AIM

Association for Information and Image Management

1100 Wayne Avenue, Suite 1100
Silver Spring, Maryland 20910
301 587-8202

Centimeter

Inches

MANUFACTURED TO AIM STANDARDS
BY APPLIED IMAGE, INC.

O

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

Quarterly Report

K. W. Karstensen

February 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, Inc.
San Diego, California

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

DISCLAIMER

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

This report has been reproduced directly from the best available copy.

Available to DOE and DOE contractors from the Office of Scientific and Technical Information, 175 Oak Ridge Turnpike, Oak Ridge, TN 37831; prices available at (615) 576-8401.

Available to the public from the National Technical Information Service, U.S. Department of Commerce, 5285 Port Royal Road, Springfield, VA 22161; phone orders accepted at (703) 487-4650.

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

Quarterly Report

K. W. Karstensen

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

**For
U.S. Department of Energy
Office of Fossil Energy
Morgantown Energy Technology Center
P.O. Box 880
Morgantown, West Virginia 26507-0880**

**By
Solar Turbines, Inc.
9250 Sky Park Court
San Diego, California 92110**

February 1994

CONTENTS		
<u>Paragraph</u>		<u>Page</u>
	INTRODUCTION	1
1.0	CHARACTERISTICS OF THE ICR GAS TURBINE CYCLE	1
1.1	Alternate ICR Gas Turbine Configurations	1
1.2	Effects of Component Performance Upon ICR Gas Turbine Performance	5
2.0	OPERATIONAL FLEXIBILITY AND OFF-DESIGN PERFORMANCE	7
2.1	Design Point Parameters	10
2.2	Single Spool Compressor	11
2.3	Two Spool Compressor	11
2.4	Thermal Efficiency vs Load	11
2.5	Variation of Other Salient Parameters With Load	11
3.0	VARIATION OF MAXIMUM POWER WITH AMBIENT TEMPERATURE	11
4.0	PRELIMINARY PHYSICAL CONFIGURATION STUDIES	25
4.1	Concept Evaluation Criteria	25
4.2	Structural Analysis Method for Preliminary Flowpath Evaluation	25
5.0	FUTURE INVESTIGATIONS	25

FIGURES

<u>Figure</u>		<u>Page</u>
1	Recuperated Engine Performance	2
2	Intercooled Recuperated Engine Performance	3
3	1-Spool ICR	4
4	2-Spool ICR	5
5	Recuperator Hot-Side Pressure Drop - Effect on Efficiency	6
6	Intercooler Pressure Drop - Effect on Efficiency	6
7	Cycle Study. Effects of Overall Compressor Pressure Ratio and Work Split	8
8	Relationship of Turbine Cooling Air, TRIT, Thermal Efficiency and Specific Power	9
9	Assumed Compressor Maps Showing Operating Lines for 1-Spool System With Fixed geometry Power Turbine	12
10	Assumed Compressor Maps Showing Operating Line for 1-Spool System With Variable Geometry Power Turbine ~ P.T. Exit Temperature Constant	13
11	Assumed Compressor Maps Showing Operating Lines for 1-Spool System With Fixed Geometry Power Turbine and Intercooler Bleed to Prevent LP Surge	14
12	Assumed Compressor Maps Showing Operating Lines for 1-Spool System With Variable Geometry Power Turbine	15
13	Assumed Compressor Maps Showing Operating Lines for 2-Spool System With Fixed Geometry Power Turbine	16
14	Assumed Compressor Maps Showing Operating Lines for 2-Spool System With Variable Geometry Power Turbine ~ PT Exit Temperature Constant	17
15	Thermal Efficiency Versus Load	18
16	Performance Parameters. ATS With 1-Spool Compressor Using Inter-compressor Bleed and Fixed Geometry Power Turbine	19

FIGURES (Continued)

<u>Figure</u>		<u>Page</u>
17	Performance Parameters. ATS With 1-Spool Compressor Using Inter-compressor Bleed and Variable Geometry Power Turbine	20
18	Performance Parameters. ATS With 2-Spool Compressor Using Fixed Geometry Power Turbine	21
19	Performance Parameters. ATS With 2-Spool Compressor Using Variable Geometry Power Turbine	22
20	Estimated Maximum Performance vs. T1 ~ 2-Spool Compressor Fixed Turbine Geometry	23
21	Estimated Maximum Performance vs. T1 ~ 2-Spool Compressor Variable PT Geometry ~ Exit Temperature 1305°F, Constant	24
22	Turbine Blade and Disc Parametric Model	27

TABLES

<u>Table</u>		<u>Page</u>
1	Effects of Changes in Component Performance and TRIT at Design Point	10
2	Engine Concept Evaluation Criteria	26

INTRODUCTION

This Quarterly Technical Progress Report covers the period August 3 through October 31, 1993, 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 and developable to 60 percent.

