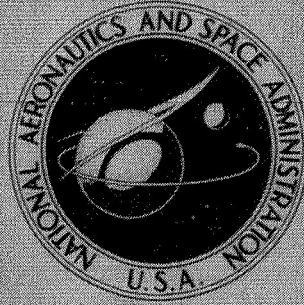


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**NASA CONTRACTOR
REPORT**



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NASA CR-1586

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**POTASSIUM TURBOALTERNATOR
PRELIMINARY DESIGN STUDY**

**I - Component Parametric Data and Initial
Integrated Turboalternator Analyses (Phase I)**

Prepared by

GENERAL ELECTRIC COMPANY

Cincinnati, Ohio 45215

for Lewis Research Center

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • SEPTEMBER 1970

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16. Abstract Parametric design data of components and configurations have been compiled in preparation of conceptual design studies of potassium turboalternators for space. 2100 ⁰ F turbine inlet, 3 to 5 years life, and 800 ⁰ to 1100 ⁰ F potassium lubricant and coolant are some of the basic design requirements. Elimination or control of turbine erosion and consideration of oil as an alternate lubricant and coolant were additional design conditions. Radial gap inductor alternators were subjected to comprehensive screening analyses to select conceptual 733 kVA designs. As the result of the screening studies, three conceptual designs at 1600 Hz, 480 V and 19,200 and 24,000 rpm were selected for detailed analysis and design. Auxiliary studies, not directly associated with the design selection analyses, were carried out to evaluate motoring capability, magnetic unbalance and reliability considerations. A number of 2 shaft, 4 bearing configurations using cradled and overhung turbines have been evolved. Performance and weight data as well as comparative risk and life expectancy have been established. Bearing/rotor response analysis has contributed an important criterion to design selection.			
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FOREWORD

The work described in this report was performed by the Nuclear Systems Programs Department of the General Electric Company in Cincinnati, Ohio, under NASA contract NAS 3-10933. The Project Manager at General Electric was Mr. E. Schnetzer, and the NASA Project Manager was Mr. J. P. Joyce, of the Lewis Research Center Space Power Systems Division.

The study was performed in two phases, Phase I to provide basic analysis of components and conceptual KTA designs for various design conditions (volume I), and Phase II to establish one preliminary design meeting specific design conditions (volume II).

General Electric was assisted by two subcontractors: The Westinghouse Aerospace Electrical Division in Lima, Ohio, responsible for the alternator design, and Mechanical Technology Incorporated in Latham, New York, contributing bearing/rotor response, as well as bearing and seal analyses.

The following people were key contributors to the work: at WAED, alternator design: A. E. King, T. C. Allen, C. C. Kouba, J. L. McCabria, and T. G. Studt; at MTI, bearing rotor response: W. D. Waldron; at GE, turbine fluid design and thermodynamics: Richard J. Rossbach and Gordon C. Wesling; mechanical design: Bluford L. Moor and Manfred O. Schnetzer; materials support: E. E. Hoffman, W. F. Zimmerman, W. R. Young, and H. A. Williams.

POTASSIUM TURBOALTERNATOR PRELIMINARY DESIGN STUDY
FINAL REPORT, PHASE I

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1. INTRODUCTION

1.1 THE TASK

On June 26, 1967, a study contract, NAS3-10933, was placed with General Electric, Space Power & Propulsion Section, for the preliminary design of a 450 kw_e turboalternator for space applications, using potassium vapor inlet temperatures of up to 2150°F. A life of three years was specified for continuous operation, with growth potential to five years. Liquid potassium at a temperature of 800 to 1100°F was to be used as a bearing lubricant, as well as alternator coolant.

As an alternate lubricant and coolant synthetic oil of 400°F was to be considered. The study was to be carried out in two phases. Phase I was to be of a conceptual nature, studying various turbine and shaft-bearing arrangements, with the alternator defined as a radial gap configuration, cradled between two bearings. Material processing information was to be compiled, the basic design problems defined, and the analytical methods established.

Phase II was planned to be of a preliminary design nature, to be based on the results of Phase I and to be carried out with the selected configuration in design details preliminary to hardware design. In accordance with the original plan, the Final Report is sub-divided into Phases I and II.

1.2 SPECIFICATIONS

The contract specifications for Phase I are the following:

KTA Design Power Output

450 KW_e at load P.F. = 0.75 Lagging

Alternator Maximum Continuous Load Rating

550 KW_e at load P.F. = 0.75 Lagging

Speed Capability, % Design = 0 to 120

Environmental Specifications = 20 G During Launch
No Operation

= 4 G In Operation Any Direction

Design Life, Years = 3, with growth potential to 5

CYCLE STATE POINTS

	<u>Aerodynamic Design</u>	<u>Structural Design</u>
Turbine Inlet Temperature, °F	2100	2150
Turbine Inlet Pressure, PSIA	165	190
Turbine Inlet Saturation Temperature, °F	2029	2080
Turbine Exit Temperature, °F	1280	1280
Turbine Exit Pressure, PSIA	7.8	7.8

TURBINE ARRANGEMENTS

Split Spool, with Interspool Moisture Separation

Single Spool, with Interstage Moisture Separation

Single Spool, with No Moisture Separation

SHAFT - BEARING ARRANGEMENTS

A.	Two Shafts	Potassium Lubricant
	Four Bearings	No Turbine Moisture Removal
	One Coupling	
B.	Two Shafts	Potassium Lubricant
	Four Bearings	Split Spool Turbine
	One Coupling	Interspool Moisture Separation
C.	Same as (A)	But Organic Lubricant
D.	Same as (B)	But Organic Lubricant

Comparative designs will be evaluated by using the following measures of performance, listed in order of importance.

Reliability

Ease of Development

Energy Conversion Efficiency

Low Weight

Special attention will be given to:

Creep

Pivot Fretting

Radiation

Erosion

Corrosion

Thermal Stress

Low Cycle Thermal Fatigue

1.3 DESIGN PROBLEMS

Of all the components required in the system, the turboalternator is probably the most crucial one as far as functional reliability and life are concerned.

For the alternator, the challenge is a combination of high speed, high temperature, inorganic insulation and liquid metal bearings. In order to obtain low weight, the rotor tip speed has to be pushed to extremes, limited by stress and creep. This implies identifying the most suitable rotor material offering the combination of high strength, low creep and attractive magnetic properties.

The employment of liquid metal bearings implies the development of a ceramic bore seal separating the potassium vapor filled rotor cavity from the stator with its windings. The bore seal, for the size requested in a 450 kw_e alternator, represents one of the major unresolved development problems.

The turbine design task is different insofar as its refractory materials of construction are new and in development and their forging and manufacturing processes are not fully established. Of special interest are weldable, high strength tantalum alloys most recently developed under NASA contracts which can be used in stator design and possibly in rotor construction.

1.4 DESIGN APPROACH

The preparations for turbine design required a compilation of material and process information. In addition to the material problems there were considerable unknowns in the mechanical design of refractory

components and their arrangements relative to each other. The biggest single unknown in designing a wet vapor turbine for 3 to 5 years of unattended operation is erosion. The Nuclear Rankine Power System, by concept, provides the turbine with only slightly superheated vapor resulting in condensation and moisture build-up within the turbine. In wet steam turbines it is known that certain moisture levels can be tolerated and that there exists a set of conditions defined by relative velocities, moisture content and materials properties forming a threshold of erosion which permits erosion free operation within certain limits. Based on long time experience with steam turbines, these limits are well established for steam power plants. They are not established, however, for potassium turbines and the refractory materials to be used in their design.

The most pertinent experience available comes from the General Electric Two-Stage Potassium Test Turbine where molybdenum TZM and TZC blades were exposed to 3.5 percent moisture entering the second stage at blade tip velocities of 760 ft/sec and only experienced minimal incipient erosion in 5000 hours of testing. This result was obtained in a 316 stainless steel loop.

Another most encouraging result was obtained in the columbium alloy two-phase corrosion loop where a sequence of nozzles and blade-like bars were used to simulate the turbine. With vapor velocities of 1000 to 1100 ft/sec and moisture levels up to 12 percent, no erosion damage was created in 5000 hours of testing.

Conceptually, a number of turbine design approaches could be distinguished. One approach simulating steam turbine practice was to

tolerate a certain threshold moisture level and to remove all excess fluid by interstage condensate extraction.

A second approach was to sub-divide the turbine into two spools and to remove condensate from the crossover ducts by in-built centrifugal separators. This approach has been used in a nuclear G. E. steam power plant. It is attractive because the centrifugal separator is much more predictable in its performance than interstage condensate extraction.

The third approach attempted operation without any condensate removal by using low blade speeds and blade loadings in conjunction with an accordingly large number of stages.

1.5 THE TEAM

In response to this most challenging design task an unusual team has been formed combining the potassium turbine know-how of GE-SPPS with the alternator design experience of Westinghouse-Lima and the expertness in bearing design and bearing/rotor response prediction of MTI.

It appeared questionable at first whether a team consisting of three different groups, working in separate places 1000 miles apart could perform effectively. It can be stated that the cooperation between the three groups was excellent and that the study in content and depth gained considerably from this most qualified team.

2.0 ALTERNATOR

2.1 INTRODUCTION

The Phase I design power output of the KTA alternator is 450 kw with a 0.75 lagging power factor. The maximum continuous rating of the alternator is to be 550 kw with a 0.75 lagging power factor. The speed range of interest is 16,000 to 32,000 rpm and the frequency range is 800 to 2000 Hz. Both potassium and organic-fluid lubricated bearing and seal systems were analyzed. The alternator is a radial gap inductor alternator and is designed to conform to the severe environmental specifications dictated by space operation.

The design study was to proceed from initial conceptual designs to preliminary designs that would be the foundation for a later, final test-hardware design. The conceptual designs to be derived from Phase I of the study; the preliminary designs to be derived from Phase II of the study; final designs will be made as part of a later study as yet undefined.

The alternator conceptual designs of Phase I are presented in this report. The designs were arrived at through the processes of elimination and optimization of candidates. Initially, parametric screening studies were carried out to eliminate all but the most desirable candidates. These parametric screening studies were concerned with (1) significant alternator internal design variables*, (2) rotor materials and (3)

* that also can exert significant influence on system considerations

alternator/system design variables such as voltage, speed, frequency and coolant temperature. Of the over forty internal design variables of interest to the alternator designer, nine were specifically studied in detail to seek optimum values. Eight different rotor materials were examined in detail and a variety of temperature, speed, frequency, and voltage combinations were evaluated as part of the system variable studies. In all, 257 designs were made and analyzed for the parametric screening studies. Separate rotor and stator detail design analyses were carried out in parallel with the parametric screening studies above and the design optimization studies below.

Three of the most promising candidate designs were selected from the screening studies and were subject to additional optimization studies to arrive at the three conceptual designs of Phase I. As a result of these studies, recommendations were made as to the most applicable designs for the preliminary designs of Phase II.

2.2 PARAMETRIC SCREENING STUDIES

Introduction

Background. Parametric screening studies of alternator design variables were performed to determine the values required to meet the primary design objectives of high reliability, ease of development, maximum energy conversion efficiency and minimum weight in the order of priority listed. These were to be met within the specification requirements regarding overload, three phase short circuit, voltage unbalance and waveform, component temperature and rotor stress limits.

Studies were performed by calculating a complete generator design for each incremental value of a given parameter within a specified range. In this manner, all aspects of the alternator design which affect the primary design objectives, the specification requirements, and material limitations could be comparatively evaluated.

Designs were compared on the basis of alternator weight and efficiency to achieve minimum system weight within the bounds of materials and specification limitations. Primary criteria and considerations used in evaluating the designs are listed in Table 2.2-1. In those cases where alternator efficiency increased as alternator weight increased, a system weight penalty per kilowatt of loss criteria was required to aid in selection of the optimum combination of weight and efficiency. A specific weight penalty of 20 pounds per kilowatt was assumed. It was found that an increased weight penalty did not significantly change the choice of parameters because the 20 pounds per kilowatt was already predominate in the influence the penalty had on the choices.

The design objectives of high reliability and ease of development were primary considerations in design tradeoff selection. For example, the choice of two conductor per slot designs and 120 degree phase belts were based primarily on these two objectives since this type of winding does not require intricate and extensive crossover connections with the relatively stiff Inconel clad silver conductors. This is an important consideration in order to eliminate damage or cracking of the conductor sheat and to prevent unnecessary exposure of the silver core in the conductor (2.2-11). A comparison of a two conductor per slot and four conductor per slot coil is shown in Figure 2.2-1. Comparison of a 120 degree phase belt and 60 degree phase belt winding is shown in Figure 2.2-2.

TABLE 2.2-1
CONSIDERATIONS FOR SCREENING DESIGNS

RELIABILITY

EASE OF DEVELOPMENT

EFFICIENCY

WEIGHT

ROTOR STRESS

ROTOR LOSSES

MAXIMUM ALLOWABLE ROTOR COOLANT TEMPERATURE

TURNS PER COIL

CONDUCTOR TEMPERATURE

VOLTAGE UNBALANCE

OVERLOAD AND THREE PHASE SHORT CIRCUIT PERFORMANCE

VOLTAGE IONIZATION AND BREAKDOWN

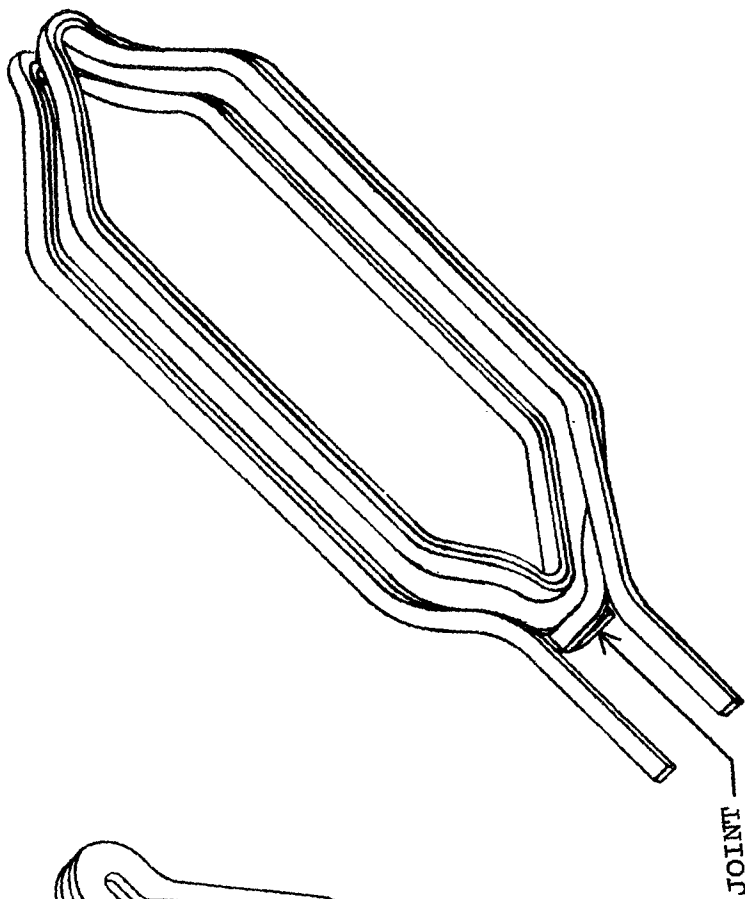
MAGNETIC UNBALANCE ROTOR FORCES

VOLTAGE WAVEFORM

REQUIRED FIELD POWER

ROTOR WEIGHT

4 CONDUCTORS / SLOT



2 CONDUCTORS / SLOT

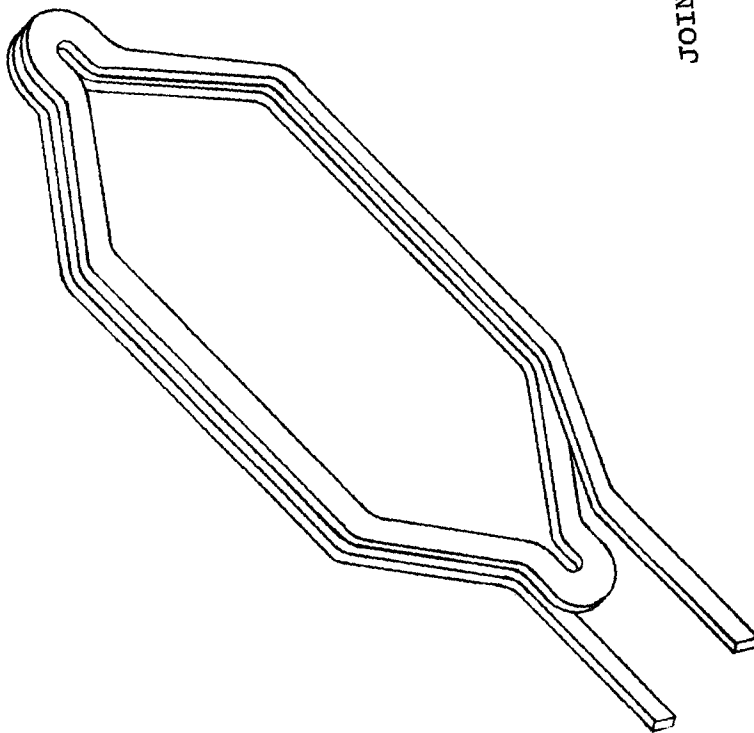


Figure 2.2-1. Coil Schematics.

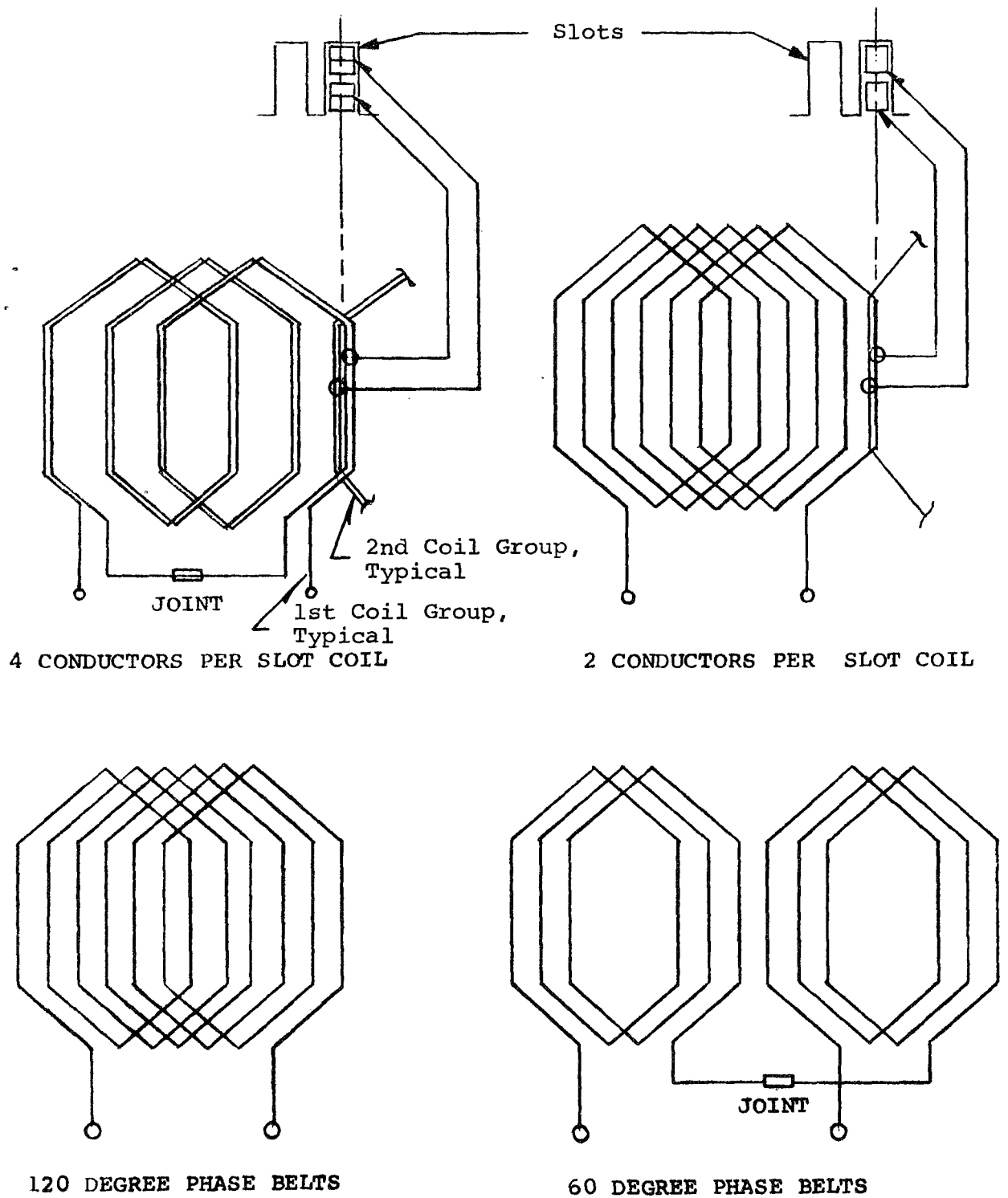


Figure 2.2-2. Coil Configurations.

Minimization of rotor pole face losses and magnetic unbalance rotor forces by proper selection of design parameters was also directed towards increasing reliability and easing required development. Low pole face losses eliminate the need for elaborate rotor cooling schemes, and reduced rotor magnetic forces alleviate the design requirements for the bearings. The comparative field power required for each design was taken into account as a secondary consideration because of its effect on voltage regulator-exciter (VRE) design. The VRE design requirements were not within the specific scope of this study.

In addition to the stator hot spot, another major mechanical design screening parameter is the rotor stress. Those designs giving too high stress during the screening studies were eliminated from further consideration for the conceptual designs. The particular criteria for establishing whether the stress was too high or not was changed during the screening studies so a few words of explanation are in order as to exactly what criteria was used.

In the past, the Westinghouse Aerospace Electrical Division has carried out similar analytical screening studies (Reference 2.2-1) where the selection criteria was as follows:

"The average tangential stress in the rotor core is calculated based on the rotor core dimensions. This stress is then increased by a stress concentration factor to account for the rotor poles attached to the core; the factor is dependent upon the number of rotor poles. This stress is then compared to the allowable stress, that which will cause less than 0.2 percent creep strain over the design life of the rotor. Those alternators having calculated average tangential rotor stresses in excess of the

allowable were said to be "stress-limited" and considered impractical."

This same criteria was proposed and used at the start of the screening studies for the KTA alternator. However, during the course of the studies, it became apparent that the above approximation was not sufficient for making a sound judgement. The basic reasons for this were that the designs at the lower temperatures were not creep strength limited and there were several borderline cases that defied selection based on the previous criteria. As a result, a more comprehensive criteria was established and the rotor stresses were analyzed in detail rather than as a single average value.

The detail calculated rotor stress used was the maximum combined (or Von Mises) stress that occurs under the pole on a plane (r-z direction) through its center; this is the most crucial and usually the largest stress to be considered in the alternator design. The Von Mises stress relationship was used to relate the biaxial stresses occurring in the rotor to the uniaxial, maximum allowable material stress. The criteria for the material stress limit was expressed in terms of the following considerations:

- (a) The stress within the rotor at 20 percent overspeed shall not exceed 90 percent* of the ultimate tensile strength of the material.

* Later in the studies, when H-11 had been selected as the rotor material, the 90 percent limit was changed to 87 percent. This is because the 87 percent line corresponds to the 72 percent yield strength limit; thus, any design meeting one of the two limits automatically met the other.

(b) The stress within the rotor at design speed shall not exceed 72 percent of the 0.2 percent offset yield strength of the material.

(c) The creep strain for 3-years operation shall be less than that required for the pole tips and bore seal to rub. The allowable strain must be determined from the initial dimensions of the actual alternator design. At the early stages of the design study it was assumed that a maximum of 0.5 percent creep strain would be permissible.

Any designs which did not meet any one of these three criteria were considered over-stressed and were dropped from further consideration. The calculation procedures used for determining the maximum principle stresses and the Von Mises stress are summarized in Appendix B of reference 2.2-2.

Other major design criteria which were used in the screening processes concern windage losses and pole face losses. The pole face calculation procedures used in the screening studies are based upon those reported in Reference 2.2-3. The windage losses were initially calculated using the results of Reference 2.2-4. Later, an improved method was derived and correlated with the original data of Smith (2.2-5). The improved method gave more accurate results and is summarized in the Appendix.

Since the separate studies were not carried out to determine configurations that minimized the windage and pole face losses until after the Base Designs had been selected, it was necessary to carry out the screening studies using fixed configurations that give minimal

losses. Unless these losses were calculated to be abnormally high (tens of kilowatts), the values calculated were not used as separate criteria for retaining or discarding a design. Rather, the calculated values were simply lumped with the rest of the losses for overall efficiency and system weight considerations. Those designs having inherently low losses were noted since they would be easient to minimize losses on later.

The goal of the screening studies was to keep the rotor windage and pole face losses near or below the one kilowatt (each) level. The one kilowatt level was not selected because of efficiency considerations but rather because these losses occur in a sensitive cooling area, the rotor cooling circuit. It was recognized that losses much over one kilowatt would obviate the chances of a simple, reliable rotor cooling configuration being sufficient.

The screening studies were carried out in successive stages according to recognized areas of separation. They were in order of their occurrence:

- a. **Internal Variables:** These are the parameters of interest to the alternator designer that do not directly effect choice of external system parameters.
- b. **Rotor Materials:** Because of the significant influence the choice of rotor material has on the design limitations and performance, separate selection studies are required which compare materials in actual designs.
- c. **System Design Variables:** These are the parameters of interest to both the alternator designer and system designers.

- d. Type of Coolant: Because of the multitude of fluids available and because there are related influences between the coolant and alternator design, separate comparative studies were made to evaluate the choices.

Selection of Coolant Temperatures. One of the intents of the screening study was to determine the best coolant temperature from the standpoint of the alternator design. Because of the interdependence between the alternator design, the initial materials selection and the limited temperature capabilities of the materials, it was necessary to assign the coolant temperature the role of a dependent variable. That is, preselected fixed values were chosen and a family of parametric data were calculated about them. Later discussion will illustrate why it is not practical to consider coolant temperature a continuously variable parameter. Also, because of the dependence of the coolant temperature on system weight, the final recommendations were based on system considerations as well as alternator considerations.

The range of coolants and temperatures to be considered in the study were:

Organic Coolant:	Rotor and Stator	-	coolant temperatures based on capabilities of fluids under consideration
Alkali Metal Coolant:	Rotor and Stator	-	800 ⁰ and 1100 ⁰ F Potassium 800 ⁰ and 1100 ⁰ F Potassium, NaK

For the various fluids considered, it was assumed that the flow could be specified such that the bulk temperature rise through the alternator would remain constant. A weighted average of film drop of 1.0⁰F was used for the potassium and NaK coolants. The purpose of these simplifications were to fix the heat sink temperature conditions. This

allowed the alternator temperature calculations to be made independent of a particular fluid or cooling scheme, thereby minimizing the number of calculations required in the preliminary stages. As a result, the discussions relate to 400°F alternators, 800°F alternators and 1100°F alternators without reference to the corresponding fluid whatever it may be. The particular reasons for the choice of 400°, 800°, and 1100°F as the fixed temperatures are discussed in Section 2.2.

Internal Design Variables

Background. It is generally recognized that system weight and performance is affected by the choice of alternator-electrical system variables such as frequency, speed, and voltage. The choice of internal alternator design variables also influences system weight and performance. Therefore, a series of trial designs were performed to comparatively evaluate the resulting trends for these variables over specified values. The trends were used to select the approximate values required to meet the specification requirements and primary design objectives prior to performing the frequency, speed, and voltage investigations. Candidate rotor materials were screened during the trial design screening studies. Rotor material screening was carried out at the lowest speed, highest frequency combination in order to prevent elimination of a candidate rotor material for reason of excessive rotor stress at a higher speed or lower frequency. The following basic design factors were held constant for the internal design variable studies.

kVa	733
Load Power Factor, lagging	0.75
Frequency	1600
Speed, RPM	16,000
Voltage (L-N)	480
Rotor Material	Hiperco-27 for 400 ^o F coolant (2.2-8) H-11 forging for 800 ^o F coolant (2.7-7,-8) 1BS1 for 1100 ^o F coolant (2.2-9)
Armature Magnetic Material	0.004 inch thick Hiperco 27 (2.2-8)
Stator Frame Material	Hiperco 27 forgings (2.2-8)
Electric Conductor Material	Anadur Insulated Inconel (2.2-6) Clad Silver
Slot Liner Material	99.5% Alumina (2.2-6)
Pole Face Configuration	Slotted
Armature Hot Spot Temperature Limits	1100 ^o or 1300 ^o F

The screening studies were performed for three coolant temperatures. The lowest temperature (400^oF) was for an organic coolant and the two higher temperatures of 800^o and 1100^oF were for a liquid potassium coolant. A temperature excursion study in the range of 200^o to 600^oF for organic coolants was performed to determine possible design benefits occurring from an organic coolant temperature other than 400^oF.

Nine internal design variables out of a possible 28 variables were chosen for inclusion in the internal design variable study. The variables selected for study were those which would significantly affect the application design goals and specification requirements and which required the knowledge of parametric trends to establish application tradeoff information for their selection. The values for other parameters were chosen based on established aerospace alternator design

principles, manufacturing procedures, material properties and material limitations.

The internal design parameters varied during the screening studies listed in Table 2.2-2. The internal design variables held constant for the trial design screening studies and their values are given in Table 2.2-3.

Results of Internal Design Parametric Screening Studies. Results of the direct axis synchronous reactance " X_d " studies for coolant temperatures of 400° , 800° , and 1100° F are given in Tables 2.2-4, 2.2-5, and 2.2-6 respectively. Results indicated that an increase in the voltage unbalance limit from 6 percent to 9 percent for the study was desirable since the specified 6 percent voltage unbalance required the value of per unit " X_d " be less than 0.7 at the expense of increased alternator weight, thereby restricting the parametric study to relatively heavier, low " X_d " machines. A per unit " X_d " of about 1.0 requires the voltage unbalance limit (for 2/3 single phase load) to be increased to nine percent. It was requested and granted that the phase unbalance be increased to 9 percent.

Rotor and stator hot spot temperatures, rotor stress at 16,000 rpm, overload and three phase short circuit limitations were found to have relatively little influence on the choice of " X_d ". Both two conductor per slot and four conductor per slot designs were examined for the 800° and 1100° F coolant temperatures. Two conductor per slot designs have the advantage of a simpler winding configuration, better transfer of the armature heat load from the slots, and a higher number of armature slots for a given " X_d " which reduces pole face loss. The disadvantage in this application is that the maximum " X_d " is limited to a relatively

TABLE 2.2-2
STUDY VARIABLES

<u>Design Variable</u>	<u>Range of Variable</u>
Direct axis synchronous reactance " x_d "	0.25 - 1.6 per unit
Armature winding current density	3000 - 9200 amp/in ²
Armature core flux density	20 - 110 kL/in ²
Armature teeth flux density	80 - 130 kL/in ²
Rotor core flux density	Range selected according to material
Radial gap	0.10 - 0.16 in.
Stator leakage flux	8 - 24 percent
Rotor leakage flux	4 - 6 percent
Phase belt	60, 120 degrees

TABLE 2.2-3
FIXED CONDITIONS

<u>Design Variable</u>	<u>Value</u>
Parallel Paths	6
Conductors in Parallel	2
Pitch Factor	0.666 or 0.8
Skew	0
Slot Configuration	Semi Closed Slot
Clearance Between End Turns	0.075 inch
Armature Conductor Insulation Thickness	0.011 inch
Ceramic Slot Cell Thickness	0.021 inch
Slot Clearance Depth	0.020 inch
Slot Clearance Width	0.018 inch
Conductor Configuration	Rectangular
Ratio of Tooth Width to Slot Pitch	0.475
Armature Stacking Factor	0.9
Armature Lamination Thickness	0.004 inch
Damper Bars on Rotor	None
Field Width Clearance (for cooling fins, insulation, etc.)	0.268 inch
Field Height Clearance (for cooling fins, insulation, etc.)	0.493 inch
Pole Embrace	0.667
Pole Face Configuration	Grooved
Field Current Density	4000 amps/in ²

TABLE 2.2-4

SUMMARY OF RESULTS OF Xd STUDY

Conditions: 733 kVA, 16000 rpm, 1600 Hz, 480 volts (L-N), 400°F coolant
 120° phase belts, radial gap = 0.14 inch
 J = 6000 amps/in², rotor core flux density = 130 kL/in²

No. Slots	Xd P.U.	Electrical Weight lbs.	Efficiency %	% Voltage Unbalance	Armature Hot-Spot, °F	Field Current Ratios	
						Overload	3Ø Short
4 CONDUCTORS PER SLOT							
162	1.54	819	89.0	15.09	876	1.23	1.74
144	1.22	799	91.5	11.57	810	1.16	1.58
126	.87	827	92.0	8.18	768	1.10	1.35
108	.69	915	92.1	6.33	741	1.06	1.20
90	.49	957	91.2	4.52	772	1.03	.96
72	.34	1062	89.4	3.12	790	1.00	.73

TABLE 2.2-5

SUMMARY OF RESULTS OF Xd STUDY CONDITIONS

Conditions: 733 kVA, 16000 rpm, 16000 rpm, 1600 Hz, 480 volts (L-N),
800°F coolant temperature, 60° phase belts

No. Slots	Xd P.U.	Electrical Weight lbs.	Efficiency %	% Voltage Unbalance	(1) Armature Hot-Spot Temp., °F	Field Current Ratios	
						Overload	3ϕ Short Circuit
4 CONDUCTORS PER SLOT							
162	1.44	768	90.7	12.83	1188	1.20	1.765
144	1.18	786	92.2	10.28	1114	1.14	1.615
126	.92	815	92.3	7.95	1098	1.10	1.465
108	.69	857	91.9	5.90	1105	1.06	1.21
90	.48	917	90.8	4.09	1096	1.02	.941
72	.32	1046	87.7	2.78	1098	.99	.602
2 CONDUCTORS PER SLOT							
198	.65	942	91.9	5.69	1081	1.05	1.13
180	.57	1018	92.2	4.91	1047	1.04	1.05
162	.50	1126	91.9	4.26	1044	1.03	.948
144	.42	1244	91.2	3.53	1042	1.02	.834
126	.35	1413	89.5	2.90	1056	1.00	.726
108	.25	1508	87.6	2.10	1058	.993	.555

(1) at 0, 0, 2/3 rated amperes per phase

TABLE 2.2-6

SUMMARY OF RESULTS OF Xd STUDY

Conditions: 733 kVA, 16000 rpm, 1600 Hz, 480 volts (L-N), 1100°F coolant⁽¹⁾
 120° phase belts, radial gap = 0.13 inch

No. Slots	Xd P.U.	Electrical Weight lbs.	Efficiency %	% Voltage Unbalance	Armature Hot-Spot, °F	Field Current Ratios	
						Overload	3Ø Short
4 CONDUCTORS PER SLOT							
180	1.57	882	89.8	14.04	1465	1.16	1.46
162	1.29	872	91.2	11.36	1449	1.42	1.38
144	1.02	872	91.7	8.85	1434	1.33	1.19
126	.77	948	91.3	6.64	1436	1.15	1.08
108	.56	1018	89.4	4.78	1416	1.13	.71
2 CONDUCTORS PER SLOT							
216	.66	1051	91.4	5.78	1384	1.13	.96
198	.59	1135	91.8	5.08	1365	1.18	.93
180	.51	1211	91.7	4.38	1360	1.07	.83
162	.46	1364	91.2	3.88	1359	1.05	.81

(1) Rotor material for 1100°F studies is experimental precipitation hardened cobalt alloy
 (1BS1) being developed on NAS3-6465.

low value due to a practical limit in the maximum number of slots, resulting in heavier designs.

A per unit "Xd" range of 0.7 to 1.0 was selected for the three coolant temperatures for use in the remaining studies based on the best combinations of alternator weights and efficiency occurring in this range.

Results of the armature winding phase belt study are summarized in Table 2.2-7. The choice of phase belts is independent of temperature therefore the study performed for the 800^oF coolant temperature is applicable to the 400^o and 1100^oF coolant temperatures. A phase belt angle of 120 degrees was selected despite a slight weight and efficiency penalty due to the 120 degree belt windings because they eliminate the crossover connections and joints required for 60^o phase belts. This improves reliability and ease of development relative to the use of clad silver conductors.

Tables 2.2-8, 2.2-9, and 2.2-10 present results of the current density studies performed for coolant temperatures of 400^o, 800^o, and 1100^oF. A current density of 7000 amperes/inch² was selected for the 400^oF organic fluid coolant temperature based on minimum system weight. A current density of 6000 amperes/inch² for the 800^oF coolant temperature gave maximum efficiency and near minimum alternator weight while limiting armature hot spot temperatures to below 1100^oF. A current density of 6000 amperes/inch² was also chosen for the 1100^oF coolant temperature based on near minimum system weight and a voltage unbalance less than 9 percent. A current density of about 2500 amperes/inch² would be required to reduce armature hot spot temperatures below 1300^oF resulting in a severe increase in system weight and a voltage

TABLE 2.2-7

SUMMARY OF RESULTS OF PHASE BELT STUDY CONDITIONS

Conditions: 733 kVA, 16000 rpm, 1600 Hz, 480 volts (L-N),
800°F coolant temperature, 180 slots, 2 conductors per slot

PHBLT DEGREES	X_d P. U.	WEIGHT LBS.	EFFICIENCY %	% VOLTAGE UNBALANCE	FIELD CURRENT RATIO		PITCH	ARMATURE HOT SPOT TEMP. ° F
					OVERLOAD	ϕ SHORT CIRCUIT		
60	.57	1018	92.2	4.91	1.04	1.05	.667	1047
120	.55	1064	91.9	4.75	1.04	1.01	.800	1072

TABLE 2.2-8

SUMMARY OF RESULTS OF CURRENT DENSITY STUDY

Conditions: 733 kVA, 16000 rpm, 1600 Hz, 480 volts (L-N), 400° F coolant,
120° phase belts, 126 slots, 4 conductors per slot

Current Density ² amps/in	Electrical Weight lbs.	Efficiency %	% Voltage Unbalance	Field Current Ratios		Armature ^o Hot Spot, °F
				Overload	3 ϕ Short	
3933	892	91.4	8.61	1.1	1.36	712
5061	864	91.9	8.39	1.1	1.38	739
5753	842	92.1	8.20	1.1	1.35	768
7472	817	92.0	8.09	1.1	1.34	865
8332	807	91.9	8.05	1.1	1.34	923
9217	800	91.7	8.03	1.1	1.34	988

TABLE 2.2-9

SUMMARY OF RESULTS OF CURRENT DENSITY STUDY CONDITIONS

Conditions: 733 kVA, 16000 rpm, 1600 Hz, 480 volts (L-N), 800 °F coolant
180 slots, 120° phase belts, 2 conductors per slot

CURRENT DENSITY AMPS/IN ²	X _d P.U.	WEIGHT LBS.	EFFICIENCY		% VOLTAGE		FIELD CURRENT RATIOS		ARMATURE HOT-SPOT TEMP. °F
			%		UNBALANCE		OVERLOAD	3 ϕ SHORT CIRCUIT	
3933	.55	1099	91.6		4.87		1.04	1.01	1017
5061	.55	1074	91.8		4.77		1.04	1.01	1045
5753	.55	1064	91.9		4.75		1.04	1.01	1072
7472	.54	1048	91.6		4.74		1.04	1.01	1144
8543	.54	1042	91.4		4.76		1.04	1.01	1204

TABLE 2.2-10

SUMMARY OF RESULTS OF CURRENT DENSITY STUDY

Conditions: 733 kVA, 16000 rpm, 1600 Hz, 480 volts (L-N), 1100°F coolant 120° phase belts, 144 slots, 4 conductors per slot, radial gap = 0.13 inch							
Current Density ² Amps/in ²	Xd P.U.	Electrical Weight lbs.	Efficiency %	% Voltage Unbalance	Armature _O Hot-Spot, °F	Field Current Ratios	
						Overload	3φ Short
9217	1.00	825	90.6	8.83	1645	1.3	1.23
8332	1.01	833	91.0	8.82	1591	1.3	1.23
7472	1.01	842	91.3	8.82	1539	1.3	1.22
5753	1.01	851	91.6	8.76	1434	1.6	1.12
5061	1.03	882	91.7	8.93	1398	1.3	1.20
3933	1.08	956	92.0	9.30	1342	1.2	1.37
3070	1.11	980	91.3	9.73	1317	1.3	1.3

unbalance above 9 percent. Alternate choices are to (1) operate at a higher current density (6000 amperes/inch²) and reduce hot spot temperatures by an improved cooling configuration for the windings or (2) use a simple configuration such as required for the 800°F coolant (conduction to stator stacks and radiation off end turns to stator frame), allow the hot spot to go as high as 1500°F and use a conductor capable of the temperature, e.g. CUBE* alloy insulated with plasma-arc sprayed alumina.

Results of the flux density studies for the three coolant temperatures are given in Tables 2.2-11, 2.2-12, and 2.2-13. Tooth densities of 130, 120, and 115 kilolines per square inch were selected for coolant temperatures of 400°F, 800°F, and 1100°F respectively based on minimum system weight, voltage unbalance, and proximity to magnetic saturation of the teeth.

An armature core flux density of 60 kilolines per square inch was chosen for the 400°F and 800°F coolant temperatures and 70 kilolines per square inch for the 1100°F coolant temperature based on near minimum system weight, low armature conductor temperature, and near minimum stator outside diameter. Alternator weight and stator outside diameter increase rapidly for density values below 60 kilolines per square inch and although a slightly lighter system weight is indicated for a lower core density (considering a weight penalty of 100 pounds per kW), the higher values were retained since very large core depths may not be fully utilized by the high frequency (1600 Hz) flux wave.

* Handy and Harman's dispersioned-strengthened copper alloy, (Reference 2.2-6)

TABLE 2.2-11

RESULTS OF FLUX DENSITY STUDIES - 400° F COOLANT

Conditions: 733 kVA, 16000 rpm, 1600 Hz, 480 volts (L-N), 120° phase belts,
 J = 7000 amps/in², 126 slots, 4 conductors per slot,
 radial gap = 0.14 inch

Flux Density kl/in ²	Electrical Weight x (lbs)	Efficiency (%)	% Voltage Unbalance (3)	Field Current Ratios		Armature Hot Spot (°F)	Xd P.U.
				Overload	3ϕ Short		
ARMATURE CORE FLUX DENSITY							
20	1164	93.2	8.04			1020	.85
30	967	92.9	8.07			948	.86
40	889	92.6	8.08			910	.86
50	845	92.3	8.09			883	.86
60	817	92.0	8.09			865	.86
70	797	91.7	8.10			853	.86
80	782	91.4	8.10			844	.86
90	771	91.1	8.11			837	.86
100	762	90.9	8.11			829	.86
110	756	90.6	8.11			824	.86
TOOTH FLUX DENSITY							
89	1077	91.3	11.65	1.18	1.58	828	1.26
99	981	91.6	10.33	1.15	1.52	837	1.11
110	867	91.8	8.92	1.12	1.41	853	.95
120	817	92.0	8.09	1.10	1.34	865	.86
130	775	92.1	7.36	1.09	1.27	882	.79

TABLE 2.2-11 (Continued)

Conditions: 733 kVA, 16,000 rpm, 1600 Hz, 480 volts (L-N), 400 °F coolant
 120° phase belts, J = 7000 amps/in²
 126 slots, 4 conductors per slot, radial gap = 0.14 inch

Flux Density Kl/in	Electrical Weight lbs.	Rotor Weight lbs.	Efficiency %	% Voltage Unbalance	Field Current Ratios		Xd P.U.
					Overload	10% Overvoltage	
ROTOR CORE FLUX DENSITY							
80.1	855	381	92.1	6.88	1.07	1.10	.77
100.2	830	346	92.1	7.37	1.08	1.10	.80
118.9	844	344	92.1	8.08	1.08	1.11	.87
130.2	817	316	92.0	8.09	1.10	1.14	.86
140.3	821	315	91.9	8.38	1.12	1.20	.89
160.3	819	294	91.4	8.70	1.16	1.28	.91

TABLE 2.2-12

SUMMARY OF RESULTS OF FLUX DENSITY STUDIES

Conditions: 733 kVA, 16000 rpm, 1600 Hz, 480 v₂lts (L-N), 800° F coolant,
120° phase belts, J = 6000 amps/in²

Flux Density Kl/in ²	Electrical Weight lbs.	Efficiency		% Voltage Unbalance	Field Current Ratios			Armature Hot Spot (°F)	Xd P.U.
		%			Overload	3φ	Short		
ARMATURE CORE FLUX DENSITY (180 slots, 2 conductors per slot)									
20	1583	93.9		4.72	1.03	1.01	1144		
30	1297	93.5		4.73	1.04	1.02	1121		
40	1191	93.0		4.74	1.04	1.02	1112		
50	1130	92.6		4.74	1.04	1.01	1090		
60	1091	92.3		4.75	1.03	1.01	1074		
70	1064	91.9		4.75	1.04	1.01	1072		
80	1044	91.6		4.75	1.04	1.00	1062		
90	1028	91.3		4.75	1.03	1.00	1058		
100	1016	91.0		4.75	1.04	1.00	1056		
110	1007	90.7		4.75	1.04	.98	1049		
TOOTH FLUX DENSITY (126 slots, 4 conductors per slot)									
80	1226	91.5		11.86	1.17	1.63	1110	1.34	
90	1101	92.0		10.36	1.14	1.56	1126	1.16	
100	1005	92.2		9.18	1.11	1.49	1126	1.03	
110	937	92.2		8.29	1.09	1.43	1128	.93	
120	885	92.3		7.51	1.09	1.36	1134	.85	
130	833	92.3		6.79	1.08	1.25	1143	.78	
ROTOR CORE FLUX DENSITY (H-11 Rotor, Air Gap = 0.13, 126 slots, 4 conductors per slot)									
70	898 (1)	92.1		7.22	1.12 (2)		1161	.83	
80	833	92.3		7.41	1.14 (2)		1134	.94	
90	886	92.1		7.85	1.39 (2)		1132	.88	
95	874	92.0		7.73	1.84 (2)		1133	.86	
100	891	91.3		7.97	2.90 (2)		1134	.89	

(1) Corresponding rotor weights are 420, 386, 379, 354 and 353 lbs.

(2) At 10% over voltage instead of at overload.

TABLE 2.2-13

SUMMARY OF FLUX DENSITY STUDIES

Conditions: 733 kVA, 16000 rpm, 1600 Hz, 480 volts (L-N), 1100° F coolant, 120° phase belts, J = 6000 amps/in², 144 slots
4 conductors per slot, radial gap = 0.14 inch

Flux Density Kl/in ²	Electrical Weight Lbs.	Efficiency %	% Voltage Unbalance	Field Current Ratios Overload	3 ϕ Short	Armature Hot-Spot, °F	Xd P.U.
ARMATURE CORE FLUX DENSITY							
20.0	1277	93.1	8.44			1479	.97
30.0	1060	92.9	8.49			1461	.97
40.0	975	92.7	8.51			1449	.97
50.0	931	92.5	8.53			1438	.97
60.0	903	92.3	8.54			1436	.97
70.0	884	92.2	8.55			1432	.98
80.0	870	92.0	8.55			1429	.98
90.0	858	91.8	8.55			1427	.98
100.0	850	91.6	8.56			1423	.98
110.0	844	91.3	8.56			1422	.98
TOOTH FLUX DENSITY							
129.6	870	90.7	7.49	2.04	0.8	1436	.87
120.3	884	92.2	8.55	1.27	1.23	1432	.98
110.0	930	92.2	9.61	1.21	1.42	1429	1.09
100.1	999	92.3	10.79	1.21	1.57	1427	1.21
89.7	1095	92.0	12.38	1.25	1.67	1427	1.37
84.1	1154	91.9	13.38	1.27	1.69	1425	1.49
ROTOR CORE FLUX DENSITY (1)							
45.2	954	92.0	7.7			1438	.90
50.2	942	92.1	8.18			1438	.95
56.2	911	92.1	8.29			1438	.95
60.4	903	92.1	8.54			1438	.97
70.4	870	92.1	8.74			1440	.99

(1) Corresponding rotor weights are 497, 483, 440 and 408 pounds.

The rotor materials selected for the 400^o, 800^o, and 1100^oF coolant temperatures for the internal design parametric screening studies were Hiperco 27, premium quality H-11 (AMS 6487), and 1BS1 experimental alloy based on a preliminary assessment of the required material strength and magnetic properties for the specified coolant temperatures. Final selection of rotor materials was performed in a latter stage of the study. Selected densities for the 400^oF coolant temperature, Hiperco-27 rotor, the 800^oF coolant temperature, H-11 rotor, and the 1100^oF coolant temperature, 1BS1 rotor were 130, 80, and 60 kilolines per square inch respectively based on the best combination of alternator weight and efficiency, rotor weight, and field current ratios for a 10 percent overvoltage condition which is a measure of the proximity of the rated operating density to the saturation level of the rotor material.

Radial gap study results are tabulated in Table 2.2-14 and 2.2-15. The choice of radial gap (0.140 inch) for the 400^o and 800^oF coolant temperatures was based on achieving minimum weight and maximum efficiency at reasonable levels of rotor stress, pole face loss, and voltage unbalance. Where a choice existed between two alternatives (0.130 or 0.140 inch) of near equal overall designs, the largest was chosen to minimize magnetic unbalance forces, provide increased radial clearance, and reduce pole face loss.

Results of the 800^oF studies are applicable to the 1100^oF coolant designs.

Results of the leakage flux study for the 800^oF coolant are tabulated in Table 2.2-16 and are applicable to the 400^o and 1100^oF coolant designs since leakage flux is independent of temperature. A

TABLE 2.2-14

RESULTS OF RADIAL GAP STUDY

Conditions: 733 kVA, 16000 rpm, 1600 Hz, 480 volts (L-N), 400^oF coolant, 120^o phase belts,
J = 7000 amps/in², 126 slots, 4 conductors per slot

<u>Radial Gap (inches)</u>	<u>Electrical Weight (lbs)</u>	<u>Efficiency (%)</u>	<u>% Voltage Unbalance</u>	<u>Rotor Weight (lbs)</u>	<u>Xd P.U.</u>
.16	783	92.2	6.66	304	.71
.15	787	92.2	7.11	306	.76
.14	769	92.1	7.35	290	.79
.13	772	92.0	7.90	293	.85
.12	757	91.9	8.20	276	.89
.10	771	91.6	9.70	296	1.10

TABLE 2.2-15

SUMMARY OF RESULTS OF RADIAL GAP STUDY

Conditions: 733kVA, 16000 rpm, 1600 Hz, 480 volts (L-N), 800°F Coolant, 120° phase belts,
J = 6000 amps/in², 126 slots, 40 conductors per slot

Radial Gap (inches)	Electrical Weight (lbs)	Efficiency (%)	% Voltage		Rotor Weight (lbs)	Xd P.U.
			Unbalance			
.12	894	91.7	8.10		397	.92
.13	880	91.8	7.42		386	.84
.14	872	91.8	6.83		378	.77
.16	891	91.7	6.21		395	.70

NOTE: The slight drop in efficiency from previous cases is due to the inclusion of rotor windage losses in the efficiency calculations.

TABLE 2.2-16

SUMMARY OF RESULTS OF LEAKAGE FLUX STUDIES

Conditions: 733 kVA, 16000 rpm, 1600 Hz, 480 volts (L-N), 800°F coolant, 120° phase belts, J = 6000 amps/in², 4 conductors per slot

% Leakage	Electrical Weight (lbs)	Efficiency (%)	% Voltage Unbalance	Rotor Weight (lbs)	Xd P.U.
STATOR LEAKAGE FLUX (126 slots)					
24.0	870	91.8	6.99	370	
20.0	868	91.8	6.89	374	
16.0	872	91.8	6.77	382	
12.0	883	91.7	6.68	396	
8.0	903	91.8	6.26	415	
ROTOR LEAKAGE FLUX (126 slots)					
4.3	875	91.6	6.55	389	.75
4.9	896	91.6	7.08	404	.80
5.1	875	91.8	6.90	376	.77
5.3	887	91.8	7.11	384	.79
5.8	911	91.8	7.45	399	.83
6.4	887	91.9	7.45	371	.82
ROTOR LEAKAGE FLUX (144 slots)					
4.3	810	91.8	8.87	348	1.00
4.9	814	91.8	9.19	336	1.02
5.1	829	91.8	9.53	347	1.06
5.3	837	91.8	9.78	351	1.08
5.8	852	91.8	10.18	358	1.12

stator leakage of 20 percent of the main path flux and a rotor leakage of 5 percent of the rotor useful flux were chosen based on minimum weights and maximum efficiencies consistent with acceptable rotor stress and voltage unbalance values.

A summary of the selected design variables resulting from the internal design variable parametric studies is presented in Table 2.2-17.

Rotor Materials Screening Studies

Background. Various rotor alloys were investigated to determine the best rotor material for the three coolant temperatures, 400^o, 800^o, and 1100^o F. A complete generator design was calculated for various values of rotor core flux density incorporating the selected internal design parameters from the previous studies. After selecting the optimum rotor core flux density for each material, generator designs were compared on the basis of efficiency, weight, rotor material limitations and specification requirements. The candidate materials selected for the rotor material screening studies, the applicable coolant temperatures and the maximum hot spot (pole face) temperatures allowed based on material property limitations are summarized in Table 2.2-18.

While the list on Table 2.2-18 might appear short relative to the importance of the rotor material on the design outcome, one must recognize the list is the result of a number of related rotor materials screening and selection studies which have been carried out over the past decade (Reference 2.2-1, -7, -8, -9, -10). Earlier studies involved little more than screening a number of candidates based upon their magnetic and mechanical property data alone and the acquisition

TABLE 2.2-17
SUMMARY OF RESULTS OF PARAMETRIC SCREENING STUDIES

Parameter	400°F Coolant	800°F Coolant	1100°F Coolant
<u>I. INTERNAL DESIGN VARIABLES</u>			
Direct Axis Synchronous Reactance, X_d	0.7 to 1.0	0.7 to 1.05	0.7 to 1.0
Phase Belt, Degrees, Pitch	120°, 0.8	120°, 0.8	120°, 0.8
Armature current density, amps/in^2	7000.	6000.	6000.
Armature core flux density, kL/in^2	60.	60.	70.
Armature tooth flux density, kL/in^2	130.	120.	115.
Rotor core flux density, kL/in^2	130.*	80.	60.
Radial gap, inches	0.14	0.14	0.14
Stator Leakage flux, percent	20.	20.	20.
Rotor Leakage flux, percent	5.	5.	5.
* For Hiparco 27 rotor material. After later rotor materials studies and stress analysis studies, the rotor material for 400°F coolant was changed to H-11 steel. The flux density was reoptimized for the H-11 at 400°F and a value of 90 kL/in^2 was selected.			

TABLE 2.2-18

MATERIALS EXAMINED DURING ROTOR MATERIALS SCREENING STUDIES

	Coolant Temp. °F	Max. Hot-Spot Temp. °F
15% Ni Maraging Steel - specified in work statement; age-hardenable high-strength steel with reasonably good magnetic properties; limited to organic coolant temperatures due to low Curie temperature that is time-temperature dependent; properties summarized in reference 2.2-8.	400	700
Hiperco-27 Alloy - high permeability soft magnetic alloy; usually considered a stator material; applicable to high temperatures but strength limited; properties summarized in reference 2.2-8.	400 800	900
H-11 Steel @ 45 R _c - high-strength tool steel softened somewhat to improve magnetic properties; considerable data available on this alloy; properties summarized in references 2.2-7 and -8.	400 800	950
1AS2 Alloy - precipitation-hardened developmental alloy; identified by heat on Contract NAS 3-6465; represents improvement over H-11 steel but is too brittle in its present developmental state; large forging techniques need to be developed also; properties summarized in reference 2.2-9.	800	1025
NIVCO - originally developed as a turbine blading material; adapted as a magnetic rotor alloy even though its combined magnetic and mechanical properties are barely adequate; represents only readily available materials capable of operation over 800°F coolant temperature; properties summarized in reference 2.2-8.	800 1100	1200
1BS1 Alloy - precipitation-hardened developmental alloy; identified by heat on Contract NAS 3-6465; represents improvement over NIVCO; only requires development of large forging techniques to make it readily available for designs; properties summarized in reference 2.2-9.	1100	1300
NASA "Superalloy" - 7.5w, 2.5Fe, 1Ti, 0.5Zr, 0.5C; developmental alloy; magnetic property data supplied in private communication with NASA IeRC personnel; considered to be similar to 1BS1 alloy per Abstract BG-6, International Congress on Magnetism, September 11-15, 1967, Boston, Mass.	1100	-
D.S. 27% Fe-Co - dispersion-strengthened developmental alloy; applicable to very high temperatures only because not competitive with lower temperature alloys at lower temperatures; properties summarized in reference 2.2-9.	1100	1300

of additional data and development on the most promising candidates. Later studies (2.2-1) involved screening materials in actual alternator designs over the 500^o to 1400^oF coolant temperature range.

At the lower coolant temperatures, it must be recognized there exist a large number of possible candidates. In order to limit the amount of analysis required to arrive at a selection, a bracketing process was used. This was possible because, as a general rule, materials having superior magnetic properties are weak structurally and those having superior mechanical properties are weak magnetically; in between, there are a host of materials having intermediate properties in both areas. By evaluating designs that bracket the spectrums from superior magnetics to superior strength, it was expected that some insight could be gained on intermediate properties that would give design improvements. Hiperco-27 and H-11 steel were considered to be the bracketing materials. As seen later, the differences in designs between these two alloys were so small as to obviate the need for examining intermediate materials. The reasons for the small difference are related to the influence the stator design has on the minimum possible size; see Section 2.2.

Relative to the mechanical design aspects of alternator rotor design, rotor materials and coolant temperature, one finds the rotor materials selection to be very dependent upon and limited by peak operating temperatures that are a function of coolant temperature and the rotor losses. This dependency is so great that as little as 100^oF difference in coolant temperature can result in significantly different material choices and resulting designs. Fortunately, for the temperatures of interest to the KTA alternator, the temperature areas

giving significant changes can be identified to various plateaus. As a result, the highest coolant temperature of the plateau can be used to represent all possible temperatures on that plateau. In other words, a single coolant temperature can be used to represent all the results that could be obtained for all temperatures on that plateau and the amount of analyses required to obtain the results are therefore minimized.

For purposes of the present discussion, the dependencies and plateaus are best illustrated by the data summarized on Figure 2.2-3. On that figure are listed the various rotor materials analyzed in the KTA study and the rotor core temperature which, for the discussion, may be taken as the coolant temperature. This is because the KTA alternator rotor losses will be minimized, thereby minimizing the temperature gradient in the core. The indicating bars on the figure show the various temperature limitations of the alloys. The "likely" temperature and "too high" temperature are self-explanatory. The "too low" temperature bar is to illustrate where the temperature is too low to fully utilize the magnetic and mechanical capabilities of the high temperature alloy; there are better alloys with lower temperature limitations that are better suited. For example, at 800^oF, the H-11 steel gives lower alternator electric specific weight (electrical weight divided by electrical rating) and has higher allowable limiting stress than the NIVCO or 1BS1 alloy. At 900^oF, however, the H-11 must be completely ruled out due to loss of creep strength and the NIVCO or 1BS1 alloy must be used. Thus, one can say the 800^oF temperature is "too low" for proper utilization of the NIVCO type alloy.

The criteria which decides whether the temperature is "too high", "too low" or "likely" is based upon the data shown within the

ROTOR CORE MATERIAL TEMPERATURE

ROTOR MTL.	200°	300°	400°	500°	600°	700°	800°	900°	1000°	1100°	1200°
			M.P.T.T. = MAX. POLE TIP TEMPERATURE								
15% NICKEL MARAGING STEEL	176 BURST	171 BURST	167 BURST 1.50	165 BURST	165 BURST 1.49	(KPSI) (#/KW)	M.P.T.T. = 700°F				
HIPERCO-27	57 YIELD	56.3 YIELD	55.3 YIELD 1.40	54 YIELD	51.6 YIELD	48.4 YEILD	30.5* CREEP 1.49	(KPSI) (#/KW)	M.P.T.T. = 900°F		
H-11 STEEL AT 45 RC	125 YIELD	122 YIELD	118 YIELD 1.48	113 YIELD	110 YIELD 1.49	105 YIELD	93 CREEP 1.50	45 C	(KPSI) (#/KW)	M.P.T.T. = 950°F	
1AS2 ALLOY, NAS3-6465 (Approx.Stress)	185 BURST	178 BURST	172 BURST	165 BURST	158 BURST 1.51		140.5 BURST 1.52	72 CREEP	(KPSI) (#/KW)	M.P.T.T. = 1025°F	
NIVCO ALLOY	M.P.T.T. = 1200°F				(KPSI) (#/KW)	72 YIELD	69 YIELD 1.68	66 YIELD	63 YIELD	57* Yorc 1.72	
1BS1 ALLOY, NAS3-6465 (Approx. Stress)	M.P.T.T.		1300°F		(KPSI) (#/KW)	80 YIELD	77 YIELD	76.5 YIELD	74.5 YIELD	71 YIELD 1.64	58.5 CREEP 41 C
NASA"SUPERALLOY" 7.5W, 2.5Fe, 1Ti, .5Zr, .5C	M.P.T.T.		1300+ °F		(KPSI) (#/KW)					YIELD 1.72	
DISPERSION STRENGTH 27% Fe-Co ALLOY NAS3-6465	M.P.T.T.						1300°F		(KPSI) (#/KW)	15* CREEP 1.50	

TOO LOW TEMP.

LIKELY TEMP. RANGE

TOO HIGH TEMP.

XX.X KPSI

CRITERIA

LBS/KW

← LIMITING STRESS

← LIMITING

← STRESS CRITERIA

SPECIFIC ELEC. WT. AT 16000 RPM

-YIELD 0.72* 0.2% OFFSET YIELD STRENGTH

-BURST, 0.9* σULTIMATE + 1.44 (20% OVERSPEED)

-CREEP, STRESSES SHOWN ARE FOR 0.4% CREEP

*INDICATES ROTOR OVER-STRESSED AT 16000 RPM

Figure 2.2-3. Rotor Material Temperature Comparison.

indicating bars, the maximum allowable pole tip temperature, the magnetic properties (typical data shown on Figure 2.2-4 thru 7) and a considerable amount of experienced design judgement. The limiting stress values and criteria are based on published property data and the three stress criteria of Section 2.2. The maximum pole tip temperatures are based upon judgement relative to both magnetic and mechanical design limitations. For instance, the 700^oF limit for maraging steel is due to magnetic limitations. The opposite is true for the 900^oF limit for Hiperc-27; it is creep strength limited. For the H-11 steel and 1AS2 alloy, both limitations are cause for the specified maximums.

Relative to the limiting stress criteria shown, the temperature at which the design becomes creep limited is generally the temperature at which it is first considered as "too high". This is because, of the three criteria, creep limitations are the least desirable. Notable exceptions shown on the chart concern H-11 steel, 1BS1 alloy and dispersion-strengthened 27% Fe-Co alloy. The H-11 may still be considered safe at 800^oF because considerable creep data is available to permit reasonable design predictions. However, the data that is available is not considered precise enough for accurate, reliable design prediction over 3 to 5 years life. If the 800^oF were used in the KTA alternator design, long-term data would have to be taken from samples made from the same forging that would be used to fabricate the actual alternator rotor. The 1BS1 alloy is shown as creep limited and "too high" at 1200^oF; since it is still superior to the dispersion-strengthened 27% Fe-Co alloy, the latter must be listed as "too low" at 1200^oF. As a result, the 1200^oF temperature appears as a "no-mans land" without any likely material.

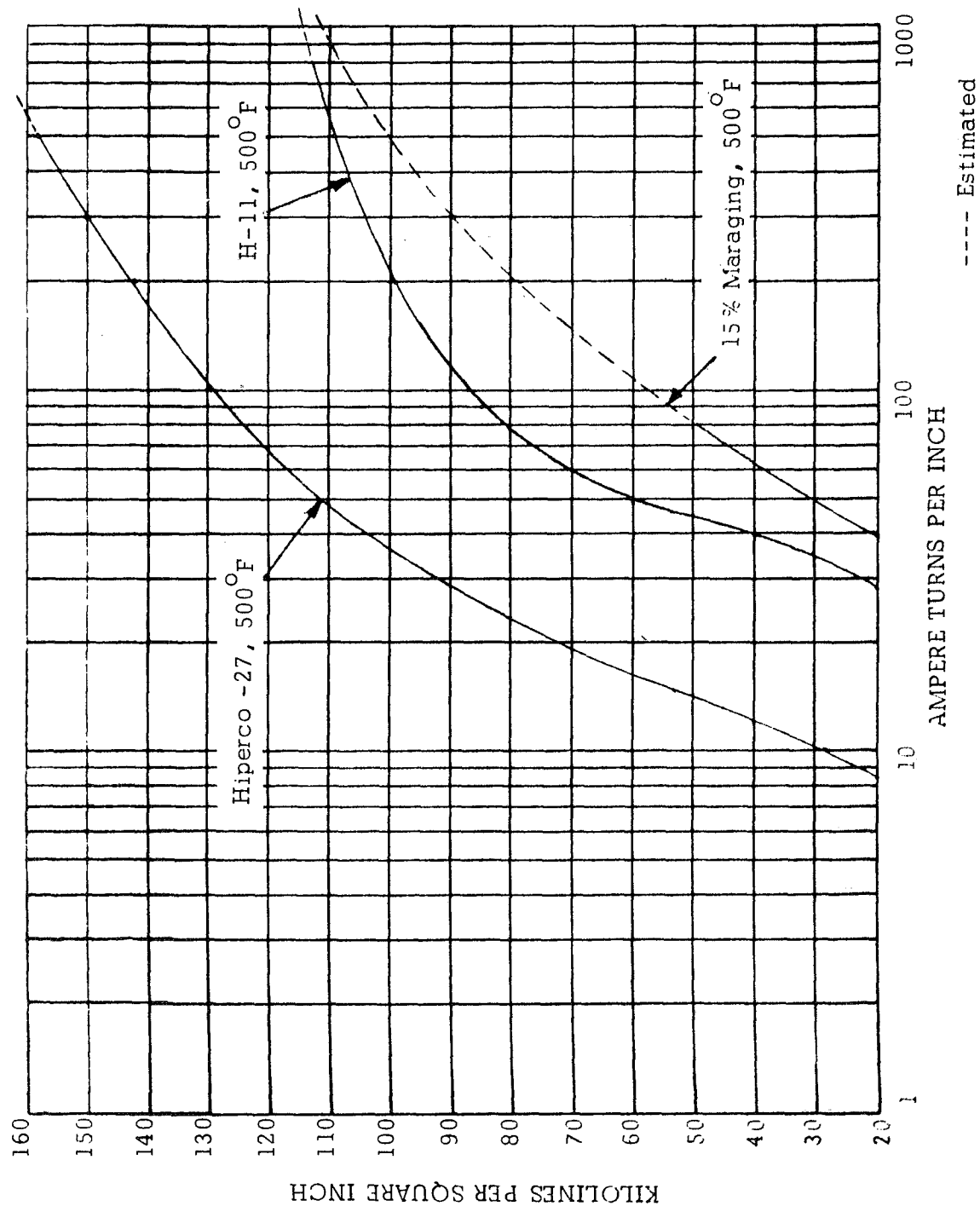


Figure 2.2-4. Magnetization Curves.

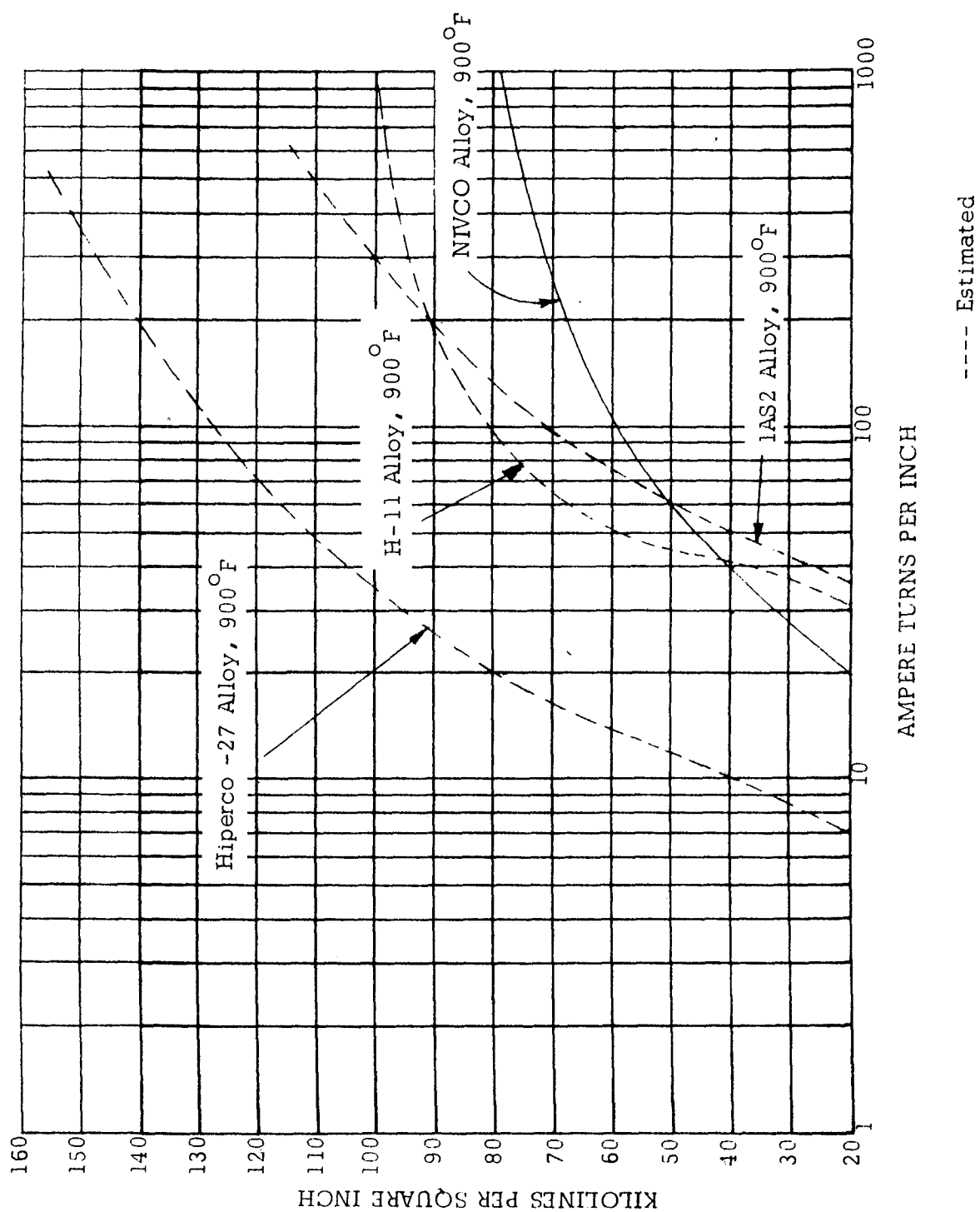


Figure 2.2-5. Magnetization Curves.

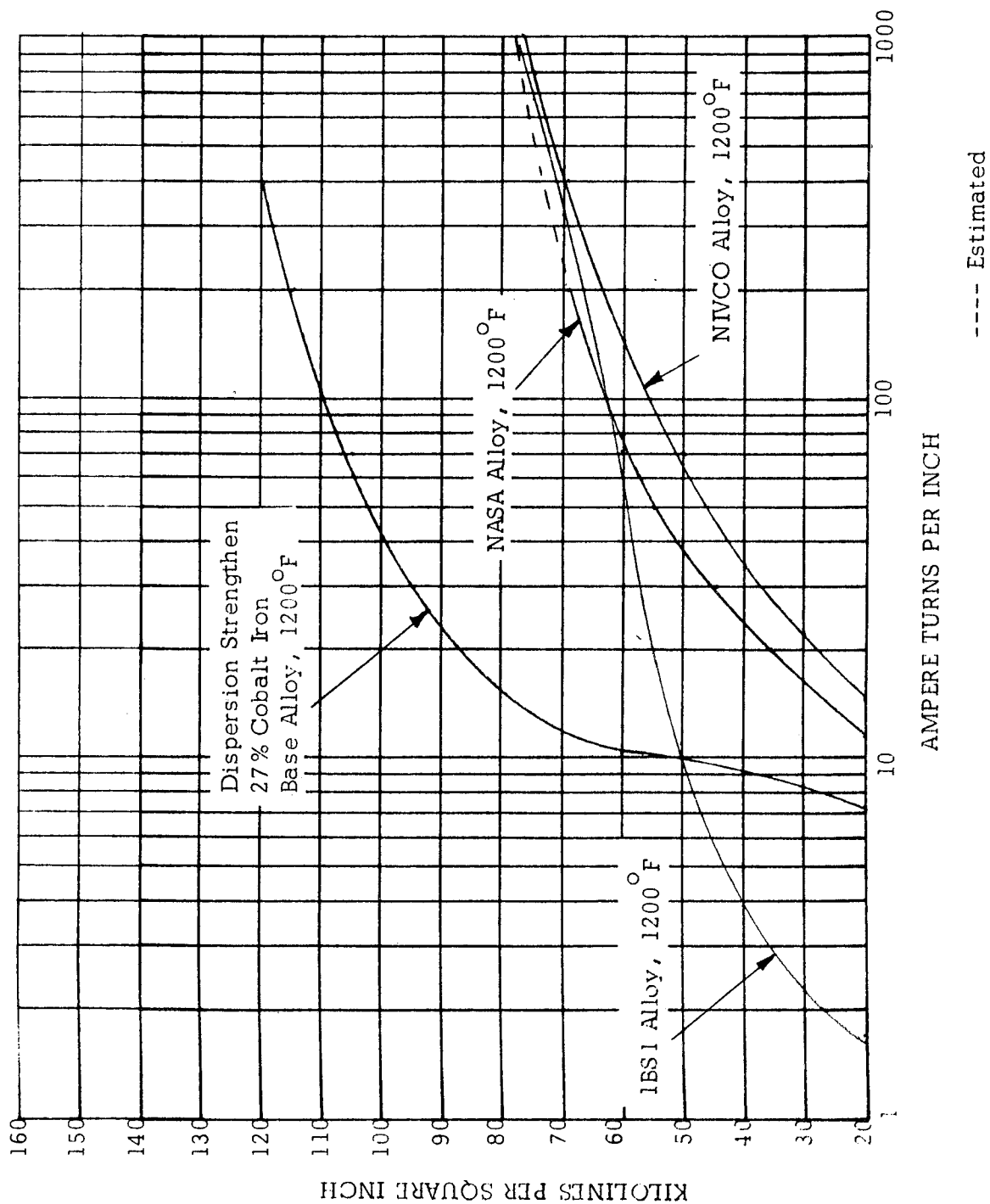


Figure 2.2-6. Magnetization Curves.

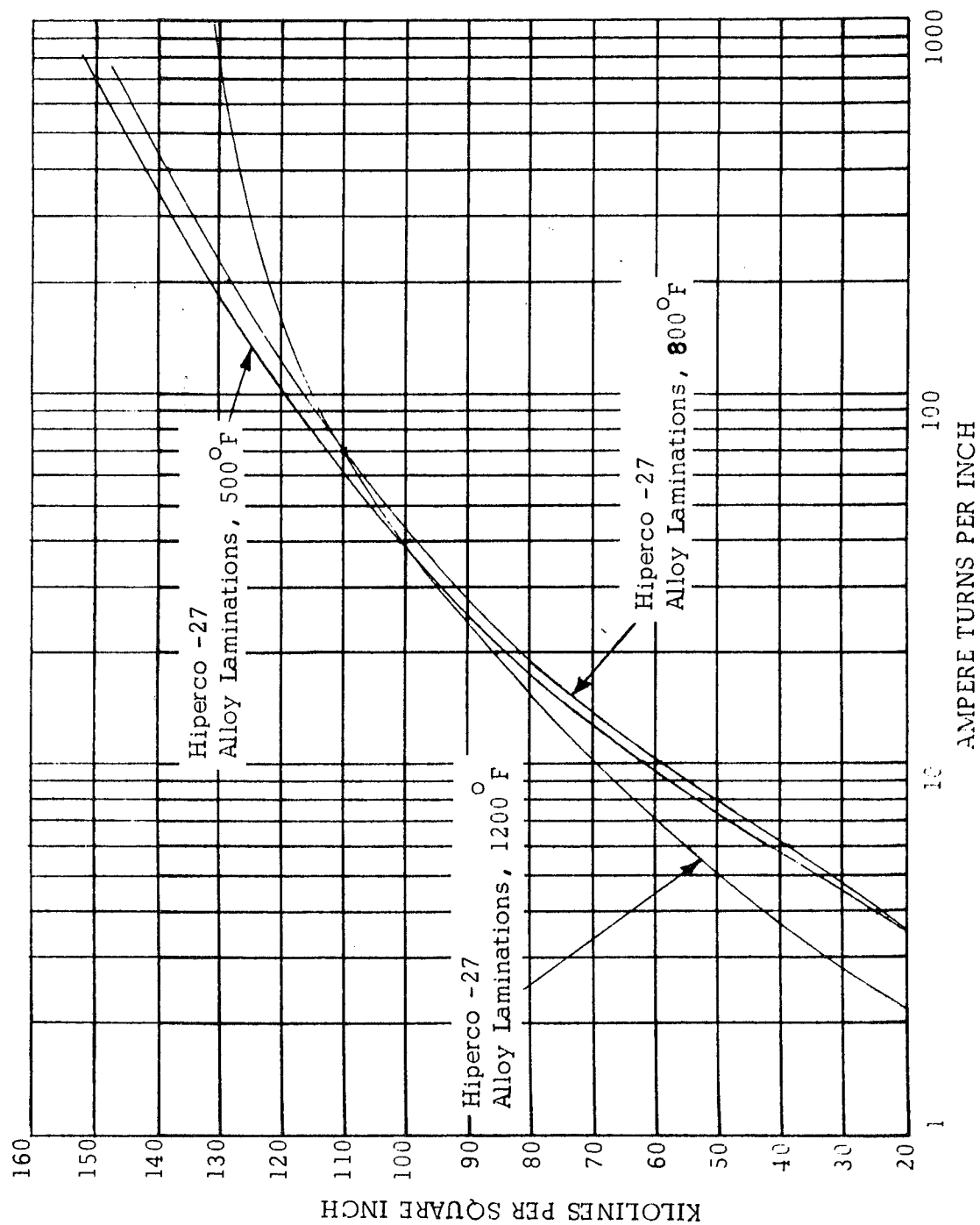


Figure 2.2-7. Magnetization Curves.

Relative to the magnetic properties which are not shown on Figure 2.2-3 an indirect comparison of their effects can be made by examining the specific weights. For instance, the first four materials have better magnetic properties and have correspondingly lower specific weights than do the last four materials. This result is not always true and it should not be taken at face value. For instance, the Hiperc-27 alloy is recognized as a superior magnetic alloy over the H-11 steel, but the specific weights are similar for the two. This is due to stator design considerations influencing the minimum rotor size. Such effects can never be known until actual designs are made and these effects serve to illustrate why comparison of magnetic and mechanical properties alone is inadequate to making the final selection of the rotor material.

An overall view of the discussion above and the distribution of the indicating bars on Figure 2.2-3 indicates there are several plateaus of temperatures and corresponding materials. The most obvious are the materials and temperatures in the 200° to 800°F range, the 900° to 1100°F range and above 1200°F. Within the 200° to 800°F range are two subtle ranges over 200° to 600°F and 600° to 800°F. As a result, the following three coolant temperatures were selected to represent designs that could cover a 200° to 1100°F range for the entire study.

- (a) 400°F - Organic coolant temperature
 - Applicable to 200° to 600°F range
 - Examined: 15% Ni Maraging Steel
 - Hiperc-27
 - H-11

- (b) 800°F - Alkali metal coolant temperature
 - Applicable to 600° to 800°F coolant
 - Examined: Hiperc-27
 - H-11 Steel
 - 1A52 Alloy
 - NIVCO

- (c) 1100^oF - Alkali metal coolant temperature
 - Applicable to 900^o to 1100^oF coolant
 - Examined: NIVCO
 - 1BS1 Alloy
 - NASA "Superalloy"
 - DS 27% Fe-Co Alloy

Results of Rotor Materials Studies. The 400^oF coolant temperature rotor materials screening study results are presented in Table 2.2-19. The three materials investigated were Hiperc-27, premium quality H-11 (AMS 6487) at Rockwell C 45 hardness, and 15 percent maraging steel. All of these rotor alloys have sufficient material strength at the 400^oF coolant temperature for operation at a rotor speed of 16,000 rpm. The Hiperc-27 and H-11 rotor materials gave generator designs with the best combination of weight and efficiency. The Hiperc-27 and H-11 rotor material designs were also superior to the 15 percent maraging steel design with respect to required field power and rotor losses. Both the Hiperc-27 alloy and H-11 rotor materials were considered candidate rotor materials; therefore, both materials were included in the 400^oF coolant temperature speed studies.

The two materials considered are rather diverse in their magnetic and mechanical properties. The Hiperc-27 is a superior magnetic steel with low strength properties and H-11 is considered a high strength steel with adequate magnetic properties. The trend in comparative properties for each of these steels is typical of most alloys. High strength alloys tend to have low magnetic capability, whereas better magnetic steels can be said to bracket a host of intermediate steels which are recognized to have better magnetic properties but less strength than H-11 or better strength but worse magnetic properties than Hiperc-27.

TABLE 2.2-19

ROTOR MATERIALS STUDY

Conditions: 733 kVA, 16,000 rpm, 1600 Hz, 480 volts (L-N) 400°F coolant,
 120° phase belts, 126 slots, J = 7000 ams/in²,
 Radial Gap = 0.14 inch

Rotor Material	Gen Wt. (lbs)	Gen Eff (%)	Rotor Weight (lbs)	Percent Voltage Unbalance	Field Current Ratios			Field Power (watts)	X _d P.U.
					Overload	3 ϕ Short	Overvoltage		
Hiperco-27	769	91.9	290	7.35	1.08	1.27	1.15	3341	0.79
H-11	815	91.8	341	6.51	1.07	1.27	1.12	4189	0.73
15% Maraging	828	91.2	351	6.41	1.06	1.07	1.15	6384	0.72

The bracketing was done in an attempt to illustrate any "gains" or "losses" that might be associated with intermediate materials without actually having to look at them in detail. As seen by the small differences in between the H-11 and Hiperco-27 designs, the bracketing appears justified. It can be concluded then, that if a Hiperco-27 rotor material design is overstressed and in need of a higher strength material, then that material might as well be H-11 rather than a lesser known intermediate.

The reasons for the small differences are several, but at the risk of oversimplifying, it may be stated that the rotor diameter can only become so small and at that point its minimum diameter is more dependent on stator design considerations. Thus, using an improved magnetic alloy beyond this point to shrink the rotor further results in little benefit with regard to generator weight. Field power is somewhat less, however. It was found during the 400⁰F speed screening studies that the best alternator designs with Hiperco-27 rotor material occurred when the flux density in the rotor was less than what the material was capable of. In other words, within the design guideline limits of this study, the Hiperco-27 rotor had better magnetic properties than was required to obtain a balanced magnetic design throughout the alternator.

Rotor materials study results at the 800⁰F coolant condition are tabulated in Table 2.2-20. Four materials were examined, two of which could be called out as candidates, H-11 alloy and 1AS2 alloy. The 1AS2 alloy is comparable to H-11 so far as generator size and efficiency goes but shows more potential so far as overspeed, overload (to saturation), and long-term stability is concerned. The 1AS2 alloy is still in the developmental stage however with only rudimentary

TABLE 2.2-20

SUMMARY OF ROTOR MATERIALS STUDIES, 800°F COOLANT

Conditions: 733 kVA, 16000 rpm, 1600 Hz, 480 volts (L-N), 120° phase belts,
 4 conductors per slot, 144 slots, J = 6000 amps/in², radial
 gap - 0.14 inch

Rotor Material	Electrical Weight lbs.	Rotor Weight lbs.	Efficiency %	Voltage Unbalance %	Field Current Ratios			Field Power Watts	Xd P.U.
					Overload	3 ϕ Short	Overvoltage		
H-11	829	347	91.8	9.53	1.12	1.54	1.14	5300	1.06
NIVCO	926	457	91.3	8.26	1.11	1.28	1.19	6700	0.96
1AS2	838	353	91.7	9.50	1.12	1.51	1.11	5900	1.06
Hiperco-27	818	299	91.2	11.75	1.20	1.58	1.15	5028	1.23

properties known compared to the extensive property data available for H-11. Thus, H-11 was the present choice. The fourth material, Hiperco-27, failed stress-wise even at the low 16,000 rpm speed. It was mainly examined for academic purposes to see the effect of minimizing the rotor size with a good magnetic material and thereby bracket the sizes between those obtainable with Hiperco-27 and those of higher strength and lesser magnetic properties.

