47	-0.24	0.24	0.48
20:1, 2550°F, 1.0	45.42	46.25	46.77	46.94	47.30	47.48
	-1.70	-0.87	-0.35	-0.18	0.18	0.36
20:1, 2550°F, 0.9	45.96	46.82	47.36	47.54	47.91	48.09
	-1.77	-0.90	-0.37	-0.18	0.19	0.37
20:1, 2550°F, 0.8	46.49	47.38	47.94	48.13	48.51	48.71
	-1.84	-0.94	-0.38	-0.19	0.19	0.39

Burner Efficiency Change (Δ) - %		
Cycle Parameters (PR, Firing Temp, Bleed Factor)	$\Delta=-0.50$ (8.5)	$\Delta=0.50$ (99.5)
12:1, 2500°F, 1.0	46.74	47.22
	-0.24	0.24
12:1, 2500°F, 0.9	47.34	47.83
	-0.25	0.25
12:1, 2500°F, 0.8	47.95	48.45
	-0.25	0.25
16:1, 2500°F, 1.0	47.40	47.89
	-0.24	0.24
16:1, 2500°F, 0.9	47.93	47.93
	-0.25	-0.25
16:1, 2500°F, 0.8	48.47	48.97
	-0.25	0.25
20:1, 2550°F, 1.0	46.88	47.36
	-0.24	0.24
20:1, 2550°F, 0.9	47.48	47.96
	-0.24	0.24
20:1, 2550°F, 0.8	48.08	48.57
	-0.25	0.25

Table 2. ATS Conventional 1-Spool - Sensitivity of Thermal Efficiency (Cont'd)

Cycle Parameters (PR, Firing Temp, Bleed Factor)	Turbine Efficiency Change (Δ) - %			
	HPT		PT	
	-0.50	0.50	-0.50	0.50
12:1, 2500°F, 1.0	46.88	47.08	46.81	47.16
	-0.10	0.10	-0.17	0.17
12:1, 2500°F, 0.9	47.49	47.68	47.41	47.76
	-0.10	0.10	-0.17	0.17
12:1, 2500°F, 0.8	48.11	48.30	48.03	48.38
	-0.09	0.09	-0.17	0.17
16:1, 2500°F, 1.0	47.53	47.75	47.47	47.81
	-0.11	0.11	-0.17	0.17
16:1, 2500°F, 0.9	48.07	48.28	48.00	48.35
	-0.10	0.10	-0.17	0.17
16:1, 2500°F, 0.8	48.62	48.82	48.55	48.89
	-0.10	0.10	-0.17	0.17
20:1, 2550°F, 1.0	47.01	47.23	46.94	47.29
	-0.11	0.11	-0.18	0.17
20:1, 2550°F, 0.9	47.61	47.83	47.55	47.90
	-0.11	0.11	-0.17	0.17
20:1, 2550°F, 0.8	48.21	48.43	48.15	48.50
	-0.11	0.11	-0.17	0.17

Total Pressure Loss Change (Δ) - %		
Cycle Parameters (PR, Firing Temp, Bleed Factor)	$\Delta=-0.50$	$\Delta=0.50$
12:1, 2500°F, 1.0	47.07	46.90
	0.09	-0.09
12:1, 2500°F, 0.9	47.67	47.50
	0.08	-0.09
12:1, 2500°F, 0.8	48.29	48.12
	0.08	-0.08
16:1, 2500°F, 1.0	47.72	47.56
	0.08	-0.08
16:1, 2500°F, 0.9	48.25	48.10
	0.08	-0.08
16:1, 2500°F, 0.8	48.79	48.64
	0.08	-0.08
20:1, 2550°F, 1.0	47.19	47.05
	0.07	-0.07
20:1, 2550°F, 0.9	47.79	47.65
	0.07	-0.07
20:1, 2550°F, 0.8	48.39	48.25
	0.07	-0.07

Table 3. Cooling and Purge Flow Temperatures and Quantities

Cycle	Blade, Vane, Wall Flow Temperature (°F)	First Disk Cavity Purge Flow Temperature (°F)	Last 2 Cavity Purge Flow Temperatures (°F)	Total Hot Section Cooling (% total mass flow)
Baseline 2-Spool	452	452	452	14.91
2-Spool with Cooling from Recup. Discharge	1074	850	287	18.7
Baseline 1-Spool	450	450	450	10.64
1-Spool with Cooling from Recup. Discharge	1134	850	287	14.12

