

A METHOD FOR THE OPTIMIZATION OF DESIGN PARAMETERS
FOR AN UNCOOLED RADIAL TURBINE

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ABSTRACT

A method to optimize the design parameters of an uncooled radial turbine has been developed. This method allows the study of overall inter-relations of stress, aerodynamic and performance parameters during the initial design of a fixed or free shaft engine. In particular it allows the effects of varying stress parameters, such as the number of blades, creep and low cycle fatigue requirements on aerodynamic and performance parameters, such as turbine efficiency and maximum turbine inlet temperature, to be studied. The method has been computerized. Extensive use is made of automatic plotting of the output data.

The mathematical model used in this study and the computer program, is based on the analysis of a simple strip section of the most highly loaded area of the turbine wheel. The results of this strip model are compared to those for complete wheel stress analysis and a factor is included in the program to improve the accuracy of the results. It is claimed that the final optimized design using this method will be an improvement over previous methods, since the inter-relationship of stress, aerodynamic and performance characteristics is more clearly understood.

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LIST OF SYMBOLS

<u>Symbol</u>	<u>Description</u>	<u>Units</u>
a) Chapter 3		
T	Temperature	°R
β	Relative Inlet Angle	degrees
α	Absolute Flow Angle	degrees
C	Absolute Inlet Velocity	FT/SEC
V	Inlet Velocity relative to the rotor	FT/SEC
U	Rotational Speed	FT/SEC
g	Gravitational Acceleration	FT/SEC ²
J	Mechanical equivalent of heat	FT, LB/Btu
C _p	Specific heat	Btu/lb, in. °R
ω	Angular Speed	RAD/SEC
r	Radius	In.
R.I.T.	Rotor Inlet Temperature	°F
η	Recovery Factor	---
P _r	Prandtl Number	---
γ	Ratio of Specific Heats	---
R	Universal Gas Constant	FT, LB/LB °F
f/a	Fuel/Air Ratio	---
P	Pressure	psia

<u>Symbol</u>	<u>Description</u>	<u>Units</u>
b) Chapter 4:		
t	Thickness	In.
r	Radius	In.
WL	Wave Length	In.
CF	Centrifugal Force	Lb
ω	Angular Velocity	Rad/Sec
σ	Stress	psi
G3	Adverse Tolerance Factor	--
TOL	Tolerance	In.
ARC	Arc between two adjacent blades	In.
BSF	Bore Stress Factor	--
ν	Poisson's Ratio	--
α	Coefficient of thermal Expansion	In/In. °F
E	Young's Modulus	psi
T	Temperature of Metal	°F
ΔT	Temperature Gradient	°F
NB	Number of Blades	--

<u>Symbol</u>	<u>Description</u>	<u>Symbol</u>
(c) Chapter 5		
W	Compressor Mass Flow	LB/SEC
NS	Specific Speed	---
PR	Pressure Ratio	---
P	Pressure	psia
T	Temperature	°F
η	Compressor Efficiency	---
N	Mechanical Speed	rpm
R	Gas Constant of Air	FT.LB/LB °F
γ	Ratio of Specific Heat	---
h	Adiabatic Work	FT.LB/LB
ρ	Density of Air	LB/FT ³

<u>Symbol</u>	<u>Description</u>	<u>Units</u>
d) Chapter 6		
NS	Specific Speed	---
N	Mechanical Speed	rpm
h, H	Adiabatic Work	FT. LB/LB
α	Absolute Flow Angle	degrees
β	Relative Flow Angle	degrees
C	Absolute Inlet Velocity	FT/SEC
V	Relative Inlet Velocity	FT/SEC
U	Rotational Speed	FT/SEC
J	Mechanical equivalent of heat	FT. LB /Btu
g	Gravitational Acceleration	FT/SEC ²
η	Efficiency	---
PR	Pressure Ratio	---
T	Temperature	°F
RIT	Rotor Inlet Temperature	°F
C_p	Specific Heat	Btu/LB. IN °R
γ	Ratio of Specific Heat	---
M	Mach Number	
R, r	Radius	In.
W	Mass Flow	LB/SEC
F/A	Fuel Air Ratio	---
P	Pressure	psia
CD	Discharge Coefficient	---
A	Area	IN ²

Subscripts

- m	Metal
- aw	Adiabatic Wall
- 3	Station at Blade Inlet
- 3S	Static at Blade Inlet
- 3T	Total at Blade Inlet
- g	Gas
- a	Air
- 3RT	Relative Total at Blade Inlet
- R	Relative
- r_i	At Radius r_i , station i from Blade Tip
- T	Tip, Total, Thermal, or Turbine
- i	Station i from Blade Tip
- M3	Metal at Blade Inlet
- aw3	Adiabatic Wall at Blade Inlet
- 2	Station at Compressor Exit
- Burner	Burner
- •	Transition or at Compressor Inlet
- CF	Due to Centrifugal Force
- SAT	Stress under Adverse Tolerance
- NOM	Nominal
- All	Allowable
- $i + 1$	At Station $i + 1$
- b, B	Bore

Subscripts

- h, H	Hub
- r,	Radial
- t	Tangential
- c	Compressor
- 1	At Compressor Inlet
- C, a	Compressor, Adiabatic
- T, T	Total to Total
- 6	Station at Exit of Turbine
- 6H	Station at Exit of Turbine at Hub Radius
- 6T	Station at Exit of Turbine at Tip Radius
- 2-3	Compressor Stage
- 4-6	Turbine Stage
- Mech.	Mechanical

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CHAPTER 1.

INTRODUCTION

In small single or free shaft engines the use of gas generator radial turbines offers certain advantages over axial turbines.

The main advantages of the radial turbine are its simplicity of design, its ruggedness and its relative insensitivity to tip clearance effects. It also allows large pressure drops and hence high work output per stage, compared to the axial turbine. One radial turbine can replace at least two stages of efficiently loaded axial turbine, and retain high efficiency. This thesis is concerned with a method of design optimization of single and free shaft radial gas turbine engines including all relevant parameters affecting this design.

The final design configuration of a radial turbine requires the definition of the values of a large number of controlling parameters. These values depend primarily on the required overall engine performance.

The parameters can generally be separated into three inter-relating specialist sections within the industry, namely aerodynamic, (Compressor and turbine), performance, and stress (including manufacturing).

If each section were asked to optimize the design for a given radial turbine engine application, the result would be three separate optimized conditions for the radial turbine design. The final overall optimized design is therefore the result of extensive trade-off studies between the sections. Since they are generally strongly

interdependent, such a trade-off study is not simple. In the author's experience the design of a radial turbine is usually the result of trade-off studies between aerodynamic and performance functions. Only after a design point is chosen is a stress check carried out.

This stress check can invalidate the chosen design point requiring a further performance-aerodynamic cycle study.

One objective of the work described in this thesis is to include stress requirements from the beginning of an overall radial turbine design cycle.

The stress requirements are included in a sufficiently accurate form to allow initial optimization of such variables as the number of blades, creep and low cycle fatigue. The effect of blade thickness tolerances on hub stresses has been included in the mathematical model and this has never been considered in past studies. The method has been computerized with automatic plotting of the output data. A further objective is to provide a computer program which will allow rapid studies to determine the potential of radial turbines for various engine applications.

CHAPTER 2

GENERAL PROCEDURE

At the beginning of a radial turbine design cycle, a number of parameters can be defined which depend on the type of engine being considered and the type of missions the engine is expected to perform.

Stress requirements such as creep, low cycle fatigue and rotor burst margin are fixed as a consequence of the above considerations. A preliminary study is carried out by the aerodynamic and performance group in order to define aerodynamic parameters from power and specific fuel consumption requirements. This initial evaluation becomes the starting point of the optimization process described by this thesis and carried out by the computer program.

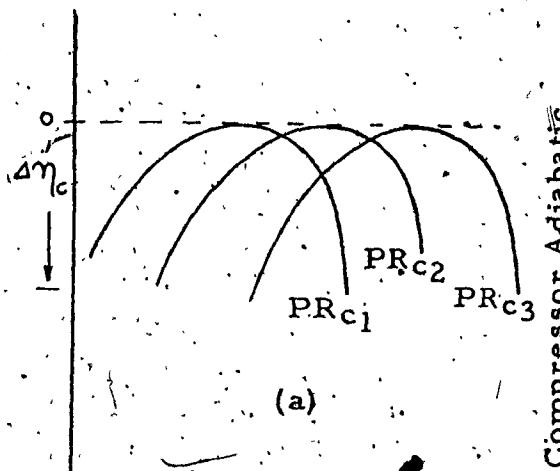
The results will indicate whether the chosen turbine wheel material is suitable, whether blade cooling or some other refinement is going to be necessary, and which direction the aerodynamic-performance trade-off should take.

The general procedure of this method is described below as it is actually handled by the computer program, from compressor inlet to turbine exit.

2.1 Compressor Data

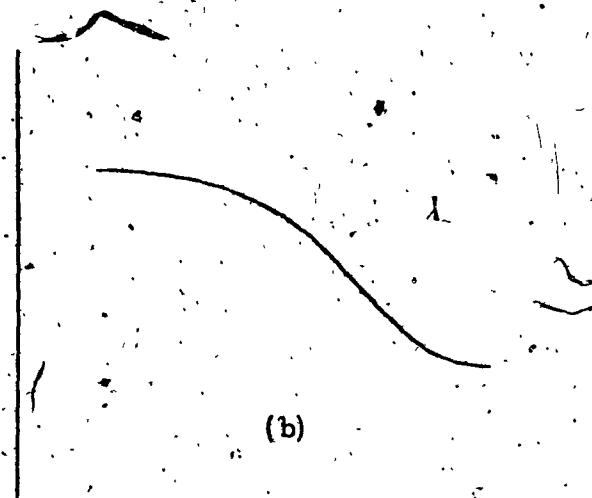
In general, the level of compressor technology within a company is defined by the following correlations (Fig. 2.1). These are results of extensive testing and they represent the company compressor state-of-the-art.

Compressor Efficiency Decrement



(a)

Compressor Adiabatic Efficiency



(b)

Compressor Specific Speed
 NS_C

Compressor Pressure Ratio
 PR_C

Fig. 2.1

Once the compressor pressure ratio, specific speed and mass flow are defined from preliminary performance studies, the compressor mechanical speed and exit conditions can be found from the theory described in Section 5.

2.2 Combustor

The combustion process is defined in terms of the effect of fuel on mass flow. From the compressor exit conditions, combustor efficiency and the required turbine inlet temperature, the increase in mass flow due to fuel added can be derived. The theory covering this is described in Sections 3.1 and 6.

Thus the entry conditions to the turbine are fully defined, allowance being made for any compressor air bleed, and pressure losses.

2.3 Turbine Data

The state of radial turbine technology is defined by correlations as shown on Fig. 2.2 below:

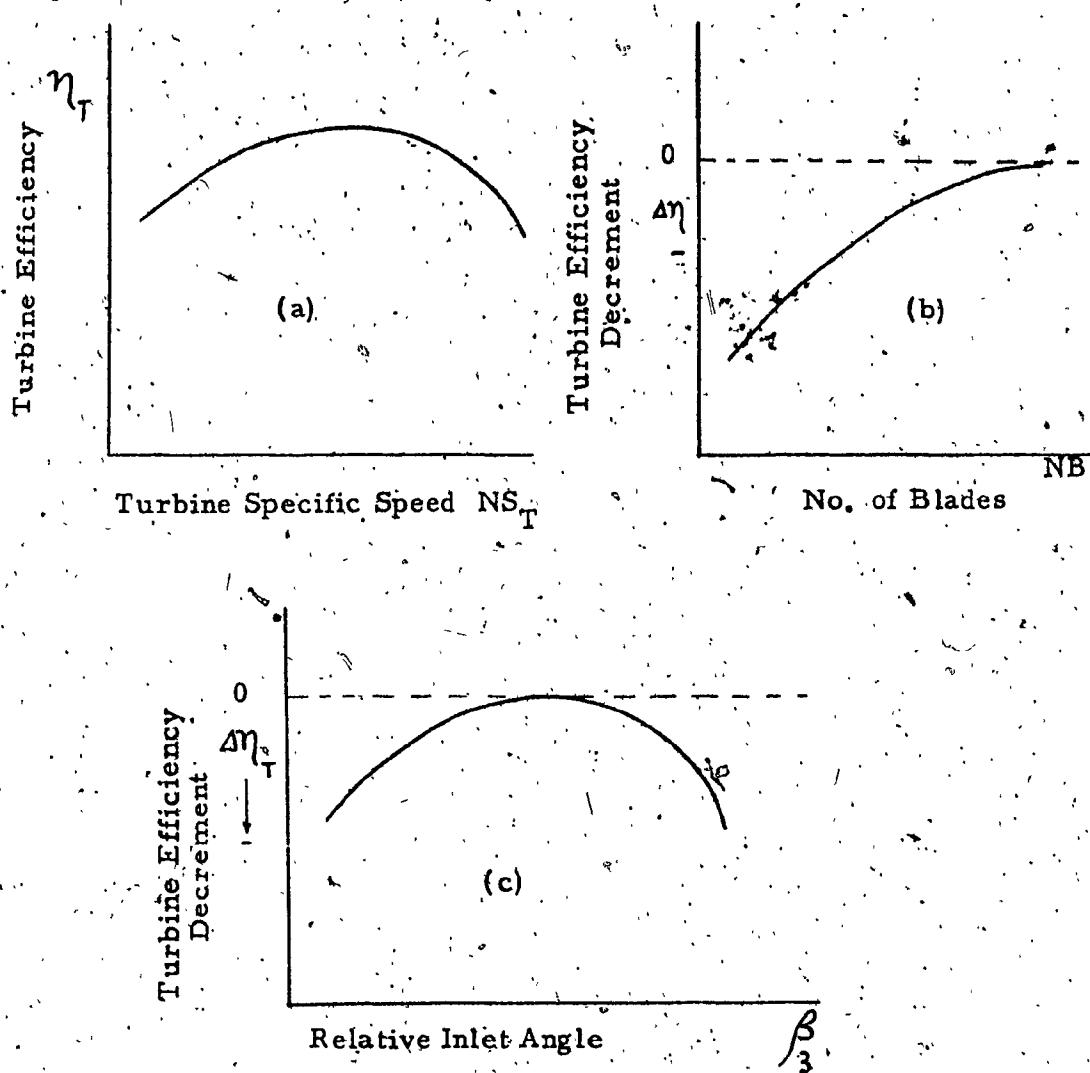


Fig. 2.2

In the case of a free shaft engine, the turbine specific speed, number of blades and relative inlet angle defines the turbine efficiency needed to calculate the turbine pressure ratio. Once the turbine pressure ratio is determined the turbine tip speed can be calculated as shown in the theory of Section 6.

2.4 Stress Analysis

Using the turbine parameters defined, strip stress analysis is used to calculate the blade thicknesses designed to lie within the limit of allowable stress at any radius.

As, in general, there is no blade cooling in radial turbines, the blade metal temperature is assumed equal to the adiabatic wall temperature. Since the value of the adiabatic wall temperature is a function of radius and its initial value at the rotor tip is a function of rotor inlet temperature (R.I.T.) and relative inlet angle (β_3), these turbine parameters have a direct control on allowable stress levels in the blade and hence on the blade thickness distribution. The actual variation of allowable stress with temperature is defined, for a given material, by the creep and burst criteria for the rotor.

The adiabatic wall temperature thus defines the variation of allowable stress with radius for the subsequent design of the blade thicknesses, and is fully covered in Section 3.

The inter-relation of stress and aerodynamic conditions are further illustrated in the following example. In turbine design there exists a positive relation between turbine efficiency, relative inlet flow angle β_3 and the number of blades as illustrated in Fig. 2.2.

From an aerodynamic viewpoint only, to maintain high turbine efficiency, the number of blades should be high and the relative angle β_3 should be negative.

However, relative angle β_3 directly affects allowable stress levels in the blade and hence the blade thickness distribution. The actual effect is that, as β_3 tends to the negative direction the metal

temperature increases, the thickness and stresses in the rotor hub increase due to the increase in the centrifugal force on the blade. Since the maximum stresses occur at the hub center or bore, and are controlled by low cycle fatigue requirements, β_3 has a direct effect on the rotor life.

Similarly with the number of blades. The more blades, the greater the total blade radial force on the hub, and thus the higher the hub maximum stress.

Consequently there exists a direct conflict between stress and aerodynamic conditions.

These conflicting effects were previously recognized.

The method described allows a more accurate trade-off to be considered.

The example shown in Section 7 indicates how this procedure of including all relevant stress and aerodynamic parameters leads to a better understanding of the overall optimization of turbine design.

It allows rapid initial studies to be carried out on a radial turbine application, indicates the degree of complexity of the final design, whether cooling is necessary, and other refinements which will give useful foresight into the possible final design conditions.

CHAPTER 3

THEORY TO CALCULATE THE BLADE METAL TEMPERATURE

If conduction effects are not present the metal temperature T_m of the blade would be equal to the adiabatic wall temperature T_{aw} .

Since M. Van Duyn (3) has found that effects in the blade tip are small it will be assumed that

$$T_m = T_{aw} \quad (3.1)$$

Figure 3.1 shows a velocity triangle at the turbine entry where the relative inlet angle $\beta_3 = 0$

The static and total

(stagnation) temper-

atures of a perfect

gas are (Ref. 4):

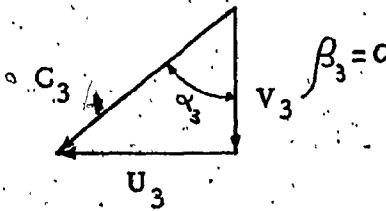


Fig. 3.1.

$$T_{3S} = T_{3T} - \frac{C_3^2}{2gJC_p g} \quad (3.2)$$

and the relation-

ship between static

and rotor relative

temperatures is

given by:

$$T_{3S} = T_{3RT} - \frac{V_3^2}{2gJC_p g} \quad (3.3)$$

Where:

- T_{3S} = Static Inlet Temperature
- T_{3T} = Total Absolute Inlet Temperature
- T_{3RT} = Total Inlet Temperature Relative to the Rotor
- C_3 = Absolute Inlet Velocity
- V_3 = Inlet Velocity Relative to the Rotor
- g = Acceleration due to Gravity
- J = Mechanical Equivalent to Heat
- Cp_g = Specific Heat of Gas

For a radial turbine, the difference between the rotor inlet relative temperature and the relative temperature at any radius r_i is related to the change in rotational velocity (Ref. 5) by:

$$T_{3RT} - T_{R(r_i)} = \Delta U = \frac{U_3^2 - U_{r_i}^2}{2gJCp_g} = \frac{\omega_{rT}^2 - \omega_{r_i}^2}{2gJCp_g}$$

or:

$$T_{3RT} - T_{R(r_i)} = \frac{\omega^2 (r_T^2 - r_i^2)}{2gJCp_g} \quad (3.4)$$

Using eqs. 3.2 and 3.3 we have:

$$T_{3T} - \frac{C_3^2}{2gJCp_g} = T_{3RT} - \frac{V_3^2}{2gJCp_g}$$

from which:

$$T_{3RT} = T_{3T} - \frac{(C_3^2 - V_3^2)}{2gJCp_g} \quad (3.5)$$

From Fig. 3.1 we have:

$$U_3^2 = C_3^2 - V_3^2 \quad (3.6)$$

That after substitution into 3.5 gives.

$$T_{3RT} = T_{3T} - \frac{U^2}{2gJCp_g} = T_{3T} - \frac{\omega^2 r_T^2}{2gJCp_g} \quad (3.7)$$

Equating eqs. 3.4 and 3.7 we have:

$$T_{R(r_i)} + \frac{\omega^2 (r_T^2 - r_i^2)}{2gJCp_g} = T_{3T} - \frac{\omega^2 r_T^2}{2gJCp_g}$$

from which:

$$T_{R(r_i)} = T_{3T} - \frac{\omega^2 (2r_T^2 - r_i^2)}{2gJCp_g} \quad (3.8)$$

Equation 3.8 is valid only for $\beta_3 = 0$.

For $\beta_3 \neq 0$ an approximate relation including the effect of β_3 on metal temperature is derived as follows, for the case when β_3 and the absolute flow angle α_3 are known.

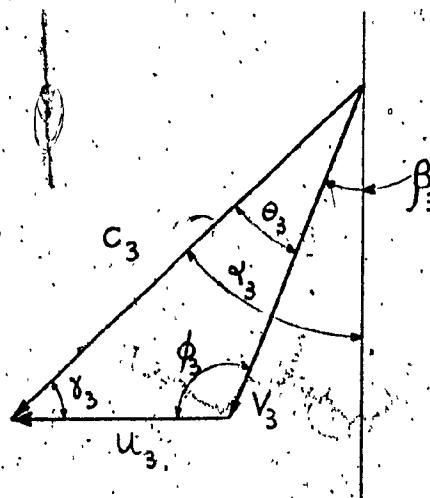


Fig. 3.2

From the velocity vector diagram at rotor inlet shown in figure 3.2:

$$\gamma_3 = 90 - \alpha_3 \quad (3.9)$$

$$\theta_3 = \alpha_3 - \beta_3 \quad (3.10)$$

and

$$\phi_3 = 180 - \gamma_3 - \theta_3$$

using 3.9 and 3.10 gives: $\phi_3 = 180 - 90 + \alpha_3 - \alpha_3 + \beta_3 = 90 + \beta_3$ $\quad (3.11)$

$$\phi_3 = 180 - 90 + \alpha_3 - \alpha_3 + \beta_3 = 90 + \beta_3$$

From equation 3.5:

$$T_{3R} = T_{3T} \frac{(C_3^2 - V_3^2)}{2gJC_P g} \quad (3.12)$$

where T_{3T} = R.I.T. (Rotor Inlet Temperature)

From Fig. 3.2 and using the sine theorem,

$$\begin{aligned} \frac{C_3}{\sin \phi_3} &= \frac{V_3}{\sin \gamma_3} = \frac{U_3}{\sin \theta_3} \\ V_3 &= \frac{U_3 \sin \gamma_3}{\sin \theta_3} = U_3 \frac{\sin(90 - \alpha_3)}{\sin(\alpha_3 - \beta_3)} \\ &\frac{U_3 \cos \alpha_3}{\sin(\alpha_3 - \beta_3)} \end{aligned} \quad (3.13)$$

$$\text{and } C_3 = \frac{U_3 \sin \phi_3}{\sin \theta_3} = \frac{U_3 \sin(90 + \beta_3)}{\sin(\alpha_3 - \beta_3)}$$

$$\frac{U_3 \cos \beta_3}{\sin(\alpha_3 - \beta_3)} \quad (3.14)$$

Substituting 3.13 and 3.14 into eq. 3.12 gives: (3.15)

$$T_{3R} = R.I.T. - \frac{U_3^2}{2gJCp_g} \left[\frac{\cos^2 \beta_3 - \cos^2 \alpha_3}{\sin^2(\alpha_3 - \beta_3)} \right]$$

If V_3 is constant at any radius r_i , then it is possible to calculate the adiabatic wall temperature at the tip by taking into account the recovery factor $\eta \approx \sqrt{P_r} \approx .85$ (P_r = Prandtl number)

Then,

$$T_{m3} = T_{aw3} = T_{3R} - \frac{V_3^2}{2gJCp_g} (1 - \eta) \quad (3.16)$$

or:

$$T_{m3} = R.I.T. - \frac{U_3^2}{2gJCp_g} \left[\frac{\cos^2 \beta_3 - 0.85 \cos^2 \alpha_3}{\sin^2(\alpha_3 - \beta_3)} \right] \quad (3.17)$$

The assumption of V_3 constant is justified by the fact that for $V_3 = 500$ ft/sec the term $\frac{V_3^2}{2gJCp_g} (1 - \eta) \approx 2^\circ F$ and therefore the effect of a variation of V_3 may be neglected. The relation between T_{3R} and $T_{R(r_i)}$ along a radial strip is:

$$\frac{T_{R(r_i)} - T_{aw(r_i)}}{\frac{\omega^2 (r_T^2 - r_i^2)}{2gJCp_g}} \approx T_{m(r_i)} = T_{m3} \quad (3.18)$$

and substituting eq. 3.17 into 3.18 gives:

$$T_{M(r_i)} = R.I.T. \cdot \frac{U_3^2}{2gJCp_g} \left[\frac{\cos^2 \beta_3 - 0.85 \cos^2 \alpha_3}{\sin^2 (\alpha_3 - \beta_3)} \right]$$

$$\frac{w^2 (r_T^2 - r_i^2)}{2gJCp_g} \quad (3.19)$$

Equation 3.19 gives the metal temperature along a radial strip and it allows the calculation of allowable stress as a function of metal temp. ($\sigma_{\text{Allowable}} = f(T_m)$) to be used in the equations of section 4.

The value of the specific heat of gas C_p , in all equations, is calculated at temperature $T_{m(r_i)}$ by means of an iterative process.

3.1 Properties of Air and Products of Combustion of Hydrocarbon Fuels

The specific heat and specific heat ratio are a function of the temperature and fuel air ratio.

For no fuel, as for the compressor calculations, the known properties of air may be used in the theory.

Figure 3.3 shows the variation of Cp_a (specific heat of air) with temperature. The curve has been used to calculate the following polynomial that closely represents its value.

$$Cp_a = .24841 \cdot T \left\{ 4.7464 \cdot 10^{-5} - T [7.0933 \cdot 10^{-8} - T \cdot (2.6510 \cdot 10^{-11} - T \cdot 3.2585 \cdot 10^{-15})] \right\} \quad (3.20)$$

SPECIFIC HEAT VARIATION WITH TEMPERATURE

K & K GAS TABLES.

(Copied from TD 103.1)

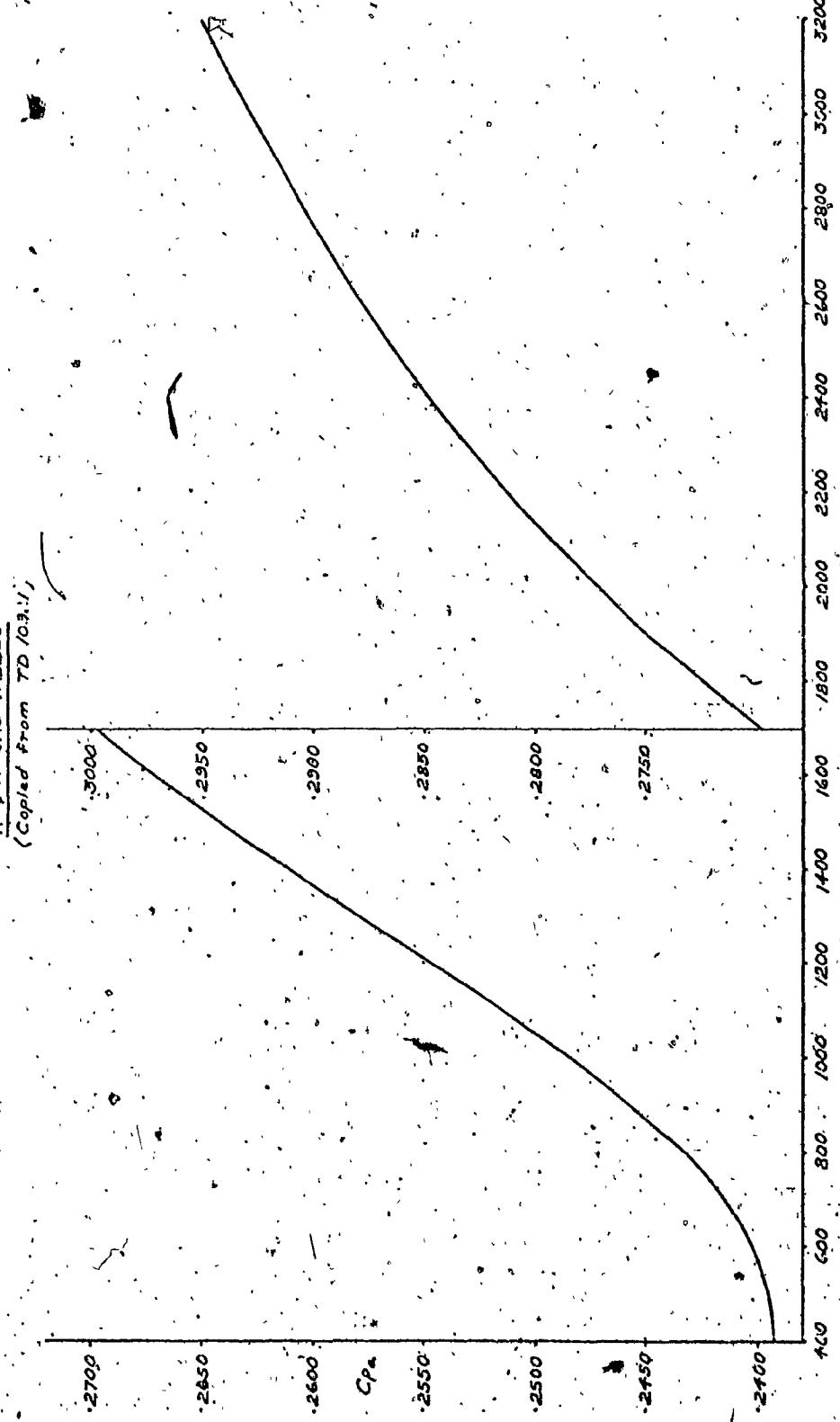


FIG. 3.3.

The value of the specific heat ratio γ is given by:

$$\gamma = \frac{C_p J}{C_p J - R} \quad (3.21)$$

Where:

$$J \approx 778.26 \quad \text{Ft. Lb/Btu}$$

$$R = 53.350 \quad \text{Ft. Lb/Lb. } ^\circ\text{F}$$

Figure 3.10 shows its variation with temperature ($f/a = 0$).

When the fuel air ratio is not zero the gas contains the products of combustion of hydrocarbon fuel. The method used to calculate their properties follows.

From the compressor theory (sect. 5) the following quantities are calculated:

P_2 - Compressor Delivery Pressure

T_2 - Compressor Delivery Temperature

If the burner efficiency η_{BURNER} and the turbine rotor inlet temperature (R.I.T.) are specified Fig. 3.4 gives the fuel-air ratio required for the combustion process, assuming an ideal temperature rise during constant pressure combustion. This fuel air ratio is then added to the compressor mass flow (after removing any compressor air bleed flow) to get the turbine mass flow.

Figure 3.4 is analytically represented by the following equations:

$$(F/A)_{400^\circ R} = 0.69379 \times 10^{-4} + 0.12447 \times 10^{-4} \Delta T + 0.4144 \times 10^{-8} \Delta T^2 \quad (3.22)$$

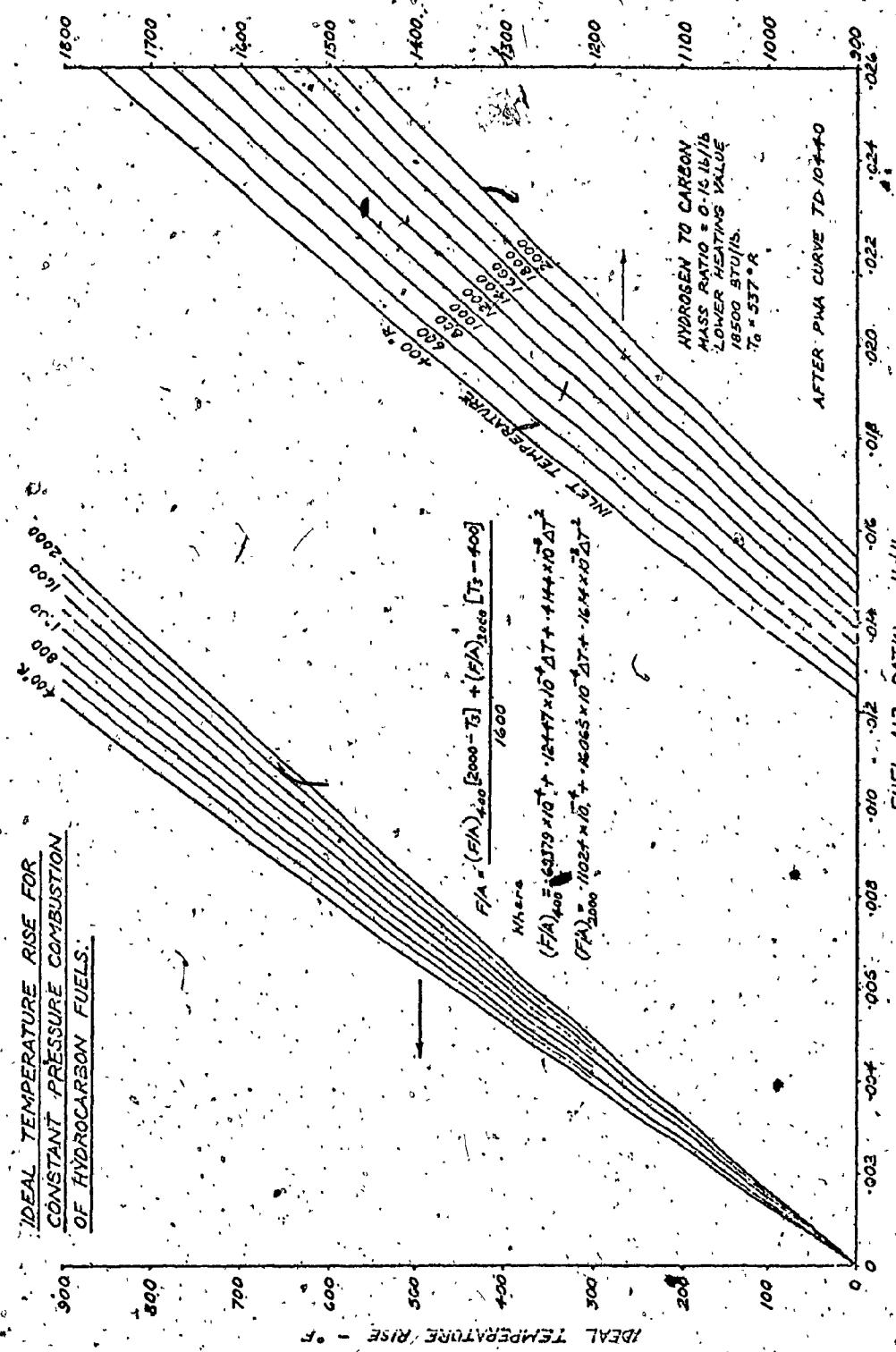


FIG. 3.4.

$$(F/A)_{2000^{\circ}R} = 0.11024 \times 10^{-4} + 0.16065 \times 10^{-4} \Delta T + \\ 0.1674 \times 10^{-8} \Delta T^2 \quad (3.23)$$

$$(F/A) = \frac{(F/A)_{400} [2000 - R.I.T.]}{1600} + \\ + \frac{(F/A)_{2000} [R.I.T. - 400]}{1600} \quad (3.24)$$

The specific heat is then calculated by using a correction factor for combustion products K_G as shown on Fig. 3.5.

Figures 3.6, 3.7 and 3.8 show the variation of C_p with temperature and fuel air ratio. All curves have been fitted into polynomial form and the resulting equation is:

$$C_{p_g} = \left\{ \frac{FAGAS}{.033} \cdot \left\{ \left[(1.212 \cdot 10^{-10} \cdot T - 1.041 \cdot 10^{-6}) \cdot \right. \right. \right. \\ \left. \left. \left. T + 3.651 \cdot 10^{-3} \right] \cdot T + .0139 \right\} + 1 \right\} \cdot C_{p_a} \quad (3.25)$$

Where C_{p_a} is determined from eq. 3.20.

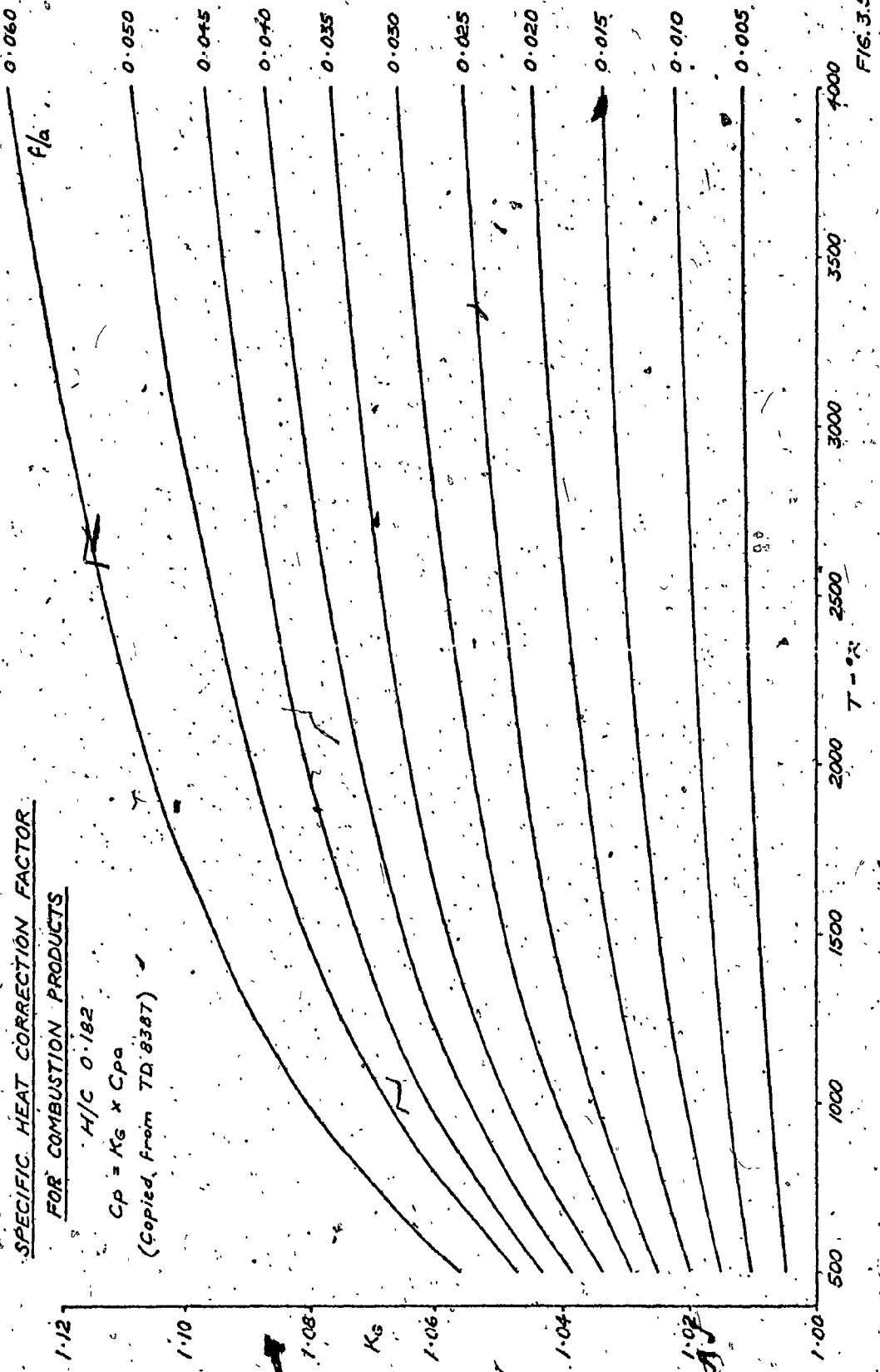
The specific heat ratio is calculated using eq. 3.21 and the results for various fuel-air ratios are shown on Fig. 3.9 and 3.10.

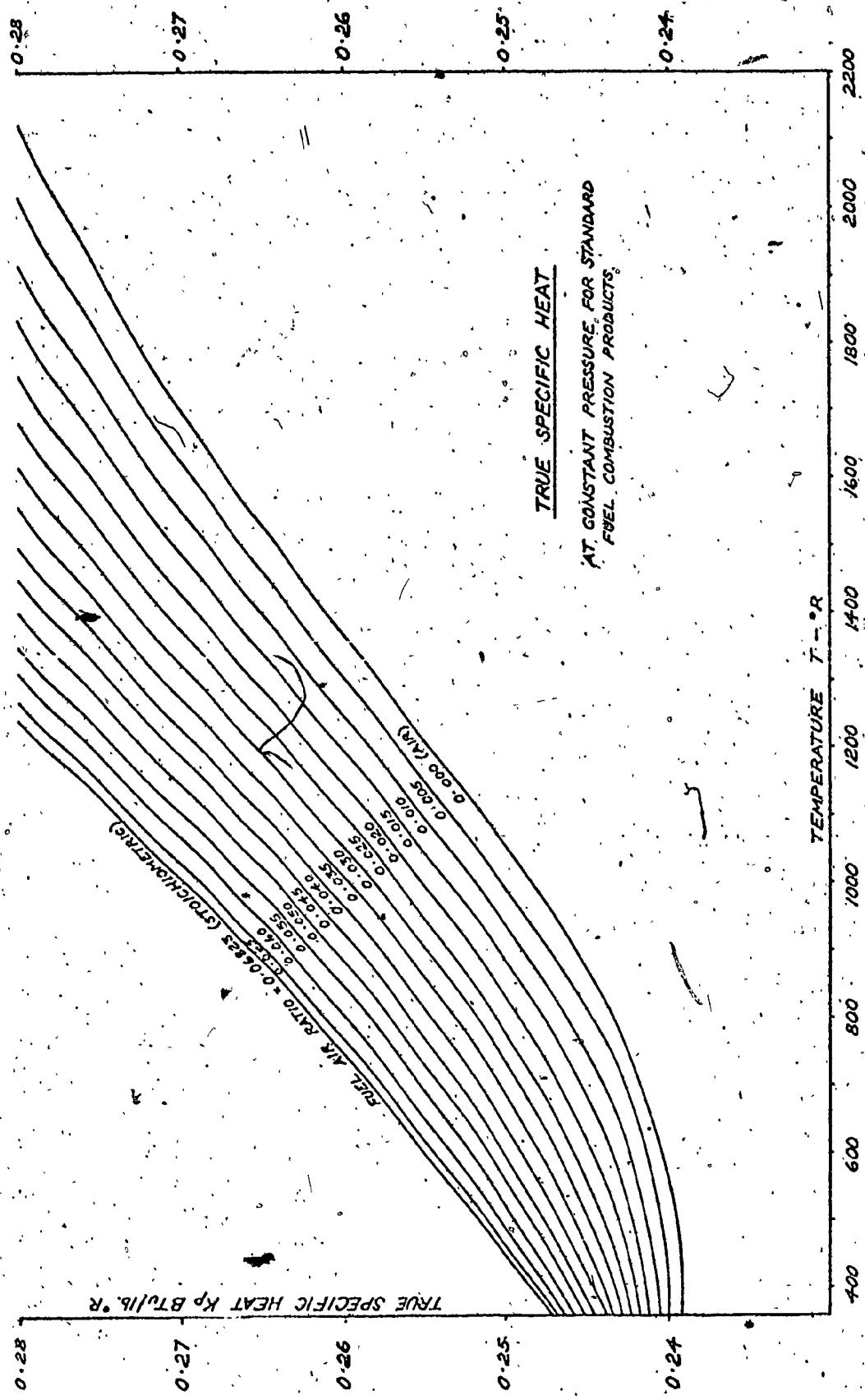
The temperature values used for the calculation of the air or gas properties are specified in each chapter where such a computation is required.