During this period, Solar's ATS Engine Design Team reviewed the intercooled and recuperated (ICR) gas turbine concept defined in the Program proposal, validated certain assumptions associated therewith, and began the process of actualizing this concept in terms of achievable turbomachinery components. Given the probable use of a free power turbine arrangement, both 1-spool and 2-spool compressor arrangements were examined with both fixed and variable turbine geometry. Off-design performance, both part-load and full-load over a range of inlet air temperatures, was examined.

During this period certain simplifying assumptions were made regarding the amount of cycle air extracted for use in turbine cooling and the distribution of its return to the cycle. The exact influence of turbine cooling air extraction on cycle performance (thermal efficiency) will be highly dependent upon turbine airfoil material selection, its life/temperature capabilities, etc. Thus, cycle performance will be subject to some degree of change as the design progresses. Even now, improvements made to the cycle performance model will result in variation in presented results. As a general rule, later results will always supersede earlier results when there is an apparent conflict.

1.0 CHARACTERISTICS OF THE ICR GAS TURBINE CYCLE

The generic concept of an intercooled, recuperated, (ICR) engine proposed for Solar's ATS was based on the work performed during Phase 1. Thermodynamically the ICR cycle has excellent efficiency and power characteristics. Figure 1 shows a temperature-entropy diagram and the efficiency relationships of a recuperated gas turbine while Figure 2 shows the same characteristics for an ICR engine. Comparing the two figures provides some insight into the advantages of the ICR arrangement. The thermal efficiency characteristic of the ICR engine is higher than the recuperated approach but more importantly is "flatter" with a broad range of pressure ratios providing high efficiency. In addition the increased output work made available through intercooling is shown to be significant. This increase in the available work is due to the fact that an intercooled or heat extraction, compressor system requires less work for compression than an adiabatic, or uncooled, system.

1.1 ALTERNATE ICR GAS TURBINE CONFIGURATIONS

There are several versions of an ICR that are being considered for the ATS. The two most promising approaches are being evaluated. These two versions are: (1) a 1-spool with a free power turbine and (2) a 2-spool approach also with a free power turbine. The single spool system has to have some control system that prevents the low pressure section of the compressor from being driven into surge. Initially an interstage air bleed and a

Q_R = HEAT RECOVERED FROM EXHAUST (RECUPERATION)

Σ = RECUPERATOR EFFECTIVENESS = $(T_a - T_2) / (T_4 - T_2)$

$Q_1 - Q_2$ = REDUCTION TO WORK OUTPUT DUE TO RECUPERATOR FLOW RESTRICTION

RE93077M

Figure 1. Recuperated Engine Performance

RE9307

Q_R = HEAT RECOVERED FROM EXHAUST (RECUPERATION)

ΔQ_R = INCREASED RECUPERATOR DUE TO TEMPERATURE REDUCTION, $T_2 T_2'$

Q_1 = INCREASED WORK OUTPUT DUE TO DECREASED WORK OF COMPRESSION

RE83079M

modulating intercooler that can vary the temperature entering the high pressure compressor were considered as candidates for surge prevention. The modulating intercooler approach has now been eliminated from the consideration because the temperatures needed at the inlet of the high pressure section have to be below ambient to prevent surge. The use of bleed at the intercooler location will allow the low pressure compressor section to operate without surge. Both the 1- and 2-spool approaches will probably require variable power turbine nozzles for starting and safety reasons. The 2-spool approach is a near ideal arrangement for meeting the goals of the program although it will be more costly to produce than the single spool. This configuration has no stall problem and has a higher thermal efficiency over a wider turn-down range than the single spool system. Schematics of the 1-spool and the 2-spool system are provided as Figures 3 and 4. These schematics show the component indices and the thermodynamic state points for the two arrangements. The indices shown will be used throughout the program to refer to particular components.