Table 4. Comparison of Design Point Thermal Efficiencies with Different Cooling Schemes

Cycle	Baseline (%)	With Recup. Discharge Cooling (%)
1-Spool	48.75	50.01
2-Spool	48.14	50.23

1-spool engine accrues a 1.26 point increase for the same change. The primary reason for the difference in efficiency change for the 1-spool versus the 2-spool is that the cooling flows in the 2-spool go through more stages and therefore more work is extracted from them. Figure 7 shows the dependency of each configuration's thermal efficiency on fractional cooling flow. The efficiency gained by changing the location for cooling air bleed is due to the high air mass flowing through the recuperator, which, in turn, lowers the turbine exhaust temperature downstream of the recuperator. Figure 7 also shows the sensitivity of the 1-spool configuration's thermal efficiency to reduction in the required temperature of the first-stage disk cooling air. This sensitivity is relatively low, at approximately 0.1 points/100°F, for changes of approximately 850 - 750°F.

Other issues must be addressed if the cooling air extraction locations covered here are to be properly evaluated as the primary sources for cooling air. The most important issues are the effect on thermal efficiency of the required turbine airfoil changes, the potential durability effects of cooler air from the LPC being used to purge the last two PT stages, and the adequacy of the available pressure at the recuperator discharge for purging first-stage disk cavities given the relatively low pressure drop expected across the catalytic combustor. A selection of cooling air bleed location will be made once these issues have been adequately reviewed.

Figure 7. Engine Thermal Efficiency Versus Total Hot Section Cooling Level

1.1.2.6 Effect of Position of Low-pressure Compressor Surge Line on Engine Performance

The studies of expected operational characteristics of ATS50 have so far been based upon assumed compressor maps reflecting current typical designs. Recent estimates of the position of LP compressor surge lines indicate that the surge margins may be notably less than so far considered. A study was done in order to evaluate the bleed air flow required to allow surge free operation at part load and to quantify the consequent thermal efficiency penalty.

For comparison purposes, the revised surge lines and working lines were superimposed on the original LP compressor maps. The new working lines were calculated by bleeding as much mass flow from the LP compressor discharge as was required to maintain surge margin. These maps are shown in Figures 8 and 9. The intercompressor bleed affects performance as shown in Figures 10 to 12. Figure 10 shows that the increase in bleed needed for the 1-spool causes only a small reduction in part load thermal efficiency, amounting to 0.2 - 0.3 points at 50 percent load. The use of bleed on the 2-spool reduces thermal efficiency by approximately one point at 50 percent load. These penalties seem relatively small because an increase in bleed causes a reduction in both efficiency and power for a given airflow and TRIT. Thus, the new point tends to fall back on the old curve. Figures 11 and 12 show how the other parameters are affected for the 1- and 2-spool

Figure 8. ATS Assumed Compressor Maps Showing New LP Surge and Operating Lines for 1-Spool Compressor

Figure 9. ATS Assumed Compressor Maps Showing New LP Surge and Operating Lines for 2-Spool Compressor

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Figure 12. ATS Performance Parameters With 2-Spool Compressor Using Variable Geometry Power Turbine Showing Effects of Inter-Compressor Bleed Needed to Prevent LP Surge Due to Lower LP Surge Line

machines, respectively. The relatively high intercompressor bleed needed at low loads for the 1-spool requires the PT nozzles to open in order to maintain rated PT exhaust temperatures to avoid overtemperature operation of the recuperator.

2.0 COMPONENT WORK

References to efficiencies are at design point, unless otherwise noted.

2.1 ENGINE COMPRESSORS

A single set of compressors was investigated for the 2-spool configuration. The low-pressure, or LP, and high-pressure, or HP, spool speeds were driven by modern turbine design considerations. Two sets of compressors were analyzed for the 1-spool; a high-speed configuration in which the rotational speed was set equal to the HP spool speed from the 2-spool, and a low-speed configuration in which the speed was set half way between the 2-spool LP and HP spool speeds.