SPECIFIC HEAT CORRECTION FACTOR
FOR COMBUSTION PRODUCTS

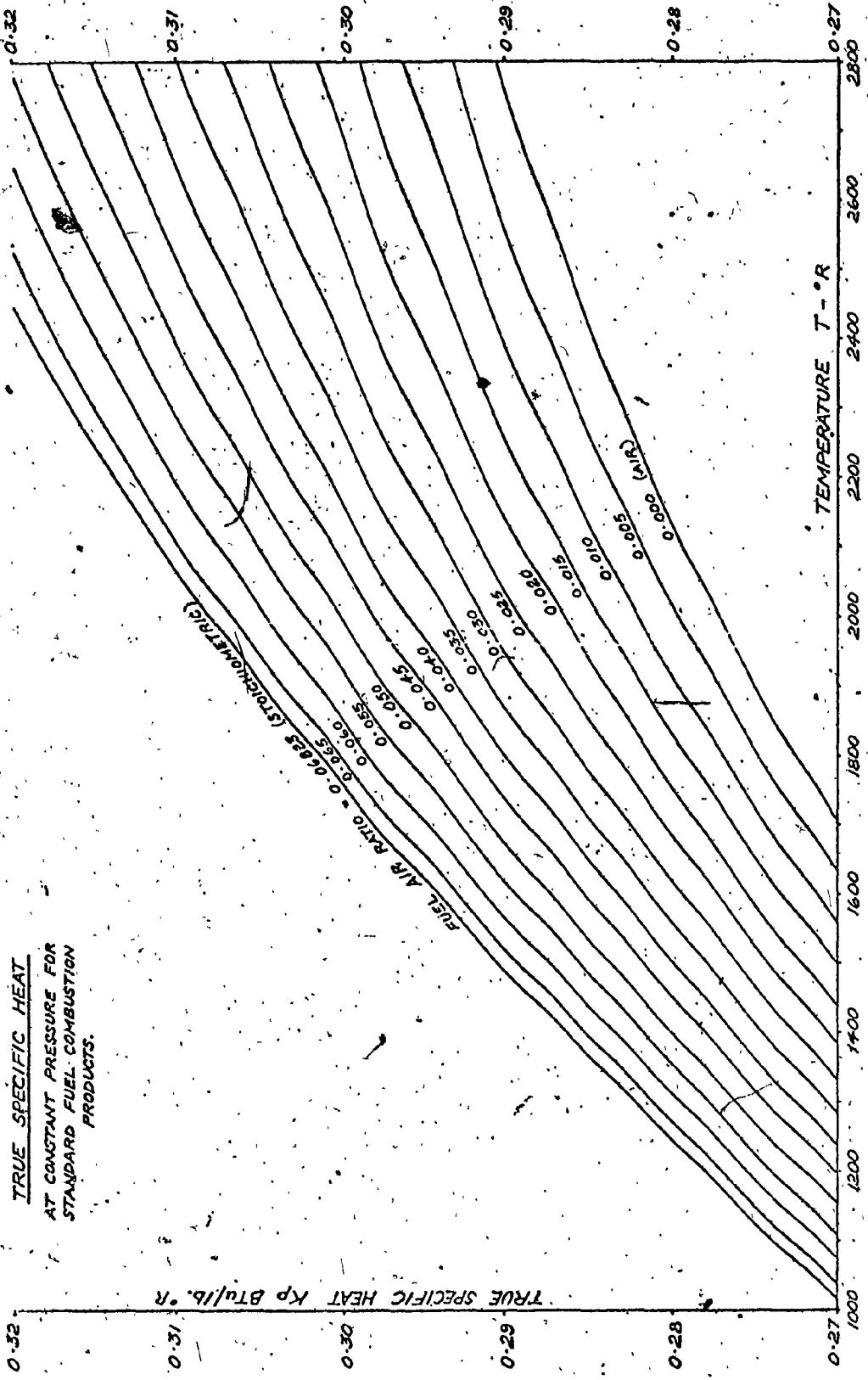
$H/C = 0.182$

$C_P = K_S \times C_{P0}$
(Copied from TD-8387)

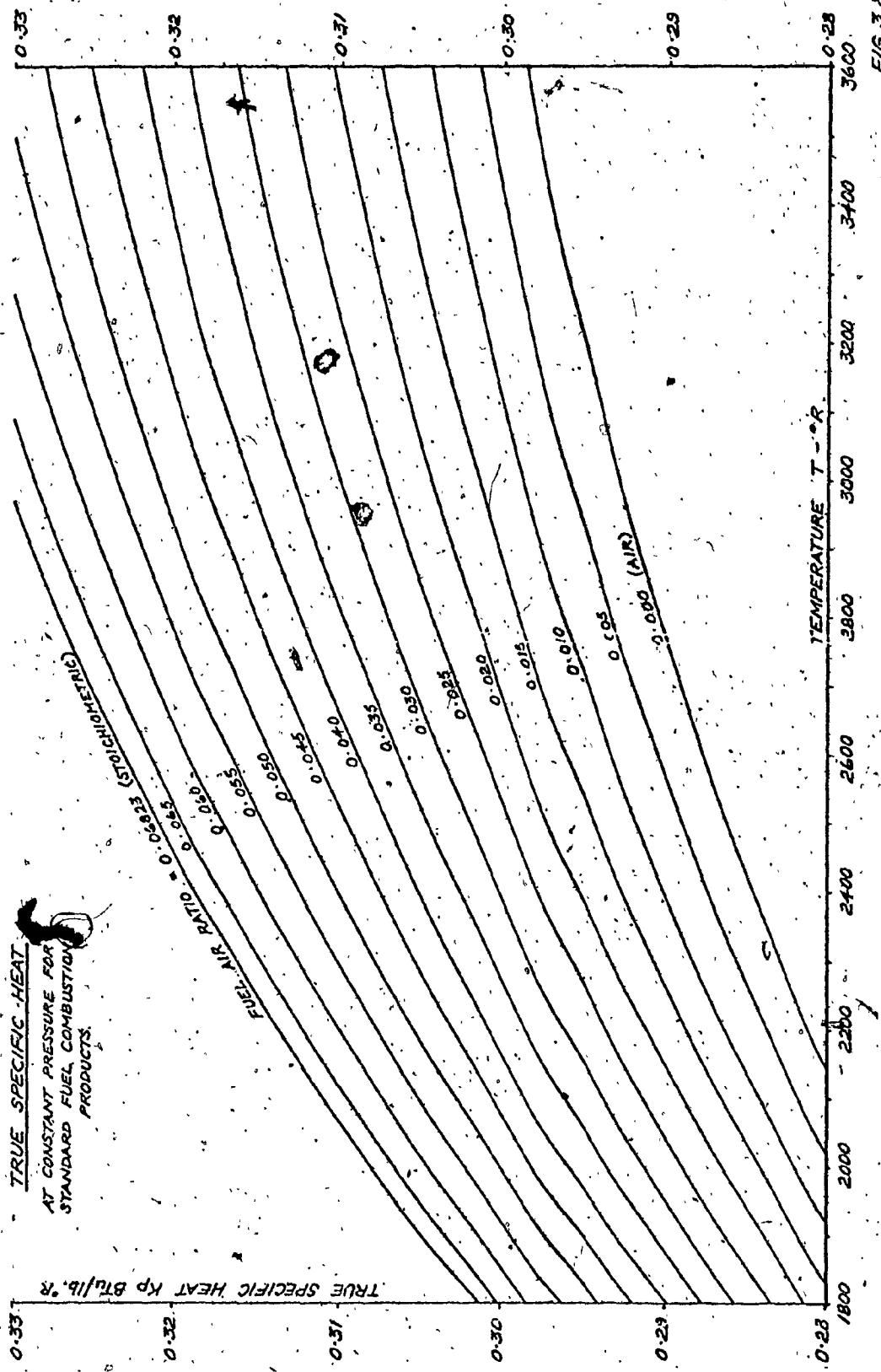


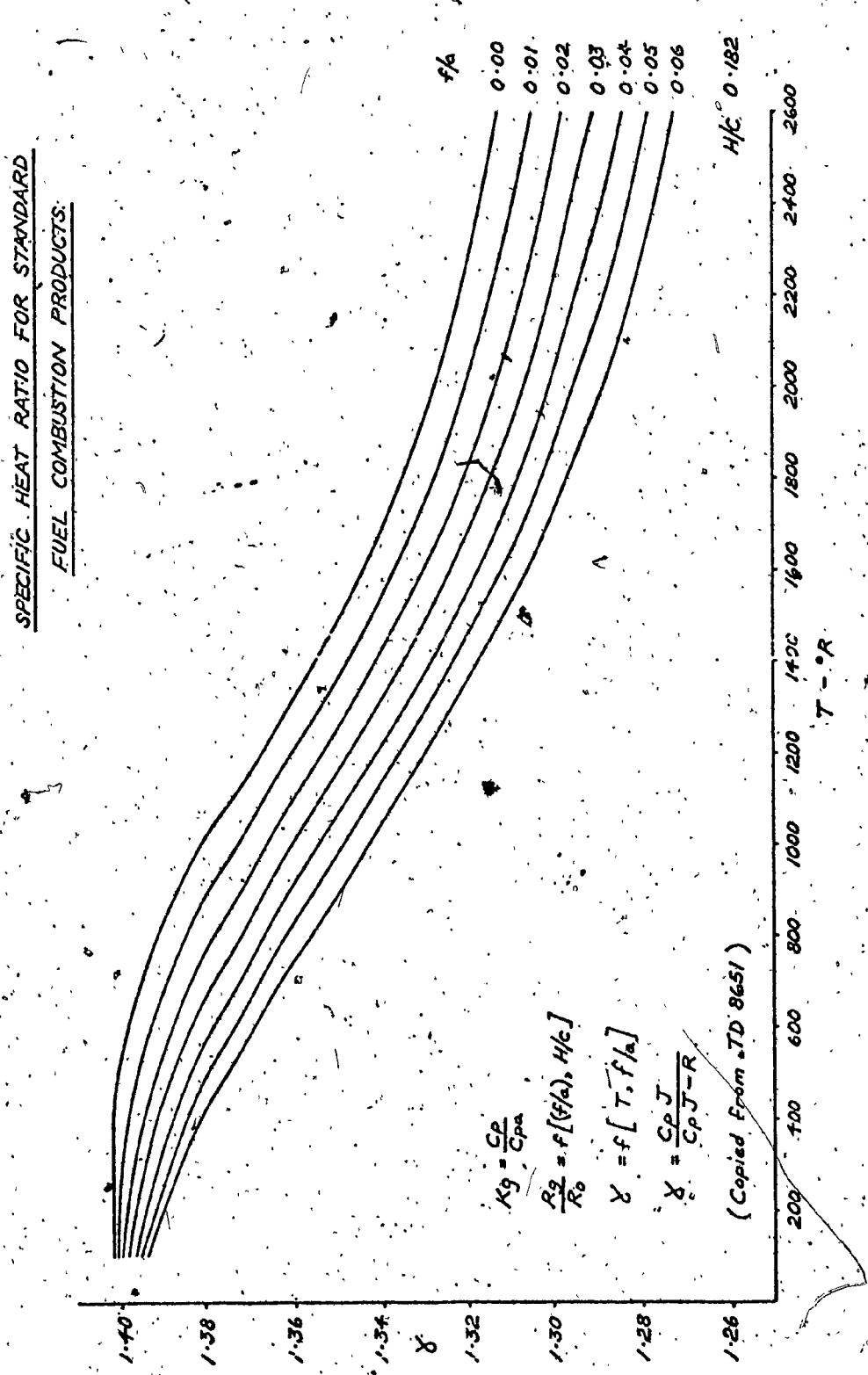


TRUE SPECIFIC HEAT
AT CONSTANT PRESSURE FOR
STANDARD FUEL COMBUSTION
PRODUCTS.



F/G-37





SPECIFIC HEAT RATIO FOR STANDARD
FUEL COMBUSTION PRODUCTS

(Copied from TD 8652)

$$\gamma = f(T, f\%)$$

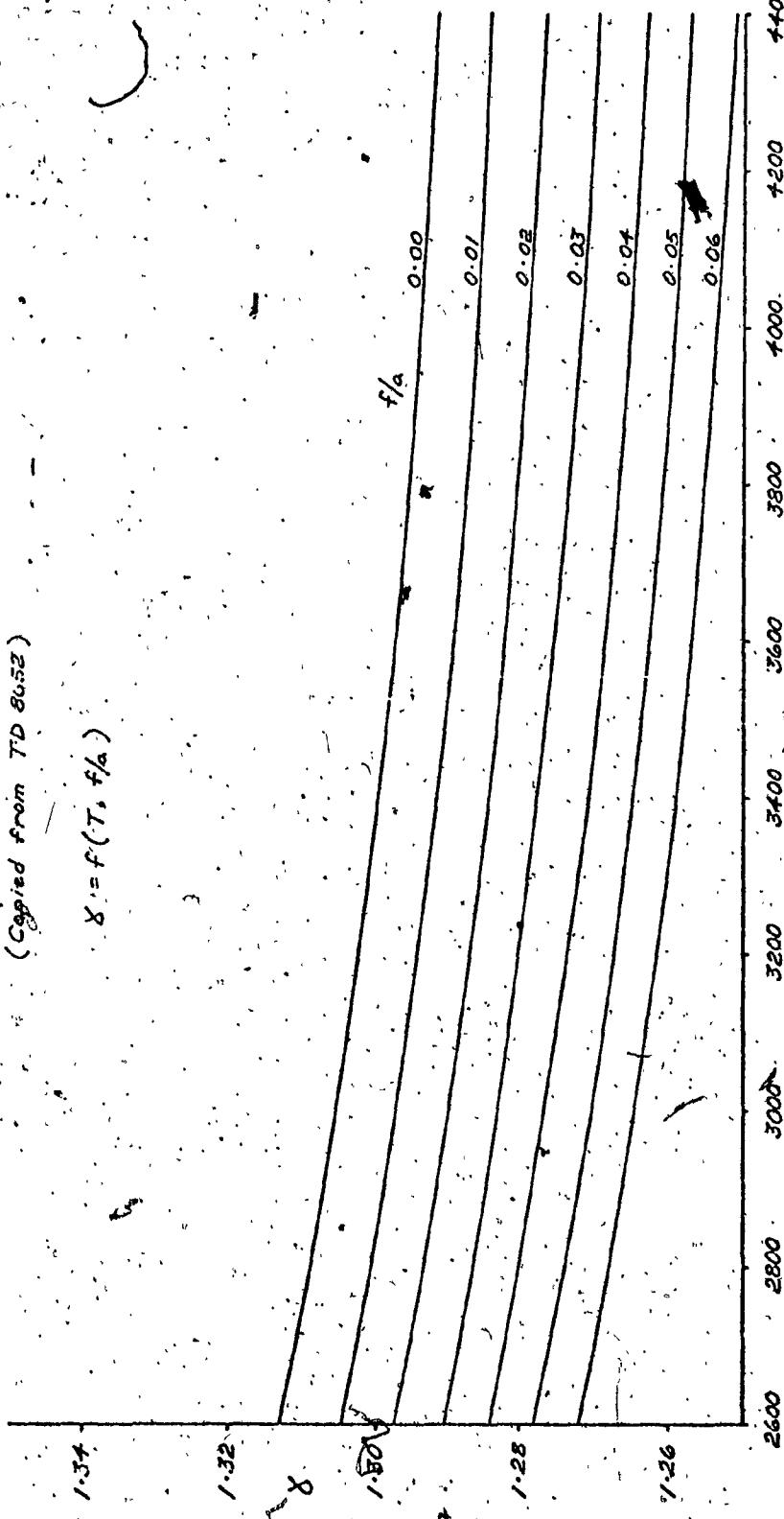


FIG. 5.10.

CHAPTER 4

STRESS ANALYSIS - GENERAL DESCRIPTION

The method described in this thesis for carrying out optimization studies of the design parameters of radial turbines, includes both aerodynamic and stress variables from the beginning of the design cycle. Since the method is intended to give rapid results, and a large number of variables are necessarily involved in turbine design, the stress model used in this new procedure is designed to be simple, to allow rapid parametric trade-offs studies. In past designs, stress conditions have been checked for each complete wheel model using a finite element program.

In an initial parametric study, this becomes impracticable due to the large amount of stress input data required for each condition. Thus some simple stress model was chosen which could be compared and corrected to give the stress results for a complete radial turbine wheel.

A simple strip analysis approach was considered and its correlation with actual stress levels in complete finite element analyses was studied.

The basic assumption in the strip analysis is that a radial strip can be used as being representative of the maximum stressed area in a radial wheel. Previous complete wheel analyses (Ref. 6) have indicated that the maximum stress at the wheel bore or centre occurs at a location as shown in Fig. 4.1.

A radial strip at this section will then be indicative of the most highly stressed area of the hub.

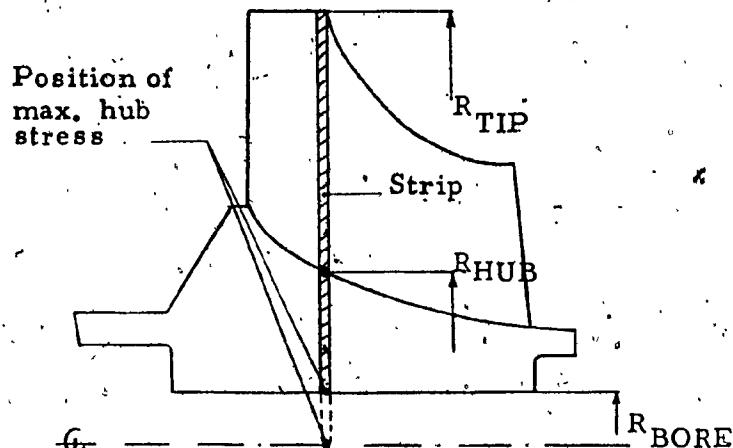


Figure 4.1

In the case of the blade, the blade strip chosen, will be justified as a realistic representation of the blade.

The stresses derived from this strip analysis for the wheel are higher than the true stresses at this location, since the strip does not allow for the interaction effect of the less highly loaded adjacent sections of the wheel. This interaction effect reduces the strip analysis hub stresses by a significant amount. The magnitude of the interaction is important since the maximum stress at the wheel bore or centre defines the maximum operating conditions for a particular wheel.

In the case of the hub centre where the effect is large, use is made of the fact that for typical radial turbine engine applications, the maximum stress is controlled by the low cycle fatigue requirements and an accurate conversion factor is evaluated here which will relate the strip analysis hub centre or bore stress to the maximum stress in an actual wheel. The derivation of this correlation factor

is further discussed in paragraph 4.3.

In the blade, the interaction effect is small. Since sections either side of the strip position are carrying much less blade, it will be assumed that the strip analysis method is conservative for the purpose of the study.

In the strip analysis, the blade is designed to maximum creep and burst criteria which are defined by engine applications, i.e. at any radius on the blade strip, the radial stress is equal to the maximum allowable stress which is defined by the metal temperature at that radius, and the creep and burst criteria being used.

Thus the blade is stressed at all radii to the maximum allowable stress. The strip analysis includes the adverse effects of manufacturing limitations, such as minimum tip thickness, tolerances, wavelength of the fluctuation in tolerance and also allows for a minimum radius between two adjacent blades.

The initial part of the strip analysis defines the blade, then distributes the total centrifugal blade load as a rim load on the remaining hub disc, and calculates the maximum hub stress at the bore or centre.

Thus the maximum allowable hub stress is found from low cycle fatigue requirements and the rotor tip speed and R.I.T. (Rotor Inlet Temperature) and the solution of the bore stress relation takes the form shown in Fig. 4.2.

By varying hub radius (R_{HUB}), the curve 'a₁c₁b₁d₁' is generated, for the given running conditions.

Thus point 'b₁' defines the hub radius at which the bore or hub centre stress equals the allowable stress. Point 'a' is defined by the minimum arc between two adjacent blades.

Generally there exists a minimum stress point 'c₁', where the corresponding hub radius results in a minimum stress level at the bore or centre, this being the optimum R_{HUB} from the viewpoint of stress.

If the R.I.T. and tip speed are increased in a manner satisfying aerodynamic constraints a condition is reached where the minimum bore or hub centre stress is equal to the allowable stress

σ_{all} , as it is shown by the curve "a₂c₂b₂d₂" in Fig. 4.2. The corresponding R.I.T. and tip speed are then the maximum which will satisfy the allowable stress requirement. This results in one type of useful output data. Increasing R.I.T. in steps to the maximum value allows the graph shown in Fig. 4.3 to be plotted. This information allows the turbine designer to check quickly the

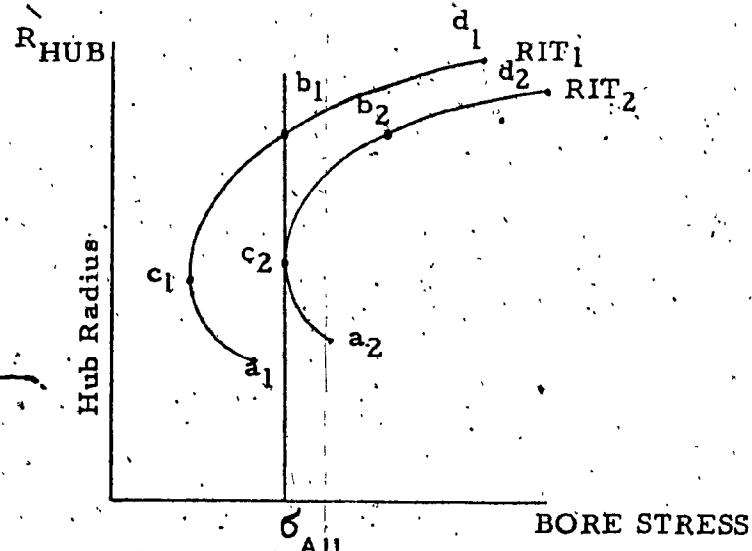


Figure 4.2

feasibility of his design. If the maximum allowable R. I. T. is below his engine requirement, then some further changes are needed to his fixed aerodynamic parameters. If, however the R. I. T. required is within the limits given by Fig. 4.3, then the designer may choose a value of R_H/R_T which satisfies the stress requirements and he is aware of the margin that exists if any, on allowable R. I. T.

Graphs like the one shown in Fig. 4.3 are automatically plotted by the computer program for different values of compressor specific speeds (NS_C), pressure ratio (PR_C) and low cycle fatigue (LCF) lives.

It should be noted that the rotor inlet temperature, R. I. T., considered in the stress calculations is the R. I. T. at stress condition. This is related to the aerodynamic R. I. T. as follows.

$$R. I. T. (\text{stress cond.}) = R. I. T. (\text{aero. cond.}) + \Delta T_u \text{ where}$$

ΔT_u is the temperature uptrim required to offset deterioration in engine performance and instrument error allowance. This distinction is made, since the aerodynamic R. I. T. defines the tip speed of the rotor (U_3), while in an actual engine the tip speed does not necessarily increase in the same manner as R. I. T. increases.

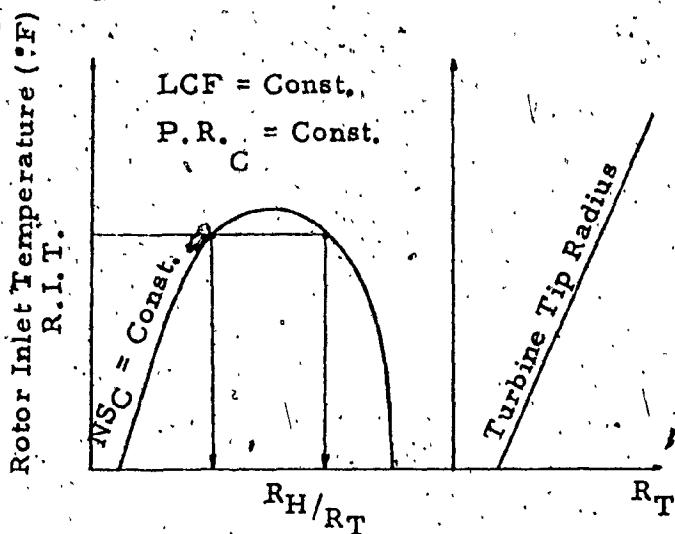


Figure 4.3

due to deterioration in engine performance. Thus the tip speed used in stress calculations is based on the aerodynamic R.I.T. and the stress condition R.I.T. is used to define metal temperature of the rotor blade. Allowances for changes in tip speed with engine deterioration are made with a tip speed correction factor ΔU , which is applied to the tip speed U_3 , calculated from the rotor inlet temperature at aero condition.

4.1 Theory to Calculate the Blade Thicknesses

At any section of the blade the stress due to centrifugal load is made equal to the allowable stress defined by the metal temperature at that radius, taking into account the adverse effect of tolerance on the blade above this section. The following parameters are defined at this point (see Fig. 4.4).

t_T - The nominal blade tip thickness. This is defined by the current state of technology, such as casting or machining limitations and allowances for erosion.

r , t - The transition radius and nominal thickness at that radius respectively. Transition radius occurs when the radial stress, under adverse tolerance, is equal to the allowable stress. Above the transition radius the blade is not stress limited.

WL - The "wave length", over which the blade profile may vary from the maximum to the minimum thickness tolerance. Defined by manufacturing limits.

TOL - The tolerance on blade thickness. The blade thickness is defined as the nominal thickness $t_r \pm TOL$.

The blade stress theory is presented in two parts, namely the theory of the tip region (defined as the blade area above the transition radius) and the second part covers the blade in the area below the transition radius.

4.1.1 Tip Region Analysis

From Fig. 4.4 the nominal blade thickness t_r is given by:

$$t_{r_i} = t_T + 2(r_T - r_i) \tan \phi \quad (4.1)$$

(for $r_T > r_i > r_o$)

$$\text{where } \tan \phi = \frac{TOL}{WL} \quad (4.2)$$

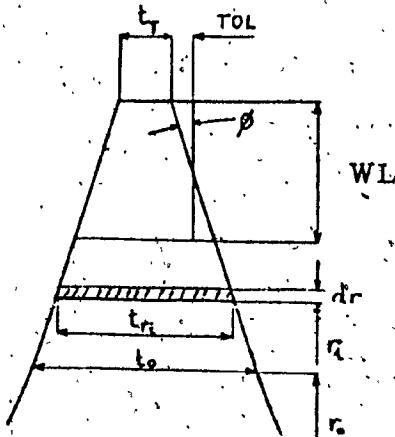


Figure 4.4

In equation 4.2 it is assumed that the blade thickness tolerance (TOL) and the tolerance wave length (WL) are selected such as to avoid reverse taper, not feasible for a cast turbine wheel and, in the case of a fully machined wheel, not desirable for vibration and manufacturing reasons. The tip thickness is generally selected to be as small as feasible for erosion and manufacturing reasons.

The equation to calculate the stress, due to the centrifugal force, at any radial station, for a unit width blade the thickness of which is varying linearly is derived as follows:

At radius r_i (see Fig. 4.5),
for a unit width, the elemental
centrifugal force (CF) is:

$$\overline{\delta CF} = \overline{\delta M} \omega^2 r_i \quad (4.3)$$

since:

$$M = \frac{\rho \cdot V}{g} \quad (4.4)$$

and:

$$V = t_{r_i} \cdot dr \cdot 1 = t_{r_i} \cdot dr \quad (4.5)$$

equation 4.3 becomes:

$$\overline{\delta CF} = \frac{\rho \omega^2}{g} \cdot (t_{r_i} \cdot dr) \cdot r_i \quad (4.6)$$

Using equation 4.1 and integrating we have:

$$\overline{CF}_{r_i} = \int_{r_i}^{r_T} \frac{\rho \omega^2}{g} \left[r_i (t_T + 2(r_T - r_i) \cdot \tan \theta) \right] dr \quad (4.7)$$

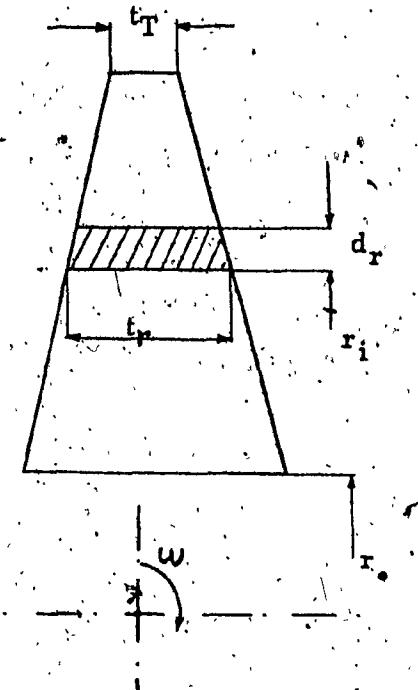


Figure 4.5.

or:

$$\overline{CF}_{r_i} = \frac{\rho \omega^2}{g} \int_{r_i}^{r_T} (r_i t_T + 2 r_i r_T \tan \phi - 2 r_i)^2 dr$$

$$\tan \phi) dr =$$

$$= \frac{\rho \omega^2}{g} \left[\frac{r_i^2}{2} t_T + r_i^2 \cdot r_T \tan \phi - \frac{2}{3} r_i^3 \tan \phi \right]$$

$$r_i^3 \tan \phi \left| \begin{array}{l} \\ \\ \end{array} \right. =$$

$$= \frac{\rho \omega^2}{6g} \left[3 r_T t_T^2 + 2 r_T^3 \tan \phi - 3 r_i^2 t_T^2 \right]$$

$$6 r_i^2 r_T \tan \phi + 4 r_i^3 \tan \phi \left] \begin{array}{l} \\ \\ \end{array} \right. =$$

$$= \frac{\rho \omega^2}{6g} \left[4 r_i^3 \tan \phi - r_i^2 (6 r_T \tan \phi + 3 t_T^2) + \right.$$

$$\left. r_T^2 (2 r_T \tan \phi + 3 t_T^2) \right] \quad (4.8)$$

The centrifugal stress at radius r_i is then:

$$\sigma_{r_i} (CF) = \frac{\overline{CF}_{r_i}}{\text{Area}} = \frac{\overline{CF}_{r_i}}{t_{r_i}} \quad (4.9)$$

Using eqs. 4.4 and 4.8 into 4.9 we have:

$$\sigma_{r_i} (CF) = \frac{\rho \omega^2 [4 r_i^3 \tan \phi - r_i^2 (6 r_T \tan \phi + 3 t_T^2)]}{6g [t_T + 2(r_T - r_i) \tan \phi]} +$$

$$+ r_T^2 (2 r_T \tan \phi + 3 t_T^2) \quad (4.10)$$

$$6g [t_T + 2(r_T - r_i) \tan \phi]$$

Equation 4.10 is valid for $r_i > r_*$ and for nominal thickness distribution.

If the effects of adverse tolerance on blade stress are included, equation 4.10 becomes:

$$\sigma_{SAT(CF)} \Big|_{r_i} = \frac{\rho \omega^2}{2g} (r_T^2 - r_i^2) \quad (4.11)$$

In fact, under adverse tolerance we have a constant thickness blade and $\tan \phi = 0$.

Equation 4.11 is valid only when:

$$r_i \geq r_* \text{ and } (r_T - r_i) \leq WL$$

It is now possible to define an adverse tolerance factor as follows:

$$C_3 = \frac{\sigma_{SAT(CF)} \Big|_{r_i}}{\sigma_{r_i} (CF)} \quad (4.12)$$

from which:

$$\sigma_{NOM} = \frac{\sigma_{All.}}{C_3} \quad (4.13)$$

Where $\sigma_{All.}$ is the allowable stress at radius r_i defined by the metal temperature $T_M(r_i)$.

When $(r_T - r_i) > WL$ the stress under adverse tolerance condition $\sigma_{SAT(CF)} \Big|_{r_i}$, is determined using eq. 4.10 replacing t_T with $(t_T + TOL)$ and r_i with $(r_i + WL)$ and adding the result to the value obtained using eq. 4.11 in which r_T has been replaced with $(r_i + WL)$. When $\sigma_{SAT(CF)} \Big|_{r_i} = \sigma_{All.} = f(T_m)$ the value of r_i determines the transition radius r_* .

4.1.2 Analysis Below Transition Radius r .

The blade nominal thickness, below the transition radius r , increases in a non linear fashion which is shown as follows:

Consider an element below the transition radius r as shown on Fig. 4.6.

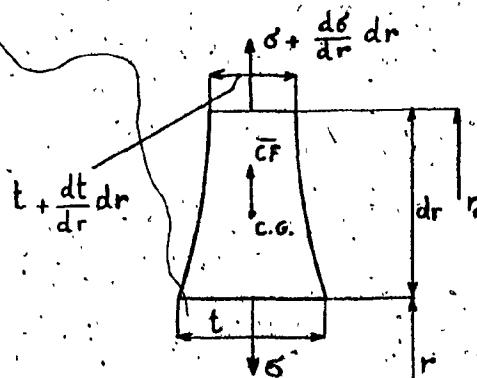


Figure 4.6

The force on the top surface neglecting 2nd order terms is:

$$(\delta + \frac{d\delta}{dr} \cdot dr) \cdot (t + \frac{dt}{dr} \cdot dr) \approx$$

$$\delta \cdot t + \delta \frac{dt}{dr} \cdot dr + t \frac{d\delta}{dr} \cdot dr \quad (4.14)$$

The force on the lower surface is:

$$-\delta \cdot t \quad (4.15)$$

The centrifugal force \overline{CF} on the element neglecting 2nd and 3rd order terms is:

$$\frac{\rho \omega^2}{2g} \cdot \left[t + t + \frac{dt}{dr} \cdot dr \right] \cdot dr \cdot \frac{(2r + dr)}{2} \approx$$

$$\frac{\rho \omega^2}{2g} \cdot t \cdot r \cdot dr \quad (4.16)$$

Summing all forces and equating to zero,

$$\sigma \frac{dt}{dr} + t \cdot \frac{d\sigma}{dr} + \frac{\omega^2 \rho}{g} \cdot t \cdot r = 0 \quad (4.17)$$

Since the allowable stress $\sigma_{All.}$ is a function of the metal temperature T_m , and T_m is a function of radius, we can express $\sigma_{All.}$ as a function of radius.

It is now required to find at each radius r_i the value of t_{r_i} which gives a stress equal the the allowable.

If equation 4.17 is rewritten as follows:

$$\sigma \frac{dt}{dr} + t \left(\frac{r\omega^2 \rho}{g} + \frac{d\sigma}{dr} \right) = 0 \quad (4.17a)$$

The equation can be stated in finite different form with reference to the indices shown in Fig. 4.7 by replacing:

$$\sigma \text{ by } \frac{1}{2} (\sigma_i + \sigma_{i+1}) \quad (4.18)$$

$$t \text{ by } \frac{1}{2} (t_i + t_{i+1}) \quad (4.19)$$

$$\frac{d\sigma}{dr} \text{ by } \frac{(\sigma_i - \sigma_{i+1})}{r_i - r_{i+1}} \quad (4.20)$$

$$\frac{dt}{dr} \text{ by } \frac{(t_i - t_{i+1})}{r_i - r_{i+1}} \quad (4.21)$$

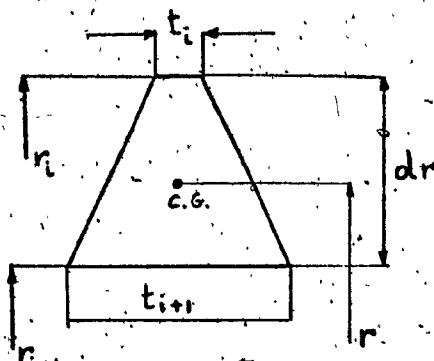


Figure 4.7

The radius r can be replaced by the radius of the centre of gravity of the element. Assuming linearity in t between the two sections gives (See Fig. 4.7):

$$\begin{aligned}
 r &= r_{i+1} + \frac{(t_{i+1} + 2t_i) dr}{3(t_{i+1} + t_i)} = \\
 &= \frac{3r_{i+1}(t_{i+1} + t_i) + (t_{i+1} + 2t_i)(r_i - r_{i+1})}{3(t_{i+1} + t_i)} = \\
 &= \frac{t_i(r_{i+1} + 2r_i) + t_{i+1}(r_i + 2r_{i+1})}{3(t_{i+1} + t_i)} \quad (4.22)
 \end{aligned}$$

Substituting eqs. 4.18 through 4.22 into eq. 4.17 a gives:

$$\begin{aligned}
 &\frac{1}{2}(\delta_i + \delta_{i+1}) \left(\frac{t_i - t_{i+1}}{r_i - r_{i+1}} \right) + \frac{1}{2}(t_i + t_{i+1}) \cdot \\
 &\left[\frac{t_i(r_{i+1} + 2r_i) + t_{i+1}(r_i + 2r_{i+1})}{3(t_i + t_{i+1})} \omega^2 p + \right. \\
 &\left. + \frac{(\delta_i - \delta_{i+1})}{(r_i - r_{i+1})} \right] = 0
 \end{aligned}$$

or finally:

$$t_{i+1} = t_i \left[\frac{\frac{6\delta_i}{g} + \frac{\omega^2 p}{g} (2r_i^2 - r_i r_{i+1} - r_{i+1}^2)}{\frac{6\delta_{i+1}}{g} - \frac{\omega^2 p}{g} (r_i^2 + r_i r_{i+1} - 2r_{i+1}^2)} \right] \quad (4.23)$$

Equation 4.23 is valid when the two conditions below are

satisfied:

$$r_{i+1} \leq r_i \text{ and } (r_T - r_i) \leq WL$$

The first time $r_{i+1} \leq r_i$, the value for σ_i is replaced by the value relative to r_i and t_i is replaced with $t_T + TOL$.

To this point the blade thicknesses calculated include the allowance for tolerance. The nominal thickness is derived as follows: (See Fig. 4.8)

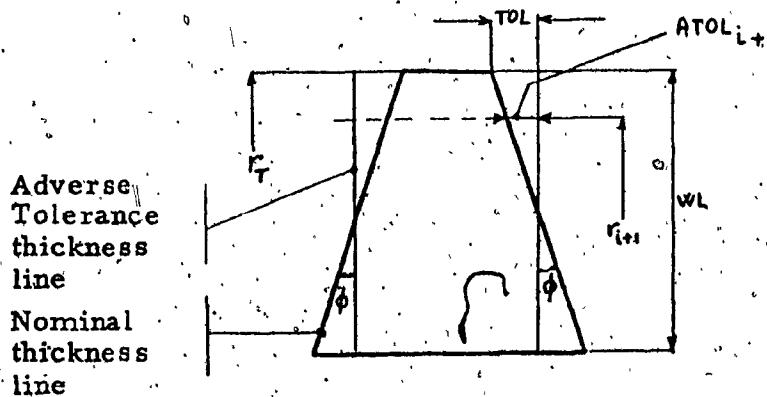


Figure 4.8

$$t_{N_{i+1}} = t_{i+1} + ATOL_{i+1} \quad (4.24)$$

$$\text{where } ATOL_{i+1} = 2(r_T - \frac{WL}{2} - r_{i+1}) \tan \phi \quad (4.25)$$

The nominal stress at each radial station r_{i+1} are obtained using equation 4.23 as follows:

$$\begin{aligned} \sigma_{N_{i+1}} &= \frac{t_{N_i}}{t_{N_{i+1}}} \left[\sigma_{N_i} + \frac{\rho u^2}{6g} (2r_i - r_i r_{i+1} - r_{i+1}^2) \right. \\ &\quad \left. + \frac{\rho u^2}{6g} (r_i^2 + r_i r_{i+1} - 2r_{i+1}^2) \right] \quad (4.26) \end{aligned}$$

The first time $r_{i+1} \leq r_*$, the values of t_{N_i} and σ_{N_i} are those corresponding to $r_i = r_*$ while $t_{N_{i+1}}$ is the nominal thickness at the $(i+1)^{th}$ station.

The result obtained by using the above set of equations is that at each radial station the blade stress under adverse tolerance condition (σ_{SAT}) will be equal to the allowable stress ($\sigma_{All.}$) defined by the blade metal temperature at that radius. The calculation proceeds using values of the preceding station until $(r_T - r_{i+1}) = WL$.

When $(r_T - r_{i+1}) > WL$ each blade section is also loaded by the centrifugal forces CF, CF1, and CF3 under adverse tolerance conditions as shown in Fig. 4.9.

CF, CF1 and CF3
are calculated
using eqs. 4.10
and 4.11 as
follows:

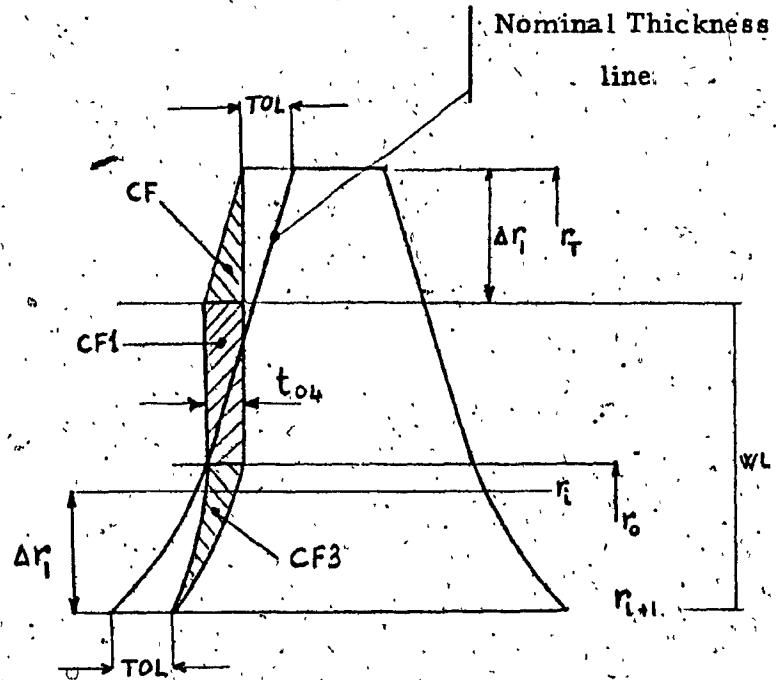


Figure 4.9

$$CF = \frac{\rho w^2}{6g} \left[4r_3^2 \tan \phi - 6r_3^2 r_{T_1} \tan \phi + \right. \\ \left. + 2r_{T_1}^3 \tan \phi \right] \quad (4.27)$$

$$CF_1 = \frac{\rho w^2}{2g} \left[r_3^2 - r_{T_4}^2 \right] \cdot t_{o4} \quad (4.28)$$

$$CF_3 = \frac{\rho w^2}{6g} \left[4r_{i+1}^3 \tan \phi - 6r_{i+1}^2 r_{T_4} \tan \phi \right. \\ \left. + 2r_{T_4}^3 \tan \phi \right] \quad (4.29)$$

where:

$$r_3 = r_{i+1} + WL \quad (4.30)$$

$$r_{T_1} = r_{i+1} + WL + \Delta r_i \quad (4.31)$$

$$r_{T_4} = r_{i+1} + \Delta r_i \quad (4.32)$$

$$t_{o4} = (r_{T_1} - r_3) \tan \phi \quad (4.33)$$

Thus the total extra centrifugal load is:

$$CF_2 = 2 (CF + CF_1 + CF_3) \quad (4.34)$$

Therefore the extra blade thickness necessary to keep the blade stresses equal to the allowable strength under adverse tolerance condition is:

$$t_{i+1} \text{ (EXTRA)} = \frac{CF2}{\sigma_{\text{All}}} \quad (4.35)$$

This thickness is added to the thicknesses obtained from equation 4.23 and therefore the nominal thicknesses become:

$$t_{Ni+1} = t_{i+1} + t_{i+1} \text{ (EXTRA)} + \text{TOL} \quad (4.36)$$

4.1.3 Minimum Arc Between Adjacent Blades

As the calculation of the blade thickness, for decreasing radius r_i , proceeds, the distance between adjacent blades decreases. The blade thickness increases while the circle perimeter decreases; thus for a chosen number of blades, at a certain point during the analysis, the blade profiles will intersect. This is not acceptable and therefore the arc distance between adjacent blades is checked against a minimum acceptable value and the analysis is stopped at that point which becomes the root of the blade.