It may appear that the choice of H-11 was based on a very limited comparison. Actually, the initial selection of H-11 is based on past studies (2.2-7) where H-11 was compared with 5 other materials having greater strengths and lower magnetic properties and/or vice versa. Literally speaking, then, it is the choice of nine possible candidates about the 800⁰F coolant temperature range.

Because of the critical nature of rotor materials selection to the final design results, the choice of H-11 should by no means be considered the ultimate. It is the practice of the Westinghouse Aerospace Electrical Division to continually search out new candidates. For example, H-11 was chosen as a better candidate than the original AISI 4340 envisioned for the SNAP-50/SPUR generator (2.2-11). To date, the choice of H-11 still appears to be the best with the 1AS2 alloy being a prime candidate for potential design improvement.

Rotor material screening studies for the 1100⁰F coolant temperature are summarized in Table 2.2-21. Of the four materials investigated, only the experimental alloy 1BS1 could be recognized as a candidate rotor material. Material strength data for the "NASA Alloy" was not available; therefore, its suitability as a candidate rotor material could not be determined. The creep strength of the NIVCO and dispersion

TABLE 2.2-21

ROTOR MATERIALS STUDY

Conditions: 733 kVA, 16000 rpm, 1600 Hz, 480 volts (L-N), 21100° F coolant
120° phase belts, 144 slots, J = 6000 amps/in
Radial gap = 0.14 inch

<u>Rotor Material</u>	<u>Electrical Weight (lbs)</u>	<u>Rotor Weight (lbs)</u>	<u>Efficiency %</u>	<u>% Voltage Unbalance</u>	<u>Field Current Ratios</u>		
					<u>Overload</u>	<u>3φ Short</u>	<u>Overvoltages</u>
1BS1	905	457	92.2	9.06	1.22	1.34	1.37
NIVCO	947	499	91.8	8.54	1.15	1.27	1.25
DS27CO	826	333	91.8	11.35	1.19	1.65	1.21
NASA Alloy	945	487	92.3	9.19	1.17	1.43	1.26
<u>Rotor</u>	<u>Field Power Watts</u>	<u>Xd</u>					
<u>Material</u>		<u>P.U.</u>					
1BS1	7409	1.03					
NIVCO	8869	.93					
DS27CO	5913	1.21					
NASA Alloy	7092	1.05					

strengthened 27% cobalt alloys were less than the induced rotor stress at 16,000 rpm which eliminated them from further consideration. The 1BS1 alloy is currently in the developmental stage; therefore, its use as a design material in the KTA application is not in accordance with the design guideline priority of ease of development.

Comparative stresses not shown on the tables, were used to determine the applicability of the material to speed and strength limitations. Typical stress plots for the rotors and materials selected are presented in Section 2.2.

Based on the results of the above alternator screening analyses, the later stress analyses and consultations with magnetic materials experts, the choices of rotor materials for the three coolant temperatures were made. These are summarized in Table 2.2-22 along with brief statements of the primary considerations leading to the choices.

System Design Variable Screening Studies

The objective of the system design variable screening studies was to find the optimum combination of alternator frequency, speed, and voltage for each of the three specified coolant temperatures on the basis of meeting the specification requirements and of achieving the primary design objectives listed below with priorities ranked in descending order.

- (1) High reliability
- (2) Ease of development
- (3) Maximum energy conversion efficiency
- (4) Minimum weight

TABLE 2.2-22

ROTOR MATERIALS CHOICES

800° F COOLANT	H-11 - 1st Choice 1. Considerable Data and is Readily Available. 2. Still insufficient creep data for 800 F coolant designs. Require 3-year data for high reliability. 1A22 - Shows Most Promise for Improvement; Needs Development 1. Reduce Brittleness by further development 2. Large Forging Techniques undefined. 3. More Data needed for reliable design predictions
1100° F COOLANT	1BS1 - 1st Choice; Needs Development 1. Large Forging Techniques undefined 2. More Data needed for reliable design predictions. NIVCO - Considered Possible Alternate 1. Almost as good as 1BS1 type 2. Must be considered only available material for 900 to 1100° F coolant temperature range.
400° F COOLANT	Either H-11 or Hiperco-27 - 1st choice 1. Both showed applicability. 2. Carried both on system speed and frequency studies 3. They bracket host of intermediate candidates.

The range and specific values for the system design variables are set forth below.

Speed, RPM	16,000 to 32,000
Frequency, Hz	800, 1200, 1600, 2000
Voltage (L-N)	120, 240, 480, 720, 960, 1200

The speeds listed below correspond to the specified frequencies and integral pole pairs.

<u>Hz</u>	<u>Poles</u>	<u>Speeds</u>
800	4, 6	24,000, 16,000
1200	6, 8	24,000, 18,000
1600	6, 8, 10, 12	32,000, 24,000, 19,200, 16,000
2000	8, 10, 12, 14	30,000, 24,000, 20,000, 17,143

The parametric data relating speed, frequency, and voltage to electrical alternator weight and efficiency were determined in an iterative cycling procedure whereby (1) frequency is varied at a fixed speed and voltage, (2) speed is varied at a fixed frequency(s) and voltage, and (3) voltage is varied at a fixed speed and frequency. The fixed values were initially selected according to optimum frequency values determined on Contract AT(04-3)-679 (2.2-1) and in the case of the frequency study, the requirement for a common speed for all frequencies. As the cycling proceeded, new values were chosen from data generated in the studies.

The data from the studies is presented in tabulated form with reference to the appropriate tables stated in the text. Presentation of primary data in curve form for the frequency, speed, and voltage studies is given in the Appendix.

Information Pertinent to Design Selection from the Standpoint of Electromagnetic Design Characteristics. The frequency and speed studies for the 400^oF organic coolant are summarized in Tables 2.2-23 and 2.2-24, respectively. A frequency of 1600 Hz results in minimum system weight while providing a good compromise between armature hot-spot temperature, rotor pole face loss, and field power. Therefore, 1600 Hz is the primary choice for frequency. Based on the same criteria, slight differences make 2000 Hz a secondary choice.

Four speed studies were required for the 400^oF organic coolant to consider both candidate rotor materials, i.e., Hiperco-27 and premium quality H-11 (AMS 6487) at frequencies of 1600 and 2000 Hz. Results of the speed studies are shown in Table 2.2-24. The preferred combination of speed, frequency and rotor material for the 400^oF coolant temperature, based on generator electromagnetic design considerations and minimum system weight is 24,000 rpm at a frequency of 1600 Hz for either a Hiperco-27 or H-11 rotor material. The Hiperco-27 generator design is slightly lighter (15 pounds) but does not have as large a stress safety factor as the H-11 design. A discussion of rotor stress considerations is presented in the next section. The second alternate choice is 30,000 rpm, 2000 Hz, H-11 rotor material and the third alternate choice is 32,000 rpm, 1600 Hz, H-11 rotor material.

The 30,000 rpm, 2000 Hz, Hiperco-27 rotor material design is obviously overstressed and therefore was excluded from electrical-mechanical design preference consideration. The 16,000, 17,143, and 20,000 rpm designs have the undesirable four conductors per slot winding configuration; therefore, these design points would be the least desirable in this application. A summary of design points listed

TABLE 2.2-23

SUMMARY OF FREQUENCY STUDY - 400° F Coolant

Conditions: 733 kVA, 24000 rpm, 480 volts (L-N), 120° phase belts,
J = 7000 amps/in², Hiperco-27 Rotor, Radial Gap = 0.14 inch

Frequency (Hz)	Gen Wt. (lbs)	Gen Eff (%)	Field Power (watts)	Percent Voltage Unbalance	Armature Hot Spot (° F)	Conductors Per Slot	Rotor Weight (lbs)	Field Current Ratios	
								3 ϕ Short	Overload
800	933	87.5	4170	8.51	1263	2	243	1.61	1.13
1200	748	91.7	3070	6.85	1033	2	242	1.38	1.09
1600	626	91.6	2904	7.97	928	2	207	1.43	1.10
2000	661	91.4	2489	5.52	851	2	236	1.10	1.05

TABLE 2.2-24

SUMMARY OF SPEED STUDIES - 400° F COOLANT

Conditions: 733 kVA, 480 volts (L-N), 120° Phase Belts, J = 7000 amps/in²
 Radial Gap - 0.14 inch

Speed (RPM)	Frequency (Hz)	Rotor Material	Gen Wt. (lbs)	Gen Eff (%)	Percent Voltage Unbalance	Armature Hot Spot (°F)	Conductors per Slot	Rotor Wt. (lbs)	Field Current Ratios		x _d P.V.
									3 φ Short	Overload	
16,000	1600	Hiperco 27	860	91.9	5.76	826	4	340	1.13	1.06	0.62
19,200	1600	Hiperco 27	762	91.7	6.42	853	2	290	1.26	1.07	0.72
24,000	1600	Hiperco 27	626	91.6	7.97	928	2	207	1.43	1.10	0.91
32,000	1600	Hiperco 27	552	91.3	7.10	1053	2	171	1.42	1.10	0.87
16,000	1600	H-11	731	91.9	8.60	948	4	288	1.44	1.10	0.95
19,200	1600	H-11	777	91.7	6.23	862	2	305	1.24	1.06	0.71
24,000	1600	H-11	643	91.8	7.09	943	2	235	1.38	1.09	0.85
32,000	1600	H-11	533	91.0	9.00	1108	2	180	1.63	1.13	1.13
17,143	2000	Hiperco 27	767	91.5	5.70	808	4	304	1.08	1.05	0.60
20,000	2000	Hiperco 27	664	91.7	6.06	849	4	253	1.15	1.06	0.66
24,000	2000	Hiperco 27	661	91.4	5.52	851	2	236	1.10	1.05	0.62
30,000	2000	Hiperco 27	546	91.5	6.22	914	2	182	1.26	1.07	0.73
17,143	2000	H-11	688	90.8	8.99	918	4	277	1.41	1.11	0.95
20,000	2000	H-11	720	91.2	5.54	850	4	306	1.17	1.05	0.62
24,000	2000	H-11	619	90.9	6.52	889	2	252	1.32	1.07	0.75
30,000	2000	H-11	505	90.7	7.58	975	2	182	1.44	1.09	0.90

in order of preference based on electromagnetic design considerations is presented in Table 2.2-25.

Results of the frequency study for a coolant temperature of 800°F are shown in Table 2.2-26. A frequency of 1600 Hz gave the best combination of generator weight and efficiency and was the primary choice. A speed study was run at both 1600 Hz and 2000 Hz to determine if an advantageous combination of speed and frequency might occur for the 2000 Hz frequency over that of the 1600 Hz frequency and also to cover the intermediate speed points of 17,143 rpm, 20,000 rpm, and 30,000 rpm. Speed points for the 1600 Hz frequency are 16,000 rpm, 19,200 rpm, 24,000 rpm, and 32,000 rpm.

Results of the speed study at a frequency of 1600 Hz is given in Table 2.2-27. Very little weight or efficiency difference exists between the 16,000 rpm and 19,200 rpm designs. Also the armature hot-spot temperatures are very close. Voltage unbalance (for a 2/3 single phase load) for the 16,000 rpm design exceeds the preliminary guideline limit of 9 percent whereas the 19,200 rpm design is well below this limit. The primary reason for this is that the 16,000 rpm design has four conductors per slot (i.e., two turns per coil) whereas the 19,200 rpm and higher speed designs have two conductors per slot (i.e., one turn per coil). Also, the direct axis synchronous reactance "X_d" is higher (1.06) for the 16,000 rpm design than for the 19,200 rpm design which has an "X_d" of 0.77. As a general rule, generator weight decreases chosen for the 16,000 rpm speed, its weight would be somewhat over 1000 pounds, efficiency about 91.9 percent, and voltage unbalance about 6 percent. Four conductor per slot designs investigated for speeds of 19,200 rpm and higher were inferior because

TABLE 2.2-25

SELECTED SPEEDS AND FREQUENCIES - 400° F COOLANT

Order of Electro- Magnetic Design Preference	Speed (rpm)	Freq. Hz	Rotor Material	Elect. Wt. (lbs.)	Eff. %	Conductors Per Slot	Voltage Unbalance %	Rotor Radial $\Delta T(^{\circ}\text{F})^*$	Armature Hot Spot Temp. ($^{\circ}\text{F}$)	Rotor Wt. (lbs.)
1	24000	1600	H-11	643	91.8	2	7.09	95	943	235
2	24000	1600	Hiperco 27	626	91.6	2	7.97	39	928	207
3	19200	1600	Hiperco 27	762	91.7	2	6.42	26	853	290
4	19200	1600	H-11	777	91.7	2	6.23	44	862	305
5	30000	2000	H-11	505	90.7	2	7.58	107	975	182
6	32000	1600	H-11	533	91.0	2	9.00	356	1108	180
7	20000	2000	Hiperco 27	664	91.5	4	6.06	113	849	253

* Temperature difference between calculated pole tip temperature and coolant temperature in rotor bore.

TABLE 2.2-26
FREQUENCY STUDY - 800° F COOLANT

<u>kVA</u>	<u>Freq. (Hz)</u>	<u>Speed (rpm)</u>	<u>Gen. Wt. (lbs)</u>	<u>Gen. Eff. (%)</u>	<u>Approx. Rotor Stress (psi)</u>	<u>Rotor Radial ΔT (°F)</u>	<u>Stator Hot-Spot Temp. (°F)</u>	<u>Conductors Per Slot</u>	<u>Volt Unbal. (%)</u>	<u>Xd P.U.</u>
733	800	24000	1079	84.0	88096		1288	2	8.34	1.09
733	1200	24000	758	91.2	79510	364	1269	2	8.90	1.11
733	1600	24000	701	91.5	78498	95	1189	2	7.78	0.93
733	2000	24000	676	90.6	77824	50	1161	2	7.32	0.83

TABLE 2.2-27

SPEED STUDY - 800° F COOLANT - 1600 Hz

kVA	Freq. (Hz)	Speed (rpm)	Gen. Wt (lbs)	Gen. Eff. (%)	Approx.			Stator Hot-Spot Temp. (°F)	Conductors Per Slot	Volt. Unbal. (%)	xd P. U.
					Ave.	Rotor Stress (psi)	Rotor Radial ΔT (°F)				
733	1600	16000	829	91.8	38728		81	1164	4	9.53	1.06
733	1600	19200	844	91.7	55520		45	1161	2	6.77	0.77
733	1600	24000	701	91.6	78498		95	1189	2	7.78	0.93
733	1600	32000	573	90.4	124344		362	1281	2	8.47	1.19

of lower generator efficiency, higher rotor stress and rotor losses. Also, a two conductor per slot design is preferable from the standpoint of ease of winding development with the stiff clad silver conductor. Because of the basic simplicity of a one turn coil winding with 120° phase belts, no end extension joints with clad silver conductors are required.

The 24,000 rpm speed design provides the best combination of generator weight and efficiency of all the speeds investigated at a frequency of 1600 Hz and meets other electrical system requirements regarding voltage unbalance, three phase short circuit, and overload. The 32,000 rpm speed design has the lowest generator weight but suffers the disadvantages of lower generator efficiency, high armature hot-spot temperature, higher rotor losses, and a voltage unbalance exceeding the 9 percent guideline limit. Temporarily disregarding any rotor stress limitations, the 24,000 rpm design would be the preferred design at a frequency of 1600 Hz (based on generator electromagnetic design considerations) with the 19,200 rpm design the second choice.

Results of the speed study at a frequency of 2000 Hz are given in Table 2.2-28. The designs at 17,143 rpm and 20,000 rpm are four conductor per slot designs and the 24,000 rpm and 30,000 rpm designs are two conductor per slot designs. This results from selection of the best design of the two configurations at each speed without greatly exceeding the 9 percent voltage unbalance guideline. Again, temporarily neglecting any rotor stress considerations, the 30,000 rpm design would be selected as the best generator electromagnetic design at a frequency of 2000 Hz, with the 24,000 rpm design the second choice and the 20,000 rpm design the third choice.

TABLE 2.2-28
SPEED STUDY - 800° F COOLANT - 2000 Hz

<u>kVA</u>	<u>Freq. (Hz)</u>	<u>Speed (rpm)</u>	<u>Gen. Wt. (lbs.)</u>	<u>Gen. Eff. (%)</u>	<u>Approx. Ave. Rotor Stress (psi)</u>	<u>Rotor Radial ΔT (°F)</u>	<u>Stator Hot-Spot Temp. (°F)</u>	<u>Conductors Per Slot</u>	<u>Volt. Unbal. (%)</u>	<u>Xd P. U.</u>
733	2000	17143	753	90.8	44064	54	1186	4	9.63	1.03
733	2000	20000	761	91.0	62651	383	1148	4	5.78	0.66
733	2000	24000	676	90.6	77821	50	1157	2	7.32	0.83
733	2000	30000	566	90.4	110676	106	1203	2	8.32	0.99

The selected speed points for the two frequencies are summarized in Table 2.2-29. It must be remembered that the selections are based on generator electromagnetic design and that stress limits were ignored in these choices. Stress limitations are covered in the next section of this report. If speeds greater than 24,000 rpm are excluded from consideration, the order of preference would be as shown in Table 2.2-30.

Grooved solid pole faces were assumed for the above designs. It should be noted here that there is some question regarding the accuracy of the load harmonic component of pole face losses as calculated by Barello's method. The calculation for the other component of pole face loss (tooth ripple) is considered reliable because the applicable value of permeability is used. Barello uses a constant value of permeability equal to 2000 in his equation which may not be applicable to high strength steels such as H-11 alloy. The use of laminated pole faces joined to the pole body would reduce the calculated tooth ripple component of pole face loss and assure that the calculated load harmonic component is not greater than the presently calculated value. The use of laminated pole tips joined to the pole body is considered feasible but is contrary to the ease of development design requirements.

Below a speed of 24,000 rpm, the 19,200 rpm, 1600 Hz design is preferred over the 20,000 rpm, 2000 Hz design not only because of higher efficiency but also because the winding is a two conductor per slot winding whereas the 20,000 rpm, 2000 Hz design is a four conductor per slot winding. Ease of winding development and reliability are greatly improved with the two conductor per slot winding because of the simplicity of this configuration and with the use of 120° phase belts, no crossover or other conductor joints, other than to the terminal or

TABLE 2.2-29
SELECTED SPEEDS AND FREQUENCIES - 800° F COOLANT, ALL SPEEDS

Order of E.M. Design Preference	kVA	Speed (rpm)	Freq. (Hz)	Gen.		Conductors		Volt. Unbal. (%)	Rotor		Stator	
				Wt. (lbs)	Eff. (%)	Per	Slot		Radial ΔT (%)	Hot-Spot Temp. (°F)	Wt. (lbs)	
1	733	24000	1600	701	91.6	2		7.78	95	1189	274	
2	733	30000	2000	566	90.4	2		8.32	106	1203	214	
3	733	24000	2000	676	90.6	2		7.32	50	1157	275	
4	733	19200	1600	844	91.7	2		6.77	45	1161	353	
5	733	20000	2000	761	91.0	4		5.78	383	1148	336	

TABLE 2.2-30
SELECTED SPEEDS AND FREQUENCIES - 800°F COOLANT, LIMITED SPEED

<u>Order of E.M. Design Preference</u>	<u>kVA</u>	<u>Speed (Rpm)</u>	<u>Freq. (Hz)</u>	<u>Gen. Wt. (lbs)</u>	<u>Gen. Eff. (%)</u>	<u>Conductors Per Slot</u>	<u>Volt. Unbal. (%)</u>	<u>Rotor Radial ΔT (%)</u>	<u>Stator Hot-Spot Temp. (°F)</u>	<u>Rotor Wt. (lbs)</u>
1	733	24000	1600	701	91.6	2	7.78	95	1189	274
2	733	24000	2000	676	90.6	2	7.32	50	1157	275
3	733	19200	1600	844	91.7	2	6.77	45	1161	353
4	733	20000	2000	761	91.0	4	5.78	383	1148	336

or bus bar, are required. Also, the two conductors in the slot are separated by phase insulation whereas with four conductors per slot, two conductors lay together separated by conductor insulation only. It is possible to form a four conductor per slot winding without making joints but experience with certain aircraft generator designs indicates voltage dielectric breakdown problems can become excessive with this configuration.

Results of the frequency study for a coolant temperature of 1100°F are shown in Table 2.2-31. A frequency of 1600 Hz gave the best combination of generator weight and efficiency with 2000 Hz a close second choice. Speed studies were run for both 1600 Hz and 2000 Hz frequencies as was done for the 800°F studies. Windage losses are not included in the 1100°F coolant temperature efficiency values (they are included in the 800°F coolant temperature efficiency values) because their magnitudes (approximately 40 kW at 16,000 rpm) were so high as to reduce other generator losses to secondary importance. The high loss is caused by the high saturated vapor pressure of potassium at 1100°F coolant temperature. Results of the speed studies are given in Tables 2.2-32 and 2.2-33.

Designs were selected using the same considerations as applied to the 800°F coolant temperature design selections. Considering generator electromagnetic design factors only and temporarily neglecting rotor stress limitations, the speed and frequency combinations in order of preference are given in Table 2.2-34.

If speeds greater than 19,200 rpm are excluded from consideration, the order of preference would be as shown in Table 2.2-35.

TABLE 2.2-31
FREQUENCY STUDY - 1100°F COOLANT

kVA	Freq. (Hz)	Speed (rpm)	Gen. Wt. (lbs)	Gen. Eff. (%)	Approx. Ave. Rotor Stress (psi)	Rotor Radial ΔT (°F)	Stator Hot-Spot Temp. (°F)	Conductors Per Slot	Volt. Unbal. (%)	Xd P.U.
733	800	24000	1200	71.7	126893	207	1467	2	7.60	1.03
733	1200	24000	836	91.3	105509	242	1486	2	9.01	1.14
733	1600	24000	770	92.7	104168	1025	1434	2	7.68	0.94
733	2000	24000	749	92.3	103096		1413	2	7.07	0.83

TABLE 2.2-32

SPEED STUDY - 1100°F COOLANT - 1600 Hz

kVA	Freq. (Hz)	Speed (rpm)	Gen. Wt. (lbs)	Gen. Eff. (%)	Approx. Ave. Rotor Stress (psi)	Rotor Radial ΔT (°F)	Stator Hot-Spot Temp. (°F)	Conductors Per Slot	Volt Unbal. (%)	Xd P. U.
733	1600	16000	905	92.2	52317	112	1431	4	9.06	1.03
733	1600	19200	935	92.7	74256	973	1411	2	6.57	0.77
733	1600	24000	770	92.7	104168	207	1434	2	7.68	0.94
733	1600	32000	627	91.4	163833	231	1492	2	9.57	1.22

TABLE 2.2-33
SPEED STUDY - 1100° F COOLANT - 2000 Hz

<u>kVA</u>	<u>Freq. (Hz)</u>	<u>Speed (rpm)</u>	<u>Gen. Wt. (lbs)</u>	<u>Gen. Eff. (%)</u>	<u>Approx. Ave. Rotor Stress (psi)</u>	<u>Rotor Radial ΔT (°F)</u>	<u>Stator Hot-Spot Temp. (°F)</u>	<u>Conductors Per Slot</u>	<u>Volt. Unbal. (%)</u>	<u>xd P. U.</u>
733	2000	17143	868	92.0	60697	112	1447	4	8.95	0.99
733	2000	20000	837	91.0	82997	973	1416	4	5.32	0.61
733	2000	24000	749	92.3	103096	107	1413	2	7.07	0.83
733	2000	30000	622	92.4	145384	231	1438	2	8.16	1.00

TABLE 2.2-34
SELECTED SPEEDS AND FREQUENCIES - 1100° F COOLANT, ALL SPEEDS

Order of E.M. Design Preference	kVA	Speed (rpm)	Freq. (Hz)	Elect. Wt. (lbs)	Eff. (%)	Conductors		Volt. Unbal. (%)	Rotor		Armature		Rotor Wt. (lbs)
						Per	Slot		Radial ΔT (%)	Hot-Spot Temp. (°F)			
1	733	30000	2000	622	92.4	2		8.16	231	1438		287	
2	733	24000	1600	770	92.7	2		7.68	242	1434		368	
3	733	24000	2000	749	92.3	2		7.07	107	1413		368	
4	733	19200	1600	935	92.7	2		6.57	96	1411		471	
5	733	17143	2000	868	92.0	4		8.95	112	1447		450	
6	733	16000	1600	905	92.2	4		9.06	202	1431		457	

TABLE 2.2-35
SELECTED SPEEDS AND FREQUENCIES - 1100° F COOLANT

Order of E. M. Design Preference	kVA	Speed (rpm)	Freq. (Hz)	Gen.		Gen. Eff. (%)	Conductors Per Slot	Volt. Unbal. (%)	Rotor		Stator	
				Wt. (lbs)	Wt. (lbs)				Radial ΔT (°F)	Hot-Spot Temp. (°F)	Hot-Spot Temp. (°F)	Rotor Wt. (lbs)
1	733	19200	1600	935	935	92.7	2	6.57	96	1411	1411	471
2	733	17143	2000	868	868	92.0	4	8.95	112	1447	1447	450
3	733	16000	1600	905	905	92.2	4	9.06	202	1431	1431	457

As stated previously, windage losses are not included in the efficiency values for the 1100°F study because they are prohibitively high. Methods to reduce windage losses for the 1100°F coolant temperature require separate evaluation regarding system penalties and are discussed more fully in Section 2.4.

Results of the voltage studies for coolant temperatures of 400°F and 800°F are summarized in Tables 2.2-36 and 2.2-37, respectively.

The establishment of a safe working voltage for inorganically insulated magnet wire requires consideration of several important factors. These are insulation composition, material form, and operating temperature. As a result of testing and evaluation performed on NAS3-4162 and NAS3-6465, magnet wire insulation based upon served glass fiber with additional inorganic fillers and glassy binders has been selected as best for operation at temperatures of 900°F to 1300°F. This insulation is known as Anadur and is produced by Anaconda Wire and Cable Company. The fully processed form of this insulation does not achieve complete density, but is porous; therefore, a designer must consider that the insulation serves as spacing rather than a complete insulating barrier. The glass-based binders in this composition begin to lose electrical insulation resistance at 1000°F and are very poor insulators above 1300°F. Thus, the electrical stresses of voltage potential must be kept low if high current losses are to be avoided. The total electric strength of Anadur, 0.0068-inch thick, at 1100°F is approximately 400 volts; this results in a typical value of about 60 volts per mil. This form of insulation must be used with supplemental ground and phase insulation composed of high purity aluminum oxide. For efficient utilization of available conductor and insulation volume in a magnetic core, the ceramic supplemental

TABLE 2.2-36

SUMMARY OF VOLTAGE STUDY - 400° F COOLANT

Conditions: 733 kVA, 1600 Hz, 24000 RPM, H-11 Rotor Material

∞	Voltage (L-N)	Gen Wt. (lbs)	Gen Eff (%)	Voltage Unbalance (%)	Armature Hot Spot (°F)	Conductors Per Slot	Xd P.U.	Rotor Radial ΔT (°F)	Field Current 3 ∅ Short	Ratios Overload
	120	544	83.4	10.10	1234'	2	1.19	> 400	3.0	1.14
	240	646	88.7	6.61	1053	2	0.79	> 400	1.36	1.08
	480	643	91.8	7.09	943	2	0.85	95	1.39	1.08
	720	598	91.7	8.43	1047	4	1.01	194	1.54	1.11
	960	659	91.4	7.27	1027	4	0.87	84	1.40	1.09
	1200	658	91.3	7.42	1031	6	0.89	136	1.40	1.09

TABLE 2.2-37

SUMMARY OF VOLTAGE STUDY - 800°F COOLANT

Conditions: 733 KVA, 1600 Hz, 19200 RPM, H-11 Rotor Material

Voltage (L-N)	Gen Wt. (lbs)	Gen Eff. (%)	Voltage Unbalance (%)	Armature Hot Spot (°F)	Conductors Per Slot	Xd P. U.	Rotor Radial ΔT (°F)	Field Current 3 ϕ Short	Ratios Overload
120	693	85.8	9.76	1296	2	1.12	>850	1.63	1.13
240	849	90.3	6.29	1161	2	0.73	848	1.27	1.07
480	837	91.6	6.75	1161	2	0.77	45	1.30	1.07
720	783	92.0	7.96	1180	4	0.92	76	1.44	1.09
960	829	91.7	6.93	1164	6	0.80	193	1.33	1.08
1200	823	91.6	7.91	1184	8	0.91	126	1.43	1.09

insulation must be of minimum thickness consistent with production and winding capabilities. The electric strength of aluminum oxide at 1100°F varies from about 60 to 150 volts per mil. For high purity alumina, about 0.018-inch thick, typical electric strength is about 100 volts per mil at 1100°F.

Although the Anadur insulation was selected for use in the screening studies (for the sake of consistency between data) an actual application would better utilize some of the lower temperature insulations recommended in Reference 2.2-6.

In order to assure a margin of safety to compensate for insulation defects created in processing, mechanical shock in operation, and transient over-voltages, an approximate operating limit of 500 volts at temperature is recommended. Electrical current leakage with accompanying excessive losses may result if higher operational voltage level is attempted.

The selected voltage for the 400°F and 800°F coolant temperature is 480 volts (L-N) based on the considerations discussed above and the combination of generator weight, efficiency, conductors per slot, and rotor temperature rise (i.e., from rotor coolant passage to the pole faces) from the parametric studies. A two conductor per slot winding configuration is considered desirable with regard to ease of development and reliability of the armature winding.

The voltage study for 1100°F coolant was held pending a decision whether to further pursue an 1100°F cooled alternator design in Phase I. The decision was made to delete the 1100°F cooled/lubricated alternator base design from further consideration and the 1100°F coolant condition voltage studies were not performed.

Information Pertinent to Design Selection from the Standpoint of Rotor Stresses. Two rotor materials were selected for further evaluation as a result of the 400⁰F rotor materials screening studies; they were H-11 steel and Hiperco-27 steel. The H-11 steel was selected over 15% Ni-maraging steel as a possible high speed rotor material. The Hiperco-27 steel was selected to show the maximum "gains" that could be achieved with a very good magnetic rotor steel recognizing that it may not work in an actual high-speed design. As seen by comparing the designs at equivalent conditions in the speed and frequency studies, the Hiperco-27 offers only slight improvements in weight and efficiency over those with H-11 steel.

The stress limitations which these materials impose upon a high speed rotor are summarized in Figures 2.2-8 and 2.2-9. These limitations are given in terms of the allowable stress intensity at design speed and were based upon the considerations given in Section 2.2.

The properties for Hiperco-27 steel were taken from NASA-CR-54091 (2.2-8). The properties for H-11 steel were taken from NASA-CR-54091 and AFAPL-TR-66-127 (2.2-7). Since H-11 is known to undergo aging, the H-11 strengths shown can be expected to be less at the end of three years.

The calculated rotor stress is the maximum combined (or Von Mises) stress that occurs under the pole on a plane (r-z direction) through its center; this is the most crucial and usually the largest stress to be considered in the alternator design, see Figures 2.2-10 to 2.2-15. The Von Mises stress relationship is used to relate the biaxial stresses occurring in the rotor to the uniaxial, maximum allowable material stress.

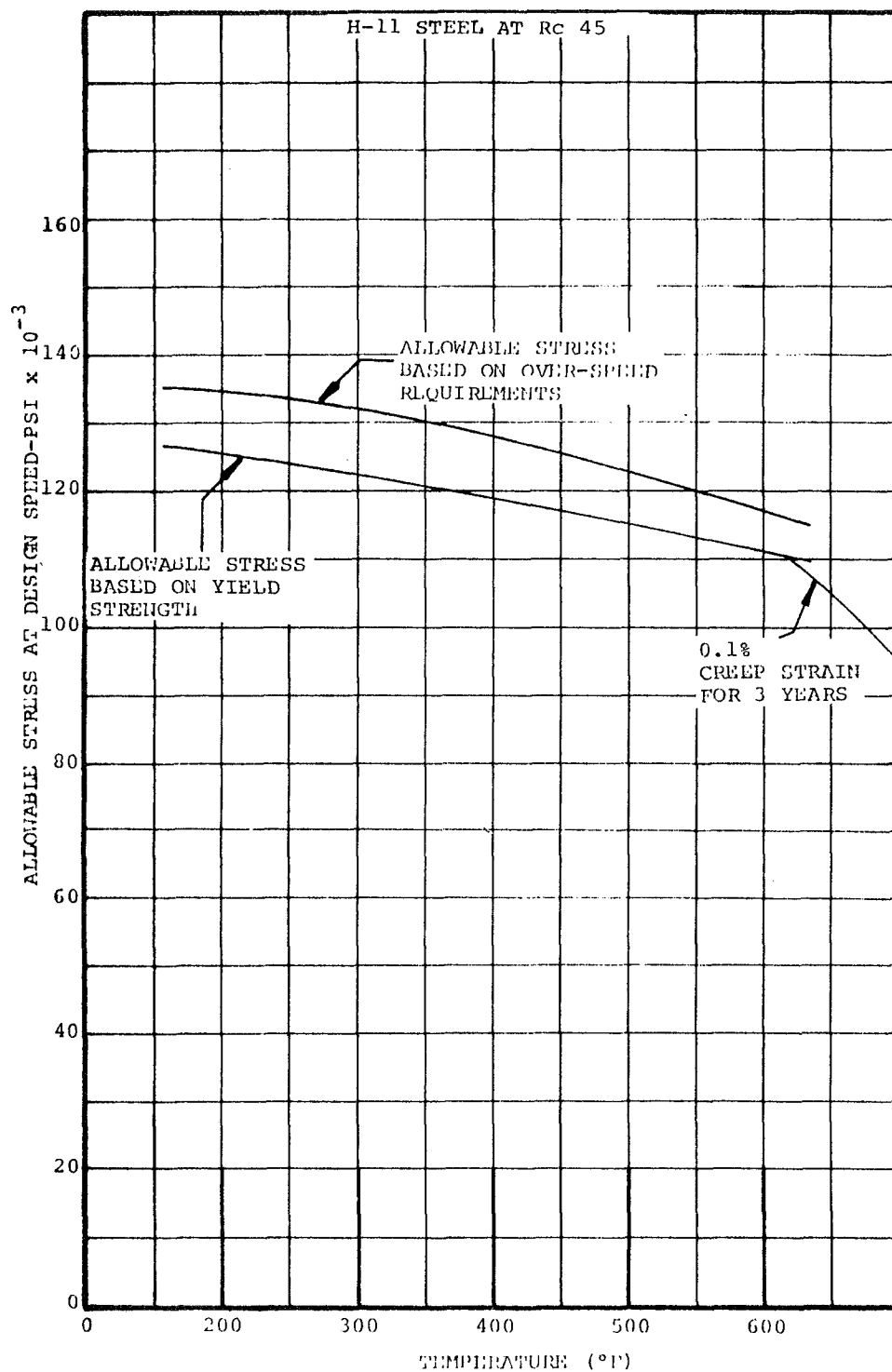


Figure 2.2-8. Interpretation of H-11 Steel Property Data at a Value Rockwell C45.

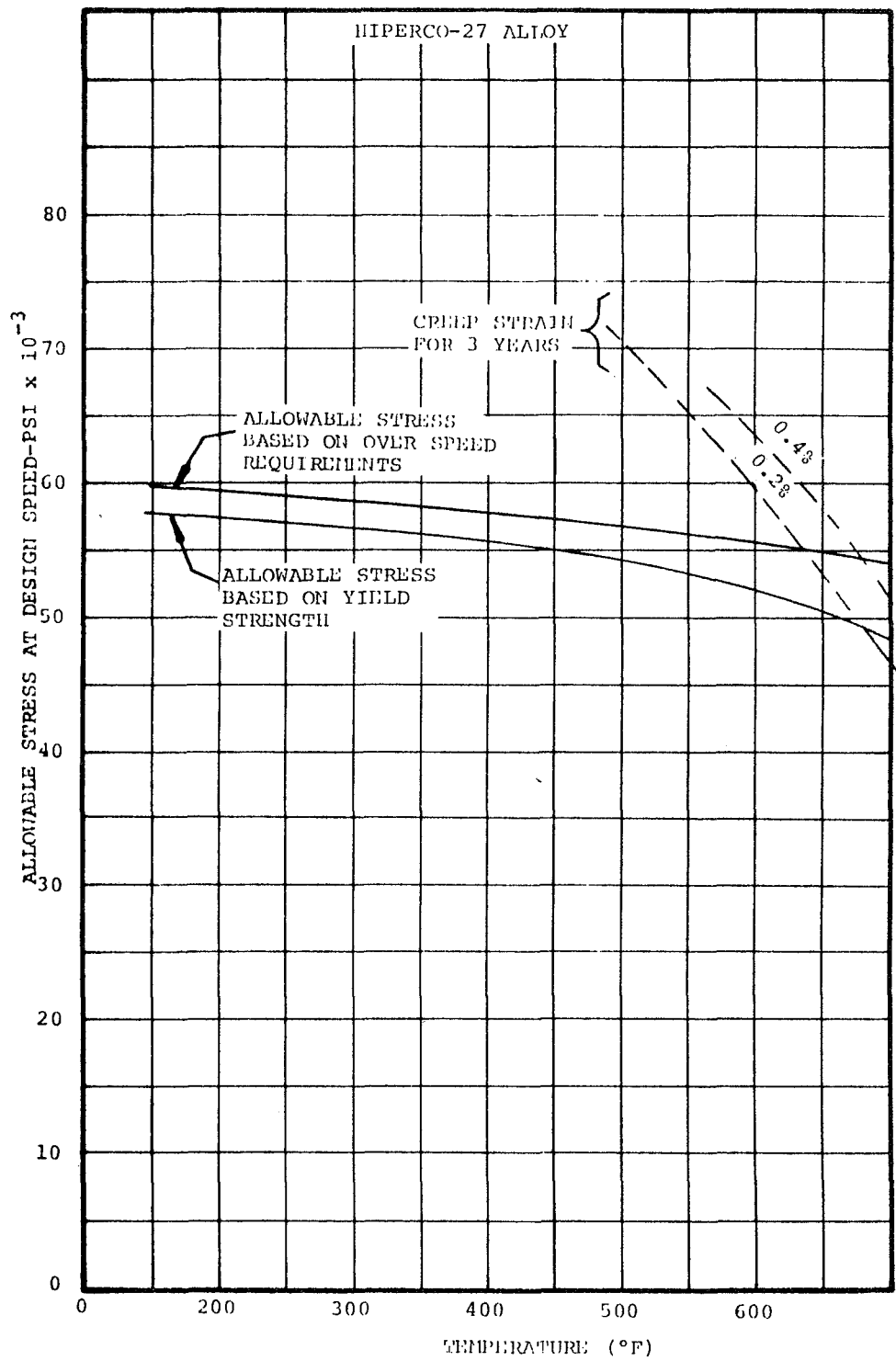


Figure 2.2-9. Interpretation of HipercO-27 Property Data.

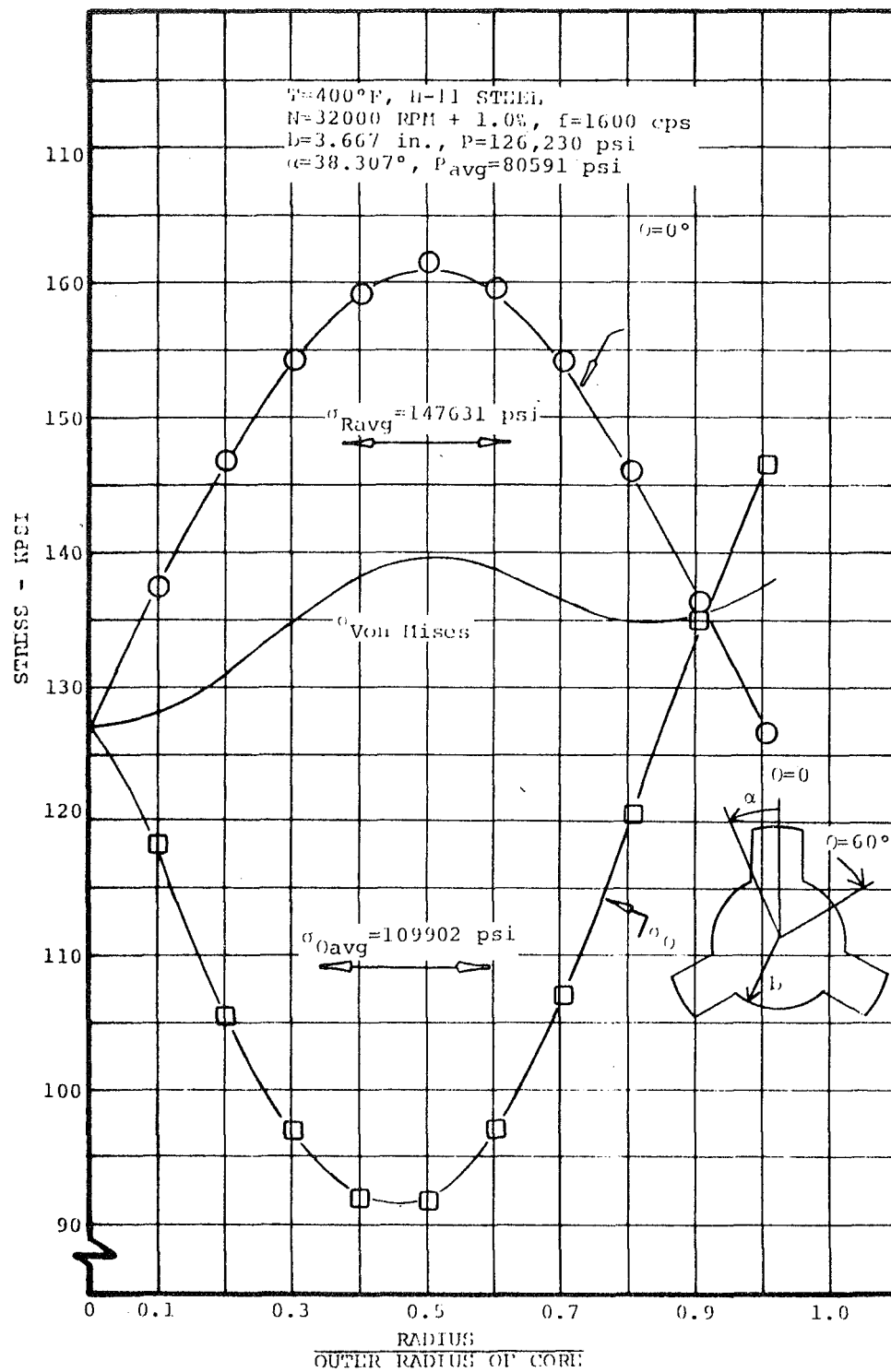


Figure 2.2-10. Rotor Stress at $\theta = 0^{\circ}$.

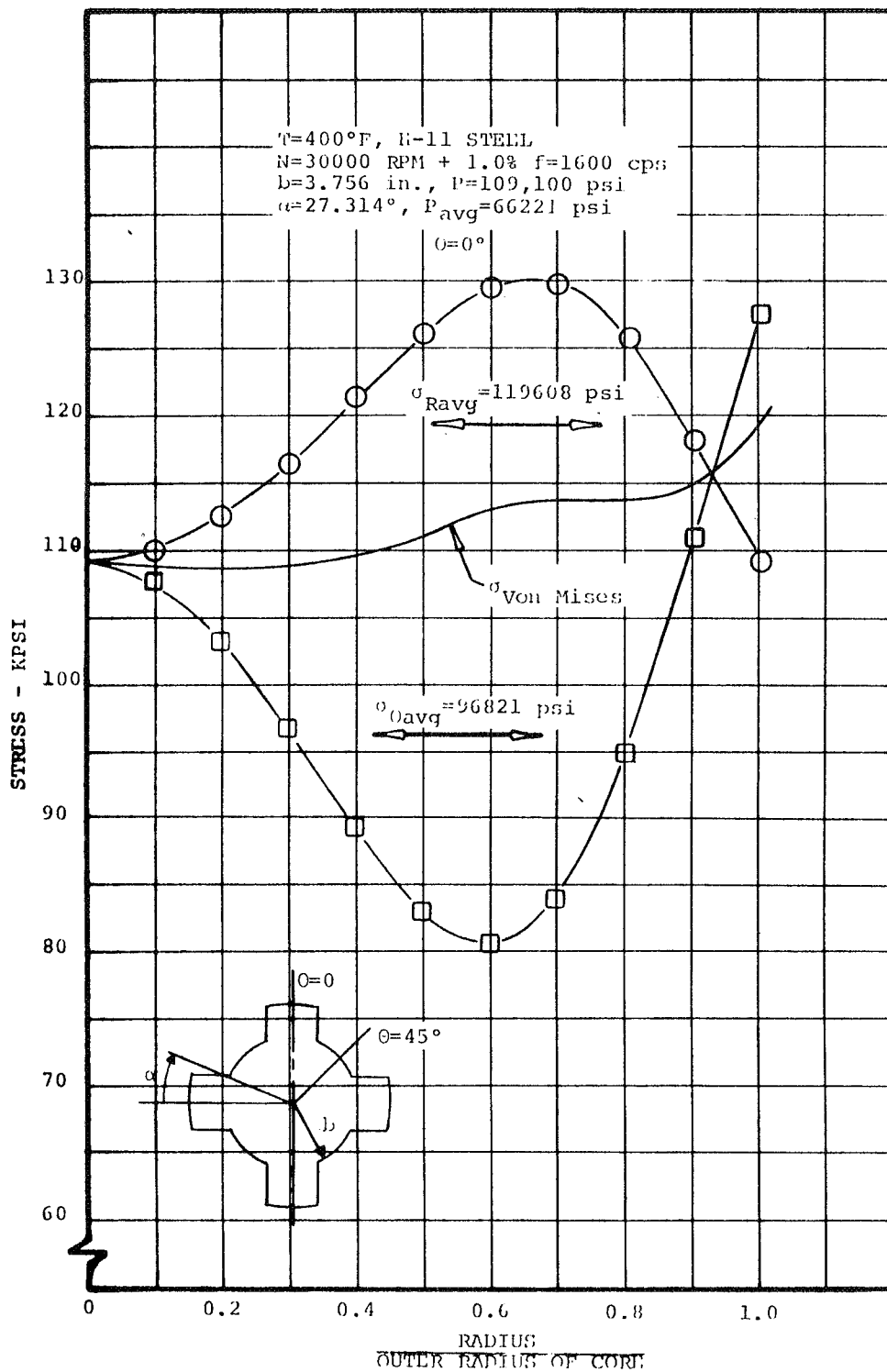


Figure 2.2-11. Rotor Stress at $\theta = 0^{\circ}$.

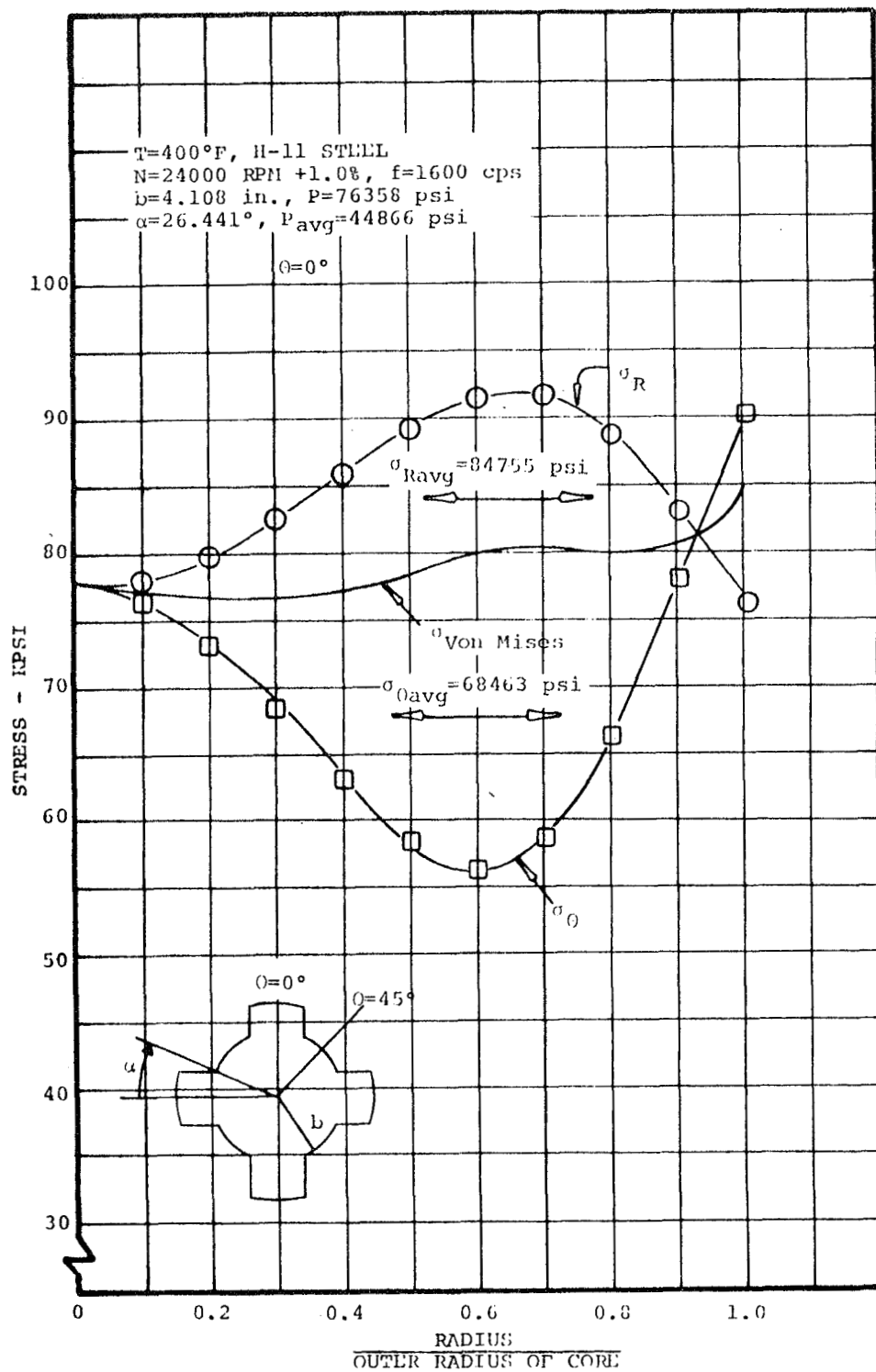


Figure 2.2-12. Rotor Stress at $\theta = 0^{\circ}$.

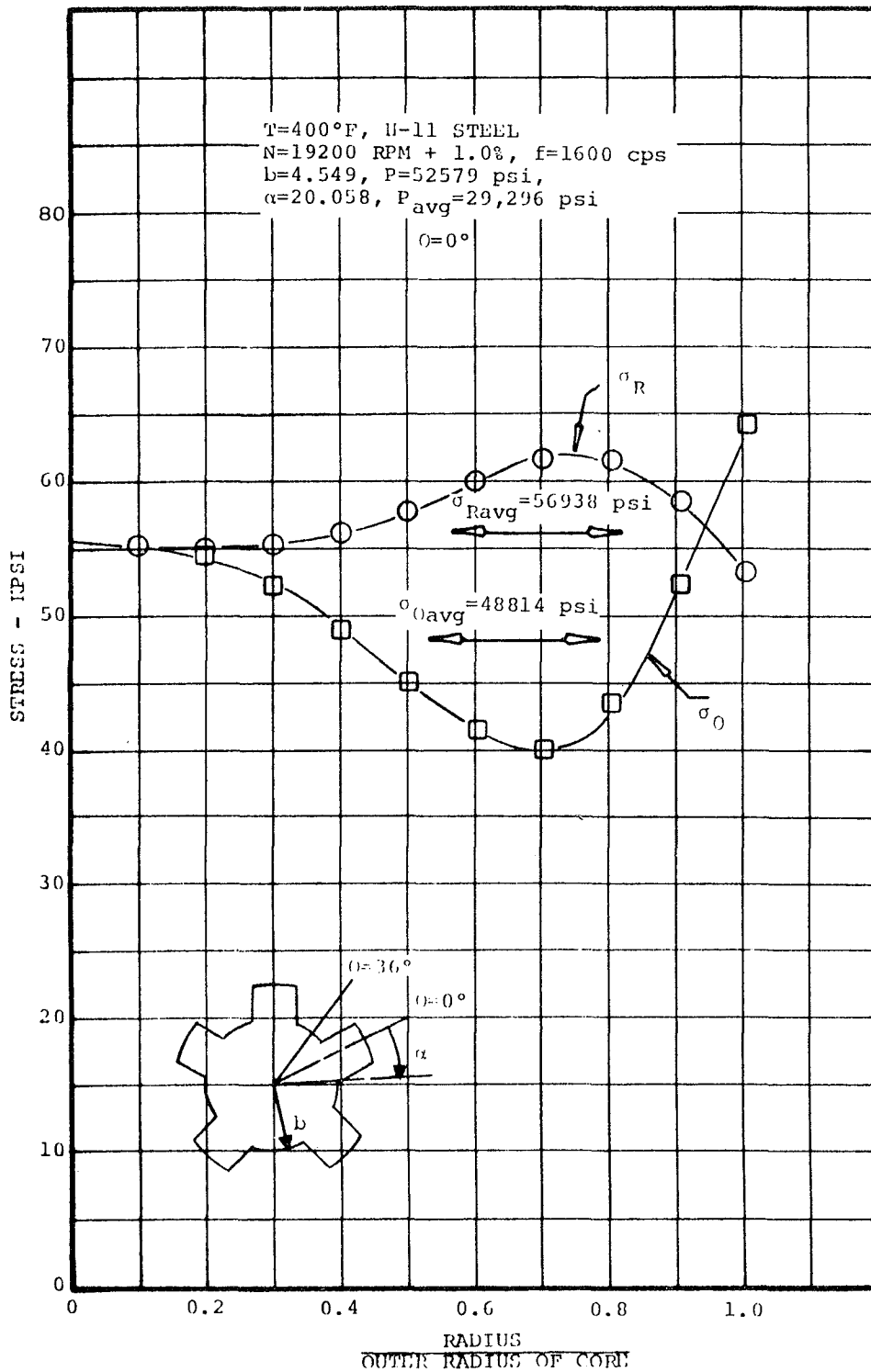


Figure 2.2-13. Rotor Stress at $\theta = 0^{\circ}$.

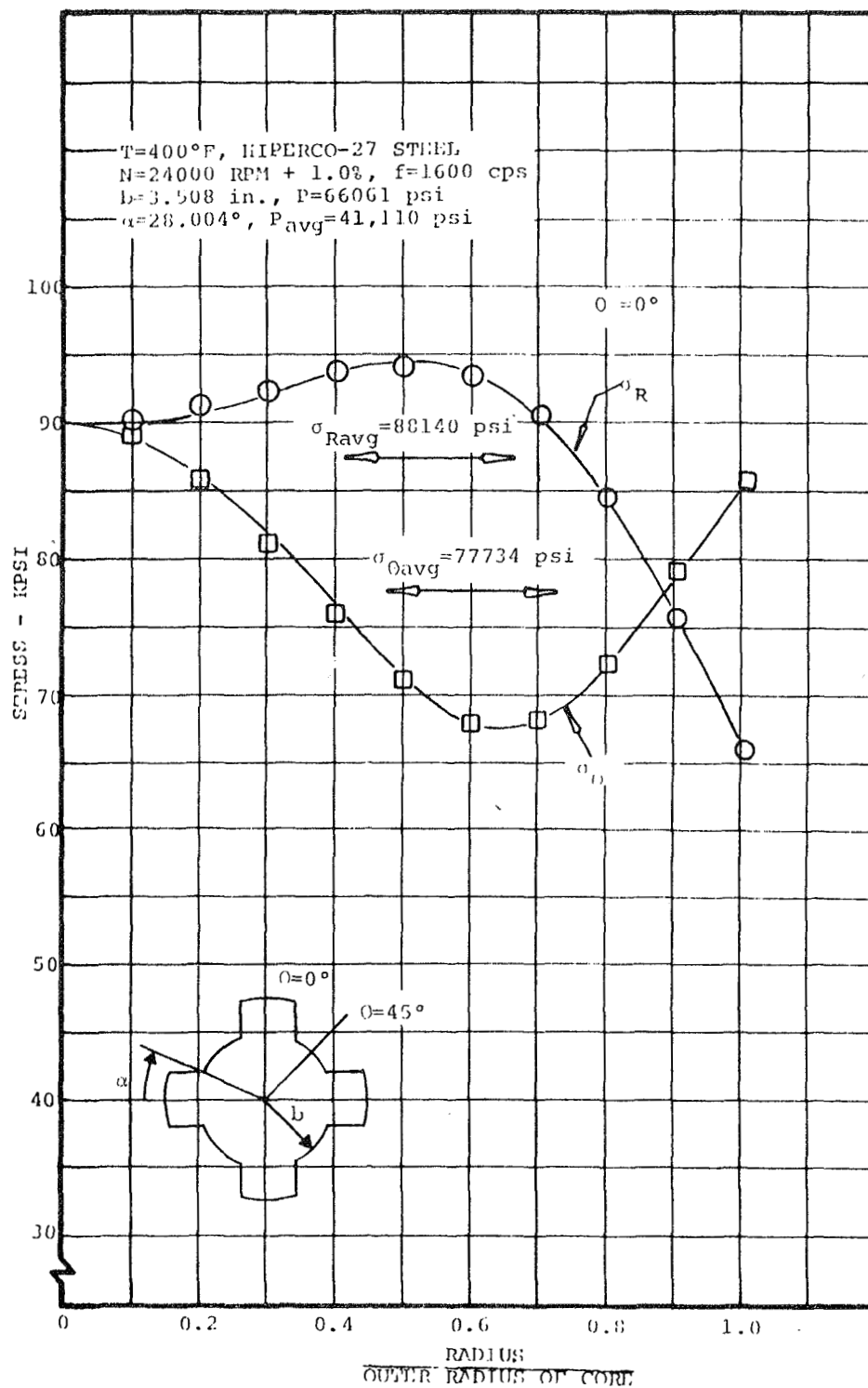


Figure 2.2-14. Rotor Stress at $\theta = 0^{\circ}$.

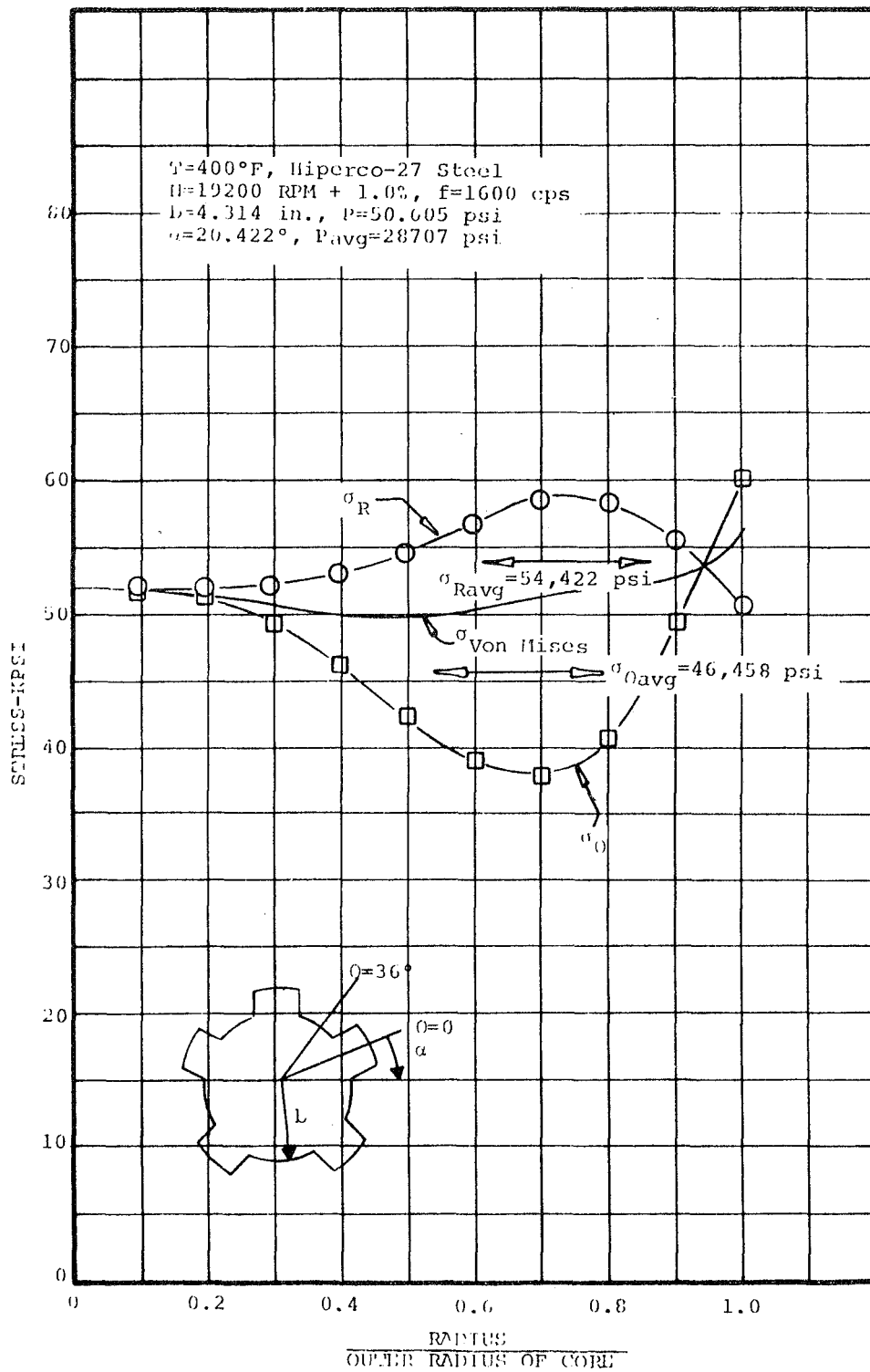


Figure 2.2-15. Rotor Stress at $\theta = 0^{\circ}$.

A safety factor, the ratio of the maximum allowable material stress to the maximum Von Mises stress is used for comparison of various designs. For both Hiperco-27 and H-11 rotor material at 400⁰F coolant, the limiting stress criteria turns out to be the yield strength consideration.

Table 2.3-2 of the next section is a summary of alternator designs for a coolant temperature of 400⁰F. The first columns give a picture of pertinent electromagnetic (EM) and mechanical design criteria. The latter columns give a summary of the mechanical stress picture and the resulting overall order of design preference.

For the H-11 alloy rotor, the stress is barely unacceptable at 30,000 rpm but certainly acceptable at speeds of 24,000 rpm or below. The stresses in the 30,000 rpm case could be made lower by designing specifically for the lowest possible stress at the expense of other electromagnetic and mechanical design criteria. However, data of the parametric studies which indicate this could be done also indicate the stress could only be lowered a few thousand psi, the result of which would be to make the speed barely acceptable with a small safety factor. From a reliability standpoint then, it would still be a low order of overall preference.

For the Hiperco-27 alloy rotor, the stress is barely unacceptable at 19,200 rpm but acceptable at the speeds below this (17,143 or 16,000 rpm). The safety factor for the 19,200 rpm, Hiperco-27 is of the same order of magnitude as that for the 30,000 rpm, H-11 design so the remarks above for making the H-11 design "barely acceptable" are applicable to the Hiperco design.

An alternate approach to making the 19,200 rpm, Hiperco-27 design more acceptable might be to substitute a slightly higher strength

material (likewise for the 30,000 rpm, H-11 case) which would undoubtedly have lower magnetic capabilities. However, substituting an intermediate, less known material between H-11 (high strength, low magnetics) and Hiperco (low strength, high magnetics) does not appear warranted since the differences between the designs at these extremes are small.

A review of the strength properties of H-11 steel and of the 1BS1 precipitation hardened alloy (modified cobalt base) was made for the 800° and 1100° F coolant temperature design condition. An interpretation of the available data for each material is graphically summarized in Figures 2.2-16 and 2.2-17, respectively. The strength data for H-11 steel was taken from report no. AFAPL-TR-66-127 (2.2-6) and report no. NASA-CR-54091 (2.2-8). Only a limited amount of data was available for the precipitation hardened, modified cobalt base alloy since only a few experiments were performed under Contract NAS3-6465 (2.2-9). The objectives of these experiments were to determine the relative merits of this type of material and not to determine the necessary property data for the design of a rotor. Therefore, the information present in Figure 2.2-17 cannot be used for the purpose of designing a rotor. The data only represents a "best estimation" of the property data that is required for the design of a rotor for 3-years life. Similarly, considerable extrapolation of the creep data for H-11 steel, shown on Figure 2.2-16, was necessary to obtain the creep strength for the 3-years time period. Thus, the creep information given by Figure 2.2-16 is also only a "best estimation" of the creep data that is required for the design of a rotor for 3-years operation. Inasmuch as H-11 is known to undergo aging, the strengths shown can be expected to be less at the end of three years. The extent of the aging is unknown at this time.

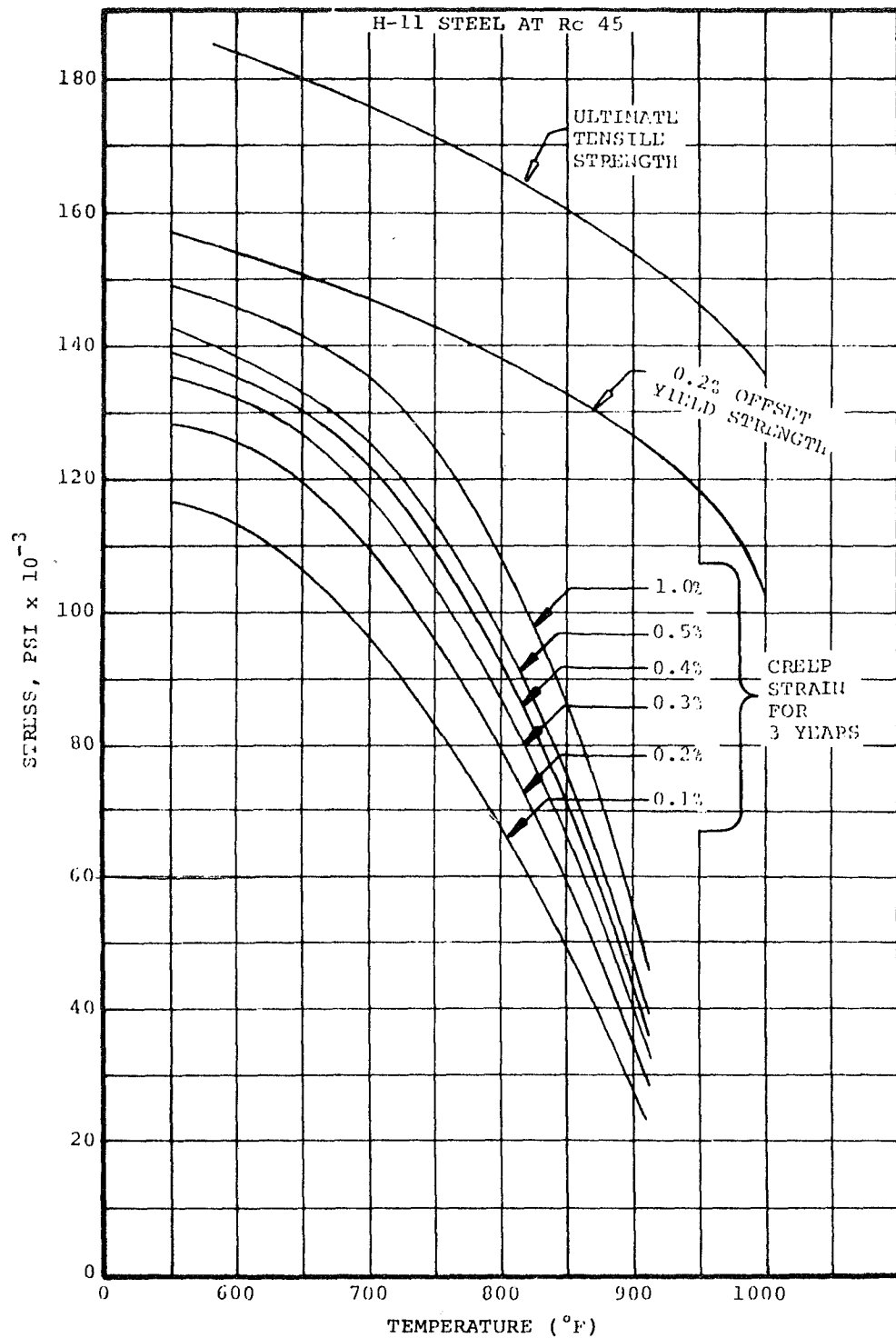


Figure 2.2-16. Interpretation of H-11 Steel Property Data at a Value Rockwell C45.

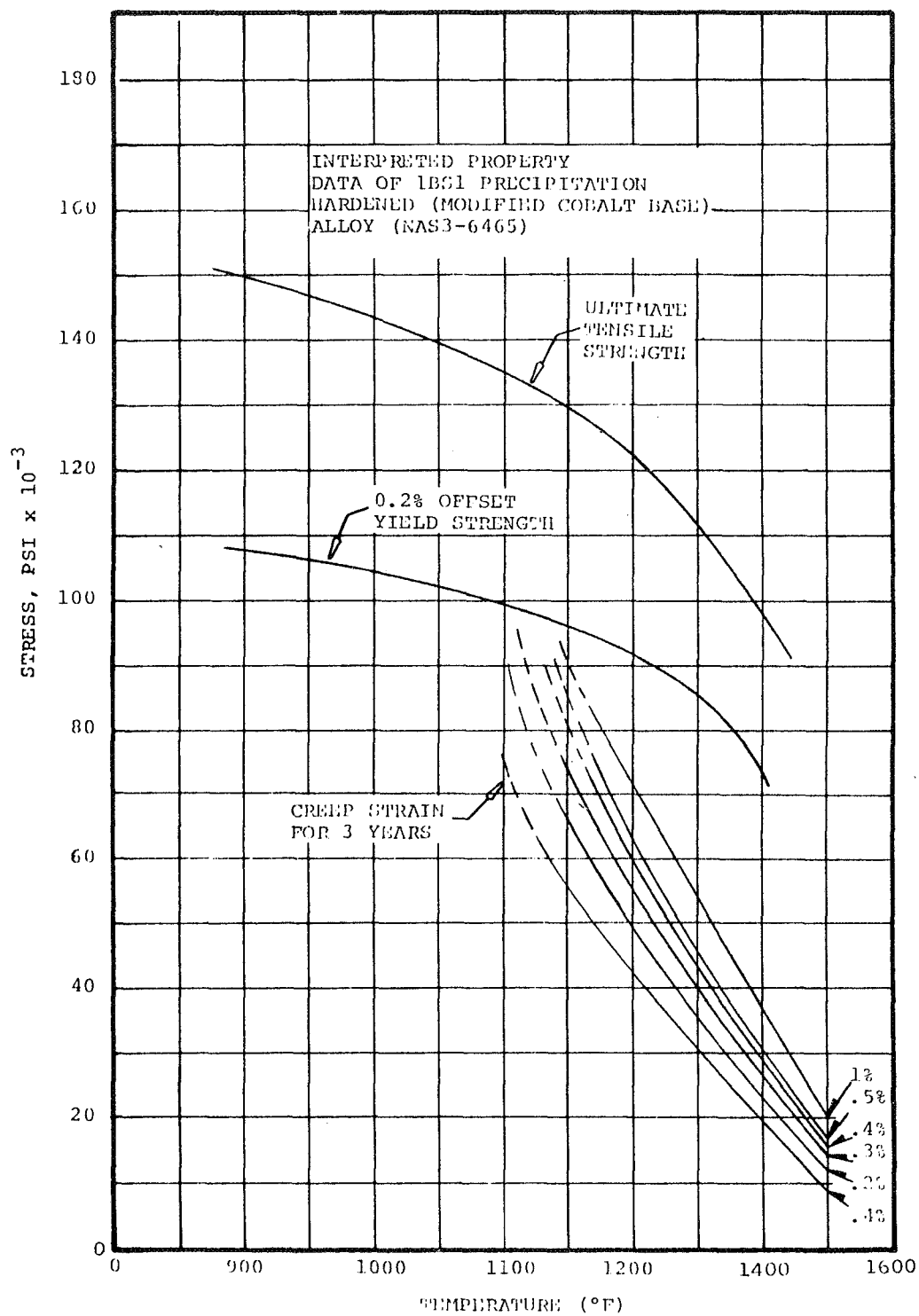


Figure 2.2-17. Interpretation of 1BS1 Precipitation Hardened Alloy Property Data.

The stress distribution for the rotors for the alternators listed in Table 2.3-1 of the next section were calculated by the procedure outlined in the Appendix of Reference 2.2-2. Four stress profiles for each rotor are graphically illustrated in Figures 2.2-20 through 2.2-43. These stress profiles represent the following:

- (a) The radial stress distribution which would exist within the rotor core if the centrifugal forces on the pole were uniformly distributed around the outer boundary of the rotor core.
- (b) The radial and tangential stress distributions in the rotor core on a radial plane (r-z plane) through the radial centerline of the poles.
- (c) The radial and tangential stress distributions in the rotor core on a radial plane (r-z plane) midway between the radial centerline of adjacent poles.
- (d) The average tangential and radial stress with respect to the radius of the rotor core that exists around the core.

In general the maximum stress intensity occurs under the poles. Therefore, the biaxial tensile stresses at this location were converted into an equivalent stress so a comparison with the uniaxial tensile data from Figures 2.2-18 and 2.2-19 could be made. The equivalent stress is shown only for the cases where the acceptability of the design is questionable.

On the basis of a comparison between the actual stresses and the stress limitations of the materials, the following conclusions are justified for the 800^oF and 1100^oF designs:

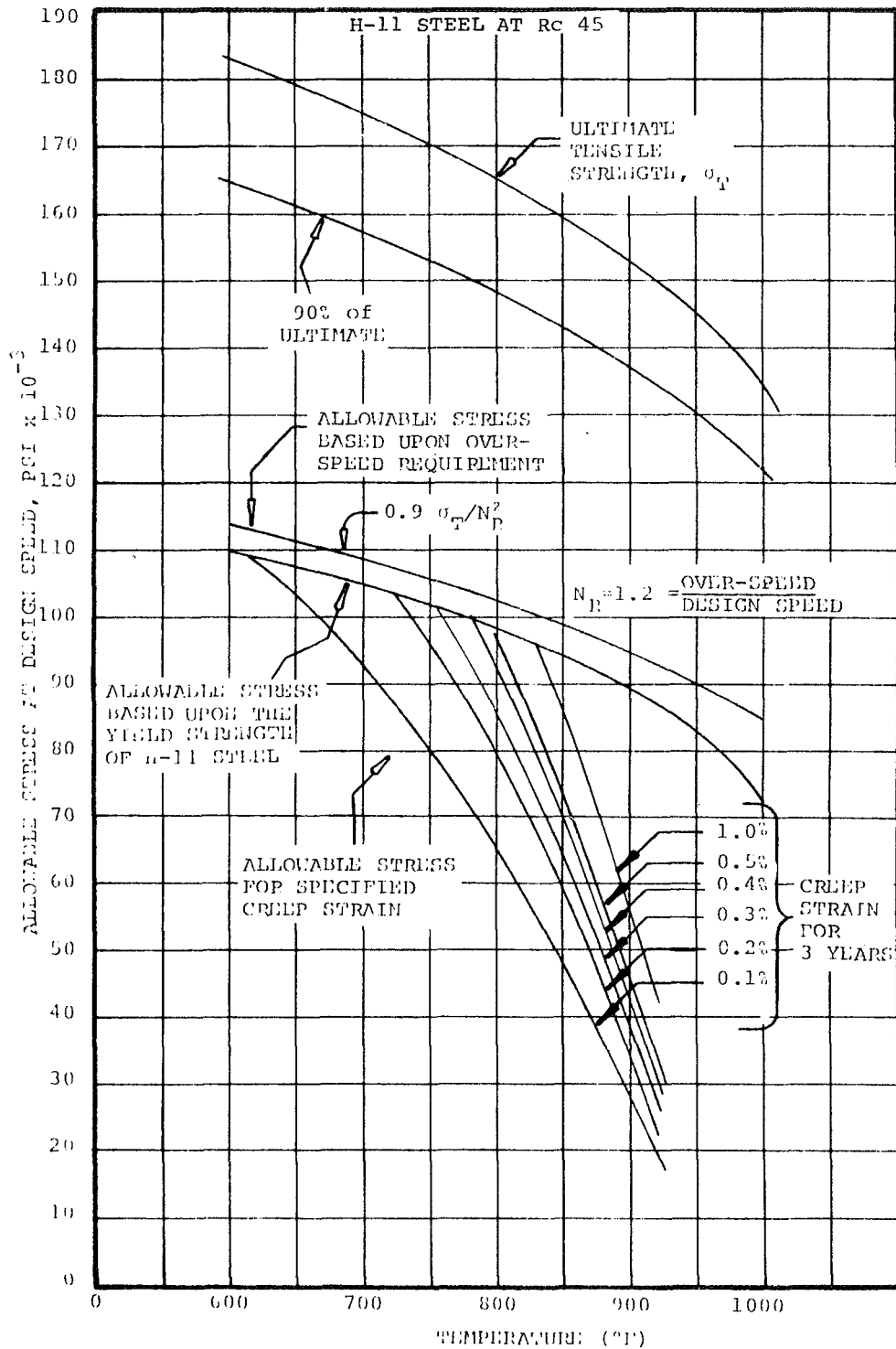


Figure 2.2-18. Interpretation of H-11 Steel Property Data at a Value Rockwell C45.

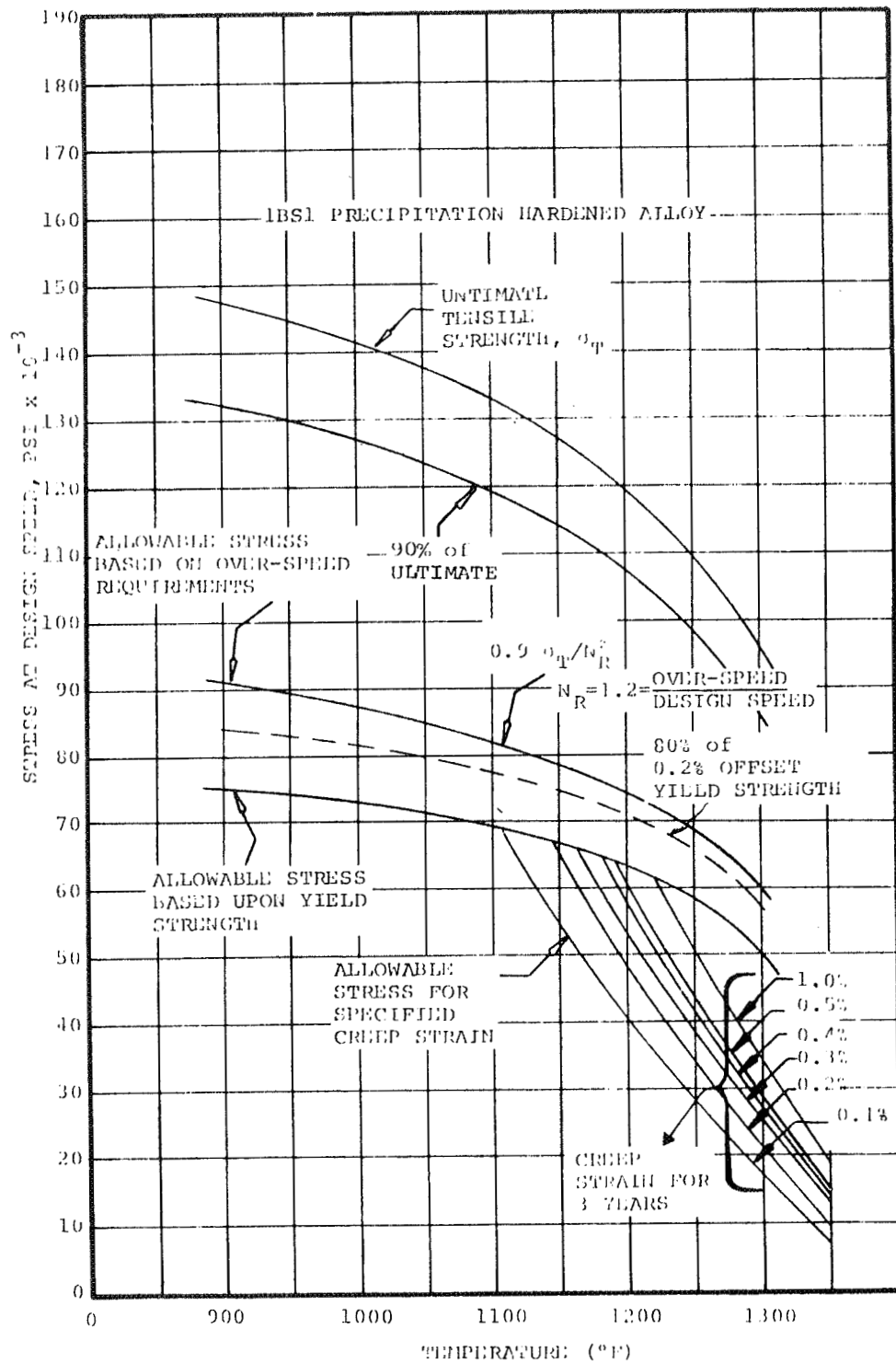


Figure 2.2-19. Interpretation of IBSI Precipitation Hardened Alloy Property Data.

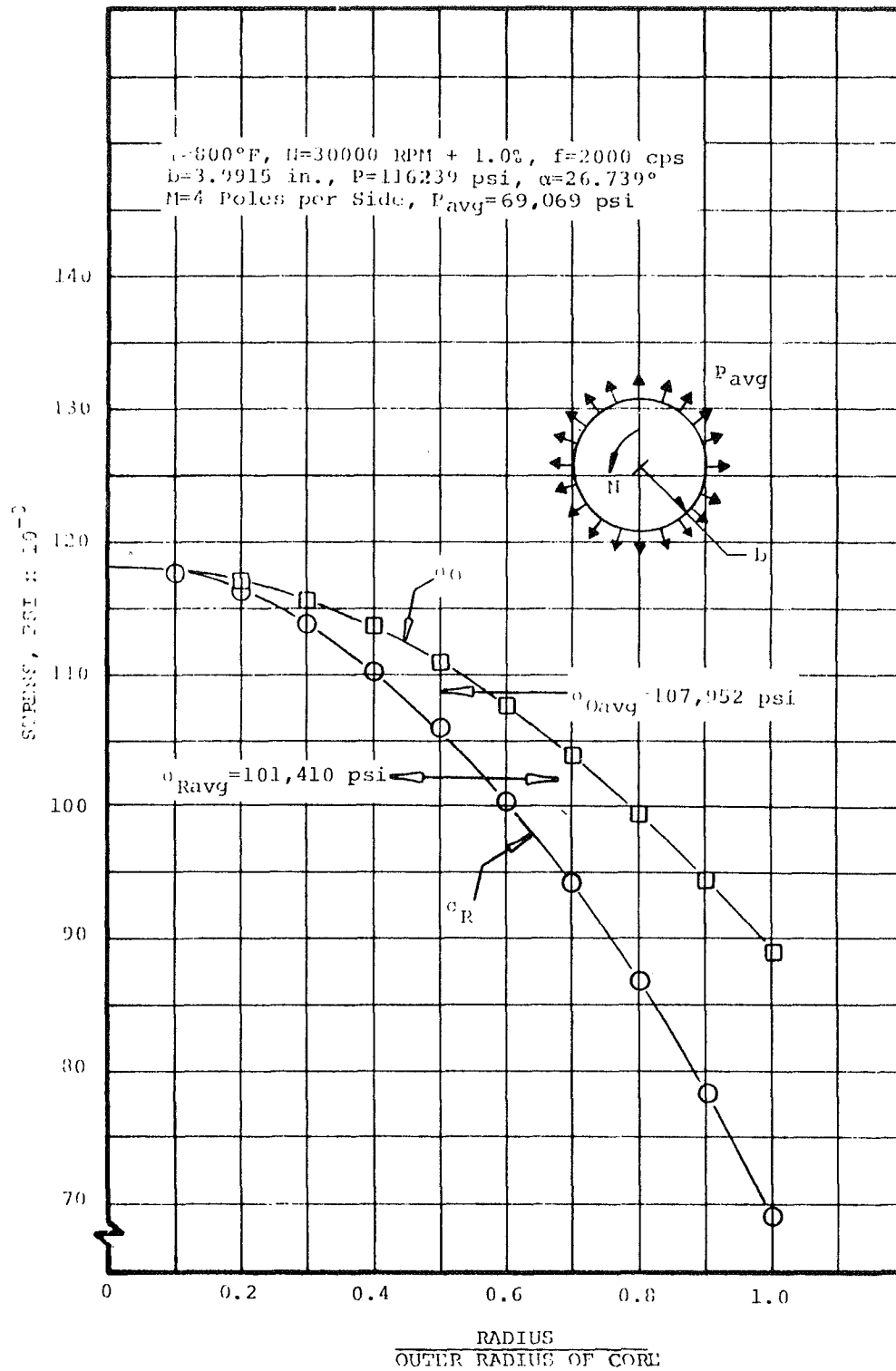


Figure 2.2-20. Rotor Stress at Uniform Load.