RE93075M

Figure 3. 1-Spool ICR

RE93076M

Figure 4. 2-Spool ICR

1.2 EFFECTS OF COMPONENT PERFORMANCE UPON ICR GAS TURBINE PERFORMANCE

The ICR cycle is not strongly affected by the efficiency level of each of the two compressors. This is due to the recuperator which tends to "recycle" the thermal inefficiency of the compressors. In addition the effects of component pressure drop on the overall efficiency are not large as shown in Figures 5 and 6. Figure 5 is a correlation of overall thermal efficiency at the optimized conditions given in Table 1 as a function of the recuperator hot-side pressure loss. As shown the pressure loss has to lie between 3 and 4% for the system to meet the program goal of 50% thermal efficiency. At a 4% loss for the recuperator the intercooler characteristics (Figure 6) reveals that a pressure drop of less than 2% will be necessary. Based on past experience these pressure loss levels should be achievable in practice.

Figure 5. Recuperator Hot-Side Pressure Drop - Effect on Efficiency

Figure 6. Intercooler Pressure Drop - Effect on Efficiency

The recuperator configuration and its operating conditions are key determinants in defining ICR cycle parameters. At the optimum pressure ratio of 12:1 for the ICR engine the recuperator inlet temperature has to be 1300°F to provide a 50% thermal efficiency. This will require that superalloys such as the Inconel 600 series be used for the recuperator air cells. These superalloys are more expensive than the more commonly used 300 series stainless materials. To operate at 1180°F, the maximum allowable temperature for a 300 series recuperator, the pressure ratio has to be increased to 17.4:1. Although the 50% thermal efficiency can still be maintained by improvements in component efficiencies and reduced pressure losses there will always be a 1% drop in efficiency (0.4% decrement) between the 12 and the 17.4:1 pressure ratio cases, everything else being equal. The cost of this 1% increase in fuel flow for the higher pressure ratio condition will be balanced against the decreased cost of the recuperator materials in evaluating the best approach to meet ATS cost-of-electricity goals.

2.0 DESIGN POINT PARAMETERS

It was first required to obtain an approximate set of design point parameters. At this time the comprehensive studies of candidate components and turbine cooling requirements had not yet been undertaken, so some rather broad assumptions were necessary. These included :

- LP and HP compressor efficiencies of 86.5 percent and 84 percent respectively. (Assuming axial LP and radial HP).
- An overall compressor pressure ratio, including a 3 percent intercooler + ducting pressure loss, of 12:1, with an LP/HP work split of 0.45/0.55. (See Figure 1)
- Overall turbine efficiency of 89 percent.
- Intercooler and recuperator effectiveness of 90 percent.
- Intercooler sink temperature the same as the ambient temperature, 59°F.
- Turbine inlet temperature of 2450°F
- A simple turbine cooling model whereby 5 percent of the compressor discharge air is extracted and re-enters the cycle at the inlet to the power turbine.
- Pressure losses (not including that between the LP and HP compressors) of 11 percent.

Table 1 shows how engine performance is affected by component efficiency, effectiveness, etc., while Figure 7 illustrates how an overall compressor pressure ratio of 12:1 with an LP/HP work split of 0.45/0.55 provides optimum thermal efficiency for the 2450°F turbine inlet temperature. Again as a side issue, Figure 8 illustrates the nature of the study which will be needed in order to determine the optimum relationship between the cooling model and the turbine inlet temperature it will allow. In the example shown, the solid lines of constant thermal efficiency reflect the simple model described earlier of compressor discharge air reentering at the PT inlet. In reality, of course, the abscissa of such a picture would represent a series of complex parameters.

Cycle Study.
Effects of Overall Compressor Pressure Ratio and Work Split.

Figure 7. Cycle Study. Effects of Overall Compressor Pressure Ratio and Work Split

Figure 8. Relationship of Turbine Cooling Air*, TRIT, Thermal Efficiency and Specific Power

Table 1. Effects of Changes in Component Performance and TRIT at Design Point

	% HP	% Heat Rate
1 point increase in LP compressor efficiency	+0.425	-0.41
1 point increase in HP compressor efficiency	+0.425	-0.4
1 point increase in overall turbine efficiency	+2.0	-1.07
1 point increase in recuperator effectiveness	-0.4	-1.0
1 point increase in intercooler effectiveness	+0.17	-0.13
1 percent increase in pressure loss	-0.555	+0.32
50°F increase in TRIT	+3.1	-0.83

2.1 OPERATIONAL FLEXIBILITY AND OFF-DESIGN PERFORMANCE

The next step is to find the best arrangement of the compressor and turbine systems with respect to the operational flexibility and off-design performance of an advanced intercooled and recuperated gas turbine capable of a thermal efficiency of 50 percent and developable to 60 percent.