Figure 4.10 shows graphically how the problem is solved.

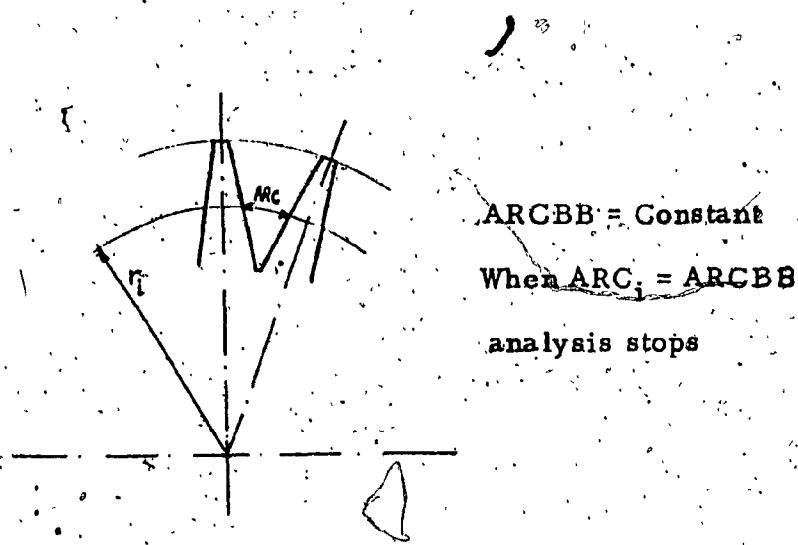


Figure 4.10

The value of ARC is defined by fatigue, machining or casting, and aerodynamic requirements.

4.2 Analysis of the Radial Turbine Hub Region

A low cycle fatigue (LCF) life of at least 10,000 cycles is usually required from a rotating component like a radial turbine. Therefore the stress level at the bore or centre (solid hub) of the rotor hub must be maintained at a reasonable stress level with an upper limit around the yield point. At this stress level plastic redistribution of stresses takes place in a mild form and therefore an elastic analysis is used for the purposes of this program.

The elastic analysis is carried out for a critical section (see Fig. 4.11) and no shear interaction is taken into account between adjacent strips.

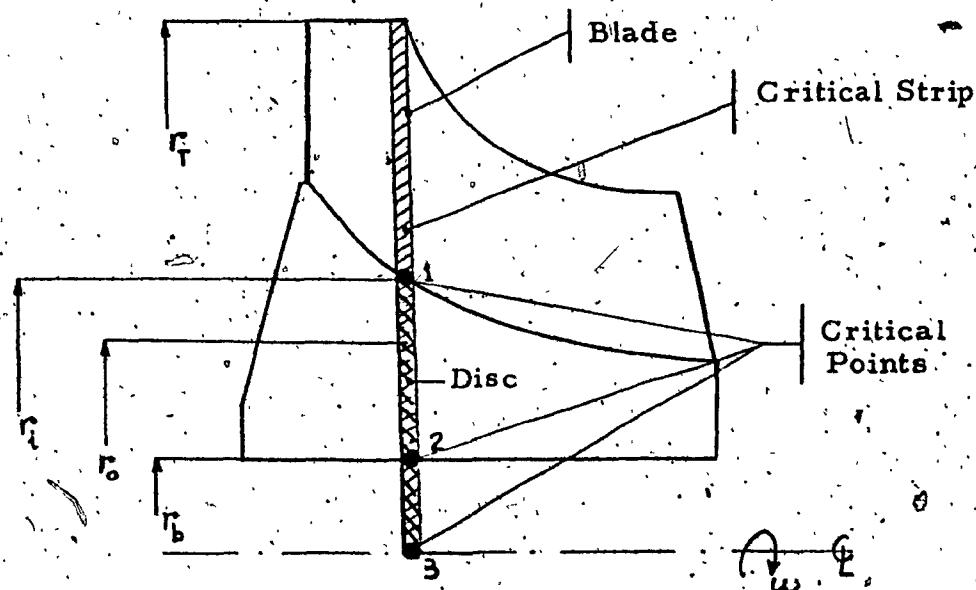


Figure 4.11

A correction factor, however, has been computed from similar wheels analyzed using a finite element computer program. Radial turbine rotors, with varying numbers of blades, with or without bored hubs, have been analyzed for stress and compared to that predicted using the method described here. The results of these two analyses permit the derivation of correlations curves which relate strip stress analysis to complete finite element stress analysis (see section 4.3). The factor is defined as:

$$\text{BSF (bore stress factor)} = \frac{\text{CF stress at bore or hub centre from strip analysis}}{\text{CF stress at bore or hub centre from finite element analysis}}$$
(4.37)

The correlation curves so derived can be used for parametric studies of new engines where geometrically similar radial turbine rotors are employed.

The effect of the thermal gradient between the bore and the rim of the hub is taken into account in the stress calculation. A linear gradient is included in the model.

4.2.1 Analysis of a Thin Disc with Solid Hub

a) Centrifugal Stresses

The tangential and radial stresses at radius r from the center of a circular disc of constant thickness and rim radius r_i , and density ρ , rotating about its axis with a uniform angular velocity ω (see Fig. 4.11), are given by the following equations (Ref. 1):

$$\sigma_r = \frac{1}{8} \frac{\rho \omega^2}{386.4} \cdot [(3+\nu) (r_i^2 - r_o^2)] \quad (4.38)$$

$$\sigma_t = \frac{1}{8} \frac{\rho \omega^2}{386.4} \cdot [(3+\nu) r_i^2 + (1+3\nu) r_o^2] \quad (4.39)$$

The maximum radial and tangential stresses occur at the centre ($r_o = 0$) and eqs. 4.38 and 4.39 become:

$$\sigma_{r_i} = \sigma_{t_i} = \frac{1}{8} \frac{\rho \omega^2}{386.4} (3+\nu) r_i^2 \quad (4.40)$$

b) Thermal Stresses

Taking a thin disc and applying a linear thermal gradient between the centre and the rim (Fig. 4.12), the tangential thermal stresses at any radius r can be determined using the following equation (Ref. 2):

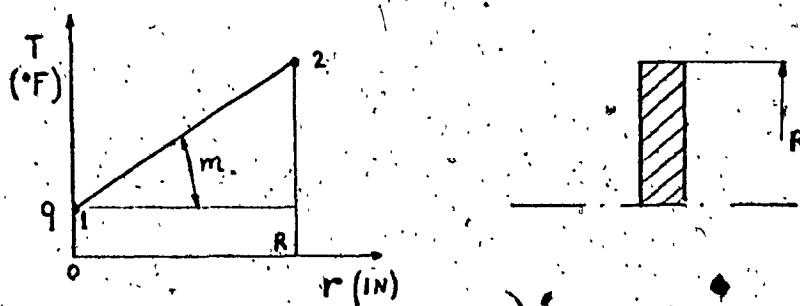


Figure 4.12

$$\sigma_{T,t} = \alpha E (-T + \frac{1}{R^2} \int_0^R Tr dr) + \frac{1}{r^2} \quad (4.41)$$

Where $T = f(r)$

A linear variation of the temperature T takes the form:

$$T = mr + q \quad (4.42)$$

Where:

$$m = \frac{T_2 - T_1}{r_2 - r_1} \quad (4.43)$$

$$q = T_1 + \frac{T_2 - T_1}{r_2 - r_1} \cdot r_1 \quad (4.44)$$

Substituting eq. 4.42 into eq. 4.41 gives:

$$\begin{aligned} \delta T_{t, t} &= \alpha E \left[- (mr + q) + \frac{1}{R^2} \int_0^R (mr + q) r dr + \right. \\ &\quad \left. + \frac{1}{r^2} \int_0^R (mr + q) r dr \right] = \frac{\alpha E m}{3} (R - 2r) \quad (4.45) \end{aligned}$$

The the centre of a solid hub disc $r = 0$ and using eq. 4.43
noting that $r_2 - r_1 = R$ gives from eq. 4.45;

$$\delta T_{t, t} = \frac{\alpha E}{3} (T_2 - T_1)$$

and putting $T_2 - T_1 = \Delta T$

$$\delta T_{t, t} = \frac{\alpha E \Delta T}{3} \quad (4.46)$$

Equation 4.46 gives the tangential thermal stress at the centre of a flat disc due to a linear temperature gradient between the hub centre and the hub rim.

c) Stress due to the Blade Centrifugal Force

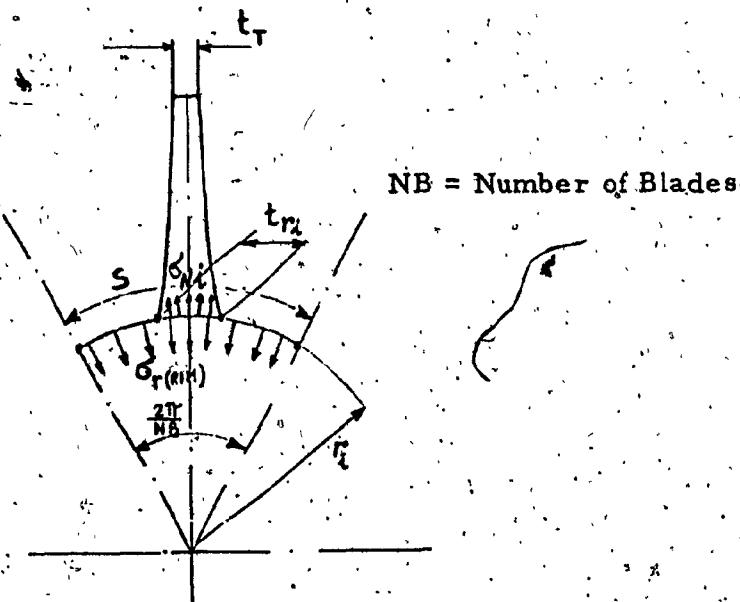


Figure 4.13

The condition for static equilibrium of radial stresses between the disc and the attached blades is given by (see Fig. 4.13):

$$\sigma_{r_i} \text{ (RIM)} \cdot s_i = t_{r_i} \cdot \sigma_{All.} \quad (4.47)$$

where:

$$s_i = r_i \left(\frac{2\pi}{NB} \right) \quad (4.48)$$

Eq. 4.48 into eq. 4.47 and solving for σ_{r_i} (RIM) gives:

$$\sigma_{r_i}(\text{RIM}) = \frac{\text{NB. } t_{r_i}}{2\pi r_i} \cdot G_{\text{All.}} \quad (4.49)$$

Then assuming the disc to be similar to a cylindrical pressure vessel under an external radial pressure σ_{r_i} (RIM).

Figure 4.14 below and two equations from reference 1 allow the following derivation for the tangential stress at the centre of the disc.

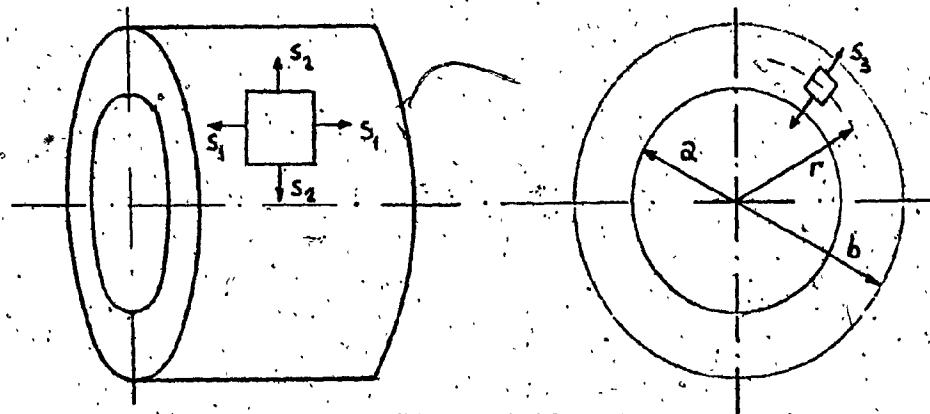


Figure 4.14

$$\sigma_1 = 0$$

$$\sigma_2 = -p \frac{b^2(a^2 + r^2)}{r^2(b^2 - a^2)} \quad * \text{Tangential stress}$$

$$\sigma_3 = p \frac{b^2(r^2 - a^2)}{r^2(b^2 - a^2)} \quad * \text{Radial stress}$$

For $a = 0$ and $r = b$ we have at the rim:

$$\sigma_2 = -p \text{ and } \sigma_3 = p$$

also for $a = 0$ and $r = 0$ at the centre of the disc:

$$\sigma_2 = -p \text{ and } \sigma_3 = p$$

and since the boundary condition here gives $\sigma = \sigma_{r_i}$ (RIM) the tangential stress at the centre of the disc due to the blade force for any radius r_i is:

$$\sigma_{t_i} (\text{CENTRE}) = \frac{\text{NB. } \sigma_{r_i}}{2\pi \cdot r_i} \cdot \sigma_{\text{All.}} \quad (4.50)$$

4.2.2 Analysis of a Thin Disc with a Bored Hub

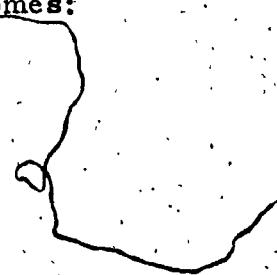
a) Centrifugal Stresses

The tangential and radial stresses at any radius r_i from the centre of a circular disc of constant thickness of radius r_i and density ρ , with a central hole of radius r_b , rotating about its axis with a uniform angular velocity ω (see Fig. 4.11), are given by the following equations (Ref. 1):

$$\sigma_{r_i} = \frac{3 + \gamma}{8} \cdot \frac{\rho \omega^2}{386.4} \cdot \left(\frac{r_i^2 + r_b^2}{r_i^2} - \frac{r_i^2 - r_b^2}{r_i^2} \right) \quad (4.51)$$

$$\sigma_t = \frac{1}{8} \cdot \frac{\rho \omega^2}{386.4} \cdot \left[(3 + \gamma) \cdot \left(\frac{r_i^2 + r_b^2}{r_i^2} + \frac{r_i^2 - r_b^2}{r_i^2} \right) - (1 + 3\gamma) \cdot \frac{r_i^2}{r_b^2} \right] \quad (4.52)$$

The maximum tangential stress occurs at the perimeter of the hole ($r_i = r_b$) and eq. 4.52 becomes:



$$\sigma_t = \frac{1}{4} \cdot \frac{\rho \omega^2}{386.4} \cdot \left[(3 + \nu) r_i^2 + (1 - \nu) r_b^2 \right] \quad (4.53)$$

While equation 4.51 gives:

$$\sigma_r = 0 \quad (4.54)$$

a) b) Thermal Stresses

Taking a thin disc of radius R_R with a central hole of radius R_B (see Fig. 4.15) applying a linear thermal gradient between the bore and the rim, the tangential thermal stress at any radius r is given by (Ref. 2):

$$\sigma_{T, t} = \frac{\alpha E}{r^2} \cdot \left(\frac{r^2 + R_B^2}{R_R^2 - R_B^2} \right)^2 \cdot \int_{R_B}^{R_R} T \cdot r \cdot dr + \int_{R_B}^r \left(T \cdot r \cdot dr - T \cdot r^2 \right) \quad (4.55)$$

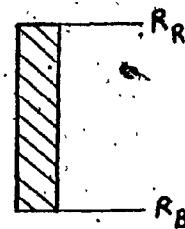
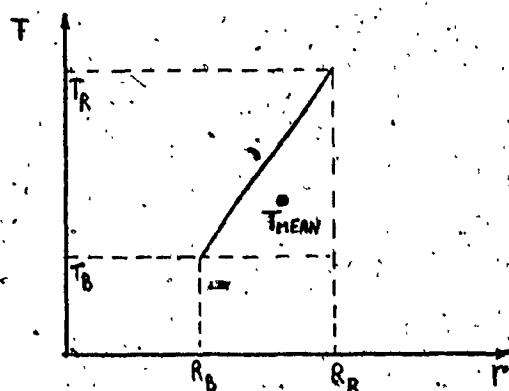


Figure 4.15

The tangential thermal stress at $r = R_B$ can be derived more simply as follows (rather than integrating eq. 4.55):

$$\sigma_{T, t} = \alpha E (T_{MEAN} - T_B) \quad (4.56)$$

Where:

$$T_{MEAN} = \frac{\int_{R_B}^{R_R} T \cdot r \cdot dr}{\int_{R_B}^{R_R} r \cdot dr}$$

$$= \frac{T_B \cdot \int_{R_B}^{R_R} r \cdot dr + \int_{R_B}^{R_R} (T - T_B) r dr}{\int_{R_B}^{R_R} r dr}$$

$$= T_B + (T_R - T_B) \cdot \left[\frac{2}{3} - \frac{R_B}{3(R_R + R_B)} \right] \quad (4.57)$$

Equation 4.57 into eq. 4.56 gives:

$$\sigma_{T, t} = \alpha E (T_R - T_B) \cdot \left[\frac{2}{3} + \frac{R_B}{3(R_R + R_B)} \right] \quad (4.58)$$

Equation 4.58 gives the tangential thermal stress at the bore of a flat disc due to a linear temperature gradient between the bore and the rim of the hub.

c) Stress due to Blade Centrifugal Force

The same approach used in 4.2.1 c) is followed yielding the equation that gives the tangential stress at the disc bore due to the blade pull:

$$\sigma_{t_i} (\text{BORE}) = \frac{\text{NB} \cdot t_i}{2\pi r_i} \cdot \sigma_{\text{All.}} \cdot \left(\frac{R_B^2 + R_R^2}{2(R_R - R_B)} \right) \quad (4.59)$$

Where:

$$\frac{R_R}{R_B} = \frac{r_i}{r_o}$$

4.2.3 Total Tangential Stresses at the Centre or at the Bore of the Hub and Their Calibration

The preceding analysis has been for a unit strip, at a section near to that of maximum stress, and does not include the interaction effect of less loaded sections of a typical radial turbine.

Since this interaction effect results in considerably reduced stresses from those evaluated by the unit strip analysis, it is necessary to carry out detailed stress analysis for various configurations of radial turbines, to determine stress factors which will allow accurate results to be obtained using the simpler strip analysis.

$$\text{Bore Stress Factor (BSF)} = \frac{\sigma_{\text{HUB}}}{\sigma_{\text{TRUE}}} \quad (4.60)$$

Where:

σ_{HUB} = Eq. 4.40 + Eq. 4.50 (4.61) for a solid hub rotor

or:

σ_{HUB} = Eq. 4.53 + Eq. 4.59 (4.62) for a bored hub rotor

and:

σ_{TRUE} = Maximum effective elastic stress at the hub centre or bore, due to CF only from a detailed stress analysis using a finite element computer program.

The total tangential stress at the centre or at the bore of the hub then becomes using eq. 4.60:

$$\sigma_{TRUE} = \frac{\sigma_{HUB}}{BSF} + \sigma_{T,t} \quad (4.63)$$

Where σ_{HUB} is given by eq. 4.61 or 4.62 and $\sigma_{T,t}$ is given by eq. 4.46 or 4.58 depending whether the disc is solid or bored.

The assumption that $\sigma_{T,t}$ is not affected by the BSF may be justified from the consideration that in a radial turbine the metal temperature is a single valued function of radius and it is not substantially affected by axial position.

4.3 Bore Stress Factor Derivation

The Bore Stress Factor (BSF) for high specific speed radial turbines is lower than that for low specific speed ones.

Similarly the B.S.F. for a solid hub is lower than that for a bored hub wheel and for low number of blades as against a high number of blades in the rotor.

For geometrically similar radial turbines, these statements are confirmed by work reported on a complete finite element analyses in Ref. 7 and 8. The final curve from Ref. 7 and 8 is shown in Fig. 4.16 while the results of three finite elements stress analysis used to construct Fig. 4.16 are shown in Fig. 4.17 through Fig. 4.20.

The usefulness of the method described in this thesis depends largely on the degree of consistency between the B.S.F. and the type of radial turbine wheel being designed (which will required continuing experiment and analytical evidence from turbine designs).

BORE STRESS FACTOR (BSF)
VS.
TURBINE SPECIFIC SPEED (NST)

$$BSF = \frac{\sigma_{HUB}}{\sigma_{TRUE}}$$

σ_{HUB} = HUB STRESS AT THE BORE
FROM STRIP ANALYSIS DUE

TO CF ONLY.
TO CF ONLY.
 σ_{TRUE} = MAX. EFFECTIVE ELASTIC
STRESS AT THE BORE, DUE
TO CF ONLY, FROM FINITE
ELEMENT STRESS ANALYSIS.

X BORED HUB - 12 BLADES
 △ " " - 9 BLADES
 ○ SOLID HUB - 12 BLADES
 □ " " - 9 BLADES

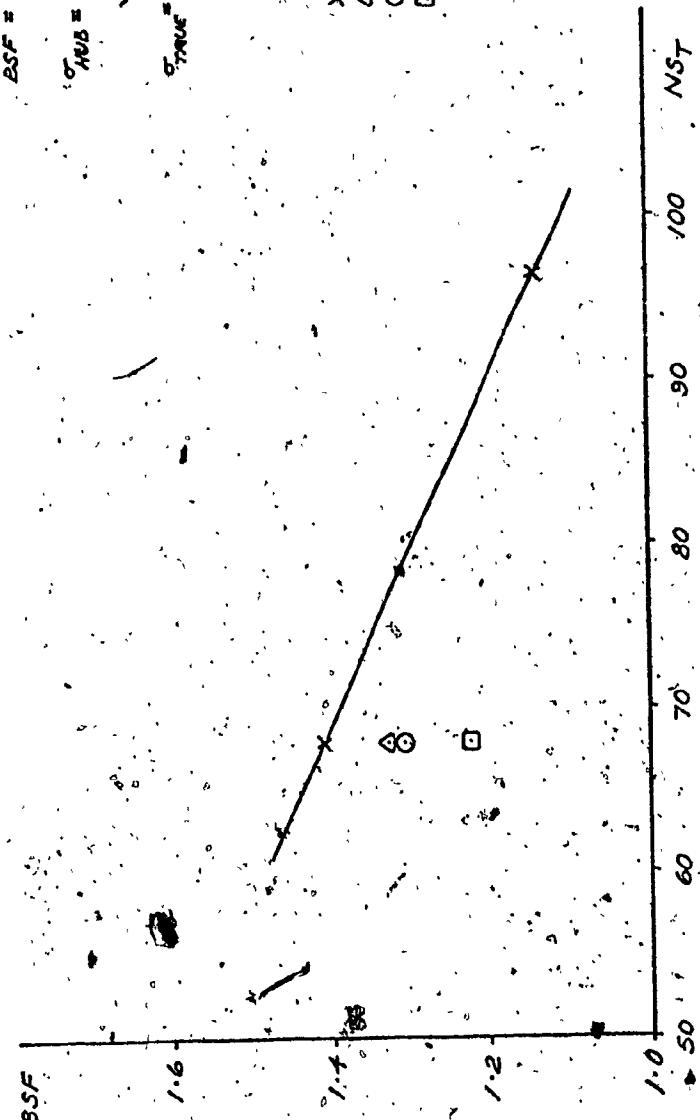


FIG. 4.16

RADIAL TURBINE

- BLADE MODIFIED TO GIVE BETTER GAS PASSAGE.
- EFFECTIVE ISOSTRESS LINES AT 60,150 R.P.M. (K.S.I.)
- ELASTIC SOLUTION.
- SOLID WHEEL.
- R.I.T. = 1975°F
- CENTRIFUGAL STRESS ONLY.
- NUMBER OF BLADES = 12.

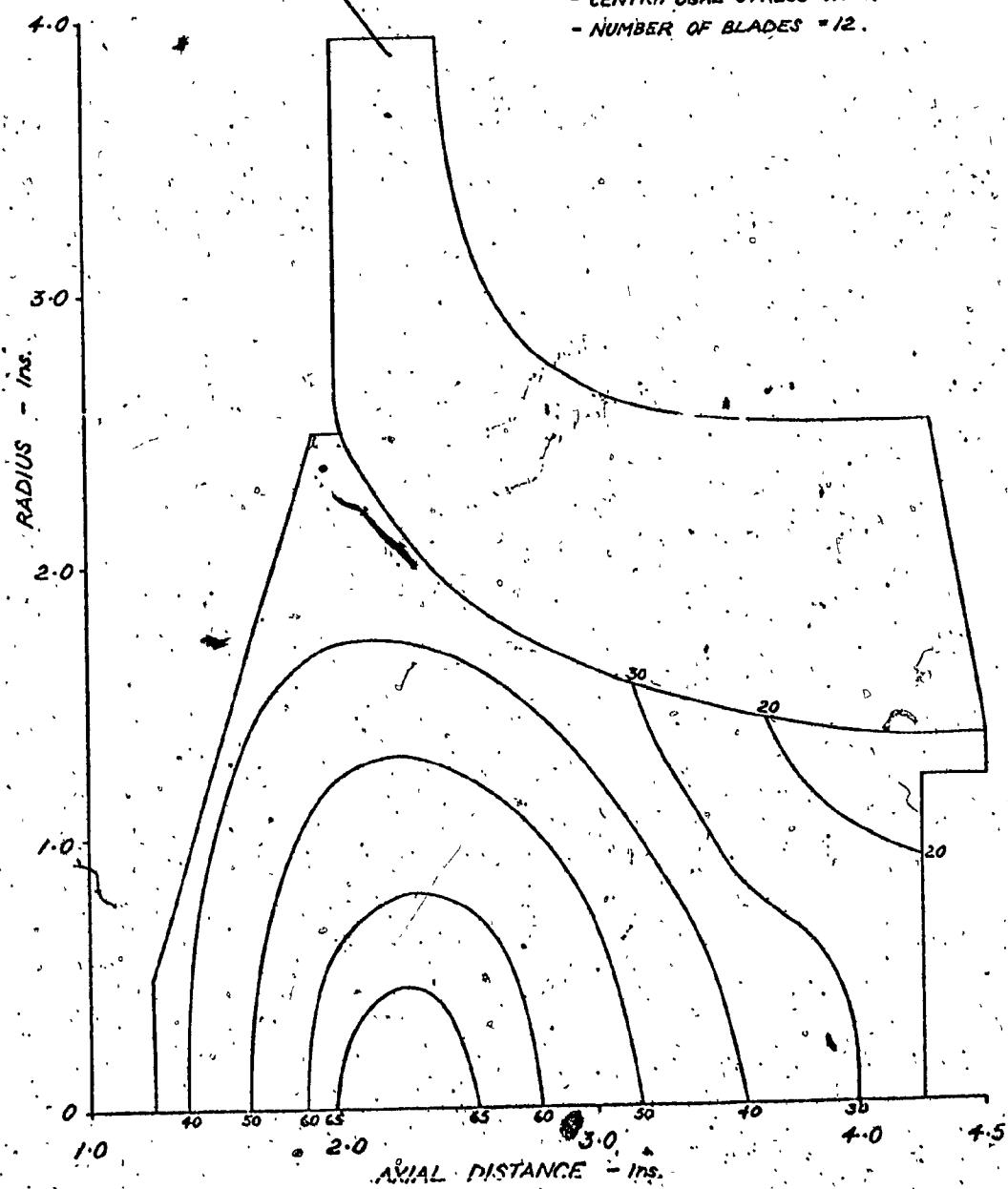


FIG. 4.17

RADIAL TURBINE

- BLADE MODIFIED TO GIVE BETTER GAS. PASSAGE.
- EFFECTIVE ISOSTRESS LINES AT 60, 150 R.P.M. (K.S.I.)
- ELASTIC SOLUTION.
- BORED WHEEL.
- R.I.T. = 1975 °F
- CENTRIFUGAL STRESS ONLY.
- NUMBER OF BLADES = 12.

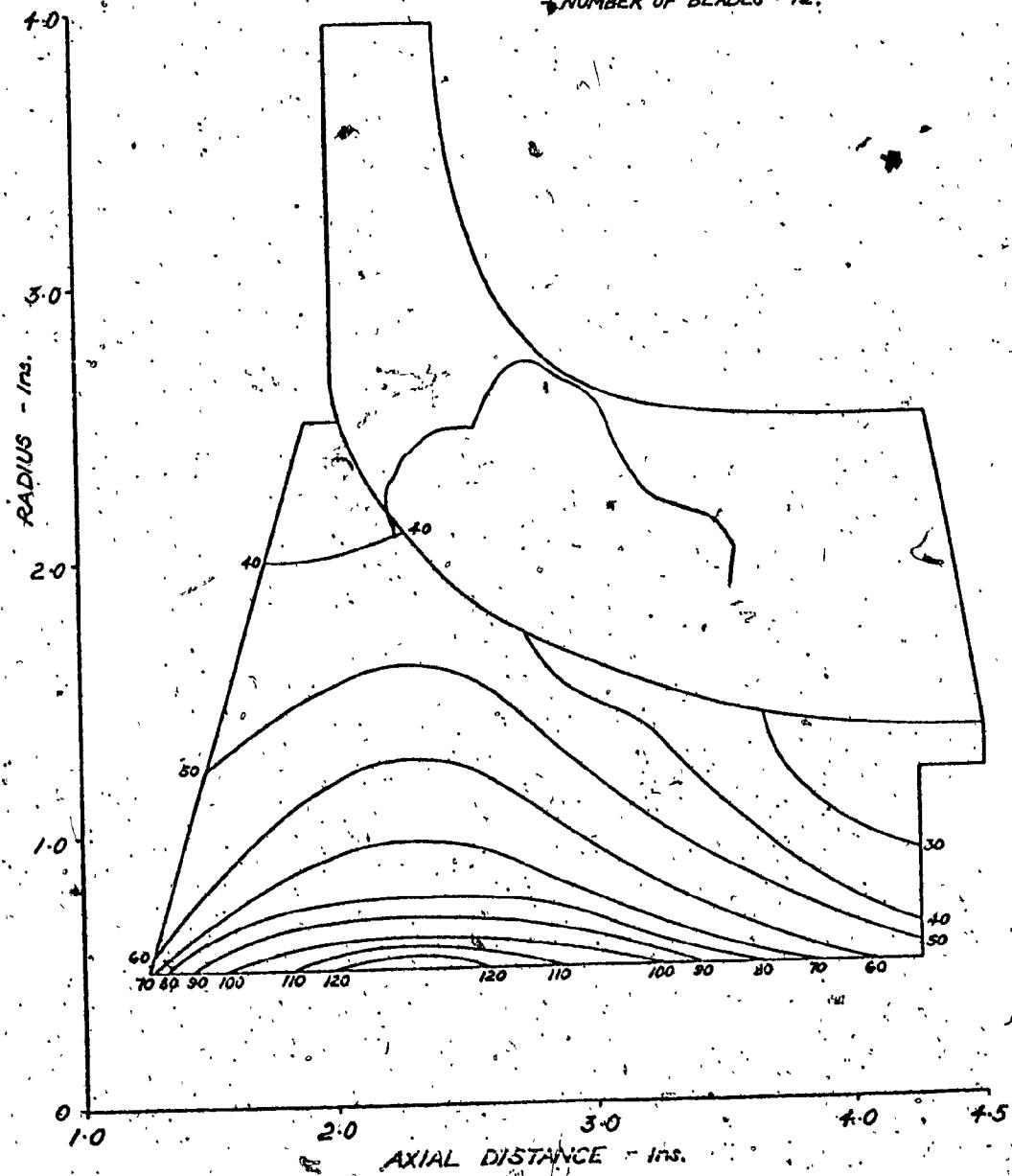


FIG. 4.18.

GAS GENERATOR TURBINE ROTOR
STRESS ANALYSIS GRID

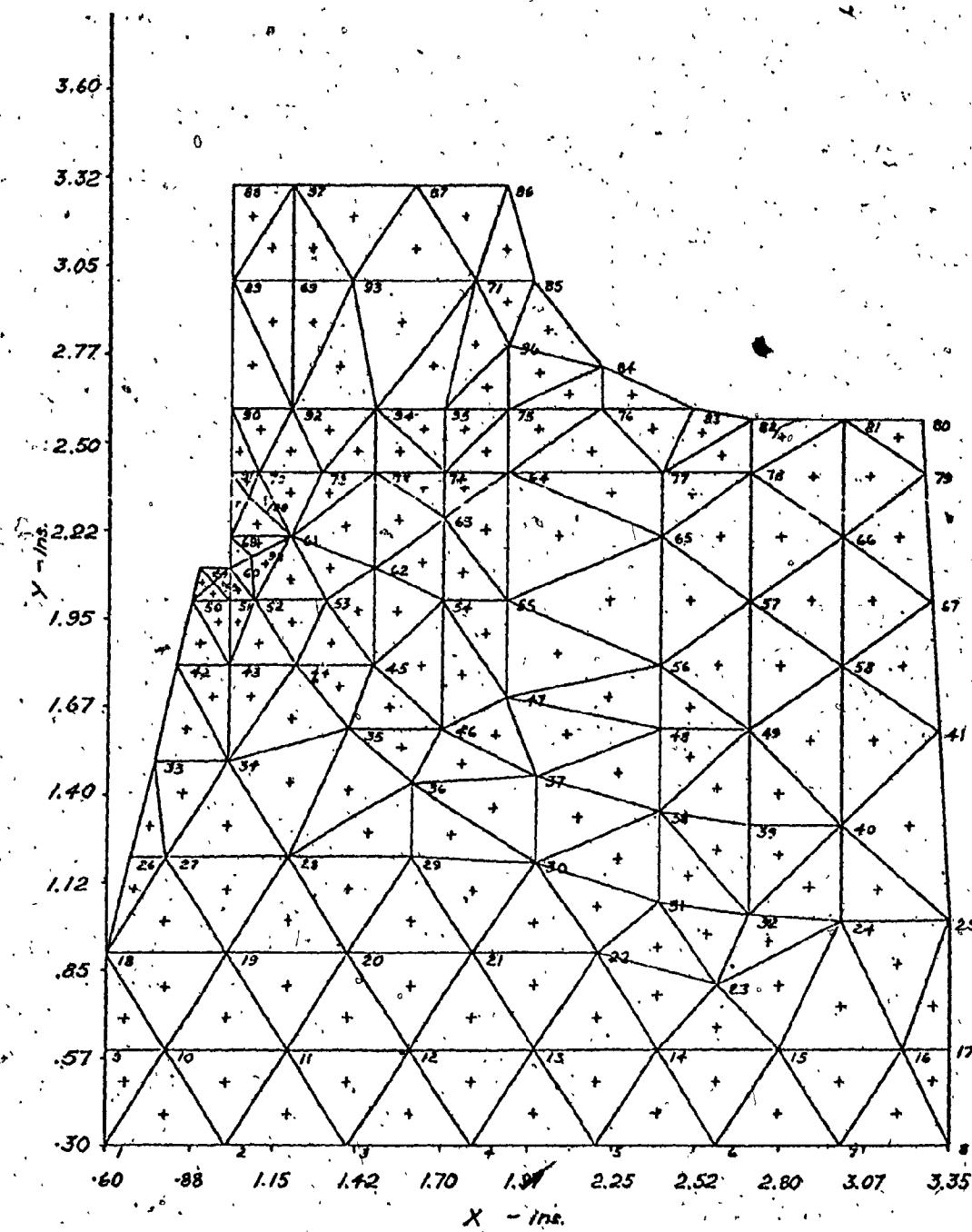


FIG. 4.19.

GAS GENERATOR TURBINE HUB
EFFECTIVE STRESS LINES

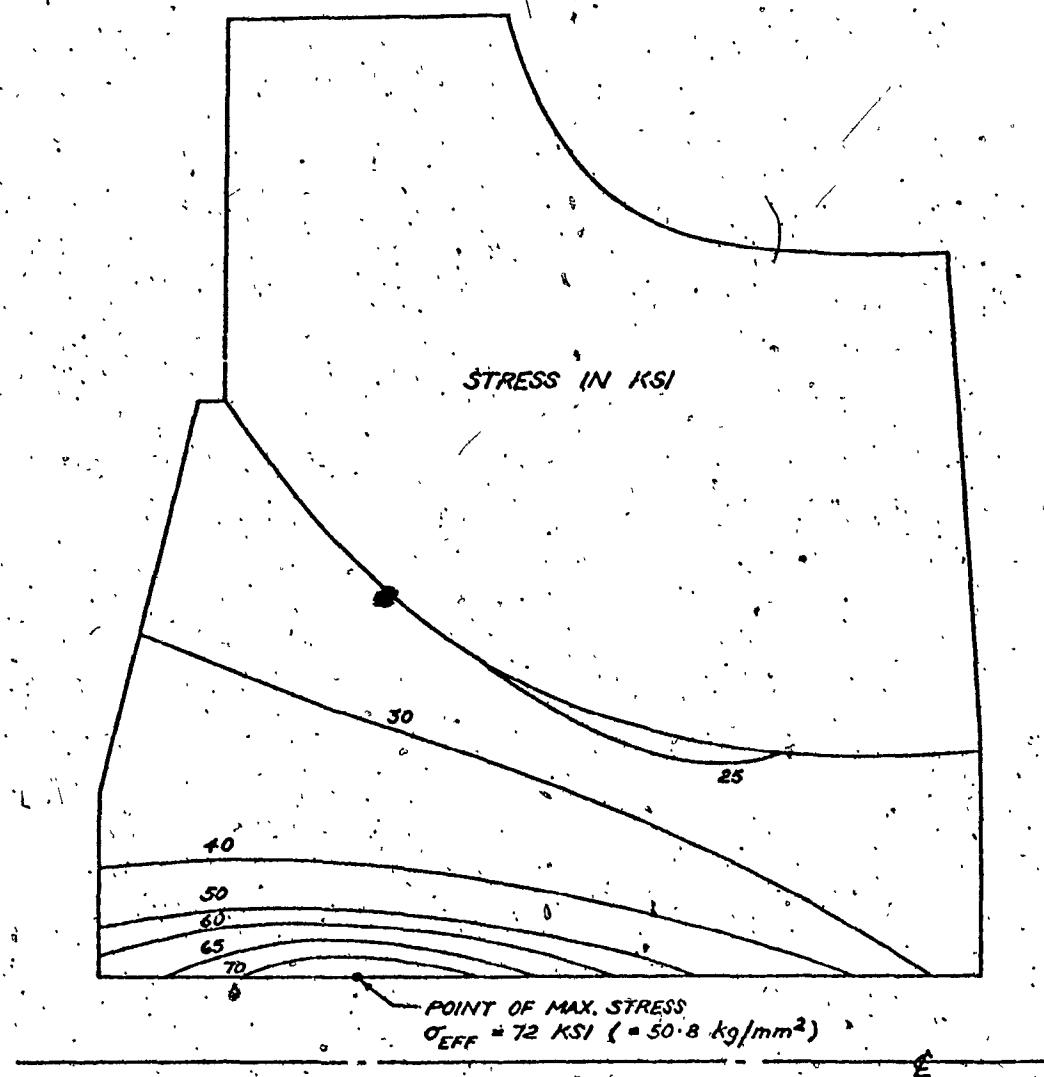


FIG. 4.20.

CHAPTER 5

COMPRESSOR AERODYNAMICS

The compressor data required, primarily depends on the level of compressor technology being considered. This technology may be described in the following manner:

- a) Compressor adiabatic efficiency versus compressor pressure ratio (see Fig. 5.1 a)
- b) Compressor efficiency decrement versus compressor specific speed for various compressor pressure ratios (see Fig. 5.1 b)

These relationships are generally based on test results and thus allow a simple modification of the efficiency in basic adiabatic compressor theory to account for the efficiency of a feasible compressor.

At the outset of an engine design the following values are generally established:

W_c = Compressor mass flow (lb/sec)

N_c = Compressor specific speed

PR_c = Compressor pressure ratio

P_1 = Compressor inlet pressure (psia)

T_o = Compressor inlet temperature ($^{\circ}$ F)

Basic compression theory then provides the following values:

$\eta_{c,a}$ = Compressor adiabatic efficiency

T_2 = Compressor delivery temperature ($^{\circ}$ F)

N_c = Compressor mechanical speed (RPM)

COMPRESSOR EFFICIENCY

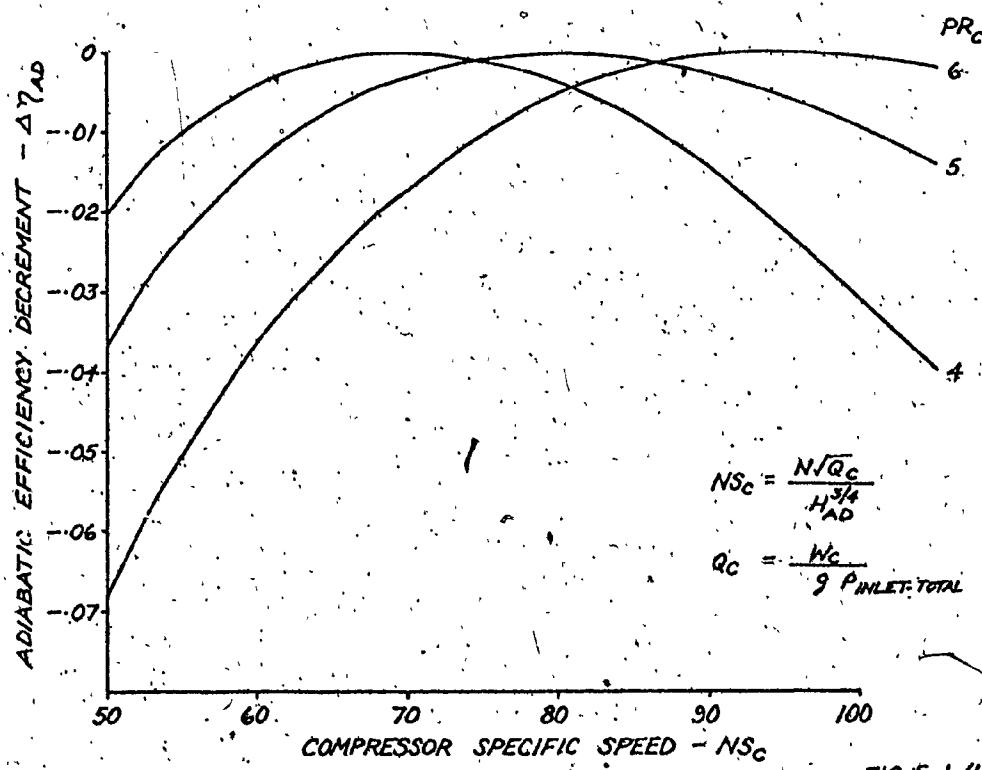


FIG. 5.1. (b)

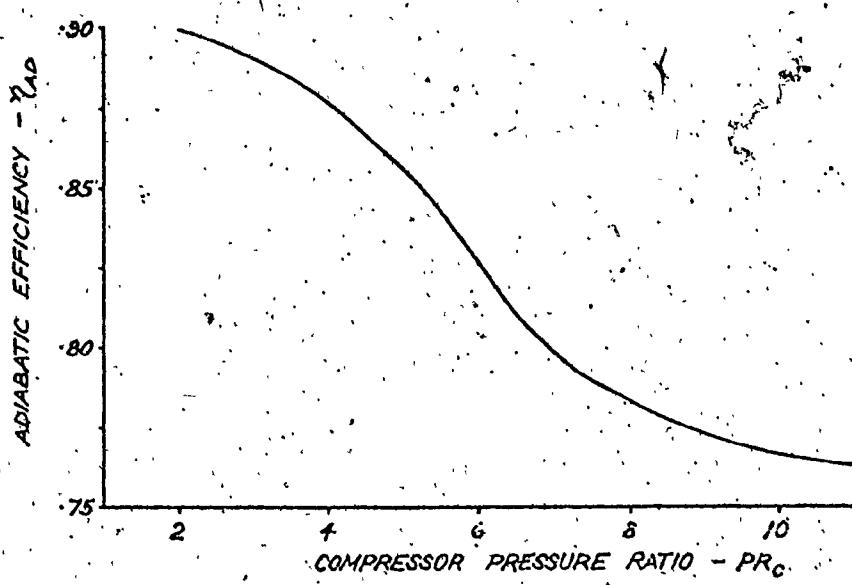


FIG. 5.1. (a)

From the values of NS_c , PR_c , P_1 and T_o defined and the use of the graphs of Fig. 5.1 a) and b) the compressor efficiency can be derived.