For the purposes of this study only axial configurations were investigated so as to maximize efficiency. Solar's axial compressor meanline performance prediction program was used to predict the performance potential. In each case the minimum number of stages which would result in a 10-15 percent surge margin at the given design speed was selected. Higher efficiencies could probably be achieved with more stages; however, comparable results were required. The impact of the number of stages will be assessed in the future, when engine layouts are available.

2.1.1 2-Spool Configuration

The LP compressor has an airflow rate of 59.29 lb/sec with a pressure ratio of 3.29:1, an adiabatic efficiency of 90.39 percent (91.84 percent polytropic) at a rotational speed of 14,938 rpm. The flowpath of this compressor is shown in Figure 13. The number of blades and the blade chord lengths are shown on the figure for each blade row. The seven-stage high-pressure compressor has a corrected airflow rate of 19.38 lb/sec with a pressure ratio of 5.01:1, and a design point adiabatic efficiency of 89.94 percent (91.91 percent polytropic) at a rotational speed of 19,500 rpm. This rotational speed was driven by modern turbine design considerations, however it is also an acceptable speed for the compressor. The flowpath, along with the blade numbers and chords, are shown in Figure 14.

2.1.2 1-Spool Configuration

The high-speed 1-spool configuration uses the same rotational speed as the 2-spool HP spool, 19,500 rpm. Thus this configuration is good for both HP compressor and turbine performance. The LP compressor performance is compromised because it is rotating faster than required, and therefore has higher shock losses than necessary and lower efficiency. The airflow rate is 57.26 lb/sec at a pressure ratio of 3.24:1, and an adiabatic efficiency of 87.29 percent (89.19 percent polytropic) at the design point. Its flowpath and blading are shown in Figure 15. The HP compressor performance is just slightly different than that of the 2-spool due to slightly different inlet conditions. The corrected airflow rate of this compressor is 19.00 lb/sec at a pressure ratio of 5.09:1, a design point adiabatic efficiency of 89.32 percent (91.84 percent polytropic) and a rotational speed of 19,500 rpm. Its flowpath and blading are shown in Figure 16.

Figure 13. ATS LP Compressor 16:1, 2-Spool

Figure 14. ATS HP Compressor 16:1, 2-Spool

Figure 15. ATS LP Compressor 16:1, 1-Spool

Figure 16. ATS HP Compressor 16:1, 1-Spool

The low-speed 1-spool was analyzed to see if by setting the speed to half way between the optima for the two compressors, the performance of each could be compromised somewhat, rather than only that of the LP, with reduced overall impact. The LP compressor at 17,210 rpm has an adiabatic efficiency 1.7 points higher than the high-speed design. The HP compressor requires that another stage be added (eight versus seven), but actually achieves an adiabatic efficiency 1.1 points higher than the high-speed design. The resulting performance for the LP compressor are 3.24:1 pressure ratio, 89.06 percent adiabatic efficiency (90.70 percent polytropic), at 17,210 rpm. No further improvement in efficiency was achieved by additional variations in speed. The corresponding HP compressor quantities are 5.09:1 pressure ratio, 90.44 percent adiabatic efficiency (92.33 percent polytropic) at 17,210 rpm. Therefore, the low-speed 1-spool configuration was selected as the preferred option. The flowpaths and blading parameters are shown in Figures 17 and 18. The performance of all of these compressors is summarized in Table 5.

The number of stages will be reviewed as the configuration evolves. More stages would improve efficiency; however the effect of this on the engine rotordynamics, cost, et cetera must also be assessed.

Table 5. Compressor Performance Characteristics

COMPRESSOR	Speed - (rpm)	Corr. Flow- (lb/sec)	Pressure Ratio (:1)	Adiabatic Efficiency (%)	Polytropic Efficiency (%)	No. of stages
2-Spool LP	14938	59.29	3.29	90.39	91.8	4
2-Spool HP	19500	19.38	5.01	89.94	91.9	7
High-speed 1-Spool LP	19500	57.26	3.24	87.29	89.2	3
High-speed 1-Spool HP	19500	19	5.09	89.83	91.8	7
Low-speed 1-Spool LP	17210	57.26	3.24	89.06	90.7	3
Low-speed 1-Spool HP	17210	19	5.09	90.44	92.3	8

2.2 TURBINES

The turbines for the 1- and 2-spool machines were designed considering compressor performance, turbine performance, cooling flows, and duct losses. Design parameters are consistent between the two machines, where possible, allowing for realistic comparison. The turbine aerodynamic design reported here are based upon cooling air extraction from the LPC discharge.