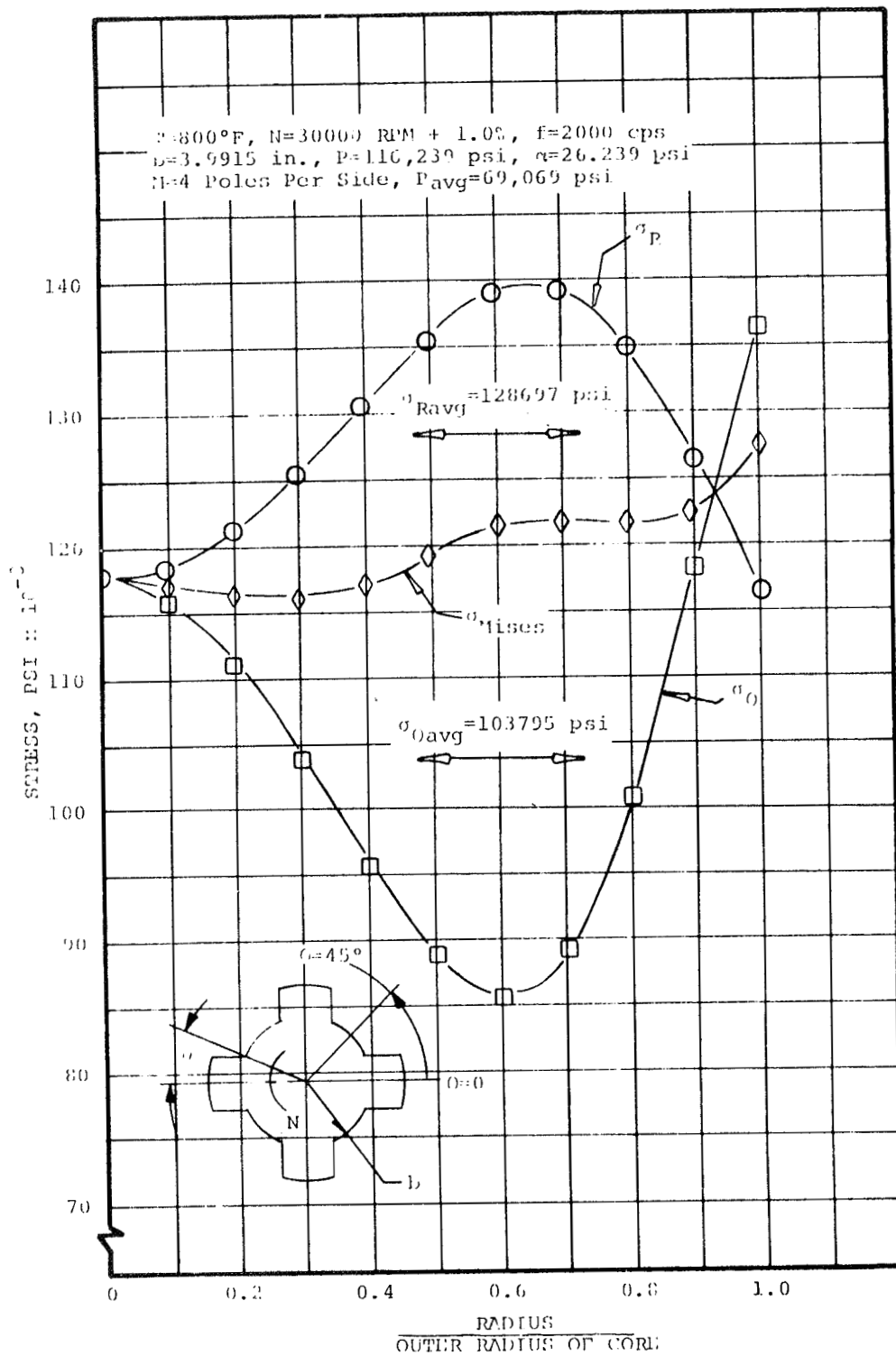


Figure 2.2-21. Rotor Stress at $\theta = 0^{\circ}$.

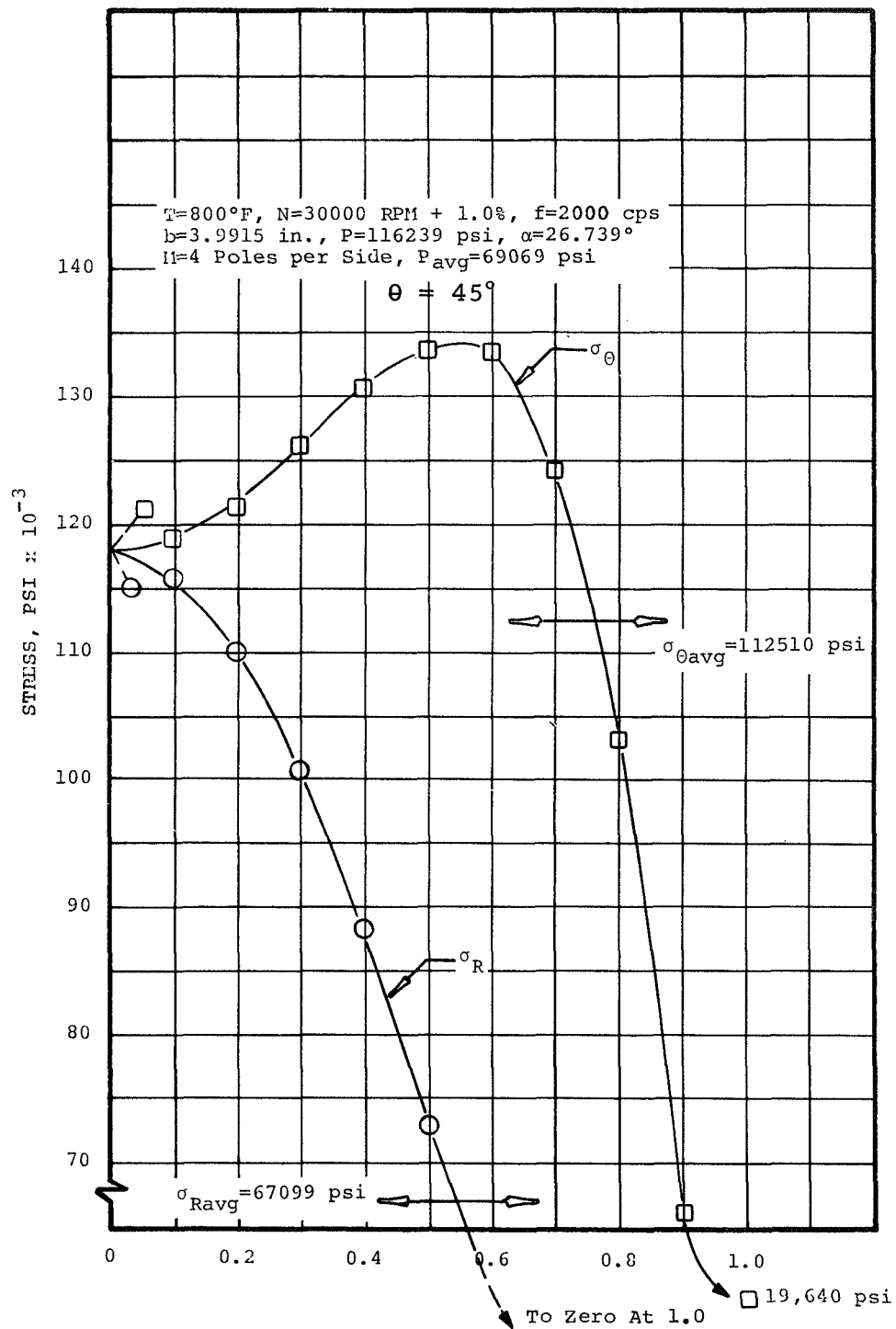


Figure 2.2-22. Rotor Stress at $\theta = 0^{\circ}$.

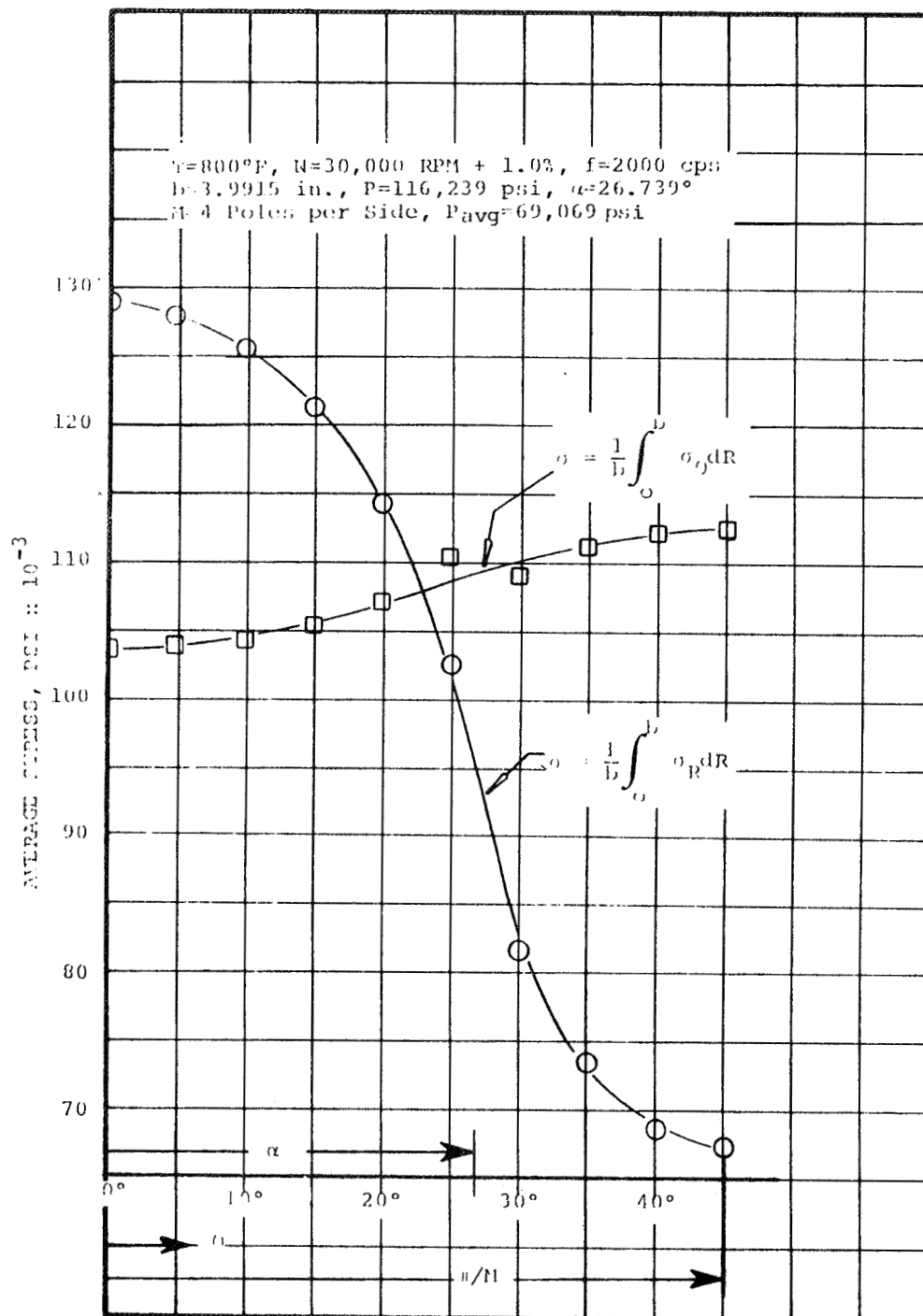


Figure 2.2-23. Rotor Stress (Average) as a Function of θ .

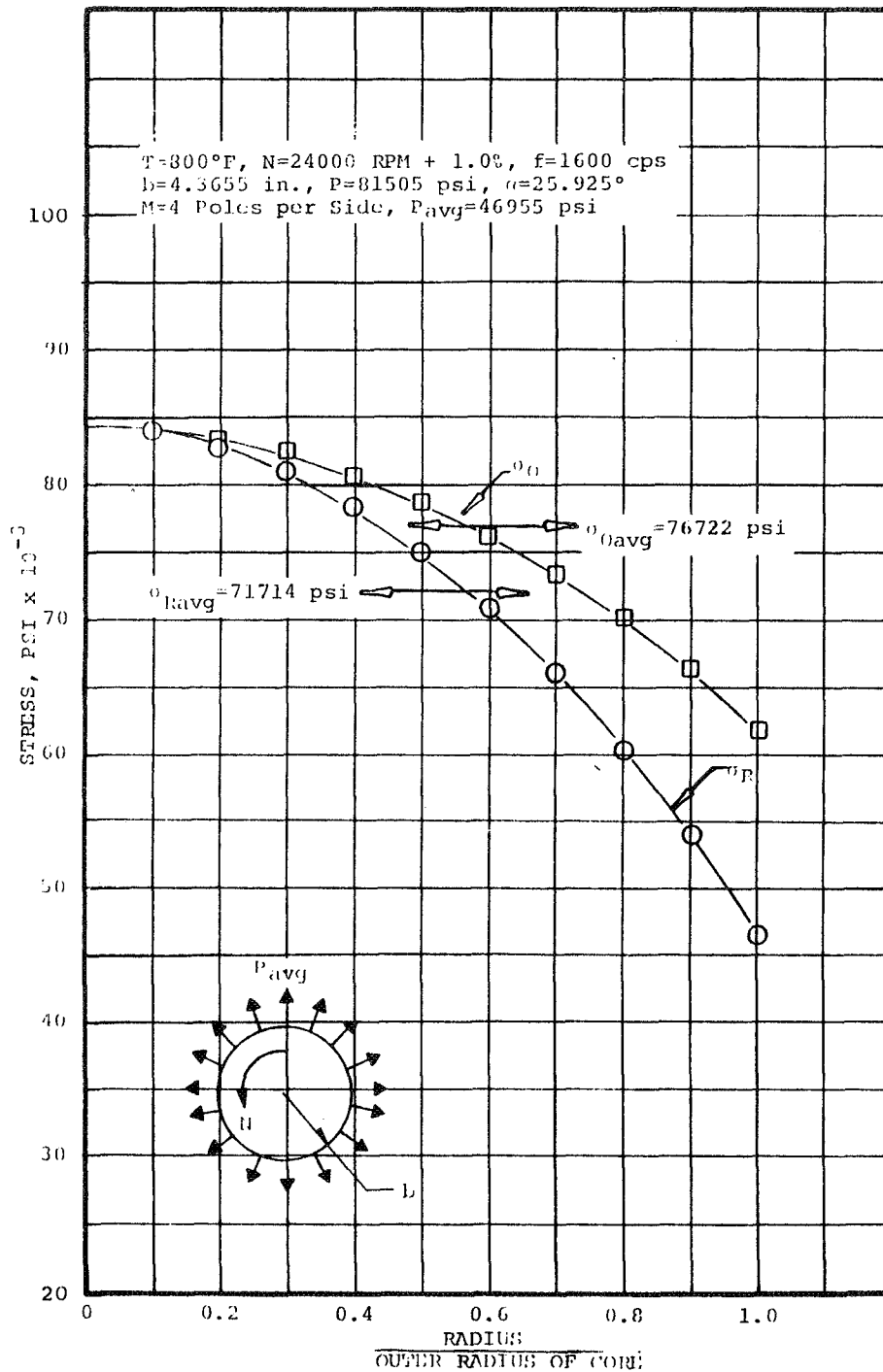


Figure 2.2-24. Rotor Stress at Uniform Load.

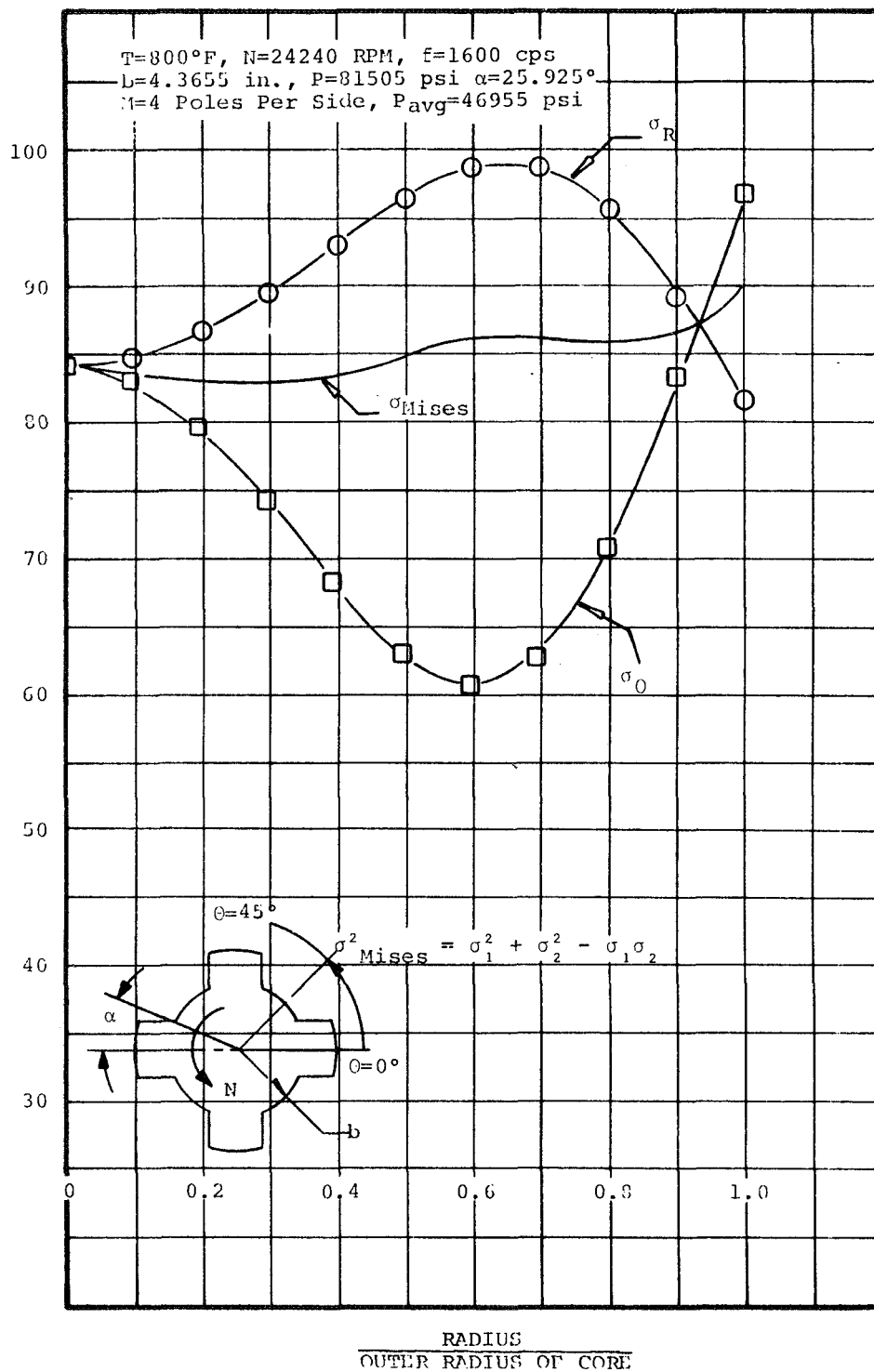


Figure 2.2-25. Rotor Stress at $\theta = 0^{\circ}$.

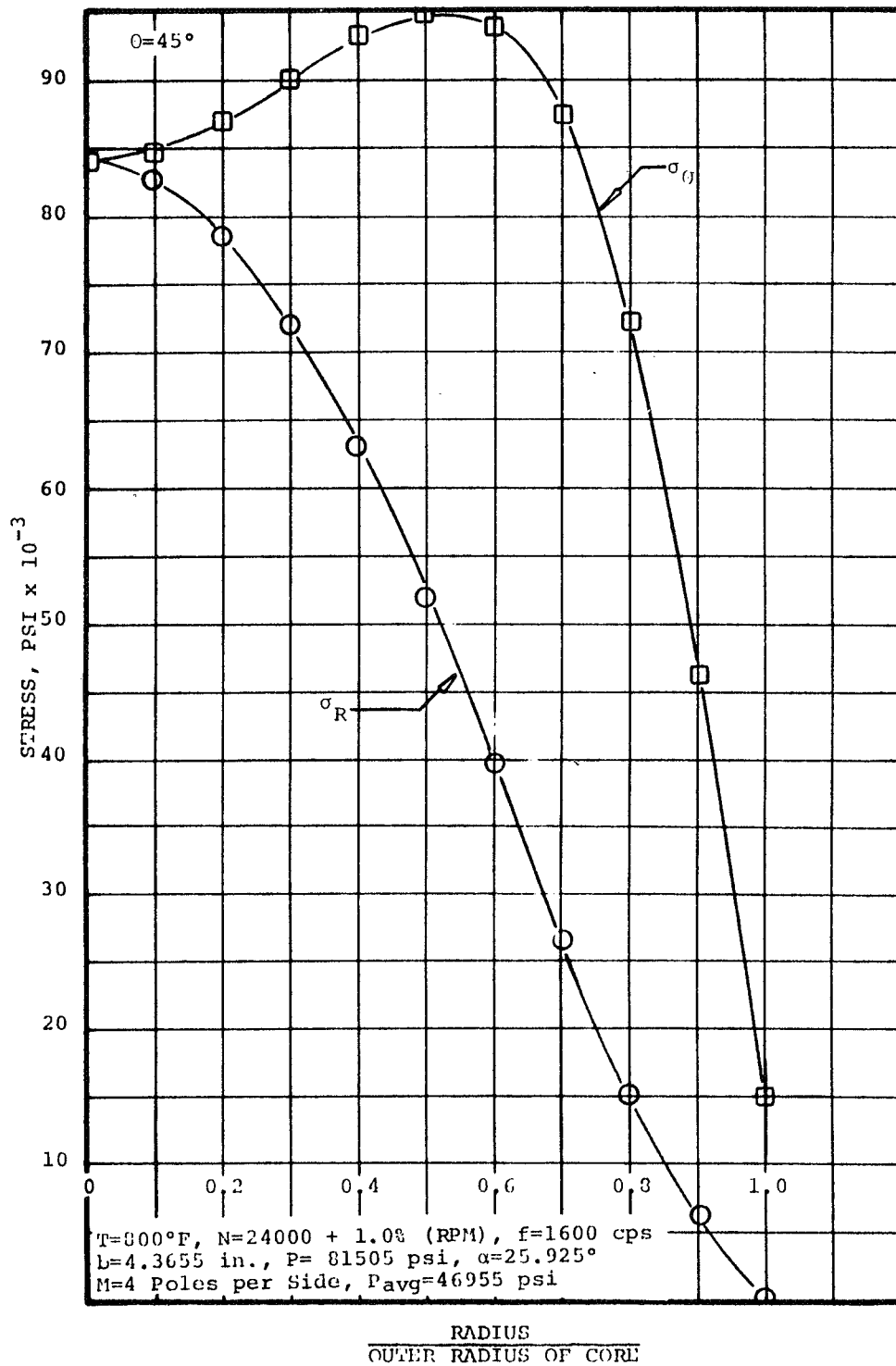


Figure 2.2-26. Rotor Stress at $\theta = 45^\circ$.

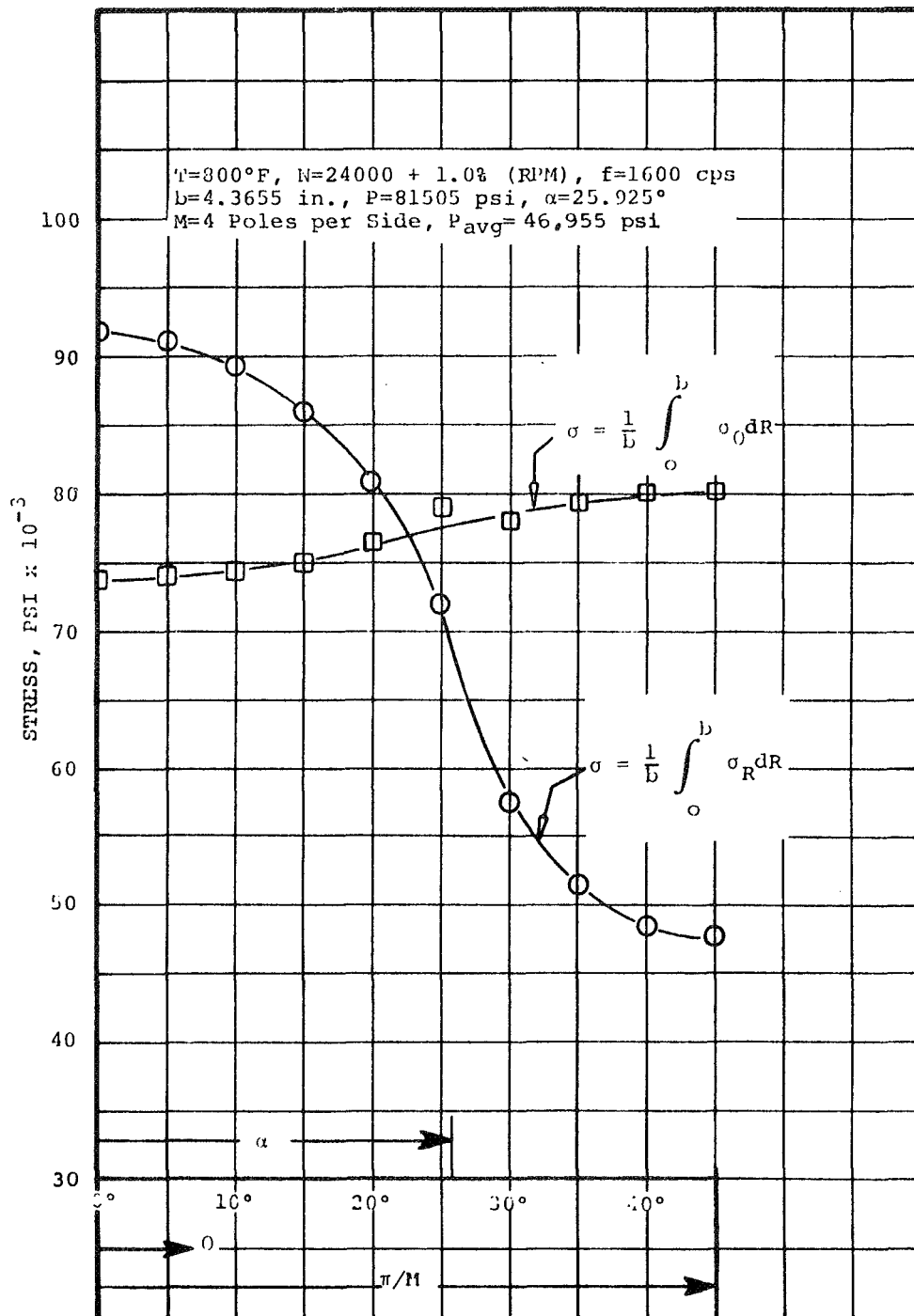


Figure 2.2-27. Rotor Stress (Average) as a Function of θ .

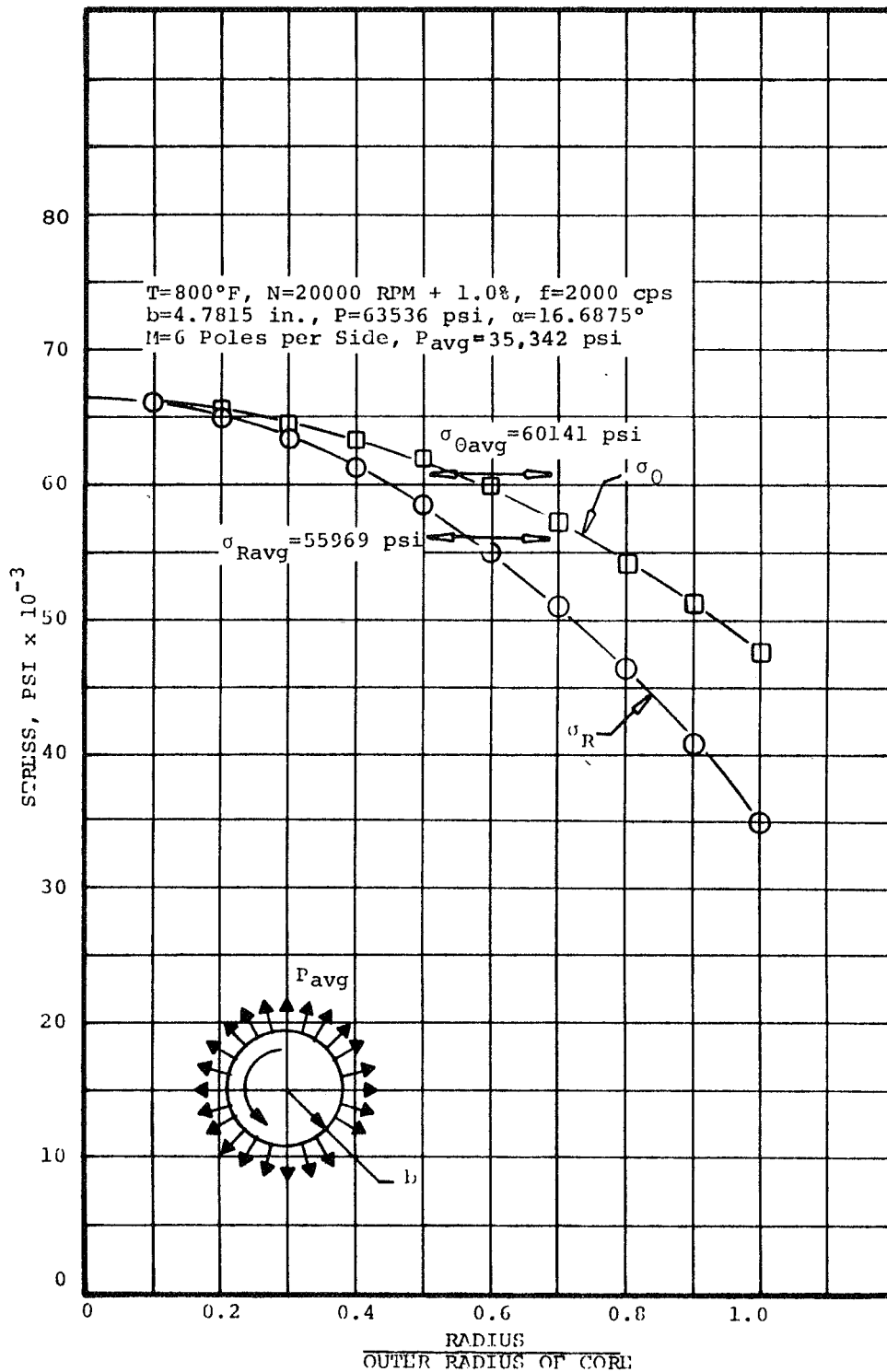


Figure 2.2-28. Rotor Stress at Uniform Load.

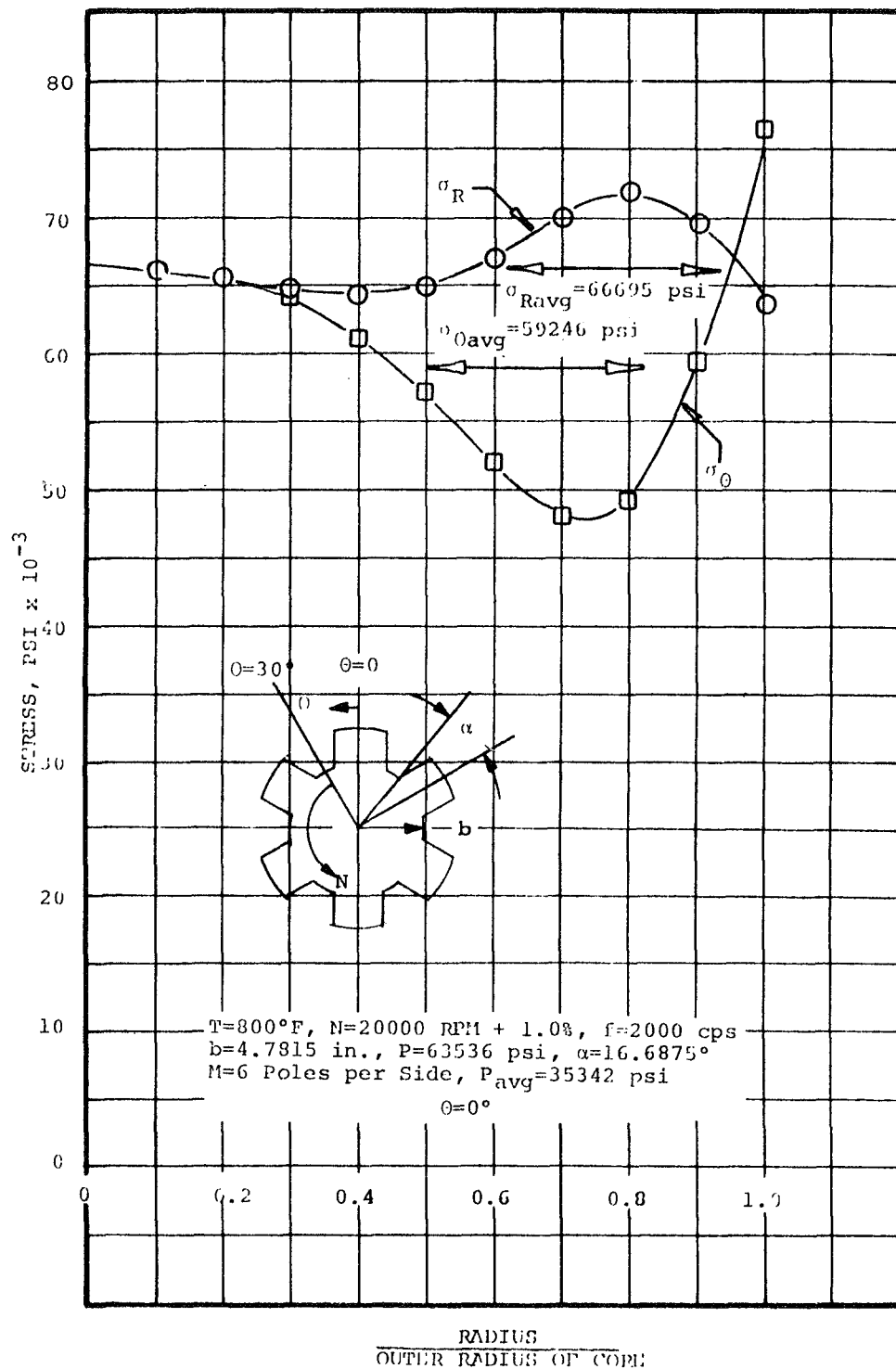


Figure 2.2-29. Rotor Stress at $\theta = 0^\circ$.

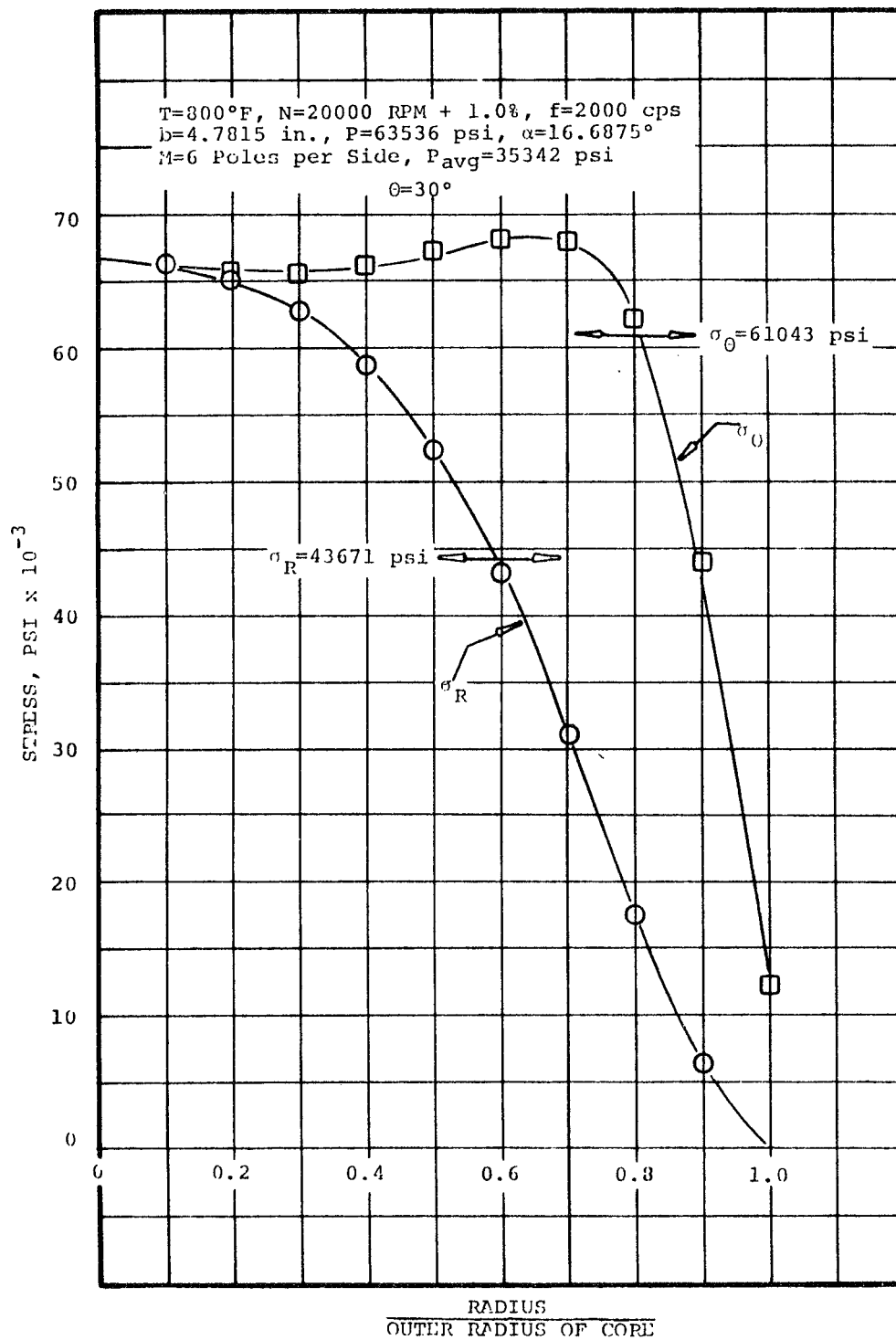


Figure 2.2-30. Rotor Stress at $\theta = 30^{\circ}$.

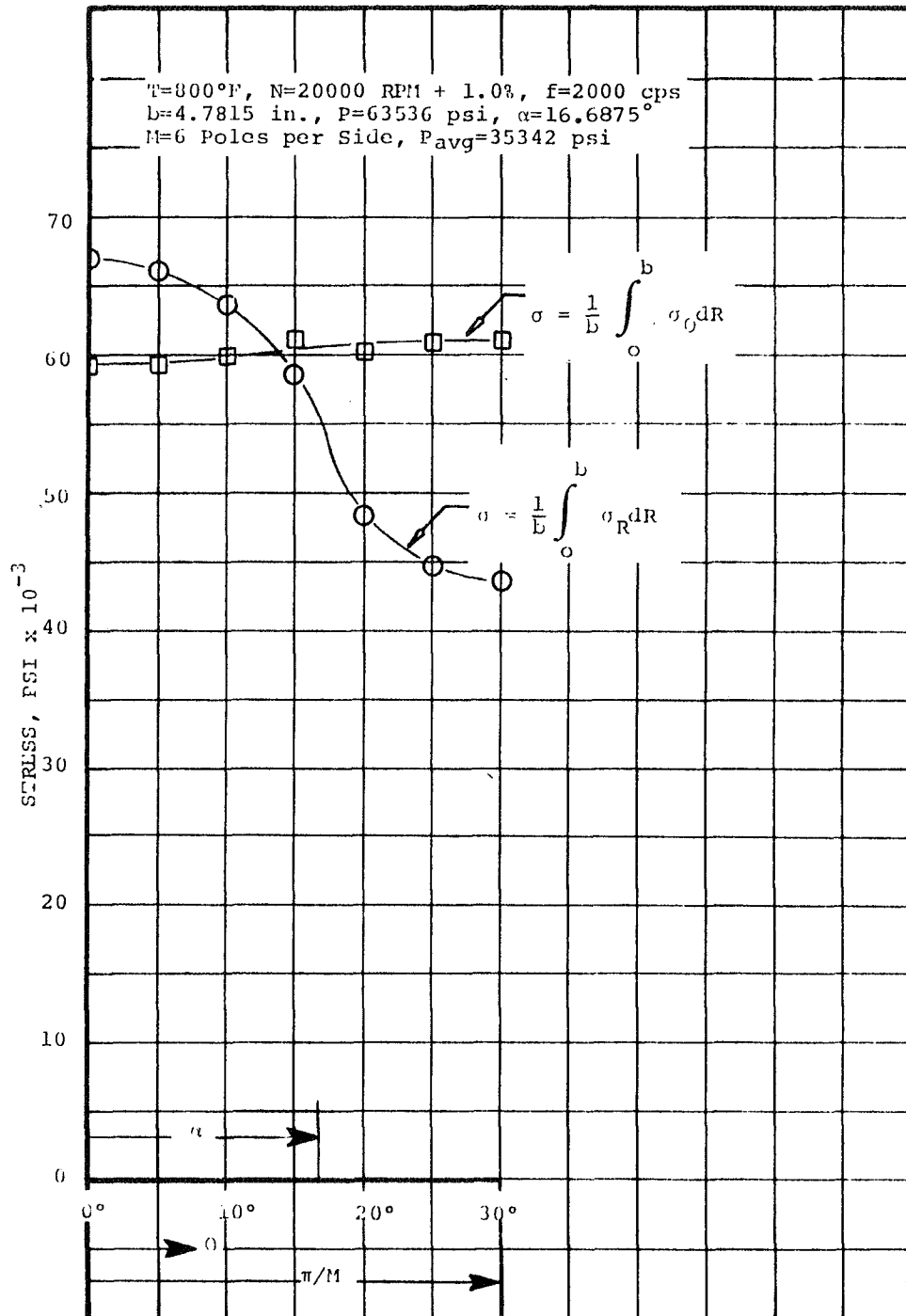


Figure 2.2-31. Rotor Stress (Average) as a Function of θ .

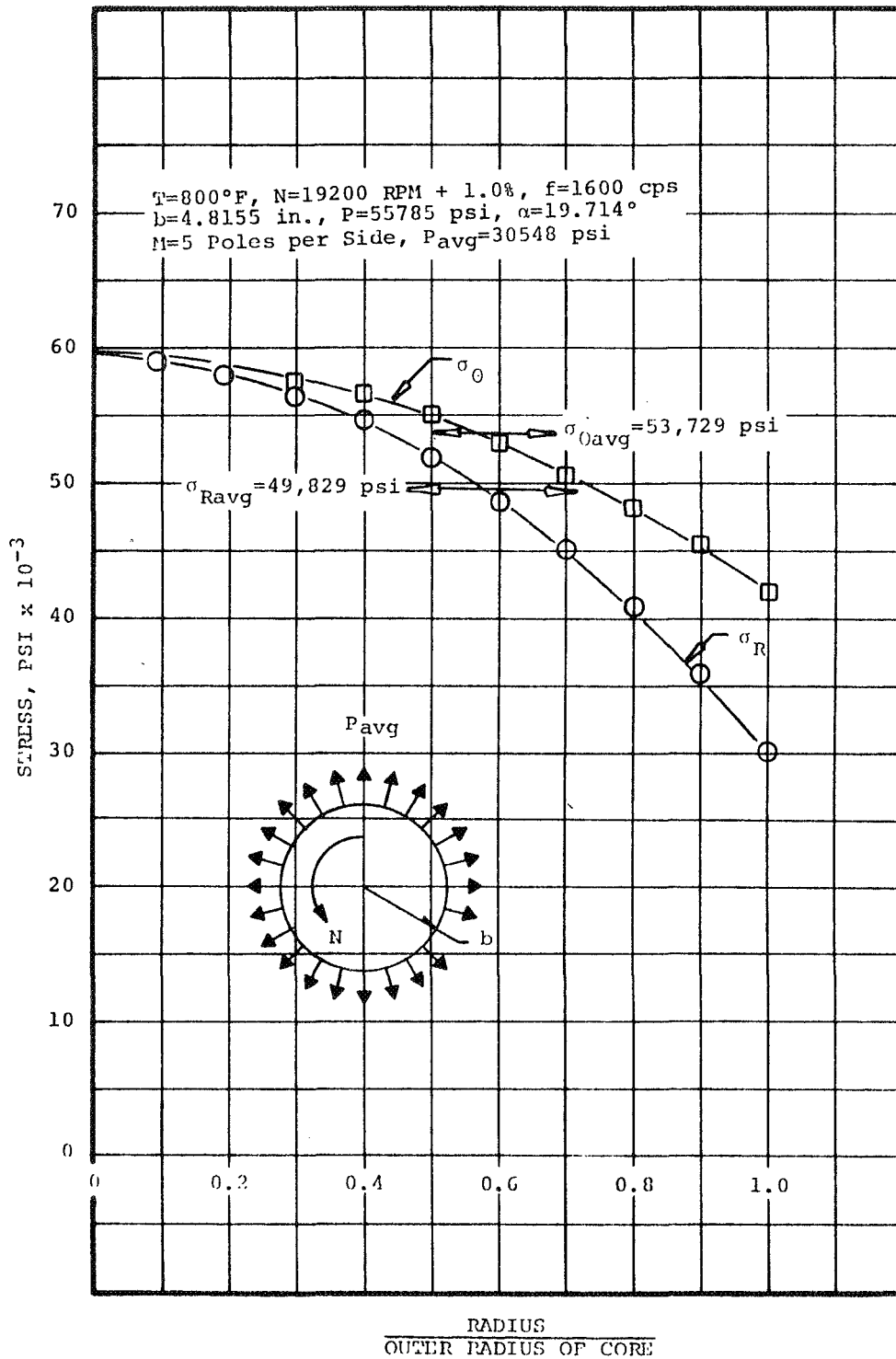


Figure 2.2-32. Rotor Stress at Uniform Load.

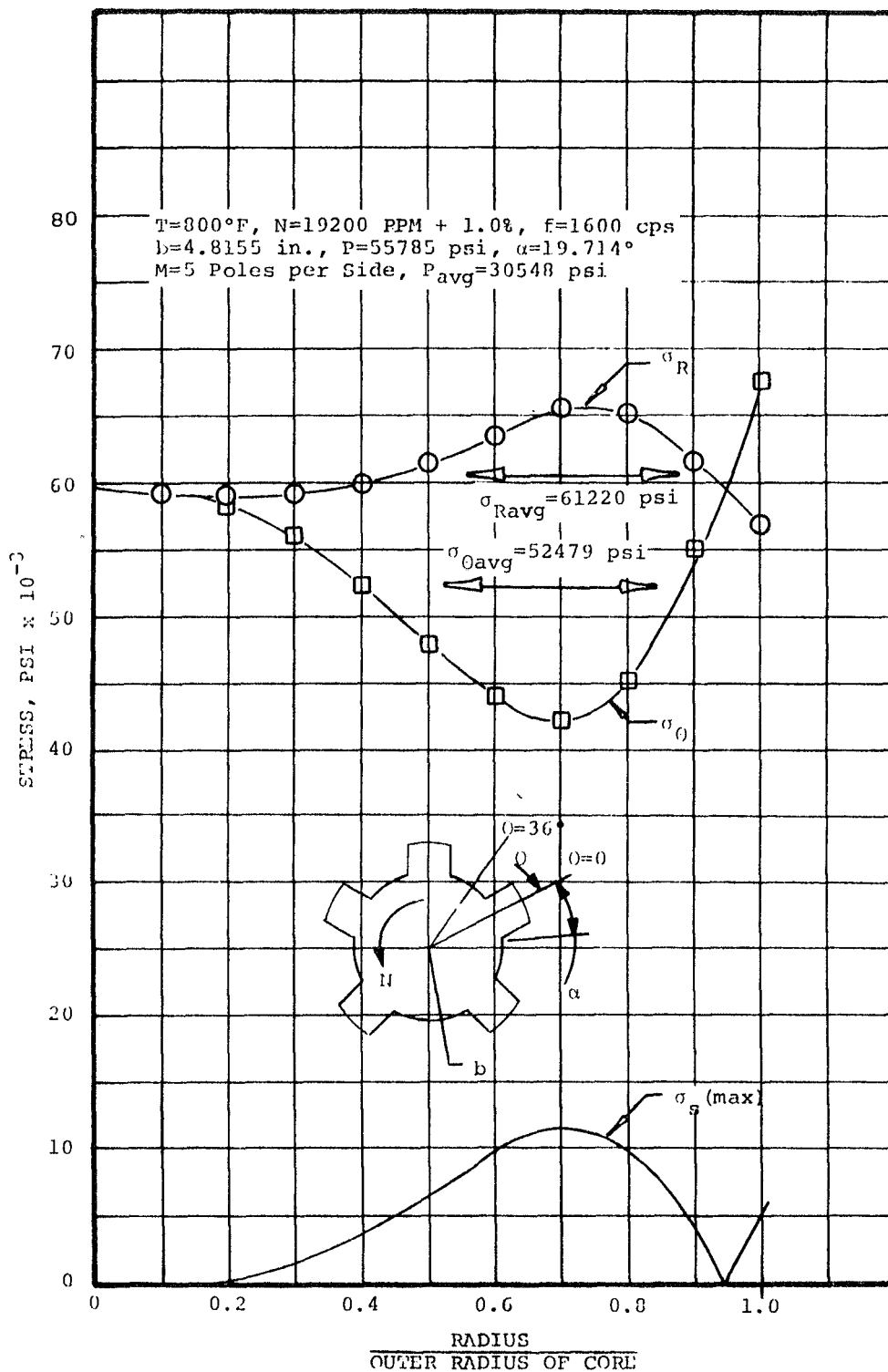


Figure 2.2-33. Rotor Stress at $\theta = 0^{\circ}$.

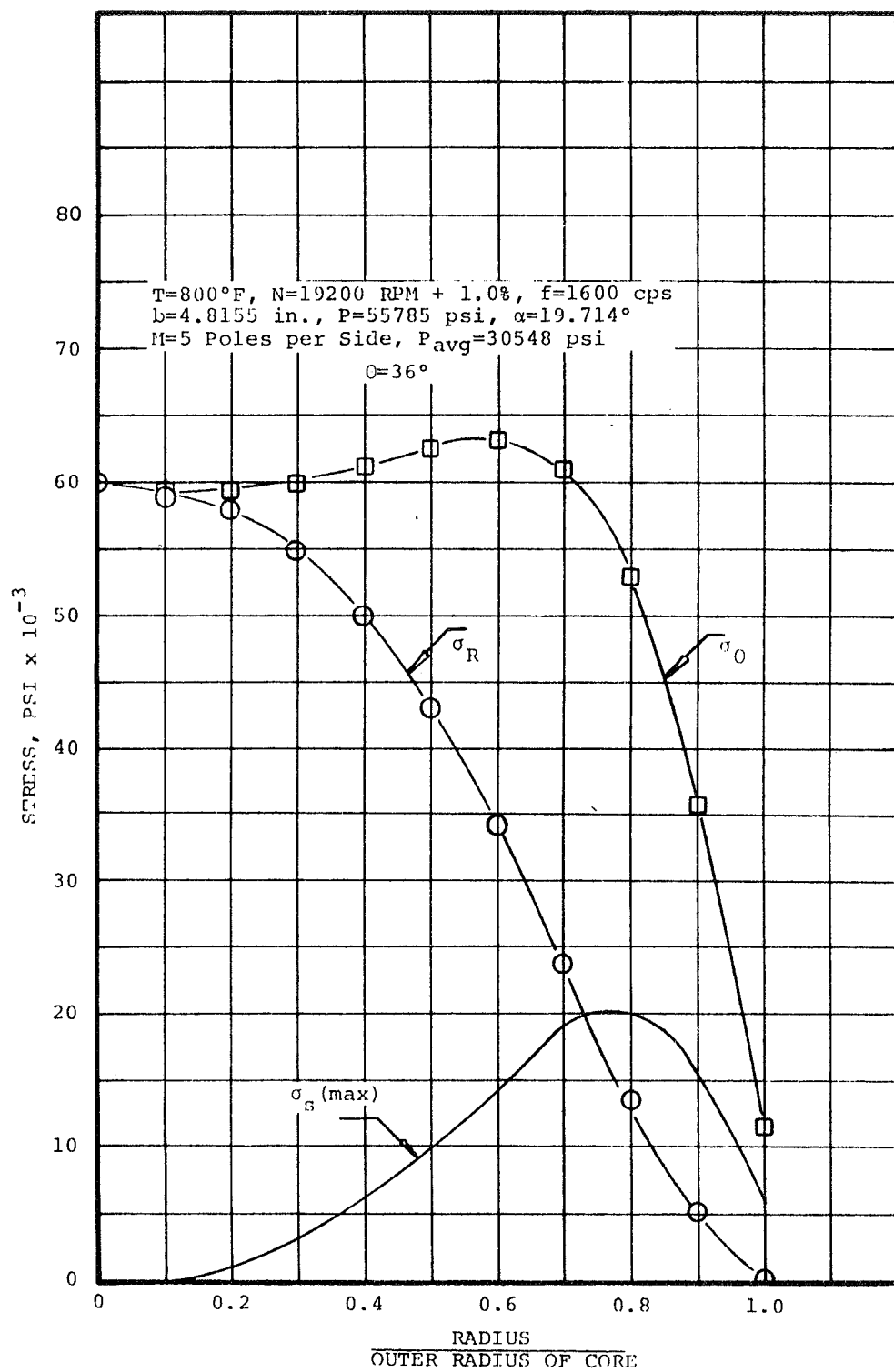


Figure 2.2-34. Rotor Stress at $\theta = 36^{\circ}$.

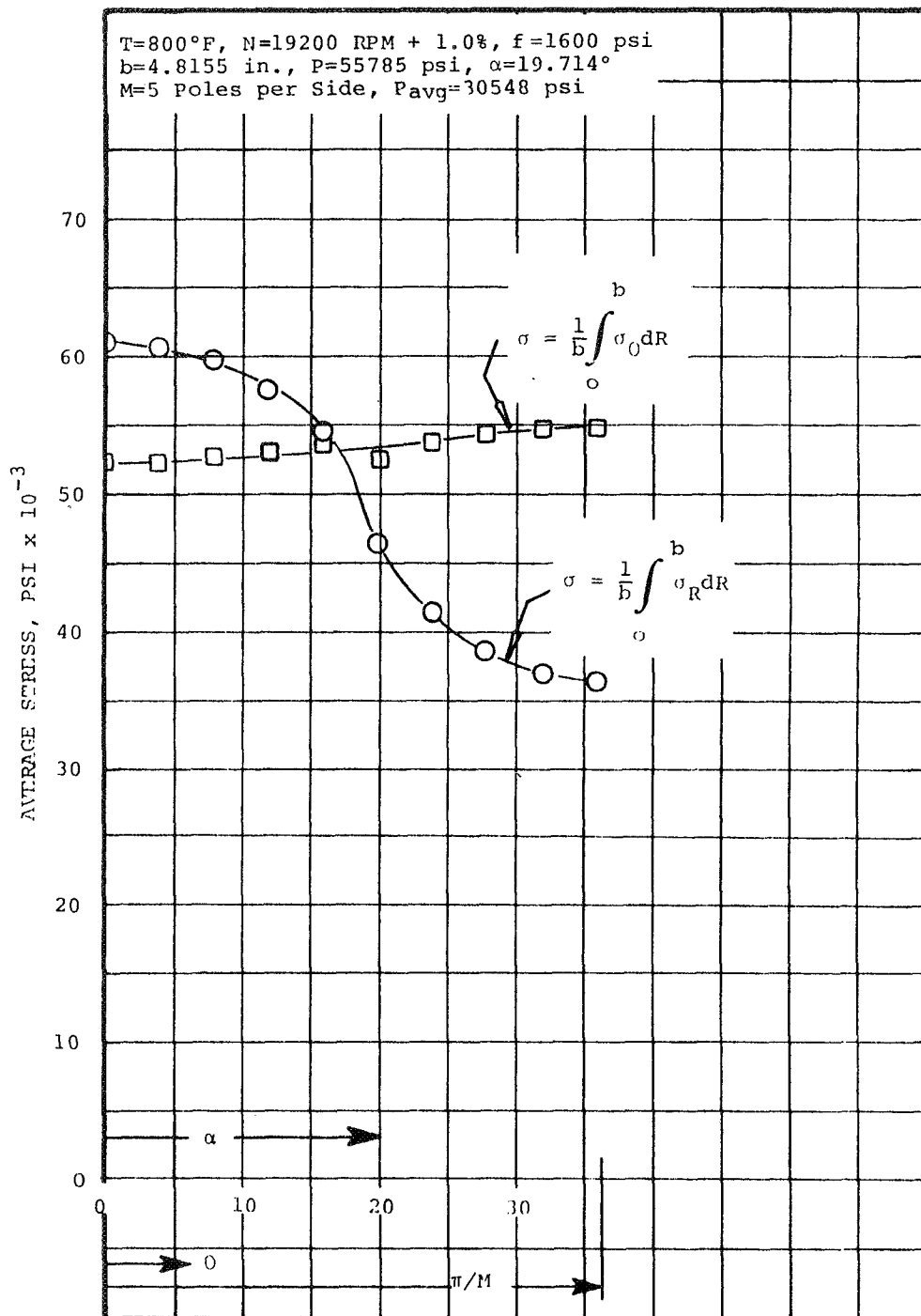


Figure 2.2-35. Rotor Stress (Average) as a Function of θ .

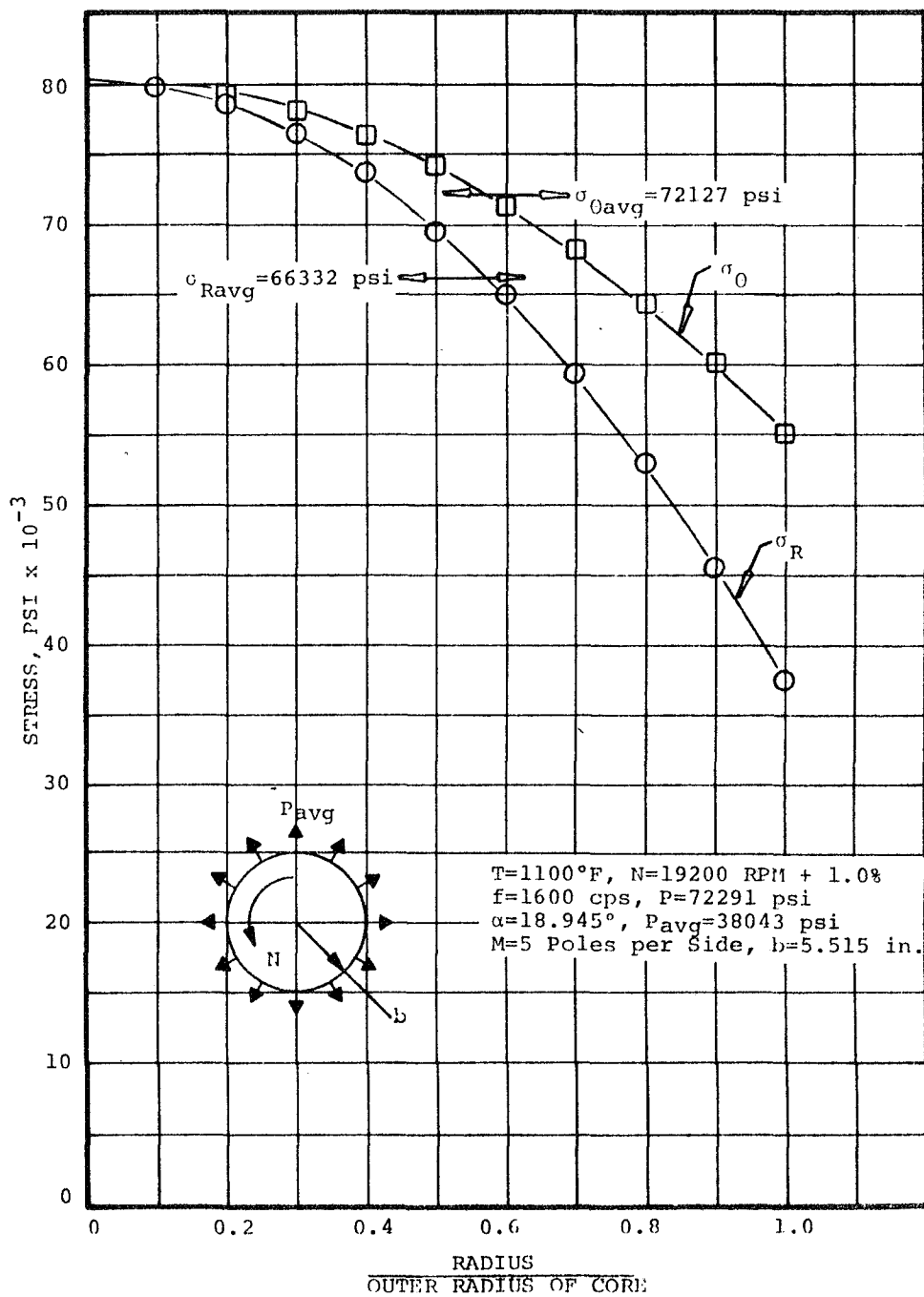


Figure 2.2-36. Rotor Stress at Uniform Load.

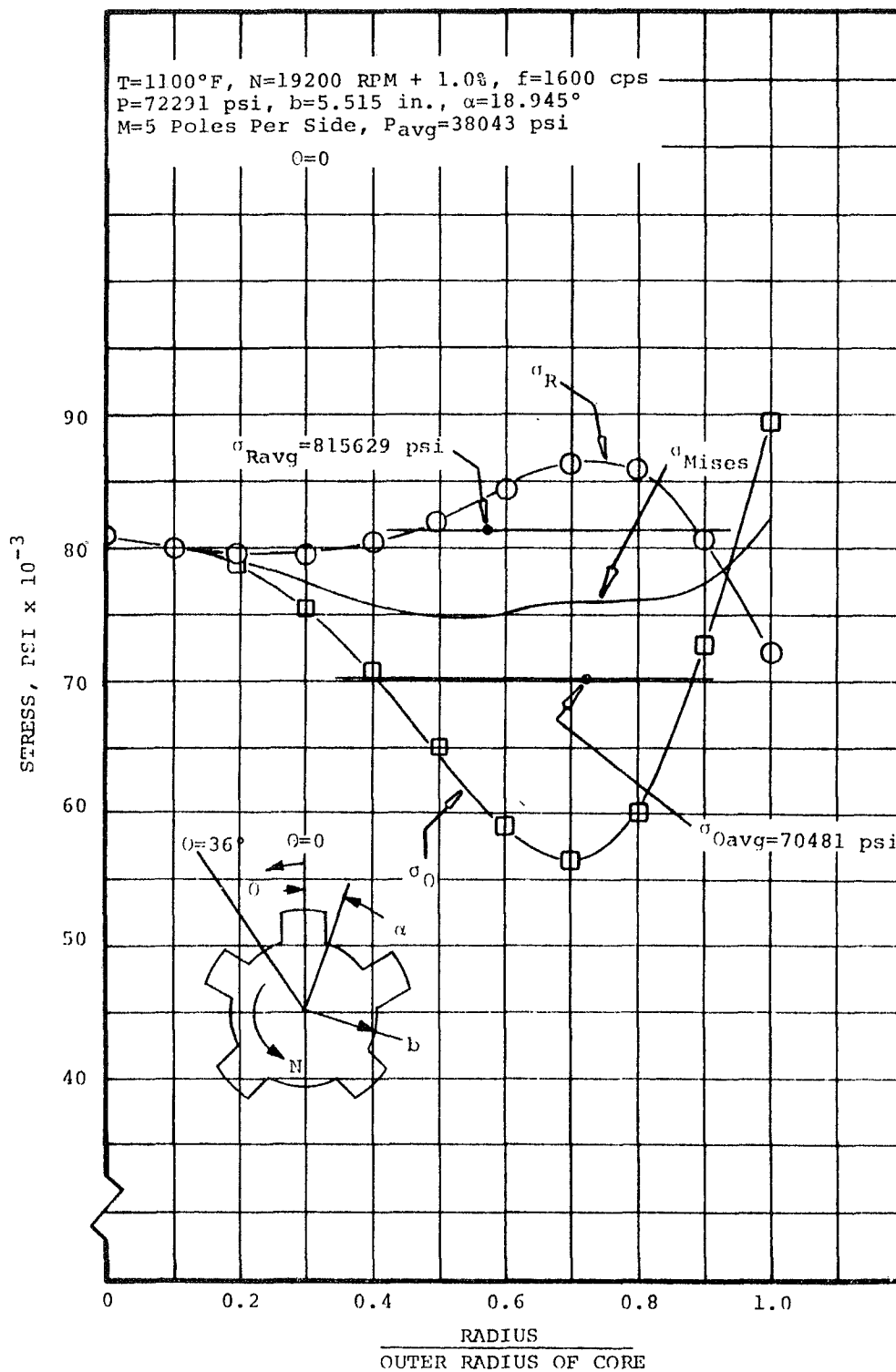


Figure 2.2-37. Rotor Stress at $\theta = 0^{\circ}$.

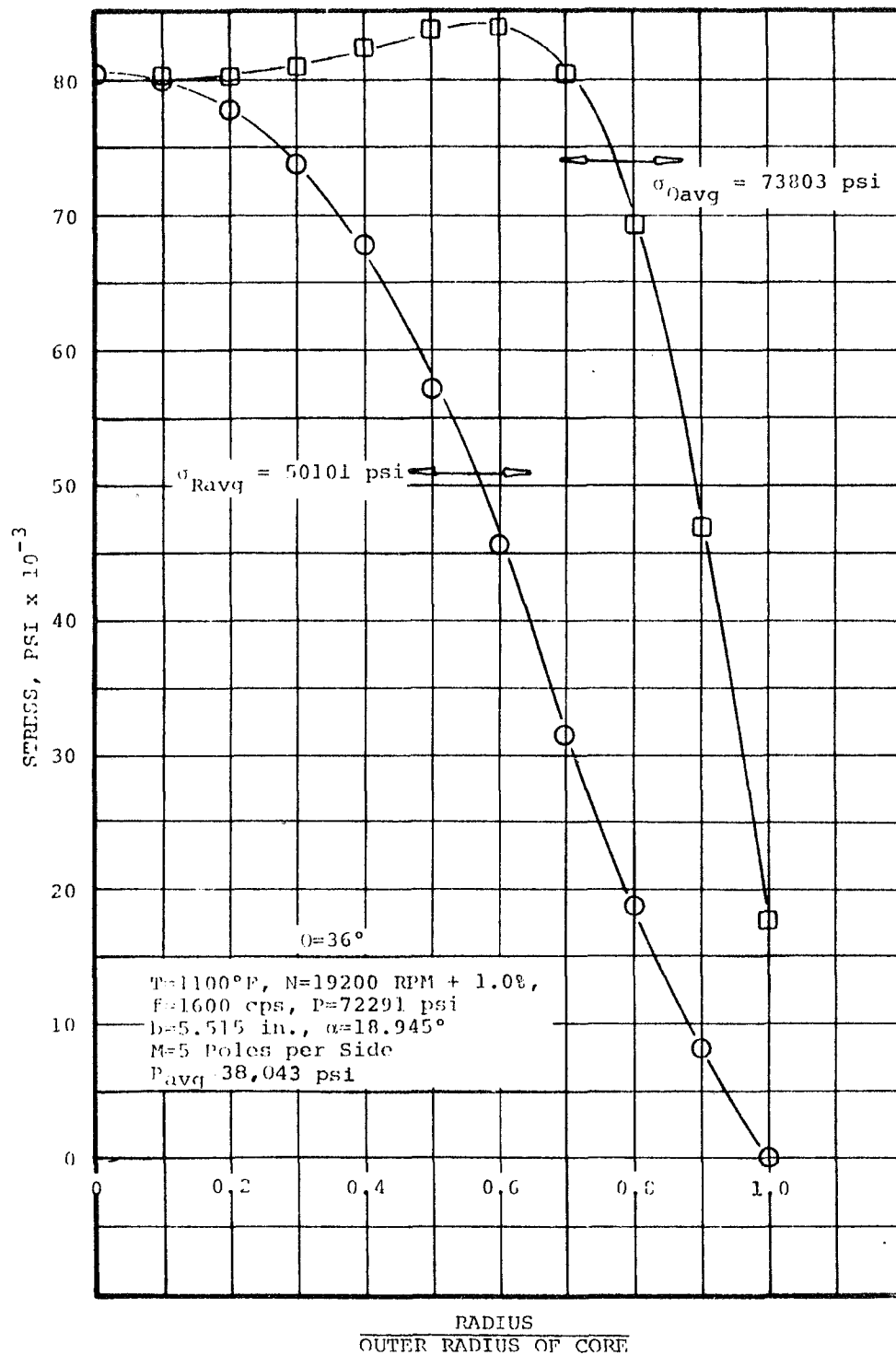


Figure 2.2-38. Rotor Stress at $\theta = 36^\circ$.

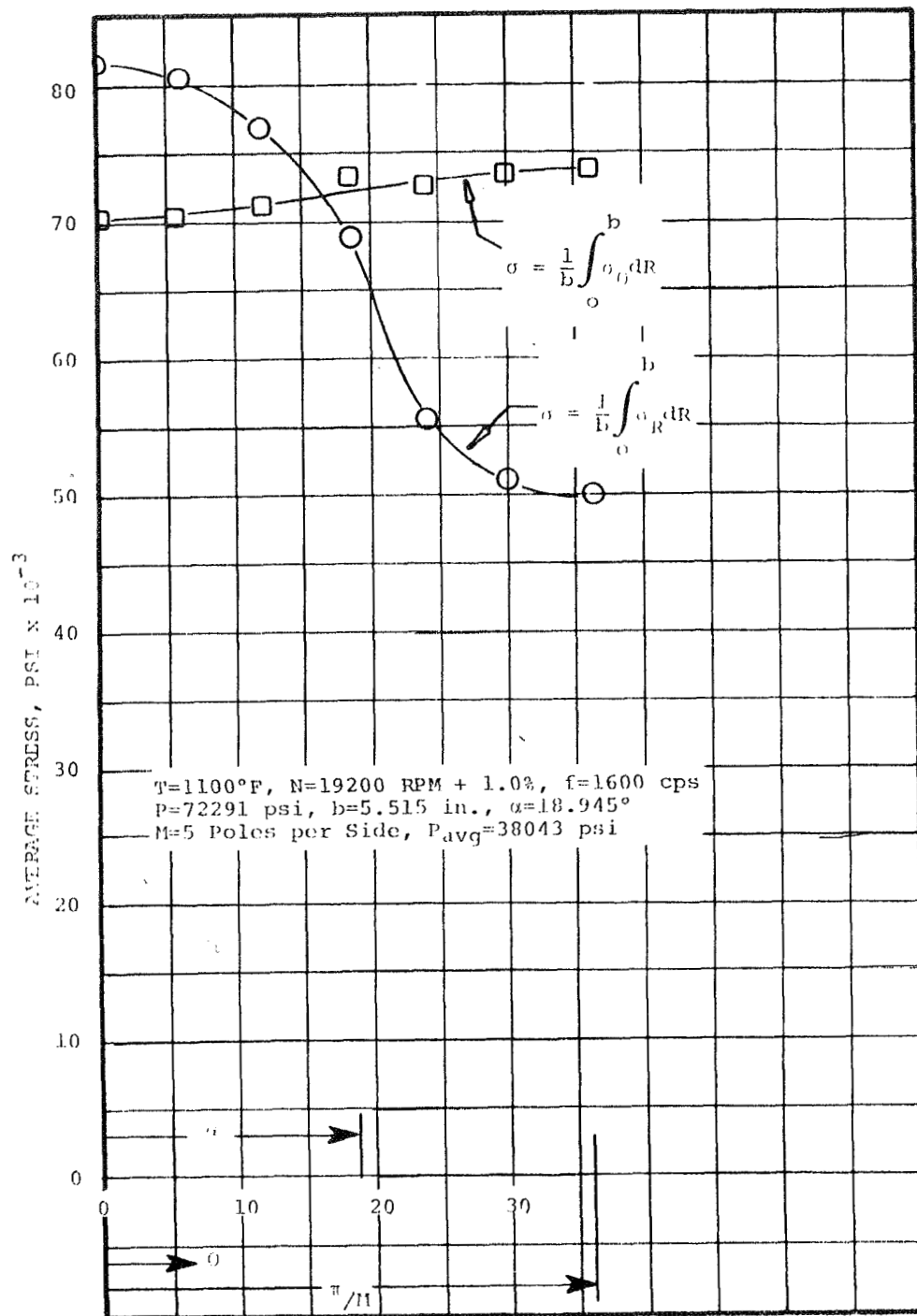


Figure 2.2-39. Rotor Stress (Average) as a Function of θ .

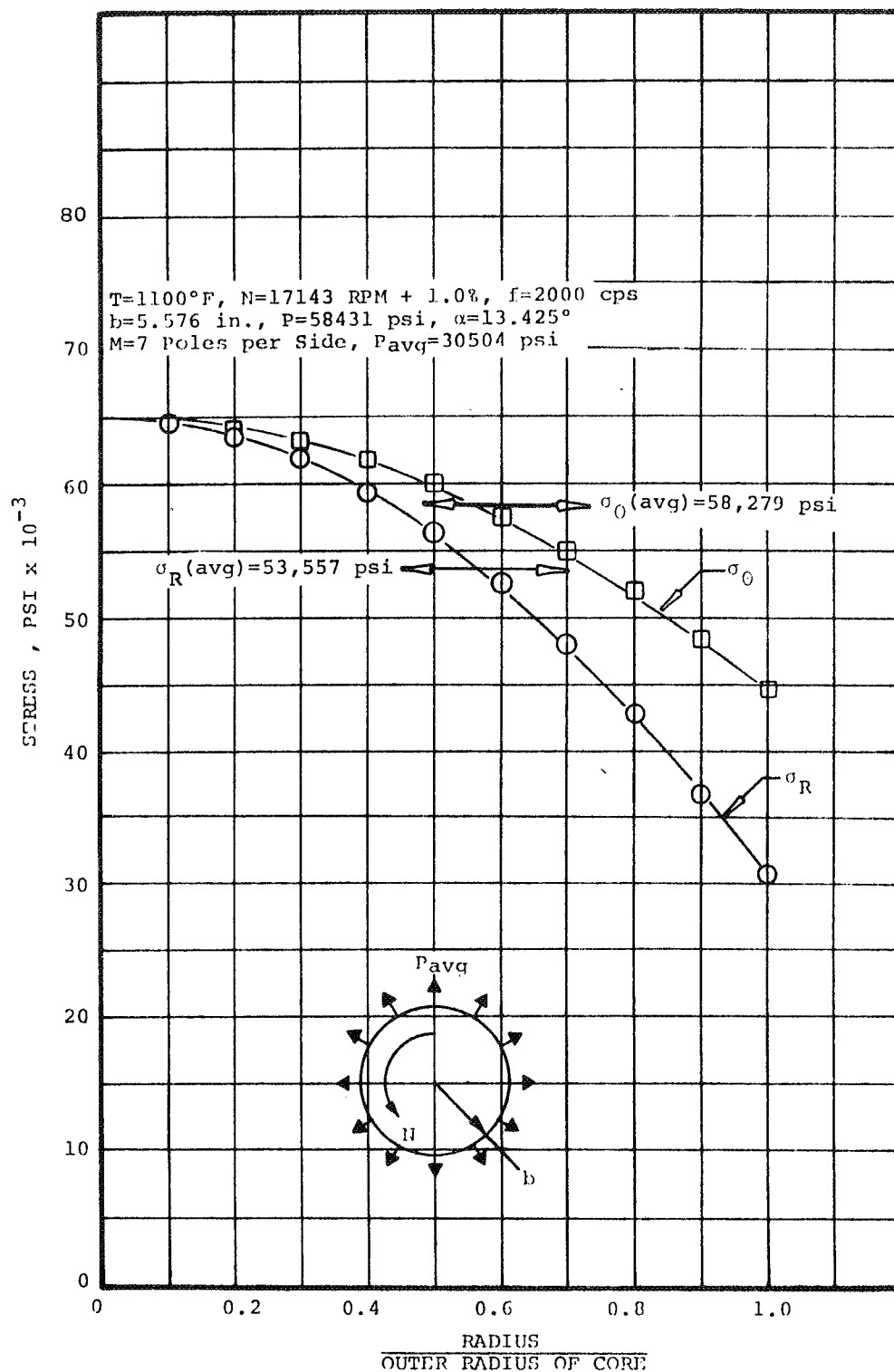


Figure 2.2-40. Rotor Stress at Uniform Load.

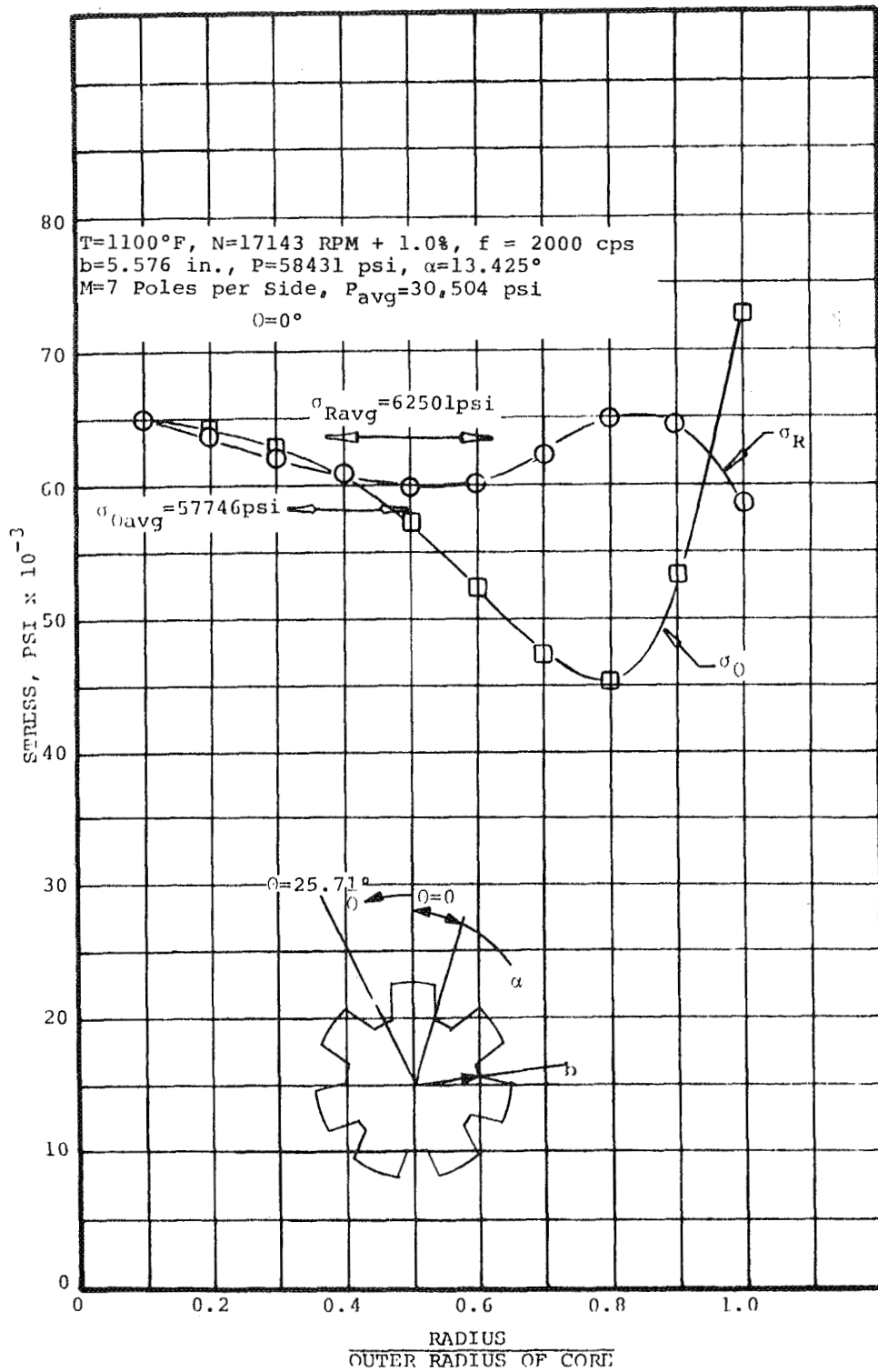


Figure 2.2-41. Rotor Stress at $\theta = 0^{\circ}$.

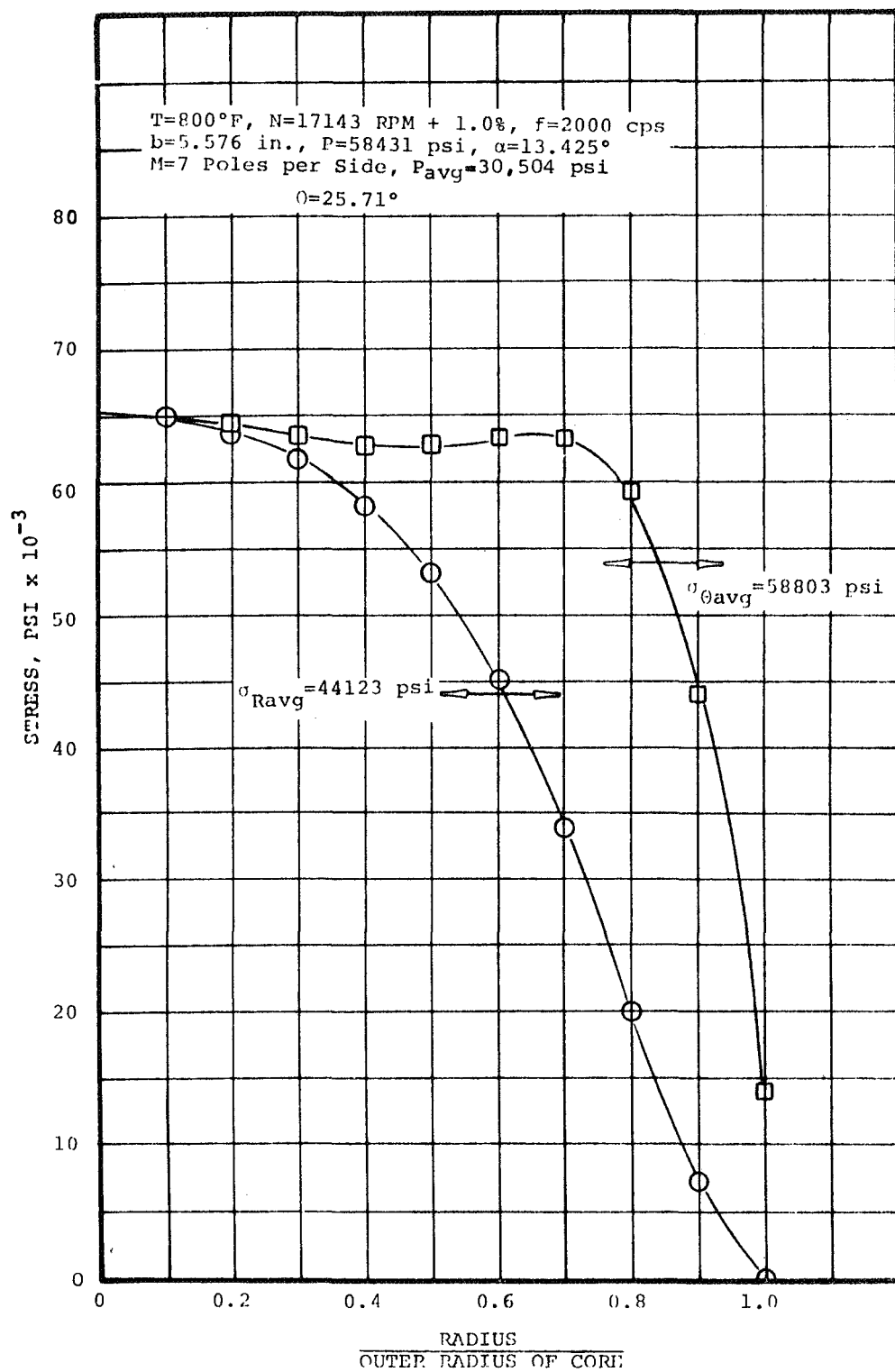


Figure 2.2-42. Rotor Stress at $\theta = 25.71^{\circ}$.