It is envisioned that such a machine will employ a gas producer system comprising a low pressure (LP) and a high pressure (HP) compressor and their drive turbine elements, followed by a free power turbine. Starting with a typical set of design parameters, approximate compressor and turbine maps were synthesized and off-design engine performance investigated for various possible arrangements. These included mounting the gas producer section on one shaft, (1-spool), or using separate shafts for the LP and HP sections, (2-spool). The use of a variable power turbine nozzle controlled to maintain rated power turbine exit temperature under normal steady state conditions at all loads and ambients was investigated. How this affects both part-load thermal efficiency and the attainable maximum performance at off-design ambients was determined.

It was concluded that :

- A 1-spool gas producer presents LP compressor surge problems at part-load. This could be avoided by the use of part-load bleed from between the compressors, but with a thermal efficiency penalty.
- A 2-spool gas producer is the best candidate since both LP and HP compressors maintain good part-load surge margins and allows the designer to optimize speeds for best efficiency but will add cost to the design.
- Although maintaining rated PT exhaust temperature with a variable PT nozzle for the intercooled cycle doesn't produce as much relative improvement in part-load thermal efficiency as it does for a non-intercooled recuperated engine, it is advantageous with respect to attainable power and thermal efficiency at ambient temperatures other than the 50°F design point. A variable PT nozzle would also be valuable for preventing PT overspeed in the event of a sudden drop-off in load, and possibly for minimizing emissions at part-load.

2.2 1-SPOOL COMPRESSOR

Figure 9 shows how the equilibrium operating lines appear relative to the assumed characteristics of the LP and HP compressors. While the HP compressor shows no problem, the LP operates dangerously close to its surge line as its speed falls towards 90 percent. Figure 10 shows that using variable PT nozzles to keep the PT exit temperature at rated level hardly affects the position of the compressor operating lines. However, the surge problem can be overcome by bleeding at the exit from the LP compressor (Figures 11 and 12).

2.3 2-SPOOL COMPRESSOR

With either a fixed or variable PT nozzle (the latter maintaining constant PT exhaust temperature) there should be no surge problem during steady state operation (Figures 13 and 14)

2.4 THERMAL EFFICIENCY VS LOAD

It was found that the part-load thermal efficiency curves were essentially identical for the 1- and 2-spool configurations except where the proximity of surge required the former to be bled. The comparisons shown on Figure 15 only shows the 1-spool curves when using bleed and it is seen that comparing the fixed PT geometry curves at say 50 percent load, the 1-spool's thermal efficiency is consequently some one point worse than that of the two spool. Also, since bleed requires a higher TRIT for a given load, the potential improvement offered by maintaining the PT exhaust temperature at the rated level is minimal for the 1-spool system. On the other hand, doing this with the 2-spool system shows a potential gain of about 1-1/2 points at 50 percent load -- assuming that PT efficiency remains the same for both fixed and variable designs.

2.5 VARIATION OF OTHER SALIENT PARAMETERS WITH LOAD

Parameters including shaft speeds, mass flow, TRIT, PT exhaust temperature, pressure ratio, and, for the 1-spool, inter-compressor bleed, are shown on Figures 16 through 19. Where a variable geometry PT is employed, the reduction in its flow capacity with falling load as needed to maintain its exhaust temperature at rated is also shown.

3.0 VARIATION OF MAXIMUM POWER WITH AMBIENT TEMPERATURE

It has been concluded from the foregoing that a 2-spool compressor system is preferred from an overall engine performance standpoint. Another important reason is, of course, the freedom to design each compressor to operate at its optimum speed. Thus only the maximum power curves for the two spool were considered. Figure 20 shows that if a fixed PT were used, the need to avoid excessive recuperator inlet temperature would severely limit the power at ambients above the design, 59°F. Also, at below 59°F, power is limited by LP speed, thus TRIT and therefore power falls below what would otherwise be attainable. When using a variable PT, however, the HP speed becomes the limiting parameter above 59°F which allows higher LP speed and TRIT and hence higher power than with the fixed nozzle (Figure 15). There is also some gain below 59°F relative to that of the fixed geometry PT because the TRIT is now on its limit -- even though the LP speed is below 100 percent and hence airflow is less than with the fixed geometry PT.

The variable geometry PT also offers higher full-load thermal efficiency levels than the fixed geometry version at both above and below 59°F (Figures 20 and 21).