Since cooling levels have a strong effect on cycle efficiency, the material assumptions leading to the cooling levels selected must be restated. The turbine blades were assumed to be manufactured from a single crystal superalloy, such as CMSX-4, coated with thermal barrier coatings (TBC's). Vanes and all other stationary hardware were assumed to be ceramic (Silicon Carbides or Silicon Nitrides). The ceramic hardware has low level cooling for the purpose of reducing thermal gradient-induced

Figure 17. ATS LP Compressor 16:1, 1-Spool Low Speed

Figure 18. ATS HP Compressor 16:1, 1-Spool Speed

stresses. The blade cooling levels were based on bulk metal temperatures of 1500°F. The blade and vane gas temperatures used for cooling represent peak values, which adds conservatism to the analysis. Lastly, the temperature of the cooling air was increased by 300°F from for the second cooled stage and 200°F for all following stages. The disc cooling was set at 60 percent of free disc pumping. The complete cooling circuit was provided to the performance model, allowing for varying individual cooling to account for changes such as more open clearances on seals and the effect of high rim seals. It will also allow for determining the effect on performance of each cooling level.

2.2.1 2-Spool Configuration

The 2-spool core turbine consists of single-stage HP and LP turbines with three-stage power turbines. The HP and LP turbines are shown in Figure 19 and the PT in Figure 20.

The HP turbine is a cooled single stage axial turbine. The design parameters are listed in Table 6. The speed of 19,500 rpm and work required were set by the HP compressor. The turbine is moderately loaded with a corrected work of 20 Btu/lbm. The relatively high speed of the turbine limits the blade height which in turn limits turbine efficiency. The uncooled total-to-total efficiency of the HP turbine is 86.2 percent, based on the assumption of a non-shrouded blade at a running clearance of .025 inches. The vane and blade counts of 36 and 56 are based on Zweifel coefficients of approximately 0.8 with the axial chords set by past design experience.

The LP turbine is also a cooled, single-stage axial turbine. The speed of 14,938 rpm is much lower than that of the HP turbine. Also the work level is considerably less with a corrected work of 13.4 Btu/lbm. The lower work level produces a more efficient turbine with an uncooled total-to-total efficiency of 87.3 percent. As with the HP turbine, the LPT radial running tip clearance is set at 0.025 inches. The design parameters are listed in Table 6.

The power turbine is an uncooled, three-stage axial turbine. The speed of 10,080 rpm is set by the exit annulus area required and the radial location of the core turbines. The work level of the power turbine stages is moderate with a total corrected work of 48.2 Btu/lbm or approximately 16.1 Btu/lbm per stage. The three-stage turbine has a total to diffuser static efficiency of 89.2 percent. The efficiency is based on a diffuser recovery of 0.5 and shrouded blades at a clearance of 0.025 inches.

Table 6. Turbine Design Parameters

	Pressure Ratio (T-T)	Corrected Speed (rpm)	Corrected Flow (lbm/sec)	Efficiency (T-T)	Corrected Work (Btu/lbm)
2-Spool					
HPT	2.04	8163.4	8.99	0.865	20
LPT	1.57	6868.1	18.21	0.876	13.5
PT	5.92	4890.3	27.84	0.892	48.3
1-Spool, High-speed					
HPT	2.88	8163.4	8.51	0.846	28
PT	5.96	4768.3	23.40	0.892	47.7
1-Spool, Low-speed					
HPT	2.89	7205.8	8.67	0.846	27.9
PT	5.97	4775.5	23.97	0.892	47.8

Figure 19. ATS50 2-Spool GP Turbines

2.2.2 1-Spool Configuration

The 1-spool core turbine was designed in two versions. These were the high and low-speed versions, as discussed for the engine compressors.

The high-speed 1-spool core turbine is shown in Figure 21. The three stage free power turbine is essentially the same as for the 2-spool machine. The HP turbine is a cooled, single-stage axial turbine. The HP turbine is highly loaded with a corrected work of 28.0 Btu/lbm. Because this is a high-work turbine, the efficiency is lower than the 2-spool at an uncooled total-to-total of 84.6 percent. As with the 2-spool this is a non-shrouded blade with a running clearance of 0.025 inches. Because the power turbine is nearly identical to that of the 2-spool core, the efficiency, shown in Table 6, is the same as the 2-spool PT efficiency of 89.2 percent.