The mechanical speed and T_2 are calculated using the following equations.

The adiabatic work is given by:

$$h_{ad} = T_o (PR_c)^{(Y-1)/Y} - 1 \cdot \frac{R \cdot Y}{Y-1} \quad (5.1)$$

where

$$\gamma = \frac{C_p}{C_v} = f \left[1/2 (T_2 + T_o) \right]$$

and

$$R = 53.350 \text{ ft-lb/}^{\circ}\text{R/lb} = \text{Gas constant for air}$$

The temperature increase through the compressor is given by:

$$\Delta T = \frac{h_{ad}}{R \cdot \eta_{c,a}} \cdot \frac{\gamma - 1}{\gamma} \quad (5.2)$$

and then:

$$T_2 = T_o + \Delta T \quad (5.3)$$

The specific speed of the compressor is defined as follows:

$$NS_c = \frac{N_c \sqrt{Q_c}}{h_{ad}^{3/4}} \quad (5.4)$$

where:

$$Q_c = \frac{W_c}{g \cdot \rho_{\text{inlet, tot}}} \quad (5.5)$$

where:

$g = 32.174 \text{ ft/sec}^2$ = acceleration of gravity

$\rho_{\text{inlet, tot}}$ = total inlet density lb/ft^3

Equation (5.4) may be rearranged to give the compressor mechanical speed N_c :

$$N_c = \frac{NS_c \cdot h^{3/4}}{\sqrt{Q_c}} \quad (5.6)$$

CHAPTER 6

TURBINE AERODYNAMICS

6.1 Single Shaft Engine

In a single shaft engine design, there is one turbine section directly fixed to the compressor. This turbine drives the compressor and also provides output power for the engine.

In order to initiate the strip stress analysis of the radial turbine it is necessary to calculate first the tip radius and tip speed of the turbine. These two variables are a function of the turbine efficiency.

The efficiency of radial turbine can be represented by means of three curves as shown on Fig. 6.1 a), b) and c). Usually these curves are the results of experimental tests and provide a method of generalizing the turbine efficiency calculation.

Thus for a given number of blades NB, relative angle β_3 and turbine specific speed, the turbine total efficiency is established.

The turbine specific speed is given by:

$$NS_T = \frac{N_T \sqrt{\Omega_6}}{h_{ad}^{3/4}} \quad (6.1)$$

where:

N_T = turbine mechanical speed

Ω_6 = turbine exit flow parameter

h_{ad} = turbine adiabatic work

Since NS_T defines turbine efficiency η_T and h_{ad} is a function of efficiency an iterative process is required to define the correct values of NS_T and η_T for the given turbine power requirements.

TURBINE EFFICIENCY

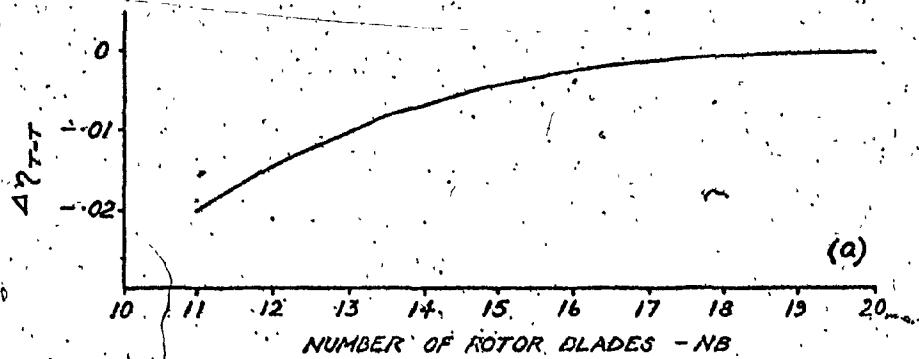
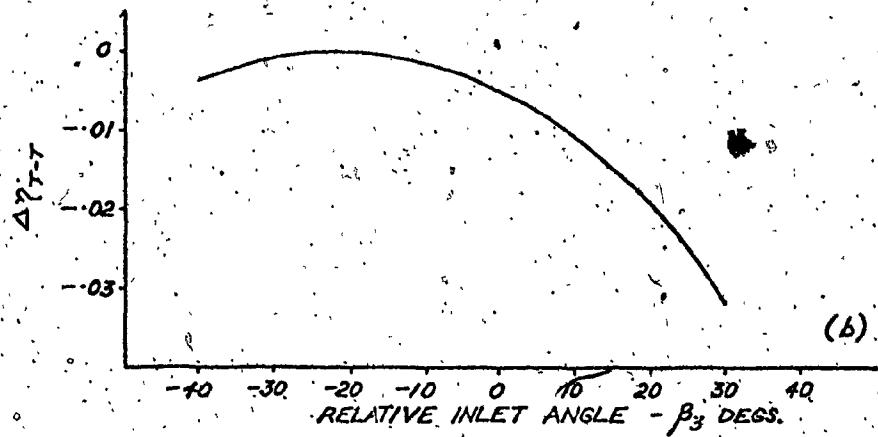
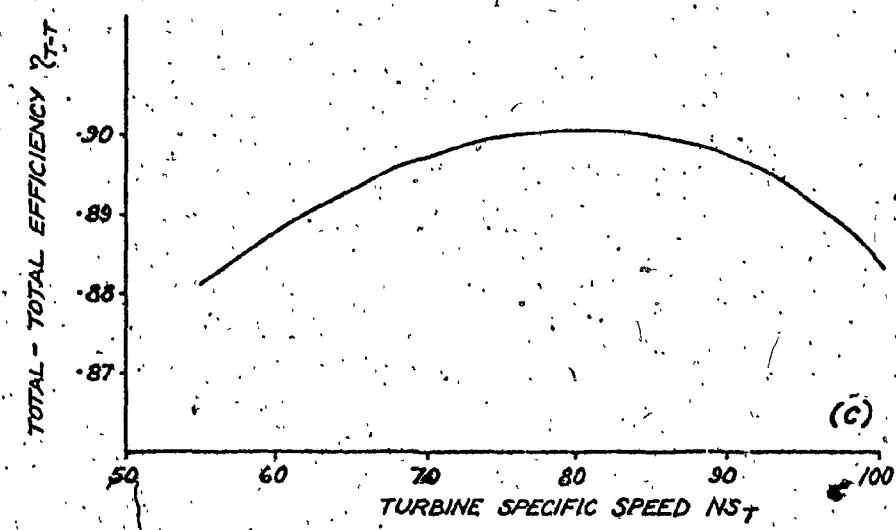


FIG. 6.1.

The turbine tip speed and tip radius are now established.

a) Wheel Tip Velocity Calculation: U_3

The tip velocity triangle

of Fig. 6.2 gives:

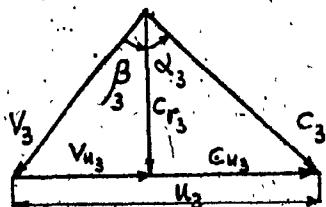


Figure 6.2

$$\operatorname{tg} \alpha_3 = Cu_3 / Cr_3$$

$$\operatorname{tg} \beta_3 = Vu_3 / Cr_3$$

or:

$$\frac{Cu_3}{\operatorname{tg} \alpha_3} = \frac{Vu_3}{\operatorname{tg} \beta_3} \quad (6.2)$$

but:

$$Vu_3 = Cu_3 - U_3 \quad (6.3)$$

Substituting equation 6.3 into 6.2 gives:

$$\frac{Cu_3}{\operatorname{tg} \alpha_3} = \frac{Cu_3 - U_3}{\operatorname{tg} \beta_3} \quad \text{from which}$$

$$Cu_3 = U_3 \cdot \frac{\operatorname{tg} \alpha_3}{\operatorname{tg} \alpha_3 - \operatorname{tg} \beta_3} \quad (6.4)$$

At the exducer exit the velocity triangle shown gives:

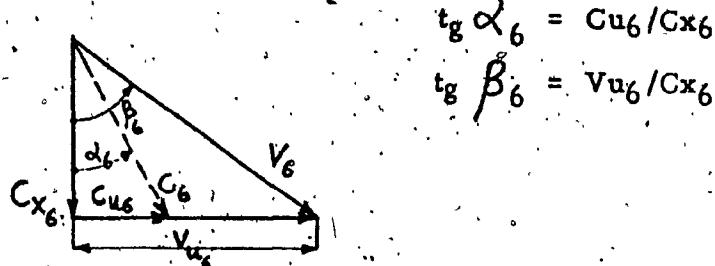


Figure 6.3

and combining:

$$\frac{Cu_6}{\operatorname{tg} \alpha_6} = \frac{Vu_6}{\operatorname{tg} \beta_6} \quad (6.5)$$

but:

$$Vu_6 = Cu_6 + U_6 \quad (6.6)$$

and substituting equation 6.6 into 6.5 gives:

$$\frac{Cu_6}{\operatorname{tg} \alpha_6} = \frac{Cu_6 + U_6}{\operatorname{tg} \beta_6} \quad \text{from which:}$$

$$Cu_6 = \frac{U_6 \operatorname{tg} \alpha_6}{\operatorname{tg} \beta_6 - \operatorname{tg} \alpha_6} \quad (6.7)$$

b) Euler's Equation

This states that:

$$g J \Delta H = U_3 \cdot Cu_3 + U_6 \cdot Cu_6 \quad (6.8)$$

and after substituting equations 6.4 and 6.7 into 6.8:

$$g J \Delta H = \frac{\frac{U_3^2}{2} t_g \alpha_3}{t_g \alpha_3 - t_g \beta_3} + \frac{\frac{U_6^2}{2} t_g \alpha_6}{t_g \beta_6 - t_g \alpha_6} \quad (6.9)$$

but $g J \Delta H$ is also given by:

$$g J \Delta H = g J C_p \Delta T = g J C_p (RIT - T_6) = \\ g J C_p \eta_{TT} RIT \left[1 - \left(\frac{1}{PRT} \right)^{\frac{x-1}{k}} \right] \quad (6.10)$$

It is now necessary to define the exit geometry. The following quantities are fixed at the beginning of the design:

M_6 = Exit Mach Number

α_6 = Exit Swirl Angle

R_{6H} = Exducer Hub Radius

From compressor analysis,

$$W_6 = (1 + F/A) W_c \quad (6.11)$$

$$P_3 = P_c (1 - K) \quad (6.12)$$

Where K is a constant defined at the beginning of the design due to the pressure loss in the combustion chamber. The exit pressure is then given by:

$$P_6 = P_3 / PR_T$$

(6.13)

Where:

$$PR_T = K_1 PR_C$$

(6.14)

K_1 is a constant defined in such a way that when multiplied by PR_C gives the PR_T . The turbine exit temperature can be derived from equation 6.10 since now η_T , RIT and PR_T are known.

In addition:

$$\frac{W_6 \sqrt{T_6}}{CD_6 A_6 P_6 \cos \alpha_6} = f(M_6)$$

(6.15)

$$A_6 = \pi (R_{6T}^2 - R_{6H}^2)$$

(6.16)

and:

$$C_6 = f(M_6) \cdot \sqrt{T_6}$$

(6.17)

$$C_{u6} = C_6 \sin \alpha_6$$

(6.18)

$$C_{x6} = C_6 \cos \alpha_6$$

(6.19)

$$\beta_6 = \operatorname{tg}^{-1} \left(\frac{V_{u6}}{C_{x6}} \right) = \operatorname{tg}^{-1} \left(\frac{C_{u6} + U_6}{C_{x6}} \right)$$

and using eq. 6.19:

$$\beta_6 = \operatorname{tg}^{-1} \left[\operatorname{tg} \alpha_6 + \frac{U_6}{C_6 \cos \alpha_6} \right]$$

(6.20)

Now from equation 6.9:

$$U_3^2 = \left[(g J \Delta H - \frac{U_6^2 \operatorname{tg} \alpha_6}{\operatorname{tg} \beta_6 - \operatorname{tg} \alpha_6}) \right]$$

$$\frac{\operatorname{tg} \alpha_3 - \operatorname{tg} \beta_3}{\operatorname{tg} \alpha_3}$$

and remembering that:

$$\frac{U_6^2 \operatorname{tg} \alpha_6}{\operatorname{tg} \beta_6 - \operatorname{tg} \alpha_6} = U_6 C_{u6} = U_6 C_6 \sin \alpha_6$$

and using eq. 6.10 gives:

$$U_3 = \left\{ \frac{\operatorname{tg} \alpha_3 - \operatorname{tg} \beta_3}{\operatorname{tg} \alpha_3} \left[g J C_p \eta_{TT} \cdot RIT \left[1 - \left(\frac{V_1}{PR_T} \right)^{5/4} \right] \right] - U_6 C_6 \sin \alpha_6 \right\}^{1/2} \quad (6.21)$$

that is the turbine tip speed.

The sequence of equations to be used to calculate the tip radius and the tip speed of the turbine is therefore as follows:

$$PR_T = K_1 \cdot PR_c$$

$$W_T = (1 + F/A) W_c$$

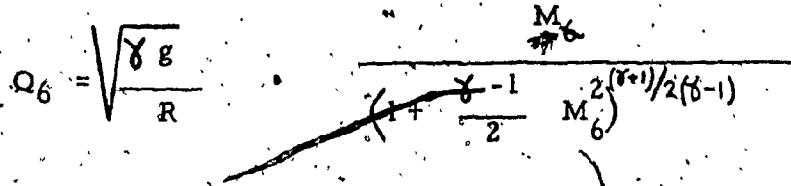
$$P_3 = P_c (1 - K)$$

$$P_{6T} = P_3 / PR_T$$

$$T_{6T} = RIT \cdot \eta_{TT} \cdot RIT \left[1 - \left(\frac{1}{PR_T} \right)^{\frac{\gamma-1}{\gamma}} \right]$$

where $\frac{\gamma-1}{\gamma}$ evaluated at the mean temperature

$$T_m = \frac{T_{6T} + RIT}{2}$$



where γ evaluated at temperature T_{6T} .

$$A_6 = \frac{W_T \cdot \sqrt{T_{6T}}}{CD_6 \cdot P_{6T} \cdot \cos \alpha_6 \cdot Q_6}$$

where:

CD_6 = .91 \pm constant = discharge coefficient

$$R_{6T} = \left(\frac{A_6}{\pi} + R_{6H}^2 \right)^{1/2}$$

$$R_6 = \left(\frac{R_{6H}^2 + R_{6T}^2}{2} \right)^{1/2}$$

$$U_6 = \pi R_6 \cdot N_c / 360$$

$$FM = \sqrt{\gamma g R} \cdot \frac{M_6}{\left(1 + \frac{\gamma-1}{2} M_6^2 \right)^{1/2}}$$

$$\gamma = f(T_{6T})$$

$$C_6 = FM \cdot \sqrt{T_{6T}}$$

$$ANG 6 = U_6 \cdot C_6 \cdot \sin \alpha_6$$

$$ANG 3 = T_g \alpha_3 / (t_g \alpha_{3e} - t_g \beta_3)$$

$$\Delta T = RIT - T_{6T}$$

$$DH = g J \cdot C_p \cdot \Delta T \quad C_p = f(T_m)$$

$$U_3 = [ANG 3 \cdot (DH - ANG 6)]^{1/2}$$

6.2 Free Shaft Engine

In a free shaft engine the turbine section is split into two sections. The gas generator turbine provides power only to meet the compressor and auxiliary requirements. The output power turbine is on a separate shaft and extracts the remaining usable power from the gas flow.

In this study we are considering a free shaft engine in which the gas generator turbine is radial while the power turbine can be either radial or axial, but it is not considered directly in the analysis.

In the free shaft design, the gas generator turbine must match the power requirements of the compressor and auxiliaries.

The compressor work is defined by the following equation:

$$H_c = \frac{W_c}{\eta_{c,a}} \left(PR_c^{\frac{r-1}{r}} - 1 \right) \cdot T_0 \cdot C_{p,2-3} \quad (6.22)$$

The turbine work is defined by:

$$H_T = W_c (1 - \Delta W) (1 + F/A) \eta_{TT} \left[1 - \left(\frac{1}{PR_T} \right)^{\frac{r-1}{r}} \right]$$

$$\therefore RIT \cdot C_{p,4-6} \quad (6.23)$$

In equations 6.22 and 6.23 both C_p and γ are functions of the mean temperature in the compressor and turbine respectively using air properties for the compressor and gas properties for the turbine.

The work balance equation becomes:

$$\frac{\left[\frac{\gamma - 1}{\gamma} \right] \cdot T_o \cdot C_{p2-3}}{P R_c - 1} = (1 - \Delta W) (1 + F/A) \eta_{TT} \cdot \eta_{c,a} \eta_{mech}$$

$$\left[1 - \left(\frac{r}{P R_T} \right)^{\frac{\gamma - 1}{\gamma}} \right] \cdot R I T \cdot C_{p4-6} \quad (6.24)$$

In equation 6.24 a mechanical efficiency η_{mech} is included to represent auxiliary drive requirements and internal gas generator power losses between turbine and compressor. From equation 6.24 the turbine pressure ratio $P R_T$ can be derived as:

$$P R_T = \left[1 - \frac{\left(\frac{\gamma - 1}{\gamma} \right) \cdot P R_c - 1}{(1 - \Delta W) (1 + F/A) \eta_{c,a} \eta_{TT} \eta_{mech}} \right]^{\frac{1}{\gamma - 1}} \cdot \frac{C_{p2-3}}{R I T}^{\frac{1}{\gamma / 1 - \gamma}} \quad (6.25)$$

The above equation, defining the turbine pressure ratio, is the only additional requirement needed to allow the analysis of a free shaft engine. In the above equation the values of $P R_c$, ΔW , η_{mech} , $R I T$, T_o are specified by engine requirements.

The turbine efficiency η_{TT} depends on turbine specific speed which is initially unknown. Hence an iterative procedure is required to define the correct value of η_{TT} .

Gas properties C_p^* and γ^* for the turbine, are defined by the mean turbine temperature. This is a function of the turbine pressure ratio, which is initially unknown, so another iterative procedure is required to define these values. Only when these two

iterative procedures employed does equation 6.25 converge to
the correct value of turbine pressure ratio.

CHAPTER 7

RESULTS AND CONCLUSIONS.

To illustrate the general method a potential study has been carried out on a possible engine configuration. The compressor mass flow is 4 lb/sec and all the other input parameters are reported in Appendix 2.

The aim of the sample calculation is to show the effect on rotor inlet temperature of the following variables:

- Compressor pressure ratio (PR_c)
- Compressor specific speed (NS_c)
- Low cycle fatigue life (LCF)
- R_H/R_T ratio

Figure 7.1 through 7.6 show the computer automatic plottings. From these graphs it is evident how the R_H/R_T ratios play an important role in the definition of the maximum rotor inlet temperature. The R_H/R_T ratio is defined by aerodynamic requirements and a value of 0.38 is usually an acceptable figure. Figure 7.1 shows the maximum RIT occurs at a value of $R_H/R_T = .5$. However, it also shows that only 10°F penalty has to be paid in order to satisfy aerodynamic requirement for an optimum efficiency level. This is a very valuable information for the rotor designer because it indicates clearly the trade-off between RIT and hub radius R_H . The behavior of the curves of Fig. 7.1 is not general but it depends on such parameter as material creep strength and LCF life required.

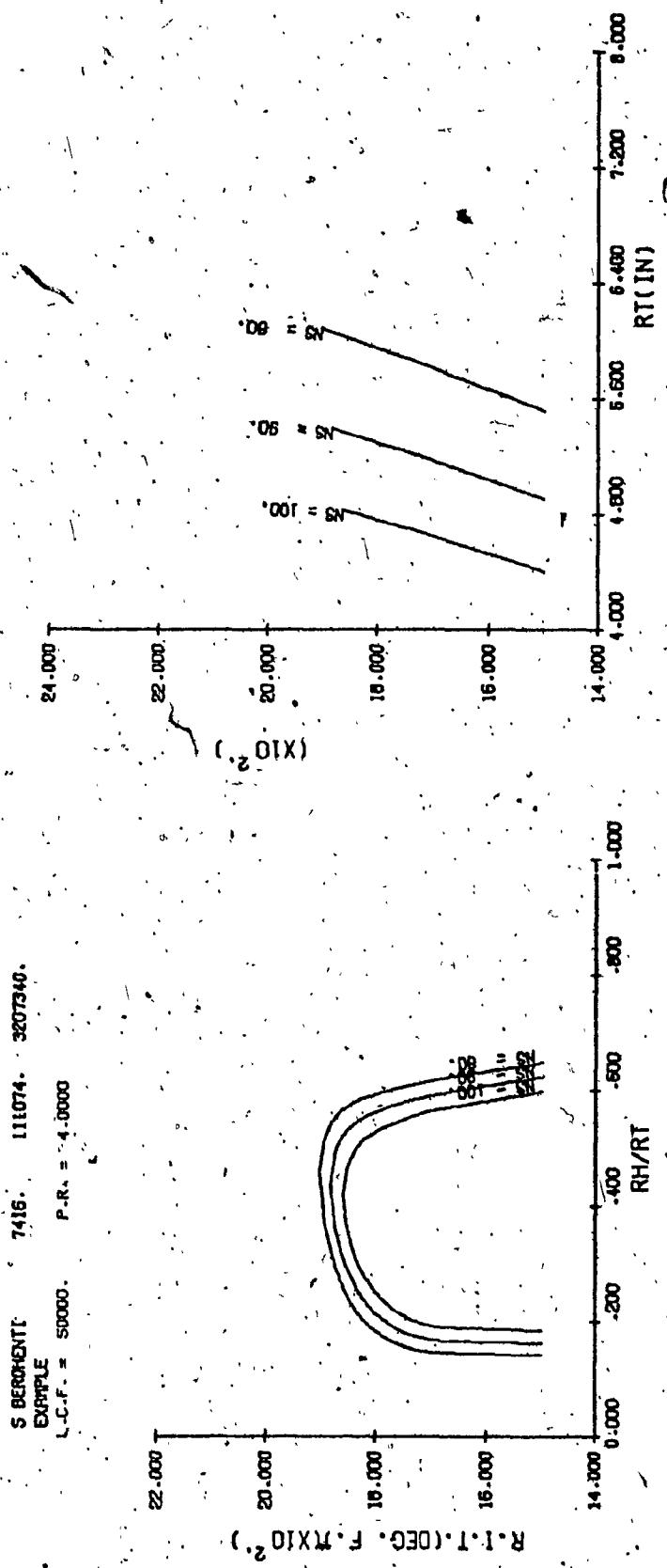


Fig. 7.1

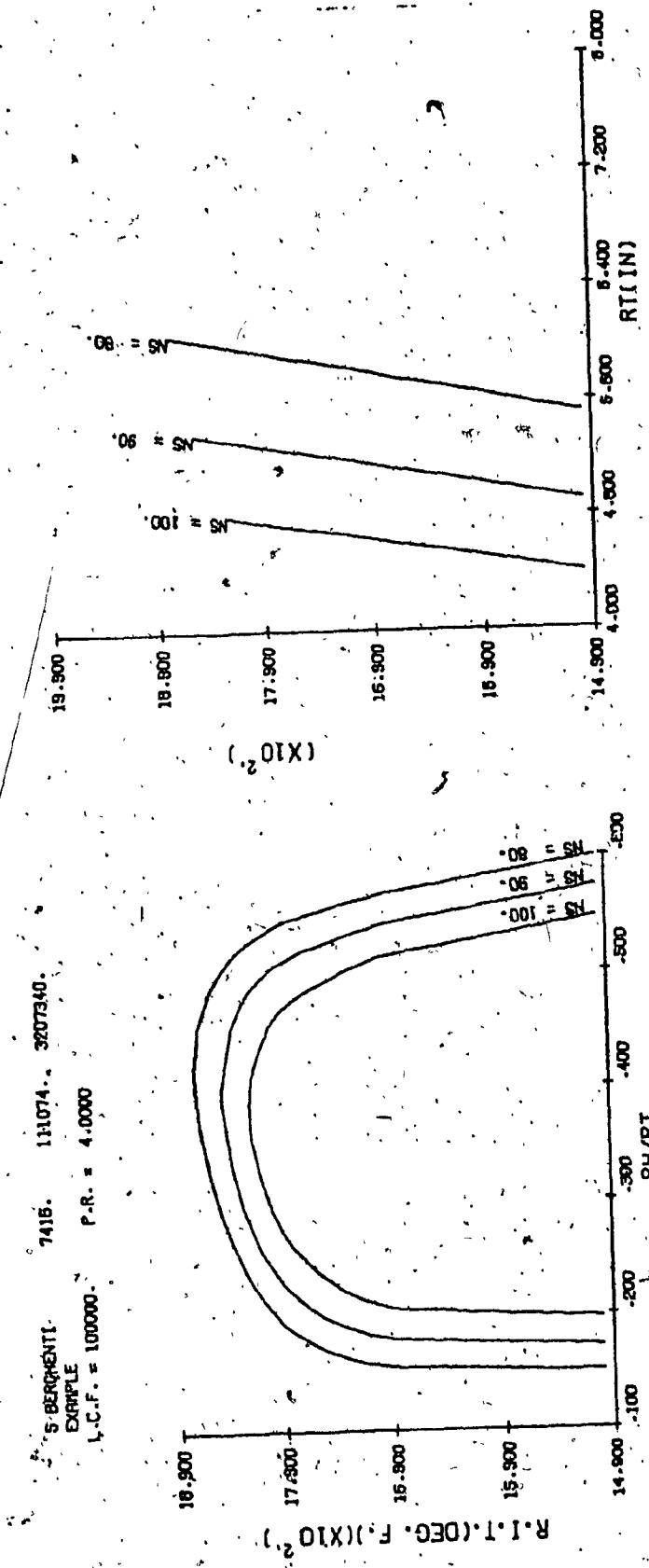


Fig. 7.2

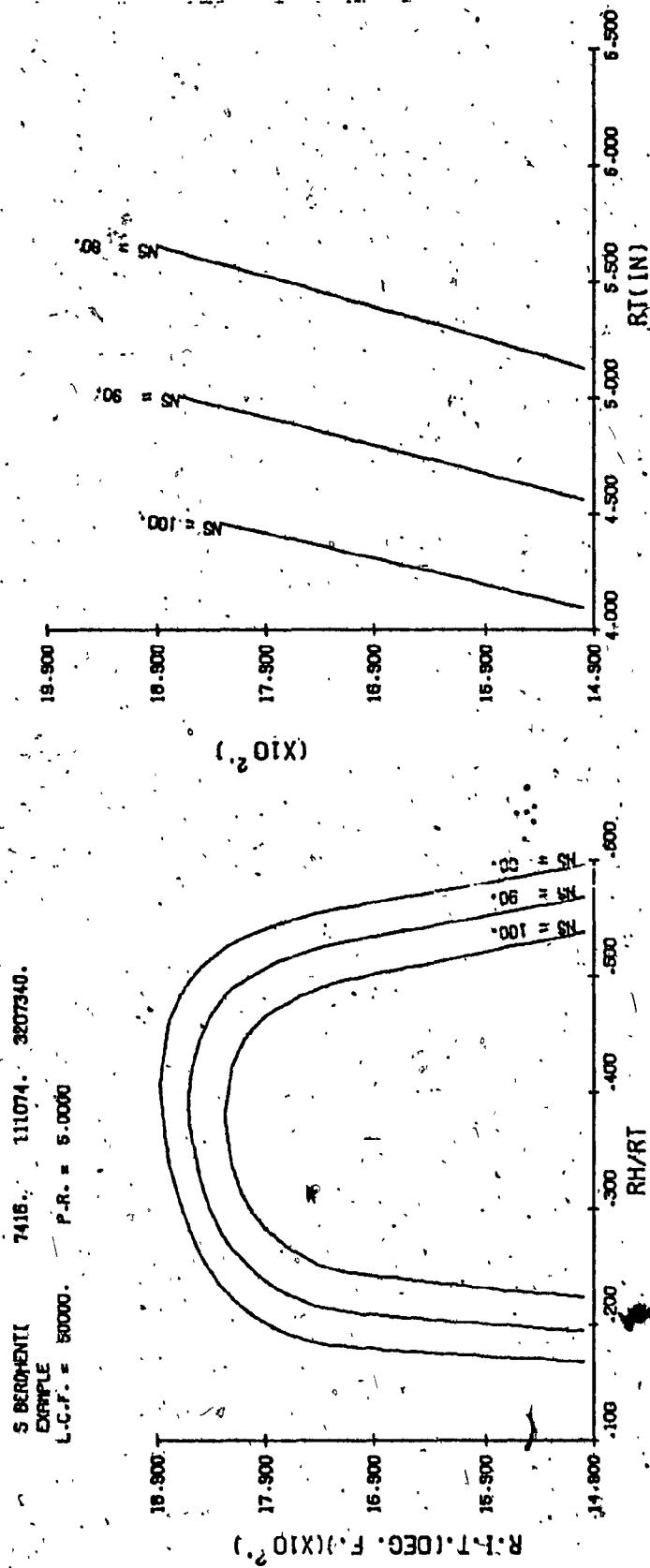


Fig. 7.3

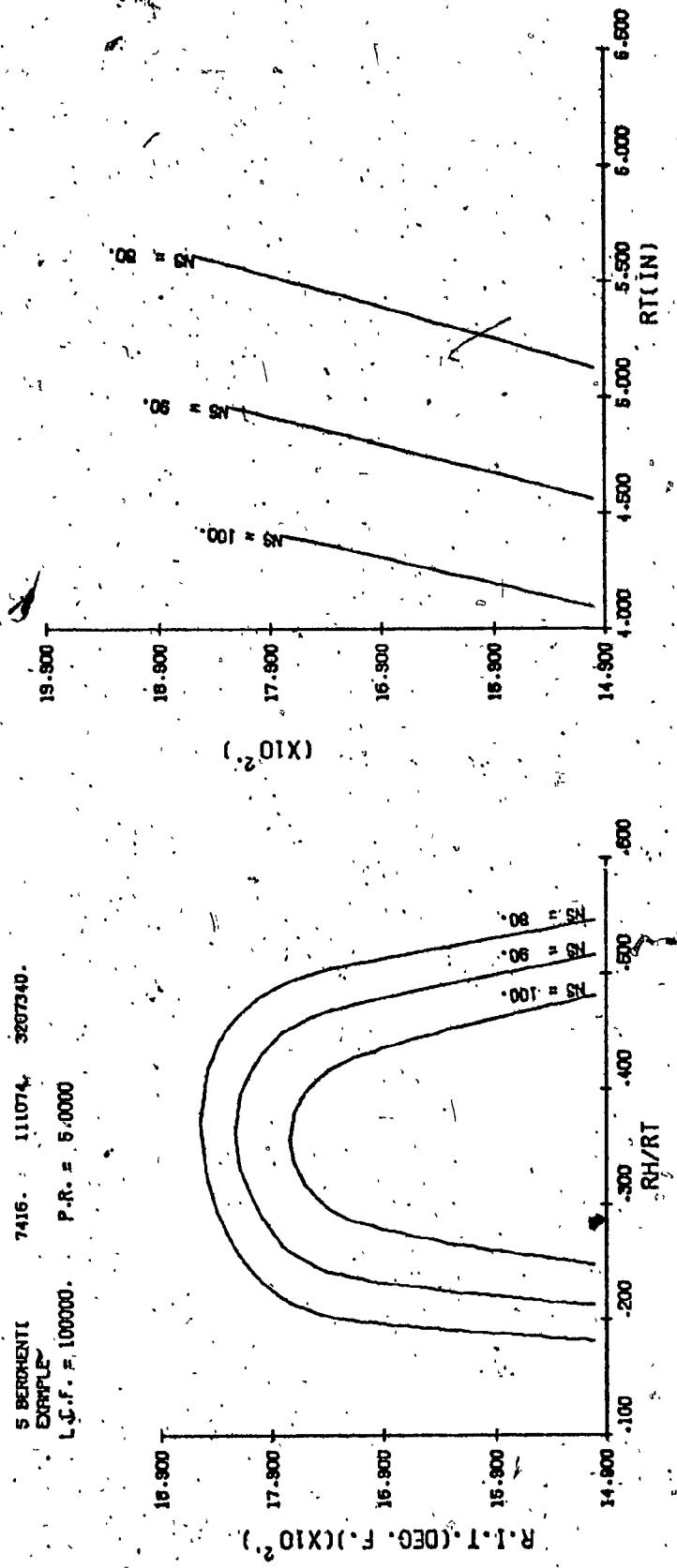


Fig. 7.4

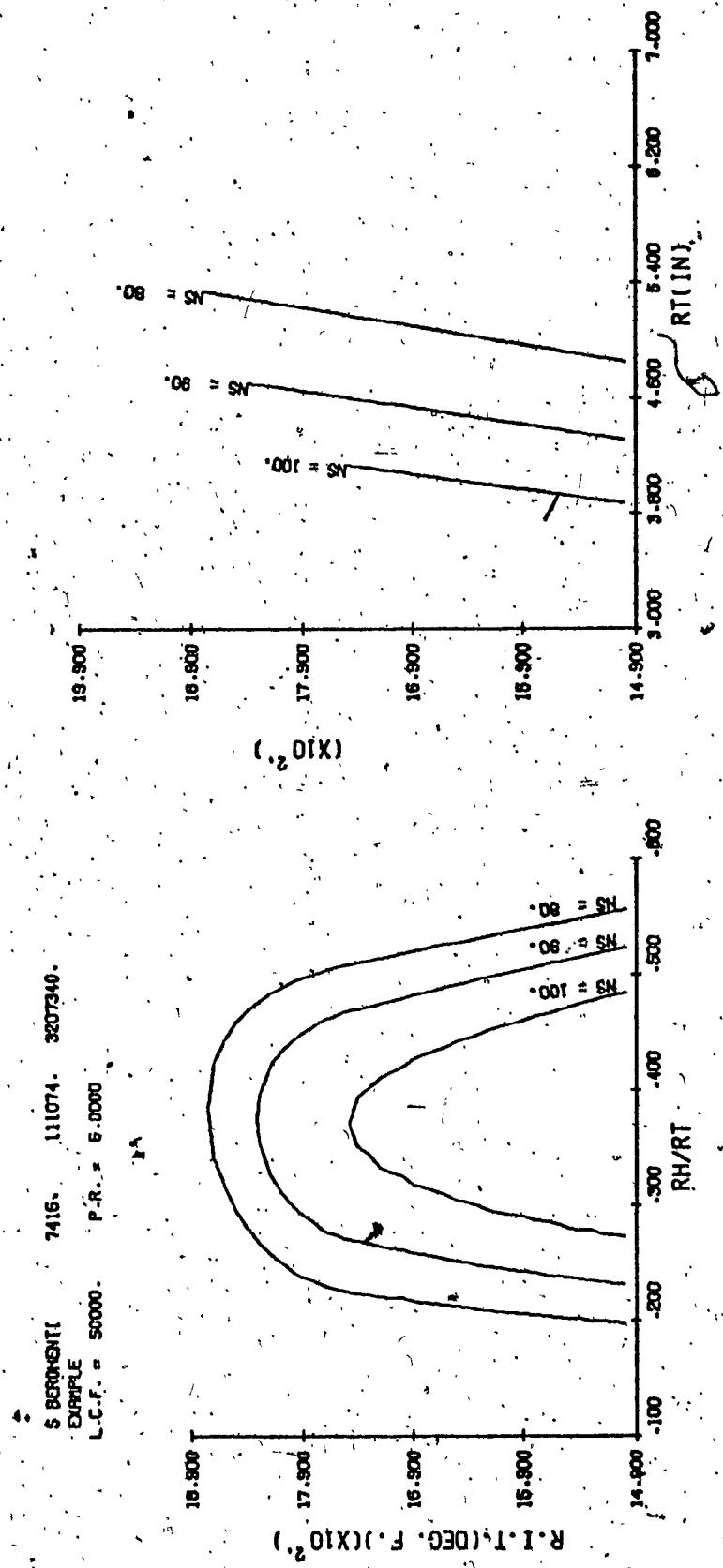


Fig. 7, 5.

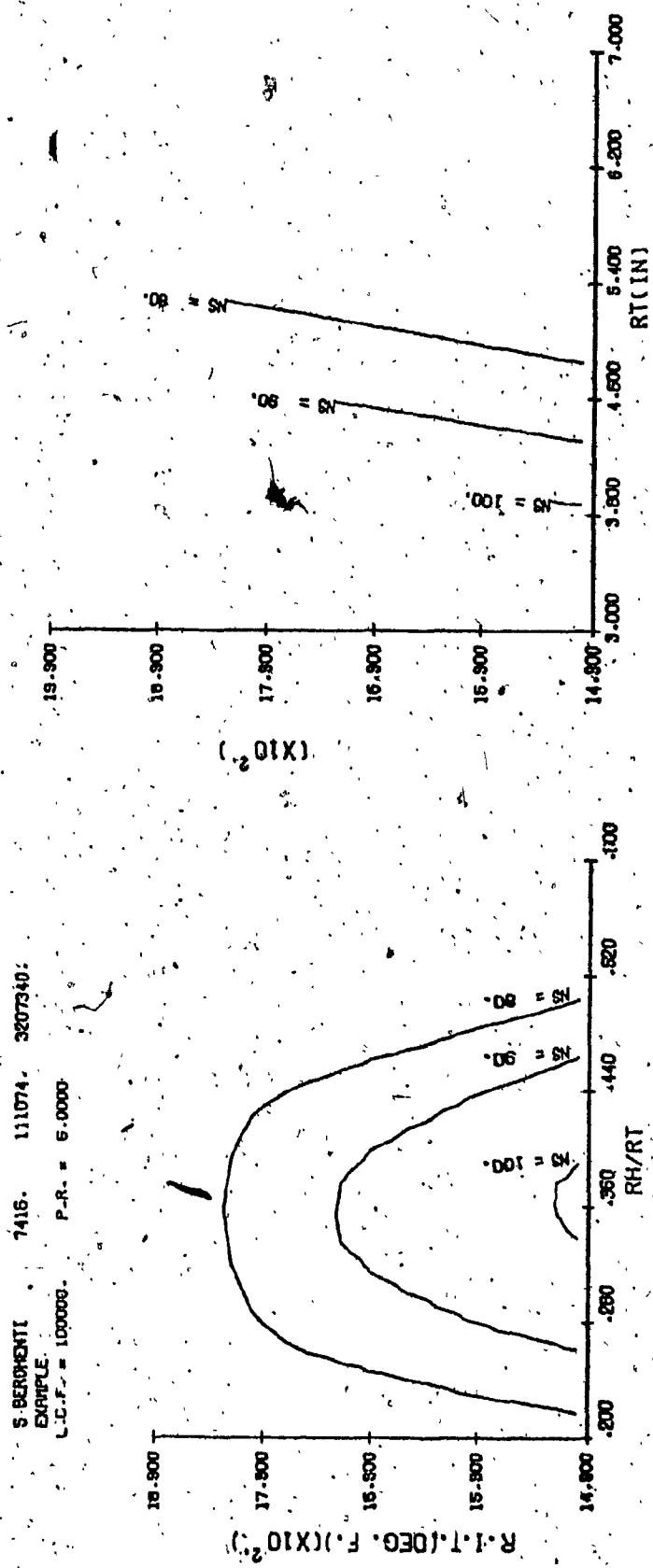


Fig. 7.6

Figures 7.7 a) and b) show the effect of compressor pressure ratio and LCF life on maximum RIT for various compressor specific speed, and a constant value of $R_H/R_T = .4$.

With the trend to higher compressor pressure ratios, a significant reduction in maximum RIT is noted.

Compressor specific speed and LCF life when increasing also have a strong adverse effect on maximum RIT.

This is only one of the many cases which can be analyzed. In particular the effect of other variables like, β_3 , α_3 , NB, TT, TOL, material, on maximum RIT can be analyzed and once performance calculations are completed the best combination of variables can be selected.

Figure 7.1 shows that the turbine tip radius increases with RIT and this is consistent for an engine in which the mass flow is kept constant but the output power increases as RIT increases.

However, a new engine design usually revolves around constant power or thrust and therefore a scale factor has to be introduced by the performance engineer. This scale factor could slightly effect the maximum RIT.

A continuation of the work described in this thesis is recommended and, in particular, the inclusion of performance calculations in the computer program will make this method capable of handling constant power engines and this will also permit engine size optimization.