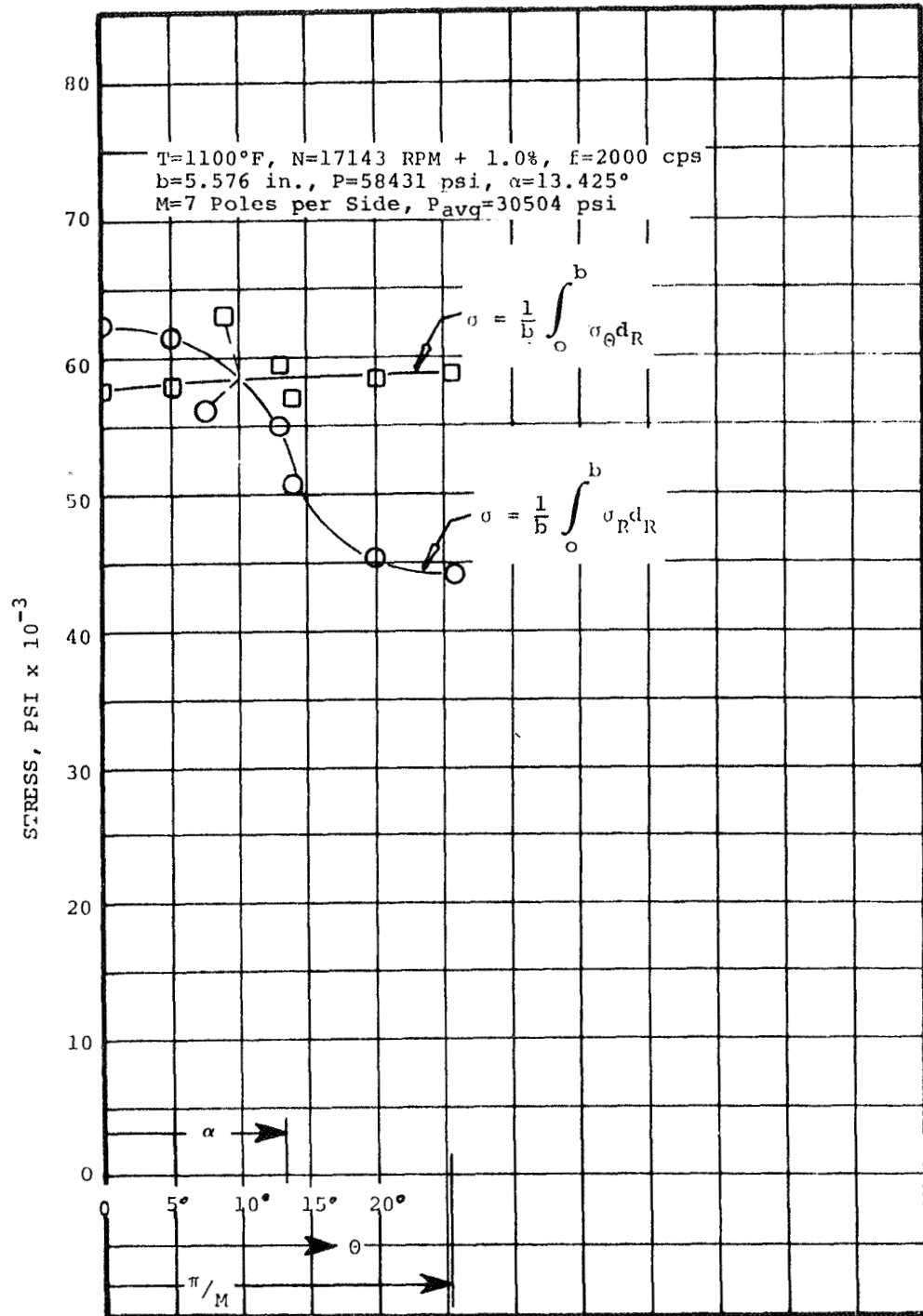


Figure 2.2-43. Rotor Stress (Average) as a Function of θ .

- (a) The stress within the rotors for the 800^oF, 30,000 rpm alternator exceed the allowable stress for H-11 steel.
- (b) The stress within the rotor for an 800^oF, 24,000 rpm alternator is within the allowable stress limit of H-11 steel. Similarly, the stress with the rotors for 800^oF and for a speed below 24,000 rpm is within the allowable stress limit of H-11 steel.
- (c) The stress within the rotor for a 1100^oF, 19,200 rpm alternator exceeds the allowable stress for the precipitation hardened, modified cobalt base alloy. The calculated stress intensity is approximately 83% of the average 0.2 percent offset strength of this material.

If one uses 72 percent of the average 0.2 percent offset strength as a limit, a material with a slightly higher yield strength would have to be developed for the application.
- (d) The stress within the rotor for an 1100^oF, 17,143 rpm alternator is within the allowable stress limit of the precipitation hardened, modified cobalt base alloy. Similarly, the stress within the rotor for 1100^oF and for speed below 17,143 rpm are within the allowable stress limit of this material.

Organic Lubricant Studies

Parametric screening Selection and Recommendations. In the selection of an organic coolant for the KTA alternator, several considerations must be taken into account. These generally include

its high temperature capabilities, its nuclear radiation resistance, its lubrication properties, its cooling capabilities, its vapor pressure and any problems associated with shaft seals, both vapor shaft seals to the rotor cavity and liquid shaft seals to the potassium in the turbine. Although as most of these considerations also affect the turbine, bearings, and shaft seals, the following remarks will be limited to those considerations unique to the alternator, namely the cooling capabilities and the vapor pressure. The cooling capabilities are unique in the area of the stator coolant where high heat flux are present and fluids giving a high film temperature drop are undesirable. The vapor pressure is unique as this causes windage losses in the alternator rotor cavity.

For the analysis of the cooling capabilities, a generalized approach was taken whereby the various film temperature drops were determined for a typical KTA alternator. The assumptions of the analysis were as follows:

Rating = 550 kWe

Efficiency = 91 percent

Total losses = 54 kW

90% of losses occur in stator = 48.5 kW

Bulk Fluid temperature rise through stator = 50^oF

Cooling Configuration = 60 tubes through frame, 3/16 O.D. by
0.032 inch thick, Single pass, straight
through, Tube wall 0.020 inch from frame I.D.

Analysis of this particular configuration results in a potassium film drop of about 1.0^oF within the tubes. Thus, the film drops calculated for the various organic fluids may be normalized relative to the potassium coolant for rapid comparison.

The summary results of the analysis are presented in Table 2.2-38 for the fluids considered. The mean temperatures shown were selected to bracket the range of likely use temperatures. The property data was obtained from supplier data sheets and are extrapolated in many cases.

As seen by the normalized film drop, the synthetic paraffinic has characteristics which would require much higher flow rates than the other organics in order to minimize the film drop. The same holds true for the fluorinated polymer at 400°F and the mineral oil at 200°F. The fluids which look the most attractive from the analysis are the two polyphenol ethers 5P4E and 4P3E. These must therefore be considered candidates relative to alternator cooling considerations.

As previously discussed in Section 2.2, a weighted film temperature drop was used in the alternator parametric screening studies. This was to permit the alternator temperature calculations to be made independent of any one particular organic fluid. The film temperature drops chosen were based on the analysis and data of Table 2.2-38 and are summarized below.

Mean Coolant Temperature (°F)	Weighted Film Temperature Drop, T, °F
200	105
400	35
600	26
800	25

After convincing oneself of the applicability or capability of an organic fluid to operate at 600°F or higher for three to five years despite the specified nuclear radiation level, one may be surprised to learn it is the vapor pressure which limits the operating fluid temperature in the alternator. This is because the high vapor pressures

TABLE 2.2-38

ORGANIC FLUID COOLING CHARACTERISTICS

Fluid	Mean Temp of	Specific Heat BTU/16° F	Required Mass Flow lb/sec	Density lb/in ³	Viscosity ² lb _m sec/in ²	Thermal Cond. BTU/hr ft ² F	Reynolds Number	Prandtl Number	Nusselt Number (2)	Normalized Film Drop ° F fluid / ° F potassium
Potassium (reference fluid)	800°	0.182	5.05	0.0269	0.2915×10^{-7}	22.65	77100	0.0039	6.38	1.0
SP4E	600°	0.529	1.74	0.0351	1.2×10^{-7}	0.0613	6430	17.3	79.9	30
Polyphenol Ether (Monsanto OS-124)	800°	0.593	1.55	0.0318	0.66×10^{-7}	0.0553 (1)	10410	11.8	100.5	26
Synthetic Paraffinic w/additives (Mobil XRM177F)	400° 600°	0.692 0.806 (1)	1.33 1.14	0.264 0.0245	6.5×10^{-7} 2.4×10^{-7} (1)	0.072 0.070	910 2120	107.3 50.5	18.0 18.2	115 125
Fluorinated Polymer (Dupont PRL43)	400° 600°	0.294 0.339 (1)	3.13 2.71	0.0566 0.0494	9.2×10^{-7} 2.4×10^{-7} (1)	0.0508 (1) 0.0489 (1)	1510 5030	88.8 27.8	20.2 80.0	141 37
4P3E (Meta) Polyphenol Ether	400° 600°	0.465 0.530	1.98 1.74	0.0377 0.0346	2.0×10^{-7} 0.9×10^{-7}	0.0765 0.075	4400 8600	20.3 10.6	63.0 82.8	30 23
Highly Refined Mineral Oil MLO-7277	200° 400°	0.521 0.613	1.77 1.50	0.0303 0.0277	12.0×10^{-7} 1.9×10^{-7}	0.075 0.0665	665 3510	139.5 29.3	18.4 54.2	105 40
(Humble 3158)										

(1) Extrapolated data

(2) For Potassium: $N_{Nu} = 0.652 (Pr)^{0.4}$ - Lubarsky & Kaufman EquationFor Re > 2300, Organic; $N_{Nu} = 0.023 (Re)^{0.8} (Pr)^{0.4}$ - Dittus & Boelter EquationFor Re < 2300, Organic; $N_{Nu} = 1.86 (Re)^{1/3} (Pr)^{1/3} (D/L)^{1/3} (u/u_s)^{0.14}$ - Sieder & Tate Equation where $(D/L)^{1/3} = 0.20$ and u_s is based on $T_s = T_{fluid} - 50^\circ F$

of the organic fluids at the higher temperatures can impose intolerable rotor windage losses on the rotor cooling system. In keeping with the self imposed limit of one kilowatt windage loss in the rotor cavity (Section 2.2), the emphasis on selection of a fluid must be directed toward one with a low vapor pressure.

In order to determine just what pressure is low enough (or too high), an analysis of the windage losses must be made for the various fluids considered. A quantitative comparison was made to illustrate which fluids might be best as well as the problems associated with them.

Various vapor conditions data gathered on the organic fluids is shown on Table 2.2-39. The data is scattered and incomplete for most vapors shown because of the general lack of its availability. Since the vapor viscosity properties are not generally measured by the manufacturers, recourse to the original suppliers was of no avail in attempting to obtain the missing data. As a result, kinetic theory of gas relations were employed to calculate the vapor viscosity. That is (2.2-12)

$$\eta = \frac{1}{3} \delta \bar{c} l$$

where

δ = density

$\bar{c}$ = mean velocity of molecule

l = length of mean free path of molecule between molecular collisions

The viscosities calculated are shown in the Table 2.2-39. The molecular weights used were derived from the basic chemical formulae for the two fluids; that is $C_{17}H_{36}$ for mineral oil and the five ring structure shown below for 5P4E.

TABLE 2.2-39

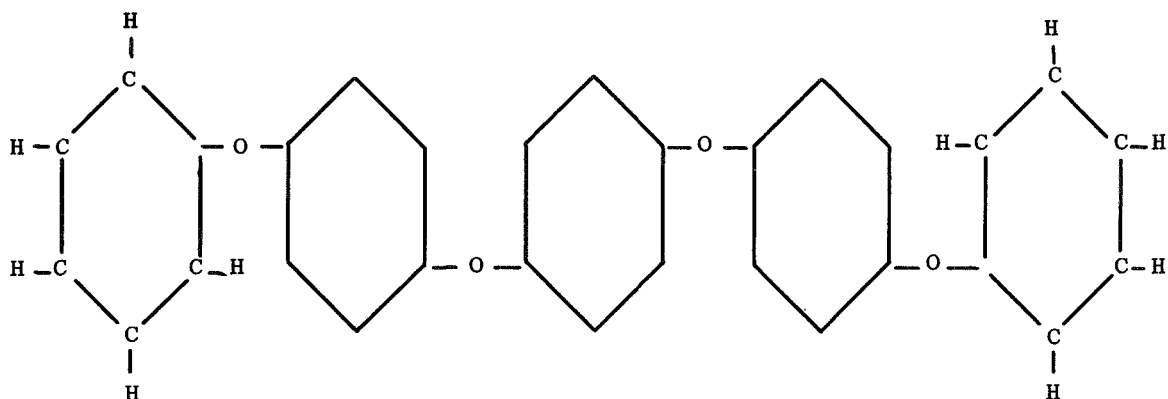
ORGANIC FLUIDS, VAPOR DATA COMPARISON

Fluid	Temp. °F	Saturated Vapor Pressure mm Hg	Saturated Vapor Viscosity lb _f sec/in ²
Potassium (reference fluid)	600°	0.58	0.159x10 ⁻⁸
	800°	8.8	0.183x10 ⁻⁸
	1100°	124.	0.216x10 ⁻⁸
5P4E (monsanto OS-124)	400°	0.093	0.0685x10 ⁻⁸ (2)
	600°	5.1	--
	800°	98.3	--
Synthetic Paraffinic (Mobil XRM 177F)	400°	--	--
	600°	--	--
Fluorinated Polymer (DuPont PR143)	400°	0.5	--
	600°	21.	--
4P3E (Meta)	400°	0.304	--
	600°	25.4	--
Highly Refined Mineral Oil (Humble 3158)	200°	0.0023 ⁽¹⁾	--
	400°	1.91	0.130x10 ⁻⁸ (3)

(1) Extrapolated Data

(2) Calculated, based on molecular weight = 446

(3) Calculated, based on molecular weight = 240



Any analysis of the effects of the various vapor pressures or windage should be viewed as a comparative first approximation since windage losses are a function of rotor speed and geometry as well as fluid properties. Also, with molecular pump shaft seals and rotor cavity scavenge system, the vapor pressure in the rotor cavity can be reduced to superheated pressures. This has the effect of reducing the windage losses to tolerable levels even though they may be intolerably high for saturated vapor. Of course, this latter statement is predicated on the condition of being able to place enough pumping capacity into the shaft seal configuration. For potassium vapor pressures much over 10 mm Hg, this is usually not possible and the same probability holds true for organic vapors.

At this point in the study, typical windage losses were calculated for mineral oil vapor and it was estimated that they could be kept less than 1 kW for vapor pressures as high as 0.005 psia. Assuming for a moment that the other organic fluids would result in losses similar to those for mineral oil, one can establish a relationship between maximum allowable vapor pressure and corresponding limiting temperature.

As seen by the data of Table 2.2-40, the fluid which has the lowest vapor pressure or maximum limiting temperature is the 5P4E. However,

TABLE 2.2-40

SATURATED VAPOR TEMPERATURE FOR $P_{sat} = 0.005 \text{ PSIA} = 0.258 \text{ mm Hg}$

5P4E - Monsanto OS-124	455 ^o F
Shell VRT Fluid E	465 ^o F
PWA 524	430 ^o F
Fluorinated Polymer - Dupont PR 143	370 ^o F
4P3E - Shell ET 378	390 ^o F
Monsanto MCS 210	380 ^o F
Mineral Oil - Humble 3158	340 ^o F

its maximum limiting temperature is still well below the desired 600°F to 800°F that would be needed if the oil were to be considered a competitive replacement for potassium.

The overall conclusion is that, for organic coolant and lubrication and due to windage loss considerations in the alternator rotor, bearing and shaft seal temperature levels of 400°F have to be used in place of 600°F or higher regardless of the thermal capabilities of the fluid. For this reason, the 400°F temperature was selected for the temperature to represent the organic coolant designs of the parametric screening studies.

Because of the need for 400°F coolant temperature (over 200° or 600°F) the final selection of the fluid to be used is somewhat flexible from the standpoint of the alternator alone; either the mineral oil, synthetic paraffinic or the polyphenyl ethers should work. Thus, the choice of which of these must be left to other criteria such as thermal stability or nuclear radiation resistance.

Over the course of three to five years of alternator life, organic fluids will progressively degrade due to thermal effects and exposure to nuclear radiation. In general, fluid degradation will result in three types of physical change:

- (a) Production of volatile decomposition products.
- (b) Changes in bulk viscosity of the fluid.
- (c) Formation and precipitation of oil-insoluble sludges.

Changes in bulk viscosity are of major immediate concern because of the direct effect on calculated bearing performance. This change is also an intermediate step in the process of sludge formation.

Viscosity changes will almost always be in the direction of an increase in viscosity, thus causing increased power losses in the bearings.

A large number of investigators have found that the effective fluid life at high temperatures can be increased many fold when oxidation effects are minimized by protection with an inert gas blanket in a closed system. Resistance of oils to radiation damage is also improved by a factor of at least two when air is excluded.

Super-refined petroleum oils are primarily straight-chain saturated hydrocarbons. Heat with or without pressure will produce cracking of the hydrocarbon chain. Increased pressure increases the amount of cracking. The products are one molecule of a saturated hydrocarbon and one molecule of an unsaturated hydrocarbon. The number of carbon atoms in each of two products will vary but the total of the two must equal the parent molecule. After several original molecules have broken, there will be a selection of unsaturated molecules of varied chain length available.

For example, a twelve carbon chain will split into molecules containing from one to eleven carbon atoms. The unsaturated hydrocarbon products are reactive and will tend to combine, producing chains of a wide variety of lengths. Thus, two long unsaturated fractions of perhaps eleven carbons would produce a new twenty-two carbon molecule. This new long molecule may be broken again at another location in the chain to produce additional active fragments. This sequence of reactions produces very short molecules down to the single carbon, saturated structure methane, and chains up to several times as long as the original. The small structures will have high vapor pressure and low viscosity. The heavy products will display high viscosity or will exist as a gummy solid.

The percentage of the various molecular weight products will be dependent upon the previously mentioned environment factors. Operation at 400°F will produce some breakdown of the oil as described above. Accurate prediction quantities and molecular weights of the products is not possible since all environmental parameters are not known quantitatively. The weight percent distribution of the aging products of a typical oil may be estimated to be as high as the following:

<u>Approximate Composition</u>	<u>Weight %</u>	<u>Nature</u>	<u>Sp. Gr. at 77°F</u>
Methane and Hydrogen	0.2,	Dissolved Gas	Up to 0.4
C ₂ to C	0.3	Dissolved Gas and Light Oil	0.4 to 0.7
C ₉ to C (original chain range)	97.5	Useful Oil	0.7 to 0.8
C ₂₀ to C ₆₀	2.0	Hard Oils and Gums	0.8 to 1.5

The original particle size of the high molecular weight gums will be very small but they tend to agglomerate. Pieces as large as several mils in diameter may be formed. These gums will tend to collect on metallic surfaces.

Polyphenyl ethers are cyclic molecules joined by oxygen atoms in an ether linkage. The cyclic or ring structure is much more resistant to heat and radiation than the straight chain petroleum molecules. The most significant change in performance characteristics is a reduction in fluid viscosity resulting from failure of the ether linkage. Typical specific gravity values at room temperature will be decreased by 800°F aging from 1.204 to perhaps 1.1. The 800°F values will decrease from 0.88 to about 0.8. Operation at 600°F will be much less severe, but

accurate estimation of fluid operating life is not feasible. Expectation of any long extended service life even at 600°F would be overly optimistic.

Organic Coolant Temperature Excursion Study. The objective of the organic coolant temperature excursion study was to determine if any specific design improvements could be attained by operating at a coolant temperature other than 400°F. The coolant temperatures chosen for investigation were 200°, 400°, and 600°F.

The organic coolant temperature excursion study results for the three coolant temperatures are presented in Table 2.2-41. The study was performed using the recommended frequency/speed/rotor material combination from the 400°F parametric studies. This combination was 1600 Hz, 24,000 rpm, and H-11 rotor material.

The temperature excursion had slight effects on generator design for constant frequency, speed, and rotor material. In general, generator weight increased slightly and generator efficiency decreased slightly with increasing coolant temperature. A 32,000 rpm generator design was made at the 200°F coolant temperature to determine any design advantages that might occur for the higher speed design. At this higher speed, the expected significant reduction in generator weight results; however, rotor stress slightly exceeded the established stress limit of the H-11 rotor material. The armature hot spot temperature and rotor radial temperature difference are both larger because the required slot combination caused higher pole face losses.

It was not considered necessary to evaluate a rotor material magnetically superior to H-11 at 200°F because of the results of the comparative evaluation made between Hiperco-27 alloy and H-11 in the

TABLE 2.2-41
ORGANIC COOLANT TEMPERATURE EXCURSION STUDY

Coolant	Average Coolant Temp. °F	Freq.	RPM	Gen. Elect. Weight lbs.	Gen. Eff. %	Voltage Unbalance %	Poles	Armature Hot-Spot Temp. °F	Rotor Radial ΔT , °F
Mineral Oil	200	1600	24000	640	92.0	7.01	8	779	95
Mineral Oil	200	1600	32000	528	91.8	8.93	6	959	354
Mineral Oil	400	1600	24000	643	91.8	7.09	8	943	95
Polyphenyl Ether	600	1600	24000	684	91.8	7.67	8	1123	133

400^oF parametric speed study and the results of the H-11, 32,000 rpm design discussed above. The 400^oF parametric speed study showed that the amount the rotor diameter can be reduced to save weight and reduce stress at a given speed (24,000 rpm in the case of the 400^oF speed studies) by using Hiperco-27, a superior magnetic material, is limited by stator design requirements. The same limitation applies to the 19,200 rpm designs. The design data presented in Table 2.3-2 shows that the Hiperco-27, 19,200 rpm design (Design No. 3) is overstressed while the H-11 19,200 rpm design (Design No. 4) is not. The allowable material stress of Hiperco-27 at the 200^oF coolant condition is only increased to approximately 57,000 psi from 55,000 psi. It is apparent from review of all the design data at 400^oF that the stator size with 200^oF coolant could not be reduced enough to permit reduction of rotor stress (by reducing rotor diameter) to much less than the 56,000 psi stress level of Design No. 3 without a significant penalty in efficiency. Thus, the 200^oF Hiperco-27 design would retain a marginal rotor design stress and only produce a weight saving over the H-11 design of slightly more than 15 pounds, efficiencies remaining the same. Finally, since the 200^oF, H-11 design is overstressed at 32,000 rpm, it is obvious that a 200^oF Hiperco-27 rotor would be overstressed at 32,000 rpm.

The temperature excursion study plus the designs at 800^o and 1100^oF for potassium provided data for an overall view of alternator weight and efficiency as a function of temperature. These data are plotted on Figure 2.2-44 and illustrate the increasing weight and decreasing efficiency trends that accompany increasing coolant temperature. The 1100^oF efficiency is misrepresentatively high because the windage losses were excluded from the total electrical losses. Because these trends

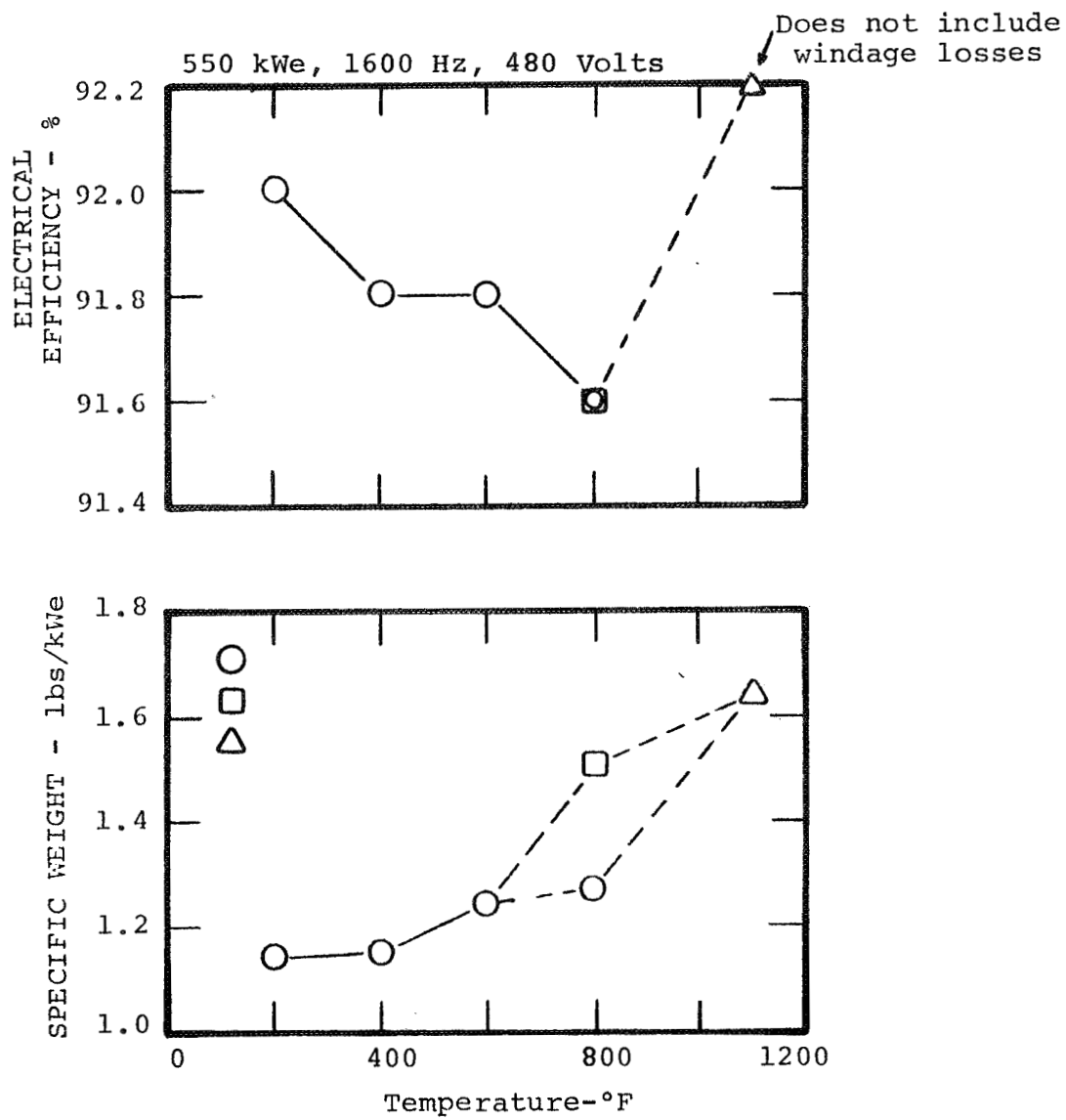


Figure 2.2-44. Weight and Efficiency Versus Coolant Temperature.

shown are contradictory to the decreasing weight of the cooling radiator as temperature increases, the system designer must evaluate the tradeoff between the two (radiator versus alternator) to arrive at a coolant temperature giving their lowest combined weight.

2.3 ALTERNATOR BASE DESIGNS

With the completion of the parametric screening studies and as a result of design study review meetings with the prime contractor, three alternator base design coolant/speed/frequency conditions were selected for the Phase I preliminary alternator designs. These constituted the three base designs of Phase I.

The purpose behind these three designs was that preliminary realistic designs were needed during the Phase I design studies prior to the definition of the conceptual designs (which were to be the result of the Phase I studies). The base designs were needed to:

- (a) Complete the alternator conceptual design studies using realistic configurations; this is to avoid doing non-essential academic analyses or an excessive number of analyses that would be required if all possible KTA alternator design configurations were considered. See Section 2.4 for explanation of what these analyses consist of.
- (b) To permit the prime contractor to carry out the bearing selection, bearing design and rotor dynamic analyses at realistic conditions.

- (c) To permit the prime contractor to carry out the KTA integration and shaft bearing arrangement studies using realistic alternator configurations.

The base designs were selected from the groups of preferred configurations determined in the course of the parametric screening studies. These preferred configurations are summarized in Tables 2.3-1 and 2.3-2.

Shown in Table 2.3-1 is a summary of the alternator designs for coolant temperatures of 800° and 1100° F. In the first column, the previous order of electro-magnetic preference is restated. The data substantiating these preferences for 800° and 1100° F coolant temperature is discussed in the previous section on parametric screening. Shown also in the table are the rotor material, the allowable stress and the type of limiting stress for each design. The most crucial stress occurs under the rotor pole and is a biaxial stress that is related to the uniaxial material stress properties by the equivalent (or Von Mises) stress at the center of the base of the pole. This combined stress also is shown in Table 2.3-1 along with a safety factor defined as the ratio of allowable to Von Mises stress. A rotor design is unacceptable when this safety factor drops below unity. Shown in the last column of the table is the order of overall preference among the alternator designs for both 800° and 1100° F coolant temperatures.

At a coolant temperature of 800° F there is no acceptable configuration at 30,000 rpm or above. The 19,200 rpm configuration is preferred over the 20,000 rpm configuration because the latter requires four conductors per slot as discussed previously. Both 24,000 rpm designs are acceptable in third and fourth preference but

TABLE 2.3-1
SUMMARY OF ALTERNATOR DESIGNS

Order of EM Design Preference	Coolant Temp., °F	Speed, rpm	Frequency, Hz	No. of Poles	Gen. Wt., lbs.	Gen. Eff., %	Conductors Per Slot	Volt Unbal., %	Rotor Radial ΔT, °F	Stator Hot Spot, °F	Rotor Wt., lbs.	Rotor Material	Allowable Stress, psi	Type	Maximum Von Mises Stress Under Pole, psi	Safety Factor	Order of Overall Preference
1	800	24,000	1600	8	701	91.6	2	7.78	95(1)	1189	274	H-11 R''C''45	97,000(2)	0.5% Creep	90,200	1.075	3
2		30,000	2000	8	566	90.4	2	8.32	106	1203	214				127,000	.764	Unacceptable
3		24,000	2000	10	676	90.6	2	7.32	50	1157	275				89,000	> 1.0	4
4		19,200	1600	10	844	91.7	2	6.77	45	1161	353				62,800	1.545	1
5		20,000	2000	12	761	91.0	4	5.78	383	1148	336				70,600	1.372	2
1	1100	30,000	2000	8	622	92.4	2	8.16	231(3)	1438	287	1BS1	71,000	Yield	---	< 1.0	Unacceptable
2		24,000	1600	8	770	92.7	2	7.68	242	1434	368				---	< 1.0	
3		24,000	2000	10	749	92.3	2	7.07	107	1413	368				---	< 1.0	
4		19,200	1600	10	935	92.7	2	6.57	96	1411	471				82,000	0.865	
5		17,143	2000	14	868	92.0	4	8.95	112	1447	450				66,600	1.066	1
6		16,000	1600	12	905	92.2	4	9.06	202	1431	457				---	> 1.0	2

(1) Values less than approximately 150°F indicate that the rotor can be of a one-piece, slotted-pole construction without coolant passages extending into the pole.

(2) Yield stress 99,000 psi.

(3) Values less than approximately 200°F indicate that the rotor can be of a one-piece, slotted-pole construction without coolant passages extending into the pole.

TABLE 2.3-2

SUMMARY OF 400°F ALTERNATOR DESIGNS

Order of EM Design Preference	Speed, rpm	Frequency, Hz	No. of Poles	Gen. Wt., lbs.	Gen. Eff., %	Conductors Per Slot	Voltage Unbalance, %	Rotor Radial ΔT^* , °F	Stator Hot Spot °F
1	24,000	1600	8	643	91.8	2	7.09	95	943
2	24,000	1600	8	626	91.6	2	7.97	39	928
3	19,200	1600	10	762	91.7	2	6.42	26	853
4	19,200	1600	10	777	91.7	2	6.23	44	862
5	30,000	2000	8	505	90.7	2	7.58	107	975
6	32,000	1600	6	533	91.0	2	9.00	356	1108
7	20,000	2000	12	664	91.5	4	6.06	113	849

Order of EM Design Preference	Rotor Wt., lbs.	Rotor Material	Allowable Material Stress, psi	Type of Limiting Criteria	Maximum Von Mises Stress Under Pole, psi	Safety Factor	Overall Order of Preference
1	235	H-11	118,000	Yield	84,700	1.395	1
2	207	Hiperco 27	55,500	Yield	--	< 1	Unacceptable
3	290	Hiperco 27	55,500	Yield	56,000	0.991	Barely Unacceptable
4	305	H-11	118,000	Yield	--	< 1	2 Unacceptable
5	182	H-11	118,000	Yield	119,000	0.990	Barely Unacceptable
6	180	H-11	118,000	Yield	139,500	0.845	Unacceptable
7	253	Hiperco 27	55,500	Yield	--	< 1	Unacceptable

* Values less than approximately 550°F for H-11 or 400°F for Hiperco 27 indicates that a one piece, slotted pole construction configuration without coolant passages directed up into the poles might be possible for the rotor. This does not mean that rotor coolant passages can be eliminated from the rotor entirely. Whatever the 400°F organic coolant passage arrangement, the design of 400°F organic coolant passage will be considerably more difficult than potassium coolant design because of problems introduced by use of organic coolant itself.

are nearly stress limited. If the coolant temperature were reduced to 600°F, the 24,000 rpm, 1600 Hz design would become the first preference both from a standpoint of electrical design and stress.

At a coolant temperature of 1100°F only two configurations are acceptable on the basis of pole center plane stress, namely the 17,143 and 16,000 rpm configurations. Both of these configurations require four conductors per slot with all the attendant undesirable features described previously.

As pointed out above in the 400°F coolant studies, four speed studies were required to consider both candidate rotor materials, i.e., Hiperco-27 and premium quality H-11 at frequencies of 1600 and 2000 Hz. As seen in Table 2.3-2, the results show that the preferred combination of speed, frequency and rotor material for the 400°F coolant temperature (based on generator electromagnetic design considerations and minimum system weight with an assumed specific weight of 100 pounds per kilowatt of loss) is 24,000 rpm at a frequency of 1600 Hz for either H-11 or Hiperco-27 rotor alloy. The first alternate choice is 19,200 rpm at a frequency of 1600 Hz with either a Hiperco-27 or H-11 rotor material. The Hiperco-27 generator design is slightly lighter (15 pounds) but does not have as large of stress safety factor as does the H-11 design. The second alternate choice is 30,000 rpm, 2000 Hz, H-11 rotor material and the third alternate choice is 32,000 rpm, 1600 Hz, H-11 rotor material. The 30,000 rpm, 2000 Hz, Hiperco-27 rotor material design is obviously overstressed and therefore was excluded from electrical-mechanical design preference consideration. The 16,000, 17,143, and 20,000 rpm designs have four conductors per slot and this feature has a deleterious effect on the ease of winding development and reliability.

The Base Designs were selected as a result of a combined NASA/General Electric/Westinghouse meeting at which the various alternative parametric designs were reviewed. The first design selected was for the 800^oF potassium coolant and was the 19,200 rpm/1600 Hz design. This design had a speed compatible with the desired turbine speed, had the desired two conductors per slot configuration and resulted in conservatively low rotor stress. This latter reason was cause for not selecting the electromagnetically preferred 24,000 rpm/1600 Hz design. While the 24,000 rpm design had stresses within those allowed for normal operation, the stresses at a sustained 20 percent overspeed condition were in excess of the 0.5 percent creep limit and 0.72 of yield strength limit. The maximum stress was still below the 0.2 percent of ultimate stress, however, so that catastrophic failure should not have occurred for any short term at overspeed. Along these same lines it was because the H-11 rotor material is not creep limited at 400^oF coolant conditions that it was possible to select the 24,000 rpm/1600 Hz design for the second of the three Base Designs. Note this is the only one of the three base designs where the electromagnetic and overall order of preferences coincided, see Table 2.3-2.

Since there appeared to be no good designs at the 1100^oF coolant conditions, the choice of the third Base Design was more difficult. After several discussions involving such alternatives as (1) the 24,000 rpm/1600 Hz/800^oF design and (2) the same with 800^oF stator and 600^oF rotor coolants, it was decided upon a 19,200 rpm/1600 Hz/1100^oF stator/800^oF rotor coolant combination. This choice was based on recognition of the fact that the use of split temperatures in the stator and rotor offers certain advantages to the analytical studies of Phase I,

realizing of course, that in a practical sense there might be physical disadvantages.

From the analytical point of view, a design with a coolant temperature of 1100°F in the stator and 800°F in the rotor offers stator design concepts for high temperature operation but does not limit the shaft speeds to 16,000 or 17,143 rpm. The rotor could be H-11 and still operate safely at 19,200 rpm. With such a configuration, it is possible to consider alternator speeds compatible with the turbine speeds and still be evaluating high temperature design concepts. The particular direction of the temperature split is important also because about 80 to 90 percent of the alternator losses occur in the stator which makes it desirable to operate the stator as hot as possible to minimize radiator weight. Other possible intermediate temperatures between 800° and 1100°F were not considered because of the following basic reasons which were also previously discussed somewhat in Section 2.2.

Choosing a 900°F or 1000°F coolant temperature in place of the 1100°F coolant would not affect the choice of alternator design speed significantly, the reason being that the basic materials would not change. In particular, the rotor material selected for the 900° or 1000°F designs would still be the same 1BS1 alloy and it would still not have enough strength at 900°F to permit operation at speeds of 19,200 rpm or above. The stator design concepts would be of no specific significance to a speed or frequency choice for either 900° , 1000° , or 1100°F because there, the most significant changes would be in the cooling configurations or armature wire selections.

Recall that it was primarily because of these considerations that the parametric screening was done at 1100°F instead of 900° or 1000°F .

The 1100°F designs would contain any features of the 900° or 1000°F designs plus any additional features for the added one -or - two hundred degrees Fahrenheit. In other words, for the analytical purposes of the Phase I studies, 1100°F was the logical technical choice for conceptual study. In a final phase hardware study, this may not be true to non-alternator considerations and as a result, any temperature between 900° and 1100°F could be picked; this would be perfectly all right from the alternator design standpoint.

Recall also that it was previously stated there is a breakpoint in the KTA alternator temperature range and it occurs in going from 800° to 900°F. Thus, what held previously for going from 1100° to 900°F does not hold for going from 900° to 800°F. At 800°F coolant, one can change the rotor material to H-11 alloy which has greater strength and magnetic capabilities (than NIVCO or LBS1 for instance), thus allowing speeds of 19,200 rpm or possibly 24,000 rpm.

In going from 800° to 700° to 600°F, the same general remarks in the first paragraph above hold. That is, 800°F designs are applicable to 600°F designs.

The three base design conditions selected are summarized in Table 2.3-3 and are hereafter identified as Design Number 1, 2, or 3.

The detail design aspects of these three designs are discussed in the following section on conceptual designs.

Note also from Table 2.3-3 that at this time in the study, mutual agreement had been reached as to which organic coolant/lubricant would be used, it was the 5P4E fluid.

TABLE 2.3-3

BASE DESIGN IDENTIFICATION

No. 1	800 [°] F Coolant, Alkali Metal (NaK or K) 19,200 rpm 1600 Hz 480 volts (L-N)
No. 2	400 [°] F Coolant, 5P4E 24,000 rpm 1600 Hz 480 volts (L-N)
No. 3	1100 [°] F Stator Coolant, NaK or K 800 [°] F Rotor Coolant, K 19,200 rpm 1600 Hz 480 volts (L-N)

2.4 CONCEPTUAL DESIGN ANALYSES

Introduction

There are certain aspects of the computer design analyses used in the parametric screening studies which, for the sake of either expediency or design flexibility, are first approximations. They may be used as first approximations because they have secondary influence on optimizing the basic electromagnetic design. The main areas concerned and the rationale for their being first approximations are summarized below.

Rotor Stress: first approximation determines the average tangential core stress and pole root radial stress to indicate magnitude of stress levels. Final stress calculations dependent upon cooling configuration, stress concentrations and thermal gradients; these items are subject to too many influences to permit practical application to generalized computer design program.

Rotor Cooling Configuration and Temperature: first approximation determines preselected point temperatures based on three alternative cooling configurations and recommends which is best based on maximum allowable pole tip temperature and coolant temperature. Final design selection and analyses subject to external and non-programmable influences, that require separate consideration, e.g., fabrication requirements.

Pole Face Losses: program size limitations require the generator design program to use a first approximation of a more elaborate program that calculates pole face losses in detail.

Windage Losses: first approximation uses input data that does not recognize loss reductions possible with "superheating" the vapor nor does it account for anything more than estimated geometry influences. It is a simple matter to calculate these losses separately once electro-magnetic design and shaft seal designs are determined.

Stator Cooling Configuration and Temperature: because of program size limits, first approximation uses coarse nodal grid to determine temperature distributions based on a fixed heat sink design. Final temperatures must be recalculated on finer nodal grid once actual cooling configuration is selected.

Stator (Structural) Stresses: too many influences to permit programming into a generalized design program of practical size.

It is the purpose of the present section, Conceptual Design Analyses, to present these separate studies and the results derived from them with one exception. That one is the rotor stress results already presented in Section 2.2. The discussions will be segregated into two areas, Rotor Studies and Stator Studies.

Rotor Studies

The basic purposes of the rotor studies were to examine the alternatives available for reducing the rotor losses (windage and pole face), evaluate the type of cooling configurations that may be required or are best suited to the design and losses, and to make recommendations for the final conceptual designs. The goal was to determine the electro-magnetic configuration giving sufficiently low losses to minimize the

cooling configuration complexities. Both the electromagnetic and cooling configurations were to be consistent with achieving a highly reliable design requiring the least development. The rotor cooling concept studies will be discussed first.

Rotor Cooling Concepts. The basic design philosophy associated with the KTA alternator rotor cooling was to minimize the losses such that, for a given coolant temperature and maximum allowable pole tip temperature, the concepts required would be consistent with least complex configurations having inherently high reliability. One of the least complex configurations is a solid rotor with the stub shaft surface area in the bearing cavity acting as the heat sink. A slightly more reliable configuration that is almost as simple is a central coolant hole bored through the center of the rotor. The latter is a more effective cooling configuration that can be easily simulated with an accurate analytical design model. It does not require separate rotor flow; it can use bearing flow and will pump at its own desired coolant flow rate by taking rotational energy from the rotor. Rotor reliability is enhanced by the central coolant hole as the center of the rotor, where inclusions could occur, is removed. Also the hole is needed to obtain uniform hardness in the core during heat treatment.

Other possible rotor coolant path configurations that are more complex include (1) individual, parallel paths (small coolant holes) through the core and/or poles, (2) radial conducting rods in the core and/or poles, and (3) heat pipe configurations. The individual, parallel coolant holes are a very effective cooling sink but suffer from fabrication complexities required to plug the ends of the drilled passages. They also suffer from operating reliability due to centrifuging

of solids in the outer passages. The design must accommodate these solids as they accumulate. The radial conducting rods add fabrication complexities but do not add much to cooling effectiveness (2.2-11). The heat pipe concept, as presented in the originally proposed KTA alternator, is a very effective cooling concept but is inconsistent with the "ease of development" design philosophy, especially in light of the fact that a very effective cooling configuration was not needed in the KTA alternator. The radial conduction rods and heat pipe concepts were not analyzed in the KTA alternator.

The analyses were done on typical KTA configurations - Base Design No. 1 and No. 2, Section 2.3 - and the results were normalized to degrees Fahrenheit temperature drop per unit loss in watts. The temperature drop was that in going from the pole tips to the rotor coolant. A summary of the normalized gradients is presented in Table 2.4-1 for each of the cooling configurations considered. With these normalized temperature gradients, it was possible to compare the multitude of watts loss, coolant temperature and limiting temperature combinations of the studies and draw conclusions on applicability.

Pole Face Loss Minimization Studies. The designer of the alternator must cope with the problem of reducing the rotor pole face losses that are produced by the time dependent component of induction in the face of the poles. There are two sources of induction from the stator which are time dependent relative to a reference point in the rotor pole. The first source is produced by the teeth and slots in the armature. At one instant, a point on the pole face is under a tooth where the flux density is a maximum and at the next instant of time this point is under a slot where the flux density is a minimum. This

TABLE 2.4-1

ROTOR COOLING CONCEPTS SUMMARY

Configuration	Coolant, °F	°F/Watt	Maximum Allowable Pole Face Limit* Losses for 900°F Tip Temperature
1. Stub Shaft	800°-k	---	---
	400°-5P4E	2.81	178 watts
2. Central Hole	800°-k	0.181	553 watts
	400°-5P4E	0.200	2500 watts
3. Parallel Passages Through Core	800°-k	0.0627	1600 watts
	400°-5P4E	0.0801	6250 watts
4. Parallel Passages Extending into Pole Bodies	800°-k	0.0181	5530 watts
	400°-5P4E	---	--- watts
* Neglects windage losses and heat carry over to or from the stator components.			

varying flux produces EMF's which in turn produce eddy-current losses. The varying flux also produces minor hysteresis loops within the pole face which cause a further loss of energy. The second source of time dependent induction results from harmonic MMF's associated with the armature windings.

Certain internal design variables can be varied to reduce the pole face losses. Those which have a significant influence on the losses are the radial gap, the ratio of the width of the slot opening to the tooth pitch*, pole pitch, and the direct axis synchronous reactance of the machine. Most of these were evaluated as part of the internal variables parametric screening studies. In general, the parameters can be varied to reduce the amplitude of flux density variations on the surface of the pole face. However, performance penalties can be incurred in the overall design when these parameters are optimized only from the standpoint of pole face losses. One is therefore obliged to consider other means which will also reduce the pole face losses.

It is generally true, unfortunately, that the materials which offer the best high temperature/strength properties for a rotor have inferior magnetic properties from the standpoint of the reduction of pole face losses. For example, the pole face losses in NIVCO alloys will be 2 to 3-fold greater than the losses in H-11 steel and the losses in H-11 are approximately 8 to 10-fold greater than the losses in a CUBEX magnetic alloy. In some cases, the superior creep strength property

* Special slot configurations to reduce pole face losses are presently a standard part of the inductor alternator design at Westinghouse Aerospace Electrical Division.

offered by a rotor material is offset by the additional heat and temperature rise produced by the higher pole face losses.

The most promising solution to this dilemma is to use a pole material which offers excellent properties for the reduction of losses in conjunction with a core material that offers the required creep strength for the highly stressed portion of the rotor. A material of much lower creep strength can be used on the pole tips since the stress is small near the surface of the pole face. In addition, the creep strain there may be allowed to approach 1 to 1.5% since the length of the pole piece will be small, whereas, a creep strain of 0.5% for other parts of the rotor could be considered excessive since this amount of strain can cause a large change of the radial gap in the alternator.

Further reduction of pole face losses can be obtained with laminated pole tips. Eddy-current losses can be decreased by approximately 2-fold due to the increase of the effective resistance offered by laminations when compared with the effective resistance of grooved pole faces. Grooving the pole face increases the effective resistance of the eddy-current path by approximately 2 to 3-fold which can decrease the pole face losses by approximately 40 percent when compared with a solid pole face.

Only one of these three techniques was considered for reduction of pole face losses during the previously discussed parametric screening of designs for the KTA. This technique consisted of slotting the pole faces with grooves at right angles to the rotor longitudinal centerline. The grooves were 0.006-inch wide by 0.10-inch deep at a slot pitch of about 30 grooves per inch. With the grooves, the resistance

of the eddy-current path was increased by a factor of 3.8 when compared with the resistance of a solid pole face. Unfortunately, the H-11 material, which offers superior d-c magnetic and strength properties for the rotor, has a very low magnetic permeabilities when it is subjected to a time-varying induction. Pole-face losses are primarily a function of $(\rho\mu)^{-1/2}$, where ρ equals the effective resistivity along the eddy-current path and μ equals the effective permeability of the material when a time-varying induction is imposed upon the steady induction. A comparison of the properties of interest for H-11 steel to a steel which offers superior properties is given in Tables 2.4-2 and 2.4-3.

It can be noted from the tables that the resistivity of the two materials are about the same magnitude. However, a comparison of the effective permeability of the material shows that the silicon steel is a much better material from the standpoint of pole-face losses. For example, if one considers the 800°F, 19,200 rpm base design, the amplitude of the time-varying induction due to stator teeth is approximately 182 gauss which means that the effective permeability of the silicon steel would be 71.5 times greater than the effective permeability for H-11 steel. If we also consider a laminated, silicon steel pole tip construction in place of a grooved H-11 steel pole face, the relative pole-face losses become approximately equal to

$$\frac{\text{Loss (laminated silicon steel)}}{\text{Loss (grooves H-11 steel)}} = \frac{304}{778} \frac{8.1}{580}^{1/2}$$

$$= \frac{1}{13.53} = 0.0739$$

For the example above, the pole face loss due to flux ripple produced by the stator teeth can be reduced by a factor of about 13.5 by using the silicon steel in a laminated construction.

TABLE 2.4-2

RESISTIVITY (MICROHMS-CM)

	H-11 Steel	Non-grain-oriented 4-1/4% Silicon Steel
Bulk material, smooth pole face	78.7	108
Effective resistivity of slotted poles	304	416
Effective resistivity of laminated pole piece, 0.010" sheet x 0.25 depth	566	778

TABLE 2.4-3

EFFECTIVE PERMEABILITY (c.g.s.)

Steady Induction = 8.75 kilogausses		
Amplitude of Time Varying Induction on Surface of Pole Face gausses	H-11 Steel	Non-grain-oriented 4-1/4% Silicon Steel
0	1.8	350
130	7.1	520
182	8.1	580
242	9.0	642
394	11.0	739
578	12.8	803
802	14.7	885

The several techniques for reducing pole face losses that were investigated are summarized for each of the alternator base designs in Table 2.4-4. The techniques investigated included (1) grooved pole tips of H-11 steel, (2) laminated pole tip of 4-1/4% silicon steel, and (3) laminated pole tip of 3-1/4% silicon, grain-oriented steel. The losses for each of these designs was compared with the losses associated with a solid pole tip of H-11 steel. In general, the pole-face losses with a solid pole tip were approximately 1.5 times greater than the losses associated with grooved pole tips and approximately 23 to 37 times greater than the losses associated with a laminated pole tip construction.

The use of a laminated pole tip construction greatly simplifies the required rotor cooling system. In order to indicate the cooling features required for each of the designs, the temperature drop required to conduct the losses through the body of the pole and from the pole face to a central passage on the axis of rotation is given in the table. With the pole face limited to a maximum hot-spot temperature of 950^oF and with 800^oF coolant, the following conclusions can be made.

- (a) A solid pole face of H-11 steel will require coolant passage within the body of each pole.
- (b) A grooved pole face of H-11 steel will require a coolant passage through the rotor core near the base of each pole.
- (c) A rotor with a laminated pole tip construction can be cooled by using a central passage on the axis of the rotor.*

* Based on negligible windage losses.

TABLE 2.4-4

POLE FACE LOSS REDUCTION

Reference Design Speed (rpm), Coolant Temperature Estimated Losses, Excluding Pole Face Losses, Kilowatts	No. 1 19200/800°F	No. 2 24000/400°F	No. 3 19200/800°/1100°F	Pole Tip Configuration
	49.4	48.0	47.9	
Efficiency, Excluding Pole Face Losses, for 550.00 Kilowatts Output	91.8%	92.0%	92.0%	
Pole Face Losses per Unit Area, watts/in ²	18.5	37.8	17.8	
Pole Face Losses, Kilowatts	2.71	4.03	2.79	
Efficiency, Including Pole Face Losses	91.3%	91.4%	91.6%	Solid Pole Tips of H-11 Steel
Temperature Drop Required to Conduct Losses From Pole Faces and into Rotor Core, °F	170°	323°	173°	
Temperature Drop Required to Conduct Losses From Pole Faces to a Central Passage 1/2 inch in Diameter, °F	490°	807°	477°	
Pole Face Losses Per Unit Area, watts/in ²	11.9	24.2	11.6	
Pole Face Losses, Kilowatts	1.74	2.57	1.80	
Efficiency, Including Pole Face Losses	91.5%	91.6%	91.7%	Grooved Pole Tips, 0.006 inches wide by 0.10 inches deep with 29.4 grooves/inch
Temperature Drop Required to Conduct Losses From Pole Faces and into Rotor Core, °F	109°	206°	113°	
Temperature Drop Required to Conduct Losses From Pole Faces to a Central Passage 1/2 inch in Diameter, °F	315°	516°	311°	
Pole Face Losses per Unit Area, watts/in ²	0.814	1.604	0.796	
Pole Face Losses, Kilowatts	0.119	0.171	0.123	

TABLE 2.4-4 (Continued)

Efficiency, Including Pole Face Losses Temperature Drop Required to Conduct Losses From Pole Faces and into Rotor Core, °F	91.7%	92.0%	92.0%	Laminated Pole Tips, 4.25% Silicon Steel, Non-Grain-Oriented, Depth of Laminations = 0.5 in., Thickness = 0.010 inches
Temperature Drop Required to Conduct Losses From Pole Faces to a Central Passage 1/2 inch in Diameter, °F	7.4%	13.7%	7.7%	
	21.5°	34.2°	38.0°	
Pole Face Losses Per Unit Area, watts/in ²	0.518	1.019	0.506	
Pole Face Losses, Kilowatts	0.076	0.108	0.078	
Efficiency, Including Pole Face Losses	91.7%	92.0%	92.0%	Laminated Pole Tips, 3.25% Silicon Steel, Grain-Oriented, Depth of Laminations = 0.5 in., Thickness = 0.010 inches
Temperature Drop Required to Conduct Losses From Pole Faces and into Rotor Core, °F	4.8%	8.7%	4.9%	
Temperature Drop Required to Conduct Losses From Pole Faces to a Central Passage 1/2 inch in Diameter, °F	13.7%	21.7%	13.5%	

- (d) A detail study of the rotor cooling scheme for very low losses may demonstrate that it will be possible to cool the rotor through the shaft extensions thereby eliminating the need for any coolant passages within the rotor.*

In addition, Table 2.4-4 shows improvements in efficiency obtained by using laminated pole tips. Although the actual improvements in efficiency are small, these differences may have more important significance when evaluated in terms of system weight penalty.

The concept proposed for the laminated pole tips is shown on Figure 2.4-1. The laminations are assembled into bundles by a stitch weld through several layers. Steel shim spacers, about 0.002 inches thick, act as interlaminar insulation between the SiFe laminations. They are recessed below the depth of penetration of the time-varying flux (about 0.10 inches). The thickness of bundle is determined by the penetration capabilities of the weld used, e.g. electron beam welder. The height is about 3/8 inch and changes with each bundle. The steps shown on the H-11 pole provide beam clearance for the bonding of each lamination bundle to the pole. Assuming a pole length of three inches and an electron (E.B.) weld penetration of 3/4 inch, only one step would be needed if the top step were started on the centerline shown. For a four-inch length and 1/2-inch penetration, three steps would be needed.

There is one detracting feature to the above configuration in that bonding silicon iron to the H-11 steel alloy presents a problem of carbon diffusion out of the H-11 into the silicon iron. The result is

* Based on negligible windage losses

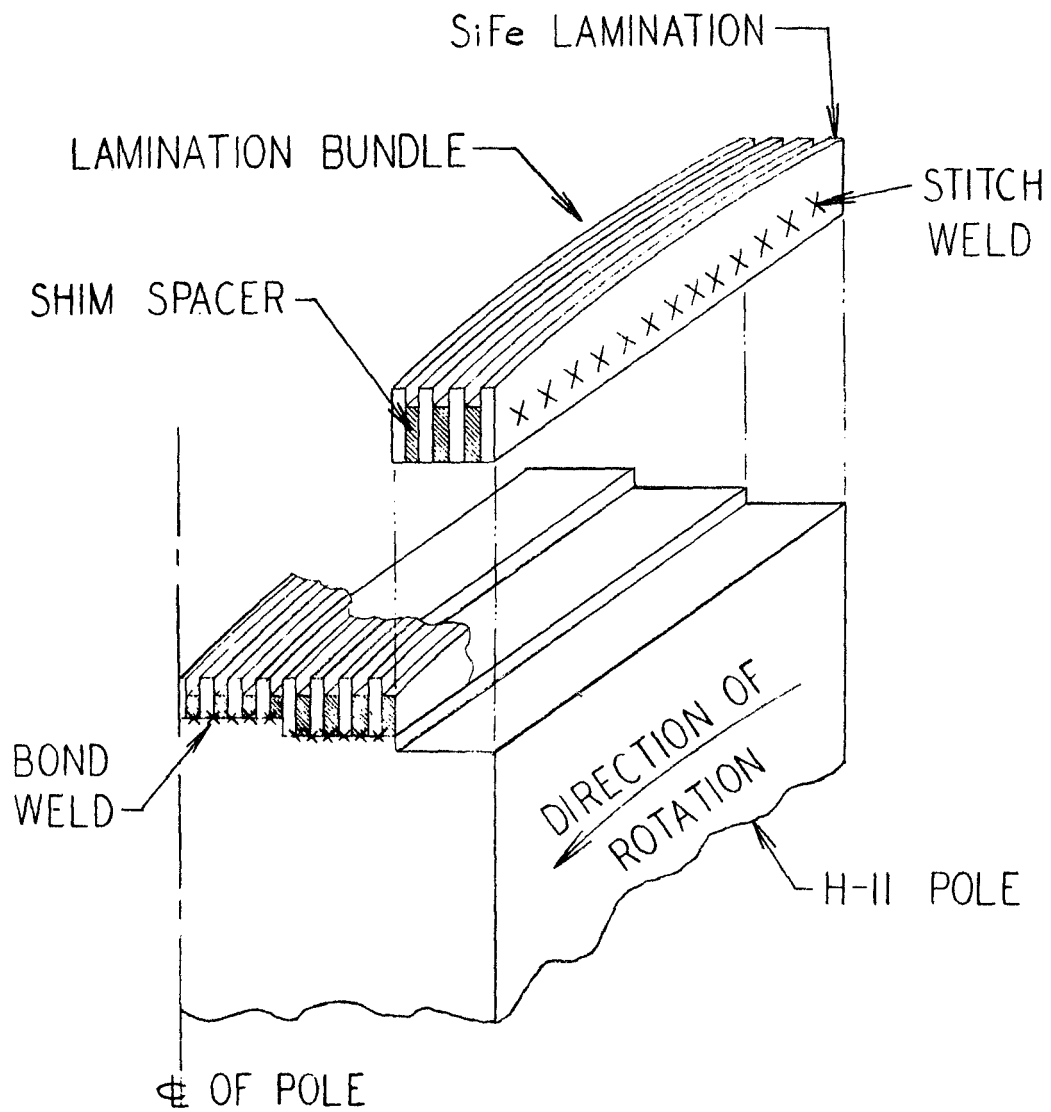


Figure 2.4-1. Laminated Pole Concept.

a brittle structure as was verified by sample pole pieces. This diffusion may degrade the magnetic properties of the silicon iron; however, no data is available to show or evaluate the extent of any degradation.

Windage Loss Calculations. As previously stated in Section 2.2, a self-imposed limit was placed on the windage losses that could be permitted in the rotor cavity. This was to maintain uncomplicated rotor cooling configurations, thereby improving reliability; it was not to improve alternator efficiency (even though some modest gains can be achieved as a result). Initially, the limit imposed was one kilowatt based on past experience with the SNAP-50/SPUR alternator and the realization that the 800^oF minimum potassium coolant temperature would make it more difficult to minimize them. As the study progressed, however, it became apparent that a limit of a few hundred watts was much more desirable and just as feasible. Therefore, this latter reduced limit became the goal of the conceptual design studies. Also, before proceeding further, it is appropriate to note that the assumption was used that the windage losses end up in the rotor-cooling circuit. This is conservative to the rotor-cooling design since, in reality, most of the losses manifest themselves as heat on the colder stationary walls of the rotor cavity.

As part of the studies and in parallel with other related studies (2.4-13), it became apparent that the calculation procedures derived by Smith (2.2-4) in the past were inappropriate to the KTA design conditions. As a result, an improved calculation procedure was derived (the Appendix) and it is upon this procedure plus the original experimental data (2.2-5) of Smith that the windage losses were based.

A survey of the several windage power loss experiments performed by Smith was made and a limited amount of data was found on a rotor configuration similar to rotor designs under investigation. A comparison of the test rotor and the designs of interest is given in Table 2.4-5. In general, the physical variations between the test rotor and the designs under investigation are small. The fact that the test model had more poles and a non-smooth stator bore means that the data from this model should be conservative from the standpoint of similarity when it is applied to the other designs.

The experimental data from the model rotor configuration is graphically illustrated in Figure 2.4-2. This data was plotted in accordance with functional Equation (6) in the Appendix. The data consisted of 16 points and is represented by the small circles shown on the figure. The ranges of the independent dimensionless parameters represented by this small amount of quantitative data are:

$$\begin{aligned} 0.134 &\leq M \leq 0.364 \\ 0.12 \times 10^6 &\leq \frac{\rho R a^*}{\mu} \leq 2.205 \times 10^6 \\ 1.71 \times 10^4 &\leq \frac{\rho R^2 \omega}{\mu} \leq 80.3 \times 10^4 \end{aligned}$$

Unfortunately, the range of these independent parameters did not provide a graphic solution of the problem for the rotor designs under investigation. The corresponding values of these parameters for the KTA alternator designs are given in Table 2.4-6.

The Mach numbers for the KTA designs are much higher than the Mach numbers associated with the experimental model. Similarly, the magnitudes of the dimensionless parameter, $\rho a^* R / \mu$, are lower than the range experienced with the test model. Therefore, it was necessary to

TABLE 2.4-5

COMPARISON OF DESIGNS AND TEST MODEL

	<u>Design No. 1 19200/800°F</u>	<u>Design No. 2 24000/400°F</u>	<u>Design No. 3 19200/1100/800°F</u>	<u>Test Rotor</u>
Rotor Diameter (inches)	15.553	14.066	15.575	7.87
<u>Core Diameter Rotor Diameter</u>	0.619	0.585	0.620	0.619
<u>Axial Length of Poles Rotor Diameter</u>	0.288	0.256	0.304	0.288
<u>Arc Length of Poles Per Side Circumference of Rotor</u>	0.345	0.344	0.335	0.329
<u>Length of Rotor Core Rotor Diameter</u>	0.786	0.770	0.806	0.921
<u>Radial Clearance Per Side Rotor Diameter</u>	0.0032	0.0032	0.0032	0.00318
Number of Poles	10	8	10	16
Condition of Stator Bore	Smooth	Smooth	Smooth	*

* 72 slots with the wedges 0.005-inches below the stator bore

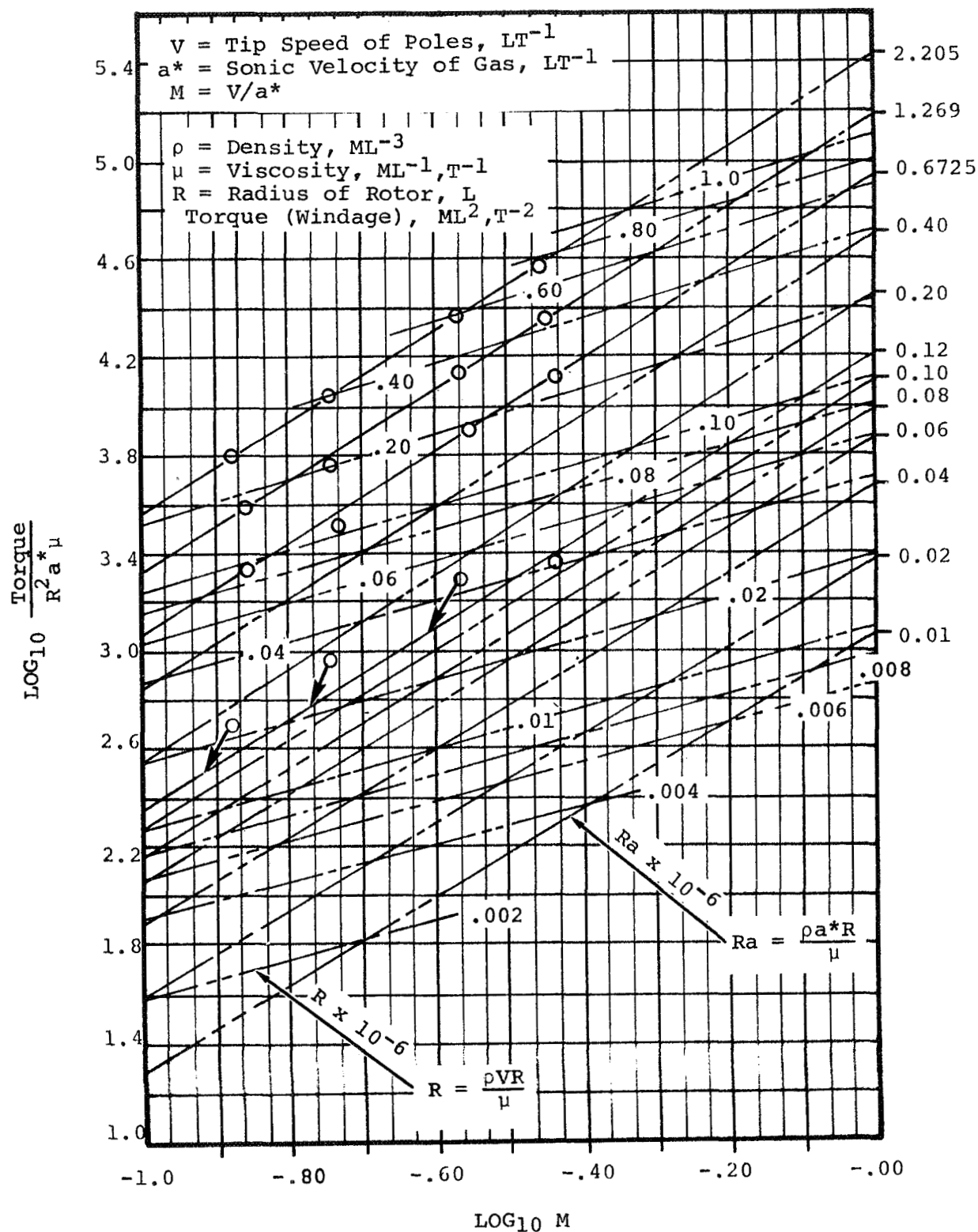


Figure 2.4-2. Windage Loss Parameters.

TABLE 2.4-6
RANGE OF DIMENSIONLESS PARAMETERS FOR ROTOR
CONFIGURATIONS UNDER INVESTIGATION

	Design No. 1 <u>19200/800°F</u>	Design No. 2 <u>24000/400°F</u>	Design No. 3 <u>19200/800/1100°F</u>
Fluid	"K"	5P4E	"K"
Fluid Temperature	800°F	400°F	800°F
Sonic Velocity of Gas, a^*	1575 fps	309 fps ^(a)	1575 fps
$M = R\omega/a^*$	0.826	4.75	0.827
$\rho R a^* / \mu$	2.96×10^4	0.304×10^4	2.95×10^4
$\rho R^2 \omega / \mu$	2.44×10^4	1.44×10^4	2.44×10^4

(a) Based on molecular weight = 446

extrapolate the parametric form of the graphic solution offered by the data from the model to gain some knowledge of the windage power loss in the KTA designs. These extrapolations were performed using a logarithmic plot of the experimental values shown in Figure 2.4-2. The extrapolations to lower values of $\rho a^* R / \mu$ were based on the condition that the dimensionless windage torque parameter is an exponential function of $\rho a^* R / \mu$ and on the condition that the windage power loss is zero when the density of the fluid is zero. The extrapolations to higher values of Mach numbers were made by using a straight-line extension of the lines on Figure 2.4-2.

It is very likely that this kind of extrapolation would not produce accurate data in the high Mach number range because the aero-dynamic drag on a pole is strongly dependent on the Mach number when this parameter is greater than 0.35 (2.4-14). For example, the data from reference 2.4-14 show that the drag coefficients on cylinders increase by 70 percent when the Mach number is increased from 0.35 to 1.0. Thus, it would be reasonable to assume that the windage loss obtained from Figure 2.4-2 must be increased by 50 to 70 percent when the Mach number approaches 1.0.

Windage power losses for the KTA alternator designs for various rotor cavity pressures were calculated using the graphical solution presented in Figure 2.4-2. These losses were increased by 50 percent to compensate for errors which may have been introduced by extrapolating the limited amount of quantitative data. The power losses are given in Table 2.4-7 for the three base designs.

The windage power losses are generally excessive when the rotor is operated within a cavity filled with a vapor at a pressure equal to the saturation pressure of the bearing lubricants. A complex cooling scheme would be required to maintain an acceptable pole tip temperature if a

TABLE 2.4-7
WINDAGE POWER LOSS

Design No. 1 and 3 19200 rpm, 800°F Potassium		Design No. 2 24000 rpm, 400°F 5P4E	
Rotor Cavity Pressure (psia)	Power Loss (watts)	Rotor Cavity Pressure (psia)	Power Loss (watts)
0.00028	10	0.0002	36
0.001	31.5	0.0004	73
0.002	58	0.0006	110
0.004	108	0.0008	147
0.006	154	0.0010	183
0.008	200	0.001075*	210
0.01	244		
0.02	450		
0.04	830		
0.06	1210		
0.08	1580		
0.10	1900		
0.155*	2850		

* Saturation pressure of bearing lubricant

considerable portion of the windage power loss were transferred to the poles. Coolant passages within the main body of the poles would be required with an 800°F coolant (saturated potassium) temperature. A reduction of the rotor cavity pressure is required to reduce the windage power loss to a magnitude that will allow the use of a simple rotor cooling scheme. A cooling scheme with a central passage can be used with an 800°F coolant if the windage power is reduced to 130 watts or less.

With the 400°F organic coolant, a cooling scheme with a central passage can be used if the windage power loss is reduced to 1500 watts or less; with additional reduction of the windage power loss it would be possible to cool the rotor through the shaft extensions. A preliminary examination of this cooling scheme indicates that a temperature drop of 1.13°F per watt of heat input into the rotor pole faces would exist between the pole face and the visco seal on the shaft extension. Thus, if the total heat input into the pole face from windage, eddy-currents, and convection from the stator bore seal can be limited to approximately 400 watts, a pole face temperature of approximately 850°F could be maintained with a 400°F coolant within the visco seals.

The pressure head developed within Holweck type of molecular vacuum pumps between the liquid seals and the rotor cavity was also evaluated. It became apparent from this evaluation that only a few stages of pumps would be required to reduce the rotor cavity pressure to a magnitude which will allow the use of a simple rotor cooling scheme.

The rotor cavity pressures for the organic coolant in Table 2.4-7 are based on the assumption that volatile products from the decomposition of the organic coolant will not accumulate within the rotor cavity. If

this happened, the rotor cavity would have to be vented to space to prevent excessive power losses. The opinion is that decomposition of the oil in the seals is very likely to occur; therefore, it must be recommended that the organic cooled/lubricated alternator be vented to space. Furthermore, with the rotor cavity vented to space, operation at pole tip peripheral velocities greater than the sonic velocity of the oil vapor, the effects of which are not completely known, is no longer of concern. In the case of a vented rotor cavity, provisions would have to be made to prevent a direct "line-of-sight" from the vapor seal outlet to the bore seal to prevent possible carbonous products (which are good electrical conductors) from depositing on the bore seal.

So far, the discussions of reducing windage losses have concerned themselves with the effects of reduced vapor pressure. This is rightly so because that is where the most significant reductions are achieved. Another method, as yet unmentioned, is to use baffling to break-up the turbulence around the poles. Three such methods are illustrated in Figure 2.4-3. All three concepts require non-magnetic material that has a high electrical resistance and all three suffer from mounting problems; the stationary baffles suffer because they must be of ceramic material and be close to the rotor wall; the rotating baffles suffer because they cannot be ceramic material in order to have sufficient mechanical strength.

While it is unknown precisely how much each of the configurations may reduce windage, some limited data by Smith (2.2-5) indicates that they could reduce losses to one-third of those for an open rotor pole configuration. Because of the minimal gains that can be achieved by

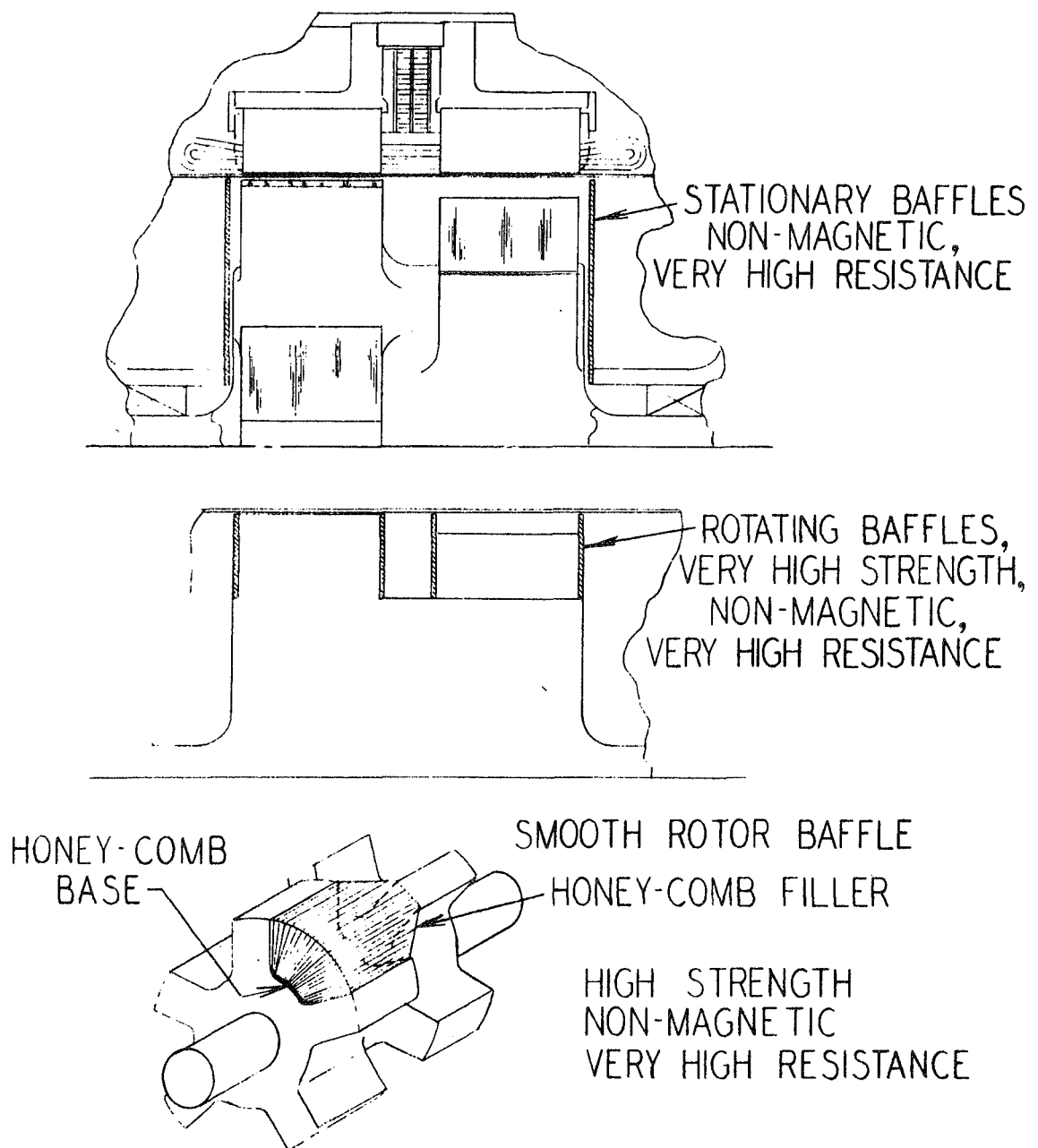


Figure 2.4-3. Windage Baffle Concepts.

baffling (compared to the more dramatic reductions achieved by reduced vapor pressure) and because of the attendant mounting complexities, it must be recommended that baffling not be used in the KTA alternator.

Rotor Study Summary. Based on the analyses of the rotor cooling concepts, the pole-face losses and the windage losses, it was possible to arrive at conclusions on which configurations or combinations thereof would be applicable to the various KTA requirements. These conclusions are consolidated into Tables 2.4-8 through 2.4-11. Summary comments relative to the applicability, advantages, and disadvantages are included in the tables.

The last table, Table 2.4-11, gives a general ordering to the preferred configurations. For instance, the top combination would be laminated pole faces with a central cooling hole and provisions for lowering the vapor pressure in the rotor cavity. Note however, that particular circumstances can influence the mix of configurations comprising the top choice. For example, if the electromagnetic design is directed toward making pole-face losses sufficiently low to permit a slotted pole configuration, the top combination might be slotted poles on pole body with central coolant hole and provision for low vapor pressure in the rotor cavity.

In summary, the listing of Table 2.4-11 can be used as a general guide to thinking but the peculiarities of the particular design must be the final guide to selecting the best configuration mix. The mix selected for the three Base Designs are tabulated in Table 2.4-12. On design number 1, laminated pole tips are recommended with a central coolant hole and a rotor cavity pressure of 0.004 psia. With the stator heat load included, the pole tip temperature is 875^oF and the creep strain at the tips is 0.12% over the three year life.

TABLE 2.4-8

ROTOR COOLING CONFIGURATIONS SUMMARY

USUALLY ONE OF TWO CHOICES

- (1) High Losses, Complex Cooling Configuration
(2) Low Losses, Simple Cooling Configuration

	<u>Configuration</u>	<u>Coolant</u>	<u>Typical °F/Watt</u>	<u>Remarks</u>
1.	Stub Shaft	800°-K	---	Impossible with 800°F coolant and calculated losses
		400°-5P4E	2.81	Feasible with very low losses and heat load
2.	Central Hole	800°-K	0.181	Feasible with low losses
		400°-5P4E	0.200	Feasible with moderate losses
3.	Parallel Passages Through Core	800°-K	0.0627	Feasible with moderate losses, requires manifolds with extra volume for storage of heavy materials that centrifuge out of the coolant
		400°-5P4E	0.0801	Not feasible due to centrifugation of heavy materials
4.	Passage in Body of Poles	800°-K	0.0181	Requires extra passage for storage of heavy materials that centrifuge out of K
		400°-5P4E	---	Not feasible due to centrifugation of heavy materials

TABLE 2.4-9

ROTOR POLE FACE LOSSES SUMMARY

BASED ON CALCULATION PROCEDURES DEVELOPED BY WESTINGHOUSE FOR TOOTH RIPPLE LOSSES; LOAD LOSSES ARE
BASED ON CALCULATION PROCEDURES DEVELOPED BY BERELLO

	<u>Typical Losses (kw)</u>	<u>Remarks</u>
Laminated Pole Tip	0.12	A. Rotor cooling scheme is simple A. 0.4% increase in efficiency D. Integrity of a laminated construction has not been defined
Slotted Iron Piece	0.19	A. 0.4% increase in efficiency A. Rotor cooling scheme is simple D. Integrity of weld has not been defined D. Complex slotting required
Slotted Pole	1.74	A. 0.2% increase in efficiency D. Coolant passages near the base of poles D. Complex slotting and drilling required
Plain Poles	2.71	A. Easy to fabricate pole faces D. Excessive pole face losses D. Complex cooling scheme into poles
A = Advantage		D = Disadvantage

TABLE 2.4-10

ROTOR WINDAGE POWER LOSS SUMMARY

CALCULATED VALUES BASED ON MODEL THEORY AND DATA FROM A SIMILAR ROTOR

- Lower Mach Numbers and Higher Reynolds Numbers
- Considerable Extrapolation of Model Data
- Margin for Error in KTA is Necessary
- Minimize by Lower Density or Baffling

		Typical Losses (kW)	Remarks
1.	Sat. Pressure in Rotor Cavity	K-800°F 0.155 psia	D. Decreases Efficiency by 0.5% D. Complex Cooling Scheme A. Simple Vapor Seals
		5P4E-400°F 0.001 psia	A. Simple Vapor Seal A. Simple Cooling Scheme D. Bore Seal Would Become a Conductor
2.	Lower Pressure	K-800°F 0.004 psia	A. Simple Cooling Scheme A. Increase in Efficiency D. Scavenge System Required with Vapor Seal
		5P4E-400°F 0.0002 psia	A. Simple Cooling Scheme D. Simple Vapor Pump Required D. Bore Seal Would Become a Conductor
3.	Vent To Space	K-800°F 0	A. Simple Cooling Scheme D. Complex Vapor Seal Required
		5P4E-400°F 0	A. Bore Seal Would Not Become Conductive
4.	Baffling (Stationary)	About 1/3 of above values	A. Vapor Seal Requirements are Diminished D. Baffling Must be Ceramic Material, Mounting Problems
A = Advantage		D = Disadvantage	

TABLE 2.4-11

ROTOR DESIGN STUDIES

ORDER OF CONFIGURATION CHOICES

1. Laminated Poles
 Slotted Poles on Iron Piece
 Slotted Poles on Pole Body
 Unslotted Poles
2. Low Vapor Pressure
 Vent to Space
 Saturated Vapor with Baffling
 Saturated Vapor
3. Central Cooling Hole
 Cooling Through Stub Shafts
 Coolant Into Core
 Coolant Into Poles Too

Expect Exceptions in Each Case Depending on Particular Circumstance

TABLE 2.4-12
CONCEPTUAL DESIGNS ROTOR DESIGN DETAILS SUMMARY

<u>DESIGN #1</u>		
19200 rpm	Laminated Pole: Pole Face Losses	= 0.076 kw
1600 Hz	0.004 psia Cavity Pressure: Windage	= 0.108 kw
800°F	Stator Heat Load	= 0.220 kw
		<u>0.404 kw</u>
	Central Hole - Pole Tip Temp. 875°F	
	- Creep Strain 0.12%	
<u>DESIGN #2</u>		
24000 rpm	Laminated Pole: Pole Face Losses	= 0.108 kw
1600 Hz	Vented to Space (Use Turbine Seals)	no losses
400°F	Stator Heat Load	= <u>0 kw</u>
		<u>0.108 kw</u>
	Coolant Thru Stub Shafts* - Pole Tip Temp. 700°F	
	- Creep Strain < 0.1%	
<u>DESIGN #3</u>		
19200 rpm	Slotted Pole: Pole Face Losses	= 1.80 kw
1600 Hz	0.004 psia Cavity Pressure: Windage	= 0.108 kw
800/1100°F	Stator Heat Load	= <u>3.50 kw</u>
		<u>5.40 kw</u>
	Coolant in Pole Tips - Pole Tip Temp. 900°F	
	- Creep Strain 0.12%	

* Borderline Cooling Configuration, Central Hole Better

On design number 2, laminated pole faces are again recommended and the rotor cavity is vented to space as previously recommended in Section 2.4. Because it is possible to use stub shaft cooling and not exceed 900^oF tip temperature for this design, the stub shaft cooling is shown. However, it must be pointed out that this design is marginal* and a central cooling hole would be better. The stator heat load is zero since the pole tip temperature is set by the stator temperature and heat load. In other words, any heat carryover from the stator to the rotor will quickly increase the pole tip temperature because the cooling path length to the stub shafts is too long to dissipate these additional losses. The pole tip will approach the stator temperature such that the heat load is negligible.

On design number 3, the coolant must be directed into the pole tips to handle the large heat load incoming from the 1100^oF stator (and still maintain a 900^oF maximum pole tip temperature). Because of this predominating heat load, the pole face losses become almost incidental and the best design to use there is the slotted poles in the pole body. The windage losses might also be allowed to increase from loss considerations alone, but because there are other advantages to a reduced cavity pressure**, it is recommended the same pump/seal provisions be included as on design number 1.