The low-speed 1-spool core turbine is shown in Figure 22. Again the power turbine is the same as the previous power turbines. This cooled single stage axial turbine is also highly loaded. The only difference between the high-speed and low-speed turbines is the change in radius produced by the lower speed. Initially the turbine was moved radially to maintain the work coefficient of the high-

Figure 20. ATS50 Power Turbine

speed design. However, this substantially increased the cooling flow as disc pumping is a function of the hub radius to the 2.6 exponent. This also added a duct loss as the HP turbine was at a larger radius than the PT. Therefore the turbine radius was reduced to line up with the PT at the hub. The end result was the same efficiency as the high-speed turbine, or 84.6 percent.

The cooling splits for the three turbines are presented in Table 7. The 1-spool core turbines both have lower cooling levels than the 2-spool core. This is due to the 2-spool having more stages and therefore more disc cavities to cool and purge. It is also due to the lightly loaded stages of the 2-spool not dropping the temperature across a stage as much as the highly loaded stage of the 1-spool.

2.3 RECUPERATOR

Additional creep and oxidation data were obtained for type 347 SS and Haynes Alloy 230. Elevated temperature tensile testing began in February by Westmoreland Mechanical Testing & Research Inc.

Table 7. ATS50 Turbine Cooling Air Distribution

2-Spool

1-Spool Low Speed

Figure 21. ATS50 High Speed 1-Spool GP Turbine

in Youngstown, Pennsylvania, to determine the needed data. Westmoreland has experience in working with thin foil testing. Material was tested at 1000°F, 1100°F, 1200°F, 1300°F, and 1400°F. Three samples of each material were tested at each temperature. These data will help predict life for a primary surface recuperator under ATS conditions.

Due to the differences in behavior of thin foils versus structural members, material test data for the materials which may be used for the foil in the PSR must be generated. There are two regimes of interest for recuperator foil materials; one which corresponds to higher pressure ratio operation where the likely failure mode is high-temperature yielding, and one which corresponds to lower pressure ratio operation in which the likely failure mode is high-temperature creep. High temperature yield testing for Type 347 stainless steel has been completed at Westmoreland Mechanical Testing Laboratory. The two percent yield and the proportional limit were obtained for a temperature range of 1000-to-1400°F. The results are summarized in Table 8.

Samples of selected materials were tested at 1000-to-1500°F at constant stress levels up to 25 ksi. The results of this test are used to ensure the material is creeping rather than yielding under the design conditions. These results are also used to verify the accuracy of existing theoretical creep

Table 8. Yield Test Results for Type 347 Stainless Steel

Temp (°F)	Load Direction	Proportional Limit (psi)	2% Yield (psi)	Ultimate strength (psi)
1000	Long	18900	25500	63100
1000	Trans	25000	30000	64800
1000	Unknown	26100	29800	66800
1050	Long	20500	29100	64200
1050	Unknown	20900	25700	62400
1050	Trans	20300	25200	57600
1100	Long	19500	28400	60500
1100	Trans	26000	28000	57600
1100	Unknown	24000	26500	62400
1200	Long	19400	29100	60500
1200	Trans	20800	25600	57300
1200	Unknown	20100	26700	59100
1300	Long	19300	23100	42500
1300	Trans	17600	22100	45400
1300	Unknown	18100	22900	42200
1400	Long	17400	20800	29400
1400	Trans	14200	20800	31800
1400	Unknown	19500	24100	33800

Thin foil creep tensile testing is in progress. Currently, a number of tests are in progress:

347 SS, T = 1300°F, stress = 11 ksi, longitudinal direction, thickness = 0.003-in

347 SS, T = 1300°F, stress = 11 ksi, transverse direction, thickness = 0.003-in

347 SS, T = 1300°F, stress = 9 ksi, longitudinal direction, thickness = 0.003-in

347 SS, T = 1300°F, stress = 9 ksi, transverse direction, thickness = 0.003-in

The above data were used to provide cycle selection guidelines to the ATS performance team.