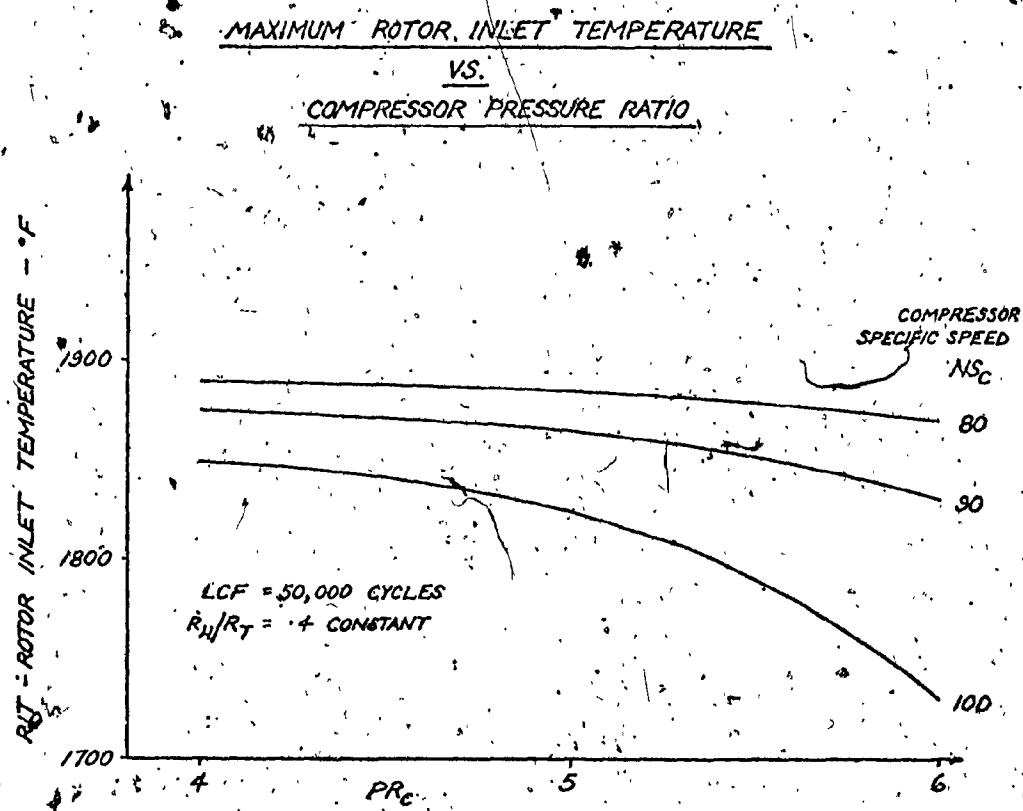


FIG. 7.7.(b)

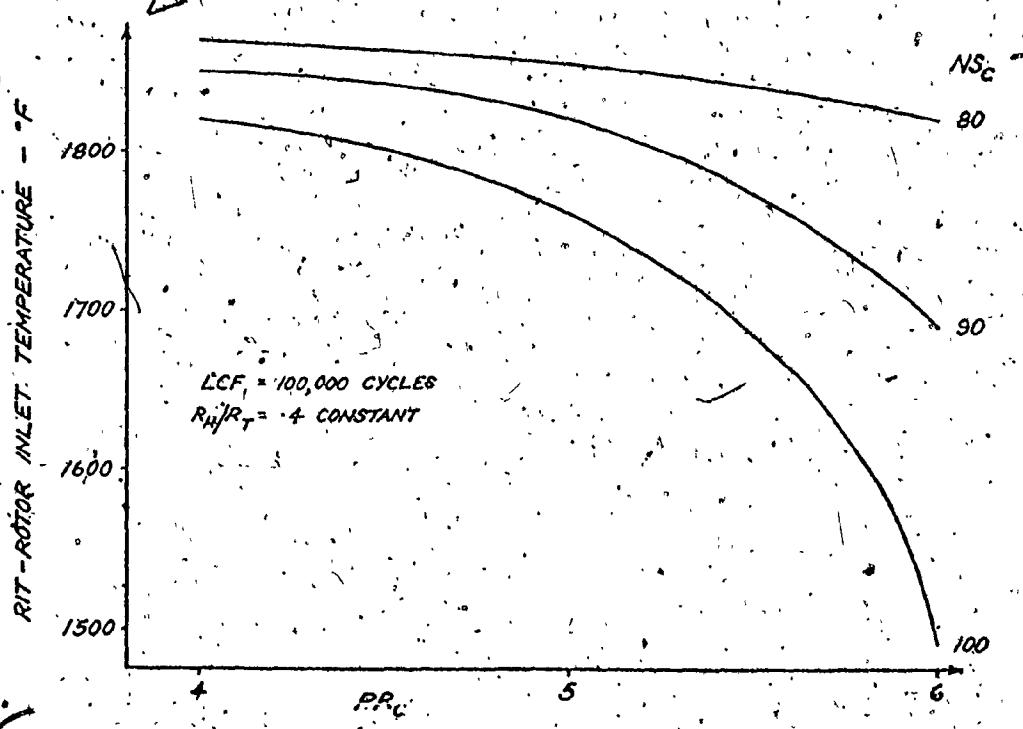


FIG. 7.7.(a)

LIST OF REFERENCES

- (1) "Formulas for Stress and Strain" by Raymond J. Roark, Fourth Edition, McGraw-Hill.
- (2) "Thermal Stress and Low-Cycle Fatigue" by S. S. Manson McGraw-Hill.
- (3) UACL T.N. #248 "The Potential of a Forged Radial Turbine", by M. VanDuyn, 1 June 1965.
- (4) "The Dynamics and Thermodynamics of Compressible Fluid Flow", Vol. I and II, A. H. Shapiro, Ronald Press Co., N. Y., 1954.
- (5) "Sawyer's Gas Turbine Engineering Handbook", Volume 1, Theory & Design, Second Edition, Gas Turbine Publications, Inc., Stamford, Connecticut 06904, 1972.
- (6) UACL T.N. #300 "AVLABS Cooled Radial Turbine Double Pass Cooling Scheme - Design and Stress Analysis" by S. Berghenti, March 1969.
- (7) UACL C. F. #3815 "Effect of Turbine Specific Speed on Radial Turbine Bore Stress Factor" by S. Berghenti, April 1973.
- (8) UACL C. F. #2825 "Calibration of Program 349 for Analysing Uncooled Radial Turbines" by E. W. Herdman, October 1970.

APPENDIX I

BLOCK DIAGRAM AND LISTING
OF COMPUTER PROGRAM

APPENDIX I.

The computer program "MARY" handles the optimization process theory described in the previous paragraphs.

The program has been written using FORTRAN IV language for a CDC 6600 computer available at United Aircraft of Canada Limited.

The final results are automatically plotted making their interpretation straight forward.

Appendix II shows a sample calculation and the relative graphs are shown and described in Section 7. From these graphs, other representations are possible to show more clearly the effect of variables on the maximum allowable Rotor Inlet Temperature.

Figure 1 shows the "program flow chart" in blocks and how they are connected by the various loops sequences.

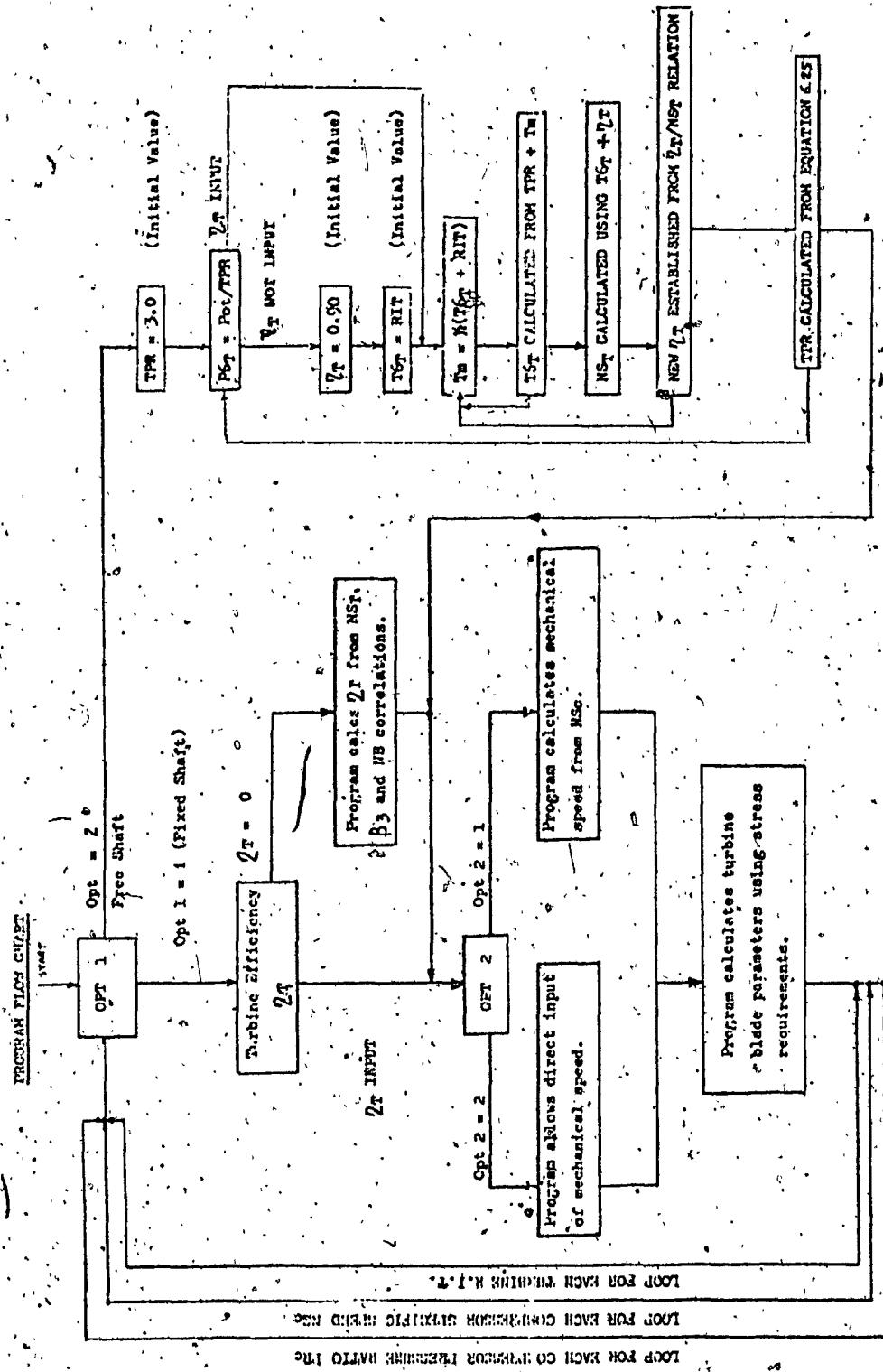


Fig. 1 App. 1

PROGRAM MARY INPUT, OUTPUT, TAPE6=INP(1), TAPE6=OUTPUT, PLOT,
1APE99=PLOT, TAPE20, TAPE30)

		OPTION	RESULT	TABLES REQUIRED
5.	C	OPT =1 =2	SOLID HUB BORED HUB	
10	C	OPT1 =1 =2	SINGLE SHAFT FREE SHAFT	
20	C	OPT2 =1 =2	MECH SPEED CALC MECH SPEED INPUT	XNS VS DETA, ETA VS PRI ETA VS PRI
25	C	ETATT=0 #0	TURB EFF. CALC TURB EFF INPUT	DETA VS BETAS, DETA VS NB, ETA VS NST
	C	COMMON/11/TIN		
	COMMON/1139/P1,T0,Y,ETAI(20),PRI(20),XNSI(20),PRC(5),DETA(20,6), IPR(5),XN,DH,CTR,ASF,XJ,G,RGAS,O,OPTZ, COMMON/34/TT,NB,TL,HL,ARCBB,MU,ROU,RB,T1,J2,T3,T4,OPT,A,B,C,D,E, 1,B1,C1,D1,A2,B2,C2,D2,AL1,AL2,AL3,AL4,AL5,AL6,E1,E2,E3,E4,E5,E6, 2,BET5,STGRANT5,OTHERM,FUDGE,ETABUT,XJ1,G1 3,TT(20),TS(20),OPT3,MATC,TMAX,TABLE(150), COMMON/34/SIGNIN(81),RHMIN(81),RHUS(81),RHL(5,81),INDEX DIMENSION RITZ(5),RH2T5),TRI3(T81)70,T81,ETATT(20),DET1 3,5,BETA3I(5),XNST3(81),ETATT3(81),TPR3(81),NBI(10),DET(10),RITZ 2,RT(81),LCF(5),R6TAR(81),TM3AR(81) COMMON/HEDR3TITLE(16),PRM DIMENSION ITEX(5) DIMENSION TABT(10),TABA(10),TATE(10) 4,0,INTEGER,OPT,OPT1,OPT2,OPT3 REAL NB, MU, LCF, NBI			

PROGRAM MARY 74174 OPT=1

C READ AND WRITE INPUT DATA

45 IFLAG=0

3 READ(60,900) TITLE(I),I=1,8),JACKASS

REWIND 30

IF (JACKASS.EQ.0.1RS) 1,2

1 IF(IPLCP.EQ.1) CALL ENDPLT

50 CALL EXIT

2 WRITE (61,901)

READ(60,999),TITLE(I),I=9,16)

WRITE(61,999) TITLE

999 FORMAT(8A10)

READ(60,904) NRIT,NSGRAN,OPT,OPT1,OPT2,OPT3,MANC,IPLCP,IGRID

904 FORMAT(9I4)

IF(IFLAG.NE.0) GO TO 16

IPLCP.EQ.0) GO TO 16

1TEX(1)=10RTHE-FOLLOW

1TEX(2)=10RING PLOT R

1TEX(3)=10REQUIRES GR

1TEX(4)=10RD PAPER

1TEX(5)=10R

CALL REGRESST(99)

CALL MESS(1TEX)

CALL NAMPLT

16 CONTINUE

READ(60,902) IT,NB,BSF1,XNST1,BSF2,XNST2

READ(60,902) TOL,WL,ARC3B,MU,RHO,RB

READ(60,902) T1,T2,T3,T4,DTHERR

READ(60,903) A,B,C,O

READ(60,903) A1,B1,C1,O1

READ(60,903) A2,B2,C2,O2

IF(OPT3.EQ.1) GO TO 300

N20=0

DC 350 I=1,20

READ(60,301) TTT(I),IS(I)

IF(TTT(I).EQ.0) GO TO 300

N20=N20+1

80 N20=CONTINUE

301 FORMAT(2E10.4)

300 IF(OPT3.EQ.2) THAX=TTT(N20)

IF(MATC.EQ.0) READ(60,909) E1,E2,E3,E4,E5,E6

IF(MATC.EQ.0) READ(60,909) AL1,AL2,AL3,AL4,AL5,AL6

FTN 4.1+PSR367 29/10/74 13.34.48.

```

PROGRAM HARRY    75774  OPT=1
FTN 4.1+PSR367° 29710774 23.34.48.

85      WRITE(61,906)
        IF (OPT.EQ.2) GO TO 30
        WRITE(61,907) OPT
        GO TO 20
90      30  WRITE(61,908) OPT
        20  IF (OPT1.EQ.1) GO TO 25
        WRITE(61,924) OPT1
        GO TO 49
25  WRITE(61,925) OPT1
        49  WRITE(61,911) NRIT,NSIGRAN,TT,NB,BSF1,XNST1,BSF2,XNST2,TOL,TL,TARCB
18,MU,RHO,PB,DTERM
95      IF (OPT3.EQ.2) GO TO 303
        WRITE(61,912)
        WRITE(61,913) A,B,C,D,T1,T2,A1,B1,C1,D1,T2,T3,A2,B2,C2,D2,T3,T4
        GO TO 304
100     303  WRITE(61,302)
        302  FORMAT(* INPUT TABLE OF STRESS V.S. TEMPERATURE*/* TEMPERATURE*6X*S
              1TRESS*/*)
        WRITE(61,305) THTH,ISET,ISET,I=1,N20)
        305  FORMAT(2F10.2)
105     304  CONTINUE
        IF (MATHC.NE.0) GO TO 2000
        WRITE(61,914)
        WRITE(61,915) E1,E2,E3,E4,E5,E6
        WRITE(61,916)
        WRITE(61,915) AL1,AL2,AL3,AL4,AL5,AL6
        2000  IF (MATHC.NE.0) CALL FILE(NAME,MATHC,TABLE)
        IF (MATHC.NE.0) WRITE(61,6666) MATHC,(TABLE(I+9),TABLE(I+29))TABULEIT+
              169),I=1,10)
        6666  FORMAT(* MATERIAL CODE*114/* TEMPERATURE
              EXPANSION*7771073E15*877)
115     1THERMAL*723X*RODULUS   YOUNGS
        WRITE(61,917)
        DO 4 I=1,NSIGRAN
        READ(60,902) NSIGRAN,ISET,ISET
        4  WRITE(61,918) SIGRAN(I),LCF(I)
        IETA=0
        WRITE(61,920)
        READ(60,902) ALPHA3,BETA3,ALPHA6,XI16,R6H
        READ(60,902) ETATT,ETABUR,RATIO,UMTRIM,UCORR,OW
        WRITE(61,921) ALPHA3,BETAS,UMPTAS,ETATT,ETABUR,RATIO,UMTRIM,R6H,
        1UPTRIM,UCORR,OW
        WRITE(61,922)

```

PROGRAM MARY 74774 OPT=1

FTN 4.1+PSR367 29710774 13.34.48.

```

      READ (60,902) (RIT(I),I=1,NRIT)
      WPITE (61,923) (RIT(I),I=1,NRIT)
      IF (ETATT.NE.0.) ETETA=1.
      IF (ETETA.EQ.0.) GO TO 8
      C   PROGRAM CALCULATES EFFICIENCY. IF ETATT INPUT AS ZERO
      C
      C   WRITE (61,937)
      DO 135 I=1,5
      READ (60,902) DETAIR(I)*BETA3(I)
      12  WRITE (61,902) DETAIR(1),BETA3(1)
      DO 111 L=2,5
      IF (BETA3-BETA3(L)) 121,122,111
      140  111 CONTINUE
      WRITE (61,905)
      STOP
      121 DETAR1=(BETA3-BETA3(I(L)))/(BETA3(I(L))-DETAIR(I(L))-DETAIR
      1(L-1))*DETAIR(I(L))
      145  WRITE (61,941)
      DO 144 I=1,10
      READ (60,902) DET(I),NBII(I)
      14  WRITE (61,902) DET(I),NBII(I)
      DO 112 L=2,10
      IF (NB-NBI(L)) 122,122,112
      112 CONTINUE
      WPITE (61,944)
      STOP
      122 DETAR2=(NB-NDI(I))/((NB-NBI(I))-NBII(I)-DETR(I)-DETR(I-1))
      DETAR = DETAR1 + DETAR2
      WRITE (61,928)
      DO 13 I=1,20
      READ (60,902) ETATT(I),XNTII(I)
      13 WRITE (61,902) ETATT(I),XNTII(I)
      160  2 READ (60,902) P1,T0,Y,W,PRLOSS,XN
      READ (60,902) XNSMAX,XNSMIN,DELXNS,ASF,ETAHECH,ETA
      READ (60,902) PR
      WRITE (61,926)
      WRITE (61,927) P1,T0,Y,W,XNSMAX,XNSMIN,DELXNS,ASF,PRLOSS,ETAHECH
      165  WRITE (61,940) PR
      T0=70*460.0
      IF (OPT2.EQ.2) GO TO 11
      READ (60,902) PRC

```

PROGRAM MARRY 74774 OPT=1

FIN 4.1+PSR367 29710774 13.34.48.

```

170      WRITE (61,938) PRC
        WRITE (61,939)
        00,6 I=1,20
        READ (760,902) XNS1(I),DETA1(I,K),R=1.5
        6* WRITE (61,902) XNS1(I),DETA1(I,K),K=1.5
        WRITE (61,929)
        007 1=E1,20
        READ (60,902) ETA1(I),PRE(I)
        7 WRITE (61,902) ETA1(I),PRE(I)
        11 WRITE (61,930)

180      C   CONSTANTS.
185      PI=3.14159265
        RAD=PI/180.
        G=32.174
        XJ=778.26
        RGAS = 53.35
        RHO1=0.076474
        Z=P1*519.0/(T0*44.7) 1
        0=W/(RHO1*Z)
        G=G*12.0
        XJ=XJ*12.0
        ALP3=ALPHA3*RAD
        BET3=BETA3*RAD
        BET4=ECOS(BET3)**2-0.05*COS(ALP3)**2
        BET5=SIN(ALP3-BET3)**2
195      NRIT1=(NRIT-I)*10+I
        NPR=0
        DO 31 I=1,5
        IF (FRIT1-NRIT=0.0) 21,31
        21  NPR=NPR+1
        31  CONTINUE
        IF (OPT2.EQ.2) GO TO 9
        IF (DELXNS.EQ.0.0) GO TO 9
        NS=(XNSMAX-XNSMIN)/DELXNS+1.01
        GO TO 15
        9  NS=1

200      C   LOOP FOR EACH COMPRESSOR PRESSURE RATIO SELECTED
        C   15 0 100 M=1,NPR
210

```

PROGRAM MARY 74/74 OPT=1

FTN 4.1+PSR367 29/10/74 13.34.48.

```

PRM=PR(M)
RTMIN=100.
BIGBIT=0.

215      C   LOOP FOR EACH SPECIFIC SPEED SELECTED UNLESS HECH SPEED INPUT
          C
          DO 110 J=1,NS
          XNS=XNSMIN+(J-1)*DELXNS
          INDEX=0
          NLAST=NRIT1
          DO 110 MM=1,NRIT1
          SIGMIN(MM)=0.0
          RHMIN(MM)=0.0
          DO 110 JJ=1,NSIGRA
          RHU(JJ,MM)=0.0
          RHC(JJ,MM)=0.0
          110 CONTINUE

220      C   CALCULATE ETA,XN FROM SUB1139
          C
          CALL SUB1139(XNS,ETA,W,PRM)
          TPRI = 3.0
          IF(OPT1.EQ.1) TPRI=PRM*RATIO
          WRITE(61,931) ETA,XNS
          931  WRITE(61,932) PRM,TIN,XN,CTR,OH

235      C   LOOP FOR EACH ROTOR INLET TEMPERATURE SELECTED
          C
          DO 1120 N=1,NRIT1
          RITO=BIT(1)+(N-1)*10
          RIT1=rito+UPTRIM
          RIT3(N)=RITO

245      C   CALCULATION OF TURBINE TIP SPEED
          C
          175  PC=P1*PRM
          PTT = PC*T1.0 - PR(CSST)
          CALL FAR(RIT1,ETABUR,FAGAS)
          HTW*(1.0+FAGAS)*(1.0-DW)
          135  P61 PUT7TPR1
          T61 = RIT1
          IF(ETA.EQ.0.1) GO TO 155

```

```

PROGRAM HARRY      74/74   OPT=1          PTN 4.1+PSR367    29/10/74  13.34.48.

      ETATT=0.90
      155 TM=(T6TT+RT1)*0.5+460.0
      CALL CGAS(ICP,GAM,AGAS,TM)
      GAM=(1.0-GAM)*GAM
      T6T=(RT1+460.0)*(1.0-ETATT*(1.0-TPR1*GAM))-460.0
      IF (ABS(T6T-T6TT)-1.0) 195,195,165
      165 T6TT=T6T
      GO TO 155
      195 CP1=CP
      196 T6T=460.0
      CALL CGAS(ICP,GAM,FAGAS,T)
      DELT=RT1-T6T
      IF (DELT.EQ.0.) GO TO 151
      GAM=2*GAM/GAM-1.0
      TOTSTAT=1.0*GAM-5*(GAM-1.0)*XM6*XM6
      T6S=(T6T+460.0)/TOTSTAT
      P6S=P6T/TOTSTAT**GAM2
      RH05=144.0+P6S/(RGAS*T6S)
      HEAD=EXJ*CP1*DELT/ETATT
      XNST=XN*SORT(HT/RH061)/HEAD**0.75
      DO 1-31 L=2,20
      1 IF (XNST-XNTI(L)) 141,141,131
      131 CONTINUE
      WRITE (61,9431)
      STOP
      141 EТАT=(XNST-XNTI(L))/((XNTI(L)-XNTI(L-1))*ETATT(L-1)+ETATT(L))
      151 EТАT=ETAT*DETA
      IF (ABS(ETAT-ETATT)-.001) 151,151,161
      161 ETATT=ETAT
      GO TO 155
      152 IF (OPT1.EQ.1) GO TO 152
      TPR2=(1.0-W*DH/(ETAMECH*WT*ETATT*CP1*(RT1+460.0)))**(1.0/GAM1)
      285 IF (ABS(TPR1-TPR2).GT.0.001) 152,152,153
      153 TPR1=TPR2
      GO TO 135
      152 SA=0.5*(GAM-1.0)
      Q6=SCRT(GAM*G/RGAS*XM6/(1.0+0.5*(GAM-1.0)*XM6)*GA
      A6=W*T*SORT(1.0/(0.91*P6T*Q6*COS(ALPH*46*RAD1)))
      R6T=SORT(R6H*R6H+R6TP1)
      R6TAR(N)=RET
      R6=SORT((R6H*R6H+R6T*R6T)/2.0)

```

PROGRAM NARY 74774 OPT=1 FTN 4.1+PSR367 29/10/74 13:34:48.

```

295      U6=PI*R6*XN/360.0
          FM=SCR1*(GAM*G+RGAS)/(1.0+0.5*(GAM-1.0)*XM6*XH6)
          C6=F'M*SQRT(1)
          ANG6=U6+C6*SIN(ALPHA6*RAD)
          ANG3=(1.-SIN(BETA3*RAD)*COS(ALPHA3*RAD))/(SIN(ALPHA3*RAD)*COS(BE
          T1A3*RAD))
          DELH=G*XJ*CP1*DELT
          U=SQRT((ANG3*(UDELH-ANG6)))
100      185 RT(N)=U*2.0/4*RAD*XN
          U=U*(1.0+UCORR)
          U3(N)=U
          RTOERT(N)
          XNST3(N)=XNST
          ETATT3(N)=ETATT
          TPR3(N)=TPRI
          NN=N
310      C   SIGMA - RH CURVE CALCULATED FROM SUB349
          C
          C   FNGCE=(BSF1-BSF2)*(XNST-XNST2)/(XNST1-XNST2)+BSF2
          C   CALL SUB349(RITO,RTO,U,N,NSIGRAN,TM3)
          C   TM3AR(N)=TM3
315      IF (INDEX-1) 120,115,120
          115 NLAST=N-2
          GO TO 125
          120 CONTINUE
          125 IF (NLAST.EQ.1) GO TO 110
          C   CALCULATION OF MAXIMUM RH AND CORRESPONDING RH
          C
          C   IBEG=1
          320 DO 200 LL=1,NSIGRAN
              WRITE (61,933) LCF(LL)
              DO 50 I=1,NLAST
                  LASTN=I
                  IF (SIGMIN(I).GT.SIGRAN(LL)) GO TO 31
                  IF (CONTINUE)
                  51 NLAST=LASTN
                  IF (SIGRAN(LL).LT.SIGMIN(I).OR.SIGMIN(LL).GT.SIGMIN(NLAST)) GOTO 220
                  DO 201 MM=2,NLAST
                      IF (SIGRAN(LL)-SIGMIN(MM)) 202,202,201
                      201 CONTINUE
335      
```

PROGRAM MARY 74774 UP#1

FTN 4.0+PSR367 29710774 13.34.68.

```

202 NX=MX
      SIGMA=(SIGGRAN(LL)-SIGMIN(MX-1))/(SIGMIN(MX)-SIGMIN(MX-1))
      RIT2(LL)=(RIT3(MX)-RIT3(MX-1))+SIGMA*RIT3(MX-1)
      TFMF=(RIT2(LL)-RIT3(LL))/(RIT3(MX)-RIT3(MX-1))
      RH2(LL)=(RHMIN(MX)-RHMIN(MX-1))*TEMP+RHMIN(MX-1)
      220 DO 250 I=1,NLAST
      WRITE(61,942) RIT3(I),RIT3(I),RTT(I),UST(I),RHCLL(I),RRATLL(I),XNST3(I),ET
      1ATT3(I),TPR3(I),SIGMIN(I)
      2,TM3ARI(I),R6TAR(I)
      250 CONTINUE
      IF(SIGGRAN(LL).LT.SIGMIN(LL).OR.SIGGRAN(LL).GT.SIGMIN(LL))GOTO 210
      WRITE(61,934) RIT2(LL),RH2(LL)
      GO TO 407
      210 WRITE(61,935) LCF(LL)
      407 IF(RHU(LL).EQ.0.) GO TO 2200
      WRITE(50,400) NLAST
      WRITE(30,401) LCF(LL)
      WRITE(30,401) XNS
      402 I=1,NLAST
      WRITE(430,401) RIT3(I),RT(I)
      IF(RT(I).LT.RTMIN) RTMIN=RT(I)
      402 CONTINUE
      IF(RHU(LL).EQ.0.) GO TO 7777
      360 DO 403 I=1,NLAST
      J=I-1
      IF(RHU(LL,I).EQ.0.) GO TO 404
      RRAT=RHU(LL,I)/RT(I)
      WRITE(30,401) RT(I),RRAT
      IF(RIT3(I).GT.BIGRIT) BIGRIT=RIT3(I)
      403 CONTINUE
      GO TO 405
      404 RRAT=RH2(LL)/RT3(I)
      WRITE(30,401) RT2(LL),RRAT
      405 CONTINUE
      00 406 I=1,JJ
      K=JJ-1
      RRAT=RHU(LL,K)/RT(K)
      WRITE(30,401) RT3(K),RRAT
      406 CONTINUE
      7777 IDOL=1RS
      WRITE(30,409) IDOL
      GO TO 206
      375

```

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PROGRAM MARY      74/74   OPT=1      FTN 4.1+PSR367    29/10/74  13.34.46.

2200 IBEG=L,L+1
      409 FORMAT(40X,1R1)
      40C FORMAT(1I5)
      401 FORMAT(2F24.6)
200  CONTINUE
      . WRITE(61,936)

385  110 CONTINUE
      REWIND 30
      IF(IFILOP.EQ.1) CALL PLOTTERINSIGRAN,LGF,RTHIN,IBEG,BIGRIT
      CONTINUE
100  GO TO 3
390  C   FORMAT STATEMENTS
      C   900 FORMAT(7A10,1A9,1R1)
      901 FORMAT(1H1,10X12HPROGRAM P559771D391UNCOOLED RADIAL TURBINE POTEN
      1ITAL STUDY/1)
      902 FORMAT(18F10.4)
      903 FORMAT(4E15.6)
      905 FORMAT(34H BETA3 GT BETA31G5) IN INPUT TABLE)
      906 FORMAT(17/11H INPUT DATA/25H DESIGN DATA AND CRITERIA/1)
      907 FORMAT(15X,9HOPT =,110,18X,10HSOLID HUB)
      908 FORMAT(15X,9HOPT =,110,18X,10HBORED HUB)
      906 FORMAT(16E12.6)
      910 FORMAT(9F8.1)
      911 FORMAT(9F8.1)
      912 FORMAT(1X,34H NUMBER OF ROTOR INLET TEMPERATURES/5X,9HNRI,D
      2,0,18X,23H NUMBER OF STRESS RANGES/5X,9RTT
      3BLADE TIP THICKNESS/5X,9HNB =,F10.4,18X,16H NUMBER OF BLADES/S
      4X,9HB,S,F. =,F10.4,18X,* BORE STRESS FACTOR VS. TURBINE SPECIFIC
      5SPEED/14X,1F10.4/14X,1F10.4/14X,1F10.4/14X,1F10.4/14X,1F10.4/14X
      5N8X,25H BLADE THICKNESS TOLERANCE/5X,9HNL =,F10.4,8X2HIN8X,42HMIN. DI STAN
      6TOLERANCE WAVE LENGTH/5X,9HARCB8 =,F10.4,8X2HIN8X,42H
      7CE BETWEEN BLADES AT HUB RADIUS/5X,9HMD =,F10.4,8X2HIN8X,42H
      8N RATIO/5X,9HRHO =,F10.4,5XAHLB/TN**35X16HMATERIAL DENSITY/5X,
      99HRA =,F10.4,8X2HIN8X,12HBORE RADIUS/5X,9HOTHERM =,F10.4,8X1
      415  *AHF9X,61HLINEAR THERMAL GRADIENT BETWEEN RIM AND CENTER OR BORE OF
      BHUB/)

912 FORMAT(1X,54HPOLYNOMIALS FOR ALLOWABLE STRESS*(PSI) VERSUS TEMP (
      1FT10X1H TEMP RANGE/)

913 FORMAT(4E16.8,F7.0,3H TO,F7.0,3H F7.0/4E16.8,F7.0,
      13H TO,F7.0/)

420

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PROGRAM MARY 75774 OPT=1

FTN 4.014PSR367 29/10/774 13.36.48.

914 FORMAT (IX,63H) POLYNOMIAL FOR YOUNG'S MODULUS OF ELASTICITY (PSI) VE

1IRSUS TEMP (F) /

915 FORMAT (6E16.8)

916 FORMAT (IX,73H) POLYNOMIAL FOR COEFFICIENT OF THERMAL EXPANSION (IN/

1IN/F) VERSUS TEMP (F) /

917 FORMAT (IX,53H) STRESS RANGES (PSI) LOW CYCLE FATIGUE LIFE (CYCLES

1)75X,6HSIGRAN,22X,3HLCF)

918 FORMAT (F11.0,19X,F9.0)

920 FORMAT (/13H TURBINE DATA / /)

921 FORMAT (5X,9HACPHAJ =,F10.4,5XTHDEGREES6X,19HABSOLUTE FLOW ANGLE/

15X,9HBETAJ =,F10.4,5X7HDEGREES6X,19HRELATIVE FLOW ANGLE/5X,9HALP

2HA6 =,F10.4,5X7HDEGREES6X,10HEXIT SWIRL/5X,9HETATT =,F10.4,5X7H

3 6X,18HTURBINE EFFICIENCY75X,9HETABUR =,F10.4,5X7H 6X

4,17HBURNER EFFICIENCY5X,9HPATIO =,F10.4,5X7H 6X13H

5 /5X,9HMG =,F10.4,18X,27 1MACH NUMBER AT EXDUCER EXIT/5X,

69HR6H =,F10.4,8X1HF9X,42HUPTRIM TEMP. FROM AERO TO STRESS CONDITION/5X,9H

7, F10.4,8X2HNGX,21HHUB RADIUS AT EXDUCER/5X,9HUPTRIM

8UCORR =,F10.4,18X,45HTURBINE TIP SPEED ENGINE DETERIORATION FACT

9DR/5X,9HODW =,F10.4,78X,16HCOMPRESSOR BLEED/7)

922 FORMAT (72H ROTOR INLET TEMPERATURES AT STRESS CONDITION SELECTED
1FOR THE STUDY (F11)

923 FORMAT (F16.4)

924 FORMAT (5X,9HOPT1 =,F10.18X,17HFREE SHAFT ENGINE)

925 FORMAT (5X,9HOPT1 =,F10.18X,19HSINGLE SHAFT ENGINE)

926 FORMAT (7716H COMPRESSOR DATA)

927 FORMAT (5X,9HP1 =,F10.4,7X4HPS1A7X,25HCOMPRESSOR INLET PRESSU

1RE/5X,9HTO =,F10.4,8X1HF9X,28HCOMPRESSOR INLET TEMPERATURE/5X

2,9HY =,F10.5,18X,2GREYNOLDS NUMBER EXPONENT/5X9RW =,F10.4,18X

310.4,6X6HLB/SEC6X,20HCOMPRESSOR MASS FLOW/5X,9HXNSMAX =,F10.4,18X

4,29HMAX COMPRESSOR SPECIFIC SPEED/5X,9HXNSMIN =,F10.4,18X,29MMIN

5COMPRESSOR SPECIFIC SPEED/5X,9HDELXNS =,F10.4,18X,35HCOMPRESSOR S

6SPECIFIC SPEED INCREMENT/5X,9HASF =,F10.4,18X,23HAERODYNAMIC SL

7IP FACTOR/5X,9HPRLOSS =,F10.4,18X,13HPRESSURELOSS/5X,9HETAMECH =,

8,FI0.4,18X,21HMECHANICAL EFFICIENCY/5X,9HETAMECH =,

928 FORMAT (59H CORRELATION OF EFFICIENCY (ETA) VERSUS SPECIFIC SPEED

1 (XN) /4X3HET11X2HXN)

929 FORMAT (759H CORRELATION OF EFFICIENCY (ETA) VERSUS PRESSURE RATIO

1 (PRY)/4X3HET11X2HPR)

930 FORMAT (1H1,12H OUTPUT DATA/ /)

931 FORMAT (20X27HCOMPRESSOR EFFICIENCY, =,F10.4,720X27HCOMPRESSOR S

1SPECIFIC SPEED =,F10.4, /)

932 FORMAT (5X,9HCOMPRESSOR PRESSURE RATIO/5X,9

455 920 FORMAT (59H CORRELATION OF EFFICIENCY (ETA) VERSUS PRESSURE RATIO

1 (PRY)/4X3HET11X2HPR)

460 930 FORMAT (1H1,12H OUTPUT DATA/ /)

PROGRAM HARY 74774 OPTED

FTN 4.1+PSR367 29/10/774 13.34.48.

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1HT2 = ,F10.4,8X1HF9X,31HCOMPRESSOR DELIVERY TEMPERATURE/5X,9HN
2   = ,F10.3,7X3HRPM8X,27HCOMPRESSOR MECHANICAL SPEED/5X,9HCTR,
3   = ,F10.4,8X2HIN8X,21HCOMPRESSOR TIP RADIUS/5X,9HH = ,F10.4,
463 46X6HBTU/LB6X,14HADIABATIC WORK/5X,9HTRP = ,28X, 22HTURBINE
5 PRESSURE RATIO/5X,9HRT = ,18X2HIN8X,18HTURBINE TIP RADIUS/5X,
69HU = ,16X6HFT/SEC6X,17HTURBINE TIP SPEED/5X,9HRT = ,18X1
7HF9X,36HROTATOR INLET TEMP AT STRESS CONDITION/5X,9HRL = ,18X2HI
8NAX,23HLOWER BLADE ROOT RADIUS/5X,9HRU = ,18X2HIN8X,23HUPPER B
9LAGE ROOT RADIUS/5X,9HRA
933 FORMAT (25H LOW CYCLE FATIGUE LIFE = ,F10.0,7H CYCLES/4X3HRTGTX2HRT
16X,*U*,10X,*RHL*,7X,*RHU*,6X,*NST*,8X,*ETATT*,6X,*TPR*,4X,*SIGMIN*
2,7X,*TM3*,7X,*R6T*)
934 FORMAT (716H MAXIMUM R.H. = ,F10.4)
935 FORMAT (/68H THIS VALUE OF SIGMA DOES NOT INTERSECT SIGMA AND THERE
1 BEFORE LCF LT ,F10.0,/)
936 FORMAT (1H)
937 FORMAT (67H CORRELATION OF EFFICIENCY (ETA) VERSUS RELATIVE FLOW A
480 1 NGLE (BETA3)/4X3HETA8X5HBETA3)
938 FORMAT (158H EFFICIENCY DECREMENT VS SPECIFIC SPEED AND PRESSURE RA
1 TIC/22X27H COMPRESSOR PRESSURE RATIOS /10X5F10.4)
939 FORMAT (4X3HXNS25X4HDETA)
940 FORMAT (150H COMPRESSOR PRESSURE RATIOS SELECTED FOR THE STUDY 75TF
485 11.0,4/))
941 FORMAT (61H CORRELATION OF EFFICIENCY (ETA) VERSUS NUMBER OF BLADE
1S (NBT)74X3HETA11X2HNA)
942 FORMAT (6F10.4,1F10.0,2F10.4)
943 FORMAT (32H XNST GT XNST(20) IN INPUT TABLE)
490 944 FORMAT (29H NB GT NB(10) IN INPUT TABLE)
END

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      SUBROUTINE S68349    74774   OPT=1   FTN 4.1+PSR367   29/10/74   13.35.07

      .SUBROUTINE SUB349(FIT,RT,U,NN,NSIGRAN,TH3)
      C
      C
      COMMON/117TIN
      COMMON/349/T1,NB,TOL,HL,ARCB8,MU,RHO,FB,T1,T2,T3,T4,OPT,A,B,C,D,A1
      A1,B1,C1,D1,A2,B2,C2,D2,AL1,AL2,AL3,AL4,AL5,AL6,E1,E2,E3,E4,E5,E6,
      28E74,BET5,SIGRAN(5),DITHERM,FUDGE,ETA3UR,XJ1,G1
      3,T1(20),TS(20),OPT3,MATC,TMAX,TABLE(150)
      COMMON/34/SIGMIN(B1),RHMIN(B1),RHUC(5,81),RHL(5,81),INDEX
      DIMENSION TABT(10),TABA(10),TABT(10),TABA(10)
      EQUIVALENCE (TABLE(10),TABT(1)),(TABLE(30),TABE(1)),(TABLE(170),TAB
      1A(1))

      REAL NA,MU
      INTEGER OPT,P,P1,P2,P3,P10,P11
      INTEGER OPT3
      U=U*12.0
      H=U/RT
      CONST=RHO*H**2/G1
      CONST0=CONST/6.0
      CONST1=CONST/12.0
      TPHI=TOL/HL
      RT2=RT*10.0
      N=RT2
      RT3=N
      DR=(RT2-RT3)*0.1
      RO=0.0
      MIN=0
      MIN1=0
      P0=0
      P1=0
      P2=0
      P3=0
      P10=0
      P11=0
      DFR=0.0
      DELR1=0.050
      R=RT-DELR
      T21000=0
      CALL FARIT(T,ETABUR,FAGAS)
      CALL EPGAS(EP,GM,FAGAS,T)
      T=T-460
      IF (R<NE-.RT) GO TO 7
      *** 00071
      *** 00072
      *** 00073
      40
      30
      25
      20
      15
      10
      5

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SUBROUTINE SUB349    74774   OPT=1   FIN 4.1*PSR367  29710774 13.35.UT.

TM3=TTT=U**2*TT2.0*G1*XJ1*CPJ*BT47BEI5
TM=TM3
GO TO 9
7 TM=TM3-R**2*(RT**2-R**2)*T1*T2*U*G1*XJ1*CPJ
9 IF(IABS (TM-T)-1.0)20,20,10
10 T=TM+460.0
GO TO 30
00078
20 IF(OPT3.EQ.2) GO TO 350
IF(TM.GE.T1.AND.TM.LE.T2) SIG=A+TM*(B+C*TH+D*TH**2)
IF(TM.GE.T2.AND.TM.LE.T3) SIG=A+TM*(31+CI*TH+DI*TH**2) 00080
IF(TM.GE.T3.AND.TM.LE.T4) SIG=A2+TM*((32+C2*TM+D2*TH**2))
IF(TM.GT.T1.AND.TM.LT.T2.AND.TM.GT.T3.AND.TM.GT.T4) GO TO 999 00081
00082
55 GO TO 360
350 IF((TM.LT.TT1(1)) GO TO 999
IF((TM.GT.TMAX) GO TO 999
GO 370 I=1.20
J=I-1
PF((TM.LT.TT1(1)) GO TO 380
370 CONTINUE
380 SIG=(TS(J+1)-TS(J))*(TM-TT1(J))/(TT1(J+1)-TT1(J))+TS(J)
360 CONTINUE
00083
65 IF(P.EQ.0.1) GO TO 130
DWL=RT-R
IF(DWL-WL)140,140,150
140 SATECONST*TR1**2=R**2)
GO TO 120
150 R1=R+WL
151 TT1=TT1+TOC
SNUH=4.*R1**3*TPhi-3.*R1**2*(2.*RT*TPhi+RT1)+(3.*TT1+2.*RT*TPhi)*R
1T**2
SUE=N(TT1*2.0*RT*TPhi)=2*U*RT*TPhi
SAT1=CONST0*SNUM/SDEN 00092
SAT2=CONST1*(R1**2-R**2)
SATE=SAT1+SAT2
120 IF(SAT-SIG150,60,60
50 DELR=DELR+.001
GO TO 40
60 R0=R
J=TT1+2.0*(RT-RO)*TPhi
00093
00094
00095
00096
00097
00098
00099
00100
SIG0=SIG
SATO=SAT
P=1
00101
00102
00103

```

SUBROUTINE SUB349 - 74774 OPT=1

FTN 4.1+PSR367 29710774 13.35.07.

```

      DELR=0.0          00117
      GO TO 40          00118
  85   DHL=RT-RO          00119
      GO TO 40          00120
      DHL=R           * * *
      IF(DWL1.LE.WL.AND.R.GE.R0)GO TO 510
      IF(DWL1.GT.WL.AND.R.GE.R0)GO TO 520
      IF(DWL1.LE.WL.OR.DWL1.GT.WL.AND.R.LT.R0)GO TO 530
      510  SAT=CONST1*(RT**2-R**2)
      SNUH=4.0*R**3*TPHI-6.0*R**2*(2.0*RT*TPhi)+(3.0*RT**2.0*RT*
      1TPHI)*RT**2
      SDEN=(TT+2.0*RT*TPhi)-2.0*R*TPhi
      STRES$CONST0* SNUM/SDEN
      TR=TT+2.0*(RT-R)*TPHI
      GO TO 470          00123
  90   530  IF(P1.EQ.1) GO TO 550          00124
      IF(P2.NE.1) GO TO 430          00125
      550  IF1=TO1-TOL          00126
      T01=TO-TOL          00127
      GO TO 440          00128
      430  T01=T1+TOL          00129
      440  T1=T0          00130
      105   R2=RC          00131
      RI=R2          00132
      R1=R2          00133
      SNUM=4.0*R1**3*TPhi-3.0*R1**2*(2.0*RT*TPhi+TT)+(3.0*TT+2.0*RT*
      1TPHI)*RT**2          00134
      SDEN=(TT+2.0*RT*TPhi)-2.0*RT*TPhi
      STRES$CONST0* SNUM/SDEN
      110   550  B5=2.0*C*R1**2-R1*R-R**2          00135
      B6=R1**2+R1*R2+R2**2          00136
      ZOL=6.C*SIG0+CONST*(2.0*R1**2-R1*R-R**2)
      VOL=6.0*SIG-CONST*(R1**2+R1*R-2.0*R**2)
      TF=(DWL1.LE.WL) GO TO 410          00137
      RT1=R+HL+DELR1
      IF(RT1.GT.RT) RT1=RT
      R3=R*WE          00138
      RT4=R+DFLR1
      T04=(RT1-R3)*TPHI          00139
      CF=CONST3*(4.0*R3**3*TPhi-6.0*RT3**2+RT1*TPhi+2.0*RT1**3*TPhi)
      DF1=CONST1*(R3**2-RT4**2)+T04          00140
      CF3=CONST0+14.*R**3*TPhi-6.*R**2*RT4*TPhi+2.*RT4**3*TPhi
      CF2=2.*CF*CF1*CF3          00141
      THIN=TO1*ZOL/VOL*CF2/SIG
      IF(P2.NE.1) GO TO 480          00142
  120
      * * *          00143
      CF=CONST3*(4.0*R3**3*TPhi-6.0*RT3**2+RT1*TPhi+2.0*RT1**3*TPhi)
      DF1=CONST1*(R3**2-RT4**2)+T04          00144
      CF3=CONST0+14.*R**3*TPhi-6.*R**2*RT4*TPhi+2.*RT4**3*TPhi
      CF2=2.*CF*CF1*CF3          00145
      THIN=TO1*ZOL/VOL*CF2/SIG
      IF(P2.NE.1) GO TO 480          00146
  125
      * * *          00147
      00150
      00151
      00152
      00153
      00154
      00155
      00156
      00157

```

SUBROUTINE SUB349 .. 74/74 OPT=1

FIN 4.1+PPSR367 - 29/10/74 13.35.07.