The need for the coolant being ducted into the rotor poles is sufficient cause for rejecting the design number 3 as a possible KTA alternator configuration. The solids which can accumulate in the pole

* Cannot accommodate many more losses than the 108 watts shown.

** Prevents cold-spot condensation within the rotor cavity and liquid impinging on the bore seal I.D.

tips cannot be filtered out (2.4-13) and additional passages must be included to store them. Since the amount of solids are dependent upon the system piping, the rotor design becomes very dependent upon non-alternator considerations. A typical example given in Reference 2.4-13 indicates as much as 2 cubic inches might accumulate. This amount of accumulation renders a highly reliable rotor cooling design impractical.

The configuration for design number 1 appears to be the most suited to the KTA alternator concept except for one aspect, the laminated pole faces. Being relatively uncommon and sensitive to fabrication anomalies, the laminated pole faces may be contrary to high reliability and ease of development requirements. Future studies will evaluate this configuration further relative to either reducing the losses and heat load or conceiving a relatively simple, reliable laminated pole configuration.

Stator Studies

The basic purposes of the Stator Studies were to (1) examine the alternative cooling configurations that would result in minimal hot-spots consistent with ease of development (or fabrication) and reliability, and (2) define the structural configurations that would result in allowable stresses consistent with minimum weight, reliability and ease of development. The structural studies were limited to those areas critical to the conceptual definitions, namely the bore seal end member configurations and stator magnetic frame and punching stack shrink-assembly fits. Structural studies of the mounting frame and end bells were deferred to Phase II studies as they were not critical to establishing the conceptual designs of Phase I. As a result, the frame and end-bell configurations shown later in Section 2.5 are heavy,

boiler-plate configurations derived from "artists-concepts" rather than engineering studies.

The reasoning behind these separate stator studies evolves from past design experiences on the first SNAP-50/SPUR alternator. There, because the required technology was in its infancy, near "equal-time" was given to the various analyses that covered the whole spectrum of the design. This was to evolve the first conceptual design (2.4-15, 2.4-16). Later, after the conceptual designs had been "frozen" and hardware development was proceeding (2.2-11) it was determined that, had more emphasis been initially placed on certain critical analyses, the stator design could have been made differently to make it more reliable and easier to fabricate. To avoid repetition of this in the KTA alternator conceptual designs, the separate stator studies below were carried out. Those associated with evaluation of the cooling configurations are discussed first.

Stator Cooling Concepts. The purpose of the study was to determine a basic stator cooling configuration that provides minimum operating temperatures without incurring electrical or mechanical design compromises. The study was performed with the use of a three dimensional heat transfer digital computer program that finds the transient and/or steady-state temperature distribution of an irregular body (2.4-17). The body is described by an analytical model made up of a nodal grid that simulates the physical configuration of the body and can accommodate changes in material properties in going from one location (node) to another. The program considers the heat transfer mechanisms of (1) conduction between internal (nodes) points, (2) conduction between internal and surface points, (3) radiation between surface points, and

(4) radiation, free convection and forced convection between surface and boundary points. The boundary points represent the fixed boundary conditions. The program also considers internal heat generation as a function of space and time and boundary temperatures that may vary as a function of time. For purposes of carrying out the study, a typical generator design was initially selected from the generator parametric screening studies. Later, after Base Designs had been selected, the studies were based on the three selected designs. All temperature distributions in the conceptual design studies were based on the 550 kW operating condition.

The initial studies were based on the following conditions and considerations:

- (a) Liquid metal as coolant at average temperatures of 800° and 1100° F.
- (b) All losses rejected to the coolant and to the end bells and housing when radiation from the armature end turns was considered.
- (c) An effective slot conductance (conductance of the a-c conductor and slot liner) of $0.2 \text{ watts/in}^2\text{-}^{\circ}\text{C}$ in directions perpendicular to current flow in the conductor.

The $0.2 \text{ watts/in}^2\text{-}^{\circ}\text{C}$ corresponds to a slot conductance that might be expected with a space vacuum ambient in the stator (2.4-13).
- (d) Maximum allowable hot-spot temperature of 1300° F.

Evaluation of the following stator cooling configurations was made during the study.

- (a) Axial cooling tubes every three degrees on the outside of the stack connected to a manifold around the field coil. This configuration is a very effective cooling sink but is complex and expensive and requires as many as 500 weld joints.
- (b) Axial cooling tubes every fifteen degrees on the outside of the stack and around the field coil. This is like number (a) above but with fewer tubes to reduce the quantity of weld joints.
- (c) Circumferential cooling manifold on the outside of the stator frame. This is to "fuse" the axial tube configurations above into one continuous duct, thereby lessening the weld joints and fabrication.
- (d) Circumferential cooling manifold on the outside of the stator frame with a separate cooling manifold around the field coil. This is like (c) above but with added cooling to minimize field coil temperatures.
- (e) Same as (d) above with cooling fins nestled between the a-c windings in the gap between the stacks and on the end turns. Heat transfer to these fins from the windings is by radiation only. This is to increase the area of material around the windings, thereby presenting more cooling sink surface area to the heat flux attempting to cross from the windings to the coolant. This is to reduce the winding temperatures.

These cooling configurations are presented schematically in Figures 2.4-4 through 2.4-16. Additional known (commercial) configurations which place the coolant either in the slots with the wire, in the wire

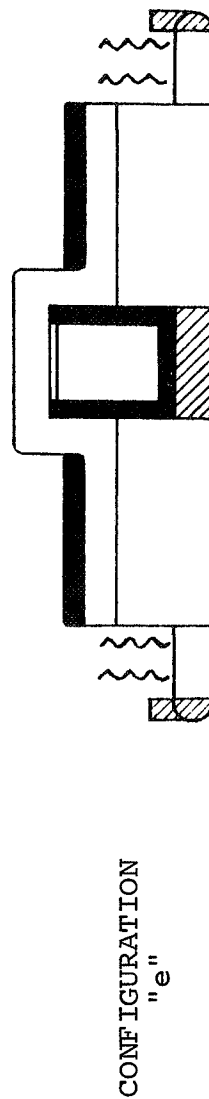
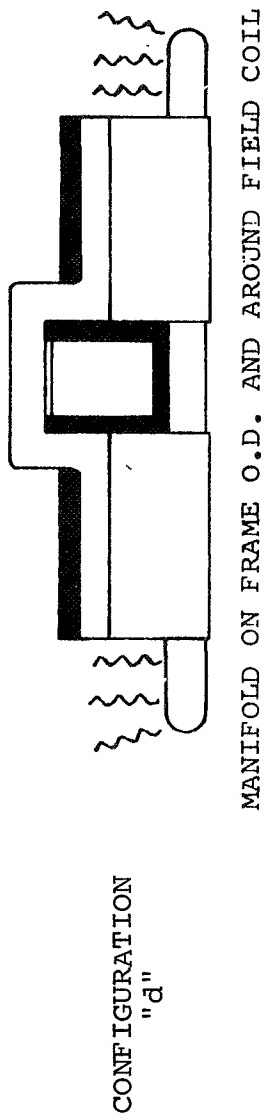
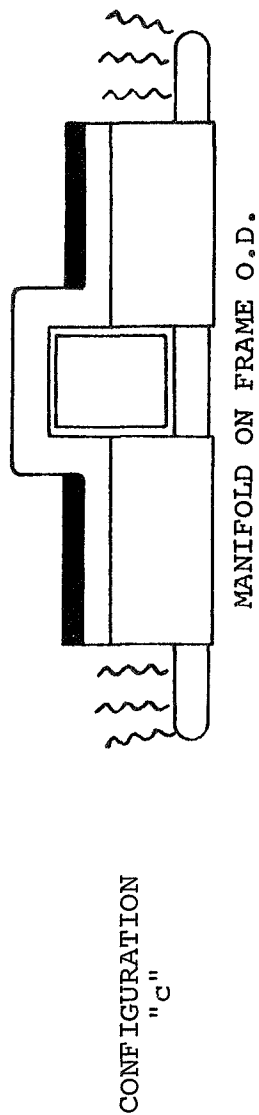
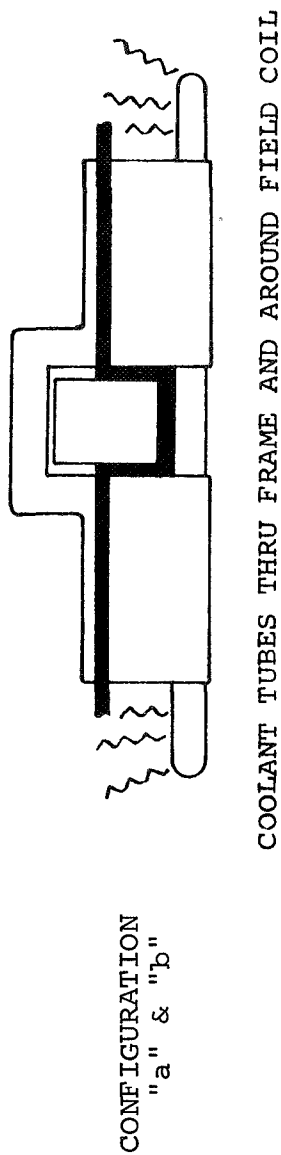


Figure 2.4-4. Stator Cooling Concepts.

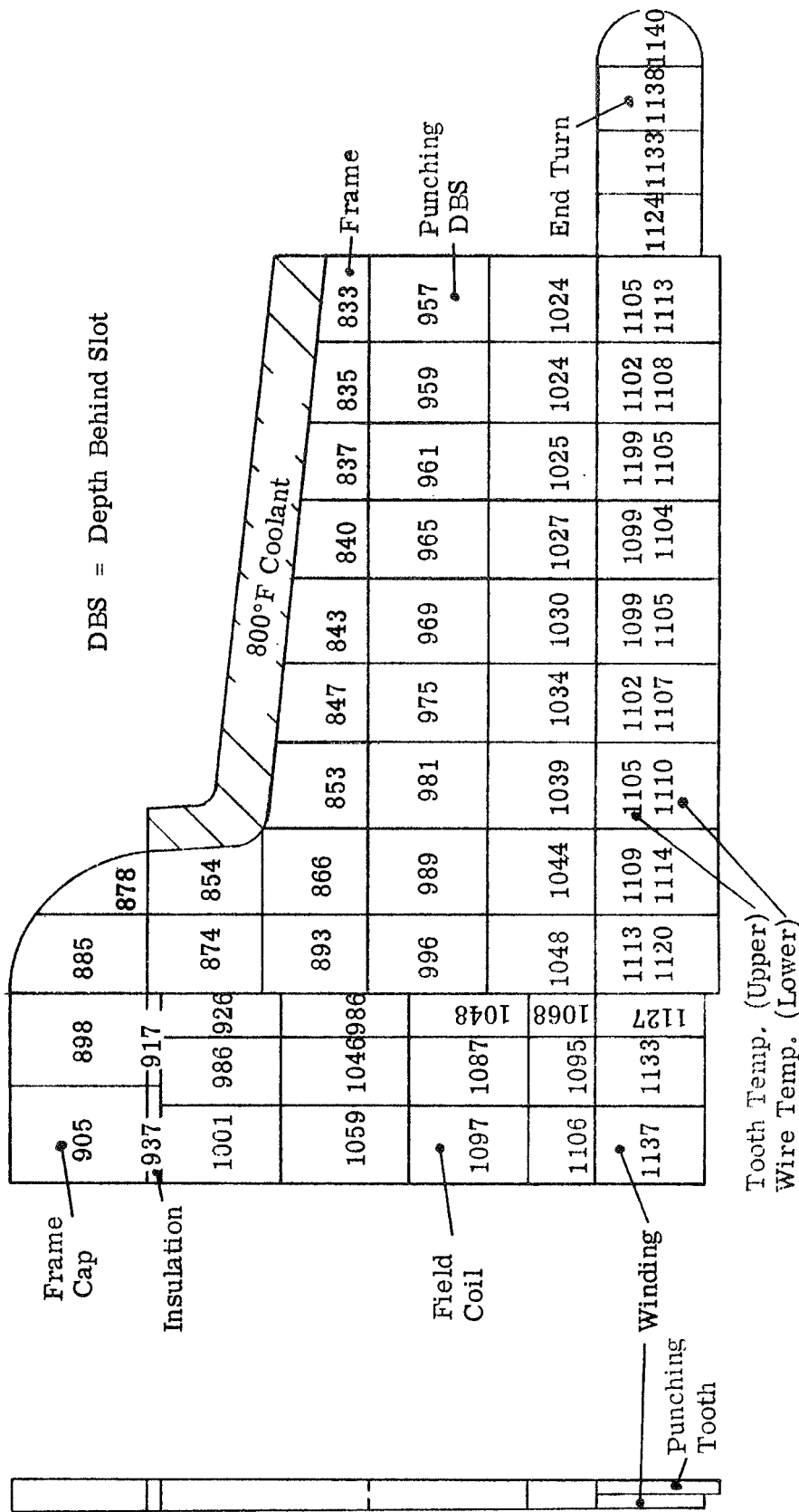


Figure 2.4-5. Stator Cooling Screening Study Temperature Distribution (°F)
Cooling Manifold on Stator Frame with Radiation from End Turns.

907	899	886	879	800°F Coolant									
939	919												
1003	989	927	875	855									
1062	1049	989	895	867	854	848	844	841	838	836	834		
1101	1091	1052	1000	993	985	979	974	970	966	964	962		
1112	1100	1073	1053	1049	1045	1040	1037	1035	1033	1033	1033		
1144	1141	1135	1121	1121	1113	1111	1110	1110	1113	1117	1123		
			1128	1123	1119	1116	1115	1116	1119	1124	1133		
												1149	1163
												1171	1174

Figure 2.4-6. Stator Cooling Screening Study Temperature Distribution (°F)
Cooling Manifold on Stator Frame Without Radiation from End Turns.

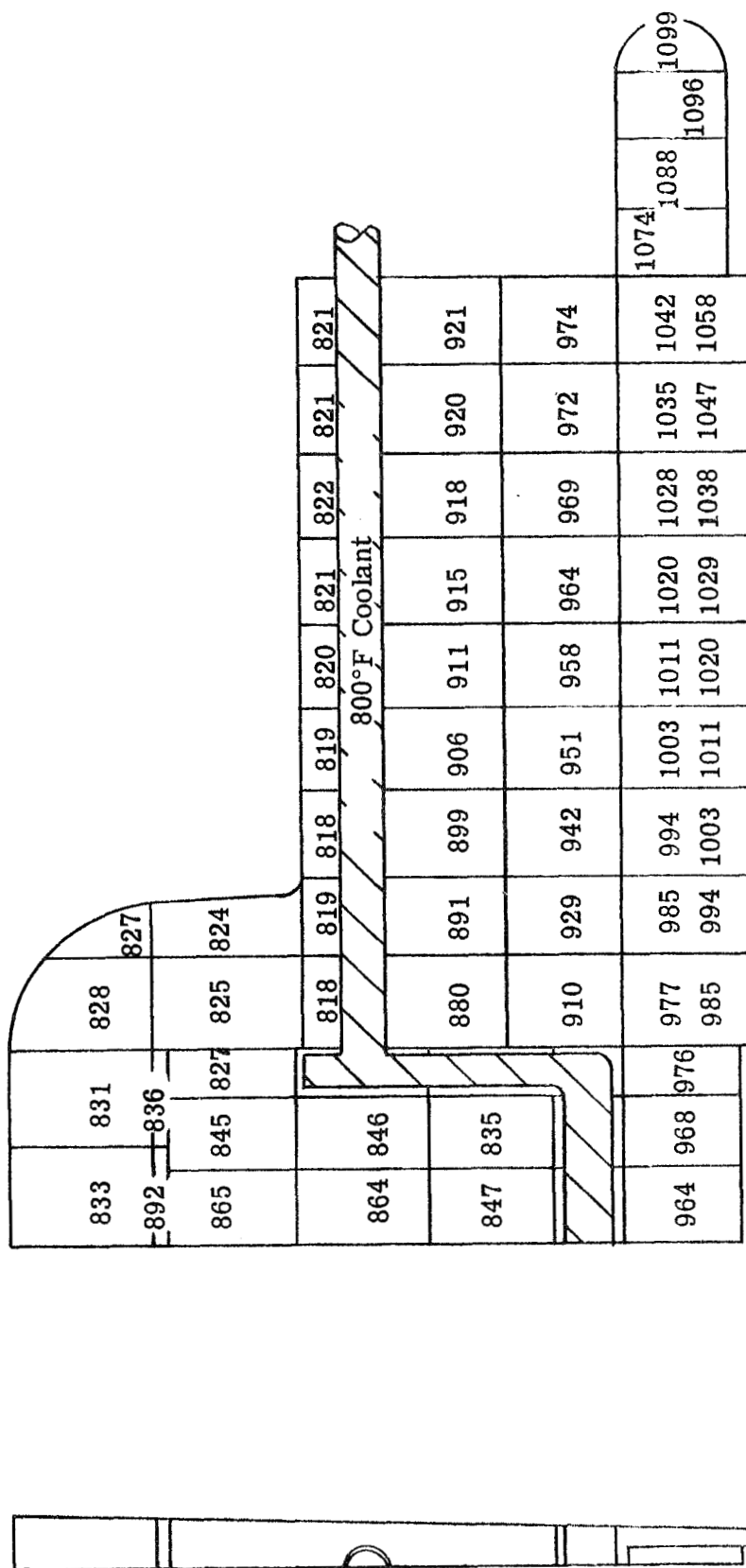


Figure 2.4-8. Stator Cooling Screening Study Temperature Distribution ($^{\circ}\text{F}$) Cooling Tubes Every Three Degrees on Stack with Cooling Manifold Around Field Coil and Without Radiation from End Turns.

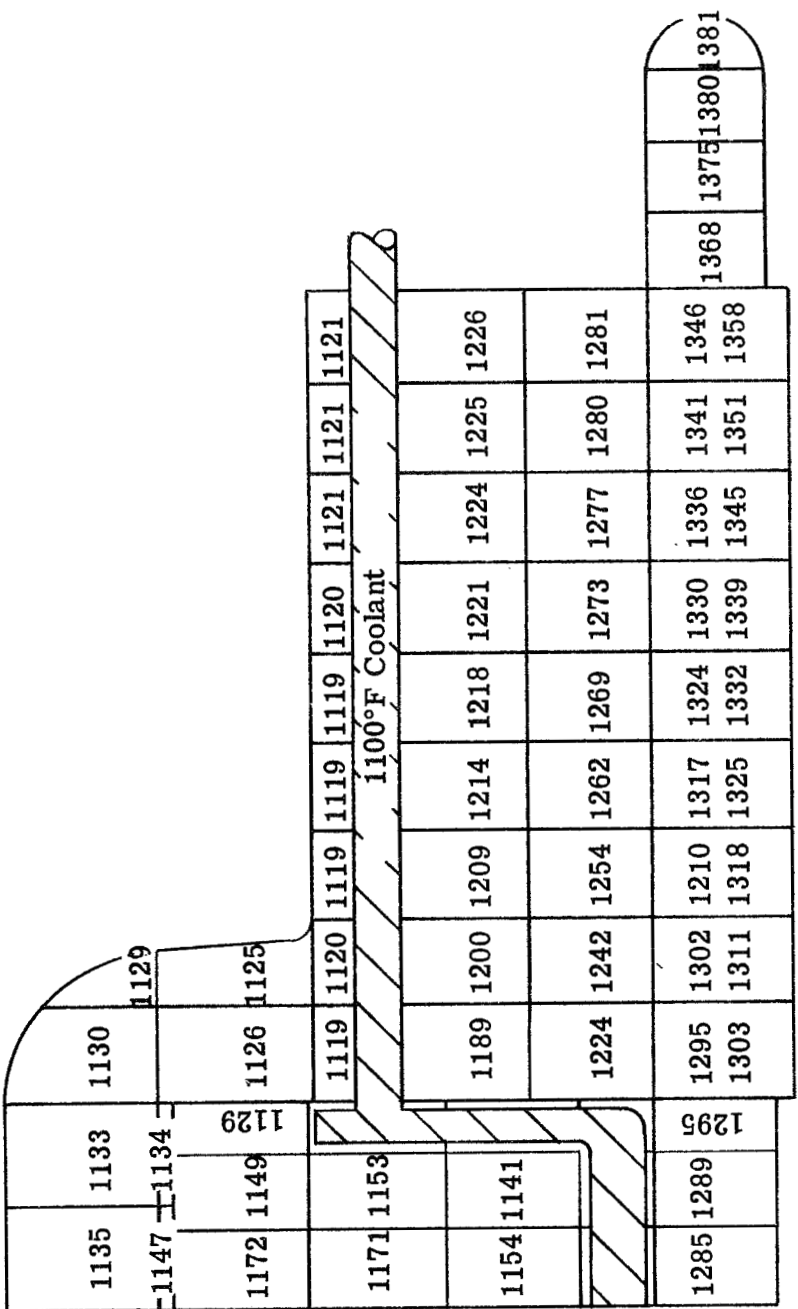


Figure 2.4-9. Stator Cooling Screening Study Temperature Distribution ($^{\circ}\text{F}$)
Cooling Tubes Every Three Degrees on Stack with Cooling
Manifold Around Field Coil.

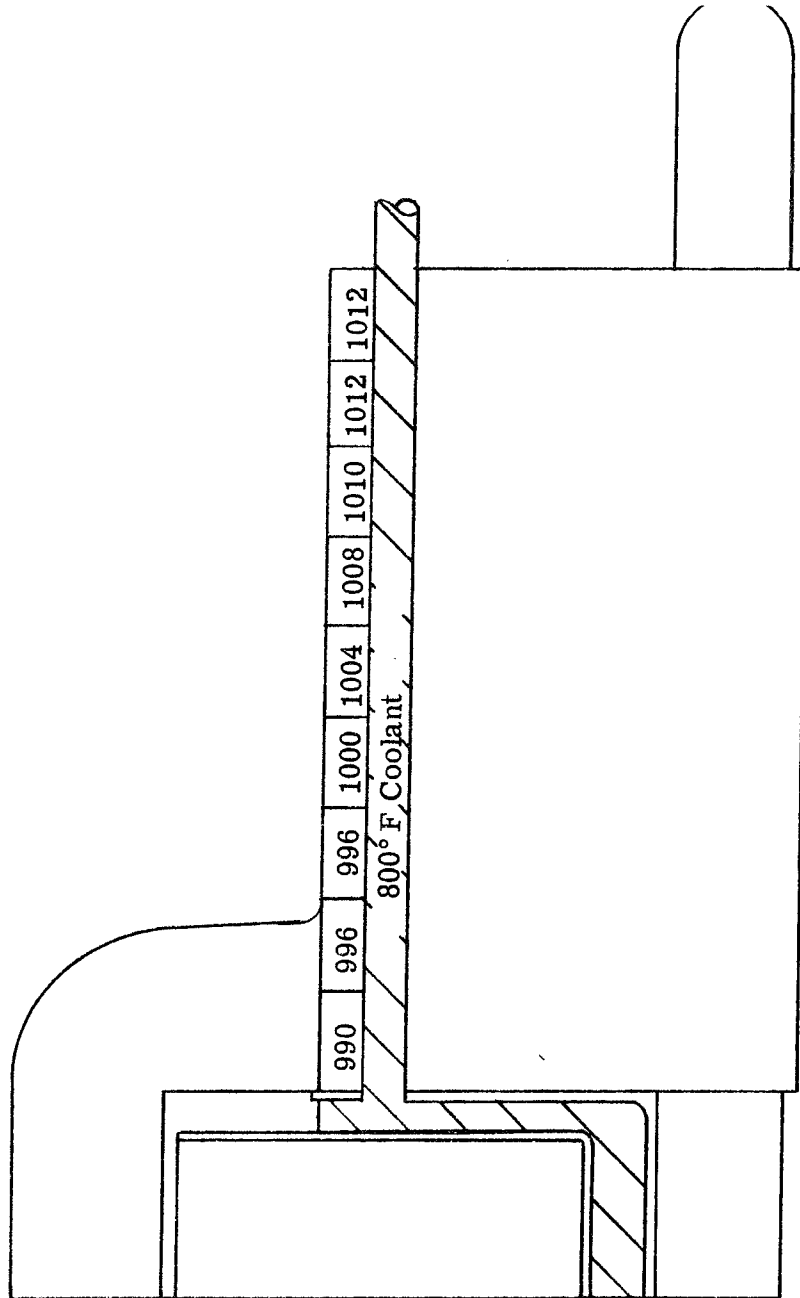
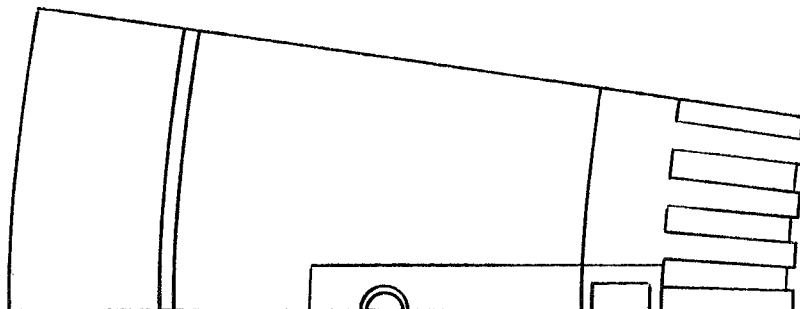


Figure 2.4-12. Stator Cooling Screening Study Temperature Distribution ($^{\circ}\text{F}$) Cooling Tubes Every Fifteen Degrees on Stack and Around Field Coil with Radiation from End Turns.

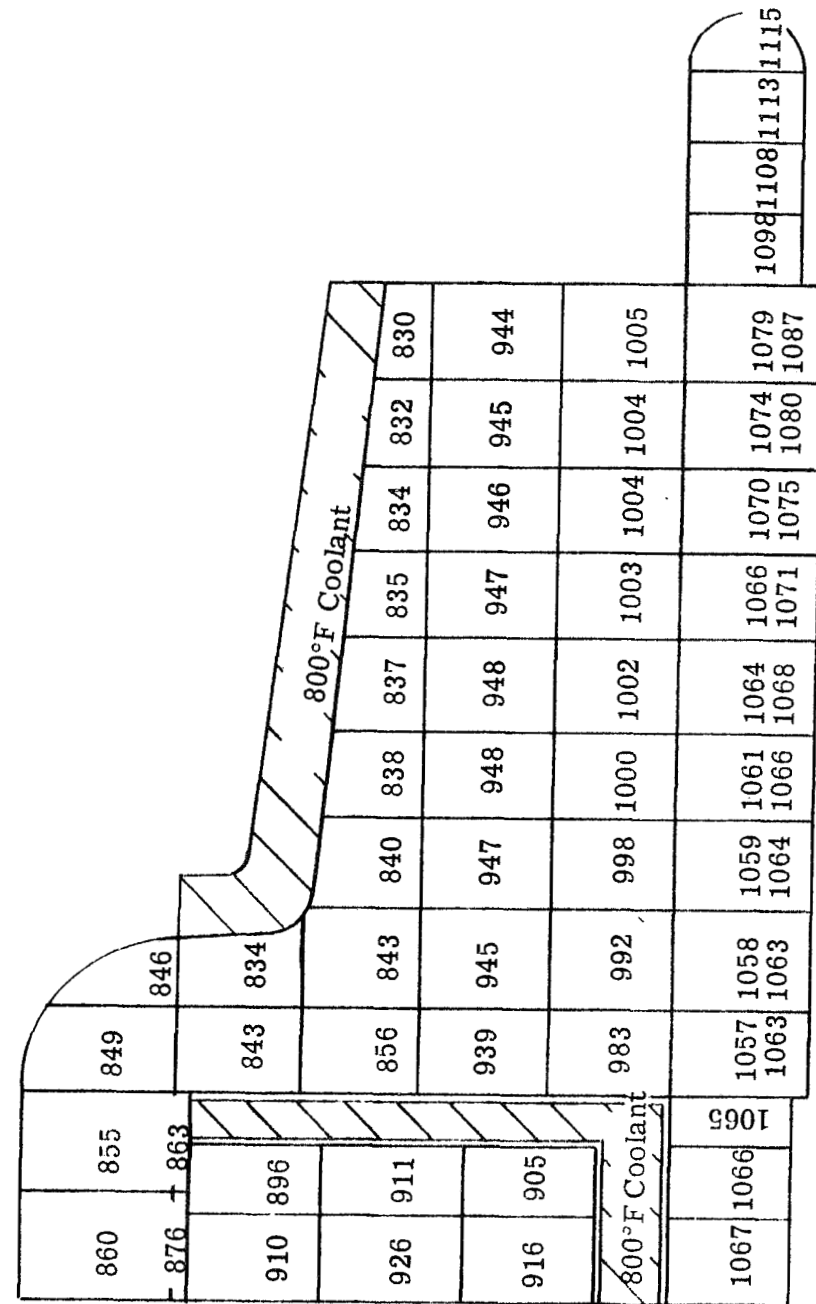


Figure 2.4-13. Stator Cooling Screening Study Temperature Distribution ($^{\circ}\text{F}$)
Cooling Manifolds on Stator Frame and Around Field Coil
with Radiation from End Turns.

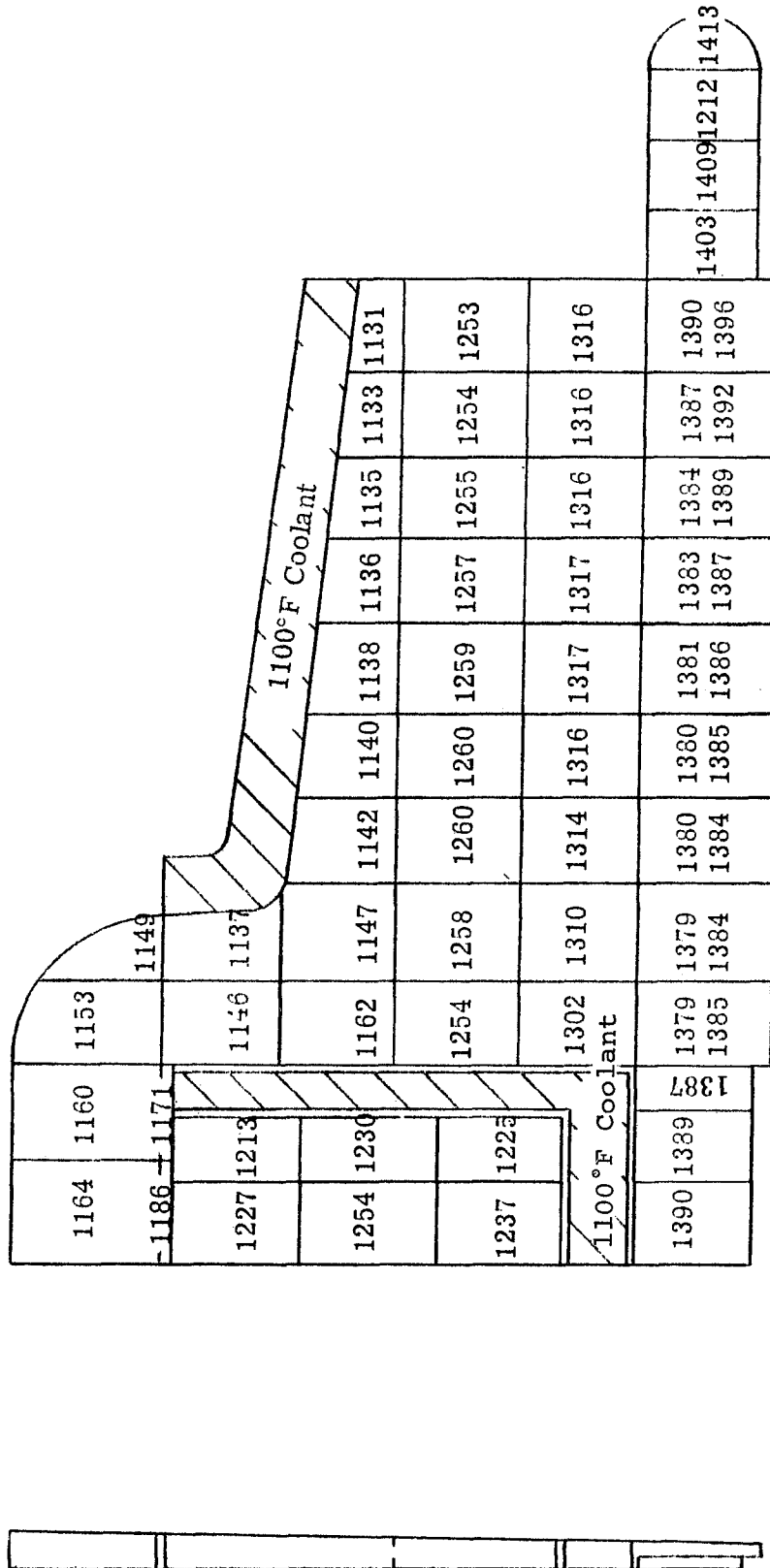


Figure 2.4-14. Stator Cooling Screening Study Temperature Distribution (°F) Cooling Manifolds on Stator Frame and Around Field Coil with Radiation from End Turns.

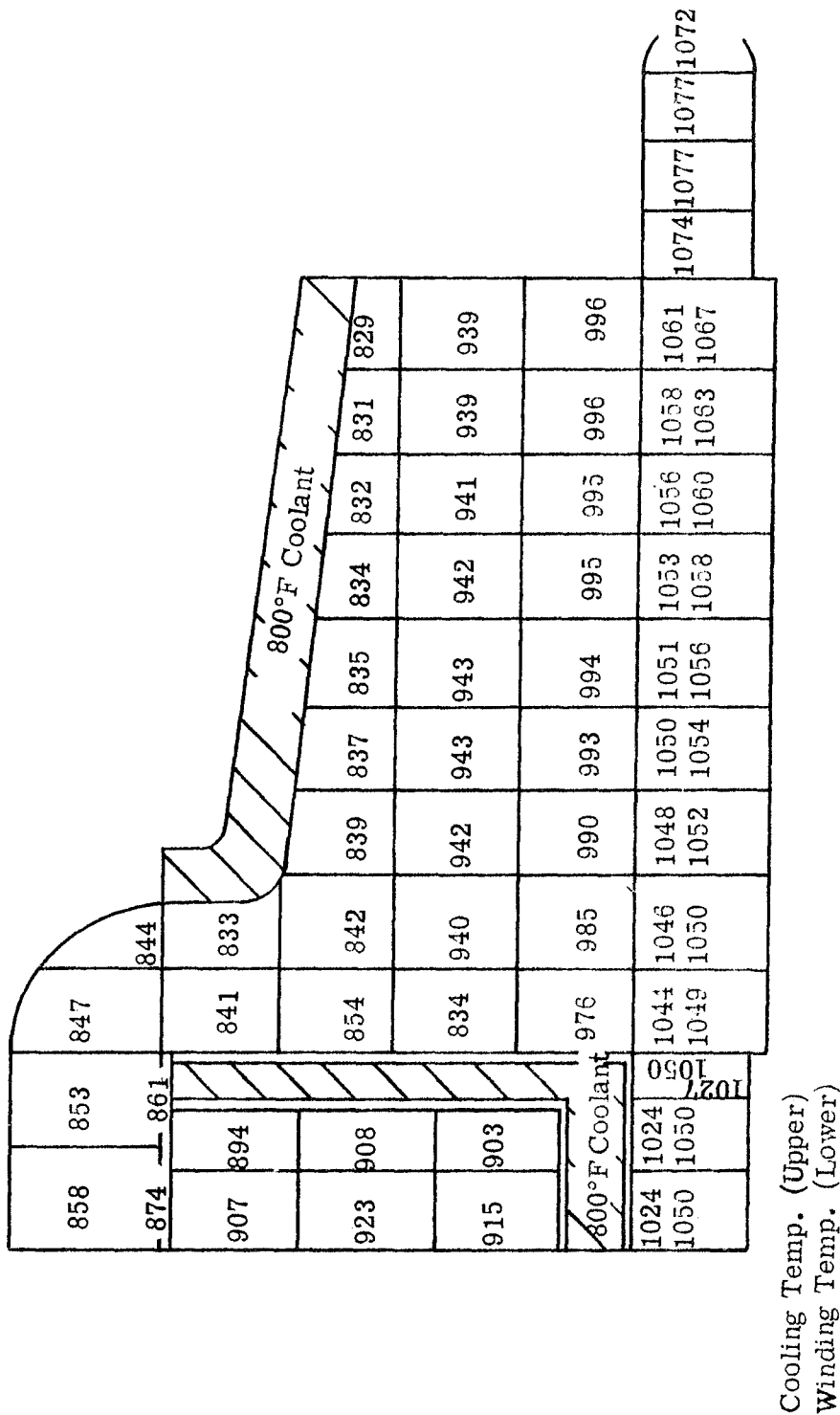


Figure 2.4-15. Stator Cooling Screening Study Temperature Distribution (°F) Cooling Manifolds on Stator Frame and Around Field Coil with End Turn Radiation and Cooling Fins Between Windings in Area Between Stacks.

itself, or in the punchings around the slots were not analyzed as the problems in fabrication for high temperature, alkali metal operation are inconsistent with the desire for ease of development.

As seen on the figures, an axisymmetric section was used for the analytical model that generally extended for one-half slot pitch plus one-half tooth pitch. Net heat exchange between the rotor and stator was neglected in these initial studies so rotor nodal points are not shown.

Temperature distributions for the above cooling configurations were determined for various combinations of 800° and 1100°F average coolant temperature with and without end turn radiation being included as a heat path. The temperatures resulting from these combinations are summarized in Table 2.4-13 with the respective maximum stator temperatures. The complete temperature distributions are presented in Figures 2.4-5 through 2.4-16.

Temperature distributions were also made for cooling configuration "e" (Figure 2.4-16) with effective slot conductances of 0.4 and 0.6 watts/in²- $^{\circ}\text{C}$ to determine the effects of any improvements (undefined) that might be possible with an increase in the thermal conductance between the conductor and the slot liner. This was because, on the 1100°F coolant designs, the heat was not being carried out of the wire as quickly or as well as desired. These results, presented in Table 2.4-13 also (Figures 2.4-17 and 2.4-18) show only negligible reductions of the winding temperatures. It would appear then that the effective (analytical) slot thermal conductivity used (in directions perpendicular to current flow) is sufficiently low and, therefore, any physical schemes to increase it over 0.2 watts/in²- $^{\circ}\text{C}$ do not greatly effect the conductor

TABLE 2.4-13

STATOR COOLING SCREENING STUDY MAXIMUM STATOR TEMPERATURES

Coolant Average Temperature	With End Turn Radiation		Without End Turn Radiation 800°F
	800°F	1100°F	
Cooling Configuration			
Cooling Tubes Every 3 Degrees	1075	1381	1099
Cooling Tubes Every 15 Degrees	1284	--	--
Manifold Cooling	1140	--	1174
Manifold and Field Coil Cooling	1115	1413	--
Manifold and Field Coil Cooling with a-c Winding Cooling Fins	1077	1372	--
Manifold and Field Coil Cooling with a-c Winding Cooling Fins - 0.4 Watts/ in ² -°C Slot Conductance	--	1371	--
Manifold and Field Coil Cooling with a-g Winding Cooling Fins - 0.6 Watts/ in - °C Slot Conductance	--	1369	--

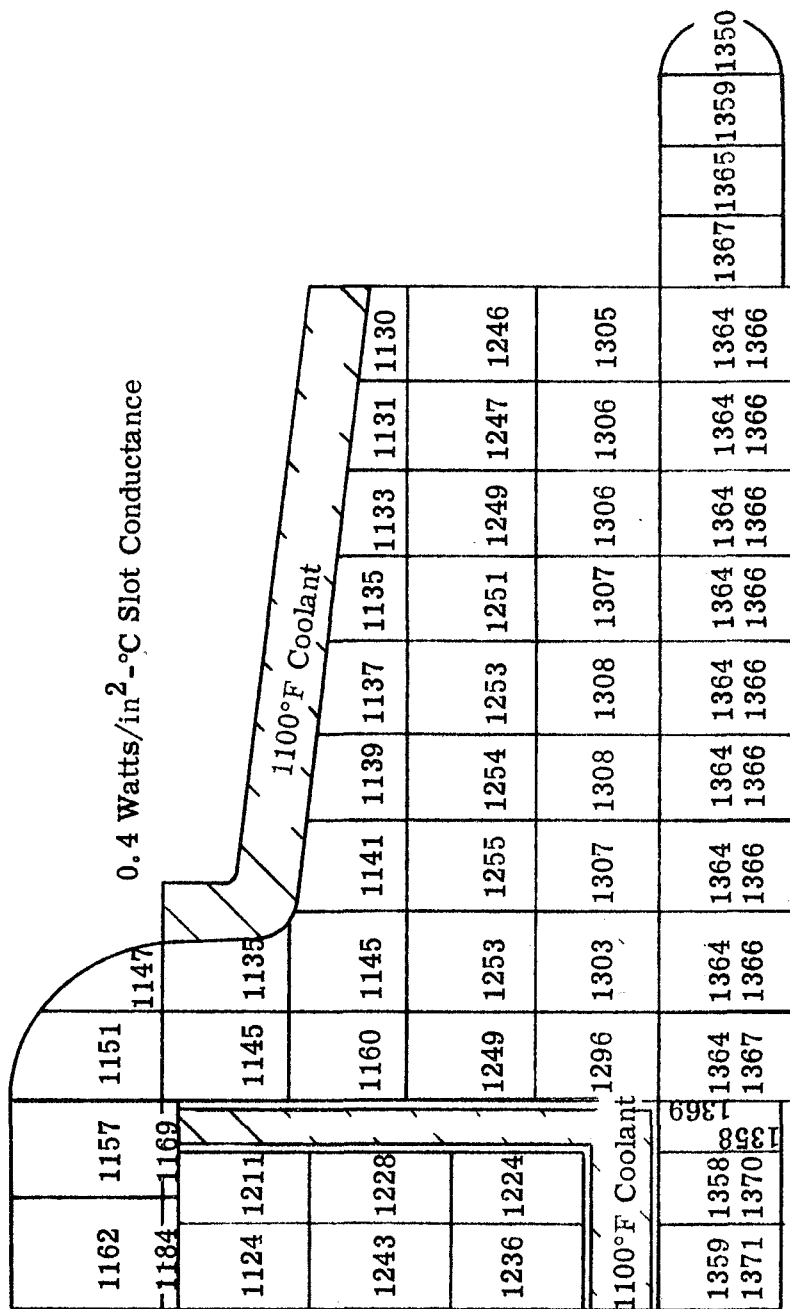


Figure 2.4-17. Stator Cooling Screening Study Temperature Distribution (°F)
Cooling Manifolds on Stator Frame and Around Field Coil
with Radiation from End Turns and Cooling Fins Between
Windings in Area Between Stacks.

temperatures. Because of this, it would be well to note at this time where the $0.2 \text{ watts/in}^2\text{-}^\circ\text{C}$ effective slot conductance is derived from.

In a physical configuration, the contacts between the wire and slot liner and punching side are subject to wide variations and irregularities. The effect is that it is impossible to establish an analytical model that describes what might or could be present. The safe approach to this analytically, then, is to be ultra conservative. This was done on the SNAP-50/SPUR Alternator (2.2-16) and the result was a prediction for a required cooling configuration similar to the configuration "a" of the present studies. In later analytical studies (2.2-2), in which most of the possible variations and irregularities were examined in a parametric comparison, the ultraconservative approach was shown to be unrealistic to the average worst case. In these studies, it was shown that on the average, the effective slot conductance may be about $0.4 \pm 0.2 \text{ watts/in}^2\text{-}^\circ\text{C}$ instead of $0.04 \text{ watts/in}^2\text{-}^\circ\text{C}$ for the ultra-conservative case. Thus, a more realistic (yet conservative) value to use for a design would be $0.2 \text{ watts/in}^2\text{-}^\circ\text{C}$. Proof of whether this rational is valid or not must wait until an alternator is built and thermally tested.

For the 1100°F coolant stators and for slot conductances that approach maximum expected levels and with end turn radiation, the maximum stator temperatures remained in excess of the maximum allowable 1300°F for the basic configurations of Figures 2.4-5 to 2.4-16. As a result, it became apparent that alternate methods for cooling of the 1100°F stator were required. A reduction of the stator losses and the length of the thermal path to the coolant are two desirable methods for reducing these stator temperatures. Based on recent experimental results

on AEC Contract AT(04-3)-679, (2.2-1, 2.4-13), evaluation of these two methods appeared to be appropriate. A temperature distribution was made for the approximate expected reductions of stack DBS length and stator losses to determine the magnitude of expected temperature reductions. The resulting temperature distribution, presented in Figure 2.4-19 reveals a maximum temperature reduction in the conductor of 119°F and a maximum stator temperature of 1272°F . These temperatures, though not completely accurate since actual reductions of stator size and losses were not available at the time, are an indication of what could be expected.

Results of the stator cooling parametric study also revealed that it is desirable that direct cooling of the field coil be included in the stator cooling configuration. The actual final selection of the particular field coil cooling configuration for conceptual design analysis will be based on cooling ability and ease of fabrication consistent with the schematic configuration above.

In summary, the 800°F nominal cooling configuration found to give reasonably low hot-spot temperatures consistent with ease of fabrication was a circumferential manifold on the stator frame with cooling fins nestled in the ac windings between the stacks. These fins are in conjunction with a separate cooling manifold around the field coil. This configuration is the "d" configuration examined with a portion of the radiation cooling fins extracted from the "e" configuration.

For 1100°F nominal coolant, the electro-magnetic design of the alternator would have to be altered specifically to reduce the winding hot spot. This is because the temperatures were well over the 1300°F limit even with cooling configurations such as configuration "b" or "c" which make maximum use of cooling sinks. It does not appear possible to

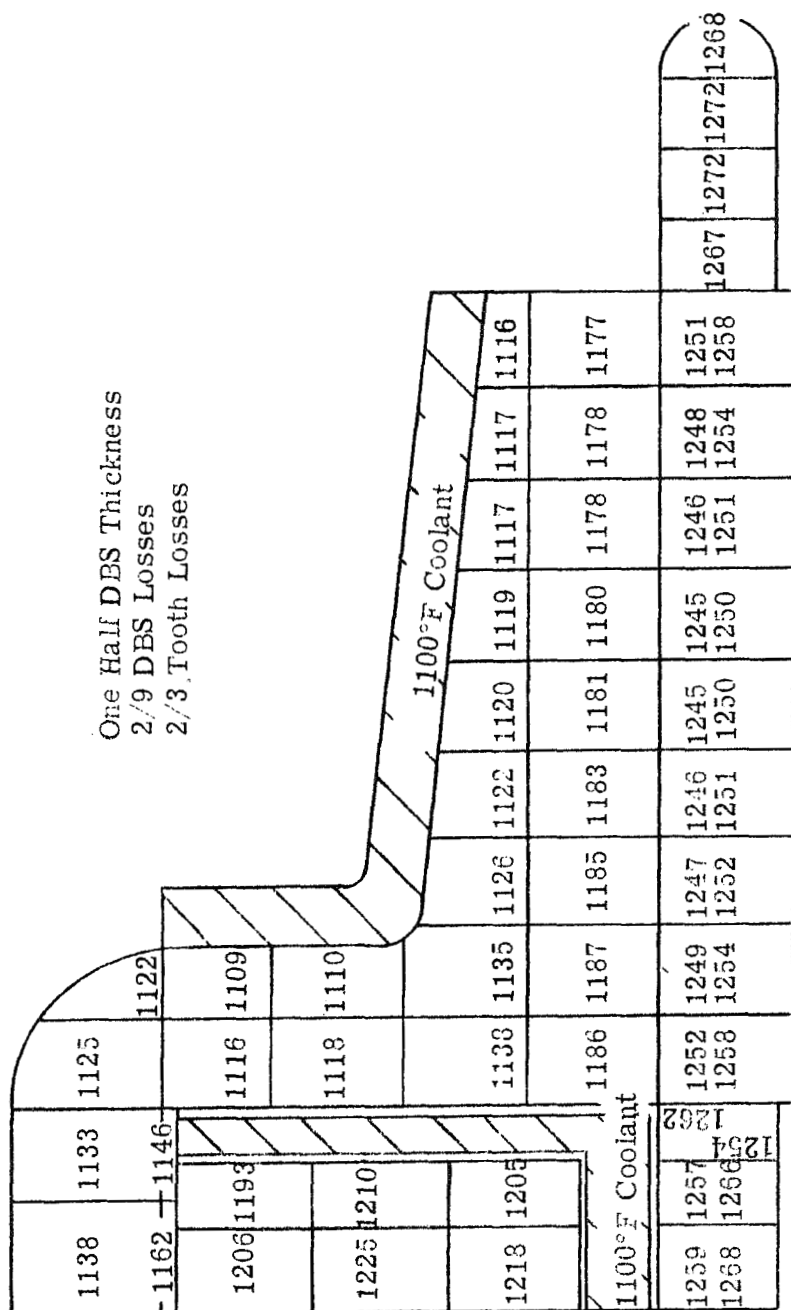


Figure 2.4-19. Stator Cooling Screening Study Temperature Distribution (°F)
Cooling Manifolds on Stator Frame and Around Field Coil
with Radiation from End Turns and Cooling Fins Between
Windings in Area Between Stacks.

add sufficiently more cooling configurations to the present electromagnetic design configurations without sacrificing ease of development.

Bore Seal End Member. The KTA design requires that liquid potassium or an organic fluid be the lubricant for the alternator. A technology for shaft seals has been acquired whereby it is anticipated that the leakage rate from the bearings can be maintained at an acceptable level. However, these seals cannot eliminate the presence of a vapor within the rotor cavity, even when the cavity is vented to the vacuum of space. At the low pressure of space, the mean free path between molecular collisions would be large (compared with a saturated or superheated vapor in a non-vented cavity), but 3 to 5 years operation with free diffusion of the molecules throughout an open stator would create problems relative to the integrity of the insulation systems. The diffusion of molecules from an organic fluid will not significantly decrease the severity of the problems from those associated with potassium since the stator temperatures will still be in the 1000^oF range. Contact of organic vapors with the hot stator components will cause a buildup of conductive deposits with time.

Protection of the stator from potassium vapor or the vapor associated with an organic fluid requires a bore-seal in the form of a thin-wall tube in the radial gap of the alternator. A thin-wall ceramic tube has been judged best suited to provide a good combination of resistance to attack by alkali metal and electric resistivity high enough to limit eddy-current losses to low values. The ceramic tube in the gap between the stator and rotor poles must be bonded to metal end members for sealing to the alternator frame in order to provide a sealed stator enclosure. In the event that both the stator and rotor

vented to space, the seal would not necessarily be hermetic nor of a high purity alumina. In that case, a term "bore-barrier" might be used to describe the ceramic tube and end members. In the discussion following, the more common term "bore seal" will be used to imply the presence of either a bore seal or a bore barrier, the choice being entirely dependent upon the cavity venting specification to be made.

The concept of designing an alternator without the bore seal was evaluated relative to the expected limitations or requirements that would be imposed on the design. Using technology and experience derived from actual tests (2.4-18), the summary of Table 2.4-14 was prepared. The limitations concern materials, voltage and electromagnetic effects. While it is recognized from the table that potassium-resistant materials are available and impose no restrictions, the deleterious effects of the voltage limitations and electromagnetic effects are sufficient cause for ruling out designs without a bore seal or bore barrier in the radial gap.

Long-time testing in potassium vapor has demonstrated assemblies of a 99.8 percent pure alumina (CE998) and the refractory alloy of 99Cb-1Zr joined by the 75Zr-19Cb-6Be braze alloy (or other active-metal brazes) to be resistant to potassium vapor at 900⁰F for more than 10,000 hours (2.4-19). The higher strength tantalum base refractory alloy (T-111, 90Ta-8W-2Hf), has displayed good brazing characteristics and the potential for being very resistant to alkali metal vapor (2.4-20). High purity low silica beryllia, although lower strength than alumina, has adequate resistance to potassium for operation up to 1300⁰F. Assemblies may be brazed using the 60Zr-25V-15Cb braze alloy. BeO material is not, however, available at this time in thin tube sections larger than 6 inches in diameter.

TABLE 2.4-14

CONSIDERATIONS IN DESIGN WITHOUT A BORE SEAL

1. MATERIAL LIMITATIONS

- (a) Organic-Insulation system must withstand carbon deposits on hot-spot even when vented to space. Applicable materials, limiting temperatures and undesirable qualities are summarized below:

Conductor Insulation:	Anadur < 1300°F (is porous) Teflon < 600°F (low radiation resistance) ML (H film) < 400°F Alumina (99.5%) < 1400°F Teflon < 600°F ML-Glass < 400°F MICA < 800 °F (poor in vacuum) Copper < 900°F Clad Silver < 1400°F
Slot Liner:	
Conductor	
Stator Magnetic Material:	Hiperco-27 CUBEX

- (b) Potassium - Requires potassium resistant system. Materials and limiting temperatures given below:

Conductor Insulation:	Plasma arc sprayed alumina
Slot Liner:	Alumina (99.5%) Beryllia (99.8%)
Conductor:	Nickel clad silver
Stator Magnetic Material:	Hiperco-27 < 1400°F CUBEX < 1100°F

2. VOLTAGE LIMITATIONS, Reference 20

- (a) For saturated potassium vapor, voltage must < 125 volts ac at 1600 Hz
(b) For superheated potassium vapor, voltage must be < 180 volts ac at 1600 Hz

3. POTASSIUM VAPOR EFFECTS IN ELECTRIC AND MAGNETIC FIELDS, Reference 20

- (a) Ionization causes extensive erosion of magnetic material and ceramics.
(b) Ceramic surfaces above 450°F become coated with a conductive film of potassium.
(c) Odd energy transfer phenomena that results in conductor destruction and erratic meter reading.

Because the technology required for the bore seal is relatively new and has not been carried to operational status for the sizes needed in the KTA alternator, the opportunity exists for coordinating the configuration design from the standpoint of both mechanical and metallurgical considerations. Past designs (2.2-11) suffered somewhat from such a lack of coordination. As a result, the designs contained both mechanical and metallurgical design compromises that did little to enhance the reliability of this very critical component in the alternator.

In those past designs, it was initially believed the most critical design area was the ceramic-to-metal joint between the tube and end member. Consequently, the majority of the design and development effort was concentrated in that area (2.4-19). Later detailed stress analyses identified equally critical aspect, excessive stress in the end member between the ceramic-to-metal joint and the stator frame (2.2-11). At the time, because hardware fabrication was proceeding, resolution of the problem was attempted by modifying the design and thickening the metal end member from 0.017 to 0.030 inch. Neither approach met with significant success since the thicker metal did not reduce stresses enough and it did detract from the successful development of the joint. Completely new and different design approaches were examined, but before little more than promising approaches were identified, the development project was terminated. The purpose of the present study was to continue evaluation of these promising concepts and select the one most suited to the KTA alternator.

The dilemma faced by the designer in selecting the best configuration is the result of two contrary requirements. The first is that the end

member must be flexible (thin) to accommodate the differential thermal expansions between the ceramic tube and the metal frame. The second is that the metal end member must be rigid (thick) enough to support any pressure differential between the rotor and stator cavity.* The solution to the dilemma is to find the proper balance between the two requirements.

The importance of this second requirement was not fully realized until large size bore seal designs were attempted and is the reason for the previously discussed problems of reference 2.2-11. Prior experience with small bore seals (up to 6 inches diameter) had given no indication of a pressure limitation or problem. As a check on the validity of the conclusions drawn on the dilemma, several of these small bore seals were stress-analyzed in detail. All showed low stress except one where calculated stresses were greater than yield strengths but less than ultimate strength. Visual examination of that particular bore seal revealed that its metal end members had indeed undergone a permanent, plastic deformation without catastrophic failure.

One of the least complicated approaches (from the standpoint of ease of development and minimum weight) to solving the two requirement dilemma is to design the end member so that the thermal expansion strains are counteracted by the pressure differential strains. The net strains and stresses are thereby minimized. The procedure of the study was to evaluate several configurations which were believed to contain features for doing this. Likewise, because transition pieces would be needed in going from the refractory alloy end members on the bore seal

*Even if the alternator is designed for near equal pressures in the rotor and stator cavity, "abnormal" situations can be foreseen where a pressure differential could exist. As the metal end member must survive these abnormal situations, the off-design requirements take on equal importance in selection of a design.

to the frame, bimetallic configurations for these pieces were included in the study. The transition pieces are needed to provide a non-refractory alloy section to weld to the generator frame on final assembly.

The comprehensive stress analyses carried out on the bore seal end members to select the conceptual designs were based on the following:

- (a) The maximum pressure differential across the bore seal and end members shall not exceed 20 psi in either direction. Internal pressurization shall be only at room temperature, external at all temperatures listed below.
- (b) The temperatures of the bore seal and end members shall be room temperature (uniform), soaking temperature (uniform), operating temperature (gradients) and the transients between them. Soaking temperature is assumed to be at least 1100⁰F.

The stress analysis was conducted with the use of a computer program that calculates stress and displacements of elastic axisymmetrical open and closed shells. This computer program was previously written and verified at another division in the Westinghouse Electric Corporation. The analysis is made by setting up a number of points (stations) along the meridional profile of the shell. At each of these points, input is furnished to describe the flexibility of the shell and the external loading. The program is based on a Myklestad-type finite difference solution set up in matrix form. The analysis takes into account the effects considered in Love's first approximation and also includes the effect of shear deflection. At the conclusion of the calculation, the stress and deformations of the shell are printed out for each of these same points.

The basic bore seal design used in the analyses consists of a thin-walled, high-purity alumina (or beryllia) cylinder brazed to refractory metal end members. The seal is completed by bonding of the end member to the alternator housing through transition pieces. This analysis of the end member stresses was based on the following assumptions relative to the basic design.

- (a) The metallic end members are 0.017 inch thick. This thickness was selected from previous studies (2.4-19) and is based on commercial experience with thin gauge metal components in producing vacuum-tight joints in brazed ceramic-refractory metal assemblies.
- (b) The bore seal diameter is 15.5 inches. This value is the approximate diameter of the bore seals of the conceptual designs.
- (c) The relative axial expansion of the bore seal is 0.014 inch; to be compensated for at one end of the bore seal only. In other words, one end of the bore seal is fixed relative to the end bell on that end.

This value was selected based on previous studies (2.2-11) as approximating relative expansion for the conceptual design configurations. Temperature distributions, from which expansions could be calculated, had not been determined at the time of these studies.

Stresses were determined for a variety of end member configurations having expansion compensating features for the 20 psi pressure differential simulating stator pressurization, 1100°F constant soak

temperature, and 0.014 inch axial displacement. These configurations included bellows, large curved members, and short conical members; all three were in both the inward and outward directions. Butt type and sandwich type metal to ceramic joints (see Figure 2.4-20) were investigated with the results showing no stress advantage of the butt joint over the sandwich joint. Results of the stress calculations, summarized in Table 2.4-15, show that the simple, straightforward short configurations are the lowest stressed. However, even then, the end member stresses were in excess of a self-imposed limit of 15,000 psi.

Due to the "compensating" configurations not resulting in low stresses and because of the desirability of providing and maintaining accurate alignment of the bore seal in the radial gap, it became necessary to consider a different configuration that provided more positive mechanical support for the bore seal and end members while maintaining flexibility. The configuration concept (for the fixed end) consists of a ring fixture which is custom-mated with the ceramic on the bore seal and fits snugly against the unpressurized side of the metal end member. The snug fit is important to providing the thin end member support. A curvic coupling (radial tooth) type arrangement accommodates for radial thermal expansion differences and supports the axial loading required to keep the fixture pressed against the end member. This concept is illustrated on the left-hand view of Figure 2.4-21.

On the opposite end of the bore seal (axial expansion compensating end), a similar support fixture arrangement is used. In order to accommodate for both radial and axial differential expansion between the seal and stator frame, two separate support fixtures (with curvic type

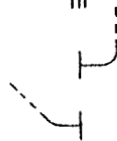
TABLE 2.4-15
KTA BORE SEAL END MEMBER STUDY (DRIVE END)

No.	Configuration	Max. Stress (psi) and Location
1	Trial run to test program input	
2	Semi-toroidal, to inside - 1" radius	604, 472 E. M. - B. S. Jt.
3	1 cycle over lap bellows, to inside - 0.5" radii	3, 847, 084 E. M. - B. S. Jt.
3A	1 cycle over lap bellows, to outside - 0.5" radii	260, 978 E. M. - B. S. Jt.
4	2 - quarter toroids, back-to-back, to inside - 1" radii	105, 225 E. M. - B. S. Jt.
5	2 - quarter toroids, back-to-back, to outside - 1" radii	48, 169 E. M. - End Bell Jt.
6	2 - quarter toroids, butt reverse together, to inside - 1" radii	71, 227 E. M. - End Bell Jt.
6A	2 - quarter toroids, butt reverse together, with straight section between to inside - 1" radii	88, 551 E. M. - End Bell Jt.
7	2 - quarter toroids, butt reverse together, to outside - 1" radii	133, 239 E. M. - B. S. Jt.
9	30° conical section, to inside - 1.5" lg., 0.5" radius bends	52, 907 E. M. - B. S. Jt.
10	15° conical section, to inside - 1.5" lg., 0.5" radius bends	70, 597 E. M. - B. S. Jt.
11	45° conical section, to inside - 1.5" lg., 0.5" radius bends	524, 283 E. M. - End Bell Jt.
12	30° conical section, to inside - 1" lg., 0.5" radius bends	30, 863 E. M. - B. S. Jt.
13	30° conical section, to inside - 0.5" lg., 0.5" radius bends	28, 891 E. M. - B. S. Jt.
13A	30° conical section, to inside - 0.5" lg., 0.5" and 0.1" radius bends	233, 427 E. M. - End Bell Jt.
13B	30° conical section, to inside - 0.5" lg., 0.5" radius bends	89, 842 E. M. - End Bell Jt.
13C	30° conical section, to inside - 0.5" lg., 0.1" radius bends	61, 116 E. M. - End Bell Jt.
14	3/4 cycle deep bellows section - 1" deep, 0.25" radius bend	473, 274 In Bend
15	15° conical section, to outside - 0.5" lg. mid. sect., 0.5" radius bends	49, 280 E. M. - End Bell Jt.
15A	15° conical section, to outside - 1" lg. mid. sect., 0.5" radius bends	100, 350 E. M. - End Bell Jt.
15B	15° conical section, to outside - 0.25" lg. mid. sect., 0.5" radius bends	37, 315 E. M. - End Bell Jt.

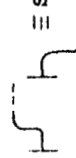
TABLE 2.4-15 (Continued)

No.	Configuration	Max. Stress (psi) and Location
15C	15° conical section, to outside - 0.0" lg. mid. sect., 0.5" radius bends	65, 171 E. M. - B. S. Jt.
16	Quarter toroid, to inside - 1" radius	86, 245 E. M. - End Bell Jt.
17	30° conical section, to outside - 0.5" radius Bends, 0.5" lg. mid. sect.	43, 238 E. M. - End Bell Jt.
18	30° conical section, to outside - 0.5" radius bends, 0.5" lg. mid. sect.	85, 484 E. M. - End Bell Jt.
19	2 - quarter toroids butt together (reverse), to inside - 1" radius bends	524, 644 E. M. - B. S. Jt.
20	30° conical sect., to outside - 0.5" radius bends, 0.5" lg. mid. sect.	70, 605 E. M. - End Bell Jt.
21	45° conical sect., to outside - 0.5" radius bends, 0.5" lg. mid. sect.	80, 173 E. M. - End Bell Jt.
22	30° conical sect., to inside - 0.5" radius bends, 0.5" lg. mid. sect.	76, 993 E. M. - End Bell Jt.
23	30° conical sect., to inside - 0.5" radius bends, 0.75" lg. mid. sect.	111, 354 E. M. - End Bell Jt.
24	15° conical sect., to outside - 1" radius bends, 0.0" lg. mid. sect.	37, 258 E. M. - End Bell Jt.

Nomenclature:



≡ butt type joint, outward and inward respectively



≡ sandwich type joint, outward and inward respectively

E. M. - B. S. Jt. * = stress occurs in end member at bore seal joint (not in joint itself but just outside (t)

E. M. - End Bell Jt. * = stress occurs in end member at joint where it attaches to the end bell (frame)

* Note: because of stress concentrations, the maximum stresses nearly always occurred at one of these two locations.

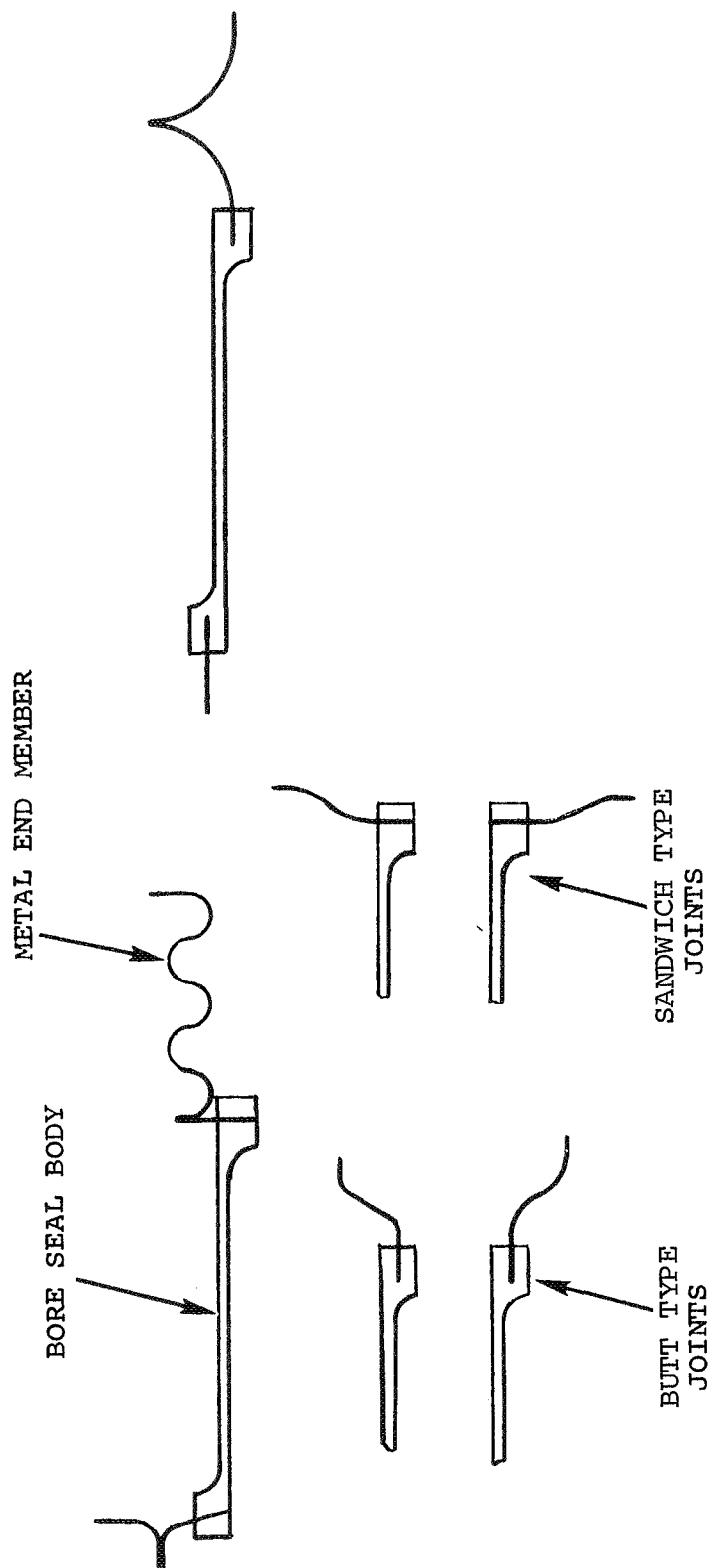


Figure 2.4-20. Typical Bore Seal Design Concepts.

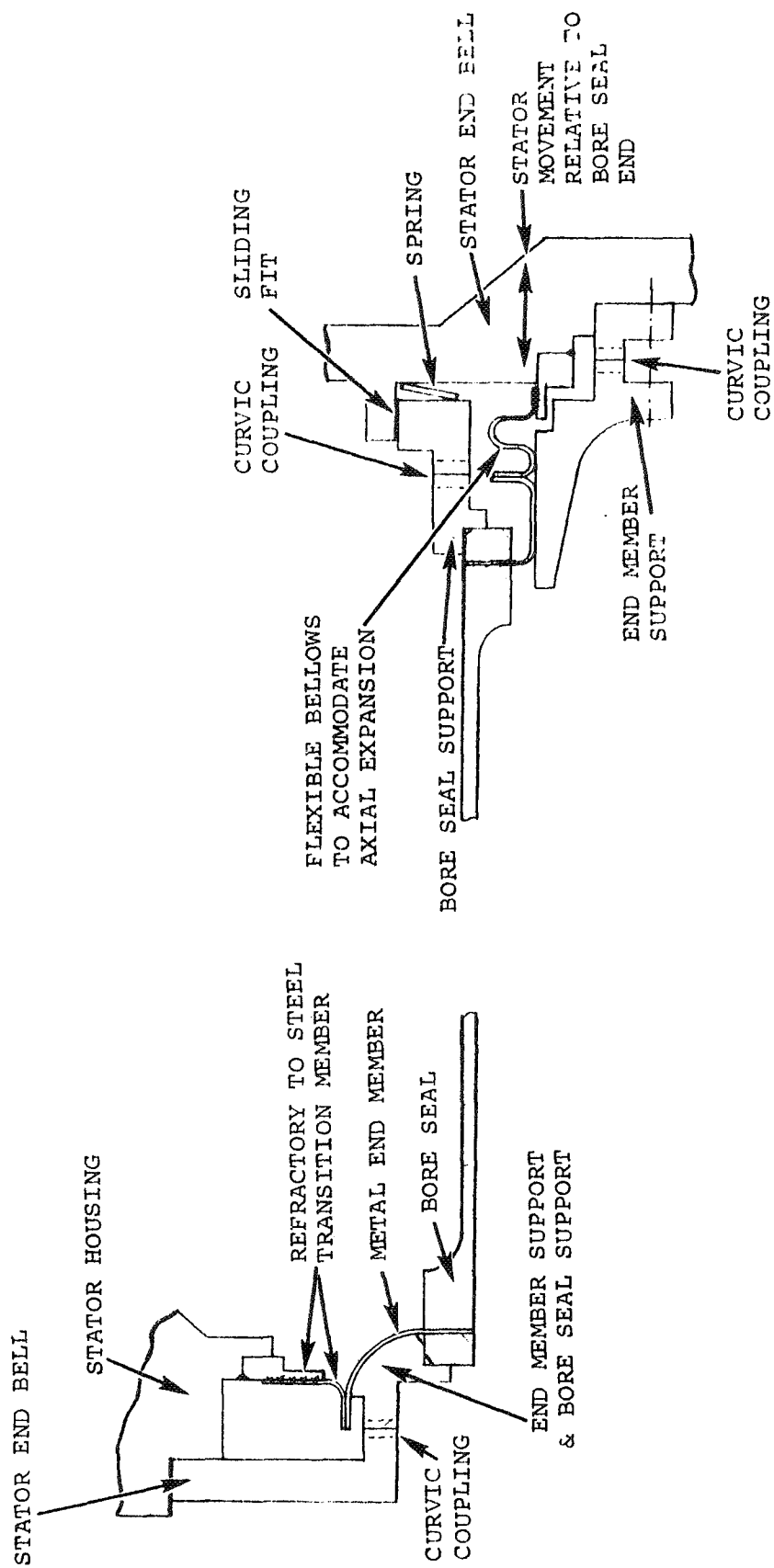


Figure 2.4-21. Proposed Bore Seal End Member Configuration.

couplings) are used. One fixture provides radial support for the bellows type metal end member and the other supports the ceramic bore seal ends that carry the axial loading. Besides providing support to reduce the stress in a fairly flexible arrangement, the use of this bore seal support arrangement will allow an increase in the thickness of the ceramic bore seal. This is because the radial teeth and positive support points provide accurate alignment of the bore seal relative to the stator. In their absence, the bore seal could conceivably become cocked* and more assembly tolerance build-up allowances would have to be accounted for in the air gap.

As the concepts illustrated on Figure 2.4-19 were conceived late in Phase I studies and as it was felt additional refinements could be conceived with continued study in Phase II, detailed stress analyses were not carried out on the configuration. The reader is referred to Part II of this report for the latest configuration and resulting stresses.

2.5 CONCEPTUAL DESIGNS

There were three conceptual designs made from the three Base Designs discussed in Section 2.3. Recall that the three base designs were selected from elimination and optimization screening studies. The base designs were then subjected to further detailed analysis to seek more refined designs. These latter designs constitute the Conceptual Designs.

*The bore seal is assembled in a loose fit in the bore at room temperature and due to differential (radial) thermal expansion, becomes even more loose at operating temperatures.

Conceptual Design Layouts

Because the basic design features of the three conceptual designs are similar in many respects, one discussion will be given for the three. Differences between the three will be pointed out. The three layouts are illustrated on Figures 2.5-1 through 2.5-3.

The significant features of the alternator electrical designs are found in the rotor and stator configurations. The remaining components that make up the complete generators are considered to be work-horse parts relative to the electrical designs and were not optimized in the Phase I studies except the bearing, shaft seal and stub shaft configurations furnished by General Electric Company.

The stators are hermetically sealed to permit operation without subjecting the stator components to corrosion damage by either air or moisture external to the generator or by potassium vapor of the rotor cavity. The bore seal in the radial gaps between the rotors and stators is high purity aluminum oxide. The stator magnetic frames consist of the iron-cobalt magnetic alloy, Hiperco-27. Electrical conductors are of Inconel-clad silver and insulated with Anadur (glass fibers). The rotor is composed of solid forgings of high-strength H-11 steel. Electrical power is taken out of the generators through bus rings and terminals located on the anti-drive ends.

Stator coolant enters and leaves through toroidal shaped manifolds located on the stator frame outside diameter. The manifolds are bonded directly to the stator frame.

Rotor coolant enters the anti-drive end and discharges out the drive end with the drive-end bearing lubricant. A special rotor-cavity

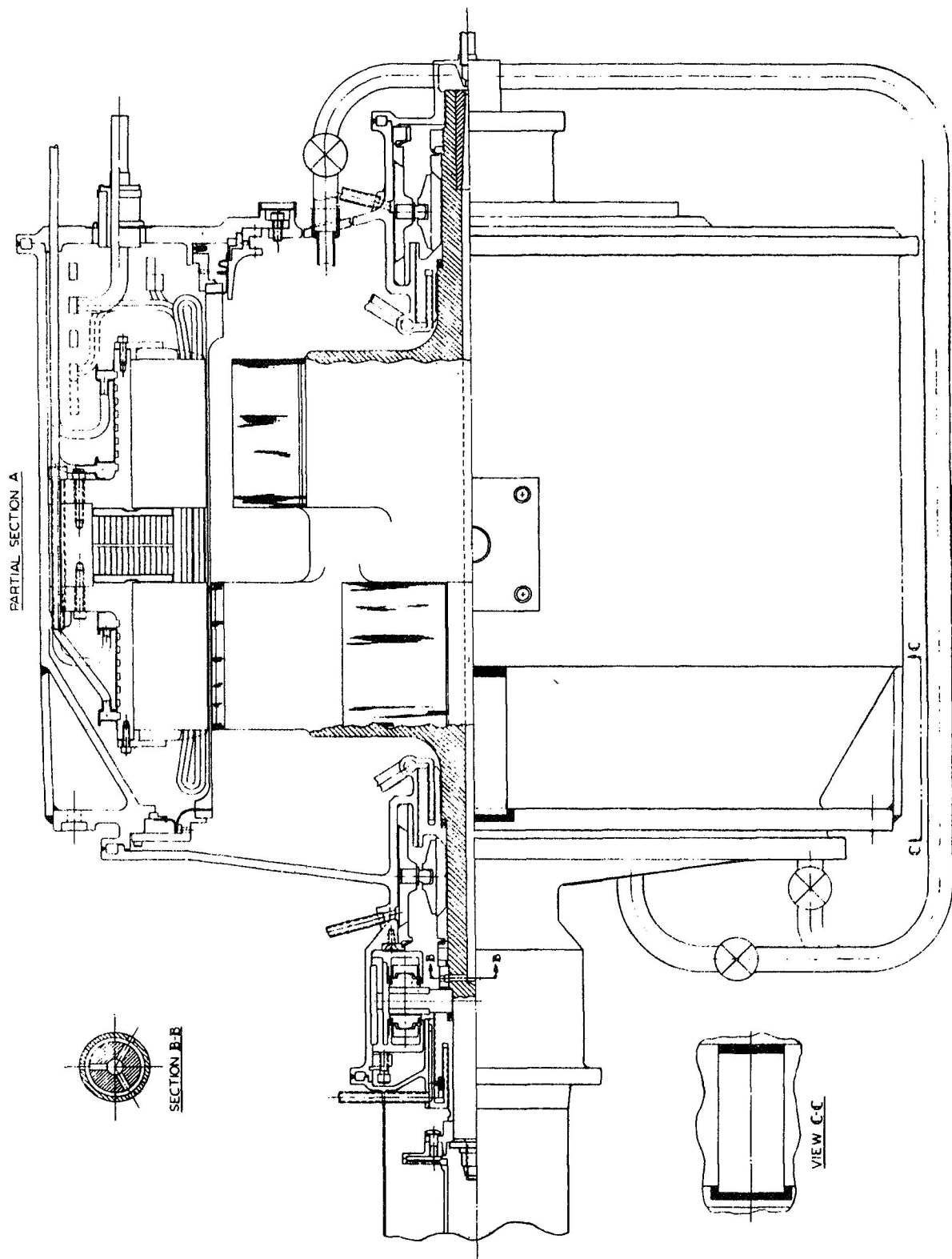


Figure 2.5-1. 733 kVA Alternator, 0.75 P.F. - 800°F, Potassium Cooled - 19,200 rpm - 1600 Hz, 480 Volts L-N.

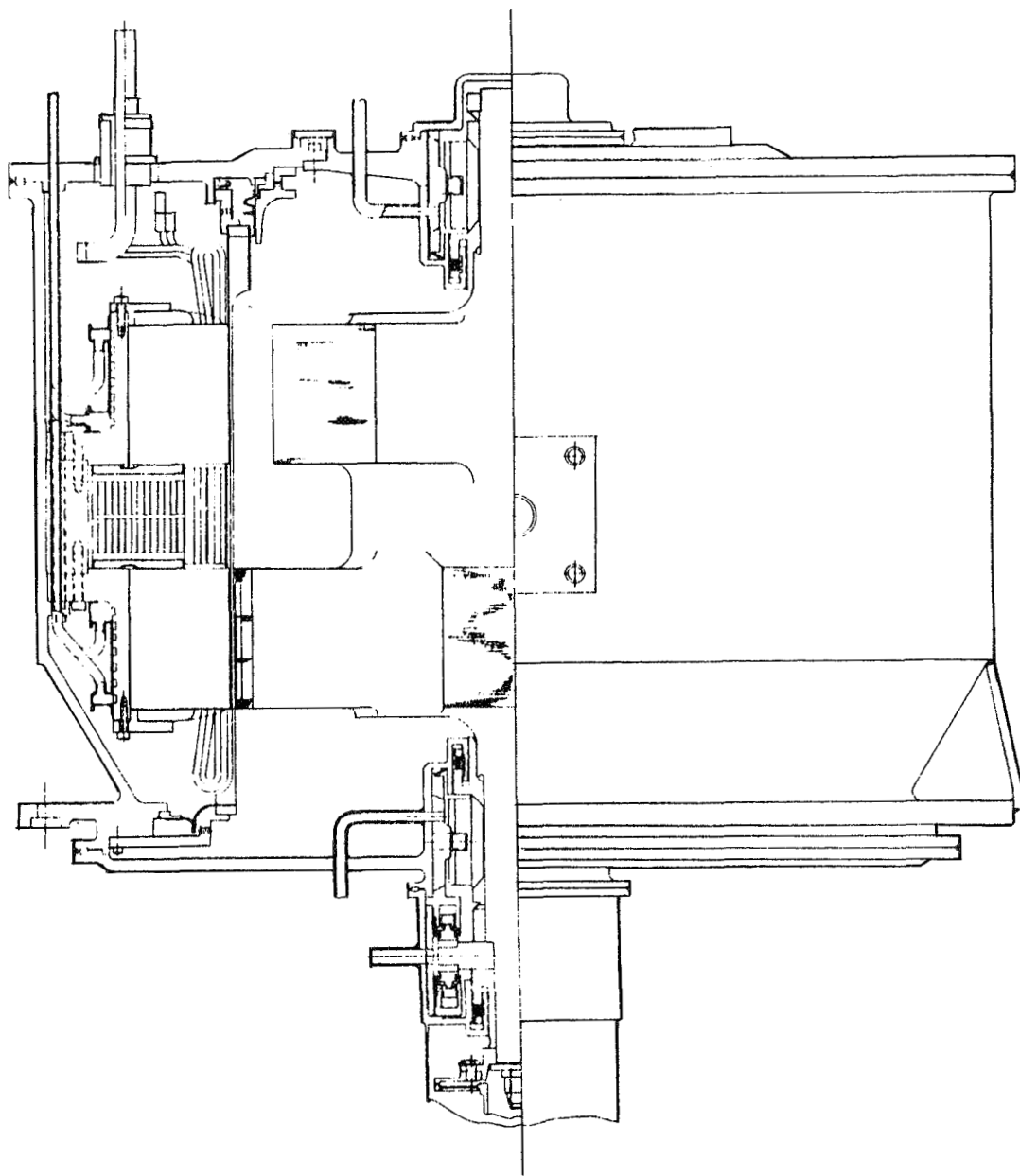


Figure 2.5-2. 733 kVA Alternator, 0.75 P.F. - 400°F Coolant Organic -
24,000 rpm - 1600 Hz, 480 Volts L-N.

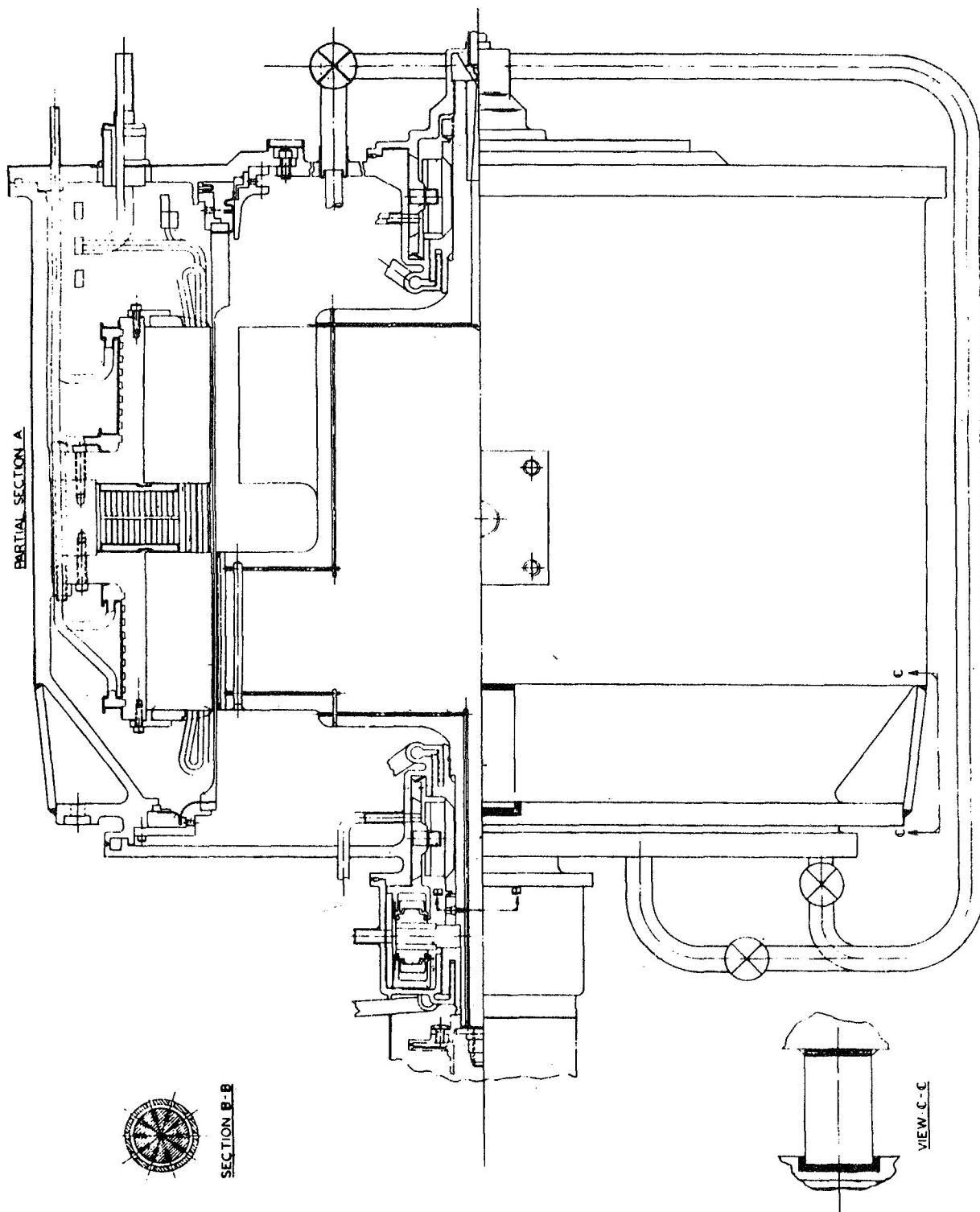


Figure 2.5-3. 733 kVA Alternator, 0.75 P.F. - 1100⁰F, Stator Potassium Cooled - 19.200 rpm - 1600 Hz - 480 Volts I-N.

scavenge system is included as part of each rotor flow circuit. The one exception to this is the 400^oF coolant organic design where cooling is through the stub shafts. None goes through the rotor as discussed in Section 2.4. The rotor coolant for the 1100^oF stator, 800^oF rotor design passes up into the poles. This undesirable feature is necessary to cool the pole tips with the large heat flux carryover from the stator to the rotor. The pole tips are laminated to minimize losses. Features of the rotor pole tip design are discussed fully in Section 2.4.