Figure 23. Design Curves for Solar PSR Foil at 0.0045 Inch Thickness

Figure 24. Design Curves for PSR Foil at 0.005 Inch Thickness

Table 9. Thin Foil Creep Testing for Type 347 Stainless Steel at 5 Percent Creep

Temperature (°F)	Stress (ksi)	Foil Thickness (Inch)	Longitudinal Rupture (hours)	Transverse rupture (hours)	Unknown Rupture (hours)
1250	17	0.0033	724	1033	---
1300	14	0.0032	488	1033	---
1300	17	0.0033	166	225	190
1200	17	0.0033	2270	1500	1209

FOIL CREEP TEST #17 & #18

Stress: 17 ksi Temperature: 1250 F Material: 347 SS

Figure 25. Thin Foil Creep Test at T = 1250°F and 17 ksi Stress Level

Figure 26. Thin Foil Creep Test at $T = 1300^\circ \text{F}$ and 14 ksi Stress Level

Figure 27. Yield Strength vs Temperature - Alloy 625 (0.003 In. Foil)

Performance testing for the high capacity recuperator, which will be used in the ATS recuperator, as well as f and j testing will be performed at the Caterpillar Technical Center. Preparation work for these tests is continuing. A hot-side exit mixer has been designed to diminish the temperature gradients.

The design of the new gas-out mixer was recently completed at Caterpillar with the assistance of Professor Moffit of Stanford University. It consists of a "top-hat" like structure with several small diameter orifices in it. This will be tested prior to the performance test to ensure that it does provide adequate gas mixing.

2.4 INTERCOOLER

Various methods for achieving a desirable intercooler effectiveness will be examined in this phase of the program. The intercooler system effectiveness that has been used for the cycle analysis to date is one that can be met using a cooling tower arrangement. Methods of improving the intercooler system performance will be evaluated. Additionally, methods to improve individual component efficiencies are currently being evaluated to increase the performance of an overall closed loop intercooler system that does not require a cooling tower.

Under certain operating conditions, water condensation will occur at the intercooler exit. This can cause problems to the first stage of the certain HPCs. A droplet coalescer is one possible solution. Operating the intercooler in a non-condensing mode is another possibility. Making the HPC more water-tolerant is yet another solution.

Water will condense downstream of the intercooler, depending upon the outlet temperature of the air leaving the intercooler and on the absolute water content of the air entering the engine (i.e., the absolute humidity). A plot of absolute humidity, in terms of pounds of water per pound of dry air, appears in Figure 28. This shows the absolute humidity that would create a water condensation situation as a function of intercooler exit temperature. Therefore, knowing the intercooler exit temperature, one can take the referenced humidity off the chart and check it against standard psychrometric data to determine the ambient temperature and relative humidity scenarios where condensation will occur.

Mist eliminators are being examined as potential droplet coalescers. These are low pressure drop devices, on the order of one inch of water, and are highly effective in coalescing water droplets, even down to the 10 micron droplet size. However, such a device may be larger than desired. Calculations are being performed to determine optimum size, performance and cost of a mist eliminator to determine whether this device should be considered further. Solar is currently working with two suppliers of mist elimination equipment to complete this phase of the program.

Alternatively, because condensing water will tend to evaporate once the air containing the water undergoes further compression in the HPC, the only requirement may be to harden the first stage or two of the HPC. This could alleviate the condensation problem.

2.5 COMBUSTOR

Activities for the ATS combustor have included definition of the combustor development plan, preliminary design layouts and subscale testing of the catalytic combustor.

Figure 28. Yield Strength vs Temperature - T-347 SS (0.003 In. Foil)

The catalytic combustor has been selected as the primary design approach to meet ATS requirements. As a back-up lean premixed combustion will be developed for ATS conditions on a parallel track. Auto-thermal reforming technology is being developed for second generation ATS growth potential. Low emissions combustor technologies that were studied include: catalytic, lean-premixed, rich-lean, H_2 enrichment, and autothermal reforming. Catalytic combustion has been selected as the primary approach because of its laboratory-demonstrated capability to meet a NOx level of lower than 5 ppmv. In addition catalytic combustion has high potential to provide a flatter temperature profile and lower pattern factor than the other technologies. Development risks with this technology are durability of the catalyst and substrate, unproven operation over the entire engine load range and combustor back-end and transition duct life. Lean-premixed combustion has fewer risks in terms of life and operation within the ATS, but will not, in all likelihood, be able to meet NOx emissions below 8 ppm at all ambient conditions as production hardware.