```

ATOL=TOL
GO TO 460
410 TMIN=T01*ZOL/VOL
480 DRERT=RL/2.0
DR2= DP1-R
ATOL=TPHI*DR2*2.0
      RERHT72.0
      IF(DR2.GE.WLH) ATOL=TOL
      460 TR=THIN*ATOL
      STRESS=TR/RT*(STRESS+CONST0*B5)+CONST0*B6
      SAT=SIG
      SIG0=SIG
      T01=THIN
      TP1=TR
      R1=R
      STRES0=SSTRESS
      P1=1
      GO TO 470
245 520 T02=TT*TOL
      R1=R+WL
      SNUM=4.0*R1**3*TPHI-3.0*R1**2*(2.0*RT*TPHI+T02)+(3.0*T02+2.0*RT
      *TPHI)*RT**2
      SDEN=(T02+2.0*RT*TPHI)-2.0*R1*TPHI
      SAT1=CONST0*SNUH/SDEN
      SAT2=CONST1*(R1**2-R**2)
      SAT=SAT1+SAT2
      SNUH=4.0*R**3*TPHI-3.0*R**2*(2.0*RT*TPHI+TT)+(3.0*TT+2.0*RT*TPHI
      *RT**2
      SDEN=(TT+2.0*RT*TPHI)-2.0*RT*TPHI
      STRESS=CONST0*SNUH/SDEN
      TT=TT+2.0*T02-RT*TPHI
      P2=1
      GO TO 470
160 470 IF(URNE,RT) GO TO 810
      C3=1.0
      GO TO 820
      810 C3=SAT/STRESS
      820 SNOM= SIG/C3
      165 VDEW*R
      SCLID=(3.0*MU)*RHO*V0**2)/(3.0*G)
      BORED=CONST1/2.0*(L3.0+MU)*R**2+(1.0-MU)*RB*RB)
      RIH=NB*TR + SIG/(6.2831853 + R)
      00194
      00195
      00196
      00197
      00199
      00200

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SUBROUTINE SUB349    74/74   OPT=1           FTN 4.1+PSR367   29/10/74  13.35.47.

      RIMBERIM*2.0*R**2/(R**2-RB*RB)
      IF(OPT.EQ.1.AND.R.EQ.RT) STRHUB=SOLID
      IF(OPT.EQ.1.AND.R.LT.RT) STRHUB=RIM+SOLID
      IF(OPT.EQ.2.AND.R.EQ.RT) STRHUB=BORED
      IF(OPT.EQ.2.AND.R.LT.RT) STRHUB=RIM8+BORED
      THA=TM-0.5*OTHERM
      THA=THA*(E2+THA*(E3+THA*(E4+THA*(E5+THA*E6)))),
      ALP=AL1+THA*(AL2+THA*(AL3+THA*(AL4+THA*(AL5+THA*AL6))))*
      GO TO 2001
      2000 J=1
      DO 2002 I=1,10
      IF(THA.LT.TABT(I)) GO TO 2003
      2002 J=I-1
      2003 E=(TABE(J+1)-TABEL(J))*((THA-TABT(J))/((TABT(J+1)-TABT(J))+TABLE(J))
      ALP=TABT(J+1)-TABT(J)*((THA-TABT(J))-TABT(J+1)-TABT(J)))
      2001 CONTINUE
      GO TO 1830,840,OPT
      830 SIGTHE=ALP*E*OTHERM*3.0
      840 F=3.0*R*RB**2-R**3-2.0*RB**3
      1900 F1=3.0*(R-RB)
      SIGTHE=ALP*E*OTHERM*(1.0+F/(F1*(R**2-RB**2)))
      850 SIGTR=STRHUB/FUDGE+SIGTHE
      ALPHA=ATAN(TR/(2.0*RI))
      DELTA=3.14159265/NB
      RIPO=SCRI(LTR/2.0)**2+R**2
      GAMMA=BFTA-ALPHA
      CH=RIPO*2.0*GAMMA
      IF(P11.EQ.1)GO TO 905
      CONTINUE
      IF(ICH-ARCB0)300,902,200
      200 IF (P3.NE.0) GO TO 220
      DELR=DR
      C
      RH1=R
      SIGTR=SIGTR
      DO 1030 JJ=1,NSIGRAN
      IF ((SIGRANKJJ)-SIGTR1) 1000,1001,1011
      1001 RHUT(JJ,NN)=R
      1000 CONTINUE
      205
      0

```

SUBROUTINE SUB349 75774 OPT=1 FTN 4.1+PSR367 29/10/74 13:35:07

```

      GO TO 230
      C
      220 IF (MIN-1) 1020,1021,1022
      1021 DELR=DELR+0.001
      SIGTR1=SIGTR
      RH1=R
      MIN=2
      MIN1=1
      GO TO 230
      1022 DECR=DELR+0.001
      GO TO 1023
      1020 DELR=DELR+0.005
      1023 SIGTPZ=SIGTR1
      SIGTR1=SIGTR
      RH2=R
      RH1=R
      IF (SIGTR1-SIGTR1) 1002,1002,1003
      1002 SIGMIN(NN)=SIGTR1
      RRMIN(NN)=RH1
      DO 1004 JJ=1,NSIGRAN
      IF (SIGRAN(JJ).LE.SIGTR2.AND.SIGRAN(JJ).GE.SIGTR1) GO TO 1005
      1005 RHU(JJ,NN)=(RH2-RH1)/(SIGTR2-SIGTR1)+(SIGRAN(JJ)-SIGTR1)+RH1
      1004 CONTINUE
      GO TO 230
      C
      C
      1003 IF (MIN1-1) 1030,1031,1032
      1030 DELLOW=DELR-0.05
      'DELR=DELR-0.15
      IF (DELR.CT.0.0) GO TO 1050
      MIN = 1
      RLOW=RH1
      >SIGLOW=SIGTR1
      DELR1=0.001
      GO TO 230
      1050 DELR = DECR+0.015
      GO TO 230
      1031 MIN=0
      DELR1=0.05
      DELR=DELLOW+0.05
      1032 DO 1006 JJ=1,NSIGRAN
      1006
  
```

SUBROUTINE SUB349 7474 OPT=1

FTN 4.1+PSR367 29/46/74 13:35:07.

1 IF (SIGRAN(JJ)-SIGRIN(NN)) .LT. 1E-6, GO TO 1008
1007 RH1(JJ,NN)=0.0
RH1(JJ,NN)=0.0
255 GO TO 1006
1008 IF (SIGRAN(JJ).GE.SIGTR2.AND.SIGRAN(JJ).LE.SIGTR1) GO TO 1009
IF (MIN1.EQ.2).GO TO 1006
SIGTR2=SIGTR1
SIGTR1=SIGLOW
RH2=RH1
RH1=RLCW
IF (SIGRAN(JJ).GE.SIGTR2.AND.SIGRAN(JJ).LE.SIGTR1) GO TO 1009
GO TO 1006
1009 1009-
RH1(JJ,NN)=(RH2-RH1)/(SIGTR2-SIGTR1)+(SIGRAN(JJ)-SIGRIN(JJ))/RH1
CONTINUE
MIN1=2
C 220 P3=1
IF (P30.EQ.1) GO TO 901
270 GO TO 40
310 P10=1
DELR=DELR+.001
901 DELR=DELR+.001
DELR1=.001
P11=1
GO TO 60
905 DELR=DELR+.001
IF (CH-ARCB9
IF (DIF.GE.0.D000) GO TO 40
902 CONTINUE
C 1010 IF (SIGRAN(JJ).NE.SIGRAN
IF (SIGRAN(JJ)-SIGTR1) 1120,1120,1011
1011 RH1(JJ,NN)=R
GO TO 1010
1120 IF (SIGRAN(JJ).GE.SIGTR1.AND.SIGRAN(JJ).LE.SIGTR1) GO TO 1121
GO TO 1010
1121 RH1(JJ,NN)=(RH1-R)/(SIGTR1-SIGTR1+(SIGRAN(JJ)-SIGRIN(JJ))/RH1)
1010 CONTINUE
C 280 GO TO 906
999 WRITE(61,998) TIT,1M
INDEX=1

SUBROUTINE SUB349 75/74 OPT=1 FTN 4.1+PSR367 29710774 13.35.07.

295 90E RETURN
908 FORMAT (//5X,10HFOR RIT = ,F5.0,24HMETAL TIP TEMPERATURE = ,F5.0,4
11H IS NOT COVERED BY YOUR STRESS POLYNOMIAL(/1)
END

```

SUBROUTINE SUB1139(XNS,ETA,W,PRM)          OPT=1           FTN 4.1+PSR367   29710774 13.35.25.

      C   CALCULATION OF COMPRESSOR EFFICIENCY AND MECHANICAL SPEED

      COMMON/11/TIN
      COMMON/1139/P1,T0,Y,ETAI(20),PRI(20),XNSI(20),PRC(5),DETAI(20,5),
     1PR(5),XN,TDH,CTR,ASF,XJ,G,RGAS,Q,OPT2
      INTEGER OPT2
      T2=T0
      IF (OPT2.EQ.2) GO TO 16
      DO 23 K=2,20
      IF (XNS-XNSI(K)) 26,26,23
      23 CONTINUE
      WRITE (61,1)
      STOP
      25
      26 K=K
      DO 24 I=2,5
      IF (PRM-PRC(I)) 11,11,24
      24 CONTINUE
      WRITE (61,3)
      STOP
      20
      21 II=I
      SL=(XNS-XNSI(KK-1))/(XNSI(KK)-XNSI(KK-1))
      DETA1=(DETAI(KK,II-1)-DETAI(KK-1,II-1))*SL+DETAI(KK-1,II-1)
      25
      DETA2=(DETAI(KK,II)-DETAI(KK-1,II))+(DETAI(KK-1,II)
      DETA=(PRM-PRC(II-1))/(PRC(II)-PRC(II-1))*(DETA2-DETA1)+DETA1
      DO 22 I=2,20
      IF (PRM-PRC(I)) 13,13,12
      12 CONTINUE
      WRITE (61,2)
      STOP
      30
      31 ETAPR=(PRM-PRC(I))/(PRI(I)-PRI(I-1))*(ETAI(I)-ETAI(I-1))+ETAI(I)
      ETAR=ETAPR+DETA
      ETAI=0-(1.0-ETAR)*IW/16.0*W
      35
      16 IF (T2.NE.T0) GO TO 18
      GAY=1.393942
      GO TO 19
      18 GAM=(1.2*(T2**5-T2**5)*1.553538E-14+25*(T2**4-T0**4)*2.102027E-
     1   1-(T2**3-T0**3)*5.566089E-5/3.)/(T2-T0)+5.124321E-5*
     2 *(T2-T0)*1.390942
      19 G1=(GAM-1)/GAN
      HAD=T0*(PRM**G1-1.0)*RGAS/W

```

SUBROUTINE SUBR39 76/74 OPT#1
 FTN 4.1+PSR367 29710774 13.35.25.
 DT=HAD/(ETA*RGAS*G1)
 T2=T
 T2=T0+DT
 45 IF (ABS(T2-T1)-0.01)>30,30,16
 30 IF (QPT2.E0.2) GO TO 22
 XN=XNS*HAD**0.75*SOR(TQ)
 GO TO 21
 22 XNS=XN*SOR(TQ)/HAD**0.75
 21 QP1=SORT(G*HAD)/20.//3.14159*XN)/SART(ETA)/ASF
 CTR=DTT*0.5
 25 DH=DT*RGAS/(XJ*G1)
 TIN=T2-460.0
 55 1 FORMAT (38H XNS .GT. THAN XNS1(20) IN INPUT TABLE)
 2 FORMAT (36H PR .GT. THAN PR1(20) IN INPUT TABLE)
 3 FORMAT (35H PR .GT. THAN PRC(5) IN INPUT TABLE)
 RETURN
 ENO

16774 ORT=1
SUBROUTINE FAR 29710774 13.35.330.

FTN 4.1+PSR367

SUBROUTINE FAR(TIT,ETABUR,FAGAS)

C CALCULATION OF FUEL-AIR RATIO

COMMON/11/TIN

TIR=TIN+460.0

DTCURN=TT-TIN

FA400=6.9379E-5+DTBURN*(1.2447E-5+DTJRN*1.3144E-9)

FA200=1.1024E-5+DTBURN*(1.6065E-5+DTURN*1.1674E-9)

FA10=(FA400*(2000.-TIR)+FA200)*(TIR-400.)/1600.

FAGAS=FA10/ETABUR

RETURN

END

9

SUBROUTINE CP GAS 74774 ONUFI FIN 4.1+PSR367 2971.0774 13.35.36.

SUBROUTINE CP GAS(1CP, GAM, FGAS, T)

C CALCULATION OF SPECIFIC HEAT AND SPECIFIC HEAT RATIO

5 CP=2.4841E-1+T*(4.7464E-5-T*(7.0933E-8-T*(2.6510E-11-T*3.2585E-15)
1))
CP=(FGAS/.0334*(((.12121E-11*T-.1041E-7)*T+.36151E-4)*T+.0139)+1.
1)*CP
GAM=778.26*CP/(778.26*CP-53.375)

10 RETURN

ENO

SUBROUTINE FILE 74774 OPT=1

```

FTN 4.1+PSR367 29710774 13.35.38.

SUBROUTINE FILE(NAME,NO,TABLE)
DIMENSION PFN(4),TABLE(11)
INTEGER TABLE,PFN,PH,CY
DATA PFN/10HDACLHATERY,BLAUTABLES,0,0/
*DATA PH/4RUACL /
DATA LFN/6LTAPE20/
DATA CY/2/
DATA IREC / 0 /
IF(IREC.EQ.0) CALL ATTACH(LFN,PFN,CY,PH)
10 IF(IREC.EQ.NO) RETURN
IPOS=NO-IPEC
IF(IPOS.GT.0) GO TO 1
IPOS=IABS(IPOS)+1
DO 2 I=1,IPOS
2 BACKSPACE 20
IPOS=1
DO 3 I=1,IPOS
3 BUFFER IN(20,1) (TABLE(1),TABLE(150))
IF(UNIT(20).EQ.5)
5 PRINT 100,NO
STOP
10D FORMAT(* MATERIAL TABLE NO *,14,* NOT AVAILABLE*)
3 CONTINUE
IREC=NO
NAME=TABLE
RETURN
END
25

```

SUBROUTINE PLOTTER 7474 OPT=1

FTN 4.1 FIPS R367 29710774 13.35.43.

```

SUBROUTINE PLOTTER(NSIGRANT,CF,RITMIN,IBEG,BIGRIT).
      DIMENSION LCF(15),RIT(100),RT(100),RH(100)
      COMMON/HEDR/ITITLE(16),PRM
      REAL LCF
      DO 9 J=IBEG,NSIGRAN
      REWIND 30
      23 READ(30,99) NLAST
      IF(EOF(30),NE.0) RETURN
      READ(30,98) CYCLE
      READ(30,98) XNS
      IF(CYCLE,NE.LCF(IJ)) GO TO 20
      DO 8 K=1,NLAST
      READ(30,98) RT(K),RT(K)
      8 CONTINUE
      15 GO TO 21
      20 READ(30,97) JACK
      IF(JACK,NE.1RS) 20,23
      21 CONTINUE
      CALL SCALE(RIT,4.,NLAST,1,10.)
      RT(NLAST)=RITMIN
      CALL SCALE(RIT,4.,NLAST,1,10.)
      YTICK=RIT(NLAST+1)
      YDEL=RIT(NLAST+2)
      XTICK=RIT(NLAST+1)
      XDEL=RIT(NLAST+2)
      CALL AXIS1(9.,4.,6HRT(IN),-6,6,0,0,RT(NLAST+1),RT(NLAST+2),10.)
      CALL AXIS1(9.,4.,1H,1,6,.90,RT(NLAST+1),RT(NLAST+2),10.)
      CALL PLOT(9.,4.,-3)
      NLAST
      REWIND 30
      READ(30,99) NLAST
      READ(30,98) CYCLE
      READ(30,98) XNS
      29 IF(CYCLE,NE.LGF(IJ)) GO TO 30
      DO 6 K=1,NLAST
      31 READ(30,98) RT(K),RT(K)
      6 READ(30,98) RT(NLAST+1)
      RT(NLAST+1)=XTICK
      RT(NLAST+2)=XDEL
      RT(NLAST+1)=YTICK
      RT(NLAST+2)=YDEL
      RT(NLAST)=YVAL
      CALL LINE(RT,RIT,NLAST,1,0,55)
      XVAL=(RT(NLAST)-RT(NLAST+1))/RT(NLAST+2)
      30
      35
      40

```

```

SUBROUTINE PLOTTER    74/74   OPT=1          FTN 4.1+PSR367    29/10/74  13.35.43.

      YVAL=(RIT(NLAST)-RIT(NLAST+1))/RIT(NLAST+2)
      CALL SYMBOL(XVAL,YVAL,.1,5HNS,7,185,5)
      CALL NUMBER(999,999,.1,XNS,85,5H,F4,0)
      95 FORMAT(1F4.0)

      30 READ(30,97) JACK
      IF(JACK.EQ.1RS) GO TO 30
      READ(30,99) NLAST
      IF(ECF(30).NE.0.) GO TO 7
      READ(30,98) CYCLE
      READ(30,98) XNS
      IF(CYCLE.NE.LCF(J)) 30,31
      7 CALL PLOT(0,0,3)
      CALL SYMBOL(-7.5,2,1,TITLE,0,80)
      CALL SYMBOL(-7.5,1,ITITLE,9,0,80)
      CALL SYMBOL(-7.4,1,8HL,C,F,=,J,8)
      CALL NUMBER(999,999,1,LCF(J),0,5HIF8,0)
      CALL SYMBOL(999,999,1,12H,P,R,=,0,12)
      CALL NUMBER(999,999,1,PRM,0,5HIF8,0)
      31 C PLOT RH VS R.I.T.
      C
      C REWIND 30
      48 READ(30,99) NLAST
      IF(ECF(30).NE.0.) GO TO 5
      READ(30,98) CYCLE
      READ(30,98) XNS
      IF(CYCLE.NE.LCF(J)) GO TO 49
      DO 66 I=1,NCAST
      READ(30,98) DUMMY
      66 CONTINUE
      LEO
      51 L=L+1
      READ(30,98) RIT(L),RH(L),JACK
      IF(JACK.EQ.1RS) 50,51
      49 READ(30,97) JACK
      IF(JACK.EQ.1RS) 48,49
      50 L=-1
      IF(L.EQ.0) GO TO 48
      RIT(L)=BIGRIT
      CALL SCALE1(PH,S,L,1,10,1)
      XTRCK=SH(L+1)
      XDEL=RH(L+2)
      80

```

SUBROUTINE PLOTTER 7474 OPT=1.

FYN 4.1+PSR367 29/10/74 13.35.43.