Pivoted-pad journal bearings support the rotors. The bearing lubricant is separated from the rotor cavities by non-contact viscosity screw-pump seals (viscoseals). The rotor cavities are further isolated by molecular vacuum pumps between the viscoseals and rotor cavity.

On the illustrations shown, a drive-end shaft seal-to-space is not shown. All that is required is a low leakage barrier or seal to separate the generator bearings from the turbine bearings. The two bearing cavities would be connected by a hermetic shroud. The actual turbine and alternator frames would be connected by an open, cold-frame fixture.

The outer mounting flanges, which double as the hermetic shrouds for the alternators, act as the support structure. The full weight of each generator is carried through two center of gravity mounting pads, one on each side at the horizontal centerline.

One of the features of the generators is the disassembly capability. Breaking the bolted joint on the anti-drive end bell and the top weld joint on the drive end bell is all that is required to permit removal of the stator assembly intact, complete with bore seal, terminals, and cooling ducts. This feature is included to permit, if necessary,

complete separation of the potassium-exposed rotor cavity from the stator cavity prior to breaking any of the hermetic (potassium exposed) seals separating them.

The required number of field coil ampere-turns may be obtained by either a large number of low current turns or a few large current turns. The choice is dependent on a number of factors such as voltage regulator exciter (VRE) requirements, heat transfer, material availability, fabrication requirements, etc. A design using a large number of turns is easiest to fabricate but difficult to cool. The latter reason led to the choice of a few straps rolled one on top of the other. The number of turns will be dependent upon the VRE requirements but is presently estimated to be about fifty. Heat is carried out of the straps on each side where thermal contact is made with the cooling ducts. Center spacers between the straps separate the two bundles. Dimensional fabrication tolerance buildup is accommodated by forcing the two bundles outward onto the cooling ducts by use of the center spacers. Interlaminar electrical insulation separates the strap layers.

The stator punchings are 0.004-inch thick Hiperco-27 magnetic alloy, have a semi-closed slot to reduce pole face losses, and are assembled into an unwelded stack to reduce iron losses. A 0.0004-inch thick layer(s) (total) of plasma arc sprayed alumina constitutes the interlaminar insulation. The block of materials shown on the ends of the stacks are to hold and align the unwelded stacks and to prevent the 0.004-inch thick teeth from flaring. The BeO ceramic heat sink between the stacks (over the armature wires, beneath the field coil I.D.) acts as a spacer to serve the same function between the stacks.

The punching diameter is about twenty inches for the three designs which is about five inches over the allowable width of rolling 0.004-inch thick Hiperco-27 on a regular, commercial basis; special rolling will be required for procurement of the material because of this.

There are four to six bus rings on the anti-drive end of each generator; three are for line connections and one to three are for neutral connections to the phases. The parallel phase groups of the armature are connected to each bus ring in such a manner that bus ring copper losses are held to a minimum which in turn permits the rings to be cooled by radiation to the stator cavity walls. The maximum bus ring temperature can be expected to be less than 1000°F with an 800°F sink temperature on the end-bell wall. The one to three neutrals are shown and the actual quantity will be finalized depending on the final design specifications stating a common, external or internal, or separate neutrals connection requirement.

Two primary considerations are the long-term metallurgical stability of the conductor joints and the containment of the silver. Silver migrates (diffuses) at elevated temperature and affects the electrical properties of insulation materials and affects the electrical properties of insulation materials around it. Sealing the wire ends with nickel provides an excellent barrier since the two metals are mutually insoluble.

Several methods of sealing the silver conductor core from the high temperature vacuum environment were considered, including nickel plating, brazing and welding. Past Westinghouse feasibility evaluations have shown that, after removal of the silver core from the conductor ends to a depth of 1/8 to 1/4 inch, pure nickel "A" plugs or TD nickel plugs could be fusion welded to the thin conductor nickel cladding. Joining

accomplished by TIG welding, electron beam welding or plasma needle-arc welding (a "fine-point" TIG process).

Depending on the choice of bus-ring material, the ends of the conductors are either brazed or welded to the bus-rings or collector rings. Choice of precipitation hardened CUBE alloy limits the process to brazing as fusion processing destroys the precipitation hardening and reverts the melt zone back to copper. To further decrease the possibility of silver diffusion out the conductor ends, the connection configuration at the bus-ring can be a "blind-hole" socket that the wire end is bonded into. Bosses are included in the joint configuration to increase the surface area, thereby reducing electrical resistance across the joint.

There is one terminal per bus ring located such that each carries not more than 2000 amperes full-load current. Because of the temperatures brought on by the high current densities and because of the limited space available, standard terminal feed-throughs cannot be used. Special terminal studs are required.

The conductors through the terminal feed-throughs are sized to have sufficient cross-sectional area to permit them to be self-cooled by radiation and/or natural convection. They are nickel plated and black oxide coated to obtain an emissivity of at least 0.4 on the outer surface. Radiation cooling alone limits the conductor temperature to less than 1000^oF at full-load current in an 800^oF ambient.

One of the more vital components in the generator is the ceramic bore seal which separates the rotor cavity from the stator cavity. This separation is necessary because the stator components are incompatible with the potassium vapor in the rotor cavity. Since the bore seal is in

a continuing state of development, the bore seal design for the conceptual design generators was not finalized; each design must be treated individually when complete operating/fabrication conditions are specified or known. The configuration shown is that previously discussed in Section 2.4.

The bore seal proposed for 800^oF coolant generators is a 99.8% pure alumina cylinder with Cb-1Zr refractory alloy metal end members. Since the bore seal will most likely be hermetically joined to non-refractory material in the generator, the use of refractory end members requires that an equally sound transition joint also be provided. Seal-weld joints are made during final assembly by electron-beam welding or TIG welding. The various development states of these bore seal concepts are summarized in the Appendix of Reference 2.2-2.

In addition to alleviating the bore seal end member design problems, the proposed bore seal end member configuration permits more accurate control of the alignment of the bore seal in the radial gap. The net effect of this is that less radial gap is required for assembly tolerances and then (1) the gap may be decreased, (2) the rotor may be allowed to creep more, or (3) the bore seal may be thickened.

The thicknesses required for the alternator designs were determined to be greater than the minimum thicknesses that could be fabricated by suppliers proficient in this field. In other words, the minimum permissible bore seal thickness is greater than the possible minimum; therefore, fabrication development problems should be reduced somewhat so far as bore seal thickness goes. Because the brazing alloy in the joints might be one of the active-metal alloys containing beryllium (reference 2.2-2, Appendix), there may be a limit to the size of the

bore seal that can be brazed with presently available brazing facilities. The reader is referred to Reference 2.4-19 for a more detailed discussion of these design considerations.

One of the unique features of the generators is the rotor cavity scavenge system. This system acts to clear the rotor cavity of liquid during startup and keep it clear during operation. It is designed as an integral part of the generator shaft seal system and works automatically without external controls. Tests which demonstrated and verified the performance of the system were completed and are reported in detail in the Appendix of Reference 2.2-11.

Bearings and Seals

The bearings are four-pad, pivoted pad hydrodynamic journal bearings that use a spherical pivot to accommodate angular misalignment. Solid carbides were chosen for the wear surfaces. Special mounts are used to compensate for the relative thermal expansion between the solid carbide pieces and the steel housings and stub shafts. For more detail see Section 5.8.

A circulating flow of potassium through each flooded bearing cavity acts as the coolant and lubricant. Special pressurized lubricant feeds in the pads are not required since the hydrodynamic action of the pads forms the wedging phenomenon necessary for full film lubrication. Volutes can be provided on the discharge side of the bearing to collect the lubricant and convert its tangential velocity into static pressure. With these volutes, a net pressure rise can occur through the bearings.

The shaft seals are described in detail in Sections 5.9 and 5.10. They are of the non-contact type and form a barrier by dynamic action of

the liquid being sealed. They consist of liquid screw-pump seals (visco-seals) and vapor shaft seals in the form of molecular vacuum pumps. A molecular vacuum pump operates on the same principle as a diffusion pump except that the momentum is imparted to the molecules by a rotating cylinder rather than by a fluid. The configuration is that of a smooth shaft (or cylinder) running in a close-fitting (e.g., 1.5 to 2 times the bearing clearance), helical threaded housing. See Section 5.9. The liquid seals prevent gross liquid from escaping the bearing cavity whenever the seal pumping head becomes greater than the pressure between the bearing cavity and the rotor cavity. At that condition, there is established within the seal a liquid-to-saturated-vapor interface which, because of the combination of high speed and low viscosity, is inherently unstable, resulting in loss of fluid. A special trapping arrangement is used to direct this fluid back into the screw seal, known as the General Electric zero leakage seal concept, Section 5.9. This liquid also may be ducted to the injection pump where it is recombined with the lubricant and coolant flow. This approach was established by Westinghouse under SPUR contract. It is shown in detail in Section 5.9. Beyond the liquid trap there is still another barrier in the form of a molecular pump in order to reduce the potential leakage of vapor to the vacuum side of the seal. The vapor seals accomplish this by pumping the rotor cavity down to a lower pressure such that the vapors must be superheated at the temperature conditions found in the rotor cavity. Thus, any wet vapors from the liquid seal interface eventually passing into the rotor cavity will have become dry in the process.

A second more beneficial effect is derived from the reduced rotor cavity pressure achieved by the vapor seals; the windage losses are reduced as discussed in Section 2.4.

Conceptual Design Performance Data

The electrical performance data for the three conceptual designs is presented in Table 2.5-1. The efficiency is about 93 percent for all three designs at both the 450 and 550 kw rating. The efficiency includes the windage friction losses but does not include the bearing and seal friction losses which are discussed in Chapters 5 and 6 and summarized in Chapter 10.

The difference between the total weights and the electrical weights given is made up of the end bells, frame housing, bus rings and terminals. The weights of these components were not minimized and are somewhat heavy as a result. They were left in a conceptual form for Phase I and will be minimized in Phase II.

The performance of the rotor design is presented in detail in Section 2.4 and will not be repeated here. The only additional consideration not previously covered is the overspeed capability of the various designs. It was determined the 19,200 rpm, 800^oF rotor designs could operate for about one year at 20 percent overspeed. This is summarized in Table 2.5-2. The 24,000 rpm, 400^oF coolant case, with proper cooling to eliminate any creep growth of significance, could operate indefinitely at 20 percent overspeed. However, the maximum stresses exceed the guidelines set forth at the start of the study (e.g., 72 percent yield strength) but they do not exceed the actual property data limitations. Thus, if the 24,000 rpm rotor survived the initial overspeed surge, it should survive indefinitely. Note however, that plastic yielding may deform the rotor such that, even though the rotor may not have failed catastrophically, it must be considered failed and unsuitable for three to five years of life.

TABLE 2.5-1

ELECTRICAL PERFORMANCE ON CONCEPTUAL DESIGNS

	No. 1			No. 2			No.3		
	450 kWe	550 kWe	550 kWe	450 kWe	550 kWe	550 kWe	450 kWe	550 kWe	
SPEED (RPM)	19200	19200	19200	24000	24000	19200	19200	550 kWe	
FREQUENCY (Hz)	1600	1600	1600	1600	1600	1600	450 kWe	550 kWe	
POLES	10	10	10	8	8	10	450 kWe	550 kWe	
COOLANT TEMPERATURE (°F)	800°F	800°F	800°F	400°F	400°F	800/1100°F	450 kWe	550 kWe	
TOTAL STATOR LOSSES (kW)	35.06	44.45	44.45	35.81	44.07	32.00	42.99		
STATOR IRON LOSSES (kW)	23.35	25.19	25.19	25.74	27.86	18.66	20.51		
ARMATURE COPPER LOSSES (kW)	8.720	14.78	14.78	7.80	12.81	9.35	16.51		
FIELD COPPER LOSSES (kW)	2.990	4.47	4.47	2.27	3.40	3.99	5.97		
TOTAL ROTOR LOSSES (kW)	0.17	0.19	0.19	0.44	0.47	0.17	0.19		
POLE FACE LOSSES (kW)	0.06	0.08	0.08	0.08	0.11	0.06	0.08		
WINDAGE LOSSES (kW)	0.108	0.108	0.108	0.36	0.36	0.108	0.108		
EFFICIENCY (%)	92.7	92.5	92.5	92.5	92.6	93.3	92.8		
WEIGHTS									
ELECTRICAL									
ROTOR (w/o STUB SHAFTS)	837	837	837	643	643	828	828		
STATOR	352	352	352	234	234	367	367		
	485	485	485	409	409	461	461		
TOTAL (EXCLUSIVE OF BEARINGS, HOUSINGS, STUB SHAFTS)	1200+	1200+	1200+	850+	850+	1250+	1250+		

TABLE 2.5-2

OVERSPEED

19,200 RPM, 800^oF ROTOR

0.12% CREEP IN 3 YEARS AT DESIGN SPEED

0.3 % CREEP IN 5 YEARS AT DESIGN SPEED

0.6 % CREEP IN 3 YEARS AT 20% OVERSPEED

CONCLUDE: COULD OPERATE AT 20% OVERSPEED FOR ABOUT A YEAR.

The temperature distribution of the stator designs was based on the designs and studies of Section 2.4. The conceptual design stator cooling configuration selected from the parametric study consists of a circumferential manifold on the stator frame with cooling fins between the a-c windings in the area between the stacks. Separate manifolds are included on the sides of the field coil. Temperature distributions were made for the above cooling configuration on all three conceptual generator designs for various conditions as shown in Table 2.5-3 and Figures 2.5-4 through 2.5-16. The temperatures determined from the studies are highlighted in Table 2.5-4. Details of these studies follow.

The temperature distributions determined for the three conceptual designs at normal operating conditions of design power output (450 kW) and maximum continuous load rating (550 kW) with slot conductance of $0.2 \text{ watt/in}^2\text{-}^\circ\text{C}$ are shown on Figures 2.5-4 and 2.5-5 and Figures 2.5-12 through 2.5-15. From this data, it is evident that this cooling configuration is not sufficient to cool the stator with 1100°F coolant at maximum continuous load rating as maximum temperatures exceed the conductor limiting temperature of 1300°F . It is apparent that, should it be desirable to use 1100°F coolant in the stator, additional means of extracting stator heat will be required.

Temperature distributions presented in Figures 2.5-6 through 2.5-9 were prepared to determine the minimum slot conductance that does not result in excessive stator temperatures. These resulting maximum stator temperatures plotted against slot conductance are shown in Figure 2.5-17 from which the minimum slot conductance of $0.04 \text{ watts/in}^2\text{-}^\circ\text{C}$ was determined. This value of slot conductance is within the range of values that can be expected for the proposed slot configuration. These

TABLE 2.5-3

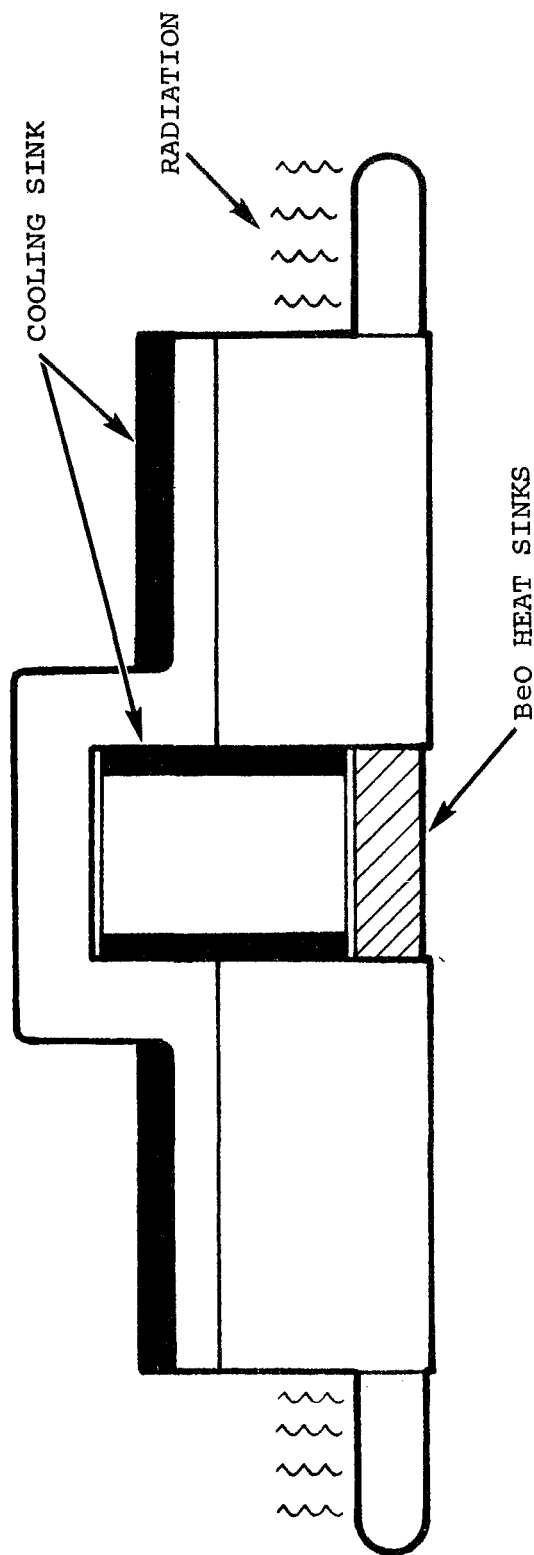
MAXIMUM TEMPERATURES OF CONCEPTUAL DESIGNS

CONCEPTUAL DESIGNS	° F		
	1	2	3
450 kW, 0.2 Slot Conductance	1024	748	1270
550 kW, 0.2 Slot Conductance	1106	861	1345
550 kW, 0.1 Slot Conductance*	1112	-	-
550 kW, 0.05 Slot Conductance*	1119	-	-
550 kW, 0.01 Slot Conductance*	1162	-	-
550 kW, 0.002 Slot Conductance*	1276	-	-
550 kW, 0.2 Slot Conductance, 3 ϕ Short Circuit	1168	-	-
550 kW, 0.2 Slot Conductance, Rotor Heating from Stator Included	1078	-	1284

* Reference Figure 2.5-7

TABLE 2.5-4

STATOR COOLING SUMMARY OF CONFIGURATION AND RESULTS



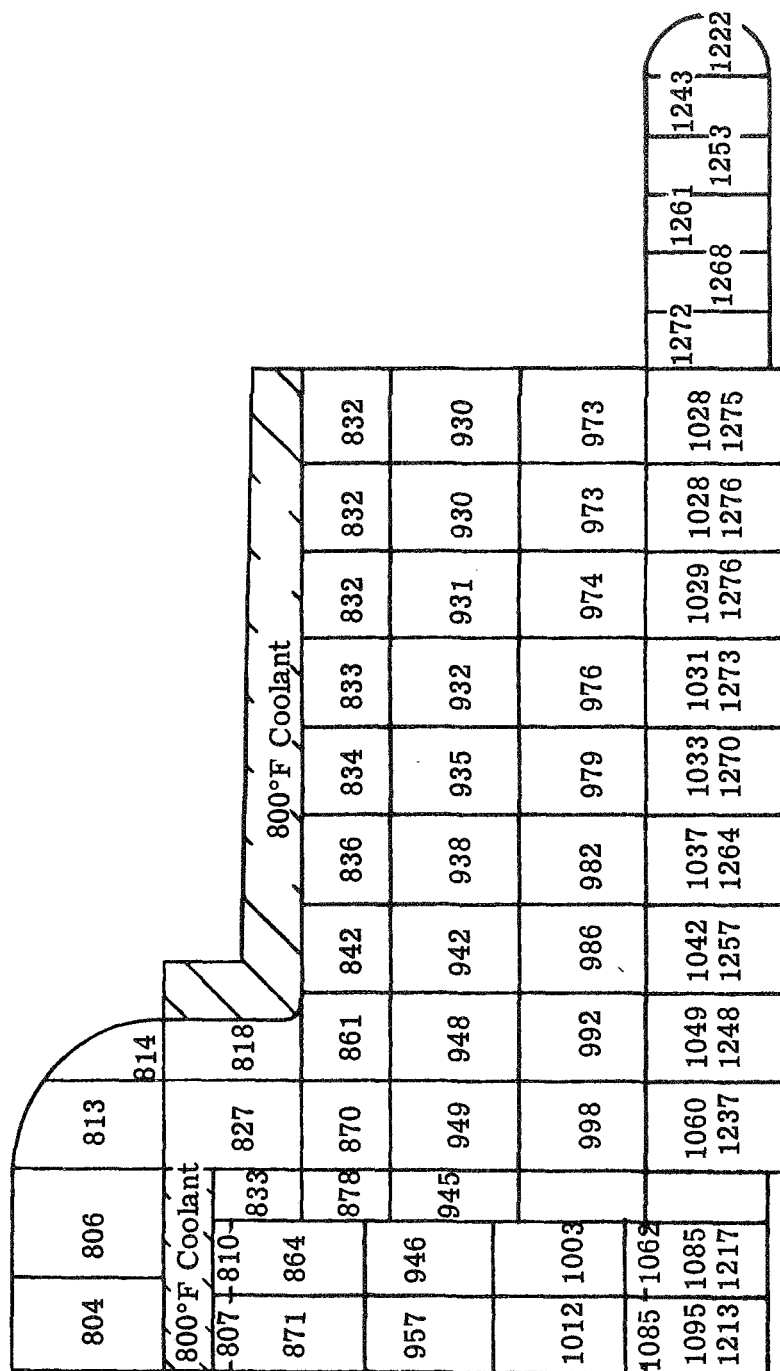
MAXIMUM TEMPERATURE	No. 1		No. 2		No. 3	
AC END TURNS	450 KW	550 KW	450 KW	550 KW	450 KW	550 KW
AC CENTER	1024°F	1106°F	748 °F	853°F	1270 °F	1345°F
BORE SEAL	1003°F	1072°F	687 °F	761°F	1258 °F	1323°F
STATOR TOOTH		964°F				1108°F
DBS	1011°F	1080°F	705 °F	779°F	1258 °F	1322°F
FRAME	959°F	1000°F	641 °F	680°F	1220 °F	1262°F
FIELD COIL	854°F	870°F	459 °F	467°F	1147 °F	1164°F
	950°F	1001°F	595 °F	639°F	1231 °F	1284°F
SINK TEMPERATURE	800°F		400°F		1100°F	

803	805	810	811										
800°F Coolant				800°F Coolant									
805	807	825	821	814	853	854	849	835	830	829	829	829	829
851	847												829
913	907	910	916	919				918	918	918	918	918	918
950	946							955	956	956	957	958	959
976	969	948	951	954									
979	979	994	998	1000	1000	999	1000	1000	1001	1003	1005	1006	1011
1003	1002	986	999	1000	1002	1003	1005	1007	1007	1009	1009	1011	1014
													1016
													1019
													1022
													1024
													1024
													1023
													1016

Figure 2.5-4. Number 1 Design Stator Temperature Distribution (°F) 450 kW Rating, 0.2 Watts/ in.₂ Slot Conductance.

804	806	813	814										
				800°F Coolant									
807	809	827	818	832	876	870	862	844	838	836	836	836	836
869	862												
948	935	950	951	949	947	946	946	946	946	946	946	946	946
998	990	994	996	996	996	996	996	996	996	997	997	998	998
1037	1026	1057	1063	1066	1068	1069	1071	1074	1077	1080	1111	1116	1116
1042	1040	1081	1081	1082	1084	1086	1088	1092	1096	1102	1116	1119	1119
1086	1085												1105

Figure 2.5-7. Number 1 Design Stator Temperature Distribution (°F) 550 kW Rating, 0.05 Watts/in.² Slot Conductance.



804	806	813	814	800°F Coolant										1151	1163	1167	1168	1165	1153				
800°F Coolant				807-810	833	827	818	800°F Coolant										1151	1163	1167	1168	1165	1153
871	863				876	870	862	844	838	836	836	836	836	836	836	836	836	836	836				
950	939								945	944	944	944	944	944	944	944	944	944	945				
1000	992																						
1034-1024					992	993	995	996	996	996	996	996	996	997	998	999							
1040	1040				1077	1082	1083	1085	1085	1087	1087	1089	1089	1093	1097	1103	1103	1165	1153				
1129	1125				1105	1103	1104	1105	1105	1107	1107	1110	1110	1114	1119	1128	1128	1168	1153				

[illegible]

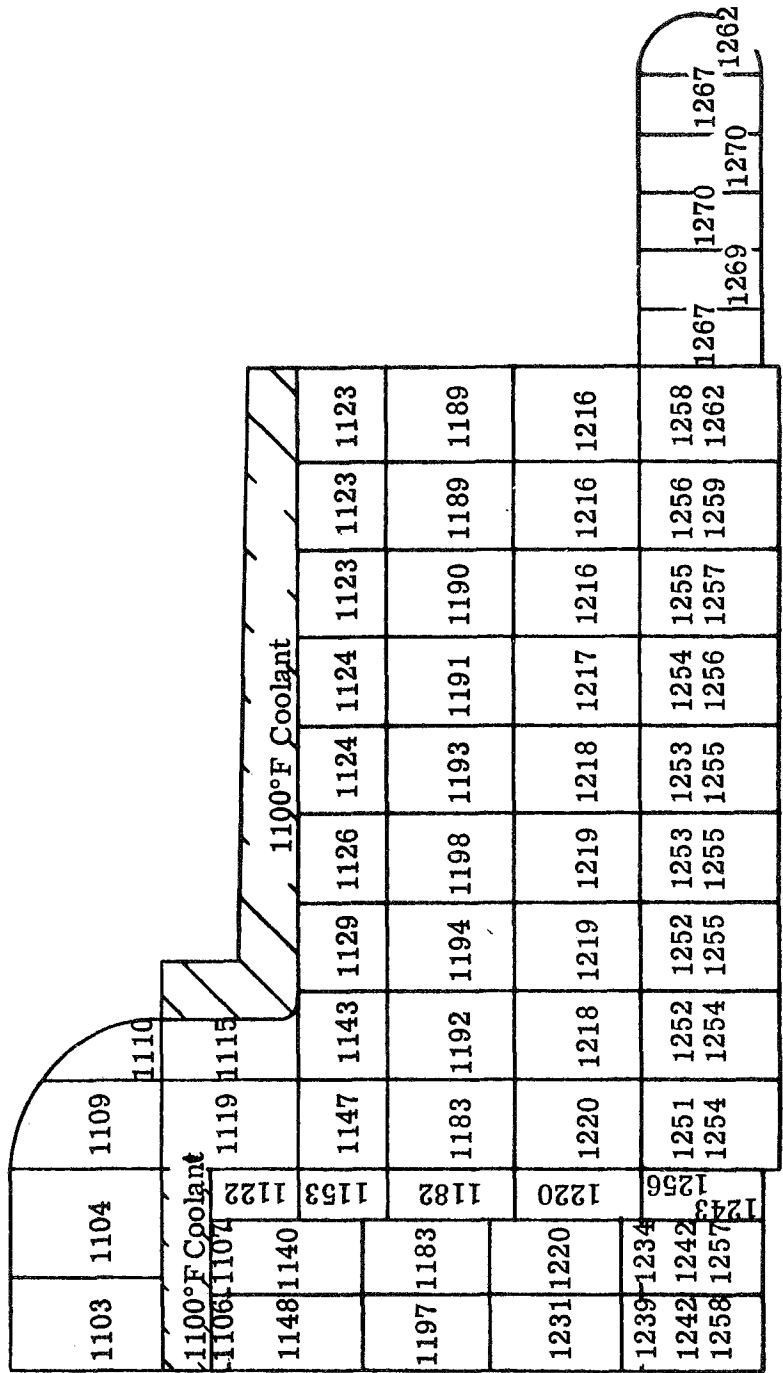
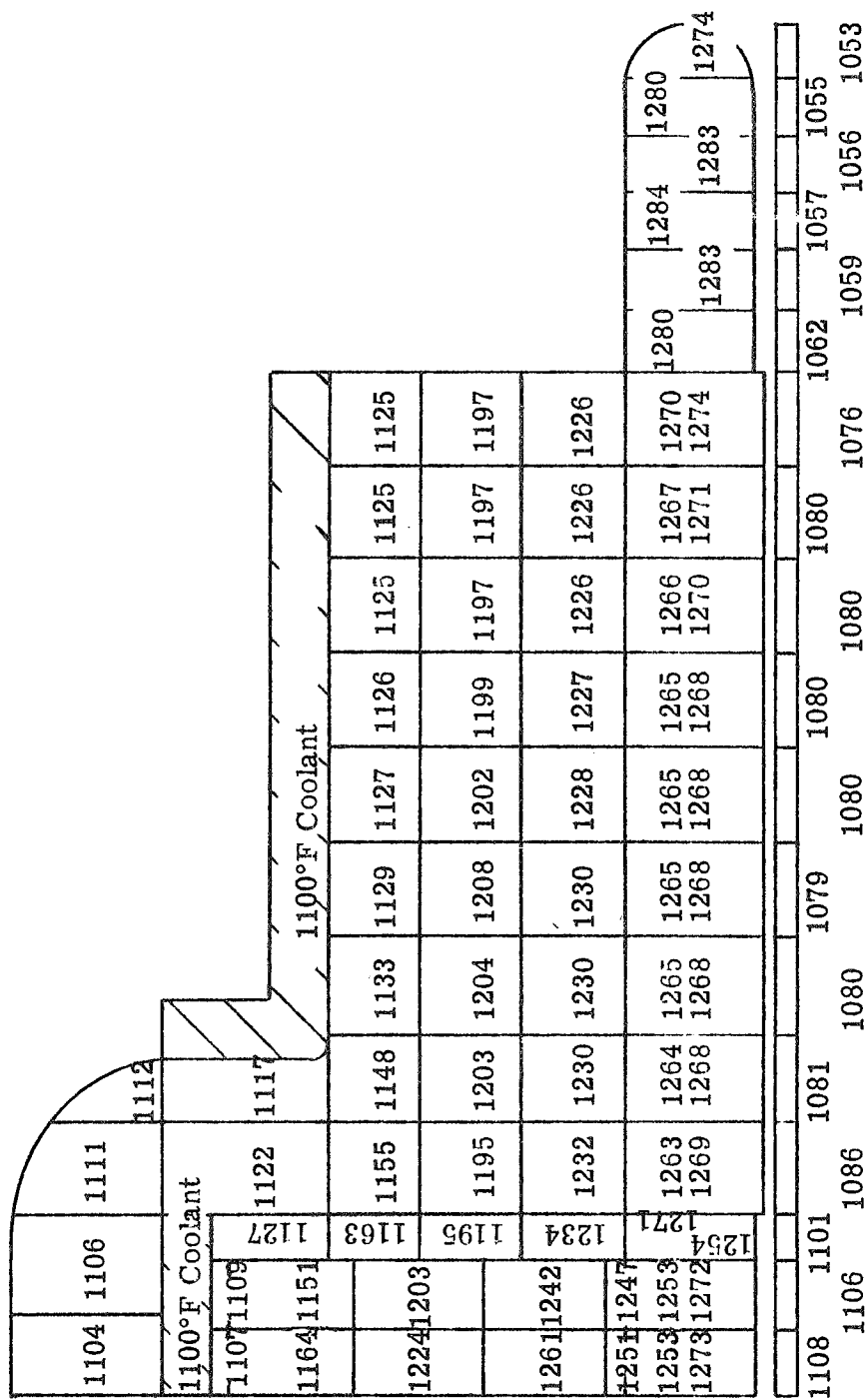


Figure 2.5-14. Number 3 Design Stator Temperature Distribution (°F) 450 kW Rating, 0.2 Watts/in. 2_oC Slot Conductance.

1104	1106	1113	111										
1100°F Coolant			1100°F Coolant										
1108	1110	1126	1100°F Coolant										
1169	1156	1172	1164	1157	1139	1135	1132	1131	1130	1130	1130	1130	
1237	1216	1212	1211	1222	1225	1230	1223	1219	1218	1217	1217	1217	
1284	1267	1264	1262	1258	1258	1259	1257	1255	1254	1254	1255	1255	
1295	1287	1299	1310	1311	1311	1311	1312	1313	1315	1318	1322	1337	
1299	1298	1320	1316	1315	1315	1315	1316	1317	1319	1323	1328	1342	
1323	1322	1299										1344	
												1331	

Figure 2.5-15. Number 3 Design Stator Temperature Distribution (°F) 550 kW Rating, 0.2 Watts/in.² Slot Conductance.



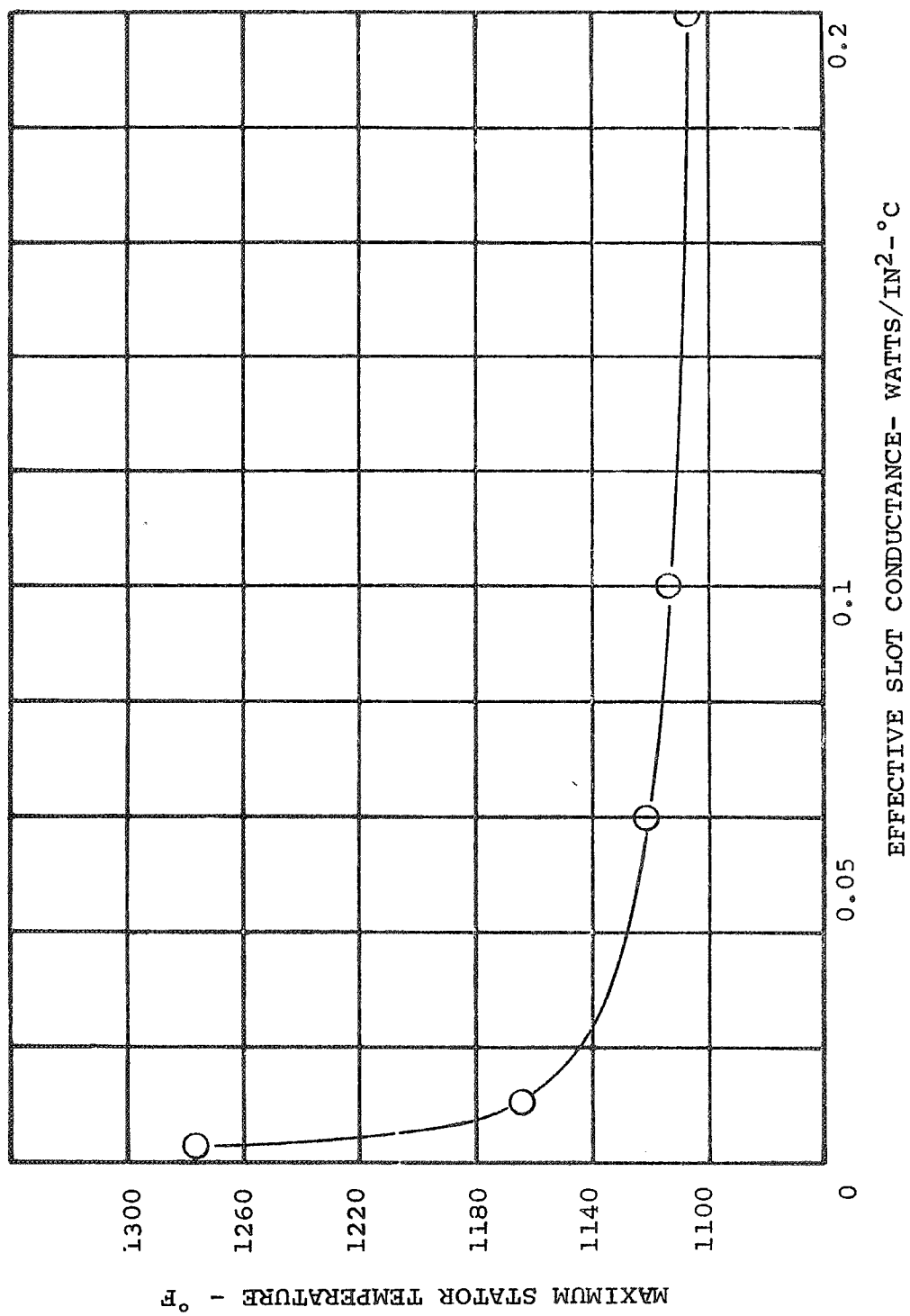


Figure 2.5-17. Stator Maximum Temperature Versus Slot Conductance.

temperature distributions were determined for normal operating conditions at maximum continuous load rating (550 kW) of the Number 1 generator conceptual design.

Figure 2.5-10 presents the temperature distribution for five seconds of three-phase short-circuit condition after steady-state conditions of normal operation at 550 kW load rating with $0.2 \text{ watts/in}^2\text{-}^\circ\text{C}$ slot conductance for Number 1 design. These data show a 62°F increase in the maximum stator temperature to 1168°F and indicates that for the lower coolant temperatures (400° and 800°F), the generator will meet the requirement of remaining operational after undergoing a three-phase short-circuit condition for five seconds. Three-phase short-circuit condition temperature distributions were not prepared for Numbers 2 and 3 designs due to lack of time. In all of the above temperature distributions, all stator losses were rejected to the stator coolant by conduction and convection and to the end bells and housing by radiation from the end turns.

An evaluation of stator temperatures and cooling for conceptual designs Numbers 1 and 3 was conducted, taking into consideration heat transfer to the rotor in addition to cooling as described above. The resulting temperature distributions are presented in Figures 2.5-11 and 2.5-16. The addition of heat transfer to the rotor results in a reduction of the stator maximum temperature by 28°F to 1078°F for design Number 1 and by 61°F to 1284°F for design Number 3. The stator losses rejected to the rotor were 2750 watts (6.2 percent of stator losses) and 7100 watts (16.5 percent of stator losses) for conceptual designs Number 1 and 3 respectively. The larger temperature reduction and greater amount of heat transferred to the rotor for the conceptual

design Number 3 is due to the greater temperature difference between the 1100^oF stator and 800^oF rotor. Heat transfer to the rotor was by radiation from the armature windings and stator teeth to the bore seal and by radiation and forced convection from the bore seal to the rotor with the temperature of the rotor being held constant.

A temperature distribution with heat transfer to the rotor for conceptual design Number 2 was not determined due to lack of time and appropriate property data on oil vapor from which to determine the heat transfer coefficient.

Investigation of field coil configuration was conducted with seven designs of both horizontal and vertical strap wound arrangements. Field coil cooling for the horizontal strap configurations was provided by two methods: (1) a central duct dividing the coil into two halves, and (2) by ducts on both sides of the coil. Cooling for the vertical strap configurations was by a duct on the O.D. of the coil. Based on both electrical and mechanical considerations, the field coil and cooling configuration selected consisted of two horizontal counter-wound strap coils separated by ceramic insulation with cooling ducts on the outside of each coil. They are electrically isolated from the coil by ceramic. This arrangement allows both coil leads to be located at the coil O.D. with the common connection at the I.D. of the coil. The cooling ducts on the coil sides provide a short direct thermal path for each turn to the coolant and additional cooling for the stack and a-c windings between the stacks. This field coil cooling configuration differs from that used in determining the temperature distributions for the conceptual designs. This is due to the fact that the final cooling configuration was selected after the majority of the temperature distributions had been

prepared using a pre-selected, fixed (preliminary) cooling arrangement. Time limitations did not allow for re-calculating temperature distributions with the new field coil cooling configuration. For comparison, however, a temperature distribution of the Number 1 design was prepared with this new field coil cooling method for the 550 kW rating and is presented in Figure 2.5-18. Comparing these temperatures with those of Figure 2.5-5 shows substantial reductions in temperatures of the field coil and the a-c windings between stacks and a 32⁰F reduction in stator hot-spot temperature.

The basic stator cooling configuration consisting of a circumferential coolant manifold on the stator frame with coolant ducts on the sides of the field coil will be finalized during Phase II of this program.

2.6 MISCELLANEOUS STUDIES

Separate studies, not directly related to deriving the conceptual designs, were carried out to provide additional data on operation and performance. These studies considered motoring startup performance, magnetic force unbalances between the rotor and stator and reliability engineering evaluations. The motoring startup study was done for purposes of identifying the inherent motoring capabilities up to 120 percent of design speed and any steps that might be taken to improve the motoring performance.

The magnetic unbalance forces were calculated for maximum force conditions. The reliability engineering studies were concerned with (1) statistical evaluations of critical materials and properties and (2) a Failure Mode, Effect and Criticality Analysis (FME & CA).

809	807	807	807	800°F Coolant										800°F Coolant			
811	805	807	807	807	807	807	807	807	807	807	807	807	807	807	807	807	807
832	818	808	807	807	807	807	807	807	807	807	807	807	807	807	807	807	807
842	822	808	807	807	807	807	807	807	807	807	807	807	807	807	807	807	807
848	827	808	807	807	807	807	807	807	807	807	807	807	807	807	807	807	807
896	883	839	807	807	807	807	807	807	807	807	807	807	807	807	807	807	807
903	902	903	903	903	903	903	903	903	903	903	903	903	903	903	903	903	903
970	969	968	967	976	987	987	987	987	987	987	987	987	987	987	987	987	987
			954	972	984	984	984	984	984	984	984	984	984	984	984	984	984
			882	910	928	941	951	958	964	968	970						
			858	880	894	904	912	917	921	924	925						
			821	827	825	825	827	828	829	830	830						

Figure 2.5-18. Number 1 Design Stator Temperature Distribution (°F) 550 kW Rating, 0.2 Watts/in. ² Slot Conductance with Field Coil Side Cooling Ducts.

Motoring*

The inductor alternator is inherently a good salient pole synchronous motor and variable speed operation can be achieved by variable supply frequency methods. Unfortunately the synchronous motor is not self-starting and the absence of rotor windings or squirrel-cage type conductors precludes the normal induction motor starting methods. The special considerations are discussed below.

Induction Performance

Eddy-current torque: Considering the rotor at standstill with balanced three-phase currents flowing in the armature windings, a rotating field will be produced. The rotating flux will "cut" the surfaces of the rotor poles and will be alternating in the pole bodies and rotor core. The flux cutting the pole surfaces will induce voltages axially across the poles causing current to flow through the rotor material. The greater part of this current will flow in a small portion of the rotor surface (skin effect) and will follow various paths along and around the rotor.

The effect of the induced currents is to produce a magnetic field tending to oppose the magnetic field which originally produced the currents (Lenz' Law) and consequently a force is set up between the rotor material and the rotating armature field. This force produces rotor torque as in a normal induction motor.

The calculation of the induced voltage presents no problem, but the calculation of the electrical resistance of the current paths can

* The motoring performance analyses was originally carried out in Phase I but was later updated in Phase II. The updated analyses is included in Part I of this report because it was not a specific task of Phase II. They are for the 600 kVA design of Phase II.

only be made by approximating the complex current paths to rather simplified patterns. The major current paths are believed to be as shown in Figure 2.6-1. Currents will also flow circumferentially across the pole faces due to the sinusoidal distribution of flux (approximately) across the pole face. Other currents will flow between poles on the same end of the machine. The prediction of the current paths at the ends of the rotor is virtually impossible with any accuracy due to the skin effect, corner effects and stray field effects.

The effective rotor resistance has been calculated for the estimated major current paths -- this resistance varies with frequency (due to skin effect) and pole surface configuration. Associated with this resistance is an effective rotor surface inductance which is dependent on the current paths and surface current magnitude. Due to the irregular "solid-body" rotor configuration, no attempt can be made to approximate this effective inductance with any degree of accuracy and consequently it has to be neglected with probable large calculation error on the eddy-current torque. Eddy-current torque can be estimated from the calculated rotor resistance -- the limit on torque is set by armature tooth saturation with the following estimated output torques for 60 Hz input.

	<u>Torque Out</u>	<u>Phase V</u>	<u>Phase I</u>	<u>Total Pin</u>
Laminated Pole	5.8 ft-lbs.	25.5 V	328 A	11.8 kW
Slotted Pole	51. ft-lbs.	26.8 V	338 A	12.0 kW

The above figures are at standstill and torque decreases rapidly with speed.

Induced current torque: The alternating flux flowing through the poles and rotor core can be utilized to induce voltage in conducting loops attached to the poles as shown in Figure 2.6-2.

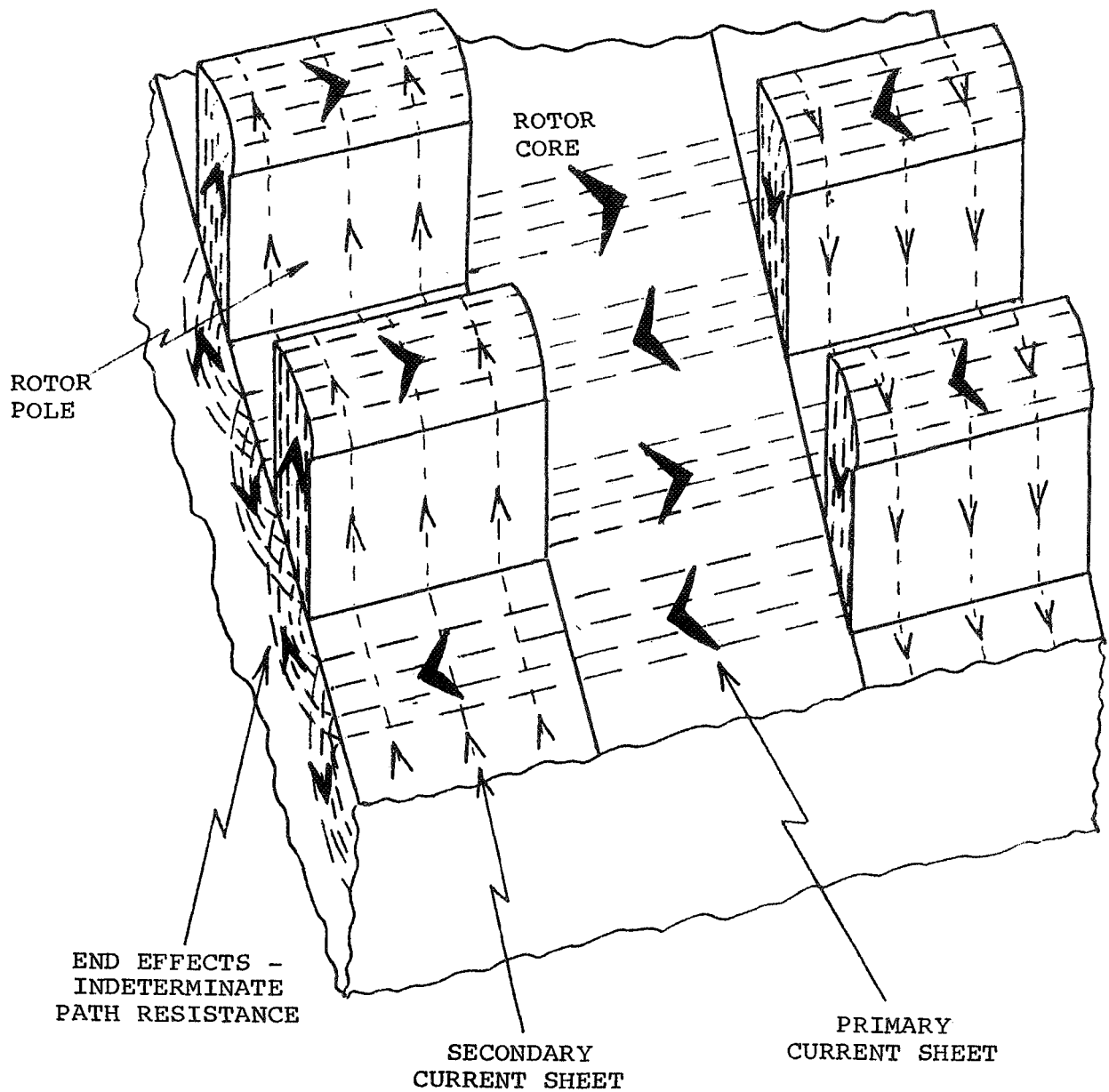


Figure 2.6-1. Planar Surface Model of Rotor Section Showing Estimated Eddy-Current Paths.

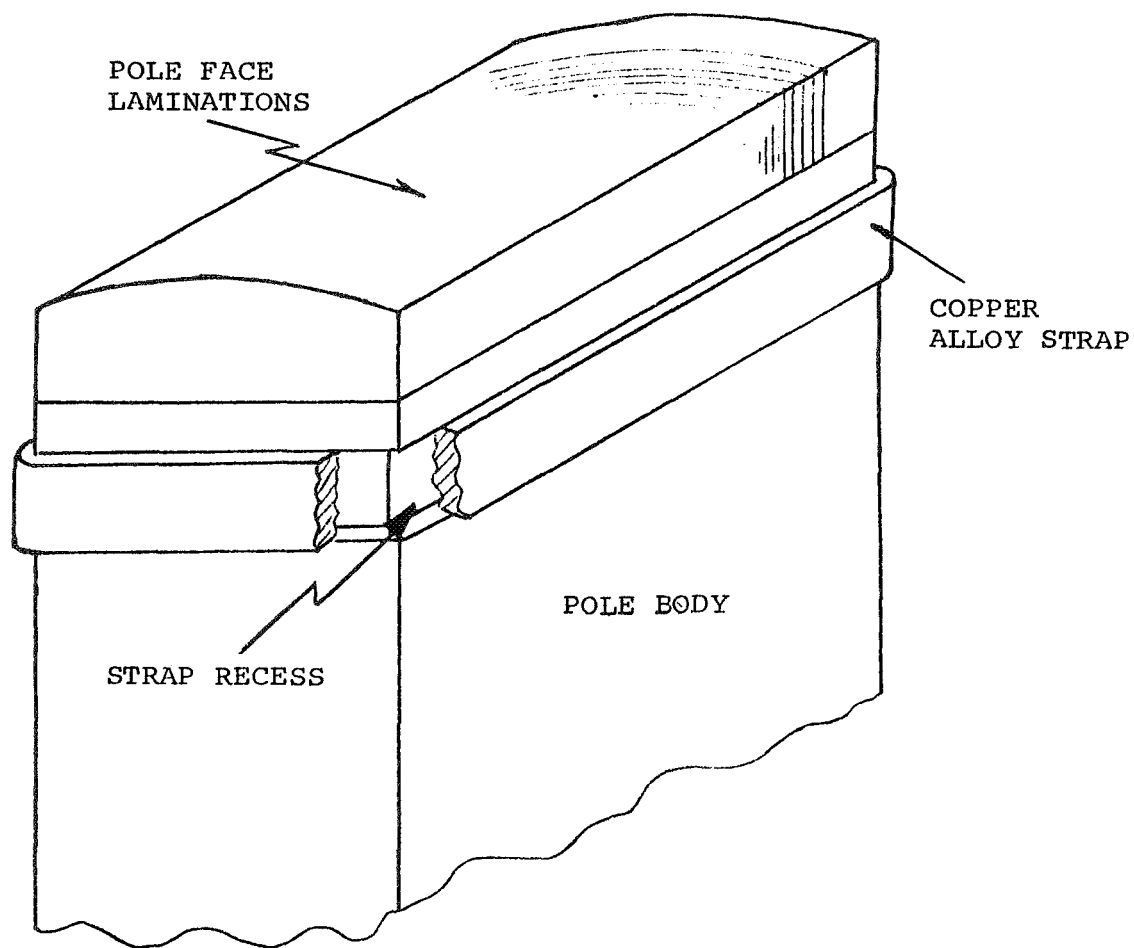


Figure 2.6-2. Construction of Conducting Loops Around the Pole Body.

The calculation of the induced voltage is easily performed and the maximum value is again determined by saturation of the armature teeth. The resistance of the conducting loop can be found from assumed dimensions. The inductance of the loop must be considered by taking the entire rotor/stator/frame flux path and a good estimation can be made by utilizing the no-load saturation curve of the alternator. Typical values of resistance and inductive reactance of such a loop are:

$$R = 1.3 \times 10^{-6} \text{ OHMS} \qquad XL = 9.27 \times 10^{-4} \text{ OHMS}$$

An approximate value of starting torque for such a configuration at 60 Hz is 62.8 ft-lbs. This is in addition to the rotor eddy-current torque. Inspection of the previous values of R and XL reveal that the resistance of the loop is almost negligible compared to its reactance and consequently the loop may be constructed of almost any material (such as H-11 steel). This consideration brings to light the existence of such "loops" as being part of the pole body and rotor core and consequently a contribution to starting torque is to be realized. The calculation of this torque is rendered almost impractical due to the undefined "loop" dimensions and non-linear embraced flux for different "loops" under consideration. It is therefore concluded that the addition of extra conducting loops would have virtually no effect upon the resulting output torque and that the torque contribution of the existing material "loops" would be greater than that calculated for the conducting loops.

Additional damper bars in the pole faces were not analyzed because of their detrimental effects on pole face losses during normal operation as an alternator and because it appeared little, if any, additional gains could be achieved with them.

Hysteresis torque: The alternating flux flowing throughout the rotor material will cause major hysteresis loops to be executed with a corresponding hysteresis loss. The magnitude of this hysteresis loss is a function of material hysteresis loss constant and material volume. The material volume is known accurately but little data is available for hysteresis loss (as distinguished from core loss) for H-11 at 60 Hz. If hysteresis loss is approximately 10 watts/pound at the induction under consideration, approximately 34 ft-lbs of hysteresis torque would be available. This torque is constant for a given induction and is independent of slip speed and applied frequency.

Total resultant torque: The above three torques will act together and the resultant torque will be the sum of these three. The following figures give maximum and minimum estimated torque values, as derived above, for the cases of slotted and laminated pole faces at 60 Hz frequency.

<u>Torque Component</u>	<u>Variation</u>	<u>Estimated Torque (S=1)</u>	
	<u>With Slip and</u> <u>Frequency</u>	<u>Maximum</u>	<u>Minimum</u>
<u>Laminated Pole Faces</u>			
Eddy-Current Torque	$S^{1/2} f^{-3/2}$	6	4
Induced Current Torque	S	200	60
Hysteresis Torque	None	300	30
Totals		506	94
<u>Slotted Pole Faces</u>			
Eddy-Current Torque	$S^{1/2} f^{-3/2}$	60	40
Induced Current Torque	S	200	60
Hysteresis Torque	None	300	30
Totals		560	130

The calculations of the above torques were made independently and the resultant output power of the combined torques may represent such a loading on the armature that excitation to these power levels is impractical, especially with the estimated maximum torques. Due to the effects of saturation and armature loading it is impractical to try to calculate required input power, etc., as the rotor effective resistance is such an indeterminate quantity and calculation errors would render the results meaningless. The only practical methods of calculating motoring torque accurately are to either perform lengthy calculation procedures or to perform practical tests on an actual alternator.

A secondary conclusion reached during this motoring evaluation was that little motoring improvement could be obtained with slotted pole faces over laminated pole faces. As a result, the choice of laminated pole tips over slotted tips is still dependent on pole face losses and cooling configuration design limitations.

Acceleration characteristics: The induction starting capability of the alternator is required to achieve a run up time of approximately one (1) second between 0 and 400 rpm. Based on a friction breakaway and running torque of approximately 14 ft-lbs over this speed range and an inertia of the rotor and turbine of 20.8 lb-in-sec^2 , it is estimated that a torque of 200 ft-lbs at starting (and decreasing with slip as $S^{1/2}$) will be required to achieve one (1) second run-up. This required torque lies between the estimated maximum and minimum induction torques and may or may not be achieved. Based on the minimum estimated torque of 94 ft-lbs (laminated rotor poles), a run-up time to 400 rpm of approximately three (3) seconds is calculated. For the slotted poles, this time would be approximately two (2) seconds. For the maximum

estimated torques of 506 and 560 ft-lbs, run-up times of less than 0.5 second are calculated.

Review of the findings with past motoring performance data indicates the starting torque may be about 60 percent of the rated alternator input torque. This places the probable torque at the low side of the estimates (i.e., near 100 ft-lbs) which means the time to 400 rpm should be closer to the three seconds maximum than the one second minimum. Breakaway torque requirements should be minimized by a strong pulsating (vibration) torque. This will enhance meeting the three seconds time.

The Synchronous Performance. Due to the decreasing relationship of torque with speed and frequency during induction starting, it is desirable to utilize the inherent synchronous motoring performance of the alternator at the lowest possible speed. It is suggested that synchronization be performed when the alternator speed is approximately 400 rpm.

Method of achieving synchronism: As soon as the rotor speed reaches approximately 400 rpm (representing a synchronous frequency of 33 Hz) the supply frequency should be steadily reduced to some value less than 33 Hz, the actual frequency being determined by the rotor speed, which will slowly decrease due to friction torque. As soon as the applied frequency is equal to or less than the synchronous frequency at that rotor speed, synchronization will be automatically effected by the reluctance torque of the machine.

Once synchronization has occurred, reluctance torque will be constant, tending to accelerate the rotor, provided that the "load angle" is non-zero. If excitation is applied to the field coil at this point, generated torque will be produced, also tending to accelerate the rotor.

However, both of these torque components will disappear if the "load angle" becomes zero. In order to produce rotor acceleration, the armature frequency must be increased steadily, whereupon the rotor will accelerate to maintain synchronism.

Synchronous acceleration: During acceleration, armature and field excitation may be varied at will to provide the best combination of torque, current and power factor. The acceleration is purely dependent on the rate of change of armature applied frequency which has a maximum value consistent with maintained synchronism and is such that an acceleration of up to 1200 rpm/sec may be obtained. It is suggested that an acceleration of less than half of this value be obtained to avoid the disastrous effects of loss of synchronism (e.g., severe armature overload currents and bearing shock). Power supply requirements during synchronous motoring depend on the required run-up time. In order to maintain rated armature current, the voltage/frequency ratio must be almost constant, as shown in Figure 2.6-3 -- consequently the power requirement for this condition will increase steadily to a maximum at full speed. If a limited power supply is to be used, the rate of change of frequency must be reduced to compensate for the reduction in maximum torque to avoid loss of synchronism. The power supply should be capable of programmed frequency and voltage variation with very close frequency regulation.

If 400 rpm is reached within 2 seconds, synchronization is affected within 1 more second, and the alternator is accelerated at 1200 rpm per second, the times to reach 40 percent design speed, rated speed and 120 percent speed are approximately 9, 19, and 21 seconds respectively. If 600 rpm per second synchronous acceleration is used, the times increase to approximately 15, 35, and 40 seconds.

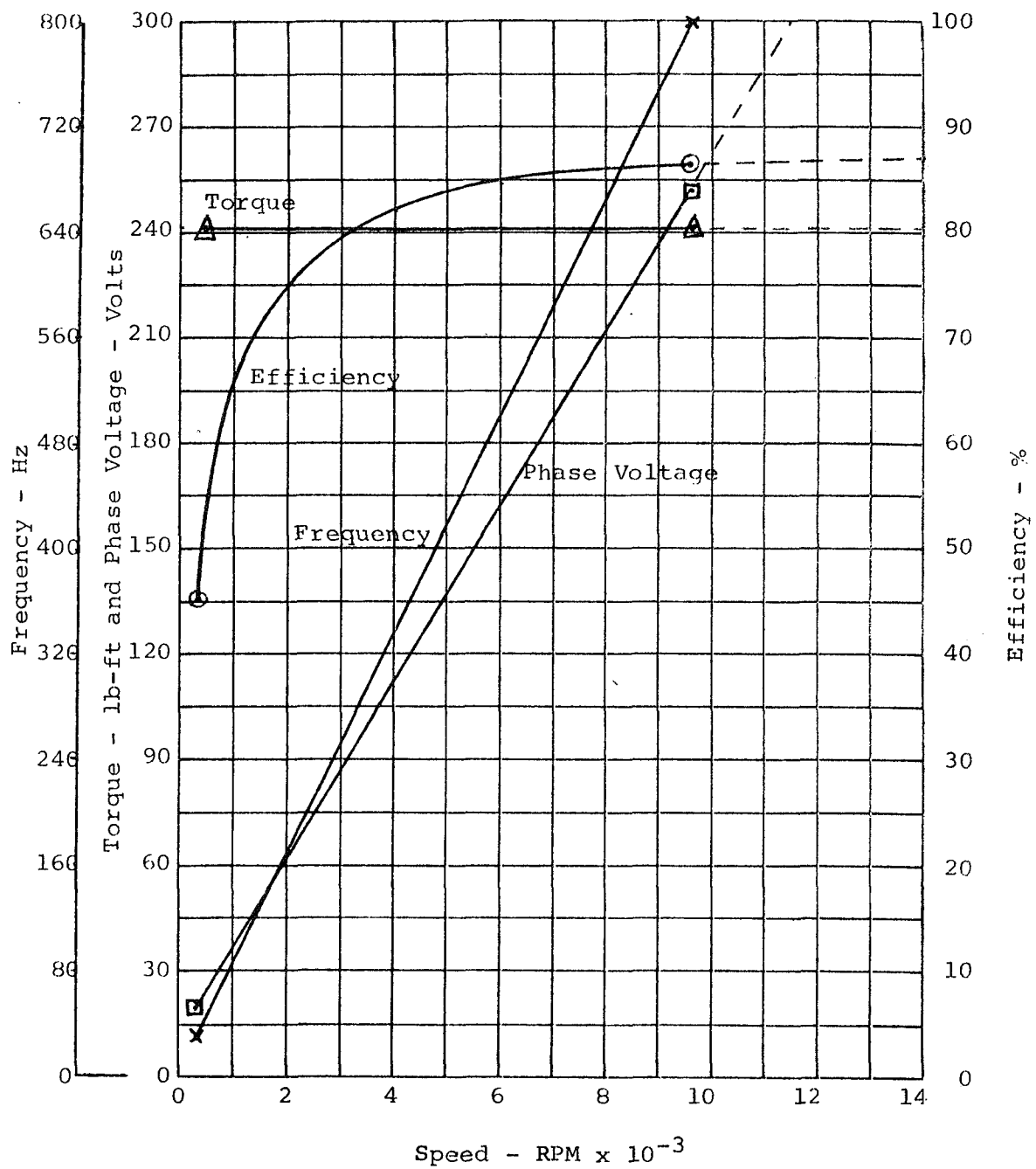


Figure 2.6-3. Synchronous Motoring Performance Unity P.F.

Magnetic Unbalance Forces

In an electromagnetic generator, a magnetic radial force can result if the rotating member is made eccentric to the stator. The magnitude of the radial force is dependent upon the degree of eccentricity between the rotor and stator and the level of field excitation. The magnetic force on the rotor will be present regardless of the conditions of gravity.

When a magnetic field "B" exists in the space between two magnetized forces, there exists a force equal to:

$$F = \frac{B^2 A}{72.13} \text{ lbs.} \quad (1)$$

B = flux density in kilolines per square inch

A = cross sectional area of the gap in square inches

The amount of energy per unit volume (i.e., energy density) in a given space is identical with pressure or force per unit surface area when the surface is taken at right angles to the direction in which the pressure can act to move a substance. Whenever there is an immediately available supply of energy, a mechanical force is in evidence. The energy stored in a magnetic field is

$$E = \frac{(B)^2}{72.13} \quad (r) \quad \frac{\text{inch-lb}}{(\text{inch})^3} \quad (2)$$

where

r = relative permeability.

For a field density of 8.49 KL/inch² in space where r = 1, there exists an energy density of 1.0 inch-lb/(inch)³ or 1.0 lbs/(inch)². Hence, the pressure tending to move the two bodies together will be equal to 1.0 lbs for each square inch of cross section of the gap.

Therefore, the force equals the energy density "E" times the cross sectional area "A". In air or free space where the relative permeability "r" is unity, the product of "E" times "A" results in Equation (1) above.

Past studies (2.4-13) have shown that the magnetic unbalance force at zero speed, no load excitation* is both a maximum and a linear function of the eccentricity between the rotor and stator. Because of this, the unbalance forces were calculated for 0.010 inches eccentricity at the maximum force conditions and the results were then fed into the bearing design considerations. This simplified approach without feedback from the bearing designs or minimization of forces is possible only because the forces calculated for this worst case are still too small to influence the bearing design. As a result, only the absolute maximum values are of interest and the actual operating values are of academic value only.

When the rotor is rotated, certain events take place which act to reduce the magnetic force unbalance on the rotor. Assume a given flux density exists at a fixed point on the face of a bottom pole. When the rotor is turned so that the reference pole takes the position of the upper pole, the flux density at the reference point is reduced due to the increased reluctance of the magnetic path. Further rotation of the rotor will return the reference point to its initial position and the flux density to its original values. Under rotation, the reference point on the pole face sees a periodically varying flux density. The variation generates a voltage in the rotor poles and therefore electric currents will flow according to the impedance of the conductance paths.

* Or thereabouts depending on the saturation characteristics of the design.

The MMF of these eddy-currents act to oppose the alternating magnetic field which caused them. Since the alternating field represents the difference in field magnitude responsible for the magnetic force unbalance on the rotor, the currents act to reduce the force. The flow of the currents will, of course, create additional rotor losses. If the armature winding has several parallel paths, the difference in the magnitude of the magnetic field will cause currents to circulate in the windings. These currents also act to reduce the magnetic force unbalance.

The impedance of the electric circuits in which these currents flow is composed of both resistance and inductive reactance. Inductive reactance is directly proportional to frequency and therefore at low rotor speeds, the magnitude of the opposing currents will be determined mostly by the resistance. At higher rotor speeds, the inductive reactance increases and the magnitude of the currents will depend increasingly on the value of this reactance. The induced voltage, like the inductive reactance, is directly proportional to rotor speed. The ratio of induced voltage to inductive reactance represents reactive current. As rotor speed is increased, the force opposing currents approach this ratio which is a constant. The rotor speed at which the opposing currents closely approximate the limiting value is dependent upon the ratio of resistance to inductance of the electric circuits.

The forces for the three base designs were determined for static conditions using the analysis techniques presented in Reference 2.4-13. The maximum forces were found to occur at about 70 percent of rated excitation and are given in Table 2.6-1. Since eccentricities will only be of the order of one to two mils, the maximum expected unbalance forces should only be thirty to sixty pounds total.

TABLE 2.6-1

MAGNETIC UNBALANCE FORCES

<u>DESIGN</u>	<u>MAXIMUM FORCE</u>
No. 1	32.3 lbs/in x 10 ⁻³
No. 2	26.7 lbs/in x 10 ⁻³
No. 3	27.6 lbs/in x 10 ⁻³

Reliability Engineering

The reliability engineering studies were concerned with two basic areas, (1) statistical evaluations of critical materials/properties to relate to reliability predictions on the alternator design and (2) a Failure Mode, Effect and Criticality Analysis (FME & CA). The FME & CA carried out was preliminary in nature and concerned itself with a typical design from Phase I. The final FME & CA was carried out on the Phase II design and is reported in Part II of this report.

The statistical studies concerned themselves with H-11 rotor material only. It was originally hoped that all candidate materials could be included in the studies, but the lack of useful data or the relative need compared to cost of analyzing them precluded all but the H-11 steel. This conclusion was reached as a result of (1) itemizing candidate materials, probable temperatures and properties to be analyzed, (2) collecting all possibly useful data on each of the materials and (3) reviewing the lists and data with experts - both mathematical and materials - who would carry out the proposed study. Lack of useful data required the majority of the materials to be dropped from further statistical considerations. The expense involved in analyzing the remainder relative to the need-to-know precluded all but the H-11 steel.

The statistical analysis of the H-11 steel is discussed in the next subsection. It is interesting to note that even for the H-11 steel, analysis of the creep strengths were not carried out. This is because of a lack of useful, consistent data with which to do a meaningful study.

Statistical Analysis of Tensile-Test Data on H-11 Steel Bars. The purpose of the statistical study was to analyze tensile-test data to determine lower limits for three tensile properties which appear in design calculations. For each property, the corresponding lower limit is supposed to be a value which one can safely assume will nearly always be exceeded. The statistical formulation of this problem is to find a lower limit such that there is a confidence γ that a proportion P (or more) of the population exceeds the limit. The "safely" of the previous sentence corresponds to γ and the "nearly always" to P . A typical choice of values would be $\gamma = 0.95$ and $P = 0.99$. The resulting limit is called a tolerance limit.

Tensile specimens from three heats of steel were previously tested (2.2-7) and for each test data set, the properties of concern were:

- a. modulus of elasticity (psi),
- b. ultimate of tensile strength (psi),
- c. offset yield strength (psi).

In addition, the specimens were heat treated to various levels of hardness and tested at various temperatures. The test temperatures represent possible temperatures of the expected environment for the device; the various hardness values represent the expected hardness gradient through the device. The purpose of the analysis was to provide lower tolerance limits which are specific to various values of temperature and hardness.

Table 2.6-2 is a list of the data on which the analysis was based. Each heat was analyzed separately. This consisted of fitting an equation involving hardness and temperature to each tensile property.

TABLE 2.6-2

H-11 PROPERTY DATA

Mod. Of Elas. (10 ⁶ psi)	Ult. Tens. Strength (psi)	Offset Yld. Strength (psi)	Hardness Rockwell "C"	Test Temp. (°F)	Ref. Spec. No.
Heat H-10283-K2 ⁽⁶⁾					
26.10	201,195	151,394	48	800	23T
25.79	191,446	159,878	48	800	24T
26.97	158,475	121,364	44.5	900	25T
27.55	163,306	118,448	44.5	900	26T
24.95	177,645	134,731	47.5	900	27T
28.66	178,536	140,421	47.5	900	28T
20.74	167,677	116,162	48.0	1000	31T
20.30	173,653	117,764	48.5	1000	32T
26.53	210,632	164,493	50	800	33T
26.89	207,753	164,016	50.5	800	34T
25.19	196,819	146,123	51	900	35T
25.36	208,583	154,691	51	900	36T
24.39	170,974	131,730	51	1000	37T
22.41	174,950	136,680	51	1000	38T
Heat D-14-K2 ⁽⁶⁾					
27.00	168,706	140,421	46	800	9X
26.94	170,500	137,410	46	800	10X
26.70	198,603	160,179	48	800	11X
27.97	203,610	164,995	48	800	12X
26.67	152,457	117,352	44	900	13X
27.58	170,061	132,382	44.5	900	14X
28.10	158,065	123,656	46	900	15X
25.51	188,064	147,944	46	900	16X
28/82	205,588	167,165	48.5	900	17X
27.21	182,547	146,940	48	900	18X
25.08	128,385	92,778	43.5	1000	19X
23.91	137,613	102,862	44	1000	20X
23.09	134,403	99,298	46	1000	21X
21.71	128,048	95,528	46	1000	22X
24.81	168,347	129,032	48	1000	23X
23.37	136,089	102,823	48	1000	24X
26.26	154,645	118,181	44	900	25X
28.28	163,672	127,246	44.5	800	7X
25.44	165,344	122,100	44.5	800	8X

TABLE 2.6-2 (Continued)

Mod. Of Elas. (10 ⁶ psi)	Ult. Tens. Strength (psi)	Offset Yld. Stregnth (psi)	Hardness Rockwell "C"	Test. Temp. (°F)	Ref. Spec. No.
Heat H-9228-K2 ⁽⁶⁾					
27.08	177,419	138,105	44.5	800	16Z
27.68	177,533	138,415	44.5	800	17Z
29.07	215,725	174,395	48	800	18Z
28.27	194,584	155,466	47.5	800	19Z
27.26	165,497	125,376	44.5	900	20Z
29.01	163,672	124,750	44.5	900	21Z
26.91	175,025	139,919	48	900	22Z
25.79	188,565	144,433	48	900	23Z
23.44	141,129	100,806	44	1000	24Z
23.71	140,421	104,313	44.5	1000	25Z
25.83	142,137	103,333	48	1000	26Z
24.51	142,427	107,823	48	1000	27Z

The "best" equation for each property was found by first fitting a full second degree polynomial in two variables (temperature and hardness) and then seeing if any terms could be omitted without significantly diminishing the predicting ability of the equation. The fitting referred to previously was carried out by least squares. To illustrate the equations that were used let Y stand for any tensile property, let H stand for hardness and let T stand for temperature. Then the full second degree polynomial is given by:

$$Y = \beta_0 + \beta_H H + \beta_T T + \beta_{HH} H^2 + \beta_{TT} T^2 + \beta_{HT} HT,$$

where the β 's are determined from the given (Y, H, T)'s using the principle of least squares. The estimated coefficients are given in Table 2.6-3. The equations represented by data of Table 2.6-3 are plotted on Figures 2.6-4 through 2.6-6. These plots cover the values of temperature and hardness that are of interest. Since the data for heat H-10283-K2 include only one point in the region of interest, this heat was not plotted and was not considered further. The plots are almost self-explanatory but a few words of explanations are appropriate. Refer to the plot of modulus of elasticity for heat D-14-K2, Figure 2.6-4. The ordinate is marked from 2.20 to 3.00. At the top of the plot the notation at 7 means that the 2.20 is 2.20×10^7 . A similar notation is found on the abscissa so that the markings are actually 800, 850, 900, etc. The graph itself is "drawn" using the letter "C" and a careful examination will reveal some lack of smoothness. This is not an intentional feature of the graph but a result of the somewhat poor resolution of a computer printer when used as a plotter. Next consider the plot of ultimate tensile strength for heat D-14-K2, Figure 2.6-5. Here two independent variables are being represented and the

TABLE 2.6-3

STATISTICAL COEFFICIENTS

Heat:	H-10283-K2	D-14-K2	H-9228-K2
Coeff:	Modulus of Elasticity (psi)		
β_O	-140,435,834	-103,847,757	-40,865,000
β_T	392,687.5	308,104.8	169,612.5
β_{TT}	-230.29	-180.60	-104.38
	Ultimate Tensile Strength (psi)		
β_O	-1,171,588	-1,302,844	-1,606,739
β_H	30,624.3	7,808.0	43,344.3
β_T	1,029.2	2,672.3	1,726.7
β_{TT}	---	- 1.59	---
β_{HT}	-27.90	---	-42.83
	Offset Yield Strength (psi)		
β_O	630,169	-1,247,146	-1,481,860
β_H	-6,525.1	7,673.2	39,572.7
β_T	-793.2	2,474.3	1,543.7
β_{TT}	---	-1.48	---
β_{HT}	12.35	---	-38.62

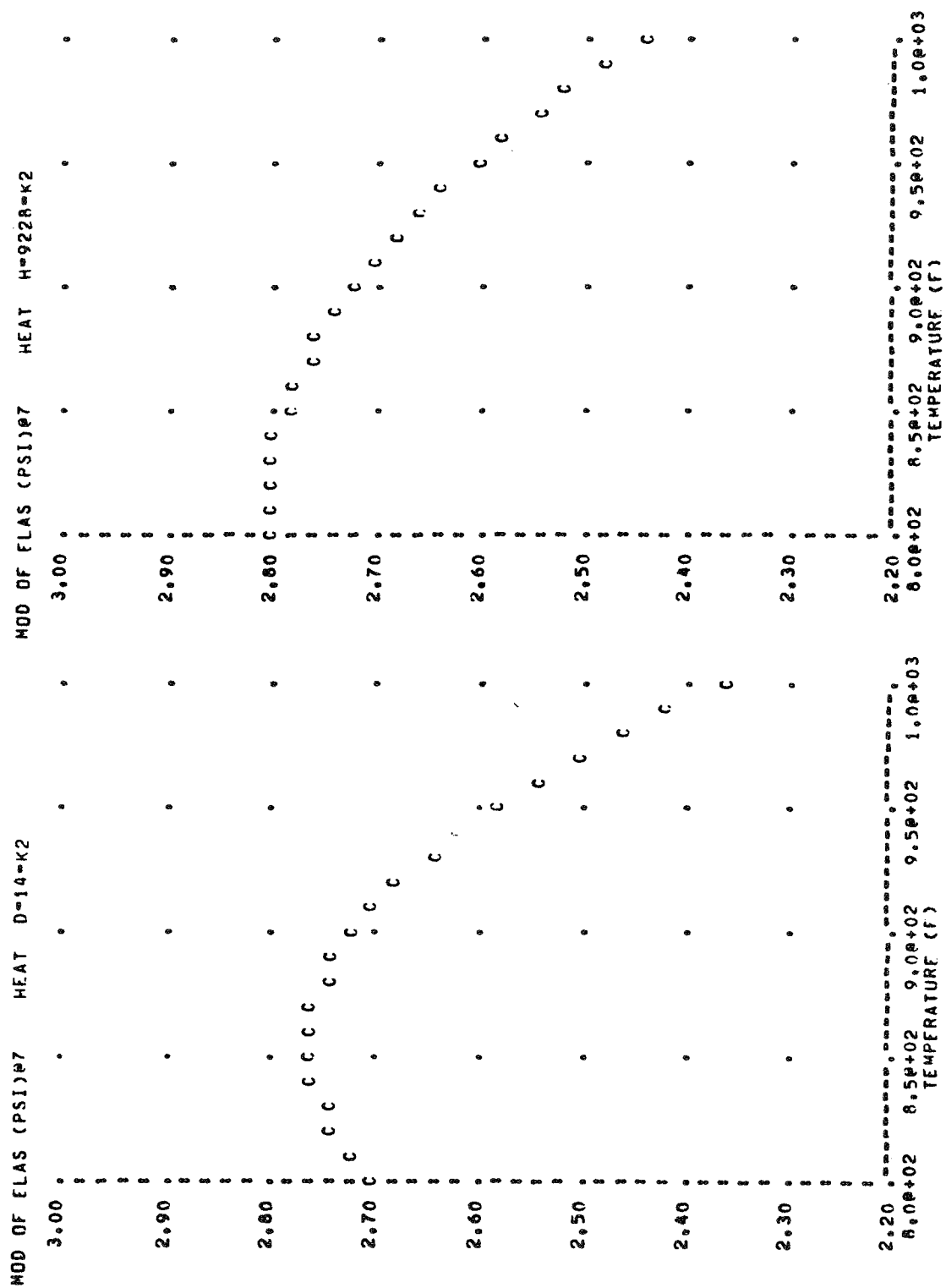
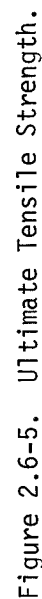


Figure 2.6-4. Modulus of Elasticity.



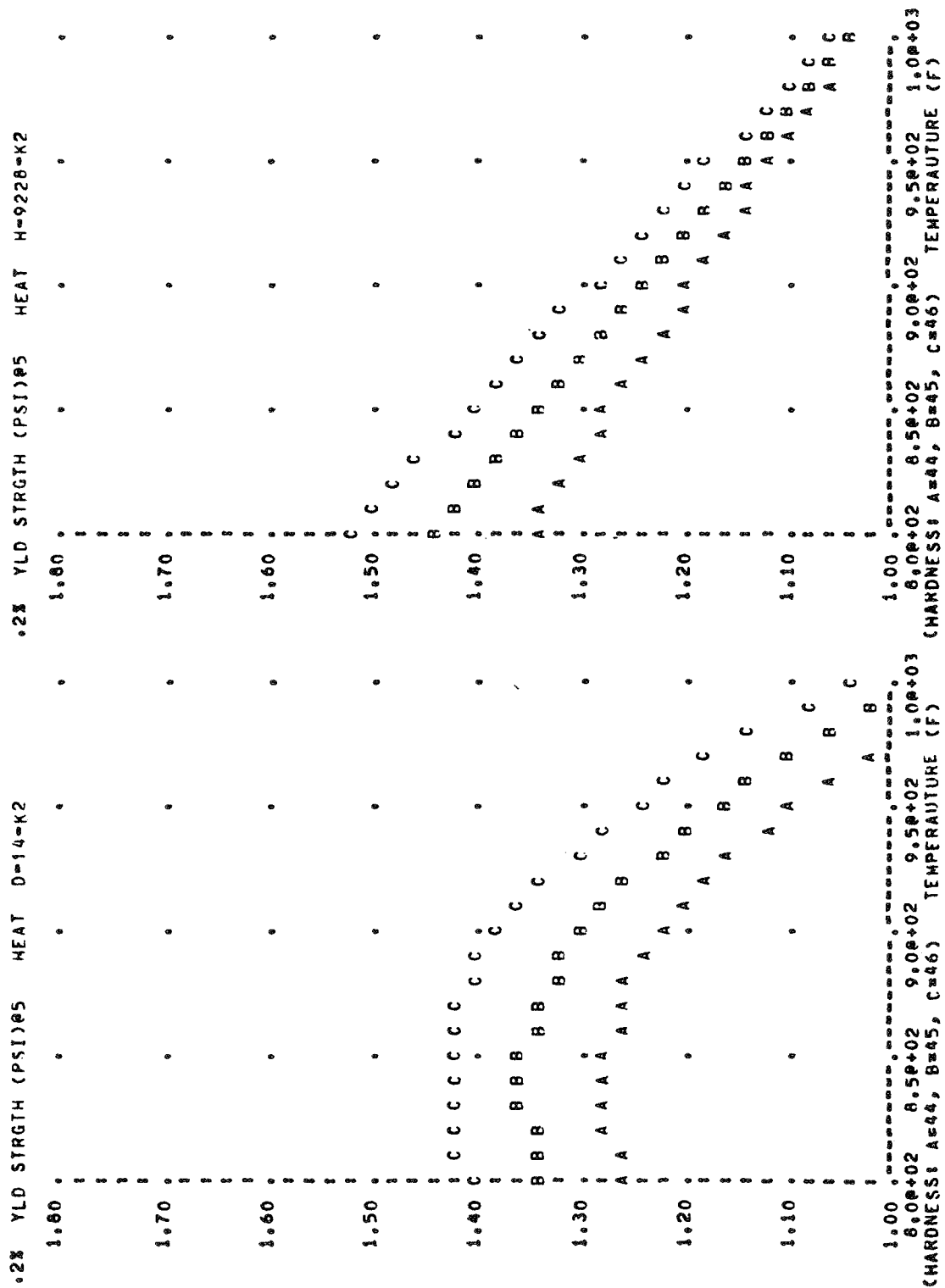


Figure 2.6-6. 0.2% Yield Strength.

extra one, hardness, is held constant at three different values as explained in the note along the abscissa. Thus the "A" graph shows the relation between tensile strength and temperature when hardness is 44. Finally, to see the resolution problem at its worst consider the plot of ultimate tensile strength for heat H-9228-K2, Figure 2.6-5. The "A" graph is representing a straight line sloping from about 175,000 down to about 142,500. Similarity is evident for the "B" and "C" graphs.

The final step in the analysis was to compute lower tolerance limits for the various tensile properties at selected values of hardness and temperature. The results of this work are presented in Tables 2.6-4 through 2.6-6. A description of the computation of the tolerance limits follows.

Suppose a linear regression analysis has been carried out (linear in the parameters, not necessarily in the independent variables) and we wish to find a lower P (proportion) tolerance limit with confidence γ for the true values η_0 of the dependent variable at a specified point $(X_{10}, X_{20}, \dots, X_{k0})$ in the independent variables. The regression analysis will provide us with an estimate of η_0 , say Y_0 , and an estimate of the underlying standard deviation, say s . The general form of the lower tolerance limit is given by

$$L = Y_0 - s(K_p^2 + [K_p^2 - ab]^{1/2})/a$$

where

$$a = 1 - K_\gamma^2/(2n),$$

$$b = K_p^2 - K_\gamma^2/N^*,$$

n = degrees of freedom for s ,

N^* = the effective number of observations,

and K_ϵ = the value of a standard normal variable which is exceeded with probability $1-\epsilon$.