A preliminary layout of the catalytic combustor for ATS applications is shown in Figure 29. The design philosophy is to pursue a multi-can annular combustor arrangement within the engine. Other layouts considered were a straight annular and a single can. The benefits of the multi-can approach are a producible catalyst size, symmetry in the premixer design to ensure circumferential temperature uniformity, and easier combustor accessibility for maintenance. The illustration shows a parallel flow arrangement where the catalyst bed is positioned around a lean-premixed injector. The combustor is started with the lean premixed injector and operated to approximately 50 percent engine load without any combustion within the catalyst bed. The combustor switches to low emissions catalytic combustion at 50 percent. Operation of the catalyst from 50-to-100 percent load will most likely require variable geometry control. Further design is in progress.

A critical piece of the catalytic combustor is the catalyst bed. Solar has been working with catalyst supplier Engelhard Corporation in system design and subscale test component development. Engelhard has been active in designing catalysts that will operate at ATS conditions. Design of the

Figure 29. Tensile Test Results - T-347 SS (0.003-0.004 In. Thick Foil)

overall system is being completed at Solar with input from Engelhard to ensure that the catalyst bed will operate with high efficiency. Testing of catalyst beds is being completed at Engelhard in a laboratory rig and at Solar within a more engine representative subscale combustor rig.

Testing at Solar in the subscale catalytic combustor has been on-going for the past two years. In these test Solar has demonstrated stable catalytic combustion at 13 atmospheres and 780°F inlet temperature with NOx emissions below 5 ppm. This work is being conducted with Engelhard-supplied catalyst beds. Testing is in progress to demonstrate catalytic combustion at ATS conditions of 1118°F combustor inlet and 15 atms pressure. Modifications to the test facility have been completed to enable testing at these conditions. Fuel profile measurements at the catalyst inlet plane have been measured to optimize the premixer design to within three percent peak-to-peak. Initial testing of a catalyst bed designed for ATS conditions are underway.

3.0 POWER PLANT INTEGRATION

Mechanical layout activity in the last quarter has concentrated on developing a concept that can be used to validate approaches to engine core flow to/from components such as the intercooler and recuperator, and to allow integration of the engine into the power systems package.

The 1-spool mechanical layout, a preliminary version of which is shown in Figure 30, is being used for flowpath design iterations to minimize the pressure losses incurred in transition from the annular LP compressor flowpath through the radial intercooler and back to the annular HP compressor flowpath and from the HP compressor to the recuperator and back into the combustor. Concurrently Solar's Systems and Packaging groups are using the same computer database to size support structures and situate driven and ancillary component to best advantage. These groups are also converting the two-dimensional layout into solid model databases to aid in visualization of the design concept as it evolves.

Figure 30. Operating Curves 347 SS (0.003 in. Thin Foil)

In parallel, a 2-spool layout is in work. This layout's first use will be in determining the rotordynamic characteristics of the engine. In parallel with the rotordynamic model development, the layout will be used for the same activities as the 1-spool layout.

4.0 FUTURE INVESTIGATIONS

Early in the next quarter more trade-off studies between the 1- and 2-spool ATS50 variants will be conducted including starting analysis and some more accurate engine component costing. The object of the trade-off studies will be to continually compare the two concepts as measured by their ability to meet the program goals with specific attention to the desirability of the final product to the customer (first cost versus life cycle cost).

APPENDIX 1

GLOSSARY

ATS	Advanced Turbine System. 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.
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.
CFATS	Coal-Fired Advanced Turbine System.
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.
GAS PHODUCER	(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.
GP	See GAS PRODUCER.
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).
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.
PATTERN FACTOR	(Abbr. PF). A parameter defining the range of temperatures extant in the exit gas flow from a gas turbine combustor. $PF = \frac{T_{MAX\ EX} - T_{AVG\ EX}}{T_{AVG\ EX} - T_{AVG\ INLET}}$
	All temperatures are absolute.
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.
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 fixed boundary 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.
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.
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.
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.
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.
ZWEIFEL COEFFICIENT	A measure of turbine blade loading based on pitch/chord ratio and gas turning angle as independent variables.

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