PA93381M

Figure 9. Assumed Compressor Maps Showing Operating Lines for 1-Spool System With Fixed Geometry Power Turbine

Figure 10. Assumed Compressor Maps Showing Operating Line for 1-Spool System With Variable Geometry Power Turbine ~ P.T. Exit Temperature Constant

PA93364M

Figure 12. Assumed Compressor Maps Showing Operating Lines for 1-Spool System With Variable Geometry Power Turbine

Figure 13. Assumed Compressor Maps Showing Operating Lines for 2-Spool System With Fixed Geometry Power Turbine

PA93368M

Figure 14. Assumed Compressor Maps Showing Operating Lines for 2-Spool System With Variable Geometry Power Turbine ~ P.T. Exit Temperature Constant

Figure 15. Thermal Efficiency Versus Load

PA83367M

Performance Parameters. ATS with Single-Spool Compressor
Using Inter-Compressor Bleed and Fixed Geometry Power Turbine

PA83368M

Figure 16. Performance Parameters. ATS With 1-Spool Compressor Using
Inter-compressor Bleed and Fixed Geometry Power Turbine

Figure 17. Performance Parameters. ATS With 1-Spool Compressor Using Inter-compressor Bleed and Variable Geometry Power Turbines

Performance Parameters. ATS with Two-Spool Compressor.
Using Fixed Geometry Power Turbine.

Figure 18. Performance Parameters. ATS With 2-Spool Compressor Using Fixed Geometry Power Turbine

Performance Parameters. ATS with Two-Spool Compressor.
Using Variable Geometry Power Turbine. 1

Figure 19. Performance Parameters. ATS With 2-Spool Compressor Using Variable Geometry Power Turbine

Figure 20. Estimated Maximum Performance vs. T_1 ~ 2-Spool Compressor Fixed Turbine Geometry

PA93373M

**Figure 21. Estimated Maximum Performance vs. T_1 ~ 2-Spool Compressor
Variable PT Geometry ~ Exit Temperature 1305°F, Constant**

4.0 PRELIMINARY PHYSICAL CONFIGURATION STUDIES

Preliminary aerothermal studies led to the concept of cycles involving non-conventional placement of components. The cycle analysis code that had been used up to this point was discontinued in favor of one that would be more appropriate to the task. The new code was developed at Solar and allows a great deal of design flexibility in component placement.

The new computer code was used to review the predictions previously made by the code used up to this point. This was to provide a clear correlation between the codes as applied to the preliminary cycle points required for Solar's proposed GFATS to meet the program goals, before extending the use of the new computer code to other configurations that will be studied during the next quarter.

4.1 CONCEPT EVALUATION CRITERIA

As proposed configurations are generated, criteria for evaluating the configurations must be available in order for meaningful comparisons to be made. Discussions were held among Solar's dedicated ATS Team to determine how to derive from the program goals specific criteria that could be used to evaluate engine concepts once sufficient information is available about the concepts for the criteria to be applied. A preliminary list of the criteria that will be used to evaluate engine concepts, and explanation of their derivation, is shown in Table 2.

4.2 STRUCTURAL ANALYSIS METHOD FOR PRELIMINARY FLOWPATH EVALUATION

ComputerVision's DesignView is the tool chosen to do the preliminary structural analysis of flowpath geometry. With this tool, a parametric model of a blade and disk can be made, with the dimensions, material properties and other pertinent quantities defined as variables, see Figure 22. This type of arrangement will be used to review preliminary turbine and compressor airfoil shapes for structural acceptability, given assumptions about operating temperature and material capability. The results of these analyses will be used to improve the aerodynamic designs from a structural standpoint.

5.0 FUTURE INVESTIGATIONS

During the next quarter, Solar's technical team will examine a number of options for arrangement of engine components from the standpoint of achievable performance, first, and then to determine how well these options support the complete set of ATS requirements including cost, availability, reliability, and maintainability. Concurrent with this effort, several options for turbine material temperature/life/cooling requirements will be evaluated to determine their ability to support ATS requirements.