The effective number of observations, N^* , is defined by the property that the standard deviation of Y_0 is given by $s/\sqrt{N^*}$. If C denotes the inverse of the moment matrix used in the regression analysis and X_0 denotes the column vector with components $X_{10}, X_{20}, \dots, X_{k0}$ then N^* is given by

$$N^* = 1/X_0' C X_0.$$

The preceding is taken from Wallis, W.A., (1951) "Tolerance intervals for linear regression", Second Berkely Symp. Math. Statist, Prob. 43-41.

In the Tables 2.6-4 through 2.6-6, lower tolerance limits are presented for various values of hardness, temperature, P , and γ as indicated. All three tensile properties are represented for each of heats D-14-K2 and H-9228-K2.

The impact that the tolerance limits have on the present design is one of adding confidence to the screening and design selection processes. In general, it was found that at the nominal 45 R_c hardness and at 800^oF, one could attach at least a 95 percent confidence level to 99 percent (proportion) of the material having strengths above the minimum yield strengths being used (72 percent of nominal 0.2 percent offset yield strength). It is not until 65 percent of the nominal yield strength is reached that a 99 percent proportion at 99 percent confidence level can be attached to the material samples. For the ultimate strength criteria (87 percent of ultimate at 20 percent overspeed), it was found that for the same hardness, temperature and confidence level, there may only be 90 percent of the material above the strengths being used in the design. This indicates that the designs are less reliable (stress-wise) than originally believed in the 20 percent overspeed condition.

TABLE 2.6-4

MODULUS OF ELASTICITY TOLERANCE LIMITS

Heat D-14-K2		MODULUS OF ELASTICITY (psi)		
TEMPERATURE = 800				
PROPORTION		.90	.95	.99
CONFIDENCE				
0.90		24837458	24353007	23423164
0.95		24564781	24046226	23045531
0.99		23944308	2336511	22153135
TEMPERATURE = 900				
PROPORTION		.90	.95	.99
CONFIDENCE				
0.90		24985368	24498023	23563022
0.95		24724283	24201889	23194379
0.55		24127151	23513371	22319589
TEMPERATURE = 1000				
PROPORTION		.90	.95	.99
CONFIDENCE				
0.90		21444135	20959673	20029831
0.95		21171448	20652893	19652198
0.99		20550975	19943177	18759801

H-11 Steel, Statistical Modulus of Elasticity, psi

Heat D-14-K2, Reference AFAPL-TR-66-127

Hardness = 44 = 46 R_C, Temperature = °F

TABLE 2.6-4 (Continued)

Heat H-9928-K2		MODULUS OF ELASTICITY (psi)		
TEMPERATURE = 800				
CONFIDENCE	PROPORTION	.90	.95	.99
	0.90	25590549	25071388	24069318
	0.95	25183344	24602095	23473398
	0.99	24109138	23336202	21822684
TEMPERATURE = 900				
CONFIDENCE	PROPORTION	.90	.95	.99
	0.90	24808049	24288888	23286818
	0.95	24400844	23819595	22690898
	0.99	23326638	22553702	21040184
TEMPERATURE = 1000				
CONFIDENCE	PROPORTION	.90	.95	.99
	0.90	21938049	21418888	20416818
	0.95	21530844	20949595	19820898
	0.99	20456638	19683702	18170184

H-11 Steel, Statistical Modulus of Elasticity, psi

Heat H-9228-K2, Reference AFAPL-TR-66-127

Hardness = 44 - 46 R_C, Temperature = °F

TABLE 2.6-5

ULTIMATE TENSILE STRENGTH TOLERANCE LIMITS

Heat D-14-K2 Ultimate Tensile Strength (psi)

Hardness = 44			Temperature = 800			Temperature = 1000		
CONFIDENCE	PROPORTION		.90	.95	.99	CONFIDENCE	PROPORTION	
0.90			137386	132480	123054	0.90	100021	95099
0.95			134112	128860	118712	0.95	96830	91558
0.99			126673	120496	108442	0.99	89560	83351
Hardness = 45			Temperature = 800			Temperature = 1000		
CONFIDENCE	PROPORTION		.90	.95	.99	CONFIDENCE	PROPORTION	
0.90			146126	141163	131631	0.90	108643	103669
0.95			143134	137807	127517	0.95	105698	100357
0.99			136270	129973	117698	0.99	98929	92610
Hardness = 46			Temperature = 800			Temperature = 1000		
CONFIDENCE	PROPORTION					CONFIDENCE	PROPORTION	
0.90			154341	149349	139764	0.90	116707	111716
0.95			151472	146106	135747	0.95	113841	108475
0.99			144856	138499	126117	0.99	107229	100871
Hardness = 44			Temperature = 900					
CONFIDENCE	PROPORTION		.90	.95	.99			
0.90			135134	130188	120687			
0.95			132065	126761	116511			
0.99			125044	118783	108571			
Hardness = 45			Temperature = 900					
CONFIDENCE	PROPORTION		.90	.95	.99			
0.90			143765	138761	129155			
0.95			140944	135563	125174			
0.99			134426	128044	115618			
Hardness = 46			Temperature = 900					
CONFIDENCE	PROPORTION		.90	.95	.99			
0.90			151806	146784	137145			
0.95			149055	143649	133218			
0.99			142679	136258	123766			

H-11 Steel, Statistical Ultimate Strength, psi
Heat D-14-K2, Reference AFAPL-TR-66-127
Hardness = R_C, Temperature = °F

TABLE 2.6-5 (Continued)

Heat H-9228-K2 Ultimate Tensile Strength (psi)					Heat H-9228-K2, Reference AFAPL-TR-66-127				
Hardness = 44					Hardness = 44				
Temperature = 800					Temperature = 1000				
CONFIDENCE	PROPORTION	.90	.95	.99	CONFIDENCE	PROPORTION	.90	.95	.99
0.90		157629	154615	148783	0.90		126436	123401	117527
0.95		154424	151035	144426	0.95		123357	119938	113273
0.99		145872	141241	132106	0.99		115087	110403	101169
Hardness = 45					Hardness = 45				
Temperature = 800					Temperature = 1000				
CONFIDENCE	PROPORTION	.90	.95	.99	CONFIDENCE	PROPORTION	.90	.95	.99
0.90		168193	165086	159078	0.90		128182	125064	119036
0.95		165468	161948	155098	0.95		125500	121965	115089
0.99		157978	153113	143565	0.99		118107	113215	103623
Hardness = 46					Hardness = 46				
Temperature = 800					Temperature = 1000				
CONFIDENCE	PROPORTION	.90	.95	.99	CONFIDENCE	PROPORTION	.90	.95	.99
0.90		177971	174803	168691	0.90		129258	126089	119977
0.95		175468	171864	164874	0.95		126756	123151	116160
0.99		168461	163450	153671	0.99		119750	114738	104218
Hardness = 44					H-11 Steel, Statistical Ultimate Strength, psi				
Temperature = 900					Heat H-9228-K2, Reference AFAPL-TR-66-127				
CONFIDENCE	PROPORTION	.90	.95	.99	Hardness = R _c , Temperature = °F				
0.90		144013	140846	134736					
0.95		141506	137904	130916					
0.99		134491	129483	119708					
Hardness = 45									
Temperature = 900									
CONFIDENCE	PROPORTION	.90	.95	.99					
0.90		149564	146312	140072					
0.95		147293	143575	136414					
0.99		140778	135579	125528					
Hardness = 46									
Temperature = 900									
CONFIDENCE	PROPORTION	.90	.95	.99					
0.90		154701	151403	145097					
0.95		152536	148755	141507					
0.99		146238	140937	130753					

TABLE 2.6-6

YIELD STRENGTH TOLERANCE LIMITS

Heat D-14-K2				Offset Yield Strength (psi)			
Hardness = 44		Temperature = 800		Temperature = 800		Temperature = 1000	
CONFIDENCE	PROPORTION	CONFIDENCE	PROPORTION	CONFIDENCE	PROPORTION	CONFIDENCE	PROPORTION
0.90	105079	0.90	100936	0.90	92976	0.90	68891
0.95	102314	0.95	97879	0.95	89309	0.95	66196
0.99	96032	0.99	90816	0.99	80636	0.99	60057
Hardness = 45		Temperature = 800		Temperature = 800		Temperature = 1000	
CONFIDENCE	PROPORTION	CONFIDENCE	PROPORTION	CONFIDENCE	PROPORTION	CONFIDENCE	PROPORTION
0.90	113539	0.90	109348	0.90	101298	0.90	77251
0.95	111013	0.95	106514	0.95	97824	0.95	74765
0.99	105216	0.99	99899	0.99	89532	0.99	69048
Hardness = 46		Temperature = 800		Temperature = 800		Temperature = 1000	
CONFIDENCE	PROPORTION	CONFIDENCE	PROPORTION	CONFIDENCE	PROPORTION	CONFIDENCE	PROPORTION
0.90	121556	0.90	117341	0.90	109246	0.90	85141
0.95	119133	0.95	114602	0.95	105854	0.95	82720
0.99	113546	0.99	108178	0.99	97721	0.99	77137
Hardness = 44		Temperature = 900		Temperature = 900		Temperature = 1000	
CONFIDENCE	PROPORTION	CONFIDENCE	PROPORTION	CONFIDENCE	PROPORTION	CONFIDENCE	PROPORTION
0.90	102198	0.90	98021	0.90	89997	0.90	80926
0.95	99606	0.95	95127	0.95	86471	0.95	78189
0.99	93677	0.99	88389	0.99	78077	0.99	71768
Hardness = 45		Temperature = 900		Temperature = 900		Temperature = 1000	
CONFIDENCE	PROPORTION	CONFIDENCE	PROPORTION	CONFIDENCE	PROPORTION	CONFIDENCE	PROPORTION
0.90	110566	0.90	106340	0.90	98228	0.90	80926
0.95	108184	0.95	103639	0.95	94867	0.95	78189
0.99	102680	0.99	97290	0.99	86796	0.99	71768
Hardness = 46		Temperature = 900		Temperature = 900		Temperature = 1000	
CONFIDENCE	PROPORTION	CONFIDENCE	PROPORTION	CONFIDENCE	PROPORTION	CONFIDENCE	PROPORTION
0.90	118436	0.90	114195	0.90	106055	0.90	80926
0.95	116113	0.95	111548	0.95	102739	0.95	78189
0.99	110728	0.99	105306	0.99	94757	0.99	71768

H-11 Steel, Statistical 0.2% Offset Yield Strength, psi
Heat D-14-K2, Reference AFAPL-TR-66-127
Hardness = R_c , Temperature = $^{\circ}F$

TABLE 2.6-6 (Continued)

Heat H-92228-K2 Offset Yield Strength (psi)									
Hardness = 44		Temperature = 800		PROPORTION	Hardness = 44		Temperature = 1000		PROPORTION
CONFIDENCE	0.90	0.95	0.99		CONFIDENCE	0.90	0.95	0.99	
0.90	122020	119663	115102		0.90	91212	88839	84245	
0.95	119514	116863	111896		0.95	88804	86131	80919	
0.99	112826	109205	102061		0.99	82337	78674	71453	
Hardness = 45		Temperature = 800		PROPORTION	Hardness = 45		Temperature = 1000		PROPORTION
CONFIDENCE	0.90	0.95	0.99		CONFIDENCE	0.90	0.95	0.99	
0.90	131859	129429	124731		0.90	93130	90692	85978	
0.95	129728	126975	121619		0.95	91033	88269	82891	
0.99	123871	120066	112599		0.99	85251	81426	73925	
Hardness = 46		Temperature = 800		PROPORTION	Hardness = 46		Temperature = 1000		PROPORTION
CONFIDENCE	0.90	0.95	0.99		CONFIDENCE	0.90	0.95	0.99	
0.90	141084	138606	133826		0.90	94525	92047	87267	
0.95	139126	136308	130841		0.95	92568	59749	84282	
0.99	133646	129728	122080		0.99	87089	83170	75522	
Hardness = 44		Temperature = 900		PROPORTION	Hardness = 44		Temperature = 900		PROPORTION
CONFIDENCE	0.90	0.95	0.99		CONFIDENCE	0.90	0.95	0.99	
0.90	108165	105688	100910		0.90	100910	97923	89158	
0.95	106205	103387	97923		0.95	97923	94775	89158	
0.99	100719	96802	89158		0.99	94775	89158	89158	
Hardness = 45		Temperature = 900		PROPORTION	Hardness = 45		Temperature = 900		PROPORTION
CONFIDENCE	0.90	0.95	0.99		CONFIDENCE	0.90	0.95	0.99	
0.90	113571	111028	106148		0.90	106148	103287	94775	
0.95	111795	108888	103287		0.95	103287	94775	89158	
0.99	106700	102634	94775		0.99	94775	89158	89158	
Hardness = 46		Temperature = 900		PROPORTION	Hardness = 46		Temperature = 900		PROPORTION
CONFIDENCE	0.90	0.95	0.99		CONFIDENCE	0.90	0.95	0.99	
0.90	118654	116074	111143		0.90	111143	108335	99926	
0.95	116961	114004	108335		0.95	108335	99926	99926	
0.99	112035	107890	99926		0.99	99926	99926	99926	

To reach the 99 percent proportion at a 99 percent confidence level, either the overspeed must be reduced (to about 10 percent) or the allowable 87 percent of ultimate being used must be reduced to 81 percent of ultimate.

Because only three heats were used, some penalty is paid in the small statistical sample. For instance, the above conclusions were drawn from results of the heat having the lowest stresses. With a larger number of heats, the confidence levels, etc., could possibly improve. Since the actual design stresses are generally lower than the above material stress criteria limits, confidence levels may be sufficiently high even for the small statistical sample used. Because of this, definition of any development test program required to prove the comparability of the design stresses and material strengths cannot be determined until the details of the final design are completely known. This is because the application of the foregoing type of result is very much dictated by the details of a particular situation. In the present instance, only a very few alternators will ultimately be manufactured and not much material is involved. Thus, when it comes time to build the alternator, it may be feasible to try several heats of steel and select one for use. This means, for example, that the worst one of the two heats investigated here could be used to obtain design values with the idea that later one can, by selection, obtain a heat at least as good. The point being raised is that we have only dealt with within-heat variability and our situation is such that we can do this. In other applications, however, it would be essential to take account of heat to heat variability. In such cases many more sample heats would be needed in the analysis.

Preliminary Fracture Toughness Considerations of H-11 Steel for Rotor Application. Although not a specific part of the reliability engineering studies, the fracture toughness in the presence of flaws of a generator rotor made from H-11 steel was considered and is reported here. The data is reported herein because of its relative value in illustrating the inherent reliability of a rotor made of H-11 steel. The analysis makes use of existing fracture toughness data. Recommendations for additional testing are also made.

The rotor material considered is, of course, H-11 steel having a room temperature yield strength of 200 ksi. Operating temperature range of the rotor is between 0°F and 800°F. Due to the thickness of the rotor, plane strain conditions prevail throughout most of the rotor. At present K_{1C} fracture toughness values of H-11 steel under the above conditions have not been generated. Some data for H-11 steel has been presented in the literature. These results are for a higher strength level and a more restricted temperature range than presently of interest. The existing data was examined and some first order approximations of the short time K_{1C} 's for the strength level and temperatures of interest were made. The term short time K_{1C} value refers to values obtained by rapidly loading the test specimen to the fracture load. The K_{1C} or fracture toughness obtained under long time creep exposure could differ appreciably from the short time results. This point will be amplified later. The estimates are only intended to act as a starting point for preliminary considerations and should not take the place of actual tests.

The H-11 data to be examined is presented in Reference 2.6-21 and 2.6-22. Not all of the H-11 data presented in these two references

will be considered. Only that portion of the data which was considered valid by ASTM recommendations of 1964 (reference 2.6-23) will be considered. This portion of the data is tabulated in Reference 2.6-24 and shown in Figure 2.6-7 and 2.6-8. Since 1964, the ASTM criteria for validity of fracture toughness test results has been modified and in general made more restrictive (Reference 2.6-25). The present recommendations for plate specimens (bend specimens, etc.) require that

$$a \geq 2.5 (K_{IC}/\sigma_{ys})^2 \quad (1)$$

$$B \geq 2.5 (K_{IC}/\sigma_{ys})^2 \quad (2)$$

where a and B are the crack length and thickness of the test specimens. For notch round specimens, B has no meaning and the corresponding recommended requirement is

$$D \geq 10 (K_{IC}/\sigma_{ys})^2 \quad (3)$$

Where D is the diameter at the unnotched section. For the notched round specimen, the crack length "a" now corresponds to the notch depth. All data points shown in Figures 2.6-7 and 2.6-8 satisfy the B or D requirement. Some of the data points shown do not satisfy the crack length requirements and are approximately designated. It is believed that the crack length requirement is not as crucial as the thickness or diameter requirement and therefore these points have not been eliminated.

The data presented in Figure 2.6-7 shows the effect of room temperature yield stress level on K_{IC} . The strength level was varied by varying the tempering temperature. All specimens but two were machined from a 8.00 inch square forged billet. The two indicated exceptions were machined from 1.0 inch diameter bars. The specimens

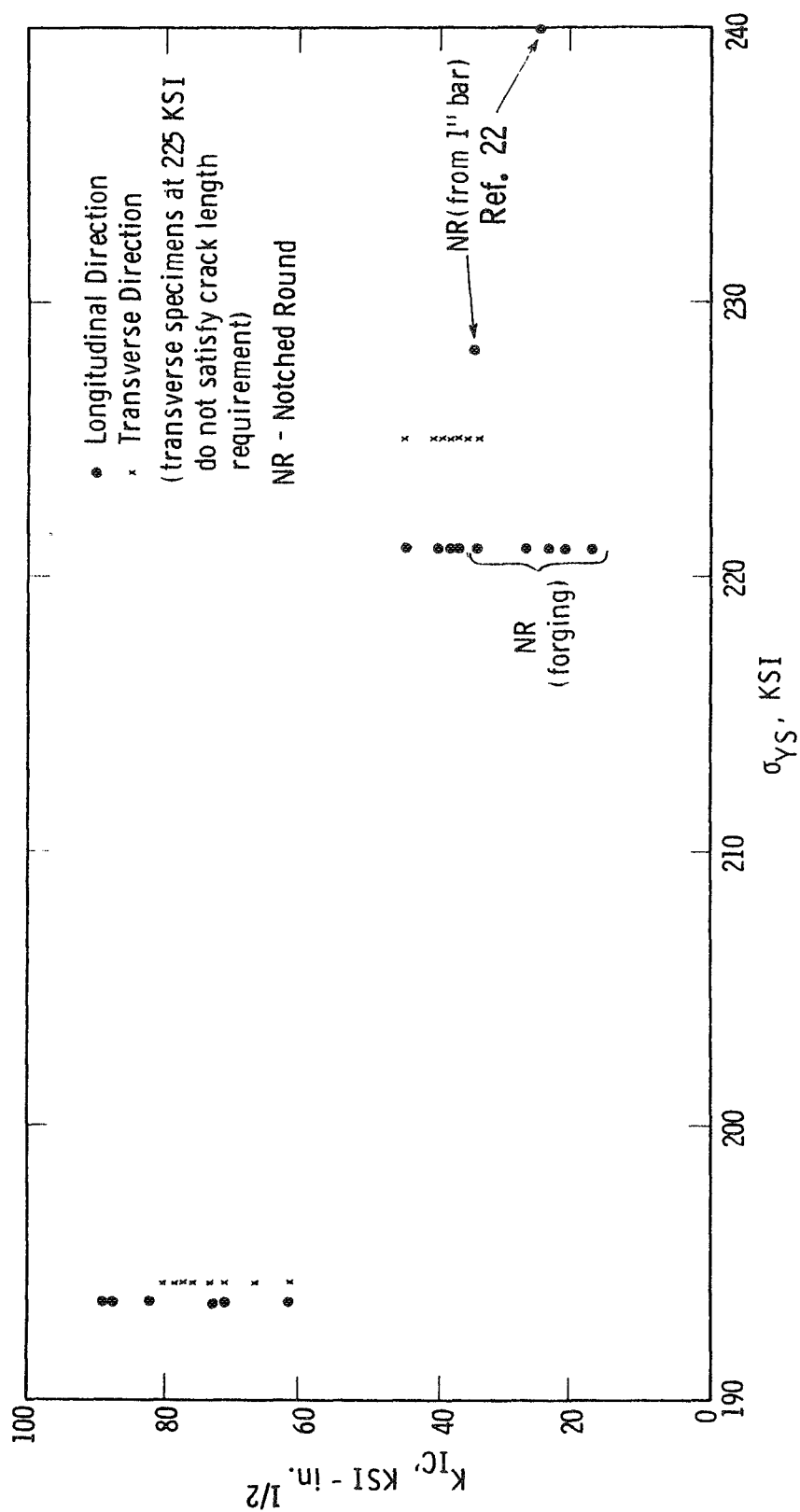


Figure 2.6-7. Data Obtained from Notched Bend Specimens Cut from 8 in. x 8 in. Square Forged Billet of H-11 Steel (Reference 21) Unless Otherwise Indicated. All Test Performed at Room Temperature.

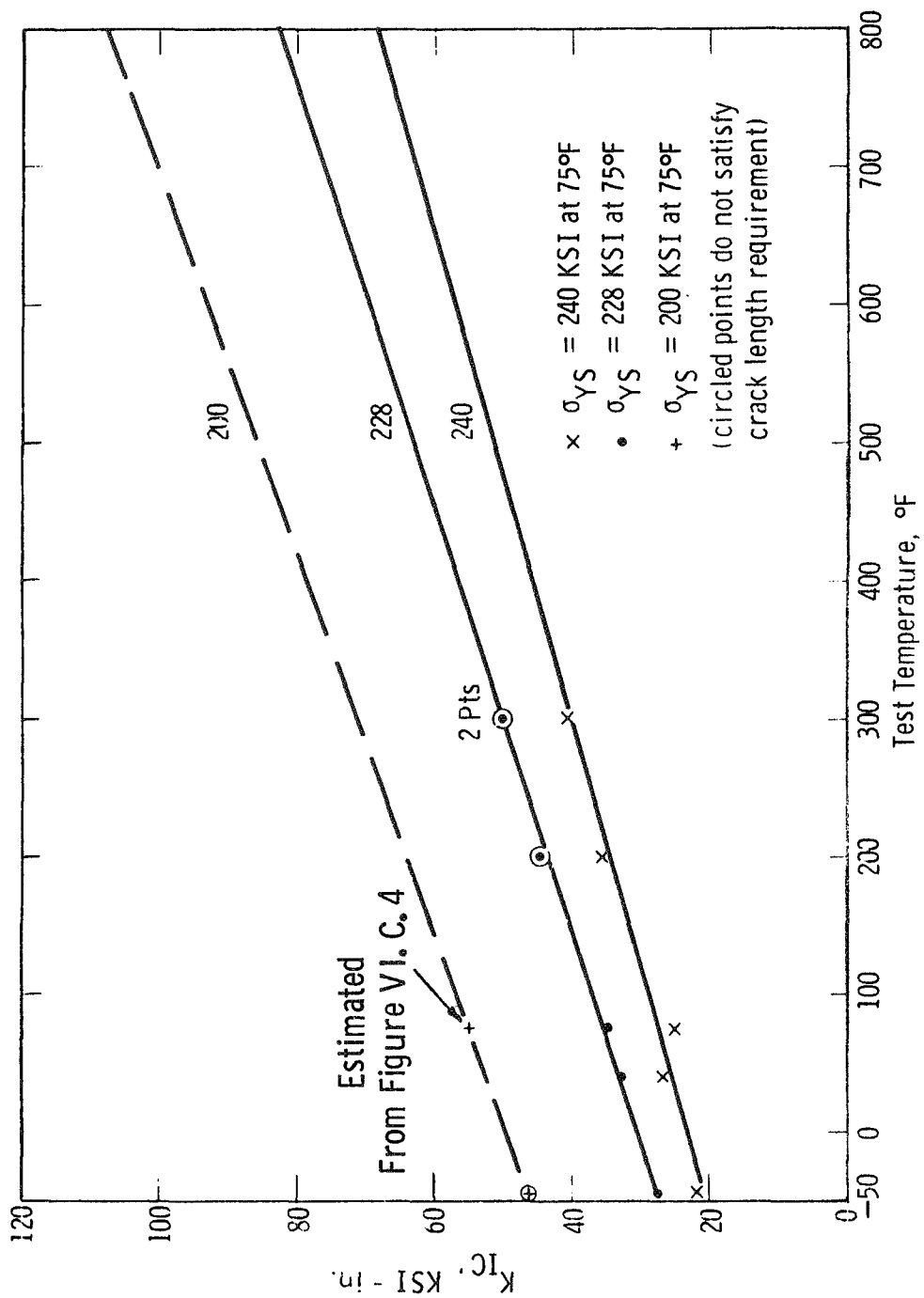


Figure 2.6-8. A11 Data Obtained from Notched Round Specimens of H-11 Steel Oriented in the Longitudinal Direction (Reference 22). A11 Specimens Cut from 1 Inch Diameter Bar.

taken from the forged billet should be more representative of the rotor condition than those machined from the bars. The data is mainly in the 195, 220, and 225 ksi yield stress ranges. The following comments on this data can be made.

- a. The notch round tests at the 220 ksi level have appreciable scatter and give lower values than the bend specimens for the same strength level.
- b. Specimen orientation does not appear to effect K_{IC} at room temperature for the particular forging considered.
- c. It is also reported in Reference 2.6-21 that for the strength levels considered in Figure 2.6-7, the position of the specimen in the forging had no significant effect on K_{IC} .

Due to the scatter shown in Figure 2.6-7, it is difficult to estimate a room temperature K_{IC} for the 200 psi yield strength level of interest. A conservative estimate can be made from a lower bound of the data. Such an estimate would give a room temperature K_{IC} of 55 ksi $\sqrt{\text{in.}}$

From Figure 2.6-8 a very rough conservative estimate of the effect of temperature on K_{IC} of the 200 ksi level can be made. The data shown in this figure was obtained from specimens machined from 1.0 inch diameter bars. Specimens machined from heavier sections would be more desirable. All specimens satisfy the unnotched diameter requirement, and all but the indicated points satisfy the crack length requirement. Data between -50°F and 300°F is shown for room temperature strength levels of 228 ksi and 240 ksi. A data point at -40°F is shown for what is believed to be a room temperature strength level of 200 ksi.

For many metals, the K_{IC} vary approximately linearly with temperature below some temperature T_0 . The K_{IC} 's shown in Figure 2.6-8 for two different strength levels vary approximately linearly with temperature. Above the temperature T_0 , the rate of increase of K_{IC} with temperature usually increases rapidly. With this point in mind, it can be stated that a linear extrapolation of the data shown in Figure 2.6-7 probably will be a conservative extrapolation. The linear extrapolation of the 228 and 240 levels are shown in Figure 2.6-8 as solid lines. The conservative estimate of K_{IC} variation for the 200 level could be obtained by either drawing a straight line between the -40°F value and the previously estimated room temperature value (55 ksi $\sqrt{\text{in.}}$) or by extrapolation of the 240 and 228 ksi strength level $-K_{IC}$ lines shown in the figure. Both approximations give roughly the indicated dashed line. This type of an extrapolation gives a K_{IC} of 110 ksi $\sqrt{\text{in.}}$ at 800°F for 200 ksi room temperature yield stress H-11 steel under short time test conditions.

From the estimated values of K_{IC} obtained in the previous paragraphs, an indication as to whether the principles of fracture mechanics are applicable to the rotor for short fracture conditions can be obtained. Where it is applicable, critical flaw sizes can be estimated. For initial considerations, the simplest and most likely defect is a penny shaped crack or flaw near the center of rotor. The relation between crack tip stress intensity, the uniform applied stress σ normal to the plane of the crack, and the radius "a" of crack is

$$K = \frac{2}{\sqrt{\pi}} \sqrt{a} \quad \sigma \quad (4)$$

The critical crack size is then

$$a_{cr} = \frac{\pi}{4} (K_{IC}/\sigma)^2 \quad (5)$$

Linear elastic fracture mechanics is only applicable to the prediction of crack instability when the plastic zone at the crack tip at instability is small compared to the crack length. To be more qualitative, it can be stated that based on experimental data (reference 2.6-25), fracture mechanics is applicable when

$$a_{cr} \geq a_m = 2.5 (K_{IC}/\sigma_{ys})^2 \quad (6)$$

where a_m indicates minimum crack size.

First room temperature conditions will be considered. The maximum tensile stress at the center of the rotor is approximately $\sigma = 100$ ksi. The estimated room temperature K_{IC} is 55 ksi $\sqrt{\text{in.}}$ and room temperature σ_{ys} is 200 ksi. From equation (5) a critical crack radius of $a_{cr} = 0.24$ in. is calculated. From equation (6), $a_m = 0.189$ in. Since $a_{cr} \geq a_m$, the applicability of fracture mechanics at this temperature is valid. Therefore, if K_{IC} is approximately 55 ksi $\sqrt{\text{in.}}$, then a penny shape flaw of 0.24 in. radius is needed for fracture. For the size rotor under consideration a flaw of this size should be detectable by non-destructive test methods.

The estimated short time K_{IC} at 800^oF was 115 ksi $\sqrt{\text{in.}}$. The yield stress at this temperature is 125 ksi. Therefore still maintaining $\sigma = 100$ ksi, the use of equations (5) and (6) results in $a_{cr} = 1.04$ in. and $a_m = 2.12$ in. From a rigorous point of view, these calculations would indicate that for short time considerations, plane strain fracture mechanics would not be applicable at this temperature ($a_{cr} < a_m$). An

estimated flaw size of $a_{cr} = 1.04$ in. for a K_{IC} of 115 ksi $\sqrt{\text{in.}}$ and $\sigma = 100$ ksi probably is realistic and could be easily detected. Flaws of this size are very unlikely however.

These calculations of the previous paragraphs show that for the estimated short time K_{IC} values, the size of flaw at the rotor center needed for catastrophic failure are large enough such that nondestructive test methods will be able to detect them. The estimate of the K_{IC} temperature dependence, although intended to be conservative, was very approximate. Fracture mechanics tests of the H-11 at the yield strength of interest and over the temperature range of interest should be carried out to see if this preliminary conclusion is a valid conclusion.

All estimates and calculations presented in this analysis are with respect to the effect of flaws on a short time basis. The influence of long time creep conditions on the reduction in ductility and subsequent reduction in fracture toughness in the presence of flaws has not been considered. There is very little data at present on the effect of creep on the fracture toughness of high strength steels. In Reference 2.6-26 it is reported that a relatively high strength material, AMS 6304, showed approximately 70 percent decrease in fracture toughness after creep exposure when toughness tested above the 250^oF toughness transition temperature. It is suggested that some testing be undertaken to determine the reduction in toughness of H-11 caused by creep exposure. Note also, the effects of cyclic loading have not been considered as the high stress areas in the rotor will be subjected to a neglectable amount of cycling.

Failure Mode, Effect and Criticality Analysis (FME & CA). The FME & CA study was carried out for purposes of identifying and cataloging the various modes and effects of component failures down to and including

subassembly levels wherever possible. The components and subassemblies were categorized by a reliability model that consisted of

Rotor and Rotor Coolant,
Stator and Stator Coolant,
Bore Seal Assembly,
Field Winding and Insulation, and
Armature Winding and Insulation Combined, and the Inert Hardware.

The bearings and seals were not considered as part of the reliability model. They are not a part of the specific design task governed in Section 2.

The specific configurations within the model were considered general in order to be applicable to all the Base Designs. The preliminary FME & CA was undertaken at a joint meeting of the materials, design, and reliability engineers most experienced in the KTA design philosophies. At this meeting, all known possible failure modes were identified, discussed, and tabulated, the purpose of which was to, in addition to discovering failure modes, provide guidance in choosing between alternate concepts in the alternator designs. The failure modes identified were correlated according to immediate or long term effects on the system. The severity of the effects were categorized as to:

Minor -- no noticeable immediate or long range effect,
Major -- no noticeable immediate effect but definite long range
effect, and
Critical - immediate and long range effect.

For the study, it was assumed the alternator was in an operating condition and successful operation constituted the ability to produce

useful power for the power system. The "power system" was defined as the KTA and its ability to produce power. The effect of the failure on the power system was evaluated from the standpoint of life, power output, overload, shorts, vibration, unbalance, overspeed capability, etc. The guiding philosophy in cataloging the failure modes was "What must the power system do to respond to the various failure modes." Because the Phase I study was preliminary in nature, the probable causes of the failure modes were included, the probability or likelihood of their occurrence was not included in the evaluations.

The tabulation of the preliminary FME & CA is presented in Table 2.6-7. In all, forty-seven major failure modes are catalogued in the table.

2.7 CONCLUSIONS AND RECOMMENDATIONS

The detail conclusions and recommendations are presented in the preceding discussions. The following will be limited to major highlights to give an overall view of the results.

It is recommended that two designs be considered for Phase II; they are the designs at either 19,200 rpm or 24,000 rpm. Both are at 1600 Hz, 480 volts and 800^oF or less potassium coolant. The 19,200 rpm case was the primary choice over the 24,000 rpm design, based on the Phase I requirements. However, with a stress-minimized design and some relaxation in the long-term overspeed requirement, the 24,000 rpm case could be made the more attractive from the electromagnetic design point of view.

The coolant temperature choice could be 800^oF, but for H-11 reliability considerations, should be 600^o to 700^oF. No apparent

TABLE 2.6-7

NASA/KTA ALTERNATOR

PRELIMINARY FAILURE MODE, EFFECT AND CRITICALITY ANALYSIS

<u>Probable Causes Of Failure</u>	<u>Mode Of Failure</u>	<u>Immediate Effect on Operation</u>	<u>Long Range Effect On Operation</u>	<u>Severity</u>	<u>Effect On Power System</u>
		<u>ROTOR</u>			
(particular reason not clearly identi- fiable)	Pole slot or lamina- tion short	Increased losses & tip temperature	Increased creep strain	Major	Shorten Life
(due to magnetic pulses, electrical vibration windage)	Individual pole lamina- tion fatigues	Cuts bore seal	Stator shorts thru open seal; excessive creep strain	Critical	No output or loss of system
(due to same as individual laminations)	Pole lamina- tions fatigue as unit	Breaks off into bore seal & stator	-----	Critical	Loss of system
(due to non- uniform pro- perties)	Non-uniform creep rate or yield	Increased vibration, creep strain	Increased creep strain	Major	Shorten life, noise
(due to loss of concen- tricity)	Stub shaft rubs seals at base	Increased vibration	Wear or seizure	Critical	Loss of system
(due to increased vibration or poor/ changed properties)	Stub shaft fatigue	Breaks off	-----	Critical	Loss of system
(due to em- brittling & high stress)	Crack propa-	Increased Strain	Rotor dis-	Critical	Loss of System

TABLE 2.6-7 (Continued)

<u>Probable Causes Of Failure</u>	<u>Mode of Failure</u>	<u>Immediate Effect On Operation</u>	<u>Long Range Effect On Operation</u>	<u>Severity</u>	<u>Effect On Power System</u>
<u>ROTOR - CONTINUED</u>					
(all due to mass transfer)	Mass transfer, stress corrosion & galvanic corrosion	Added solids into coolant	Increased stress, strain & possible loss of rotor	Major	Shorten life, loss of system
(due to corona & $\bar{U} \times \bar{B}$ in plasma)	MHD & plasma erosion on poles	Reduce output	Loss of pole tip or bore seal	Major	Reduce life or output
(due to flexibility of mounting configuration)	Windage baffle or support fatigues	Rubs on rotor	Increased vibration	Minor	Increase noise
(baffle coats with potassium or carbon)	Windage baffle shorts with conductive film	Increased losses	Increased rotor creep strain	Major	Shorten life
(due to fit or bearing wear)	Minor loss of concentricity	Increased vibration	Additional creep strain	Major	Shorten life
(due to loss of bearing)	Major loss of concentricity	Rub and/or loss of rotation	----	Critical	Loss of system
(embrittlement, creep resistivity, permeability)	Material property change	Change in power output	Excessive creep or yield	Major	Shorten life

TABLE 2.6-7 (Continued)

<u>Probable Causes of Failure</u>	<u>Mode of Failure</u>	<u>Immediate Effect on Operation</u>	<u>Long Range Effect on Operation</u>	<u>Severity</u>	<u>Effect On Power System</u>
<u>ROTOR COOLANT</u>					
(due to solids accumulation on system malfunction	Flow blockage (decreased or stopped)	Increased rotor tem- perature or vibration	Excessive creep or yield	Critical to major	Shorten life or loss of system
(in coolant passage)	Leak in weld	Increased cavity pressure & windage	Liquid erosion	Critical	Shorten life
(due to high fluid pressures)	Excessive stress corrosion effects	Added solids into coolant	Excessive stress, or strain, or loss of rotor	Major to critical	Shorten life or loss of system
(due to system malfunction)	Excessive coolant temperature	Possible plastic yield or rupture	Excessive creep or yield	Critical to major	Loss of system or shorten life
(due to seal failure or vent plugging)	Loss of cavity vacuum control	Increased cavity pressure & windage	Excessive rotor creep	Major	Shorten life

TABLE 2.6-7 (Continued)

<u>Probable Causes Of Failure</u>	<u>Mode of Failure</u>	<u>Immediate Effect on Operation</u>	<u>Long Range Effect on Operation</u>	<u>Severity</u>	<u>Effect On Power System</u>
<u>BORE SEAL ASSEMBLY</u>					
(due to fatigue in joint, tube, bellows or end member; stress corrosion, mass transfer or plasma erosion, instability of brazing alloy)	Loss of hermetic seal	Allows rub or increased vibration	Stator shorts	Critical	Loss of System or output
(due to potassium or carbon coating)	Seal I.D. shorts with conductive film	Increased losses	Increased rotor creep strain	Major	Shorten life
(due to stack or broken stator parts)	Fretting wear	Short out stack teeth	Loss of hermetic seal	Critical to major	Shorten life or loss of system
(due to abnormal electrical opera- tion)	Abnormal temperature excursion	Excessive thermal stress	Loss of hermetic seal	Critical to major	Shorten life or loss of system
<u>STATOR & STATOR COOLANT</u>					
(due to fatigue stress corrosion mass transfer)	Leak in cooling duct	Stator shorts increased stator cavity pressure to induce corona	----	Critical	Loss of output or system

TABLE 2.6-7 (Continued)

<u>Probably Causes Of Failure</u>	<u>Mode of Failure</u>	<u>Immediate Effect on Operation</u>	<u>Long Range Effect on Operation</u>	<u>Severity</u>	<u>Effect on Power System</u>
<u>STATOR & STATOR COOLANT - Continued</u>					
(due to solids accumulation or system malfunction or stopped)	Flow blockage decreased or stopped	Excessive temperatures and losses	Accelerate insulation deterioration	Major	Shorten life
(due to bond or contact pressure loss)	Duct separating from frame	Excessive temperatures and losses	Accelerate insulation deterioration	Major	Shorten life
(due to mechanical or electrical fatigue)	Stack clamps fatigue	Separation of stack, loss of power	Fretting wear on bore seal	Critical	Noise, loss of system
(due to system malfunction)	Excessive coolant temperature	Excessive losses, loosen stack fits	Accelerated deterioration or increased vibration	Major	Shorten life
(due to loose stack fit, fretting wear or shorting)	Loss of inter- laminar insulation	Increased losses and excessive temperatures	Accelerated deterioration	Major	Shorten life
(due to mechanical or electrical vibration)	fatigue of teeth	Breaks off, flux unbalance	Decrease output wears thru bore seal	Major to critical	Shorten life or loss of system
	Short circuit effects on stack	Turn Stacks in frames	Increase pressure and wear on insulation	Major	Shorten life

TABLE 2.6-7 (Continued)

<u>Probable Causes Of Failure</u>	<u>Mode Of Failure</u>	<u>Immediate Effect On Operation</u>	<u>Long Range Effect On Operation</u>	<u>Severity</u>	<u>Effect On Power System</u>
<u>WINDINGS & INSULATION</u>					
(wire expands in slot and tries to straighten end turns)	Short circuit operation	Cracks slot liners or end turn insulation or wedges. Straightens wires to bus rings	Accelerated insulation deterioration	Major to Minor	Shorten life, reduces output
(due to mechanical & electrical vibration)	Slot cell insulation fatigue	Loss of slot liner or wed- gers, increased temperatures	Stator shorts	Major	Reduces output
(due to mechanical & electrical vibration)	Anadur frets away	Loosening of windings, increased temperatures	Accelerated insulation deterioration	Major	Shorten life
(due to hand- ling abuse, electrical stress, poor manufacturing)	Cladding breaks	Silver migrates out of wire	Shorts insula- tion or opens wire	Major to critical	Reduces output
(due to plating or cladding breaks or chemical reaction)	Joint cladding instability	Loss of solid bond	Silver migration	Major to critical	Reduces output or losses output

TABLE 2.6-7 (Continued)

<u>Probable Causes Of Failure</u>	<u>Mode Of Failure</u>	<u>Immediate Effect On Operation</u>	<u>Long Range Effect On Operation</u>	<u>Severity</u>	<u>Effect On Power System</u>
<u>WINDINGS & INSULATION - Continued</u>					
(due to mechanical or electrical failure)	A/C opens	Reduction in output, phase unbalance	----	Major to critical	Reduces output or looses out- put, wave form quality poor
(due to mechanical or electrical failure)	F/C opens	Loss of output over speeds	----	Critical	No output
(due to system malfunction)	Excessive coolant temperatures	Increased losses, bus rings & wires sag	Accelerated deterioration possible short	Major	Shorten life losses output
(due to mechanical & electrical vibration)	Bus rings & leads fatigue	Open circuits	----	Major	Reduces output or losses out- put
(due to vibration)	Terminal cracks or breaks	Loss of her- metic seal, shorts out	----	Critical	Reduces output or losses out- put
(due to vibration)	Heat shorts fatigue	Excessive temperatures and losses	Accelerates insulation deterioration	Major	Shortens life

TABLE 2.6-7 (Concluded)

<u>Probable Causes Of Failure</u>	<u>Mode of Failure</u>	<u>Immediate Effect On Operation</u>	<u>Long Range Effect On Operation</u>	<u>Severity</u>	<u>Effect On Power System</u>
<u>INERT HARDWARE</u>					
(due to joint failure or fatigue crack)	Loss or rotor cavity hermetic seal	Lower cavity pressure, less windage	Loss of potassium	Major	Shorten life
(due to system malfunction)	Excessive coolant temperature	Thermal stress & distortions	Excessive creep loosens fits; lose concentricity	Major to critical	Shorten life or loss of system
(due to excessive vibration)	Fatigue of housings, endbells, etc.	Increase vibration	Loss of potassium	Major	Shorten life
(due to evaporation of coating)	Reduction of emissivity on stator ID	Excessive armature temperatures increase losses	Accelerate insulation deterioration	Major	Shorten life
(due to vibration)	Mounting flange loosens	Increased vibration, loss of concentricity	Accelerated fatigue & breaking-up	Major	Shorten life, loss of system

benefits appear in going to higher or lower temperatures than the 600^o to 800^oF range. Higher temperatures require different rotor materials that limit the KTA speed to 16,000 rpm. This lower speed penalizes the KTA and system. The lower temperatures with organic coolants do not offer significant reduction in design complexity nor do they allow speeds over 24,000 rpm, the maximum allowed for the potassium coolant also. Besides the system weight and coolant life penalties with the lower coolant temperature, the complexities of potassium-to-organic turbine shaft seals preclude any recommendations for organic fluids in the KTA alternator.

Deposits of organic fluids in the stator are hardly less harmful than those of potassium. Carbonized deposits become conductive and result in corona effects similar to potassium. The introduction of an organic lubricant would not eliminate the need for bore seal, but it would introduce additional problems of gumming up and collecting in the rotor and coupling cavity.

The recommended temperature of 600^o to 800^oF is tied to the choice of the H-11 rotor material. The studies also indicated H-11 steel at 45 R_c is the best choice of temperatures as low as 200^oF. Above 800^oF, magnetic materials are available for rotors but their strength capabilities penalize the KTA by limiting speeds. Magnetic properties of the materials evaluated appear adequate. As a guide to future materials development, the following statement can be made based on the present findings. "If a rotor material could be developed which had 1100^oF magnetic and strength properties similar to those for H-11 at 800^oF, then an 1100^oF KTA alternator could be developed as easily as and be as attractive as one for 800^oF coolant." The stator designs and

materials used are not limited until about the 1100°F temperature range so any higher temperatures are keyed upon the rotor material limitations.

In the stator design, it was determined that the same basic materials and concepts would be either desirable or adequate for 400°F designs, 800°F designs or 1100°F designs. Only slight material changes would be made between the three temperatures. The low temperature designs could use some standard low temperature materials, but because of the concept to design at high power density, the insulation and wire materials must still be capable of 800°F hot spots. The 1100°F designs would use slightly higher temperature wire and insulation materials and be supported better to prevent excessive creep or sagging of the Hiperco -27. Neither of these changes have any significance to the design concepts. Designs at all three temperatures make maximum use of direct radiation off the wire end turns and cooling ducts in direct contact with the stator frame and field coil to have sufficient cooling in a space vacuum ambient.

Regardless of whether both or either cavities (rotor and stator) are vented to space or whether potassium or organic coolant is used, a ceramic bore carrier or seal is required in the radial gap between the rotor and stator. The fabrication of the ceramic (99.8% alumina) body for 800°F or below coolant is within today's technology. At 1100°F, beryllia ceramic is required and this material will require development to fabricate the large size bodies required. The ceramic-to-metal joints and the metal end member design and fabrication will require the most development for any temperature or material choice once a design is finalized and fabrication starts. Venting the rotor cavity to space would obviate the need for a hermetic bore seal and a simpler

bore barrier concept would suffice. This would introduce the need for potassium shaft seals to space. Of the two - bore seals or shaft seals to space - the hermetic bore seal approach appears the most appropriate to the KTA requirements.

2.8 APPENDIX A

GRAPHICAL PRESENTATION OF PARAMETRIC ELECTRICAL DESIGN DATA

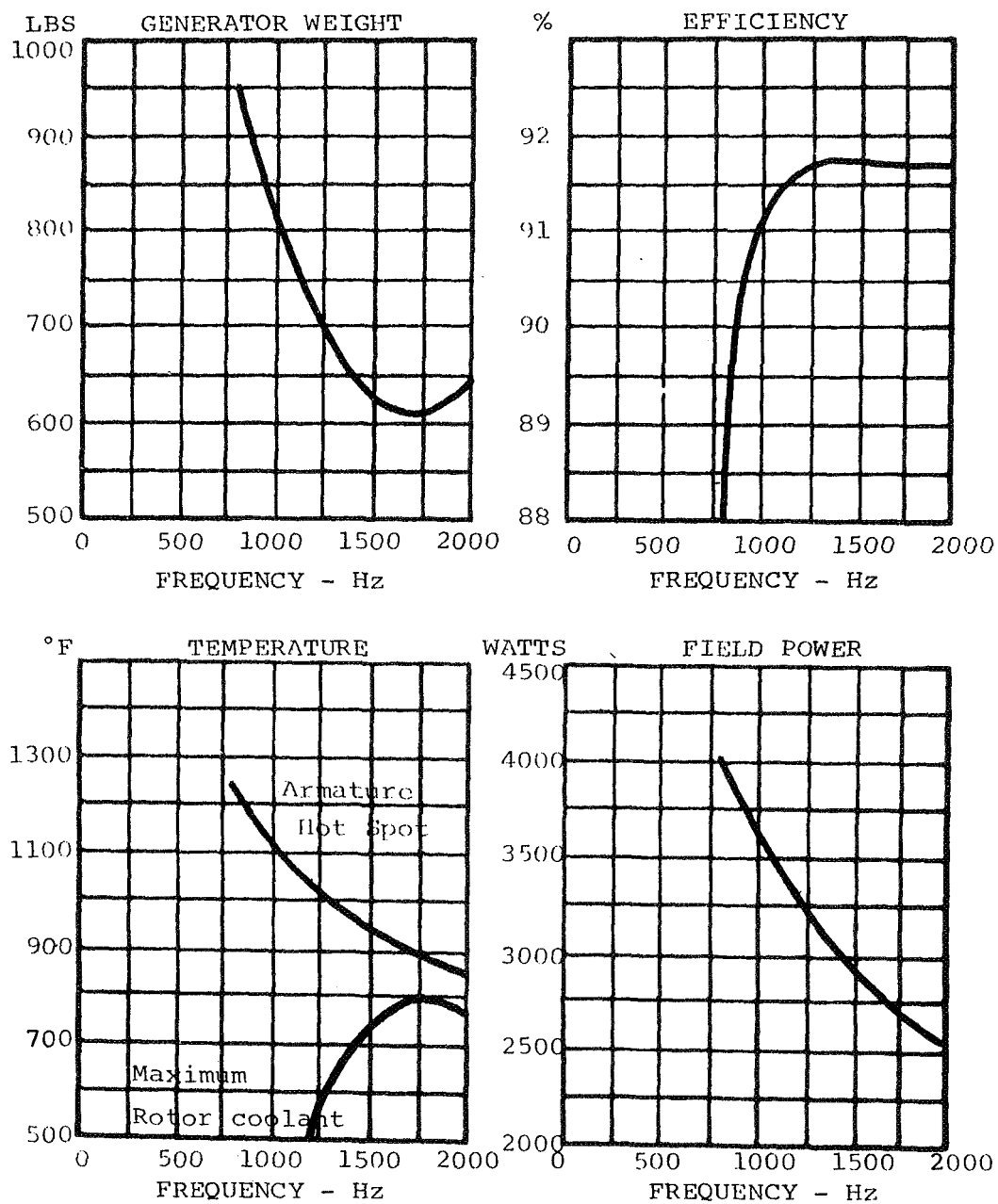


Figure 2.8-1. Frequency Study - 400°F.

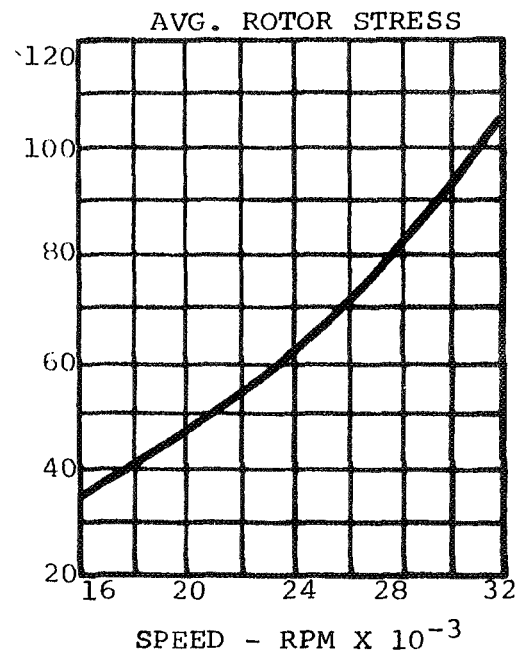
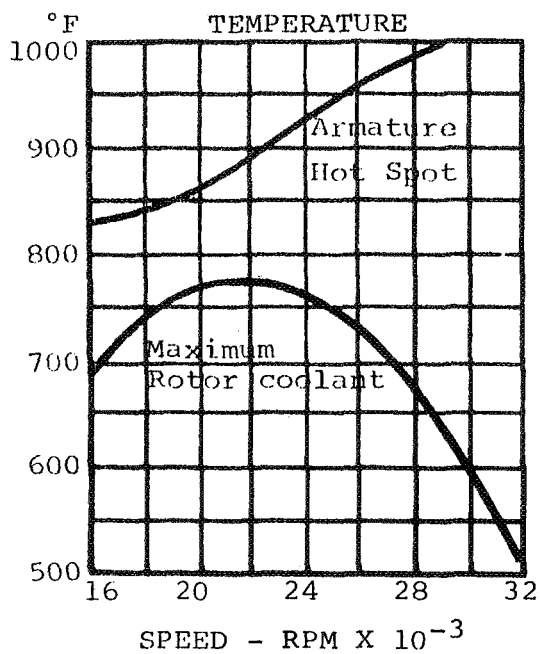
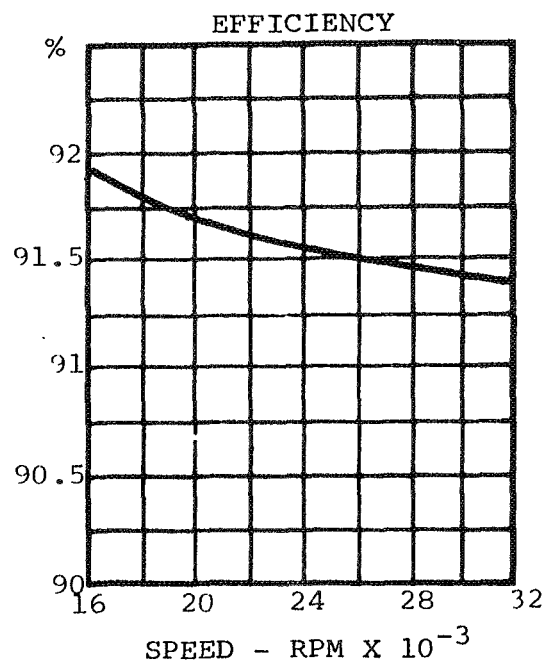
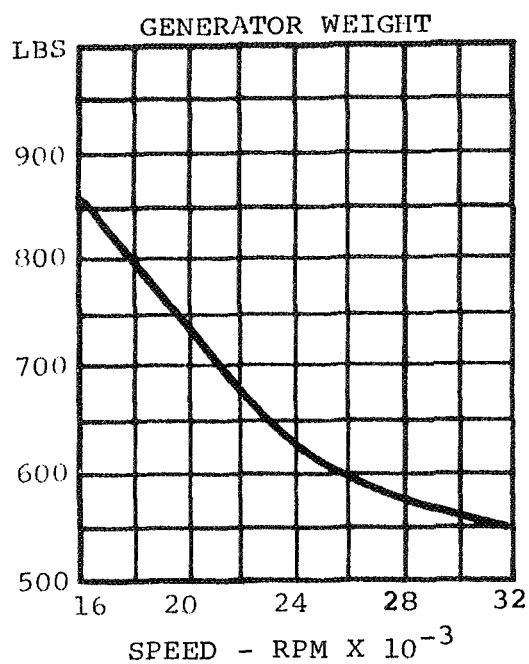


Figure 2.8-2. Speed Study - 400°F.

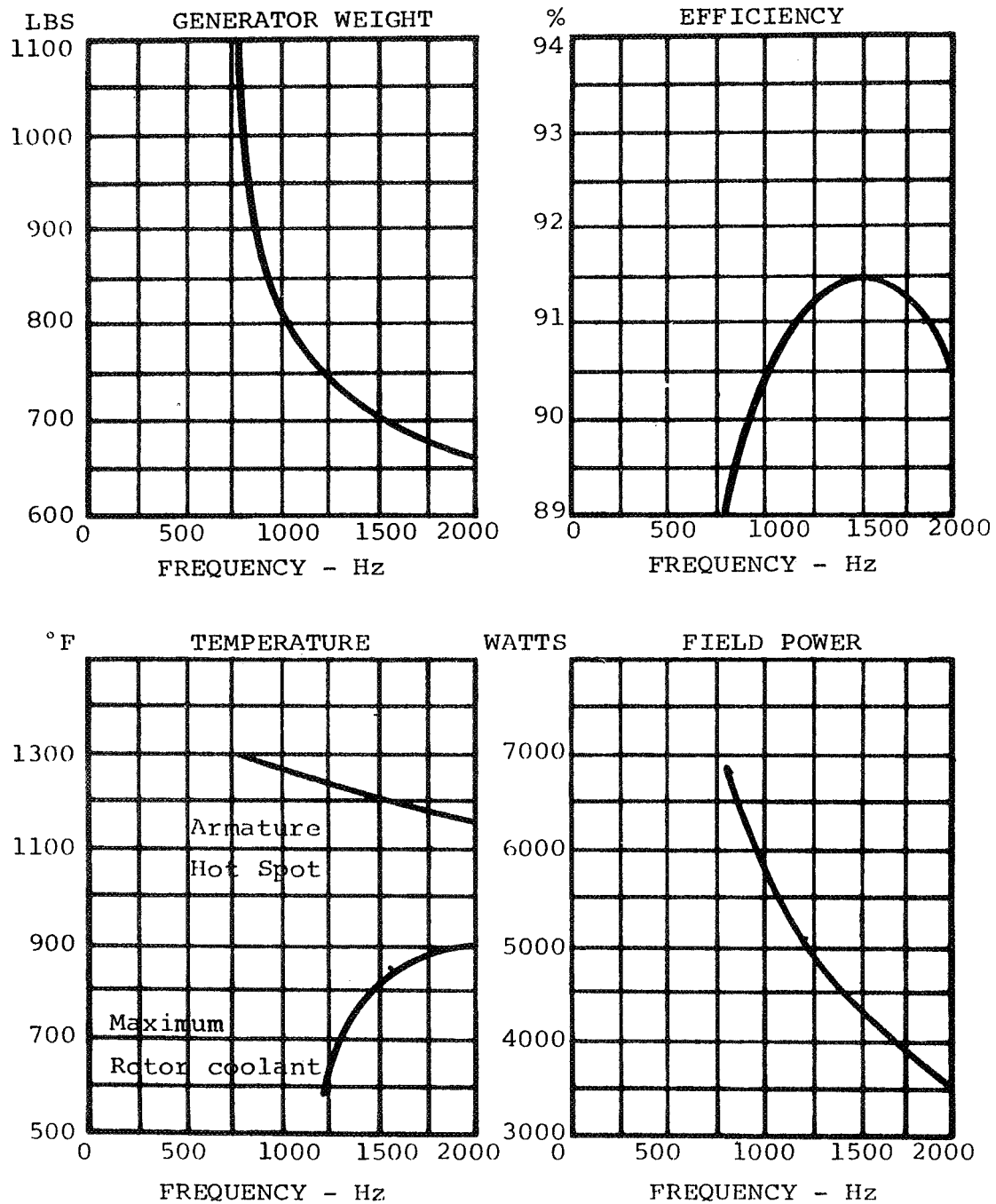


Figure 2.8-3. Frequency Study - 800°F.

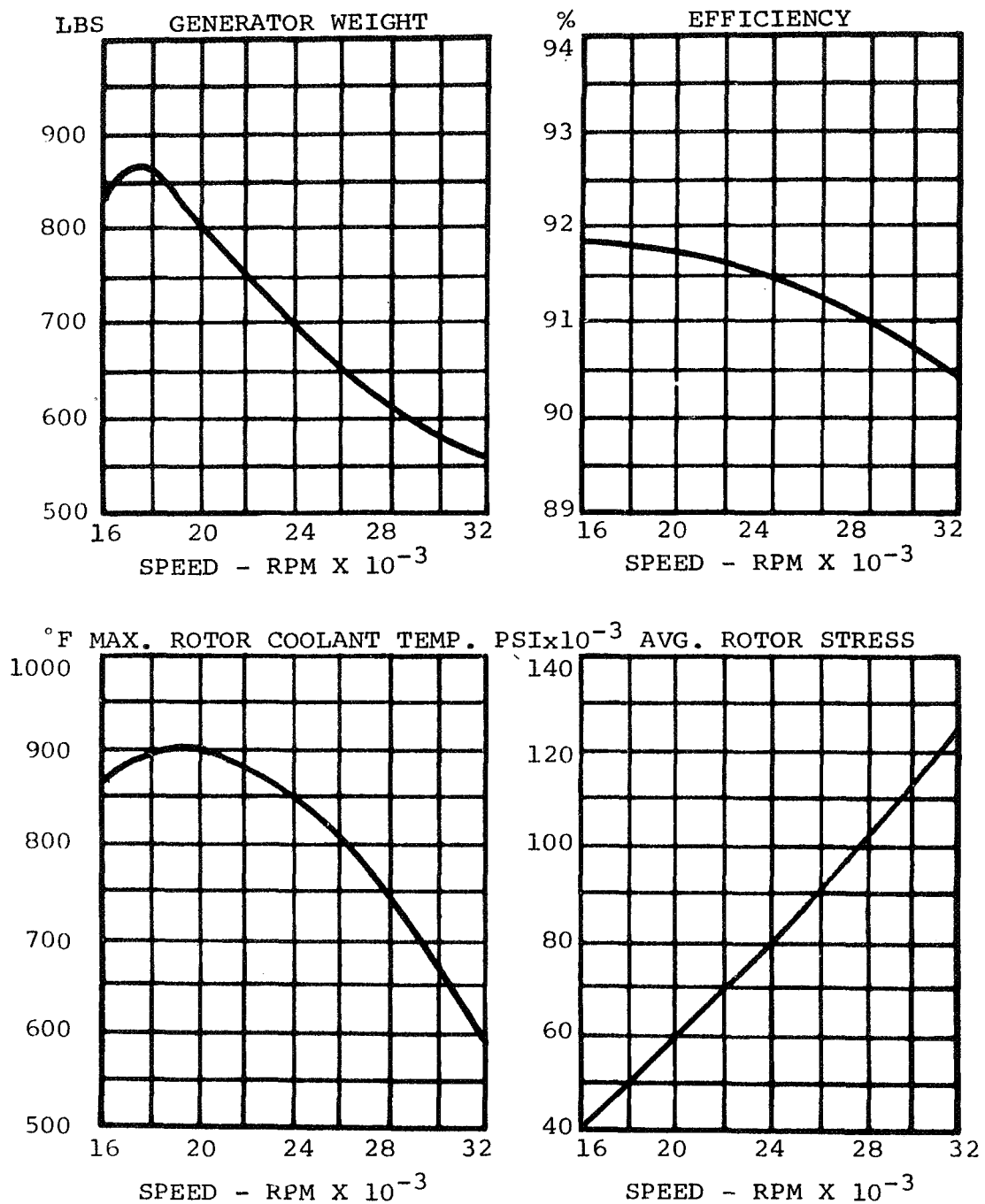


Figure 2.8-4. Speed Study - 800°F.

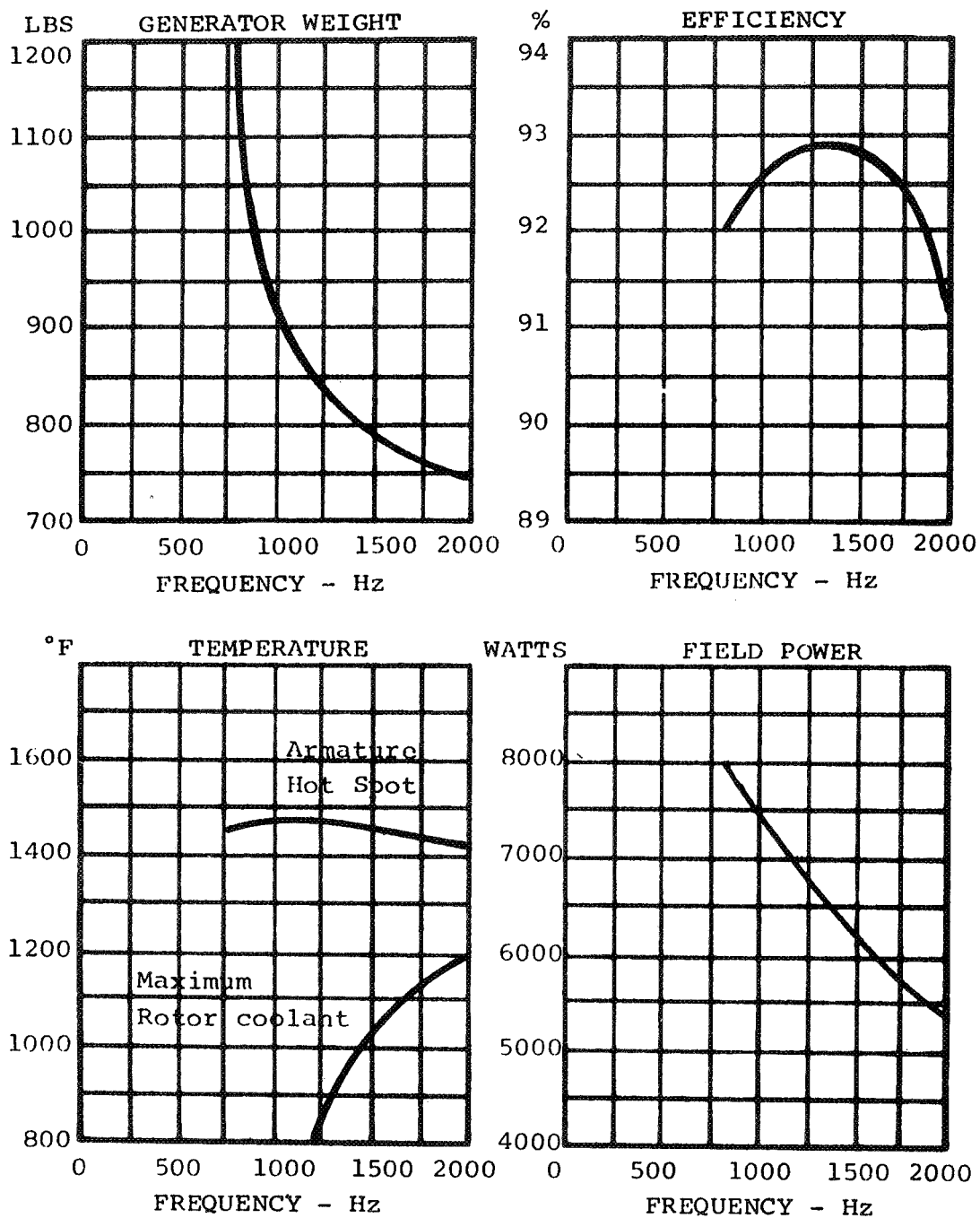


Figure 2.8-5. Frequency Study - 1100°F.

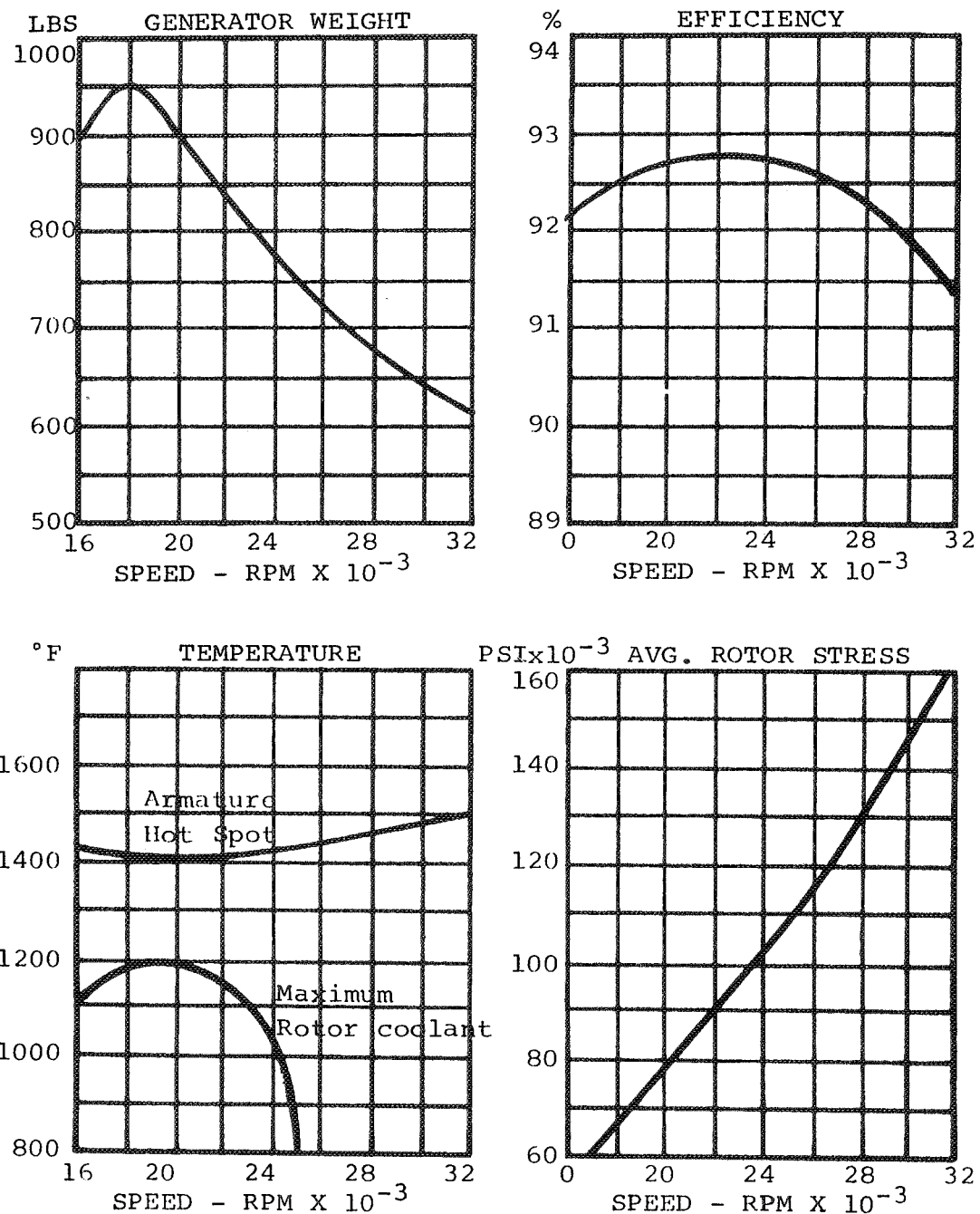


Figure 2.8-6. Speed Study - 1100°F.

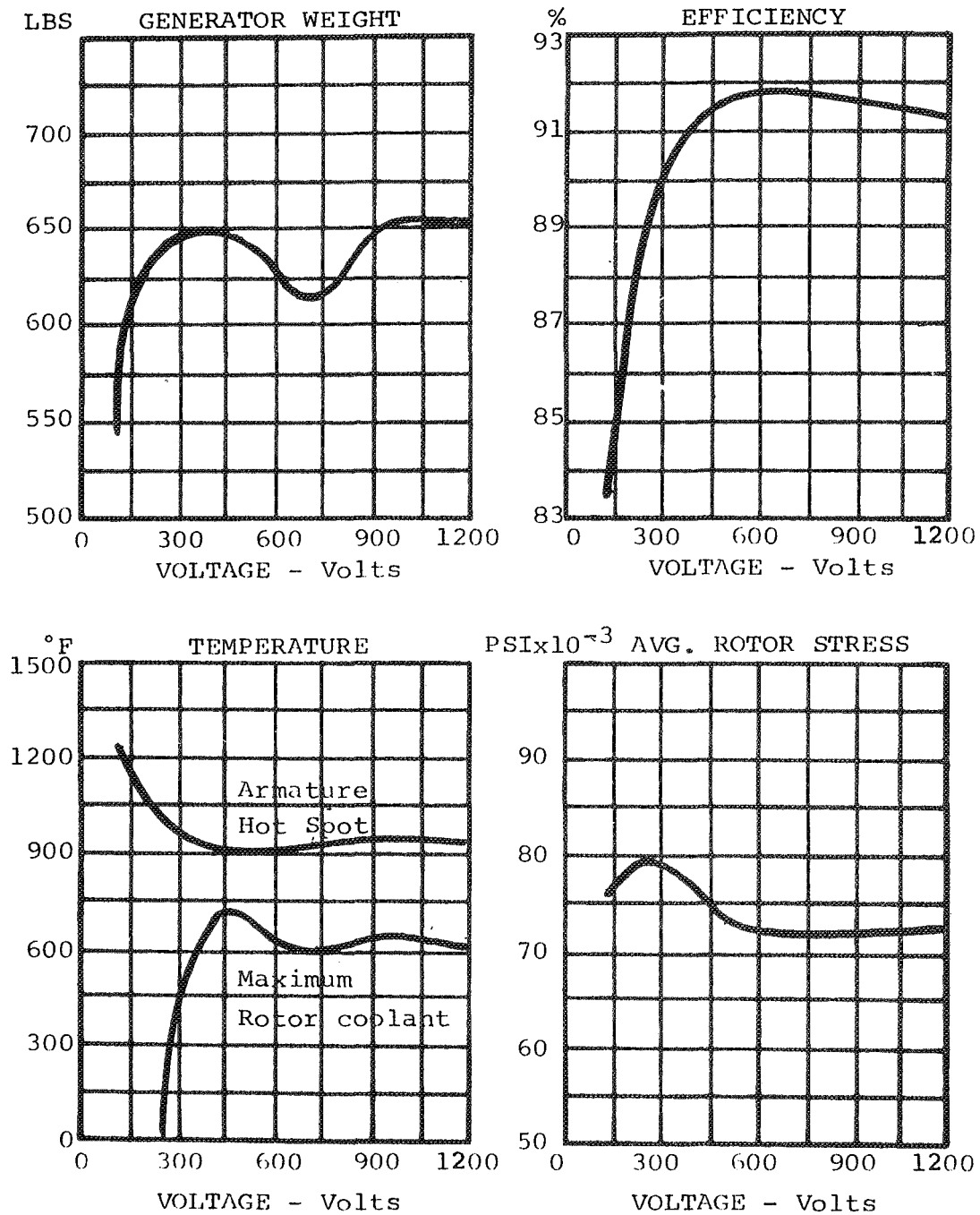


Figure 2.8-7. Voltage Study - 400°F.

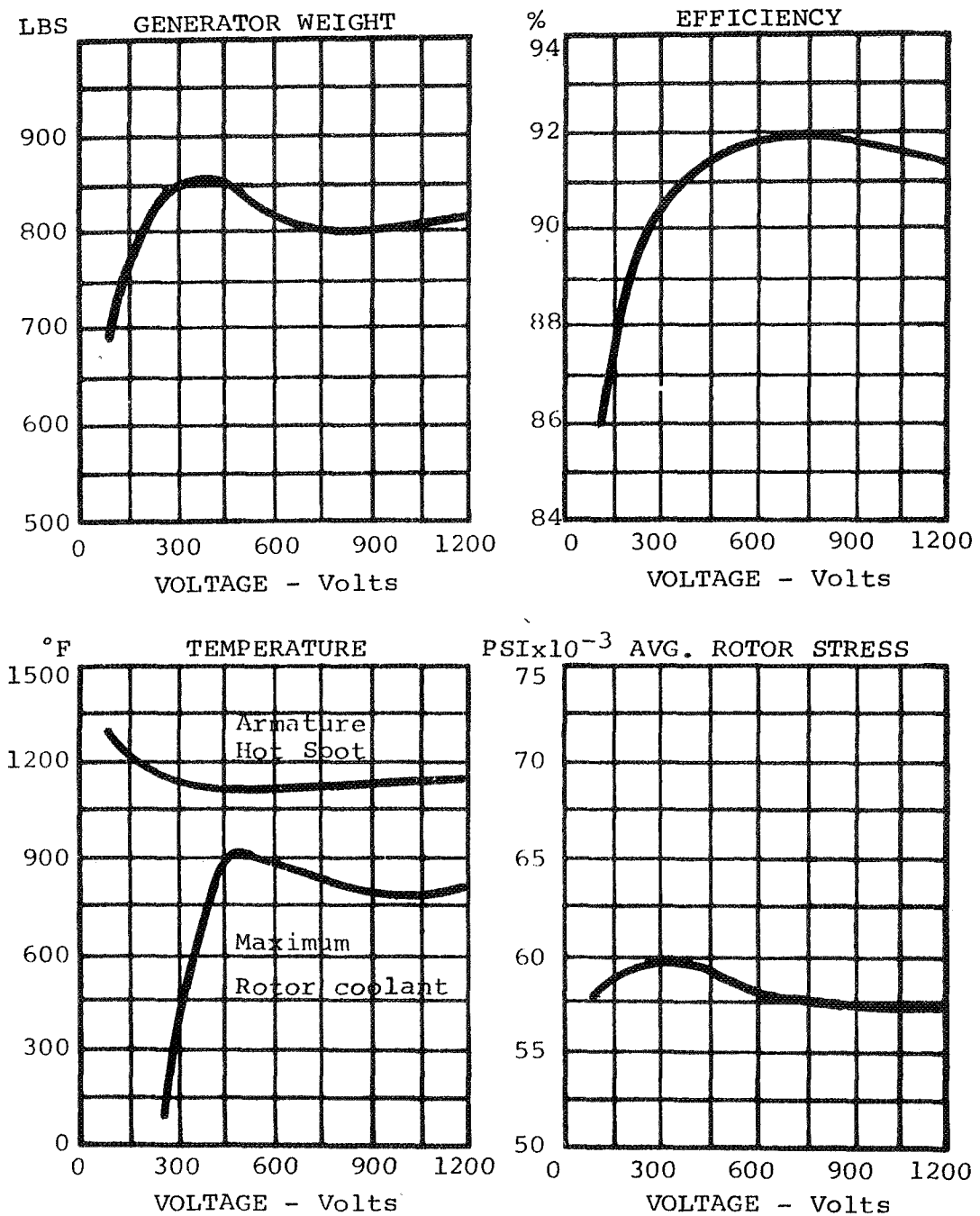


Figure 2.8-8. Voltage Study - 800°F.

2.9 APPENDIX B

COMMENTS ON WINDAGE LOSS CALCULATIONS

COMMENTS ON WINDAGE LOSS CALCULATIONS.

A refinement of the windage power loss calculations was made for analysis of the KTA alternator base designs using model theory and experimental data from a rotor configuration similar to the KTA rotor configurations. A discussion of the analytical refinements is presented in the following paragraphs.

The windage power loss within a rotor cavity is necessarily predicted from test data obtained from an operating model. To accomplish this task, it is necessary to (1) describe the significant variables associated with the fluid which influence the losses, (2) describe the variables which define the dynamic state of the rotor, (3) describe the dependent variable of interest, (4) reduce the number of variables involved by using dimensional analysis, and (5) obtain the performance characteristics of the geometrically similar rotor configuration in terms of the dimensionless parameters defined by the analysis.

The dimensional analysis for this case does not include all of the physical dimensions of the configuration since these variables are a linear function of one characteristic dimension of the rotor.

The dependent variable of interest from the standpoint of power loss is the windage torque, T . Therefore, this quantity was treated as the dependent variable. The independent variables associated with the fluid within the rotor cavity were assumed to be (1) the density, ρ ,

(2) the viscosity, μ , and (3) the velocity of progradation of pressure impulses or the sonic velocity within the fluid, a^* . The dynamic state of the rotor was defined by speed of rotation, ω . The dimensions of the rotor configuration were represented by one characteristic dimension, R , or the radius of the rotor.

The torque on the rotor can be written as an infinite series as:

$$T = \sum C_i [\omega^a, \rho^b, \mu^c, R^d, a^{*e}] \quad (1)$$

In order for Equation (1) to be correct, the dimensions of each of the series on the right must have the same dimensions as the dependent variable on the left side of the equation. Substituting the fundamental dimensions (M, L, and t) for each term the dimensional equation becomes:

$$ML^2t^{-2} = (t^{-1})^a, (ML^{-3})^b, (ML^{-1}t^{-1})^c, (L)^d, (Lt^{-1})^e \quad (2)$$

Equating exponents of M, L, and t gives:

$$1 = b + c$$

$$2 = -3b - c + d + e \quad (3)$$

$$-2 = -a - c - e$$

and solving this set of equations for c, d, and e in terms of a and b gives:

$$c = 1 - b$$

$$d = 2 + a + b \quad (4)$$

$$e = 1 - a + b$$

Rewriting the series Equation (2) for "T" using the exponents of equation, Equation (4) gives:

$$T = C_i [\omega^a, \rho^b, \mu^{1-b}, R^{2+a+b}, a^{*1-a+b}] \quad (5)$$

Next, collecting terms of similar known and unknown exponents, which according to Bridgman's statement of the π theorem gives n-m terms that

are dimensionless and contain all of the variables, one obtains,

$$\frac{T}{R^2 a^* \mu} = f \left[\left(\frac{WR}{a^*} \right)^a, \left(\frac{\rho R a^*}{\mu} \right)^b \right] \quad (6)$$

For this case, the total number of variables, n , is six and the total number of fundamental dimensions, m , is three which gives $n-m = 3$ or three groups of dimensionless parameters. With these three groups of dimensionless parameters experimental data can be illustrated graphically as shown by the solid lines in Figure 2.8-1. The graphical solution offered by this treatment of quantitative data can be used to predict the windage power loss for a similar case even if the rotor has a different diameter and is operated within a strange fluid. The accuracy of this graphical solution is limited by the assumed variables. If pertinent variables were not included in the functional Equation (5), the accuracy of the solution would be limited accordingly. The solution can be proved to be satisfactory only by experience.

A previous dimensional analysis of the problem by Smith (2.2-4, -5) did not consider the velocity of propagation of pressure impulses, a^* , as a pertinent variable. The dimensionless groups of terms for this case can be obtained by setting $e = 0$ in the previous exponent equations, (4), and solving for "a" and "c" in terms of "b". This gives

$$a = 1 + b$$

$$c = 1 - b$$

$$d = 3 + 2b$$

Rewriting the series Equation (2) for "T" using these new exponent equations gives

$$T = C_1 \left[\omega^{1+b}, \rho^b, \mu^{1-b}, R^{3+2b} \right]$$

Collecting terms of similar known and unknown exponents gives

$$\frac{T}{R^3 \omega \mu} = f \left[\frac{\rho \omega R^2}{\mu} \right]^b \quad (8)$$

In order to compare this functional equation with Equation (6), which was obtained by considering "a*" to be a pertinent variable, both sides of Equation (8) are multiplied by $R\omega/a^*$. This gives

$$\frac{T}{R^2 a^* \mu} = f \left[\frac{R\omega}{a^*} \left(\frac{\rho \omega R^2}{\mu} \right)^b \right] \quad (9)$$

This gives two functional relationships which contain the same dimensionless groups of terms on the left hand side of Equations (6) and (9).

The dimensionless torque parameter in Equation (9) is (1) an exponential function of the Reynolds number $\frac{\rho \omega R^2}{\mu}$, and (2) a linear function of the Mach number, $\frac{R\omega}{a^*}$, associated with the pole tips. Experience has proved that the dimensionless torque parameter is not a linear function of the Mach number. Thus, the accuracy of the functional equation obtained by a dimensional analysis which does not include a^* is limited accordingly. The difference between the two functional relationships, Equation (6) and (9), also can be illustrated graphically in Figure 2.8-9. After a plot of the quantitative data has been made as illustrated by the solid lines in Figure 2.8-9, a line through points of constant Reynolds number can be drawn as shown by the broken lines in Figure 2.8-9. In this instance, the Reynolds number equals the product of $R\omega/a^*$ and $\rho R a^*/\mu$. This line may not be straight; it probably has an upward curvature when the Mach number is large. On the other hand, the functional relationship given by Equation (9) infers that all of these points fall on a straight line as indicated in Figure 2.8-9. Thus, when experimental data is plotted according to the functional relationship

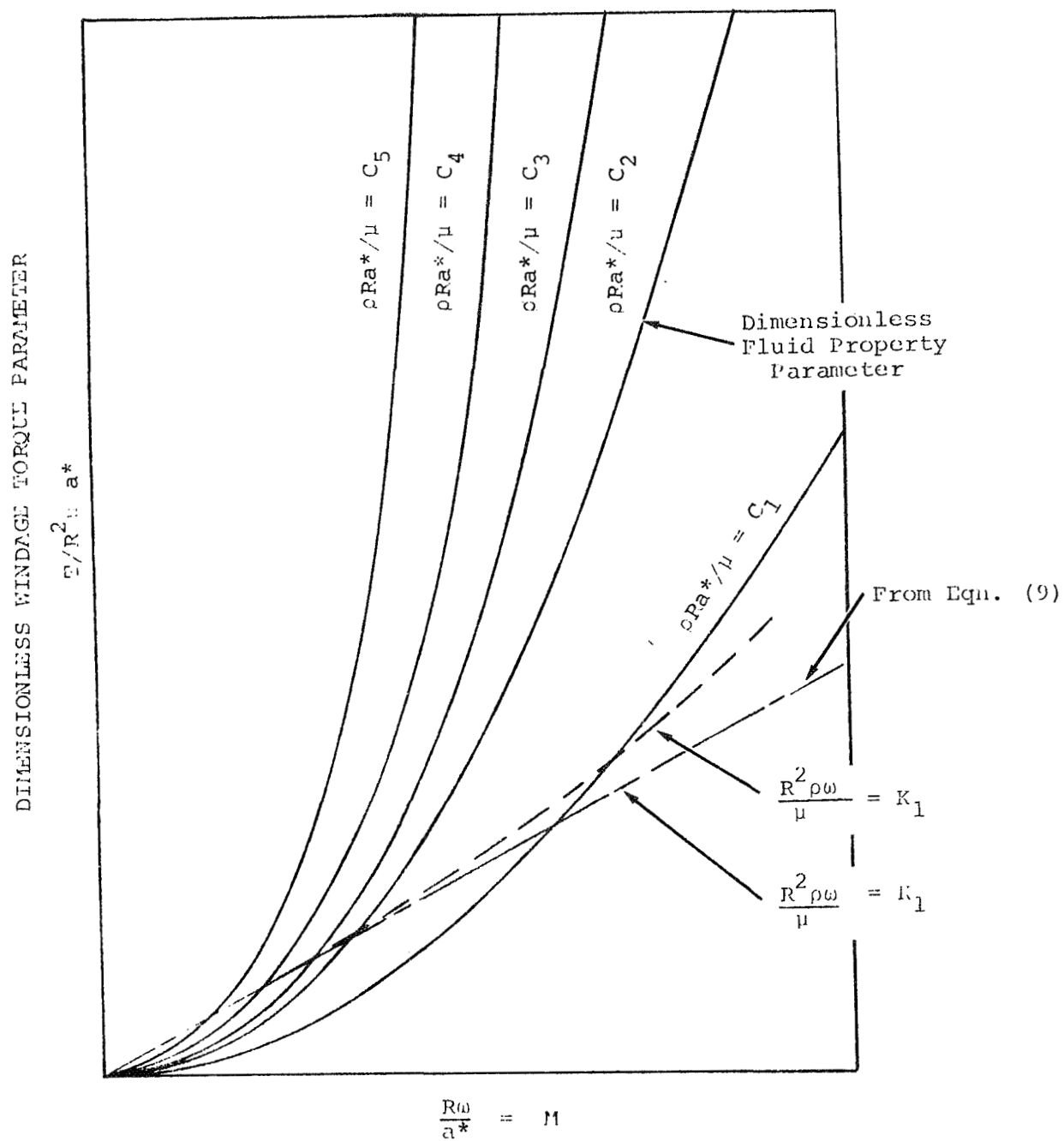


Figure 2.8-9. Dimensionless Rotor Speed Parameter.

defined by Equation (9), the effects of Mach number are omitted. Error will be introduced when this data is used to predict the windage power loss for cases which have a Mach number different than the test case.