```

 85      YEL
        RIT(L+1)=YTICK
        RIT(L+2)=YDEL
        CALL AXIS1(-7.,0.,60RH/RIT,-6.,50.,0.,RH(L+1),RH(L+2),10.)
        CALL AXIS1(-7.,0.,15HR.I.T.(OEG. F.),35.,4.,90.,RIT(N+1),RIT(N+2),1
        10.)
        CALL PLOT(-7.,0.,-3)
        REWIND 30
        READ(30,99) NLAST
        IF(EOF(30).NE.J.) GO TO 5
        READ(30,98) CYCLE
        READ(30,98) XNS
        IF(CYCLE.NE.CYCLE) GO TO 47
        DO 4 IN=1,NLAST
        READ(30,98) DUMMY
        4 CONTINUE
        L=0
        53 L=L+1
        READ(30,98) RIT(L),RHT(L),JACK
        IF(JACK.EQ.1RS) 52,53
        57 READ(30,92) JACK
        IF(JACK.EQ.1IRE) 60,67
        52 L=L-1
        RH(L+1)=XTICK
        RH(L+2)=XTICK
        RIT(L+1)=YTICK
        RIT(L+2)=YDEL
        CALL LINE(RH,RIT,L,1,0.55)
        XVAL=(RH(L)+RH(L+2))/2.1-RH(L+1))/RH(L+2)
        YVAL=(RIT(L)+RIT(L+2))/2.1-RIT(L+1))/RIT(L+2)
        115 CALL SYMBOL(XVAL,YVAL,.1,SHNS=.92,.97)
        CALL NUMBER(99.,999.,.1,XNS,.95,.9H1F4,0)
        GO TO 60
        5 CONTINUE
        CALL PLOT(12.,-4.,-3)
        9 CONTINUE
        REWIND 30
        RETURN
        99 FORMAT(1I15)
        98 FORMAT(2F20.0,1R1)
        125  97 FORMAT(40X,1R1)
        END

```

APPENDIX II

SAMPLE CALCULATION

APPENDIX II

A sample calculation is presented to show the flexibility of the computer program and it's handling of cases of varying complexity.

The input data are show from page 6 to page 12 and they are described below.

Page 6

NRIT	= No. of Rotor Inlet Temperature to be analyzed.
NSIGRAN	= No. of stress ranges to be analyzed.
OPT	= Solid or Bored Hub
OPT 1	= Single or free shaft engine
OPT 2	= Input specific speed or mechanical speed
OPT 3	= Allowable stresses of material used input as a table or as 3rd order polynomials.
MAT	= A code that defines the material used.
PLOT	= Plotting or not plotting of results
TT	= Blade tip thickness
NB	= Rotor number of blades
BSF 1	= A linear variation of the "bore stress factor" as a function of specific speed of turbine (see Fig. 4.16)
BSF 2	
NST 1	
NST 2	
TOL	= Blade thickness tolerance (\pm)
WL	= Wave length over which the blade thickness varies from min. to max.
ARCBB	= Minimum arc allowed between two adjacent blades
MU	= Poisson's ratio of material

RHO	= Density of material.
RB	= Bore radius of turbine hub
T1 T2 T3 T4}	= Metal temperatures defining the region over which the stress polynomials are valid
Δ THERM	= Thermal gradient between center or bore, and hub rim (Positive when bore or center colder than rim).
A, B, C, D. A1, B1, C1, D1 A2, B2, C2, D2	= Coefficient of 3rd order stress polynomials as function of metal temperature

Page 7

T G}	= Table of allowable stress of material versus metal temperature, if stress polynomials are not used (See Fig. 1)
---------	---

Page 8

E1 E2 E3 E4 E5 E6}	= Coefficients of 5th order polynomial giving the Young's Modulus as a function of metal temperature (Only if MAT = 0)
AL1 AL2 AL3 AL4 AL5 AL6}	= Coefficients of 5th order polynomial giving the coefficient of thermal expansion as a function of metal temperature (Only if MAT = 0)
SIGRAN (I)	= Stress range at the bore or hub center capable of giving a max. low cycle fatigue life. LCF (I)
LCF (I)	= Low cycle fatigue life corresponding to SIGRAN (I)

Page 9

ALPHA 3	= Absolute flow angle at turbine inlet
BETA 3	= Relative flow angle at turbine inlet
ALPHA 6	= Absolute flow angle at turbine exit
M'6	= Mach Number at turbine exit
R6H	= Radius of the hub at turbine exit
ETATT	= Turbine efficiency
ETABUR	= Burner efficiency
RATIO	= Constant to calculate the turbine pressure ratio in a single shaft engine
UPTRIM	= Rotor Inlet Temperature uptrip due to engine deterioration
U CORR	= Turbine tip speed correction due to engine deterioration
ΔW	= Air bleed
RIT	= Rotor inlet temperature to be analyzed

Page 10

DELETA	= Turbine efficiency decrement as a function of relative flow angle (see Fig. 6.1 b)
BETA 3	
DELTB	= Turbine efficiency decrement as a function of turbine number of blades (see Fig. 6.1 a)
N.B.	
ETATT	= Turbine efficiency as a function of
NST	turbine specific speed (see Fig. 6.1 c)

Page 11

P_1	= Compressor inlet pressure
T_0	= Compressor inlet temperature
γ	= Reynold's Number Exponent
W_c	= Compressor mass flow
PR LOSS	= Pressure loss in combustor
XN	= Mechanical speed of compressor
XNSMAX	= Max. specific speed to be analyzed
XNSMIN	= Min. specific speed to be analyzed
DELXNS	= Specific speed decrement
ASF	= Aerodynamic Slip Factor of compressor
ETAMECH	= Mechanical efficiency of gearbox
ETA	= Compressor efficiency
PR _c	= Pressure ratios of compressor to be analyzed.
XNS _c	= Compressor efficiency decrements as a function of specific speed and compressor pressure ratio (see fig. 5.1 b).
$\Delta\eta_c$	

Page 12

ETA PR1}	= Compressor adiabatic efficiency
PR 1}	versus pressure ratio (see fig. 5.1 a)

The computer program output follows and they are self-explanatory (see Section 7 for more on the output interpretation).

MINIMUM PROPERTIES FOR THE DESIGN
OF THE RADIAL TURBINE BLADE

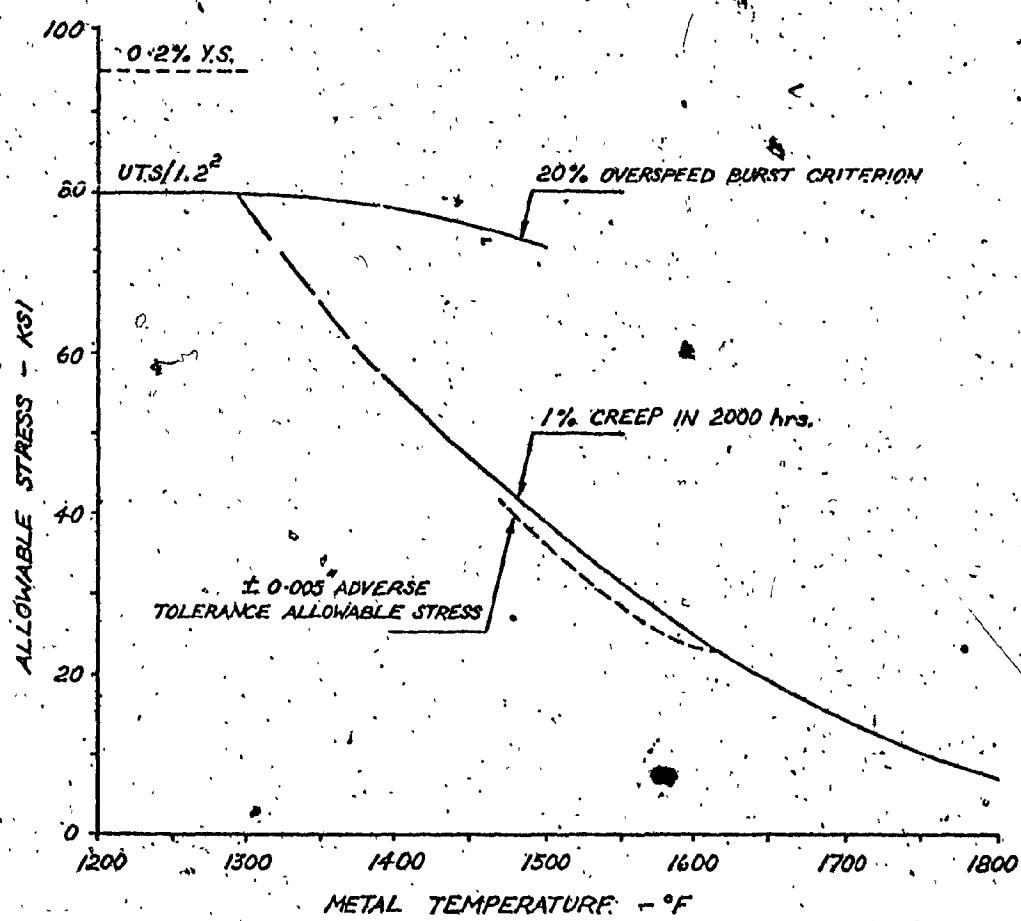


FIG. I.
APPENDIX 2.

PROGRAM MARY

NAME	LOCAL	DATE	COMP. IDENT.
S. Berghenti	7416	11/10/74	3207340
DESCRIPTION OF RUN		70H	
EXAMPLE		70H	

DESIGN DATA AND CRITERIA

NRIT	NSIGRAN	OPT	OPT1	OPT2	OPT3	MAT	PLOT	I4.
6	2	2	1	1	2	3	1	
TT(in)	NB	BSF1	NST1	BSF2	NST2			F10
.040	12.	1.41	68.	1.14	96.4			

OPT = 1 SOLID HUB
 = 2 BORED HUB
 OPT1 = 1 SINGLE SHAFT
 = 2 FREE SHAFT
 OPT2 = 1 INPUT SPECIFIC SPEED
 = 2 INPUT MECHANICAL SPEED
 OPT3 = 1 STRESS POLYNOMIALS
 = 2 STRESS VS.TEMP. TABLE
 PLOT = 0 NO PLOT
 = 1 PLOT

TOL(ir)	WL(in)	ARCBB(in)	MU	RHO(1b/in ³)	RB(in)	F10
.005	.5	.2	.3	.28	.5	

T1(°F)	T2(°F)	T3(°F)	T4(°F)	DITHERM(°F)	MAX NRIT = 8
				150.	F10 MAX NSIGRAN = 5

Polynomials for Allowable Stress vs Temp.

$$\text{ALLOWABLE} = A + B*T + C*T^2 + D*T^3 \quad (\text{PSI})$$

A	B	C	D
0			
A1	B1	C1	D1
0			
A2	B2	C2	D2
0			

ALWAYS 3 CARDS

4E15

PROGRAM MARY

T(F)	G (PSI)
1. 70.	80,000.
2. 1200.	80,000.
3. 1290.	80,000.
4. 1350.	66,000.
5. 1400.	55,000.
6. 1450.	46,500.
7. 1500.	38,500.
8. 1550.	31,000.
9. 1600.	25,000.
10. 1650.	19,000.
11. 1700.	14,000.
12. 1750.	10,000.
13. 1800.	7,000.
14. 1900.	1,000.
15. 0	
16.	
17.	
18.	
19.	
20.	

F10

NB: A BLANK CARD MUST FOLLOWS THE LAST T AND G.

PROGRAM MARYDESIGN DATA AND CRITERIAPOLYNOMIAL FOR YOUNG'S MOD. OF ELAST. VS TEMP. (PSI)

$$E = E_1 + E_2 \cdot T + E_3 \cdot T^2 + E_4 \cdot T^3 + E_5 \cdot T^4 + E_6 \cdot T^5$$

E1	E2	E3	E4	E5	E6	6E12

POLYNOMIAL FOR COEFF. OF THERMAL EXPANSION VS TEMP. (in/in °F)

$$\alpha = AL_1 + AL_2 \cdot T + AL_3 \cdot T^2 + AL_4 \cdot T^3 + AL_5 \cdot T^4 + AL_6 \cdot T^5$$

AL1	AL2	AL3	AL4	AL5	AL6	6E12

SIGRAN(I) PSI	LCF(I) CYCLES	F10-
1 100,000.	50,000.	
2 90,000.	100,000.	
3		
4		
5		

- NB. 1) MAX. OF 5.
 2) NO. OF LINES MUST EQUAL NSIGRAN.
 3) LCF(I) MUST BE IN INCREASING ORDER.

DISREGARD BLANK LINES

NB: IF MAT # 0 NO BLANK CARDS FOR E & α

PROGRAM MARYTURBINE DATAALL F10.0

ALPHA3	BETA3	ALPHA6	M6	R6H
68.	10.	5.	.4	1.2

ETATT	ETABUR	RATIO	UPTRIM	UCORR	ΔW
0.	.98	.94	90.	0.0	0.0

Note: If ETATT is input as zero, program calculates ETATT from the tables on Page 4.
 If ETATT is input, Page 4 can be ignored.

RIT(I)	1500.	1600.	1700.	1800.	1900.	2000.	

Note: 1) The number of RIT(I) must equal NRIT input on Page 1.
 2) Maximum number of RIT(I) = 8; Minimum number of RIT(I) = 2.
 3) RIT(I)'s must be in increasing order and in 100°F increments.

PROGRAM MARYTURBINE DATAALL F10.0

	DELETA	BETA3
1	.003	5.
2	.005	0.
3	.008	5.
4	.011	10.
5	.015	15.

	DELETB	N.B.
1	.02	11.
2	.015	12.
3	.010	13.
4	.007	14.
5	.0045	15.
6	.0025	16.
7	.0015	17.
8	.0005	18.
9	.0002	19.
10	0.0	20.

	ETATT	NST
1	.881	55.
2	.888	60.
3	.8935	65.
4	.8975	70.
5	.9000	75.
6	.9010	80.
7	.9000	85.
8	.8975	90.
9	.8925	95.
10	.8840	100.
11	0	
12	0	
13	0	
14	0	
15	0	
16	0	
17	0	
18	0	
19	0	
20	0	

TURBINE EFFICIENCY DECREMENT VS RELATIVE
INLET FLOW ANGLE

NB. 5 CARDS MUST BE INPUT

TURBINE EFFICIENCY DECREMENT VS NUMBER
OF BLADES ..

NB. 10 CARDS MUST BE INPUT

TURBINE EFFICIENCY VS TURBINE SPECIFIC
SPEED.

NB. 20 CARDS MUST BE INPUT.

PROGRAM MARY

COMPRESSOR DATA

P1	T0	Y	Wc	PRLOSS	XN	6F10
14.7	59.	.05	4.0	.02		

XNSMAX	XNSMIN	DELXNS	A.S.F.	ETAMECH.	ETA	6F10
100.	80.	10:	.92	1.0		

COMPRESSOR PRESSURE RATIOS

4.	5.	6.			5F10
4.	5.	6.			

Note: If OPT2 = 1 input XNS; If OPT2 = 2 input XN and ETA

If XN and ETA are input no blanks or any data for XNSC vs. η decr.

COMPRESSOR EFFICIENCY DECREMENT VS. SPECIFIC SPEED FOR VARIOUS PRESSURE RATIOS

XNSC	COMPRESSOR PRESSURE RATIOS			5F10
	4.	5.	6.	
50.	- .68	- .037	- .020	
55.	- .050	- .023	- .010	
60.	- .037	- .014	- .004	
65.	- .025	- .007	- .001	
70.	- .017	- .003	- 0.0	
75.	- .010	- .001	- .001	
80.	- .005	- 0.0	- .004	
85.	- .002	- .001	- .008	
90.	- .001	- .003	- .015	
95.	- 0.0	- .006	- .022	
100.	- .001	- .010	- .031	
105	- .002	- .014	- .040	
0				
0				
0				
0				
0				
0				
0				
0				
0				

NB. 21 CARDS MUST BE INPUT

PROGRAM MARY

COMPRESSOR DATACOMPRESSOR EFFICIENCY VS. COMPRESSOR PRESSURE RATIO

	ETAPRI	PRI
1	.90	2.
2	.89	3.
3	.8775	4.
4	.855	5.
5	.825	6.
6	.798	7.
7	.7825	8.
8	.7725	9.
9	.7670	10.
10	.7625	11.
11	0	
12	0	
13	0	
14	0	
15	0	
16	0	
17	0	
18	0	
19	0	
20	0	

2F10

NB. 20 CARDS MUST BE
INPUT

NB: IF XN AND ETA ARE INPUT DISREGARD THIS INPUT SHEET

UNCOOLED RADIAL TURBINE POTENTIAL STUDY

S'BERGHEUTI EXAMPLE
7416. 141074. 3207340.

INPUT DATA

DESIGN DATA AND CRITERIA

OPT	=	2	BORED HUB
OPT1	=	1	SINGLE SHAFT ENGINE
NRIT	=	6	NUMBER OF ROTOR INLET TEMPERATURES
NSTGRAN	=	2	NUMBER OF STRESS RANGES
TT	=	.0400	IN
NB	=	12.0000	BLADE TIP THICKNESS
B.S.F.	=	1.4100	NUMBER OF BLADES
		6.8.0700	BORE STRESS FACTOR VS. TURBINE SPECIFIC SPEED
		1.1400	
TOL	=	96.4000	
HL	=	.0050	IN
ARCBB	=	.5000	IN
MU	=	.2000	IN
RHO	=	.3000	MIN. DISTANCE BETWEEN BLADES AT HUB RADIUS
RB	=	.2800	POISSON RATIO
OTHERM	=	.5000	MATERIAL DENSITY
		.5000	IN
		.5000	BORE RADIUS
		.5000	LINEAR THERMAL GRADIENT BETWEEN RIM AND CENTER QR. BORE OF HUB

INPUT TABLE OF STRESS VS TEMPERATURE

TEMPERATURE STRESS

70.00	80000.00
1200.00	80000.00
1290.00	80000.00
1350.00	66000.00
1400.00	55000.00
1450.00	46500.00
1500.00	38500.00
1550.00	31000.00
1600.00	25000.00
1650.00	19000.00

MATERIAL CODE	TEMPERATURE	YOUNG'S MODULUS	THERMAL EXPANSION
3			
7C0C00E+02	309000E+08	.720000E-05	
560100E+03	294000E+08	.744000E-05	
900000E+03	276000E+08	.770000E-05	
120000E+04	261000E+08	.803000E-05	
130000E+04	255000E+08	.815000E-05	
150000E+04	242000E+08	.870000E-05	
160000E+04	234000E+08	.895000E-05	
170000E+04	227500E+08	.921000E-05	
180000E+04	220000E+08	.950000E-05	
190000E+04	214000E+08	.980000E-05	
STRESS RANGES (PSI) LOW CYCLE FATIGUE LIFE (CYCLES)			
SIGRAN	LCF		
100000.	500000.		
900000.	1000000.		
TURBINE DATA			
ALPHAJ	= 6.6.0000	DEGREES	ABSOLUTE FLOW ANGLE
BETAJ	= 10.0.0000	DEGREES	RELATIVE FLOW ANGLE
ALPHAB	= 5.0000	DEGREES	EXIT SWIRL
ETATT	= -0.0.0000		TURBINE EFFICIENCY
ETABUR	= .9800		BURNER EFFICIENCY
RATIO	= .9400		
M6	= .4600		MACH NUMBER AT EXDUCER EXIT
R6H	= .2000	IN	HUB RADIUS AT EXDUCER
UPTRIM	= 9.0.0000	F	UPTRIM TEMP. FROM AERO TO STRESS CONDITION
UCORR	= -0.0.000		TURBINE TIP SPEED ENGINE DETERIORATION FACTOR
DW	= -0.6000		COMPRESSOR BLEED
ROTOR INLET TEMPERATURES AT STRESS CONDITION SELECTED FOR THE STUDY (F)			
1500.0000			
1600.0000			
1700.0000			
1800.0000			

1900.0000

2000.0000

CORRELATION OF EFFICIENCY (ETA) VERSUS RELATIVE FLOW ANGLE (BETA3)

ETA	BETA3
-0.0030	-5.0000
-0.0050	-0.0000
-0.0080	5.0000
-0.0110	10.0000
-0.0150	15.0000
0.0150	15.0000
0.0200	1.0000
0.0250	12.0000
0.0300	13.0000
0.0350	14.0000
0.0450	15.0000
0.0025	16.0000
0.0015	17.0000
0.0005	18.0000
-0.0002	19.0000
-0.0000	20.0000
0.0010	20.0000
0.0020	20.0000
0.0030	20.0000
0.0040	20.0000
0.0050	20.0000
0.0060	20.0000
0.0070	20.0000
0.0080	20.0000
0.0090	20.0000
0.0100	20.0000
0.0110	20.0000
0.0120	20.0000
0.0130	20.0000
0.0140	20.0000
0.0150	20.0000
0.0160	20.0000
0.0170	20.0000
0.0180	20.0000
0.0190	20.0000
0.0200	20.0000
0.0210	20.0000
0.0220	20.0000
0.0230	20.0000
0.0240	20.0000
0.0250	20.0000
0.0260	20.0000
0.0270	20.0000
0.0280	20.0000
0.0290	20.0000
0.0300	20.0000
0.0310	20.0000
0.0320	20.0000
0.0330	20.0000
0.0340	20.0000
0.0350	20.0000
0.0360	20.0000
0.0370	20.0000
0.0380	20.0000
0.0390	20.0000
0.0400	20.0000
0.0410	20.0000
0.0420	20.0000
0.0430	20.0000
0.0440	20.0000
0.0450	20.0000
0.0460	20.0000
0.0470	20.0000
0.0480	20.0000
0.0490	20.0000
0.0500	20.0000
0.0510	20.0000
0.0520	20.0000
0.0530	20.0000
0.0540	20.0000
0.0550	20.0000
0.0560	20.0000
0.0570	20.0000
0.0580	20.0000
0.0590	20.0000
0.0600	20.0000
0.0610	20.0000
0.0620	20.0000
0.0630	20.0000
0.0640	20.0000
0.0650	20.0000
0.0660	20.0000
0.0670	20.0000
0.0680	20.0000
0.0690	20.0000
0.0700	20.0000
0.0710	20.0000
0.0720	20.0000
0.0730	20.0000
0.0740	20.0000
0.0750	20.0000
0.0760	20.0000
0.0770	20.0000
0.0780	20.0000
0.0790	20.0000
0.0800	20.0000
0.0810	20.0000
0.0820	20.0000
0.0830	20.0000
0.0840	20.0000
0.0850	20.0000
0.0860	20.0000
0.0870	20.0000
0.0880	20.0000
0.0890	20.0000
0.0900	20.0000
0.0910	20.0000
0.0920	20.0000
0.0930	20.0000
0.0940	20.0000
0.0950	20.0000
0.0960	20.0000
0.0970	20.0000
0.0980	20.0000
0.0990	20.0000
0.1000	20.0000
0.1010	20.0000
0.1020	20.0000
0.1030	20.0000
0.1040	20.0000
0.1050	20.0000
0.1060	20.0000
0.1070	20.0000
0.1080	20.0000
0.1090	20.0000
0.1100	20.0000
0.1110	20.0000
0.1120	20.0000
0.1130	20.0000
0.1140	20.0000
0.1150	20.0000
0.1160	20.0000
0.1170	20.0000
0.1180	20.0000
0.1190	20.0000
0.1200	20.0000
0.1210	20.0000
0.1220	20.0000
0.1230	20.0000
0.1240	20.0000
0.1250	20.0000
0.1260	20.0000
0.1270	20.0000
0.1280	20.0000
0.1290	20.0000
0.1300	20.0000
0.1310	20.0000
0.1320	20.0000
0.1330	20.0000
0.1340	20.0000
0.1350	20.0000
0.1360	20.0000
0.1370	20.0000
0.1380	20.0000
0.1390	20.0000
0.1400	20.0000
0.1410	20.0000
0.1420	20.0000
0.1430	20.0000
0.1440	20.0000
0.1450	20.0000
0.1460	20.0000
0.1470	20.0000
0.1480	20.0000
0.1490	20.0000
0.1500	20.0000
0.1510	20.0000
0.1520	20.0000
0.1530	20.0000
0.1540	20.0000
0.1550	20.0000
0.1560	20.0000
0.1570	20.0000
0.1580	20.0000
0.1590	20.0000
0.1600	20.0000
0.1610	20.0000
0.1620	20.0000
0.1630	20.0000
0.1640	20.0000
0.1650	20.0000
0.1660	20.0000
0.1670	20.0000
0.1680	20.0000
0.1690	20.0000
0.1700	20.0000
0.1710	20.0000
0.1720	20.0000
0.1730	20.0000
0.1740	20.0000
0.1750	20.0000
0.1760	20.0000
0.1770	20.0000
0.1780	20.0000
0.1790	20.0000
0.1800	20.0000
0.1810	20.0000
0.1820	20.0000
0.1830	20.0000
0.1840	20.0000
0.1850	20.0000
0.1860	20.0000
0.1870	20.0000
0.1880	20.0000
0.1890	20.0000
0.1900	20.0000
0.1910	20.0000
0.1920	20.0000
0.1930	20.0000
0.1940	20.0000
0.1950	20.0000
0.1960	20.0000
0.1970	20.0000
0.1980	20.0000
0.1990	20.0000
0.2000	20.0000

COMPRESSOR DATA

P1	=	14,7000	PSIA	COMPRESSOR INLET PRESSURE
T0	=	59.0000	F	COMPRESSOR INLET TEMPERATURE
Y	=	.3500		REYNOLDS NUMBER EXPONENT
H	=	4.0000	LB/SEC	COMPRESSOR MASS FLOW
XNSMAX	=	100.0000		MAX COMPRESSOR SPECIFIC SPEED
XNSMIN	=	60.0000		MIN COMPRESSOR SPECIFIC SPEED
DELXNS	=	10.0000		COMPRESSOR SPECIFIC SPEED INCREMENT
ASF	=	.9200		AERODYNAMIC SLIP FACTOR
PRLOSS	=	.0230		PRESSURE LOSS
ETAMECH	=	1.0000		MECHANICAL EFFICIENCY

COMPRESSOR PRESSURE RATIOS SELECTED FOR THE STUDY

4.0000
5.0000
6.0000
-0.0000

INTRODUCTION

COMPRESSOR PRESSURE RATIOS					
XNS	4.6930	5.8000	6.0000	-0.0000	-0.0000
50.0000	-0.680	-0.0370	-0.0200	-0.0000	-0.0000
55.0000	-0.0500	-0.0230	-0.0100	-0.0000	-0.0000
60.0000	-0.0370	-0.0140	-0.0040	-0.0000	-0.0000
65.0000	-0.0250	-0.0070	-0.0010	-0.0000	-0.0000
70.0000	-0.0170	-0.0030	-0.0000	-0.0000	-0.0000
75.0000	-0.0100	-0.0010	-0.0010	-0.0000	-0.0000
80.0000	-0.0050	-0.0000	-0.0040	-0.0000	-0.0000
85.0000	-0.0020	-0.0019	-0.0080	-0.0000	-0.0000
90.0000	-0.0010	-0.0030	-0.0150	-0.0000	-0.0000
95.0000	-0.0000	-0.0060	-0.0220	-0.0000	-0.0000
100.0000	-0.0010	-0.0160	-0.0310	-0.0000	-0.0000
105.0000	-0.0020	-0.0140	-0.0400	-0.0000	-0.0000
-5.0000	-0.0000	-0.0000	-0.0000	-0.0000	-0.0000
-0.0010	-0.0000	-0.0000	-0.0000	-0.0000	-0.0000
-0.0020	-0.0000	-0.0000	-0.0000	-0.0000	-0.0000
-0.0030	-0.0000	-0.0000	-0.0000	-0.0000	-0.0000
-0.0040	-0.0000	-0.0000	-0.0000	-0.0000	-0.0000
-0.0050	-0.0000	-0.0000	-0.0000	-0.0000	-0.0000
-0.0060	-0.0000	-0.0000	-0.0000	-0.0000	-0.0000
-0.0070	-0.0000	-0.0000	-0.0000	-0.0000	-0.0000
-0.0080	-0.0000	-0.0000	-0.0000	-0.0000	-0.0000
-0.0090	-0.0000	-0.0000	-0.0000	-0.0000	-0.0000
-0.0100	-0.0000	-0.0000	-0.0000	-0.0000	-0.0000

CORRELATION OF EFFICIENCY (ETA) VERSUS PRESSURE RATIO (PR)

ETA	PR
.9000	2.0000
.8900	3.0000
.8775	4.0000
.8550	5.0000
.8250	6.0000
.7980	7.0000
.7825	8.0000
.7725	9.0000
.7670	10.0000
.7625	11.0000
.0.0000	0.0000
0.0000	-0.0000
-0.0000	-0.0000
-0.0000	-0.0000
-0.0000	-0.0000
-0.0000	-0.0000
-0.0000	-0.0000
-0.0000	-0.0000
-1.0000	-0.0000

OUTPUT DATA

COMPRESSOR EFFICIENCY = .8002
 COMPRESSOR SPECIFIC SPEED = 80.0000

	PR	4.0000	COMPRESSOR PRESSURE RATIO				
	12	= 34.3.6848	F	COMPRESSOR DELIVERY TEMPERATURE			
N	= 35334.444	RPM	MECHANICAL SPEED				
CTR	= 4.6225	IN.	COMPRESSOR TIP RADIUS				
H	= 68.6724	BTU/TB	ADIABATIC WORK				
TPR	=	IN	TURBINE PRESSURE RATIO				
RT	=	FT/SEC	TURBINE TIP SPEED				
U	=	F	ROTQR. INLET TEMP. AT STRESS CONDITION				
RIT	=	IN	LOWER BLADE ROOT RADIUS				
RHL	=	IN	UPPER BLADE ROOT RADIUS				

LOW CYCLE FATIGUE LIFE = 50000. CYCLES

	RT	U	RHL	NST	ETATT	TPR	SIGMIN	TN3	RGT
1.500.0000	5.5082	169.8.4722	.77.92	3.5638.	70.4287	8715	3.7600	55.971	1234.3786
1.510.0000	5.5234	170.3.1447	.7814	3.5648	70.3.31	8715	3.7600	56105.	1263.3250
1.520.0000	5.5385	170.7.7901	.7845	3.5659	70.2.583	8714	3.7600	56308.	1272.2775
1.530.0000	5.5535	171.2.4238	.7876	3.5669	70.1.740	8713	3.7600	56497.	1281.2316
1.540.0000	5.5685	171.7.0458	.7898	3.5679	70.0.904	8713	3.7600	56635.	1290.1871
1.555.0000	5.5834	172.1.6562	.7923	3.5689	70.0.173	8712	3.7600	56838.	1299.1440
1.566.0000	5.5983	172.6.2550	.7950	3.5699	69.9.447	8711	3.7600	56978.	1308.1024
1.576.0000	5.6132	173.0.8425	.7980	3.5709	69.8.127	8711	3.7600	57183.	1317.0622
1.586.0000	5.6281	173.5.4185	.8014	3.5719	69.7.613	8710	3.7600	57389.	1326.0234
1.596.0000	5.6429	173.9.9833	.8043	3.5729	69.6.304	8709	3.7600	57532.	1334.9859
1.606.0000	5.6576	174.4.5368	.8088	3.5739	69.6.704	8709	3.7600	57744.	1343.9490
1.616.0000	5.6724	174.9.0792	.8118	3.5749	69.5.203	8708	3.7600	57891.	1352.9149
1.626.0000	5.6871	175.3.6104	.8159	3.5755	69.4.410	8708	3.7600	58106.	1361.8813
1.636.0000	5.7017	175.8.1307	.8187	3.5761	69.3.322	8707	3.7600	58254.	1370.8490
1.646.0000	5.7163	176.2.6400	.8226	3.5761	69.2.940	8706	3.7600	58474.	1379.8178
1.656.0000	5.7309	176.7.1384	.8269	3.5759	69.2.562	8706	3.7600	58698.	1388.7879
1.666.0000	5.7455	177.1.6260	.8297	3.5762	69.1.290	8705	3.7600	58848.	1397.7592
1.676.0000	5.7600	177.6.1028	.8337	3.5760	69.0.323	8705	3.7600	59074.	1406.7315
1.686.0000	5.7745	178.0.5690	.8364	3.5744	68.9.760	8704	3.7600	59228.	1415.7051
1.696.0000	5.7889	178.5.0245	.8400	3.5722	68.9.002	8703	3.7600	59436.	1424.6797
1.706.0000	5.8033	178.9.4695	.8465	3.5709	68.8.249	8703	3.7600	59816.	1433.6554
1.716.0000	5.8177	179.3.9040	.8532	3.5700	68.7.504	8702	3.7600	60187.	1442.6321
1.726.0000	5.8321	179.8.3280	.8641	3.5681	68.6.758	8701	3.7600	60728.	1451.6099

MAXIMUM RIT	MAXIMUM RH	LOW CYCLE FATIGUE LIFE = 1,000,000 CYCLES	RH	RH	NST	ETATT	TPR	SIGMIN	TH3	R6T3
RT	U	RT	U	RHL	RT	RT	RT	RT	RT	RT
1510.0000	5.5082	1598.7722	8277	3.2997	70.4237	.8715	3.7600	55.971	1254.3786	3.4768
1510.0000	5.5234	1703.1447	8304	3.3007	70.3431	.8715	3.7600	56105	1263.3250	3.4817
1520.0000	5.5385	1707.7951	8341	3.3016	70.2583	.8714	3.7600	56301	1272.2775	3.4866
1530.0000	5.5535	1712.4238	8378	3.3019	70.1740	.8713	3.7600	56497	1281.2316	3.4915
1540.0000	5.5685	1717.0458	8405	3.3021	70.0934	.8713	3.7600	56635	1290.1871	3.4964
1550.0000	5.5834	1721.6562	8442	3.3018	70.0073	.8712	3.7600	56838	1299.1440	3.5013
1550.0000	5.5983	1726.2550	8468	3.3019	69.9237	.8711	3.7600	56978	1308.024	3.5061
1570.0000	5.6132	1730.8425	8506	3.3015	69.8427	.8711	3.7600	57183	1317.0622	3.5109
1580.0000	5.6281	1735.4135	8555	3.3010	69.7613	.8710	3.7600	57389	1326.0234	3.5158
1590.0000	5.6429	1739.9333	8589	3.3011	69.6804	.8709	3.7600	57532	1334.9859	3.5206
1600.0000	5.6576	1744.5368	8641	3.3005	69.6012	.8709	3.7600	57744	1343.9498	3.5253
1610.0000	5.6724	1749.0792	8675	3.3005	69.5213	.8708	3.7600	57891	1352.9149	3.5301
1620.0000	5.6871	1753.6134	8722	3.2999	69.4410	.8708	3.7600	58106	1361.8013	3.5349
1630.0000	5.7017	1758.1237	8756	3.2999	69.3622	.8707	3.7600	58254	1370.6490	3.5396
1640.0000	5.7163	1762.6400	8803	3.2992	69.2840	.8706	3.7600	58474	1379.8178	3.5444
1650.0000	5.7309	1767.1384	8850	3.2985	69.2152	.8705	3.7600	58698	1388.7879	3.5491
1660.0000	5.7455	1771.6260	8882	3.2984	69.1290	.8705	3.7600	58848	1397.7592	3.5538
1670.0000	5.7600	1776.1023	8928	3.2976	69.0523	.8704	3.7600	59074	1406.7315	3.5585
1680.0000	5.7745	1780.5693	8960	3.2974	68.9730	.8704	3.7600	59228	1415.7051	3.5632
1690.0000	5.7889	1785.0245	9001	3.2967	68.9062	.8703	3.7600	59436	1424.6779	3.5678
1700.0000	5.8033	1789.4695	9098	3.2901	68.8249	.8703	3.7600	59616	1433.6554	3.5725

1715-70000	5.8177	1793-9000	.9186	3.2877	68.7501	.8702	3.7600	60187.	1462.6321	3.5771
1720.0010	5.8321	1798-3280	.9308	3.2839	68.6758	.8704	3.7600	60728.	1451.6099	3.5817
1730.0010	5.8464	1802.7417	.9466	3.2785	68.6014	.8701	3.7600	61468.	1460.5887	3.5863
1740.0010	5.8607	1807.1450	.9658	3.2730	68.5285	.8703	3.7600	62241.	1469.5685	3.5909
1750.0010	5.8749	1811.5382	.9927	3.2641	68.4556	.8700	3.7600	63380.	1478.5493	3.5955
1760.0010	5.8891	1815.9211	1.0195	3.2562	68.3831	.8699	3.7600	64559.	1487.5310	3.6061
1770.0010	5.9033	1820.2939	1.0566	3.2443	68.3110	.8699	3.7600	66006.	1496.5137	3.6047
1780.0000	5.9175	1824.6566	1.0966	3.2329	68.2394	.8698	3.7600	67399.	1505.4972	3.6092
1790.0000	5.9316	1829.0094	1.1436	3.2192	68.1682	.8697	3.7600	68938.	1514.4817	3.6138
1800.0000	5.9457	1833.3521	1.2096	3.1988	68.0974	.8697	3.7600	70934.	1523.4670	3.6183
1810.0000	5.9597	1837.6850	1.2781	3.1777	68.0272	.8696	3.7600	72829.	1532.4532	3.6228
1820.0000	5.9737	1842.0081	1.3587	3.1520	67.9572	.8696	3.7600	74903.	1541.4403	3.6273
1830.0000	5.9877	1846.3214	1.4557	3.1186	67.8877	.8695	3.7600	77176.	1550.4281	3.6318
1840.0000	6.0017	1850.6250	1.5764	3.0741	67.8167	.8695	3.7600	79686.	1559.4168	3.6362
1850.0000	6.0156	1854.9189	1.7264	3.0143	67.7500	.8694	3.7600	82453.	1568.4063	3.6407
1860.0000	6.0295	1859.2032	1.9228	2.9244	67.6810	.8693	3.7600	85486.	1577.3965	3.6452
1870.0000	6.0434	1863.4779	2.2280	2.7478	67.6133	.8693	3.7600	88811.	1586.3875	3.6496
1880.0000	6.0572	1867.7432	0.0600	0.0000	67.5465	.8692	3.7600	92466.	1595.3792	3.6540

MAXIMUM RIT
RH AT MAXIMUM = 2.5112

COMPRESSOR EFFICIENCY = .8840
COMPRESSOR SPECIFIC SPEED = 90.0000

	COMPRESSOR PRESSURE RATIO			COMPRESSOR DELIVERY TEMPERATURE			COMPRESSOR MECHANICAL SPEED			COMPRESSOR TIP RADIUS		
PR	=	4.0000	F	=	342.4885	RPM	=	39751.498	IN	=	4.1001	IN
CTR	=	6.8.3811	BTU/LB	=	6.8.3811	BTU/LB	=	6.8.3811	BTU/LB	=	6.8.3811	BTU/LB
TPR	=	1.0000	IN	=	1.0000	TURBINE TIP-RADIUS	=	1.0000	TURBINE TIP-SPEED	=	1.0000	TURBINE TIP-SPEED
RIT	=	4.9005	U	=	4.9005	U	=	4.9005	U	=	4.9005	U
RIT.	=	9.141	FT/SEC	=	9.141	FT/SEC	=	9.141	FT/SEC	=	9.141	FT/SEC
RHL	=	4.9277	IN	=	4.9277	IN	=	4.9277	IN	=	4.9277	IN
RHU	=	4.9412	IN	=	4.9412	IN	=	4.9412	IN	=	4.9412	IN

TURBINE FATIGUE LIFE = 500000. CYCLES

	RT	RHL	RHU	NST	ETATT	TPR	SIGHIN	TM3	R6T
1540.0000	4.9005	1699.9769	.7913	3.0499	.79.155	.8748	3.7600	60.874	3.4759
1510.0000	4.9141	1704.6896	.7934	3.0510	.79.088	.8747	3.7600	61.023	3.4808
1520.0000	4.9277	1709.3905	.7966	3.0521	.78.9927	.8747	3.7600	61.256	3.4857
1530.0000	4.9412	1714.0795	.8002	3.0526	.78.8974	.8747	3.7600	61.495	3.4905
1540.0000	4.9547	1718.7568	.8031	3.0530	.78.8027	.8747	3.7600	61.646	3.4954
1550.0000	4.9681	1723.4224	.8076	3.0529	.78.7086	.8747	3.7600	61.888	3.5002
1560.0000	4.9815	1728.0764	.8105	3.0532	.78.6152	.8746	3.7600	62.044	3.5051
1570.0000	4.9949	1732.7189	.8149	3.0535	.78.5224	.8746	3.7600	62.293	3.5099
1580.0000	5.0083	1737.3499	.8178	3.0533	.78.4303	.8746	3.7600	6244.9	3.5147
1590.0000	5.0216	1741.9696	.8221	3.0535	.78.3381	.8746	3.7600	6270.1	3.5195
1600.0000	5.0349	1746.5779	.8249	3.0532	.78.2473	.8746	3.7600	6286.1	3.5242
1610.0000	5.0481	1751.1750	.8278	3.0535	.78.1575	.8746	3.7600	63025	3.5290
1620.0000	5.0613	1755.7609	.8320	3.0531	.78.0673	.8745	3.7600	6328.0	3.5337
1630.0000	5.0745	1760.3357	.8348	3.0533	.77.9787	.8745	3.7600	6344.8	3.5385
1640.0000	5.0877	1764.8995	.8394	3.0528	.77.8321	.8745	3.7600	6371.0	3.5432
1650.0000	5.1008	1769.4523	.8421	3.0531	.77.8021	.8745	3.7600	6387.7	3.5479
1660.0000	5.1139	1773.9942	.8461	3.0524	.77.7726	.8745	3.7600	6414.1	3.5526
1670.0000	5.1270	1778.5252	.8489	3.0525	.77.6279	.8745	3.7600	6431.2	3.5573
1680.0000	5.1400	1783.0455	.8539	3.0523	.77.5417	.8744	3.7600	6458.3	3.5619
1690.0000	5.1530	1787.5552	.8597	3.0479	.77.4500	.8744	3.7600	6437.9	3.5666
1700.0000	5.1660	1792.0541	.8669	3.0462	.77.3768	.8744	3.7600	6526.1	3.5712
1710.0000	5.1789	1796.5425	.8770	3.0434	.77.2362	.8744	3.7600	6581.9	3.5758
1720.0000	5.1913	1801.3234	.8876	3.0407	.77.1201	.8744	3.7600	6640.7	3.5805
1730.0000	5.2047	1805.4878	.9003	3.0369	.77.1185	.8744	3.7600	6717.9	3.5851
1740.0000	5.2175	1809.9449	.9196	3.0318	.77.0355	.8743	3.7600	6813.9	3.5896

LOW CYCLE FATIGUE LIFE = 100000 CYCLES		RHT	RHL	RHU	NST	ETATT	TPR	SIGHIN'	TH3	RHT
1750.000	5.2303	1814.3916	9419.	3.0254	76.9530	.8743	3.7600	69396.	1477.6788	3.5942
1760.000	5.2431	1818.8281	9662	3.0188	76.8709	.8743	3.7600	70591.	1486.6430	3.5988
1770.000	5.2559	1823.2543	9951	3.0119	76.7896	.8743	3.7600	72057.	1495.6081	3.6033
1780.000	5.2686	1827.6705	1.0383	2.9999	76.7586	.8743	3.7600	73949.	1504.5741	3.6079
1790.000	5.2813	1832.0765	1.0813	2.9873	76.6279	.8743	3.7600	75709.	1513.5410	3.6124
1800.000	5.2940	1836.4725	1.1331	2.9731	76.5479	.8742	3.7600	77655.	1522.5087	3.6169
1810.000	5.3066	1840.8586	1.1936	2.9562	76.4681	.8742	3.7600	79778.	1531.4773	3.6214
1820.000	5.3193	1845.2347	1.2665	2.9350	76.3894	.8742	3.7600	82107.	1540.4467	3.6259
1830.000	5.3318	1849.6019	1.3531	2.9069	76.3108	.8742	3.7600	84565.	1549.4170	3.6304
1840.000	5.3444	1853.9575	1.4603	2.8714	76.2327	.8742	3.7600	87491.	1558.3880	3.6348
1850.000	5.3569	1858.3043	1.5688	2.8352	76.1552	.8742	3.7600	89990.	1567.3598	3.6393
1860.000	5.3694	1862.6414	1.7300	2.7704	76.0779	.8742	3.7600	93299.	1576.3324	3.6437
1870.000	5.3819	1866.9688	1.9398	2.6701	76.0012	.8741	3.7600	96750.	1585.3057	3.6481
1880.000	5.3944	1871.2867	0.0000	0.0000	75.9249	.8741	3.7600	100486.	1594.2798	3.6525

MAXIMUM RHT = 1878.6995

RH AT MAXIMUM = 2.3616

1750.0000	5.2303	1.014	3916	1.0305	2.7549	76.9520	.8743	3.7600	69306.	1477.6788	3.5942
1760.0000	5.2431	1.818	.8281	1.0601	2.7432	76.8719	.8743	3.7600	70591.	1486.6430	3.5988
1770.0000	5.2559	1.823	.2543	1.0972	2.7291	76.7854	.8743	3.7600	72057.	1495.6081	3.6033
1780.0000	5.2686	1.827	.6705	1.1526	2.7583	76.7064	.8743	3.7600	73949.	1504.5741	3.6079
1790.0000	5.2813	1.832	.0765	1.2083	2.6867	76.6273	.8743	3.7600	75709.	1513.5410	3.6124
1800.0000	5.2940	1.836	.4725	1.2772	2.6601	76.5473	.8742	3.7600	77655.	1522.5087	3.6169
1810.0000	5.3066	1.840	.8586	1.3588	2.6263	76.4664	.8742	3.7600	79778.	1531.4773	3.6244
1820.0000	5.3193	1.845	.2347	1.4601	2.5817	76.3864	.8742	3.7600	82107.	1540.4467	3.6259
1830.0000	5.3318	1.849	.6010	1.5898	2.5184	76.3168	.8742	3.7600	84665.	1549.4170	3.6304
1840.0000	5.3446	1.853	.9575	1.7725	2.4176	76.2327	.8742	3.7600	87491.	1558.3880	3.6346
1850.0000	5.3569	1.858	.3043	2.0910	2.1520	76.1581	.8742	3.7600	89990.	1567.3598	3.6393
1860.0000	5.3694	1.862	.6414	0.0000	0.0000	76.0779	.8741	3.7600	93299.	1576.3324	3.6437

MAXIMUM RIT = 1850.0313
RM AT MAXIMUM = 2.1312

COMPRESSOR EFFICIENCY = 884.0
COMPRESSOR SPECIFIC SPEED = 101.0000

		COMPRESSOR PRESSURE RATIO		COMPRESSOR DELIVERY TEMPERATURE		COMPRESSOR MECHANICAL SPEED	
PR	=	4.0000	F				
T2	=	342.4885	RPM				
N	=	44168.332	IN				
CTR	=	3.6901	BTU/LB.				
H	=	68.3811	ADIABATIC WORK				
TPR	=		TURBINE PRESSURE RATIO				
RT	=		TURBINE TIP RADIUS				
U	=		TURBINE TIP SPEED				
RIT	=		F _a	ROTOR INLET TEMP AT STRESS CONDITION			
RHL	=		IN	LOWER BLADE ROOT RADIUS			
RHU	=		IN	UPPER BLADE ROOT RADIUS			

LONG CYCLE FATIGUE LIFE = 50000 CYCLES

		RHL	RHU	NST	EATAI	TPR	SIGHIN	TH3	R6T
1.500.0000	4.4008	1696.2507	.8059	2.6265	.88.0176	.8727	3.7600	6621.8	1255.0446
1.510.0010	4.4132	1701.0281	.8111	2.6261	.87.9089	.8728	3.7600	66492.	1263.9609
1.520.0020	4.4256	1705.7935	.8140	2.6264	.87.8009	.8728	3.7600	66657.	1272.8787
1.530.0030	4.4379	1710.5469	.8168	2.6267	.87.6937	.8729	3.7600	66819.	1281.7979
1.540.0040	4.4502	1715.2885	.8214	2.6262	.87.5872	.8729	3.7600	67098.	1290.7186
1.550.0050	4.4625	1720.0183	.8242	2.6264	.87.4815	.8730	3.7600	67265.	1299.6466
1.560.0060	4.4747	1724.7365	.8287	2.6258	.87.3765	.8730	3.7600	67553.	1308.5641
1.570.0070	4.4869	1729.4433	.8315	2.6260	.87.2722	.8731	3.7600	67725.	1317.4888
1.580.0080	4.4991	1734.1388	.8359	2.6253	.87.1686	.8731	3.7600	68015.	1326.4149
1.590.0090	4.5112	1738.8215	.8386	2.6254	.87.0658	.8732	3.7600	68186.	1335.3423
1.600.0010	4.5234	1743.4936	.8430	2.6246	.86.9636	.8732	3.7600	68491.	1344.2709
1.610.0020	4.5355	1748.1543	.8457	2.6247	.86.8621	.8733	3.7600	68667.	1353.2008
1.620.0030	4.5475	1752.8039	.8483	2.6248	.86.7617	.8733	3.7600	68838.	1362.1319
1.630.0040	4.5596	1757.4422	.8541	2.6239	.86.6617	.8734	3.7600	69148.	1371.0642
1.640.0050	4.5716	1762.0694	.8576	2.6239	.86.5617	.8734	3.7600	69324.	1379.9977
1.650.0060	4.5835	1766.6855	.8634	2.6229	.86.4620	.8735	3.7600	69642.	1388.9323
1.660.0070	4.5955	1771.2907	.8668	2.6228	.86.3647	.8735	3.7600	69823.	1397.8681
1.670.0080	4.6074	1775.8449	.8702	2.6227	.86.2671	.8736	3.7600	70006.	1406.8049
1.680.0090	4.6193	1780.4683	.8746	2.6222	.86.1702	.8736	3.7600	70250.	1415.7428
1.690.0010	4.6312	1785.5409	.8812	2.6205	.86.0733	.8737	3.7600	70639.	1424.6818
1.700.0020	4.6430	1789.6028	.8905	2.6143	.85.9783	.8737	3.7600	71214.	1433.6218
1.710.0030	4.6548	1794.1540	.8995	2.6110	.85.8832	.8738	3.7600	71792.	1442.5629
1.720.0040	4.6666	1798.6946	.9151	2.6061	.85.7888	.8738	3.7600	72574.	1451.5049
1.730.0050	4.6783	1803.2247	.9329	2.5999	.85.6943	.8739	3.7600	73555.	1460.4479
1.740.0060	4.6901	1807.7444	.9486	2.5943	.85.6017	.8739	3.7600	74466.	1469.3918

LOW CYCLE FATIGUE LIFE = 100000. CYCLES	RHT	RHL	NST	ETATT	TPR	SIGMIN	TM3	R&T
1750.0000	4.7013 1.812.2536	9779	2.5842	85.5090	.8739	3.7600	75897.	1478.3367
1760.0000	4.7134 1.616.7525	1.0057	2.5741	85.4169	.8740	3.7600	77466.	1487.2825
1770.0000	4.7250 1.821.2167	1.0407	2.5624	85.3238	.8740	3.7600	79082.	1496.2367
1780.0000	4.7366 1.825.6665	1.0835	2.5483	85.2353	.8740	3.7600	80889.	1505.1931
1790.0000	4.7481 1.830.1062	1.1345	2.5306	85.1453	.8741	3.7600	82871.	1514.1503
1800.0000	4.7596 1.834.5357	1.1960	2.5089	85.0559	.8741	3.7600	85055.	1523.1084
1810.0000	4.7713 1.838.9553	1.2692	2.4811	84.9670	.8741	3.7600	87446.	1532.0674
1820.0000	4.7826 1.843.3650	1.3399	2.4549	84.8786	.8741	3.7600	89496.	1541.0271
1830.0000	4.7939 1.847.7647	1.4473	2.4083	84.7906	.8741	3.7600	92307.	1549.9877
1840.0000	4.8053 1.852.1546	1.5904	2.3366	84.7036	.8741	3.7600	95400.	1558.9490
1850.0000	4.8166 1.856.5348	1.7765	2.2257	84.6168	.8742	3.7600	98341.	1567.9111
1860.0000	4.8280 1.850.9052	0.0000	0.0000	84.5336	.8742	3.7600	101834.	1576.8740
MAXIMU1 RHT RH AT MAXIMUH = 1854.7488 RH AT MAXIMUH = 2.0228								
1505.0000	4.4008 1.696.2507	8657	2.4056	86.0176	.8727	3.7600	66218.	1255.0446
1510.0000	4.4132 1.701.0281	8717	2.4034	87.9029	.8728	3.7600	66492.	1263.9609
1520.0000	4.4256 1.705.7935	8750	2.4032	87.8029	.8728	3.7600	66657.	1272.8787
1530.0000	4.4379 1.710.5469	8783	2.4029	87.6927	.8729	3.7600	66819.	1281.7979
1540.0000	4.4502 1.715.2885	8835	2.4016	87.5872	.8729	3.7600	67098.	1296.7186
1550.0000	4.4625 1.720.183	8867	2.4013	87.4815	.8730	3.7600	67265.	1299.6406
1560.0000	4.4747 1.724.7365	8919	2.3998	87.3725	.8730	3.7600	67553.	1308.5641
1570.0000	4.4869 1.729.4430	8951	2.3994	87.2722	.8731	3.7600	67725.	1317.4888
1580.0000	4.4991 1.734.1380	9003	2.3978	87.1666	.8731	3.7600	68015.	1326.5149
1590.0000	4.5112 1.738.8215	9043	2.3973	87.0658	.8732	3.7600	68186.	1335.3423
1600.0000	4.5234 1.743.4936	9111	2.3955	86.9636	.8732	3.7600	68491.	1344.2709
1610.0000	4.5355 1.748.1543	9151	2.3950	86.8624	.8733	3.7600	68667.	1353.2008
1620.0000	4.5475 1.752.8039	9190	2.3944	86.7513	.8733	3.7600	68838.	1362.1319
1630.0000	4.5596 1.757.4422	9262	2.3925	86.6611	.8734	3.7600	69148.	1371.0642
1640.0000	4.5716 1.762.0694	9300	2.3918	86.5617	.8734	3.7600	69324.	1379.9977
1650.0000	4.5835 1.766.6855	9365	2.3897	86.4628	.8735	3.7600	69642.	1388.9323
1660.0000	4.5955 1.771.2907	9402	2.3889	86.3647	.8735	3.7600	69823.	1397.8681
1670.0000	4.6074 1.775.8679	9440	2.3882	86.2671	.8736	3.7600	70006.	1406.8049
1680.0000	4.6193 1.780.4683	9488	2.3867	86.1702	.8736	3.7600	70250.	1415.7428
1690.0000	4.6312 1.785.0409	9579	2.3836	86.0739	.8737	3.7600	70639.	1424.6818
1700.0000	4.6433 1.785.6028	9710	2.3784	85.9783	.8737	3.7600	71214.	1433.6218
1710.0000	4.6548 1.794.1540	9837	2.3729	85.8732	.8738	3.7600	71792.	1442.5629
1720.0000	4.6666 1.798.6946	1.0008	2.3611	85.7888	.8738	3.7600	72574.	1451.5049
1730.0000	4.6783 1.803.2247	1.0250	2.3502	85.6549	.8739	3.7600	73555.	1460.4479
1740.0000	4.6901 1.807.7444	1.0461	2.3396	85.6017	.8739	3.7600	74466.	1459.3918
1750.0000	4.7018 1.812.2536	1.0837	2.3217	85.5029	.8739	3.7600	75897.	1478.3367
1760.0000	4.7134 1.816.7525	1.1221	2.3034	85.4159	.8740	3.7600	77466.	1487.2825

1770.0000	4.07250	1821.2167	1793.691	2.2806	85.3253	.8740	3.7600	79082.	1496.2367	3.6034
1780.0000	4.07366	1825.6665	1792.293	2.2523	85.2353	.8740	3.7600	80889.	1505.1931	3.6079
1790.0000	4.07481	1830.1062	1794.068	2.2148	85.1453	.8741	3.7600	82871.	1514.1503	3.6124
1800.0000	4.07596	1834.5357	1795.957	2.1632	85.0553	.8741	3.7600	85055.	1523.1084	3.6170