Table 2. Engine Concept Evaluation Criteria

Criterion	Explanation
Effect on Overall Turbine Efficiency	Turbine efficiency is the most important of the component efficiencies in an ICR operating at the conditions of Solar's proposed GFATS
Ducting and Flow Losses	These losses, while having only one third of the turbine's effect on overall cycle efficiency, per percentage change, ducting and flow losses are much more likely to vary widely with different engine configurations and are therefore deemed to be important enough to overall cycle efficiency to be considered separately in preliminary evaluations.
Effect on Efficiency of Other Components	Components other than the turbine section are considered as a group, and any particular configuration's impact on achievable component efficiencies will be weighed here.
Off Design Performance	This is a measure of the performance of the system both at off load conditions and at different site conditions.
Mechanical Complexity	This criterion is meant to be one of the measures of the potential first cost and overhaul and maintenance costs of the system, and of the potential for reduced system availability.
Control Complexity	Same as above criterion. Control complexity is added specifically because control issues are thought to be important enough to warrant inclusion and separate consideration.
Other Availability Issues	This criterion covers issues such as availability of a particular configuration when one of the other system components is out of service. It forces evaluators to consider the important issue of equipment availability carefully.
Hot Section Life	Life of the hot section must be considered because it will be an important driver of cost of power and of mechanical arrangements if replacement of hot section parts outside of the normal overhaul cycle.
First Cost	First cost must be considered separately because it is an important customer consideration.
Technical Risk	In order to optimize the choice of concept, the technical risks associated with the concept must be assessed.
Market Acceptance	A particular concept may measure well by all the other criteria, but it must be acceptable in the psychology of the marketplace.

Preliminary Airfoil Hub and Disk Stress Calculations Airfoil shape represents mass distribution

RSJ 12/8/83

Airfoil Hub Stress Calculations

Airfoil Constants and Given

nbldes=40 rpm=13000
r=7.75 pi=3.14159
rho=11.2 air_mean=.9
densd=.283 a_mean=1.8
taper=.5 diskthick=1.5*air_hub

Blade Equations

omega=pm*2*pi/60
vol=a_mean*bh
cpul=densd/386.4*vol*roent*omega*2
sigmahub=cpul/a_hub
dead_rim_pul=nbldes*cpul
hbh=(ro-r)/2

$$aneq=pi*(ro^2-r^2)*rpm^2$$

$$aneq=34710791736$$

$$hah=air_hub/2$$

$$hah=air_hub/2$$

$$hah=.5*diskthick$$

$$avehthickmean=2*pi*(hbh+r)*.08/(nbldes)$$

$$a_hub=2*a_mean/(1+taper)$$

$$a_tp=2*a_mean*taper/(1+taper)$$

$$air_hub=a_hub*(air_mean/a_mean)$$

$$air_tp=a_tp*(air_mean/a_mean)$$

$$a_mean=a_mean*avehthickmean$$

$$a_hub=2*a_mean/(1+taper)$$

$$a_tp=2*a_mean*taper/(1+taper)$$

Results

vol=0.739 in³
omega=1361.356 rad/sec
cpul=6317 pounds per blade
sigmahub=32604 psi
dead_rim_pul=372661

$$a_hub=0.286$$

$$a_tp=0.143$$

$$avehthickmean=0.119$$

Uniform Thickness Disk Stresses

Disk Constants and Given

a=1.5 disk bore
b=r-postheight disk live rim (b=7.000)
postheight=.75 (0-bladed disk)
g=386.4
h=diskthick
densd=.283 (disk material density)
attach_factor=.751=solid dead rim, no air pockets
mu=.3
nu=.001
r=7.000 Radius to evaluate stress

$$postheight=1.5*(2*pi*r/(2*nbldes))$$

Disk Equations

$I_rim=dead_rim_pul+attach_factor*pi*(r^2-b^2)*h*((densd+densd)/2)/386.4*((r+b)/2)*omega^2$
disk_elg_1=term1*term2*term3+term4*term5
disk_elg_r=term1*term2*term6+term4*term7
elg=(1/2*.5)*(disk_elg_1^2+disk_elg_r^2+(disk_elg_1-disk_elg_r)^2)*.5
sigtave=densd*omega^2*(b^2+a*b+a^2)/(3*g)+I_rim/(2*pi*h*(b-a))
term1=(3+mu)/8
term2=(densd*omega^2)/g
term3=a^2+b^2+a^2*b^2/r^2-(r^2*(1+3*mu)/(3+mu))
term4=I_rim*b/(2*pi*h*(b^2-a^2))
term5=1+(a^2/r^2)
term6=a^2+b^2-(a^2*b^2/r^2)-r^2
term7=1-(a^2/2r^2)
live_rim_stress=I_rim/(h^2*pi*b)

Disk Results

disk_elg_1=25823
disk_elg_r=10640
elg=22478
live_rim_stress=10640
I_rim=842349
sigtave=1481

$$postheight=0.750$$

Figure 22. Turbine Blade and Disc Parametric Model

DATE

FILMED

7 / 14 / 94

END