1810.0000	4.07710	1838.9553	1799.291	2.0835	84.9673	.8741	3.7600	87446.	1532.0674	3.6214
1820.0000	4.07825	1843.3650	1801.7051	1.9585	84.8786	.8741	3.7600	89496.	1541.0271	3.6259
1830.0000	4.07939	1847.7647	1803.0000	0.0000	84.7905	.8741	3.7600	92307.	1549.9877	3.6304

MAXIMUM RIT = 1821.7936

RH AT MAXIMUM = 10.8343

COMPRESSOR EFFICIENCY = .8638
 COMPRESSOR SPECIFIC SPEED = 83.0000

		COMPRESSOR PRESSURE RATIO			
		F	RPM	IN	BTU/LB
CTR	=	45.4579		IN	BTU/LB
H	=	84.0030			
TPR	=			IN	TURBINE PRESSURE RATIO
RT	=			IN	TURBINE TIP SPEED
U	=			FT/SEC	TURBINE TIP SPEED
RIT	=			FT	ROTOR INLET TEMP AT STRESS CONDITION
RHL	=			IN	LOWER BLADE ROOT RADIUS
RHU	=			IN	UPPER BLADE ROOT RADIUS

END CYCLE FATIGUE LIFE = 5000 CYCLES		NST	ETATT	TPR	SIGHMIN	TMJ	R6T
RTI	U	RHL	RHU				
1500.0000	5.1246 1812.1670	.8619	3.0525	71.5769	.8721	4.7000	65027.1218.9298
1510.0000	5.1388 1817.2907	.8679	3.0515	71.4902	.8720	4.7000	65342.1227.7122
1520.0000	5.1530 1822.2019	.8719	3.0513	71.4040	.8720	4.7000	65546.1236.4965
1530.0000	5.1671 1827.2006	.8782	3.0503	71.3185	.8719	4.7000	65869.1245.2826
1540.0000	5.1812 1832.1869	.8821	3.0500	71.2335	.8719	4.7000	66478.1254.0705
1550.0000	5.1953 1837.1610	.8878	3.0488	71.1491	.8719	4.7000	66493.1262.8601
1560.0000	5.2093 1842.1228	.8916	3.0484	71.0653	.8718	4.7000	66617.1271.6513
1570.0000	5.2233 1847.0724	.8973	3.0472	70.9821	.8718	4.7000	66950.1280.4443
1580.0000	5.2372 1852.0103	.9013	3.0468	70.8994	.8717	4.7000	67162.1289.2389
1590.0000	5.2512 1856.9356	.9060	3.0464	70.8173	.8717	4.7000	67376.1298.0351
1600.0000	5.2651 1861.8492	.9132	3.0449	70.7758	.8716	4.7000	67720.1306.8328
1610.0000	5.2789 1866.7513	.9177	3.0445	70.6547	.8716	4.7000	67931.1315.6322
1620.0000	5.2928 1871.6410	.9252	3.0429	70.5742	.8716	4.7000	68283.1324.4330
1630.0000	5.3065 1876.5193	.9296	3.0424	70.4943	.8715	4.7000	68499.1333.2354
1640.0000	5.3203 1881.3763	.9366	3.0407	70.4150	.8715	4.7000	68860.1342.0423
1650.0000	5.3339 1886.2055	.9409	3.0401	70.3356	.8714	4.7000	69079.1350.8560
1660.0000	5.3476 1891.0233	.9452	3.0395	70.2584	.8714	4.7000	69201.1359.6712
1670.0000	5.3612 1895.8296	.9524	3.0377	70.1308	.8713	4.7000	69663.1368.4877
1680.0000	5.3747 1900.6246	.9577	3.0373	70.0138	.8712	4.7000	69890.1377.3056
1690.0000	5.3882 1905.4053	.9660	3.0351	70.0272	.8712	4.7000	70261.1385.1249
1700.0000	5.4017 1910.1608	.9717	3.0343	69.9311	.8711	4.7000	70492.1394.9454
1710.0000	5.4152 1914.9422	.9767	3.0335	69.8756	.8710	4.7000	70717.1403.6735
1720.0000	5.4286 1919.6924	.9846	3.0314	69.8304	.8710	4.7000	71094.1412.6202
1730.0000	5.4420 1924.4317	.9908	3.0303	69.7258	.8709	4.7000	71387.1421.4447
1740.0000	5.4554 1929.1600	1.0036	3.0258	69.6516	.8709	4.7000	72000.1430.2705

LOW CYCLE FATIGUE LIFE = 100000 CYCLES										
RIT	RT	RHL	RHU	NST	ETATT	VPR	SIGHMIN	TH3	R6T	
1500.000	5.1246	1812.1670	.9311	2.7982	71.5769	*8721	4.7000	65027.	1218.9298	
1510.000	5.1388	1817.1907	.9377	2.7962	71.4912	*8720	4.7000	65342.	1227.7122	
1520.000	5.1520	1822.2019	.9421	2.7953	71.4041	*8720	4.7000	65546.	1236.4965	
1530.000	5.1671	1827.1996	.9491	2.7932	71.3135	*8719	4.7000	65869.	1245.2826	
1540.000	5.1827	1832.1869	.9543	2.7854	71.2335	*8719	4.7000	66078.	1254.075	
1550.000	5.1953	1837.1620	.9624	2.7960	71.1491	*8719	4.7000	66403.	1262.8601	
1560.000	5.2093	1842.1228	.9678	2.7889	71.0653	*8718	4.7000	66617.	1271.6513	
1570.000	5.2233	1847.0724	.9757	2.7866	70.9821	*8718	4.7000	66950.	1280.4443	
1580.000	5.2372	1852.0190	.9808	2.7854	70.8994	*8717	4.7000	67162.	1239.2389	
1590.000	5.2512	1856.9356	.9859	2.7842	70.8173	*8717	4.7000	67376.	1298.0351	
1600.000	5.2651	1861.8492	.9936	2.7816	70.7358	*8716	4.7000	67726.	1306.8328	
1610.000	5.2789	1866.7510	.9984	2.7804	70.6547	*8716	4.7000	67931.	1315.6322	
1620.000	5.2926	1871.6410	1.0081	2.7776	70.5742	*8716	4.7000	68283.	1324.4330	
1630.000	5.3066	1876.5140	2.7763	70.4943	*8715	4.7000	68499.	1333.0354	3.4945	
1640.000	5.3203	1881.3763	1.0233	2.7733	70.4150	*8715	4.7000	68660.	1342.0423	3.5029
1650.000	5.3339	1886.2055	1.0290	2.7718	70.3364	*8714	4.7000	69079.	1350.8560	3.5086
1660.000	5.3476	1891.0233	1.0347	2.7703	70.2584	*8714	4.7000	69321.	1359.6712	3.5133
1670.000	5.3612	1895.8296	1.0435	2.7672	70.1808	*8713	4.7000	69663.	1368.4877	3.5179
1680.000	5.3747	1900.6246	1.0491	2.7555	70.1038	*8712	4.7000	69890.	1377.3056	3.5226
1690.000	5.3882	1905.4683	1.0595	2.7622	70.0272	*8712	4.7000	70261.	1386.1249	3.5272
1700.000	5.4017	1910.1808	1.0658	2.7604	69.9511	*8711	4.7000	70492.	1394.9454	3.5319
1710.000	5.4152	1914.9422	1.0731	2.7587	69.8756	*8710	4.7000	70717.	1403.7673	3.5365
1720.000	5.4286	1919.6924	1.0833	2.7550	69.8004	*8710	4.7000	71094.	1412.6202	3.5411
1730.000	5.4420	1924.4317	1.0911	2.7524	69.7258	*8709	4.7000	71387.	1421.4447	3.5457

1750.0000	5.4554	1929.1633	1.1079	2.7455	69.6116	8709	4.7000	72000	1430.2705	3.5503
1750.0000	5.4688	1933.9774	1.1252	2.7385	69.5779	8708	4.7000	72595	1439.0975	3.5548
1760.0000	5.4821	1938.5840	1.1473	2.7288	69.5146	8707	4.7000	73369	1447.9257	3.5594
1770.0000	5.4953	1943.2798	1.1696	2.7194	69.4118	8707	4.7000	74095	1456.7540	3.5639
1780.0000	5.5086	1947.9648	1.2041	2.7033	69.3394	8706	4.7000	75248	1465.5854	3.5685
1790.0000	5.5218	1952.6393	1.2488	2.6819	69.2375	8706	4.7000	76623	1474.4169	3.5730
1800.0000	5.5350	1957.3031	1.2956	2.6589	69.2160	8705	4.7000	77994	1483.2495	3.5775
1810.0000	5.5482	1961.9565	1.3531	2.6240	69.1449	8705	4.7000	79559	1492.0831	3.5820
1820.0000	5.5613	1966.5993	1.4247	2.5857	69.0743	8704	4.7000	81336	1500.9177	3.5865
1830.0000	5.5744	1971.2317	1.5128	2.5342	69.0340	8704	4.7000	83353	1509.7534	3.5905
1840.0000	5.5875	1975.8538	1.6304	2.4606	68.9342	8703	4.7000	85703	1518.5901	3.5954
1850.0000	5.6005	1980.4656	1.8155	2.3362	68.8648	8702	4.7000	88359	1527.4277	3.5999
1860.0000	5.6135	1985.0671	0.0000	0.0000	68.7359	8702	4.7000	91257	1536.2663	3.6043

MAXIMUM RIJ = 1855.6625
RHT MAXIMUM = 2.0938

COMPRESSOR EFFICIENCY = 8610
 COMPRESSOR SPECIFIC SPEED = 90.0300

		COMPRESSOR PRESSURE RATIO			
		COMPRESSOR DELIVERY TEMPERATURE			
		COMPRESSOR MECHANICAL SPEED			
		H	T	IN	COMPRESSOR TIP RADIUS
		84.2771	BTU/LB	ADIABATIC WORK	
TPR	=			IN	TURBINE PRESSURE RATIO
RT	=			FT/SEC	TURBINE TIP SPEED
U	=			IN	ROTOR INLET TEMP AT STRESS CONDITION
RIT	=			F	
RHL	=			IN	LOWER BLADE ROOT RADIUS
RHU	=			IN	UPPER BLADE ROOT RADIUS

CYCLES	RT	U	RHL	RHU	NST	ETATT	TPR	SIGMIN	TM3	R6T
1500.0000	4.5585	1813.4867	.8878	2.5864	80.4695	.8750	4.7000	71434.	1218.5001	3.4357
1510.0000	4.5713	1816.5484	.8914	2.5860	80.3716	.8750	4.7000	71651.	1227.2697	3.4406
1520.0000	4.5839	1823.5911	.8976	2.5842	80.2744	.8750	4.7000	72037.	1236.0432	3.4454
1530.0000	4.5966	1828.6213	.9015	2.5837	80.1779	.8749	4.7000	72259.	1244.8184	3.4503
1540.0000	4.6092	1833.6391	.9063	2.5831	80.0822	.8749	4.7000	72484.	1253.5954	3.4551
1550.0000	4.6218	1838.6445	.9146	2.5812	79.9869	.8749	4.7000	72775.	1262.3741	3.4599
1560.0000	4.6343	1843.6377	.9190	2.5806	79.8923	.8749	4.7000	73096.	1271.1545	3.4647
1570.0000	4.6468	1848.6186	.9266	2.5785	79.7984	.8749	4.7000	73497.	1279.9366	3.4695
1580.0000	4.6593	1853.5875	.9311	2.5778	79.7652	.8749	4.7000	73732.	1288.7203	3.4743
1590.0000	4.6718	1858.5442	.9355	2.5770	79.6126	.8748	4.7000	73960.	1297.5055	3.4790
1600.0000	4.6842	1863.4891	.9528	2.5747	79.5265	.8748	4.7000	74374.	1306.2924	3.4838
1610.0000	4.6966	1868.4220	.9471	2.5739	79.4291	.8748	4.7000	74607.	1315.0868	3.4885
1620.0000	4.7090	1873.3430	.9516	2.5730	79.3363	.8748	4.7000	74643.	1323.8707	3.4932
1630.0000	4.7213	1878.2523	.9606	2.5706	79.2482	.8748	4.7000	75260.	1332.6621	3.4979
1640.0000	4.7336	1883.1499	.9659	2.5696	79.1385	.8747	4.7000	75500.	1341.4550	3.5026
1650.0000	4.7459	1888.0359	.9754	2.5670	79.0395	.8747	4.7000	75928.	1350.2493	3.5073
1660.0000	4.7582	1892.9104	.9805	2.5659	78.9311	.8747	4.7000	76173.	1359.0450	3.5120
1670.0000	4.7704	1897.7733	.9856	2.5647	78.8332	.8747	4.7000	76422.	1367.8421	3.5156
1680.0000	4.7826	1902.6248	.9941	2.5618	78.8359	.8747	4.7000	76862.	1376.6405	3.5212
1690.0000	4.7948	1907.4650	.9989	2.5607	78.7192	.8747	4.7000	77104.	1385.4403	3.5259
1700.0000	4.8069	1912.2939	1.0048	2.5594	78.6330	.8746	4.7000	77360.	1394.2414	3.5305
1710.0000	4.8190	1917.1145	1.0149	2.5563	78.5474	.8746	4.7000	77803.	1403.0437	3.5351
1720.0000	4.8311	1925.9130	1.0234	2.5538	78.4623	.8746	4.7000	78180.	1411.8773	3.5397
1730.0000	4.8431	1926.7134	1.0341	2.5504	78.3777	.8746	4.7000	78640.	1420.6823	3.5443
1740.0000	4.8552	1931.4978	1.0419	2.5478	78.2937	.8746	4.7000	79007.	1429.4886	3.5489

1750.0000	4.8672	1935.2712	1.0591	2.5408	78.2102	8746	4.7000	7978.0	1438.2960	3.5534
1760.0000	4.8791	1941.0337	1.0820	2.5314	78.1272	8745	4.7000	8075.8	1447.1045	3.5580
1770.0000	4.8911	1945.7853	1.1026	2.5227	78.0447	8745	4.7000	8163.8	1455.9142	3.5625
1780.0000	4.9030	1950.5261	1.1291	2.5117	77.9527	8745	4.7000	8268.5	1464.7250	3.5670
1790.0000	4.9149	1955.2563	1.1705	2.4926	77.8813	8745	4.7000	8431.8	1473.5369	3.5715
1800.0000	4.9268	1959.9757	1.2136	2.4682	77.8003	8745	4.7000	85895.6	1482.3498	3.5760
1810.0000	4.9385	1964.6846	1.2655	2.4424	77.7193	8745	4.7000	87699.	1491.1638	3.5805
1820.0000	4.9504	1969.3829	1.3138	2.4177	77.6393	8744	4.7000	89236.	1499.9768	3.5849
1830.0000	4.9622	1974.0707	1.3900	2.3766	77.5603	8744	4.7000	91473.	1508.7948	3.5894
1840.0000	4.9739	1978.7481	1.4889	2.3188	77.4813	8744	4.7000	94181.	1517.6117	3.5938
1850.0000	4.9857	1983.4151	1.6321	2.2288	77.4027	8744	4.7000	97057.	1526.4296	3.5983
1860.0000	4.9974	1988.0718	1.8663	2.0555	77.3243	8744	4.7000	99745.	1535.2485	3.6027
1870.0000	5.0091	1992.7183	0.0000	77.2470	8744	4.7000	103107.	1544.0683	3.6071	

MAXIMUM RIT = 1860.7585
RH AT MAXIMUM = 1.9614

LOW CYCLE FATIGUE LIFE = 100000 CYCLES

RIT	RT	U	RHL	RHU	NST	ETATT	TPR	SIGMIN	TH3	R6T
1500.0000	4.5585	1813.4867	0.9678	2.3497	80.4696	8750	4.7000	71434.	1218.5001	3.4357
1510.0000	4.5713	1818.5484	0.9730	2.3484	80.3716	8750	4.7000	71651.	1227.2697	3.4406
1520.0000	4.5839	1823.5941	0.9816	2.3451	80.2744	8750	4.7000	72037.	1236.0432	3.4454
1530.0000	4.5966	1828.6213	0.9867	2.3436	80.1779	8749	4.7000	72259.	1246.8184	3.4503
1540.0000	4.6092	1833.6391	0.9917	2.3420	80.0821	8749	4.7000	72484.	1253.5954	3.4551
1550.0000	4.6218	1838.6445	1.0008	2.3385	79.9869	8749	4.7000	72875.	1262.3741	3.4599
1560.0000	4.6343	1843.6377	1.0068	2.3369	79.8923	8749	4.7000	73096.	1271.1545	3.4647
1570.0000	4.6468	1848.6186	1.0170	2.3339	79.7984	8749	4.7000	73497.	1279.9366	3.4695
1580.0000	4.6593	1853.5875	1.0231	2.3312	79.7052	8749	4.7000	73732.	1288.703	3.4743
1590.0000	4.6718	1858.5475	1.0289	2.3294	79.6126	8748	4.7000	73960.	1297.5055	3.4790
1600.0000	4.6842	1863.4891	1.0388	2.3252	79.5205	8748	4.7000	74374.	1306.9224	
1610.0000	4.6966	1868.4220	1.0444	2.3232	79.4261	8748	4.7000	74607.	1315.0808	3.4835
1620.0000	4.7090	1873.3430	1.0500	2.3211	79.3363	8748	4.7000	74843.	1323.8707	3.4932
1630.0000	4.7213	1878.2523	1.0616	2.3165	79.2482	8748	4.7000	75260.	1332.5621	3.4979
1640.0000	4.7336	1883.1499	1.0684	2.3144	79.1555	8747	4.7000	75500.	1341.4550	3.5026
1650.0000	4.7459	1888.0359	1.0806	2.3096	79.0665	8747	4.7000	75928.	1350.2493	3.5073
1660.0000	4.7582	1892.9104	1.0872	2.3072	78.981	8747	4.7000	76173.	1359.0450	3.5120
1670.0000	4.7704	1897.7733	1.0937	2.3047	78.892	8747	4.7000	76422.	1367.8421	3.5167
1680.0000	4.7826	1902.6248	1.1057	2.2993	78.8059	8747	4.7000	76862.	1376.6405	3.5213
1690.0000	4.7948	1907.4650	1.1132	2.2967	78.7112	8747	4.7000	77104.	1385.4403	3.5259
1700.0000	4.8059	1912.2939	1.1209	2.2937	78.6310	8746	4.7000	77360.	1396.0414	3.5305
1710.0000	4.8190	1917.1215	1.1335	2.2878	78.5474	8746	4.7000	77803.	1403.0437	3.5351
1720.0000	4.8311	1921.9180	1.1442	2.2827	78.4623	8746	4.7000	78180.	1411.8773	3.5397
1730.0000	4.8431	1926.7134	1.1589	2.2761	78.3777	8746	4.7000	78640.	1420.6823	3.5443
1740.0000	4.8552	1931.4978	1.1706	2.2710	78.2937	8746	4.7000	79007.	1429.4986	3.5489
1750.0000	4.8672	1936.2712	1.1938	2.2586	78.2132	8746	4.7000	79780.	1438.2960	3.5534

1763.000	4.8795	1951.0337	1.2260	2.2414	78.1372	.8745	4.7000	80758.	1447.1045	3.5580
1770.000	4.8911	1945.7853	1.2561	2.2246	78.0447	.8745	4.7000	81638.	1455.9142	3.5625
1780.000	4.9033	1950.5261	1.2937	2.2032	77.9627	.8745	4.7000	82685.	1464.7250	3.5670
1790.000	4.9179	1955.2563	1.3576	2.1634	77.8813	.8745	4.7000	84318.	1473.5369	3.5715
1800.000	4.9268	1959.9757	1.4300	2.1172	77.8003	.8745	4.7000	85895.	1482.3498	3.5760
1810.000	4.9386	1964.6846	1.5204	2.0480	77.7198	.8745	4.7000	87699.	1491.1638	3.5805
1820.000	4.9504	1969.3829	1.6486	1.9543	77.6398	.8744	4.7000	89236.	1499.9788	3.5849
1830.000	4.9622	1974.0707	0.0000	0.0000	77.5603	.8744	4.7000	91473.	1508.7948	3.5894

MAXIMUM RH = 1823.4167
RH AT MAXIMUM = 1.8056

COMPRESSOR EFFICIENCY = .8544
COMPRESSOR SPECIFIC SPEED = 100.0000

PR	5.000	F	COMPRESSOR PRESSURE RATIO
T2	410.2277	RPM	COMPRESSOR DELIVERY TEMPERATURE
N	50651.449		COMPRESSOR MECHANICAL SPEED
CTR	3.5859	IN	COMPRESSOR TIP RADIUS
H	84.9236	BTU/LB	ADIABATIC WORK
TPR		IN	TURBINE PRESSURE RATIO
RT		FT/SEC	TURBINE TIP SPEED
U		F	ROTOR INLET TEMP AT STRESS CONDITION
RTT		IN	LOWER BLADE ROOT RADIUS
RHL		IN	UPPER BLADE ROOT RADIUS
RHU		IN	

L0W CYCLE FATIGUE LIFE = 50000 CYCLES

RIT	RT	U	RHL	RHU	NST	ETATT	TPR	SIGNIN	TH3	R6T
1.500.0000	4.0917	1808.5966	.9164	2.1994	.89.4666	.8729	4.7000	7.8242.	1.220.0487	3.4367
1.510.0000	4.1033	1813.7189	.9209	2.1986	.89.3562	.8721	4.7000	7.8476.	1.228.8012	3.4416
1.520.0000	4.1148	1818.8284	.9291	2.1957	.89.2466	.8722	4.7000	7.8829.	1.237.5555	3.4464
1.530.0000	4.1264	1823.9254	.9335	2.1948	.89.1378	.8722	4.7000	7.9166.	1.246.3116	3.4512
1.540.0000	4.1379	1829.0099	.9379	2.1937	.89.0297	.8723	4.7000	7.9410.	1.255.0693	3.4560
1.550.0000	4.1493	1834.0819	.9464	2.1907	.88.9224	.8723	4.7000	7.9869.	1.263.8287	3.4608
1.560.0000	4.1608	1839.1416	.9506	2.1897	.88.8158	.8724	4.7000	8.0104.	1.272.5897	3.4656
1.570.0000	4.1722	1844.1839	.9562	2.1885	.88.7100	.8724	4.7000	8.0355.	1.281.3523	3.4703
1.580.0000	4.1836	1849.2241	.9660	2.1851	.88.6045	.8725	4.7000	8.0828.	1.290.1166	3.4751
1.590.0000	4.1950	1854.2471	.9711	2.1835	.88.5005	.8725	4.7000	8.1072.	1.298.8323	3.4798
1.600.0000	4.2063	1859.2582	.9764	2.1824	.88.3961	.8726	4.7000	8.1331.	1.307.5496	3.4845
1.610.0000	4.2176	1864.2570	.9857	2.1786	.88.2931	.8726	4.7000	8.1808.	1.316.4184	3.4892
1.620.0000	4.2269	1869.2449	.9909	2.1772	.88.1915	.8727	4.7000	8.2072.	1.325.1807	3.4939
1.630.0000	4.2401	1874.2192	.9958	2.1757	.88.0898	.8727	4.7000	8.2327.	1.333.9604	3.4986
1.640.0000	4.2514	1879.1826	1.0061	2.1717	.87.9881	.8728	4.7000	8.2820.	1.342.7335	3.5033
1.650.0000	4.2626	1884.1342	1.0131	2.1701	.87.8886	.8728	4.7000	8.3081.	1.351.5080	3.5079
1.660.0000	4.2738	1889.0743	1.0192	2.1683	.87.7881	.8729	4.7000	8.3344.	1.360.2839	3.5126
1.670.0000	4.2849	1894.0028	1.0301	2.1639	.87.6901	.8729	4.7000	8.3852.	1.369.0611	3.5172
1.680.0000	4.2960	1898.9197	1.0360	2.1620	.87.5916	.8730	4.7000	8.4120.	1.377.8396	3.5218
1.690.0000	4.3071	1903.8252	1.0418	2.1603	.87.4931	.8730	4.7000	8.4391.	1.386.6194	3.5264
1.700.0000	4.3182	1908.7194	1.0529	2.1552	.87.3961	.8731	4.7000	8.4916.	1.395.4005	3.5310
1.710.0000	4.3292	1913.6022	1.0600	2.1530	.87.3001	.8731	4.7000	8.5192.	1.404.1828	3.5358
1.720.0000	4.3403	1918.4738	1.0682	2.1502	.87.2043	.8732	4.7000	8.5523.	1.412.9663	3.5402
1.730.0000	4.3513	1923.3542	1.0830	2.1432	.87.1094	.8732	4.7000	8.6162.	1.421.7809	3.5447
1.740.0000	4.3622	1928.1836	1.0953	2.1378	.87.0147	.8732	4.7000	8.6665.	1.430.5670	3.5493

1750-0000 483732 1935-0218 201202 562202 07338 40700 87653 1439-6912 55558-2

1760.0000	4.3841	1937.8491	1.1417	2.1144	86.8273	8733	4.7000	88530.	1448.71425	3.5583
1770.0000	4.3950	1942.6654	1.1698	2.1001	86.7344	8734	4.7000	89579.	1456.9320	3.5628
1780.0000	4.4059	1947.4709	1.2045	2.0806	86.6422	8734	4.7000	90819.	1465.7225	3.5673
1790.0000	4.4167	1952.2655	1.2481	2.0555	86.5505	8735	4.7000	92285.	1474.5140	3.5718
1800.0000	4.4275	1957.0494	1.3101	2.0182	86.4593	8735	4.7000	94099.	1483.3066	3.5763
1810.0000	4.4383	1961.8227	1.3894	1.9658	86.3687	8736	4.7000	96148.	1492.1001	3.5807
1820.0000	4.4491	1966.5853	1.5101	1.9739	86.2787	8736	4.7000	98458.	1500.8947	3.5852
1830.0000	4.4599	1971.3373	0.0000	0.0000	86.1892	8737	4.7000	101094.	1509.6902	3.5896

MAXIMUM RIT = 1825.8496
RH AT MAXIMUM = 1.7033

TOW CYCLE FATIGUE LIFE = 100000 CYCLES										
RIT	RT	U	RHL	NST	ETATT	TPR	SIGNIN	TM3	RT	R6T
1500.0000	4.0917	1808.5966	1.0145	1.9666	89.4666	8720	4.7000	78242.	1220.0487	3.4367
1510.0000	4.1033	1813.7189	1.0206	1.9644	89.3562	8721	4.7000	78476.	1228.8012	3.4416
1520.0000	4.1148	1818.8284	1.0318	1.9591	89.2466	8722	4.7000	78929.	1237.5555	3.4464
1530.0000	4.1264	1823.9254	1.0378	1.9567	89.1378	8722	4.7000	79168.	1246.3116	3.4512
1540.0000	4.1379	1829.0099	1.0437	1.9541	89.0297	8723	4.7000	79410.	1255.0693	3.4560
1550.0000	4.1493	1834.0819	1.0567	1.9482	88.9224	8723	4.7000	79869.	1263.8287	3.4608
1560.0000	4.1608	1839.1416	1.0637	1.9455	88.8158	8724	4.7000	80104.	1272.5897	3.4656
1570.0000	4.1722	1844.1889	1.0709	1.9424	88.7100	8724	4.7000	80355.	1281.3523	3.4703
1580.0000	4.1836	1849.2241	1.0838	1.9355	88.6049	8725	4.7000	80828.	1290.0146	3.4751
1590.0000	4.1950	1854.2471	1.0954	1.9323	88.5005	8725	4.7000	81072.	1298.8823	3.4798
1600.0000	4.2063	1859.2580	1.0973	1.9288	88.3958	8726	4.7000	81331.	1307.6496	3.4845
1610.0000	4.2176	1864.2570	1.1159	1.9213	88.2938	8726	4.7000	81808.	1316.4164	3.4892
1620.0000	4.2289	1869.2440	1.1203	1.9175	88.1915	8727	4.7000	82072.	1325.1887	3.4939
1630.0000	4.2401	1874.2192	1.1283	1.9137	88.0898	8727	4.7000	82327.	1333.9604	3.4986
1640.0000	4.2514	1879.1826	1.1428	1.9053	87.9889	8728	4.7000	82820.	1342.7335	3.5033
1650.0000	4.2626	1884.1342	1.1521	1.9012	87.8886	8728	4.7000	83081.	1351.5080	3.5079
1660.0000	4.2738	1889.0743	1.1617	1.8967	87.7889	8729	4.7000	83344.	1360.2839	3.5126
1670.0000	4.2849	1894.0028	1.1790	1.8863	87.6900	8729	4.7000	83852.	1369.0611	3.5172
1680.0000	4.2960	1899.9197	1.1881	1.8812	87.5916	8730	4.7000	84120.	1377.8396	3.5218
1690.0000	4.3071	1903.8282	1.1971	1.8758	87.4939	8730	4.7000	84391.	1386.6194	3.5264
1700.0000	4.3182	1908.7294	1.2171	1.8641	87.3969	8731	4.7000	84916.	1395.4005	3.5310
1710.0000	4.3292	1913.6022	1.2279	1.8582	87.3004	8731	4.7000	85192.	1404.1828	3.5356
1720.0000	4.3403	1918.4738	1.2404	1.8507	87.2045	8732	4.7000	85523.	1412.9663	3.5402
1730.0000	4.3513	1923.3342	1.2666	1.8328	87.1094	8732	4.7000	86162.	1421.7809	3.5447
1740.0000	4.3622	1928.1836	1.2901	1.8181	87.0147	8732	4.7000	86665.	1430.5670	3.5493
1750.0000	4.3732	1933.0218	1.3083	1.7830	86.9207	8733	4.7000	87653.	1439.3542	3.5538
1760.0000	4.3841	1937.8491	1.3886	1.7453	86.8273	8733	4.7000	88530.	1448.1425	3.5583
1770.0000	4.3950	1942.6654	1.4741	1.6730	86.7344	8734	4.7000	89579.	1456.9320	3.5628
1780.0000	4.4059	1947.4709	0.0000	0.0000	86.6422	8734	4.7000	90819.	1465.7225	3.5673

MAXIMUM RIT = 1773.3933

REATTACHMENT = 1.05754

COMPRESSOR EFFICIENCY = .8319
 COMPRESSOR SPECIFIC SPEED = 80.0000

PR	6.0000	F	COMPRESSOR PRESSURE RATIO
T2	470.7340	F	COMPRESSOR DELIVERY TEMPERATURE
N	44824.529	RPM.	COMPRESSOR MECHANICAL SPEED
CTR	4.3925	IN	COMPRESSOR TIP RADIUS
H	99.7897	BTU/LB	ADIABATIC WORK
TPR			TURBINE PRESSURE RATIO
RT		IN	TURBINE TIP RADIUS
U		FT/SEC.	TURBINE TIP SPEED
RIT		F	ROTOR INLET TEMP AT STRESS CONDITION
RHC		IN	LOWER BLADE ROOT RADIUS
RHU		IN	UPPER BLADE ROOT RADIUS

TOW CYCLE FATIGUE LIFE = 50000 CYCLES

	RHL	RHU	NST	EATL	TPR	SIGNIN	TH3	R6T
1500.0000	4.6458 1895.5358	.9562	2.6920	.72.5815	5.6400	73.694.	1191.0937	3.4042
1510.0000	4.6593 1900.8136	.9656	2.6894	.72.4939	5.6400	74.131.	1199.7475	3.4090
1520.0000	4.6728 1906.0785	.9713	2.6883	.72.4065	5.6400	74.365.	1208.4033	3.4139
1530.0000	4.6862 1911.3304	.9776	2.6870	.72.3205	5.6400	74.490.	1217.0612	3.4187
1540.0000	4.6995 1916.5695	.9867	2.6842	.72.2347	5.6400	75.935.	1225.7211	3.4235
1550.0000	4.9130 1921.7959	.9922	2.6829	.72.1495	5.6400	75.203.	1234.3829	3.4283
1560.0000	4.9263 1927.4095	.9977	2.6799	.72.0648	5.6400	75.662.	1243.0467	3.4331
1570.0000	4.9396 1932.2106	1.0063	2.6784	.71.9806	5.6722	5.6400	75.935.	1251.7123
1580.0000	4.9529 1937.3991	1.0129	2.6773	.71.8973	5.6722	5.6400	76.212.	1260.3798
1590.0000	4.9661 1942.5752	1.0233	2.6737	.71.8143	5.6722	5.6400	76.673.	1269.0839
1600.0000	4.9793 1947.7388	1.0297	2.6721	.71.7314	5.6721	5.6400	76.955.	1277.7555
1610.0000	4.9925 1952.8992	1.0360	2.6704	.71.6501	5.6721	5.6400	77.241.	1286.4288
1620.0000	5.0056 1958.0293	1.0467	2.6669	.71.5683	5.6720	5.6400	77.716.	1295.51039
1630.0000	5.0187 1963.1563	1.0533	2.6652	.71.4880	5.6720	5.6400	77.995.	1303.7807
1640.0000	5.0318 1968.2711	1.0656	2.6613	.71.4078	5.6720	5.6400	78.493.	1312.4592
1650.0000	5.0448 1973.3740	1.0726	2.6594	.71.3281	5.6719	5.6400	78.777.	1321.1392
1660.0000	5.0578 1978.4648	1.0796	2.6575	.71.2489	5.6719	5.6400	79.064.	1329.8209
1670.0000	5.0708 1983.5438	1.0912	2.6533	.71.1702	5.6718	5.6400	79.565.	1338.5042
1680.0000	5.0838 1988.6109	1.0982	2.6511	.71.0920	5.6718	5.6400	79.869.	1347.1891
1690.0000	5.0967 1993.6663	1.1060	2.6489	.71.0144	5.6718	5.6400	80.164.	1355.8754
1700.0000	5.1096 1998.7099	1.1202	2.6442	.70.9372	5.6717	5.6400	80.679.	1364.5633
1710.0000	5.1225 2003.7420	1.1277	2.6419	.70.8605	5.6717	5.6400	80.967.	1373.2526
1720.0000	5.1353 2008.7624	1.1355	2.6394	.70.7843	5.6716	5.6400	81.271.	1381.9434
1730.0000	5.1481 2013.7714	1.1484	2.6342	.70.7085	5.6716	5.6400	81.600.	1390.6356
1740.0000	5.1609 2018.7689	1.1571	2.6315	.70.6333	5.6716	5.6400	82.008.	1399.3292

1750.000	5.1736	2023.7551	1.1657	2.6288	70.5585	* 8715	5.6400	82411.	1408.0241	3.5215
1760.000	5.1863	2028.7245	1.1798	2.6235	70.4042	* 8715	5.6400	82928.	1416.7223	3.5260
1770.000	5.1990	2033.6620	1.1981	2.6160	70.4107	* 8714	5.6400	83605.	1425.4290	3.5306
1780.000	5.2115	2038.5864	1.2229	2.6059	70.3377	* 8714	5.6400	84456.	1434.1370	3.5351
1790.000	5.2241	2043.5037	1.2428	2.5973	70.2551	* 8713	5.6400	85166.	1442.8462	3.5396
1800.000	5.2366	2048.4680	1.2708	2.5852	70.1829	* 8713	5.6400	86067.	1451.5568	3.5441
1810.000	5.2492	2053.3013	1.3158	2.5638	70.1212	* 8712	5.6400	87534.	1460.2685	3.5486
1820.000	5.2616	2058.1836	1.3573	2.5435	70.0499	* 8711	5.6400	88818.	1468.9844	3.5530
1830.000	5.2741	2063.0551	1.4119	2.5161	69.9791	* 8711	5.6400	90360.	1477.6956	3.5575
1840.000	5.2865	2067.9158	1.4623	2.4786	69.9086	* 8710	5.6400	92197.	1486.4108	3.5619
1850.000	5.2989	2072.7656	1.5686	2.4285	69.8386	* 8710	5.6400	94241.	1495.1272	3.5664
1860.000	5.3113	2077.6050	1.6865	2.3546	69.7690	* 8709	5.6400	96560.	1503.8447	3.5708
1870.000	5.3236	2082.4336	1.8204	2.2602	69.6998	* 8709	5.6400	98523.	1512.5633	3.5752
1880.000	5.3359	2087.2517	0.0000	0.0000	69.6310	* 8708	5.6400	101400.	1521.2830	3.5796

MAXIMUM RIT = 2.0751338
RH AT MAXIMUM = 2.0526

LOW CYCLE FATIGUE LIFE = 1000000 CYCLES

RIT	RT	U	RHL	RHU	NST	ETATT	TPR	SIGMIN	TM3	R6T
1500.000	4.8458	1895.5358	1.0534	2.4355	72.5815	* 8725	5.6400	73694.	2191.0937	3.4042
1510.000	4.8593	1900.8136	1.0655	2.4309	72.4939	* 8725	5.6400	74131.	2199.7475	3.4090
1520.000	4.8728	1906.0785	1.0728	2.4285	72.4069	* 8724	5.6400	74395.	2120.8.4033	3.4139
1530.000	4.8862	1911.3304	1.0810	2.4259	72.3205	* 8724	5.6400	74490.	21217.0612	3.4187
1540.000	4.8996	1916.5695	1.0926	2.4208	72.2347	* 8724	5.6400	74935.	21225.7211	3.4235
1550.000	4.9130	1921.7959	1.0996	2.4181	72.1495	* 8723	5.6400	75203.	21234.3829	3.4283
1560.000	4.9263	1927.0095	1.1107	2.4128	72.0648	* 8723	5.6400	75662.	21243.0467	3.4331
1570.000	4.9396	1932.2166	1.1190	2.4098	71.9808	* 8722	5.6400	75935.	21251.7123	3.4378
1580.000	4.9529	1937.3991	1.1271	2.4068	71.9273	* 8722	5.6400	76212.	21260.3798	3.4426
1590.000	4.9661	1942.5752	1.1401	2.4010	71.8143	* 8722	5.6400	76673.	21269.0339	3.4473
1600.000	4.9793	1947.7388	1.1479	2.3977	71.7319	* 8721	5.6400	76955.	21277.7555	3.4521
1610.000	4.9925	1952.8902	1.1570	2.3941	71.6501	* 8721	5.6400	77241.	21286.4288	3.4568
1620.000	5.0056	1958.0293	1.1729	2.3874	71.5688	* 8720	5.6400	77716.	21295.1039	3.4615
1630.000	5.0187	1963.1563	1.1816	2.3838	71.4880	* 8720	5.6400	77995.	21303.7807	3.4662
1640.000	5.0318	1968.2711	1.1965	2.3764	71.4078	* 8720	5.6400	78493.	21312.4592	3.4709
1650.000	5.0448	1973.3740	1.2059	2.3725	71.3281	* 8719	5.6400	78777.	21321.1392	3.4755
1660.000	5.0578	1978.4648	1.2159	2.3684	71.2489	* 8719	5.6400	79064.	21329.8209	3.4802
1670.000	5.0708	1983.5438	1.2326	2.3605	71.1702	* 8718	5.6400	79565.	21338.5042	3.4848
1680.000	5.0838	1988.6109	1.2425	2.3559	71.1920	* 8718	5.6400	79869.	21347.1891	3.4894
1690.000	5.0967	1993.6663	1.2524	2.3533	71.1144	* 8718	5.6400	80164.	21355.6754	3.4941
1700.000	5.1096	1998.7099	1.2724	2.3420	70.9372	* 8717	5.6400	80679.	21364.5633	3.4987
1710.000	5.1225	2003.7420	1.2828	2.3371	70.8605	* 8717	5.6400	80967.	21373.2526	3.5033
1720.000	5.1353	2008.7624	1.2955	2.3317	70.7843	* 8716	5.6400	81271.	21381.9434	3.5076
1730.000	5.1481	2013.7714	1.3135	2.3213	70.7085	* 8716	5.6400	81800.	21390.6356	3.5124
1740.000	5.1609	2018.7689	1.3257	2.3154	70.6333	* 8716	5.6400	82108.	21399.3292	3.5170

1750.0000	5.1736	2023.7551	1.3375	2.3095	70.5585	.8715	5.6400	82411.	1408.0241	.305215
1760.0000	5.1863	2028.7245	1.3578	2.2984	70.4462	.8715	5.6400	82920.	1416.7223	.305260
1770.0000	5.1990	2033.6620	1.3871	2.2820	70.4107	.8714	5.6400	83605.	1425.4290	.305306
1780.0000	5.2115	2038.5884	1.4245	2.2599	70.3377	.8714	5.6400	84456.	1434.1370	.305351
1790.0000	5.2241	2043.5037	1.4568	2.2397	70.2351	.8713	5.6400	85166.	1442.8462	.305396
1800.0000	5.2366	2048.4980	1.5031	2.2106	70.1329	.8713	5.6400	86067.	1451.5568	.305441
1810.0000	5.2492	2053.3513	1.5885	2.1522	70.1212	.8712	5.6400	87534.	1460.2685	.305486
1820.0000	5.2616	2058.1836	1.6848	2.0786	70.0493	.8711	5.6400	88818.	1468.9814	.305530
1830.0000	5.2741	2063.0551	0.0000	0.0000	69.9791	.8711	5.6400	90360.	1477.6956	.305575

MAXIMUM RIT = 1827.6656
RH AT MAXIMUM = 1.08919

COMPRESSOR EFFICIENCY = $\frac{1}{\text{COMPRESSOR SPECIFIC SPEED}} \cdot 8215$

PR	=	6.0000	F	COMPRESSOR PRESSURE RATIO	
T2	=	475.8018	RPM	COMPRESSOR DELIVERY TEMPERATURE	
N	=	50425.436	IN	COMPRESSOR MECHANICAL SPEED	
CTR	=	3.9289	BTU/LB	COMPRESSOR TIP RADIUS	
H	=	1.01.0399	IN	ADIABATIC WORK	
TPR	=	FT/SEC	IN	TURBINE PRESSURE RATIO	
RT	=	F	FT/SEC	TURBINE TIP SPEED	
U	=	RT	IN	ROTOR INLET TEMP AT STRESS CONDITION	
RIT	=	U	IN	LOWER BLADE ROOT RADIUS	
RAL	=	RAL	IN	UPPER BLADE ROOT RADIUS	
RHU	=	RHU	IN		
LOW CYCLE FATIGUE LIFE = 50000 CYCLES					
RIT	RT	U	RHE	NST	ETATT
1.500.0000	4.3090	1896.1744	.9965	2.2538	81.603
1.510.0000	4.3212	1901.5266	1.0027	2.2519	81.5106
1.520.0000	4.3333	1906.8657	1.0146	2.2478	81.4115
1.530.0000	4.3454	1912.1916	1.0211	2.2458	81.3132
1.540.0000	4.3575	1917.5050	1.0277	2.2438	81.2155
1.550.0000	4.3696	1922.8052	1.0390	2.2390	81.1085
1.560.0000	4.3816	1928.0927	1.0444	2.2368	81.0122
1.570.0000	4.3936	1933.3674	1.0508	2.2345	80.9165
1.580.0000	4.4055	1938.6295	1.0643	2.2292	80.8215
1.590.0000	4.4175	1943.8791	1.0718	2.2267	80.7271
1.600.0000	4.4294	1949.1161	1.0793	2.2241	80.6333
1.610.0000	4.4412	1954.3418	1.0919	2.2185	80.5402
1.620.0000	4.4530	1959.5224	1.0990	2.2156	80.4482
1.630.0000	4.4647	1964.6835	1.1071	2.2129	80.3571
1.640.0000	4.4764	1969.8301	1.1233	2.2085	80.2565
1.650.0000	4.4881	1974.9647	1.1313	2.2035	80.1766
1.660.0000	4.4997	1980.0873	1.1395	2.2002	80.0872
1.670.0000	4.5114	1985.1979	1.1544	2.1929	79.9984
1.680.0000	4.5229	1990.2967	1.1636	2.1893	79.9102
1.690.0000	4.5345	1995.3836	1.1727	2.1856	79.8226
1.700.0000	4.5460	2000.4588	1.1895	2.1771	79.7355
1.710.0000	4.5575	2005.5224	1.1982	2.1731	79.6489
1.720.0000	4.5690	2010.5743	1.2083	2.1689	79.5629
1.730.0000	4.5805	2015.6147	1.2284	2.1598	79.4775
1.740.0000	4.5919	2020.6436	1.2383	2.1552	79.3926

1750.0000	4.6033	2025.6613	1.2460	2.1517	79.3082	.8748	5.6400	90.899.	1407.3175	3.5200
1760.0000	4.6147	2030.6671	1.2692	2.1395	79.2243	.8747	5.6400	91.615.	1416.0018	3.5245
1770.0000	4.6260	2035.6620	1.2066	2.1300	79.1410	.8747	5.6400	92.156.	1424.6874	3.5290
1780.0000	4.6374	2040.6455	1.3230	2.1093	79.0582	.8747	5.6400	93.227.	1433.3742	3.5335
1790.0000	4.6487	2045.6179	1.3551	2.0907	78.9758	.8747	5.6400	94.099.	1442.0623	3.5380
1800.0000	4.6599	2050.5792	1.3966	2.0646	78.8943	.8747	5.6400	95.171.	1450.7516	3.5425
1810.0000	4.6712	2055.5295	1.4519	2.0270	78.8127	.8747	5.6400	96.448.	1459.4422	3.5469
1820.0000	4.6824	2060.4687	1.5320	1.9730	78.7319	.8746	5.6400	97.924.	1468.1339	3.5514
1830.0000	4.6936	2065.3970	1.6763	1.8570	78.6515	.8746	5.6400	99.699.	1476.8268	3.5558
1840.0000	4.7048	2070.3144	0.0000	0.0000	78.5716	.8746	5.6400	101769.	1485.5208	3.5603

MAXIMUM RIT = 1831.4545
RH AT MAXIMUM = 17.7669

LOW CYCLE FATIGUE LIFE = 100000. CYCLES

RIT	RT	U	RHL	RHU	NET	ETATT	TPR	SIGHIN	TH3	R6T
1500.0000	4.3090	1896.1744	1.1246	1.9941	.81.6003	.8748	5.6400	81264.	1190.8435	3.4031
1510.0000	4.3212	1901.5266	1.1335	1.9897	.81.5006	.8748	5.6400	81562.	1199.4716	3.4079
1520.0000	4.3333	1906.8657	1.1484	1.9809	.81.4015	.8748	5.6400	82087.	1208.1017	3.4127
1530.0000	4.3454	1912.1918	1.1503	1.9764	.81.3032	.8748	5.6400	82194.	1216.7338	3.4175
1540.0000	4.3575	1917.5050	1.1685	1.9716	.81.2055	.8749	5.6400	82485.	1225.3678	3.4223
1550.0000	4.3696	1922.8552	1.1826	1.9618	.81.1085	.8749	5.6400	83023.	1234.0038	3.4271
1560.0000	4.3816	1928.0927	1.1935	1.9566	.81.0122	.8749	5.6400	83320.	1242.6416	3.4318
1570.0000	4.3936	1933.3674	1.2037	1.9512	.80.9165	.8749	5.6400	83620.	1251.2813	3.4366
1580.0000	4.4055	1938.6295	1.2244	1.9391	.80.8215	.8749	5.6400	84174.	1259.9229	3.4413
1590.0000	4.4175	1943.8791	1.2357	1.9326	.80.7271	.8750	5.6400	84480.	1268.6011	3.4460
1600.0000	4.4294	1949.1161	1.2467	1.9258	.80.6333	.8750	5.6400	84789.	1277.2467	3.4507
1610.0000	4.4412	1954.3408	1.2596	1.9122	.80.5402	.8750	5.6400	85347.	1285.8940	3.4554
1620.0000	4.4530	1959.5247	1.2826	1.9047	.80.4432	.8750	5.6400	85660.	1294.5527	3.4601
1630.0000	4.4647	1964.6835	1.2948	1.8969	.80.3571	.8750	5.6400	85961.	1303.2146	3.4648
1640.0000	4.4764	1969.8301	1.3237	1.8788	.80.2635	.8750	5.6400	86550.	1311.8341	3.4694
1650.0000	4.4881	1974.9647	1.3381	1.8694	.80.1756	.8749	5.6400	86856.	1320.5523	3.4741
1660.0000	4.4997	1980.0873	1.3536	1.8589	.80.0872	.8749	5.6400	87181.	1329.2221	3.4767
1670.0000	4.5114	1985.1979	1.3861	1.8362	.79.9934	.8749	5.6400	87772.	1337.8934	3.4834
1680.0000	4.5229	1990.2967	1.4036	1.8228	.79.9102	.8749	5.6400	88086.	1346.5663	3.4880
1690.0000	4.5345	1995.3836	1.4254	1.8087	.79.8226	.8749	5.6400	88404.	1355.2407	3.4926
1700.0000	4.5460	2000.4588	1.4707	1.7725	.79.7355	.8748	5.6400	89029.	1363.9166	3.4972
1710.0000	4.5575	2005.5224	1.4975	1.7512	.79.6489	.8748	5.6400	89352.	1372.5940	3.5018
1720.0000	4.5690	2010.5763	1.5346	1.7186	.79.5629	.8748	5.6400	89678.	1381.2728	3.5063
1730.0000	4.5805	2015.6147	0.0000	0.0000	79.4775	.8748	5.6400	90306.	1389.9530	3.5109

MAXIMUM RIT = 1725.1248
RH AT MAXIMUM = 17.6286

COMPRESSOR EFFICIENCY = .8065
COMPRESSOR SPECIFIC SPEED = 100.0000

PR	=	6.0000	COMPRESSOR PRESSURE RATIO		
T2	=	483.3984	COMPRESSOR DELIVERY TEMPERATURE		
N	=	56024.635	RPM	COMPRESSOR MECHANICAL SPEED	
CTR.	=	3.5689	IN	COMPRESSOR TIP RADIUS	
H	=	102.9129	BTU/LB	ADIABATIC WORK	
TPR	=		IN	TURBINE PRESSURE RATIO	
RT	=		FT/SEC	TURBINE TIP SPEED	
UT	=		FT	ROTOR INLET TEMP AT STRESS CONDITION	
RTT	=		IN	LOWER BLADE ROOT RADIUS	
RHU	=		IN	UPPER BLADE ROOT RADIUS	

LOW CYCLE FATIGUE LIFE = 50000. CYCLES

RTT	RT	U	RHU	NST	ETATT	TPR	SIGHMIN	TH3	R6T
1500.0000	3.8670	1890.6256	1.0540	1.8721	90.7340	.8714	5.6400	89452.	1192.6637
1510.0000	3.8781	1896.0398	1.0623	1.8689	90.6214	.8715	5.6400	89767.	1261.2734
1520.0000	3.8891	1901.4551	1.0701	1.8659	90.5103	.8716	5.6400	89887.	1209.8972
1530.0000	3.9000	1906.7574	1.0852	1.8581	90.4001	.8716	5.6400	90511.	1218.5231
1540.0000	3.9110	1912.0967	1.0906	1.8548	90.2906	.8717	5.6400	90814.	1227.1508
1550.0000	3.9219	1917.4231	1.0975	1.8512	90.1818	.8717	5.6400	91121.	1235.7804
1560.0000	3.9327	1922.7367	1.1166	1.8418	90.0738	.8718	5.6400	91767.	1244.4118
1570.0000	3.9436	1928.0375	1.1255	1.8376	89.9666	.8718	5.6400	92080.	1253.0451
1580.0000	3.9544	1933.3257	1.1342	1.8331	89.8601	.8719	5.6400	92396.	1261.6862
1590.0000	3.9652	1938.6312	1.1511	1.8226	89.7543	.8719	5.6400	93046.	1270.3512
1600.0000	3.9759	1943.8643	1.1617	1.8177	89.6493	.8720	5.6400	93367.	1278.9902
1610.0000	3.9867	1949.1149	1.1721	1.8125	89.5449	.8720	5.6400	93693.	1287.6309
1620.0000	3.9974	1954.3531	1.1824	1.8072	89.4412	.8721	5.6400	94022.	1296.2733
1630.0000	4.0021	1959.5791	1.2024	1.7941	89.3383	.8721	5.6400	94695.	1304.9173
1640.0000	4.0187	1964.7927	1.2248	1.7874	89.2260	.8722	5.6400	95030.	1313.5629
1650.0000	4.0294	1969.9943	1.2288	1.7803	89.1344	.8722	5.6400	95370.	1322.2101
1660.0000	4.0400	1975.1837	1.2524	1.7641	89.0335	.8723	5.6400	96066.	1330.8588
1670.0000	4.0506	1980.3611	1.2672	1.7561	88.9332	.8723	5.6400	96411.	1339.5091
1680.0000	4.0612	1985.5266	1.2807	1.7479	88.8336	.8724	5.6400	96739.	1348.1609
1690.0000	4.0717	1990.6801	1.2947	1.7376	88.7246	.8724	5.6400	97091.	1356.8141
1700.0000	4.0822	1995.8219	1.3277	1.7147	88.6263	.8725	5.6400	97791.	1365.4688
1710.0000	4.0927	2000.9518	1.3447	1.7026	88.5286	.8725	5.6400	98148.	1374.1248
1720.0000	4.1032	2006.0701	1.3659	1.6871	88.4215	.8726	5.6400	98510.	1382.7823
1730.0000	4.1136	2011.1768	1.4179	1.6508	88.3250	.8726	5.6400	99234.	1391.4411
1740.0000	4.1240	2016.2719	1.4485	1.6220	88.2492	.8727	5.6400	99601.	1400.1013
1750.0000	4.1344	2021.3554	1.4719	1.5940	88.1540	.8727	5.6400	100170.	1408.7628

MAXIMUM RIT = 1747.0115
RH AT MAXIMUM = 1.5359

LOW CYCLE FATIGUE LIFE = 100000, CYCLES		RHL	RHU	NST	ETATT	TPR	SIGMIN	TIM3	R6T
RT	U								
1500.0000	3.8670	1890.6250	1.3062	1.5046	90.7310	.8714	5.6400	89752.	1192.6637 ~
1510.0000	3.8781	1896.0398	1.3366	1.4756	90.6214	.8715	5.6400	89767.	1201.2734
1520.0000	3.8891	1901.4051	1.3843	1.4649	90.5163	.8716	5.6400	89887.	1209.8972
1530.0000	3.9000	1906.7574	0.0000	0.0000	90.4001	.8716	5.6400	90511.	1218.5231
									3.4186

MAXIMUM RIT = 1521.8106
RH AT MAXIMUM = 1.4694