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3. TURBINE FLUID DESIGN

3.1 FLUID DESIGN PROCEDURE

The basic tool used for the fluid design of the KTA turbines is a multistage (15) turbine pitch-line design computer program. The program is based on the thermodynamic properties of potassium, and has three options for the calculation of vapor expansion in a stage; equilibrium and supersaturated with or without subsequent reversion to equilibrium conditions. The fluid expansions are calculated with total pressure losses based on blade row loss correlations, Reference 3.1-1, and include profile, secondary flow and tip clearance losses. This computer program gives sufficient information for preliminary design. Spanwise and blade-to-blade solutions are computerized but only give information needed for final design.

Inputs to the program include inlet conditions, flow rate, rotative speed, number of stages and exit pressure. Also input for each stage are pitch line wheel speed, nozzle efflux angle and velocity, enthalpy change, tip clearance and rotor exit to inlet area ratio. The program iterates on stage enthalpy drop to satisfy the exit pressure specified. Program outputs include flow path dimensions, pitch line velocity diagrams, total pressure losses, stage loading, reaction and turbine efficiency.

The initial design specifications were 450 kw_e output with inlet vapor conditions of 2150°F and 214 psia and exit conditions of 1330°F and 10.6 psia. A rotative speed between 16,000 and 32,000 rpm was to be selected as a result of the study. Several "rule of thumb" turbine design limits, based on judgment and experience, were used to guide the turbine design procedure. These are listed in Table 3.1-1.

The turbine fluid designs were based on supersaturated expansions in each stage with reversion to equilibrium conditions at the exit of each stage. This calculation model was used for the two-stage potassium turbine and the test results were in good agreement with the calculations as shown in Figures 3.1-1 and 3.1-2, where the symbols represent the test data and the lines represent the calculated performance.

3.2 PARAMETRIC STUDY

In order to make manageable the number of designs undertaken, the speed range was covered by considering rotative speeds of 16,000, 20,000, 24,000, and 32,000 rpm only. Initially, preliminary designs were made for turbines with in-line flow paths for the following ten configurations:

<u>RPM</u>	<u>Number of Stages</u>
16,000	10
16,000	12
20,000	8
20,000	10
20,000	12
24,000	8
24,000	10

TABLE 3.1-1
TURBINE FLUID DESIGN LIMITS

TARGET MINIMUM BLADE HEIGHT, IN.	0.4
MAXIMUM ROTOR-HUB TURNING ANGLE, DEG.	130
MAXIMUM TIP SPEED, FPS	900
MAXIMUM STATE-TO-STAGE ANNULUS AREA RATIO	1.45
MINIMUM HUB REACTION, PERCENT	0
MAXIMUM RATIO LAST- TO FIRST-STAGE PITCHLINE SPEED	$(1.045)^n$

n = NUMBER OF STAGES

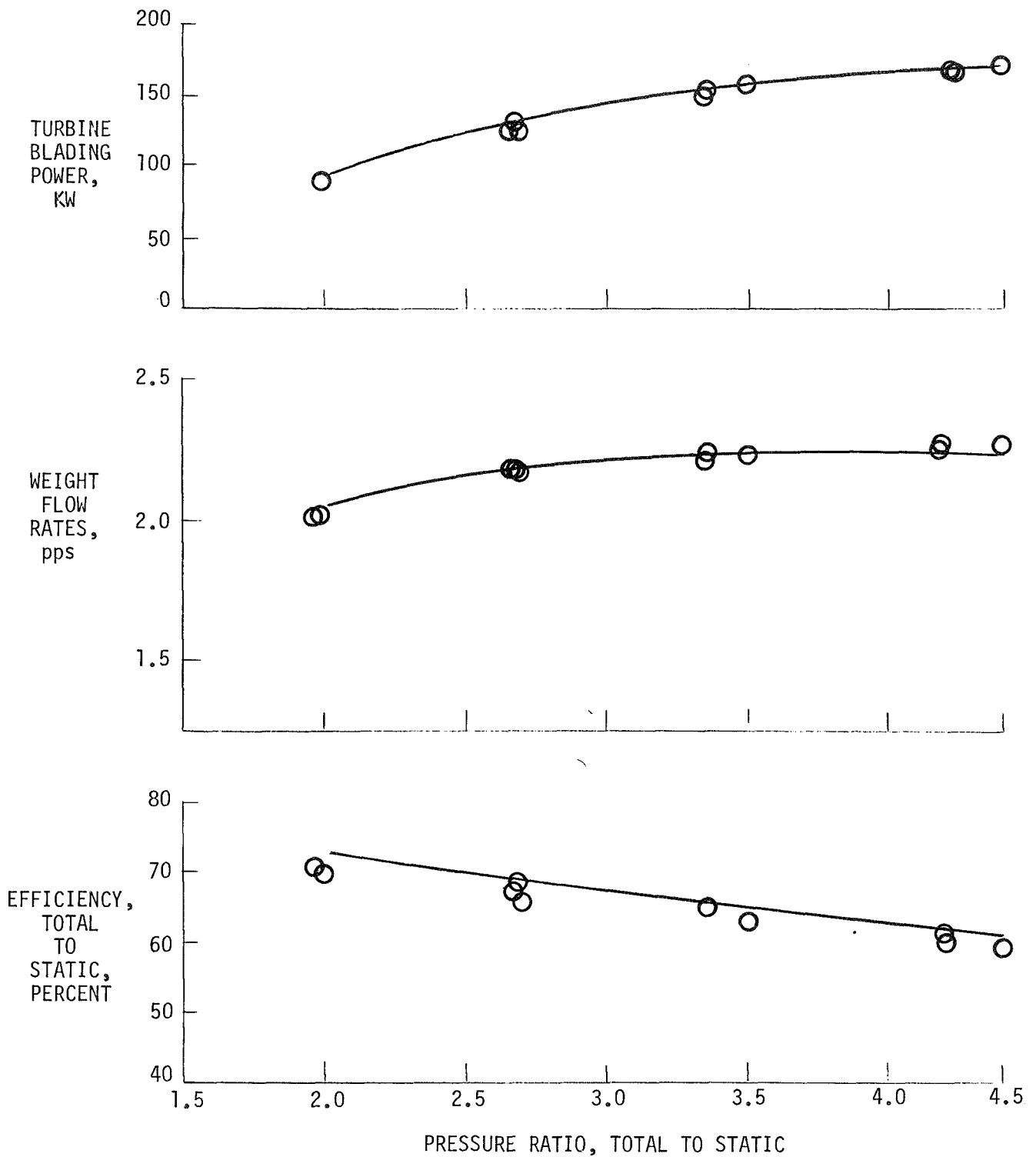


Figure 3.1-1. Comparison of Measured and Calculated Power, Weight Flow and Efficiency Versus Total to Static Pressure Ratio Inlet Temperature, 1550°F, Rotative Speed 15,400 rpm.

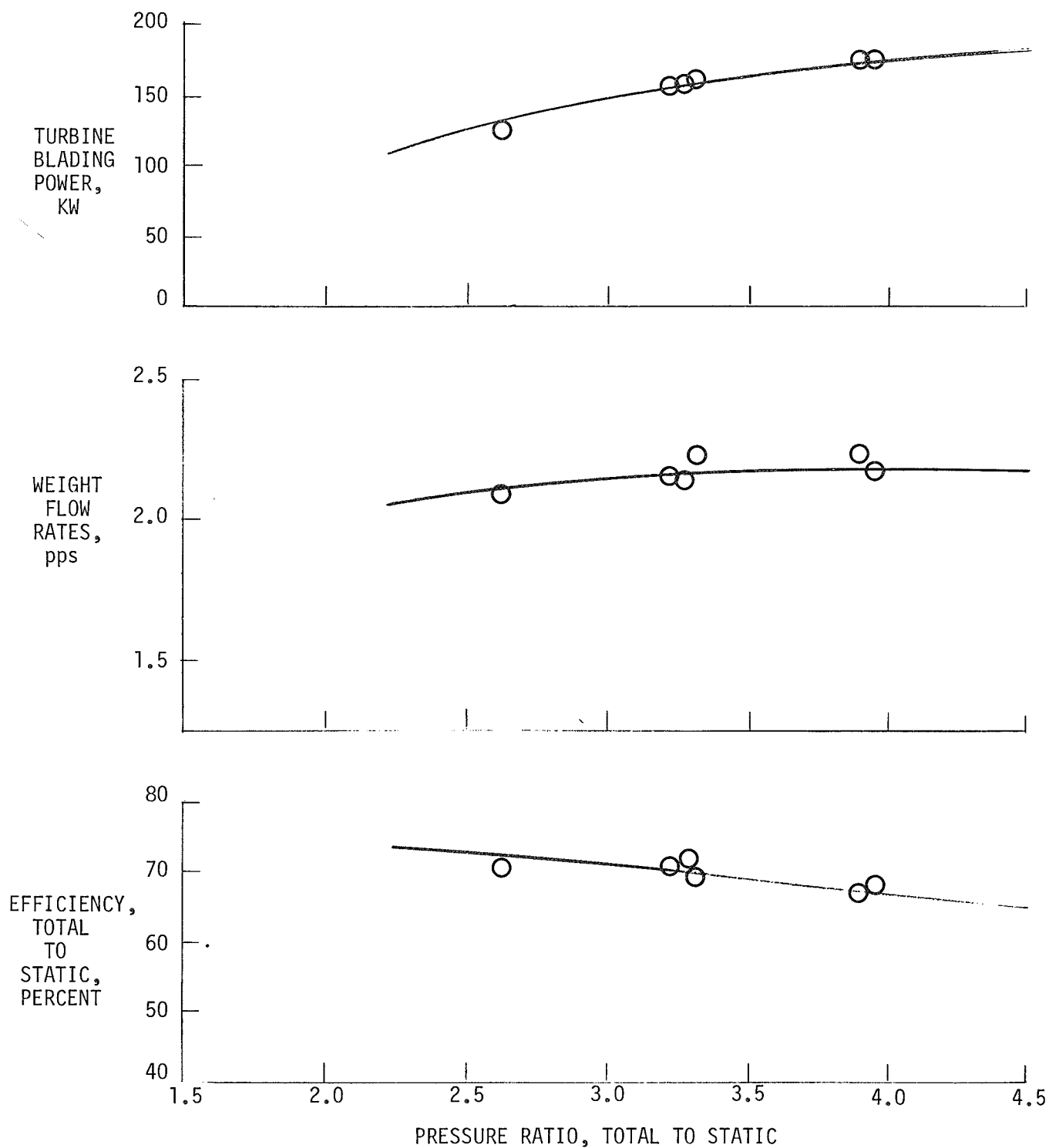


Figure 3.1-2. Comparison of Measured and Calculated Power, Weight Flow and Efficiency Versus Total to Static Pressure Ratio Inlet Temperature, 1550°F, Rotative Speed 19,200 rpm.

<u>RPM</u>	<u>Number of Stages</u>
24,000	12
32,000	6
32,000	8

Preliminary designs were also made for each of the ten turbines in the split spool arrangement. After a vapor quality of about 90 percent was attained in the high pressure turbine, the flow was taken out through a scroll to a centrifugal separator in which 80 percent of the moisture was assumed to be removed. The drier vapor was then expanded in a low pressure turbine. A pressure drop of 1.5 psi was used between the high and low pressure sections of the turbine. This pressure drop was based on pressure losses of 0.5 of a velocity head in the exit scroll, one velocity head in the separator, and 0.25 of a velocity head in the inlet scroll to the low pressure turbine.

After three month's work, instructions were received from NASA altering the turbine inlet and exit conditions. The turbine inlet conditions were changed to 2100°F and 165 psia, and the exit conditions were changed to 1280°F and 7.8 psia. At this time, the rotative speed of 20,000 was changed to 19,200 rpm because of alternator frequency considerations. Sixteen turbine designs were carried out for the revised cycle conditions, 11 at 19,200 rpm, the preferred rotative speed for the alternator, three at 24,000 rpm, and one each at 16,000 and 32,000 rpm. Shown in Figure 3.2-1 is the variation in turbine efficiency with speed and number of stages for single flow turbines. These values are based on aerodynamic and thermodynamic losses and do not include parasitic losses due to bearings and seals. As expected, turbine efficiency increases with rotative speed and number of

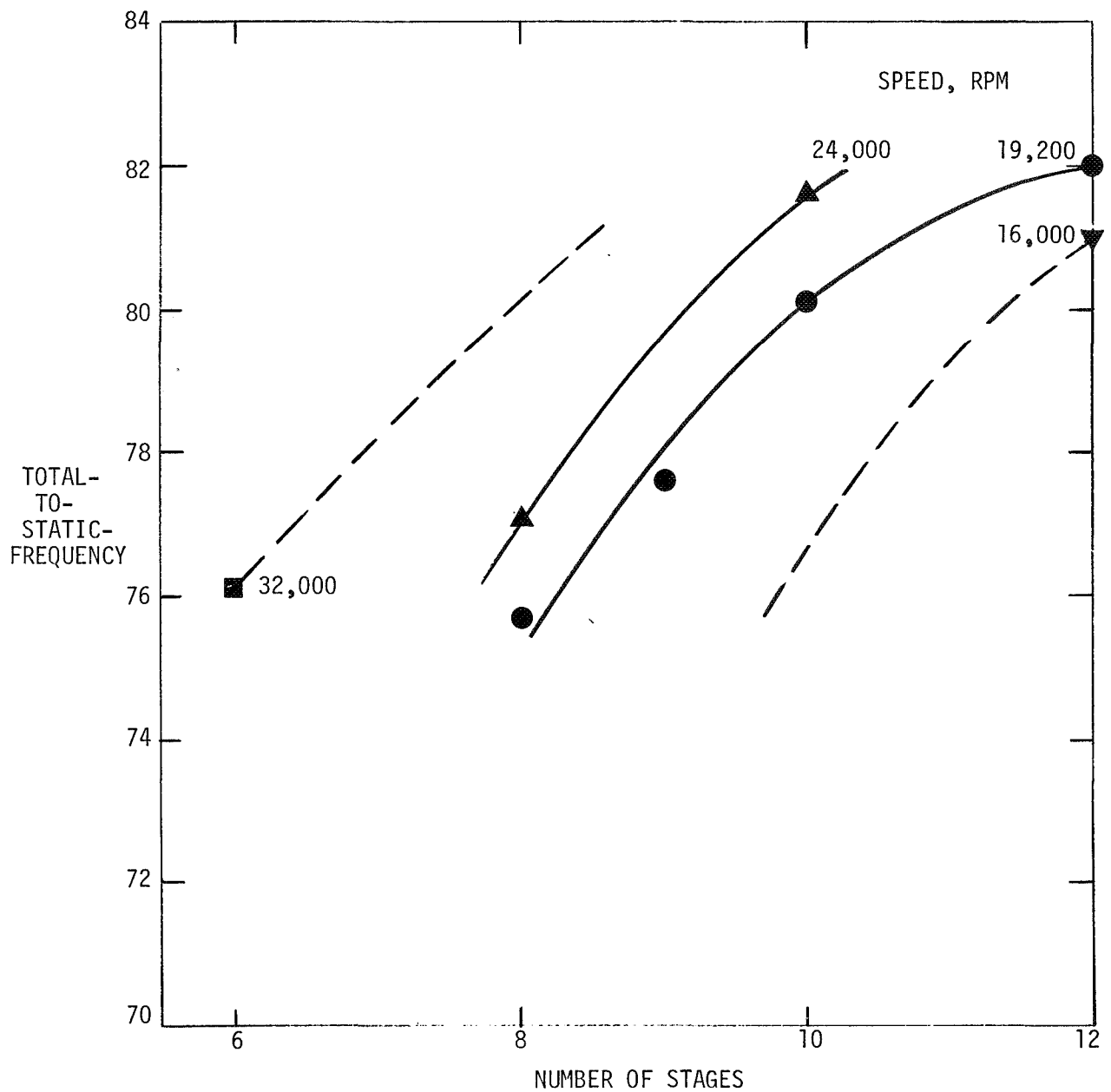


Figure 3.2-1. Effect of Speed and Number of Stages on Efficiency.

stages. A greater rotative speed results in a larger blade height for a given wheel speed, reducing the tip clearance loss. A larger number of stages permits either lighter stage loading and/or lower wheel speeds.

The parameters of a ten-stage, 19200 rpm single spool turbine design are shown in Figures 3.2-2 and 3.2-3 and are typical of the KTA turbines. The characteristics shown in Figure 3.2-2 are input and the ones in Figure 3.2-3 are output results. The pitchline loading, reaction and nozzle angle distributions shown in Figure 3.2-2 are not unusual. The average pitchline loading (ratio of turbine head to pitchline wheel speed required) is 1.78. The wheel speed is low (450 ft/sec) in the first stage to maximize the blade height and increases to 700 ft/sec in the last stage. The parameters in Figure 3.2-3 are shown with the target or limiting values as dashed lines. A drawing of this turbine is shown in Figure 3.2-4. This particular design has a high axial thrust (1590 lb) which must be balanced by a thrust bearing and has high labyrinth seal leakage losses (26 KW), particularly at the front of the rotor. These losses are shown in Figures 3.2-5 and 3.2-6. The leakage flow rate in Figure 3.2-5 shows the leakage in each stage and the total leakage. The difference is the leakage at the front of the rotor. The power loss shown in Figure 3.2-6 is based on the fact that the leakage flow is not available to do work in the stage.

The turbine blading losses are shown in Figure 3.2-7 and include profile, secondary flow and tip clearance losses based on the data of Reference 3.1-1. The blading efficiency is 81 percent. The total loss breakdown for this particular turbine is shown in Table 3.2-1. It can

10 STAGE, 19,200 RPM SINGLE SPOOL TURBINE

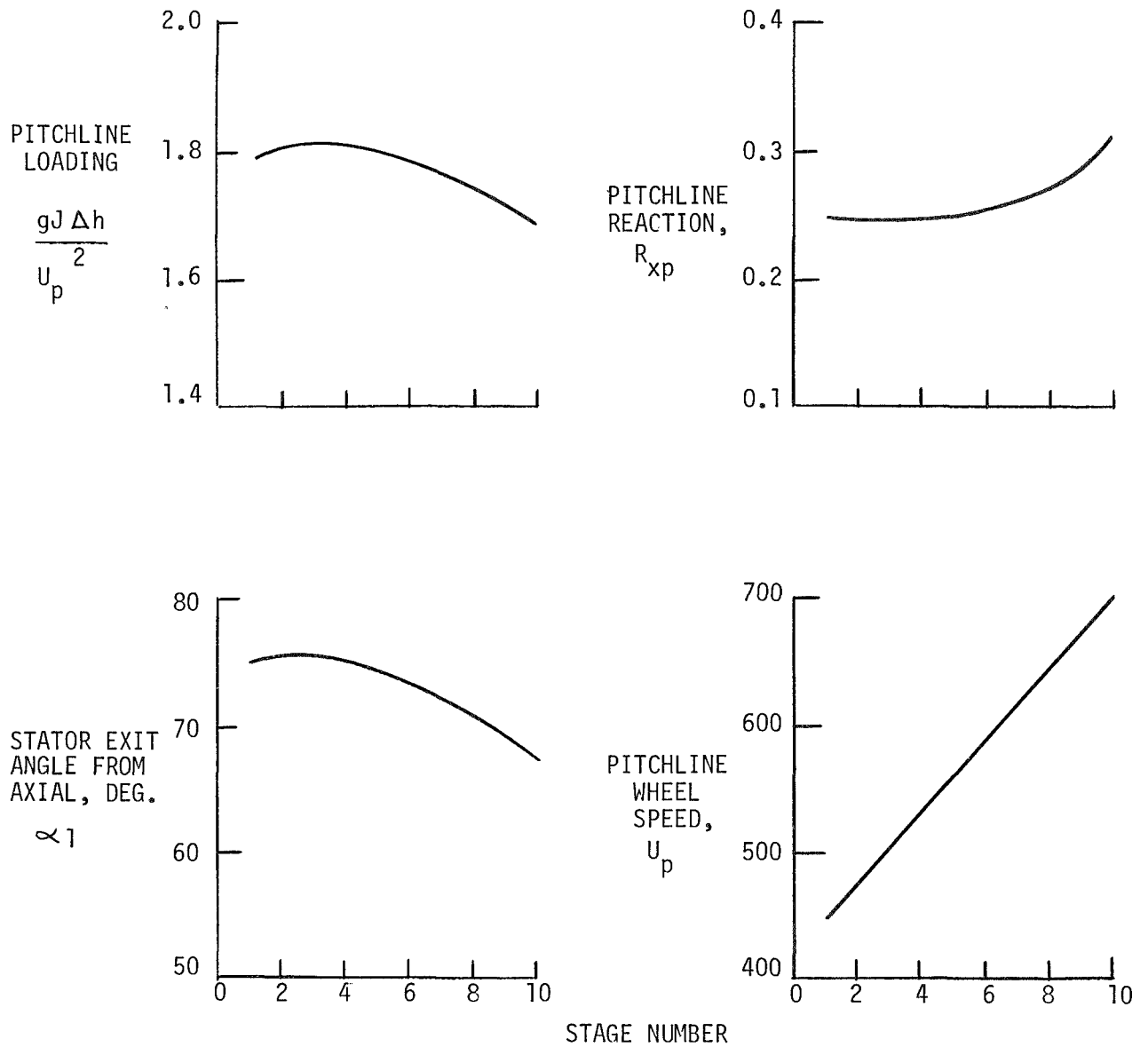


Figure 3.2-2. Typical Turbine Design - Stage Characteristics.

10 STAGE, 19,200 RPM, SINGLE SPOOL TURBINE

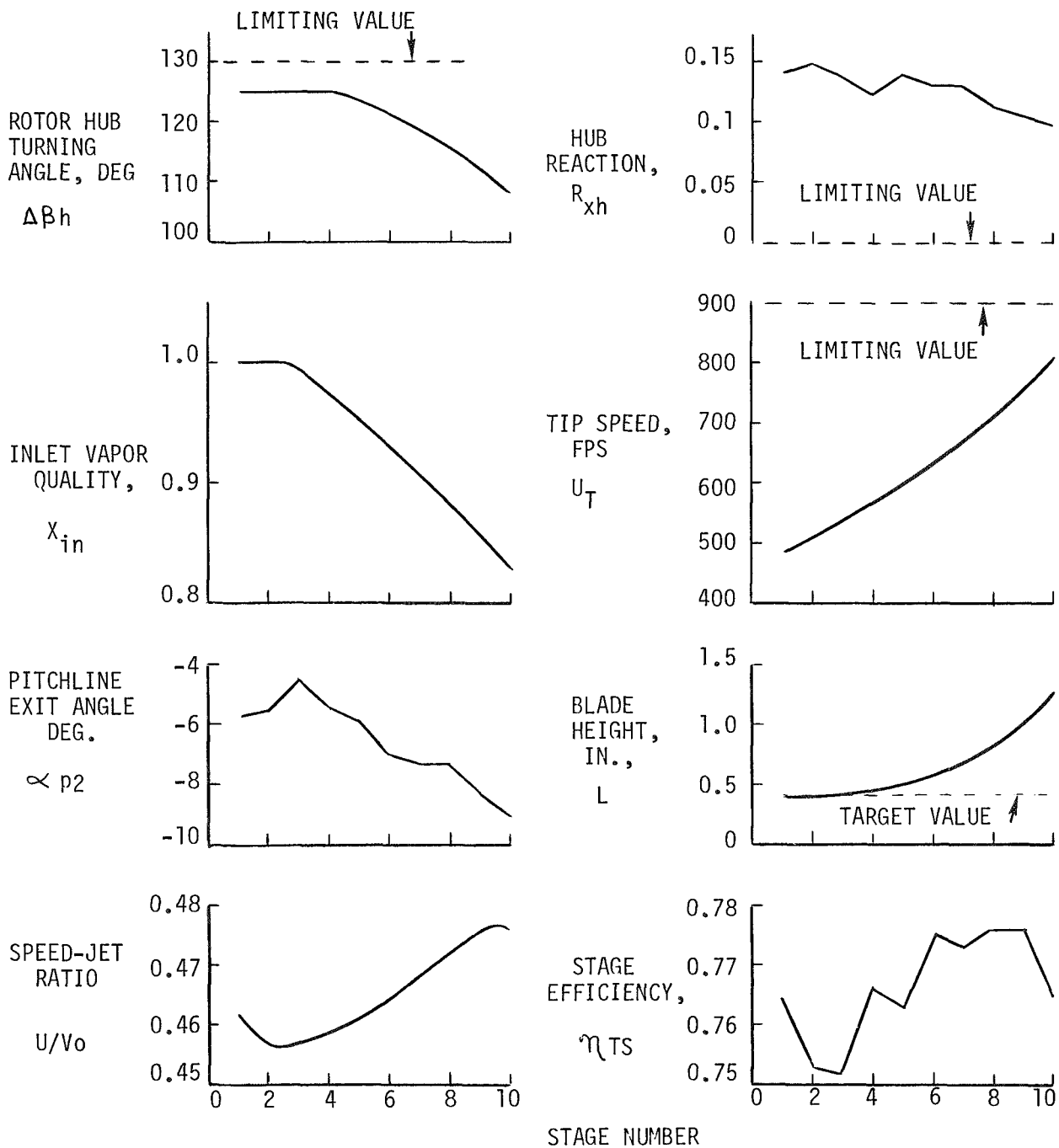
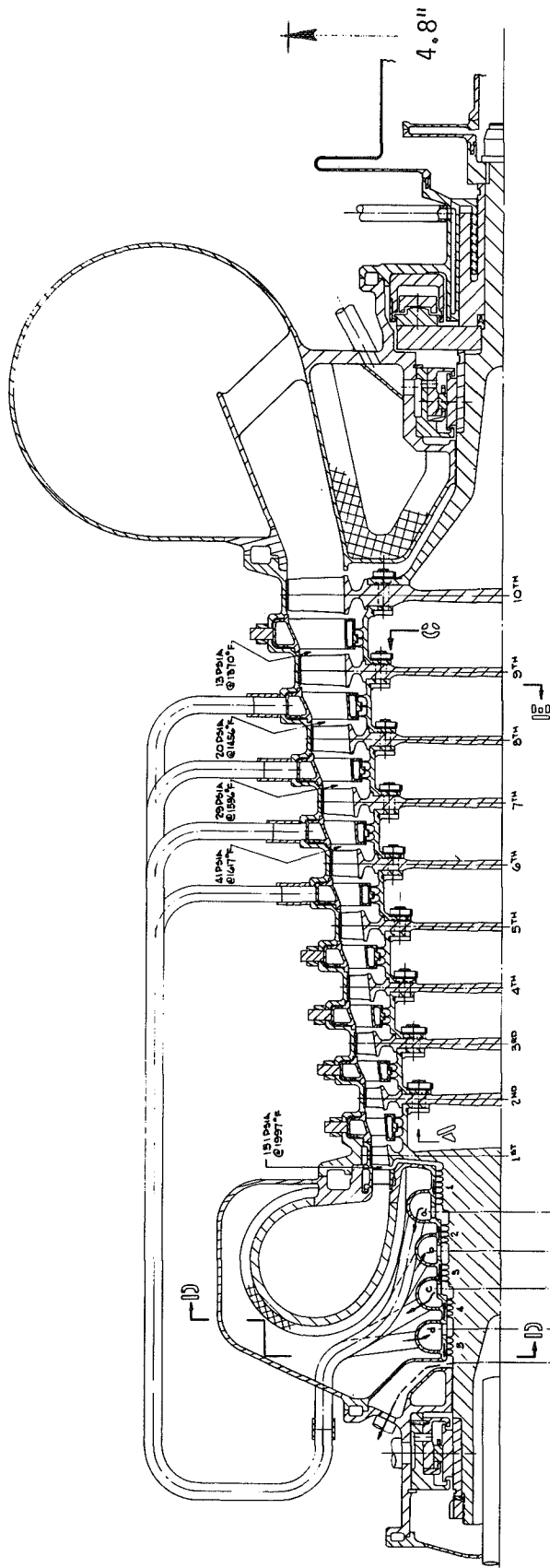


Figure 3.2-3. Typical Turbine Design - Stage Parameters.



TYPICAL TURBINE DESIGN

10-Stage, 19,200 rpm, Single-Spool

Blading Efficiency, percent	81.0
Rotor Axial Thrust, lb	1590
Bearing and Seal Losses, KW	34.3
Shaft Efficiency, percent	76.0

Figure 3.2-4. Typical Single-Spool Turbine Design.

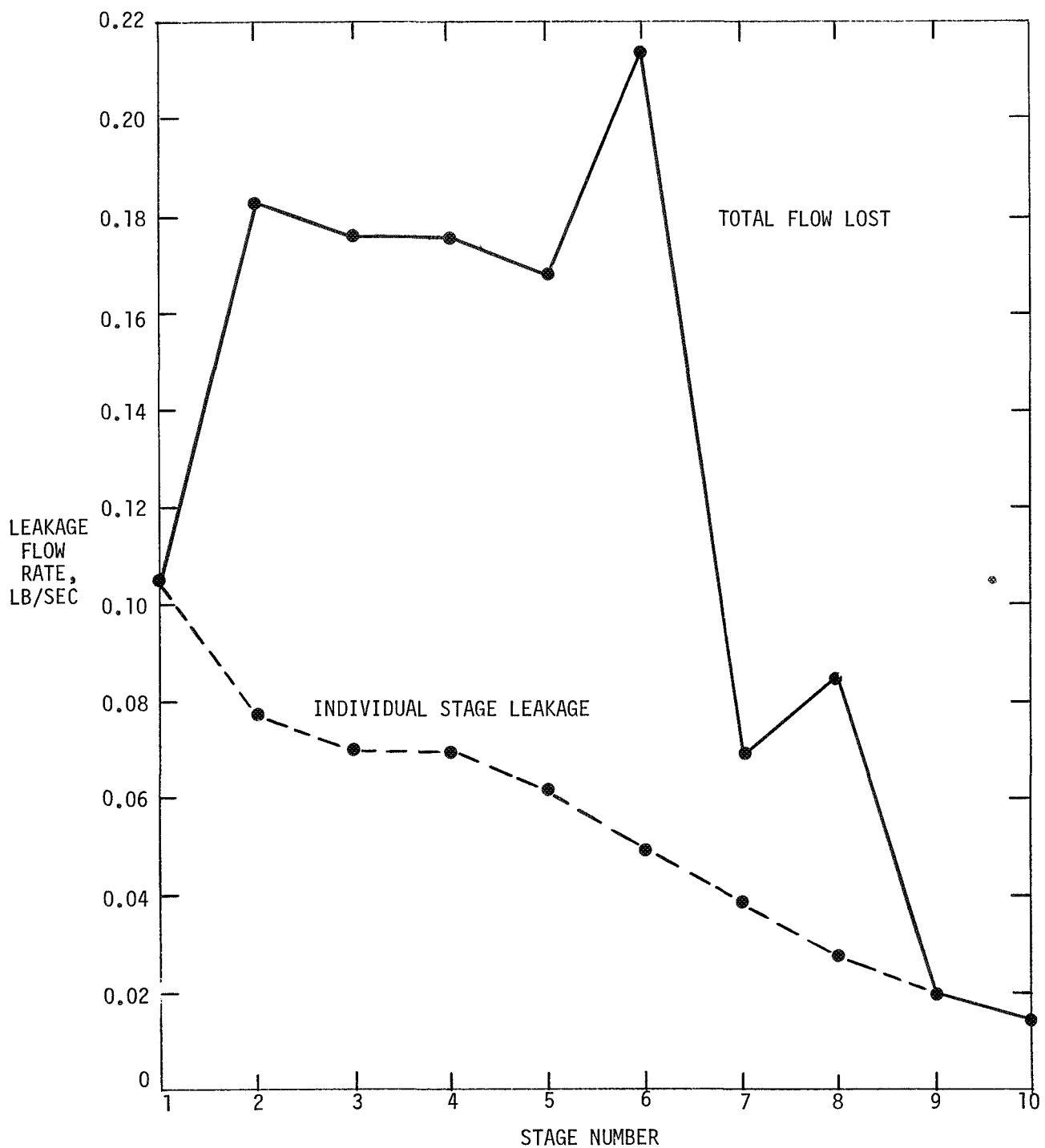


Figure 3.2-5. Typical Turbine Labyrinth Seal Leakage Flow Rate.

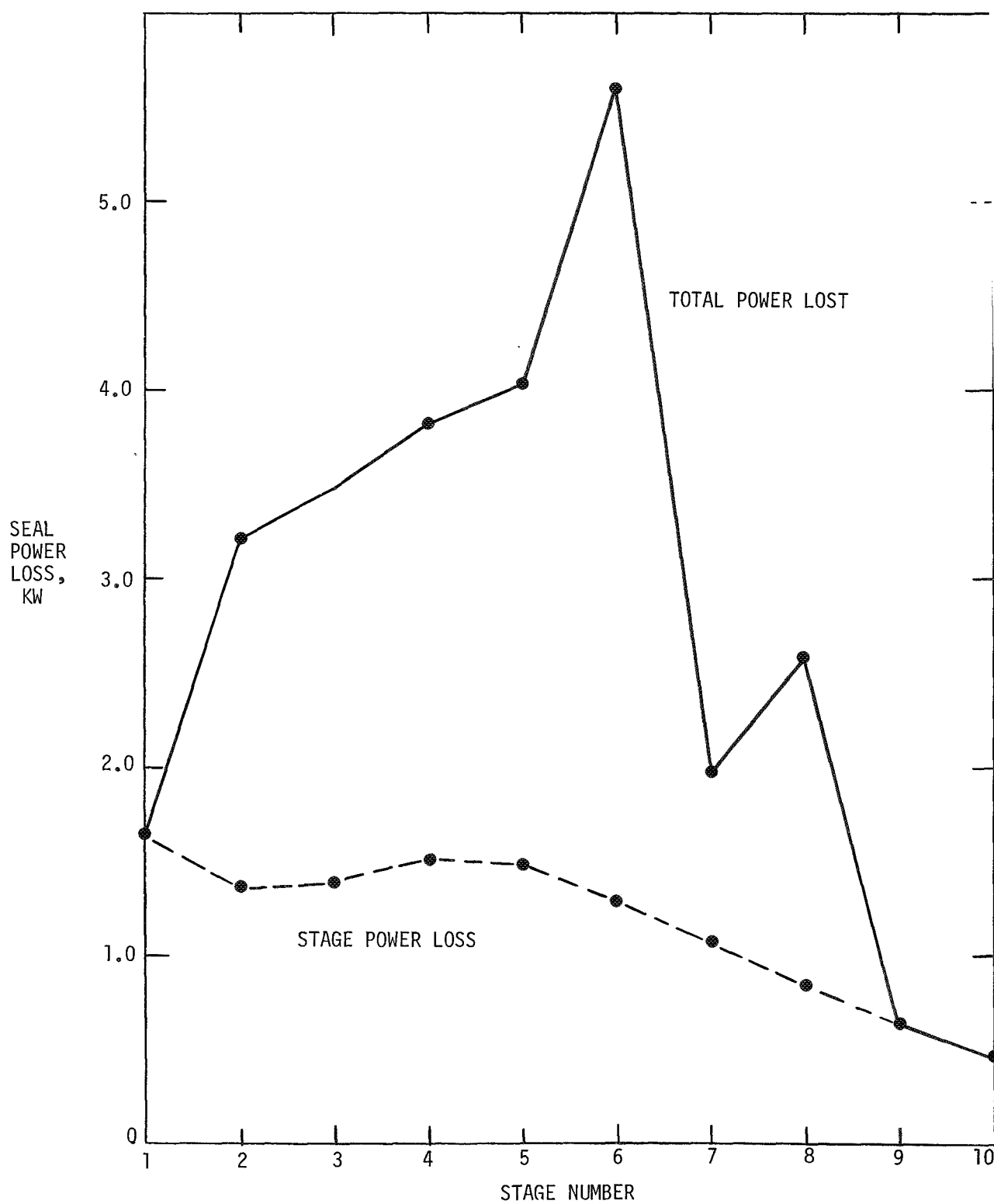


Figure 3.2-6. Typical Turbine Labyrinth Seal Leakage Power Loss.

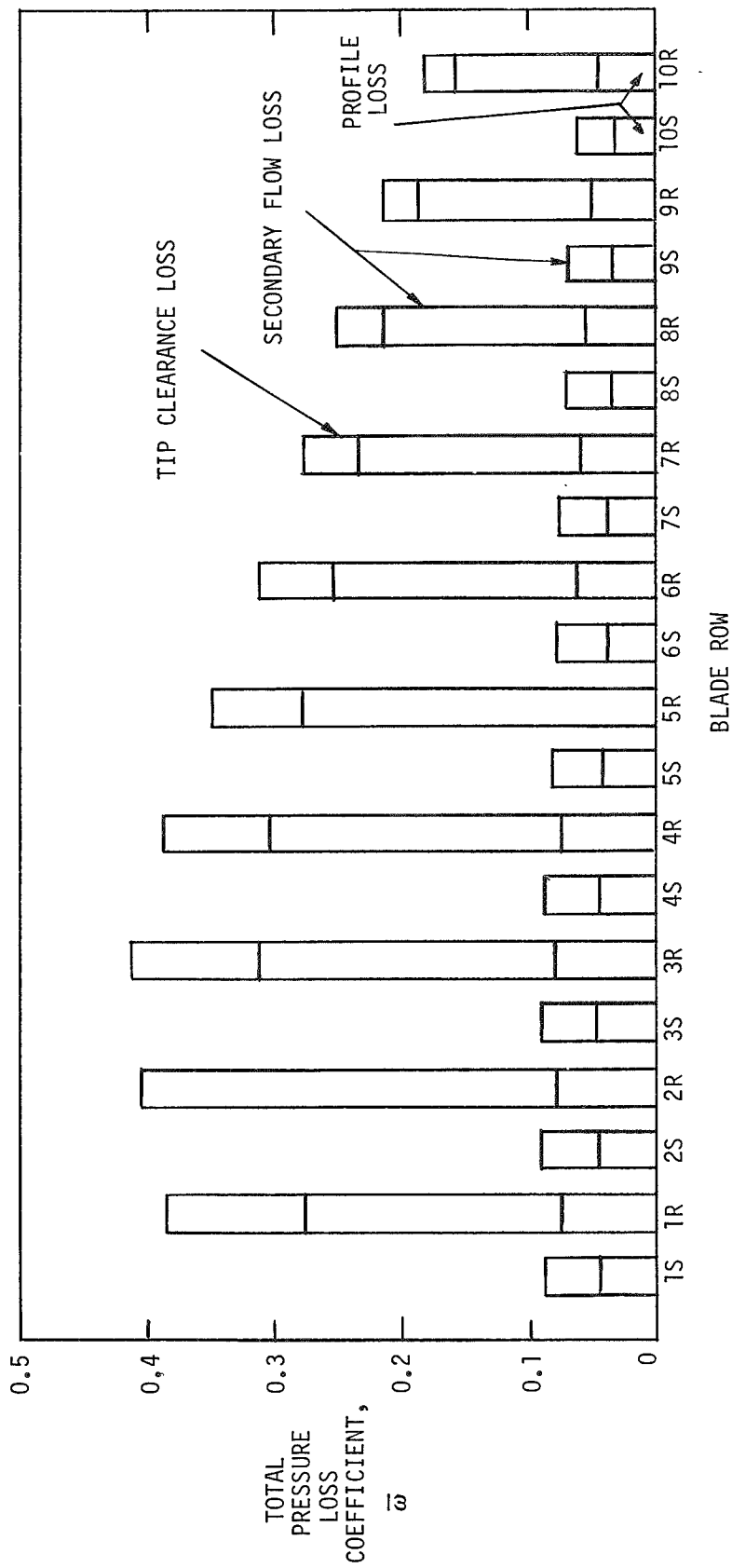


Figure 3.2-7. Typical Turbine Design Blading Losses.

TABLE 3.2-1

TYPICAL TURBINE LOSS BREAKDOWN

10-Stage, 19,200 rpm, Single Spool

IDEAL ENERGY, KW		686
BLADING LOSSES, KW	54	
DRY VAPOR OUTPUT, KW		632
<u>MOISTURE LOSSES</u>		
QUALITY LOSSES, KW	53	
DROPLET DRAG LOSSES, KW	29.5	
TURBINE BLADING OUTPUT, KW		549.5
TOTAL-TO-STATIC EFFICIENCY	0.801	
<u>PARASITIC LOSSES</u>		
LABYRINTH SEAL LEAKAGE LOSS, KW	26.0	
JOURNAL BEARING POWER LOSS, KW	1.4	
THRUST BEARING POWER LOSS, KW	6.4	
SCREW SEAL POWER LOSS, KW	0.5	
NET TURBINE POWER, KW		515.2
OVERALL EFFICIENCY	0.751	

be seen that moisture losses are very significant (82.5 KW).

The effect of tip clearance is shown in Figure 3.2-8 for shrouded and unshrouded blades. For unshrouded blades the turbine efficiency decreases 1.4 percentage points for each additional 10 mils increase in tip clearance. Because there is very little differential thermal expansion between the rotor and stator a radial clearance of 0.01-inch was determined to be feasible. For this clearance the difference in turbine efficiency is less than one point and unshrouded blades were selected based on mechanical design considerations.

In contrast to the typical design at 19,200 rpm, two designs at the extreme ends of the speed range are interesting. Shown in Figure 3.2-9 is a six stage design at 32,000 rpm. This design is attractive because of its compactness, but the higher wheel speeds (650 to 750 ft/sec) permitted by larger blade height increases the stresses as can be seen by the wide disks. The turbine stage parameters are shown in Figures 3.2-10 and 3.2-11. The blading efficiency is 77.2 percent. This is a highly loaded turbine (pitchline loading, 1.94) to minimize the number of stages. At the other extreme, a twelve stage turbine at 16,000 rpm is shown in Figure 3.2-12. This is a conservative design (pitchline speed range 400 to 670 ft/sec) and the lower stresses are reflected in the thin disks. The stage parameters are shown in Figures 3.2-13 and 3.2-14. The blading efficiency is 81.8 percent, resulting from an average pitchline loading of 1.69. The only really significant way to improve turbine efficiency is to reduce stage loading. This idea was applied to the nine-stage 19,200 rpm design by increasing the wheel speeds to the range of 600 to 740 ft/sec. This resulted in lightly loaded stages (with an average pitchline loading of

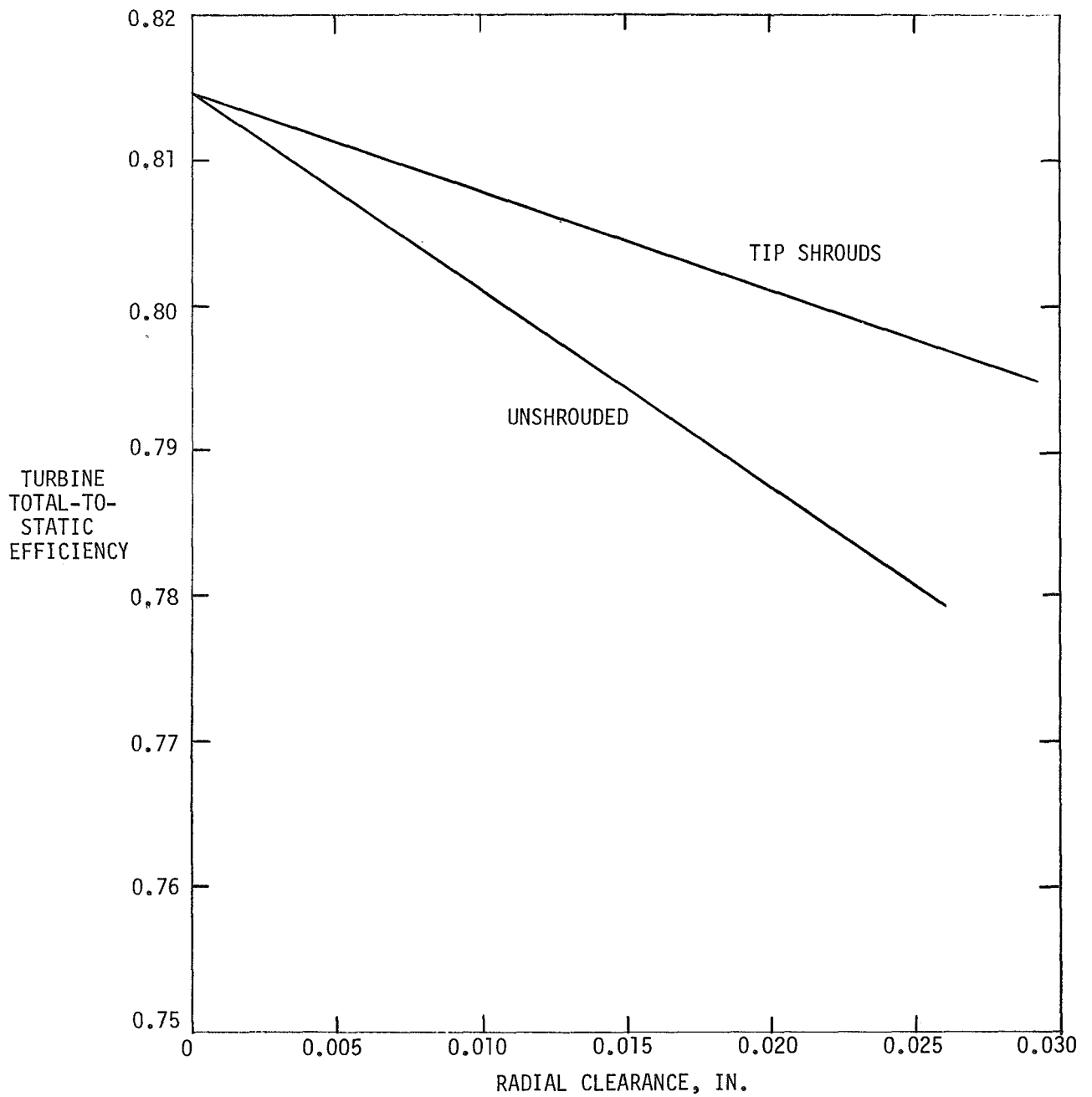
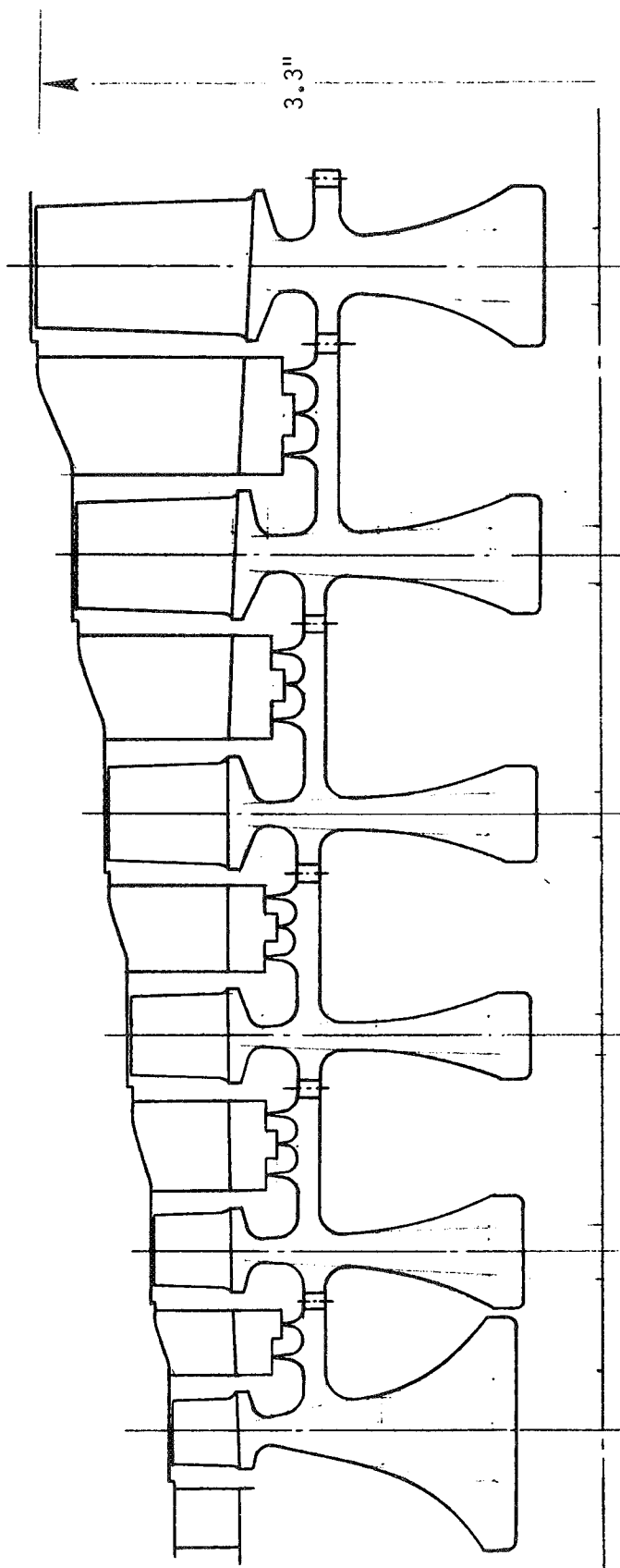


Figure 3.2-8. Typical Effect of Tip Clearance on Efficiency.



SINGLE-SPOOL, SINGLE-FLOW TURBINE DESIGN

6-Stage, 32,000 rpm

Blading Efficiency, percent 77.2

Figure 3.2-9. Single-Spool, Single-Flow Turbine Design.

6 STAGE, 32,000 RPM, SINGLE SPOOL, SINGLE FLOW DESIGN

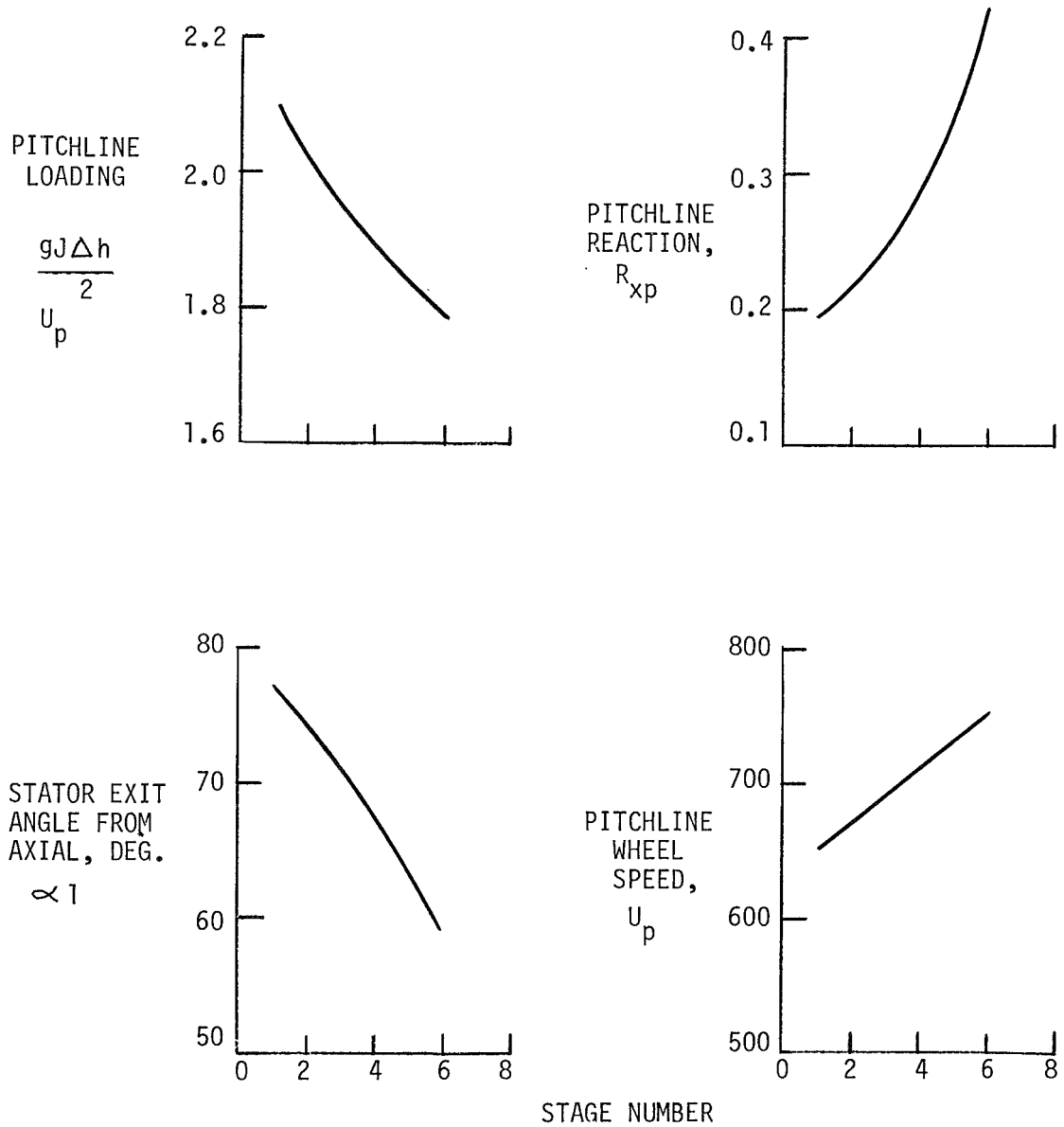


Figure 3.2-10. Turbine Design - Stage Characteristics.

6 STAGE, 32,000 RPM, SINGLE SPOOL, SINGLE FLOW DESIGN

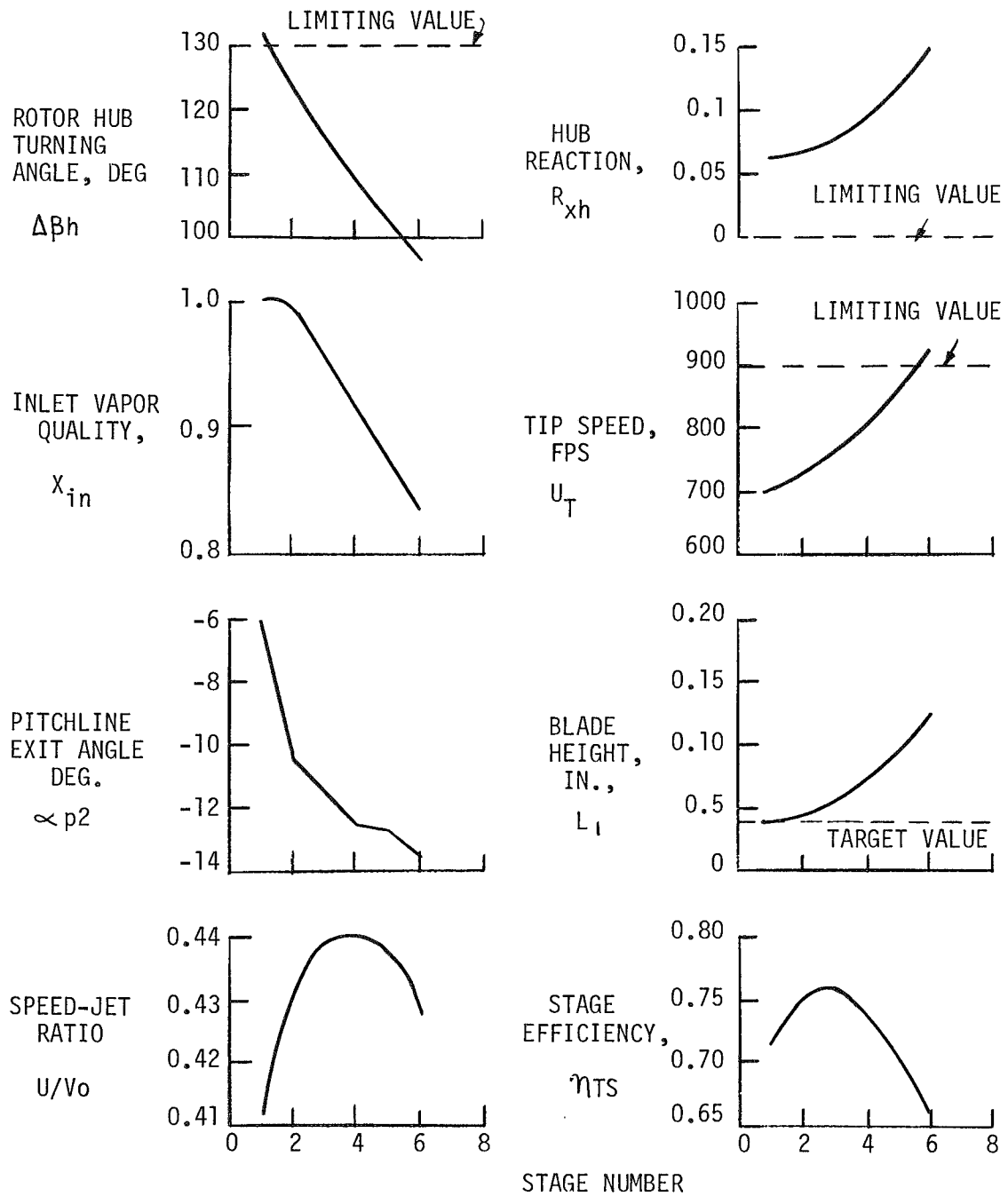
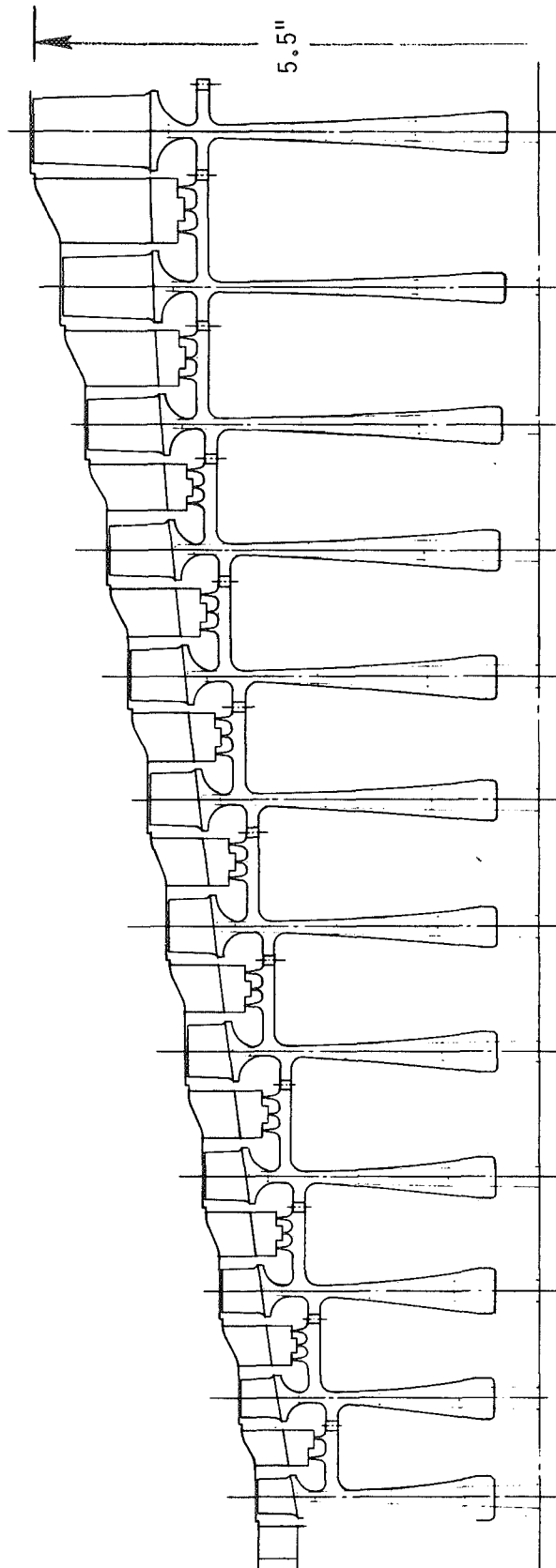


Figure 3.2-11. Turbine Design - Stage Parameters.



SINGLE-SPOOL, SINGLE-FLOW TURBINE DESIGN

12-Stage, 16,000 rpm

Blading Efficiency, percent 81.8

Figure 3.2-12. Single-Spool, Single-Flow Turbine Design.

12 STAGE, 16,000 RPM, SINGLE SPOOL, SINGLE FLOW DESIGN

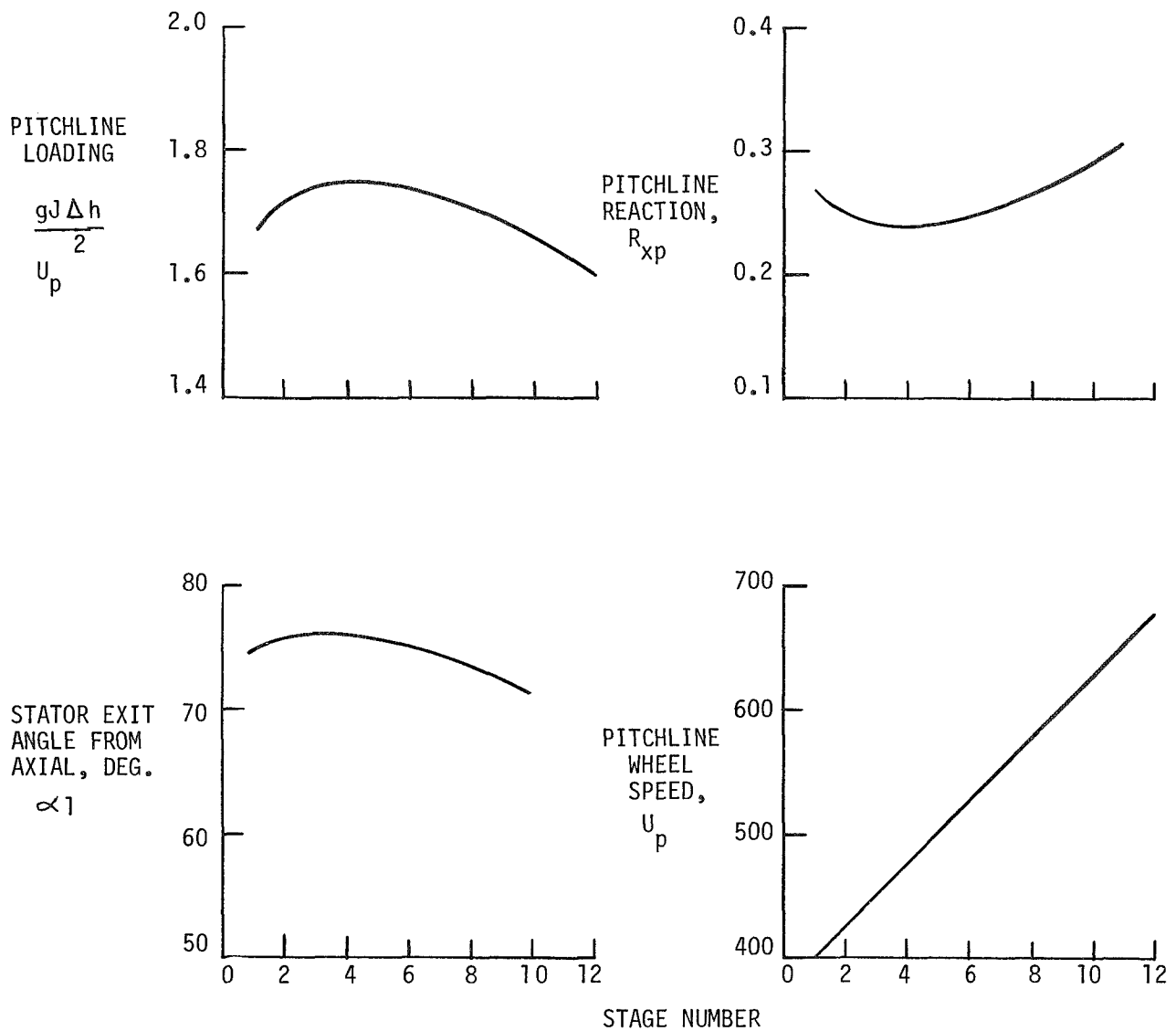


Figure 3.2-13. Turbine Design - Stage Characteristics.

12 STAGE, 16,000 RPM, SINGLE SPOOL, SINGLE FLOW DESIGN

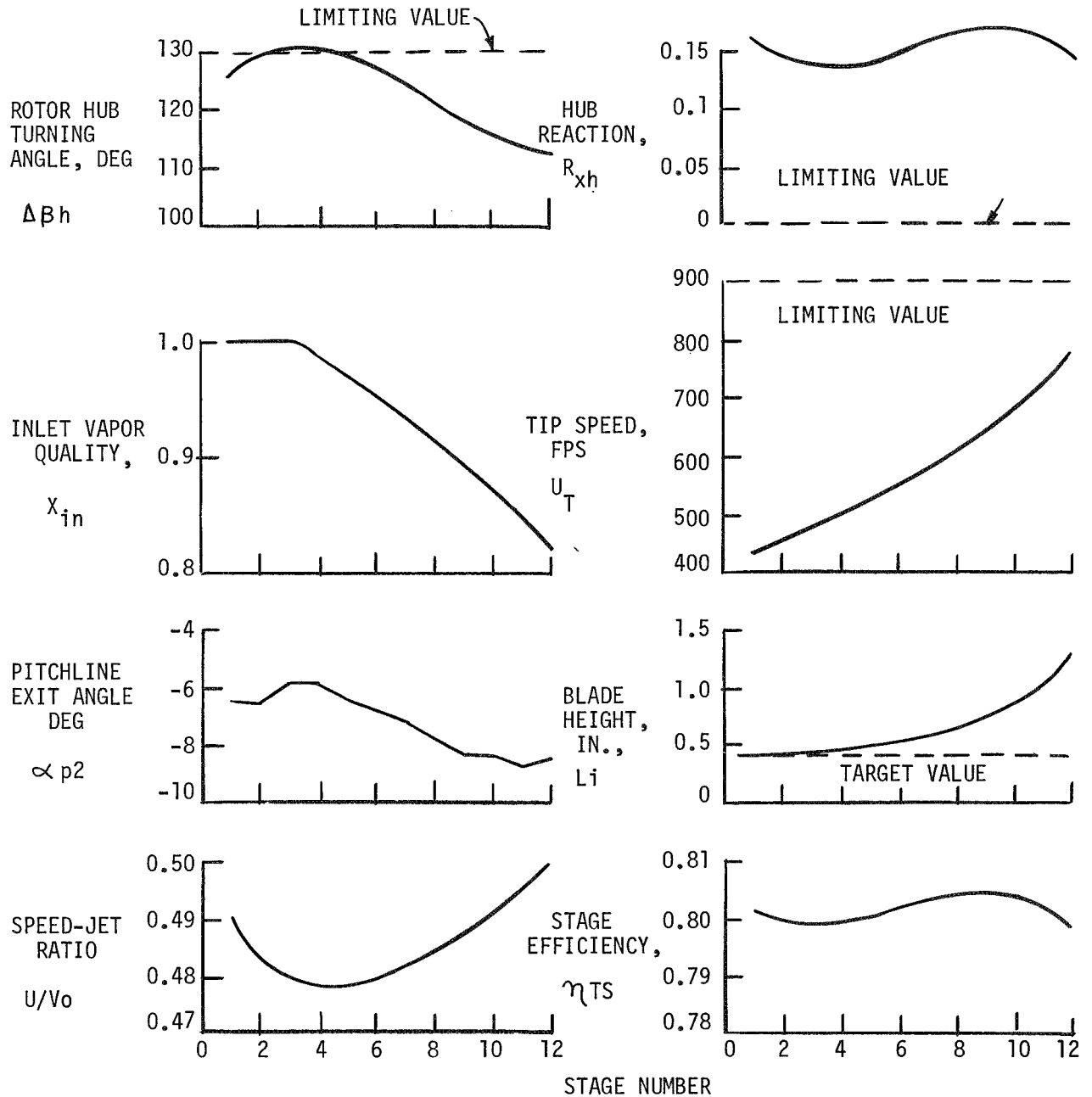


Figure 3.2-14. Turbine Design - Stage Parameters.

1.52) as shown in Figure 3.2-15, with the other stage parameters shown in Figure 3.2-16. Although it was possible to increase aerodynamic efficiency about 5 percent in this way to an efficiency of 83 percent, the blade heights were quite small and the higher wheel speeds increased the stresses considerably. A drawing of this special design is shown in Figure 3.2-17.

9 STAGE, 19,200 RPM, HIGH-EFFICIENCY TURBINE DESIGN

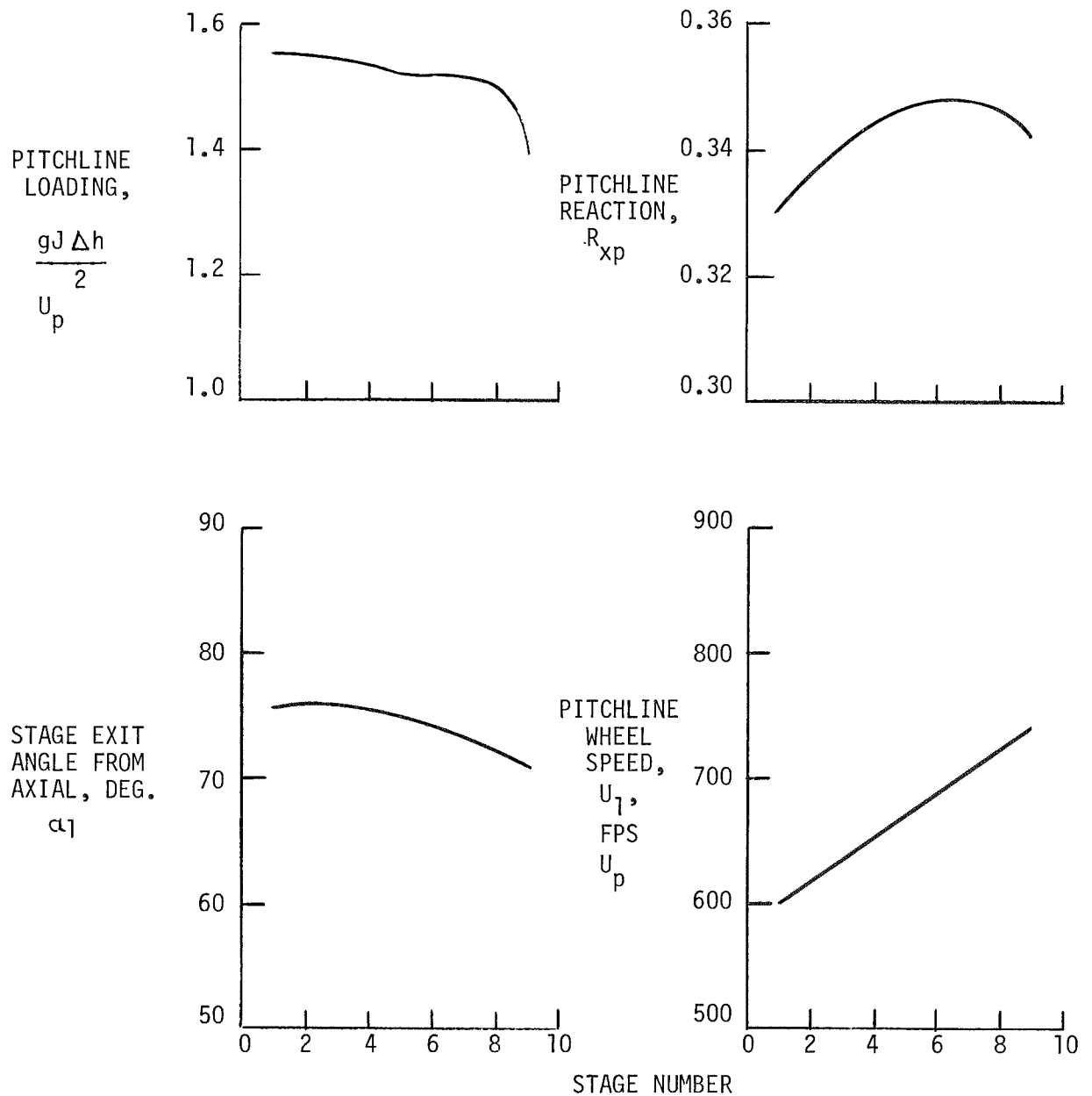


Figure 3.2-15. Stage Characteristics.

9 STAGE, 19,200 RPM, HIGH-EFFICIENCY TURBINE DESIGN

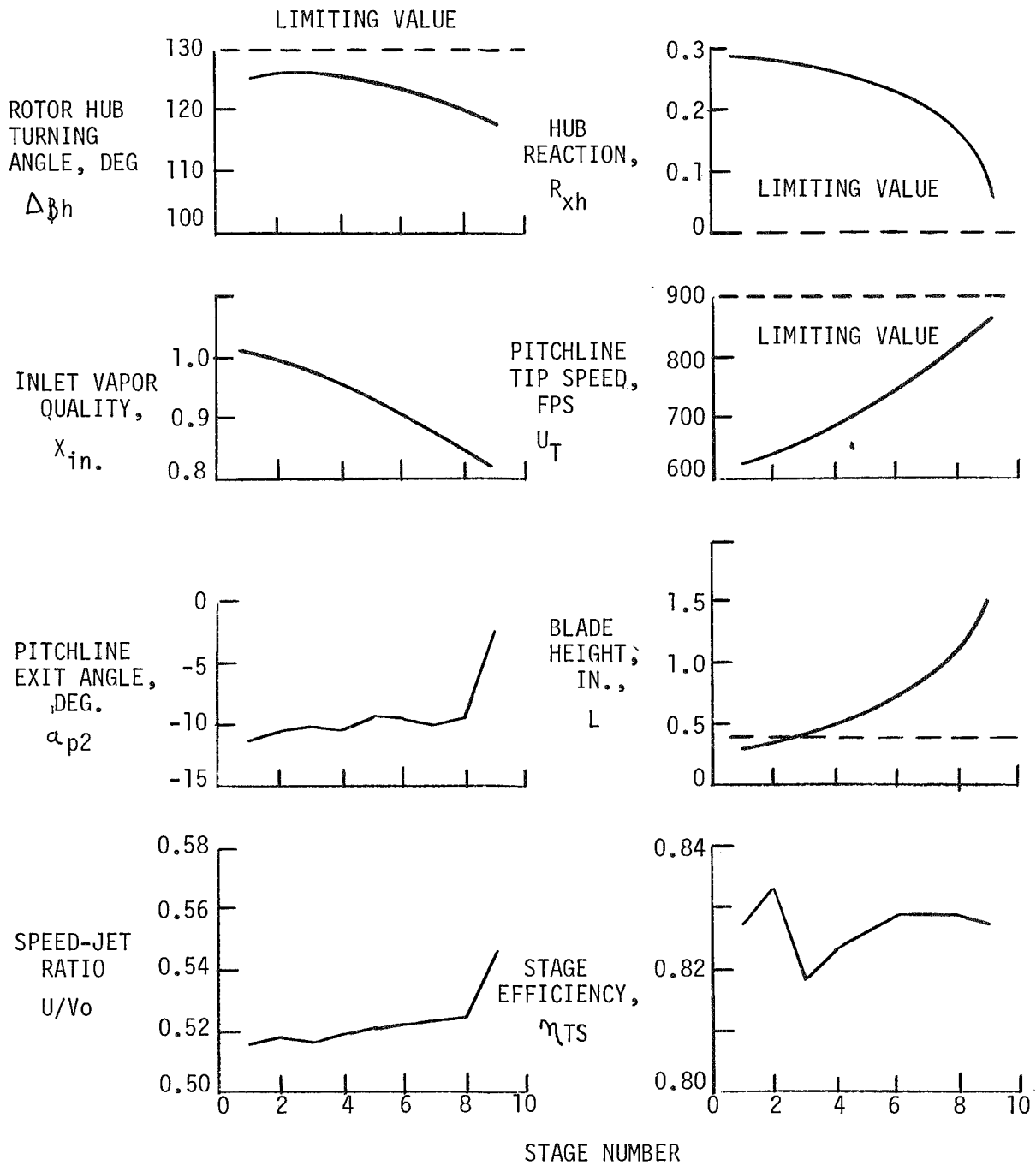
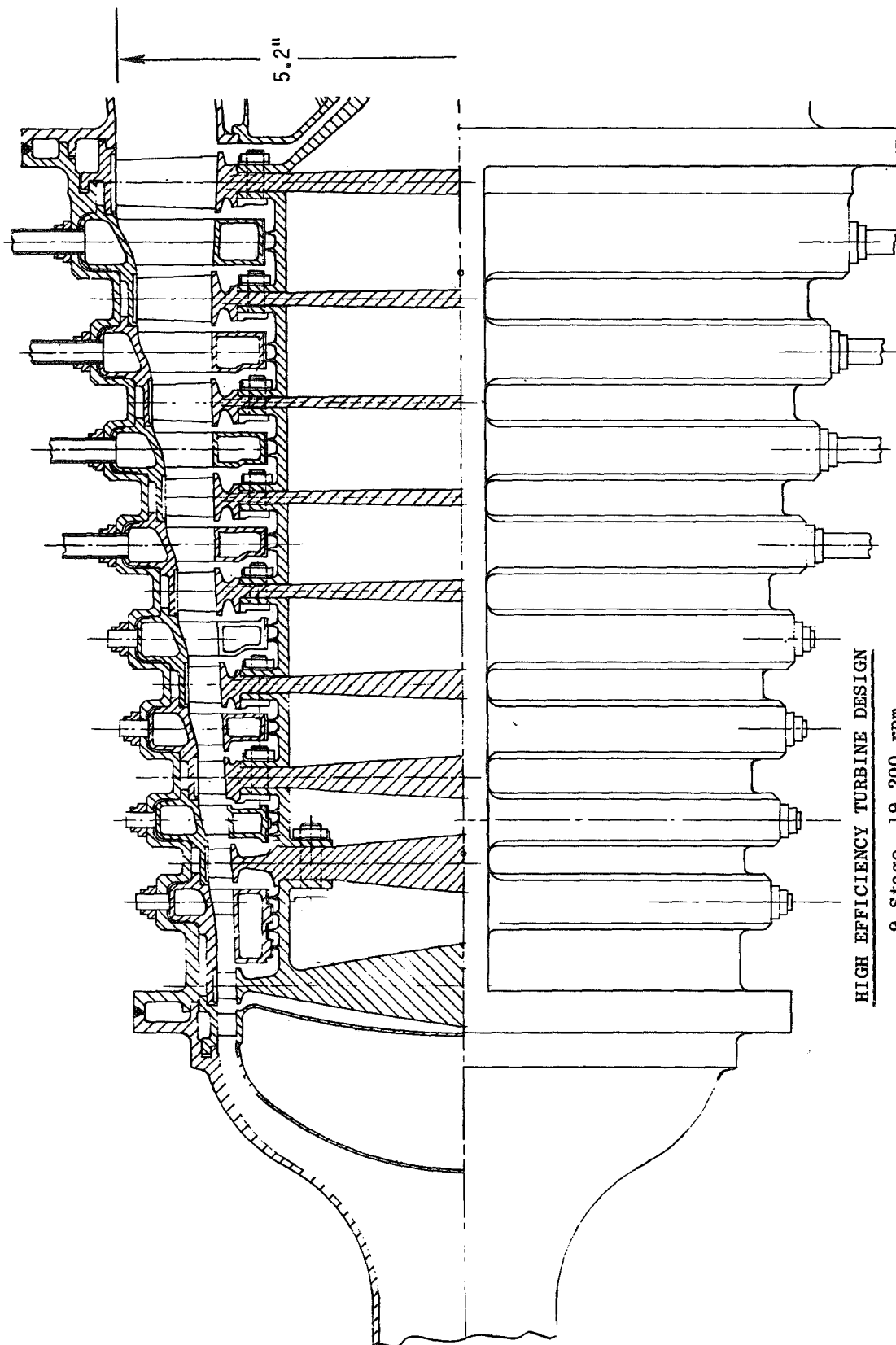


Figure 3.2-16. Stage Parameters.



Blading Efficiency, percent 83.0

Figure 3.2-17. High Efficiency Turbine Design.

4.0 TURBINE MATERIALS

4.1 MATERIALS SELECTION

During Phase I, both weldable and nonweldable alloys were considered for the rotating turbine components which were required to carry high stresses. While the nonweldable alloys had generally higher rupture and creep strength-to-density ratios, the advantages of the weldable tantalum-base alloys dictated their consideration. These advantages were the ability to weld assemble complex rotor structures, avoidance of either bolting or the use of curvic couplings, and the absence of the low-ductility characteristics associated with the molybdenum-base alloys at low temperature. These advantages were not sufficient to offset (1) the better strength characteristics available in the molybdenum-base alloys and (2) the teardown advantages of a mechanically joined molybdenum-base alloy rotor. These latter advantages were evident in studies of the feasibility of a tie-bolted, curvic-coupled rotor.

The readily weldable T-111 and Cb-1Zr alloys, which have received, and which will continue to receive, so much use in experimental alkali metal containment applications, were selected as ducting alloys for the containment of potassium liquid and vapor and for structural applications. For strength reasons, T-111 was used at the higher temperatures and Cb-1Zr was used at lower temperatures where its properties permitted its application. It was decided in Phase I to use T-111 in the nozzles,

casing, and structure in the third stage and beyond.

The chemical composition, density, and thermal expansion of each of the alloys considered are given in Table 4.1-1. The strength-to-density characteristics of the alloys are compared in Figure 4.1-1 at the 1 percent creep level and in Figure 4.1-2 at rupture. In Figure 4.1-1, the curves for the molybdenum alloys were produced from 0.5 percent creep instead of 1 percent creep; thus, these molybdenum alloy creep strength-to-density curves are slightly lower than they should be with respect to the tantalum and columbium alloy data and, therefore, are conservative. The obvious strength advantages of the molybdenum-base alloys over the tantalum-base alloys are clearly shown in these figures, both at the 1 percent creep level and at rupture. Note that, in the absence of rupture data, the rupture properties of the ASTAR 811C alloy and of the molybdenum alloys KDTZM-1175 and TZC have been estimated from the 1.0 percent and 0.5 percent creep data available for these alloys using constant conversion factors over the entire parameter range. The factors are based on observed conversion factors taken from T-111 data in the case of the ASTAR 811C data and taken from TZM in the case of the KDTZM-1175 data. The application of such "flat" factors across the parameter range must introduce errors in the estimate and this possibility must be recognized. These curves should be recognized as being based on limited and extrapolated data under less than optimized conversion assumptions. These strength/density curves are intended to show only the relative properties of the various alloys in those rotating machinery applications where the stresses developed are a function of material density.

TABLE 4.1-1

TURBINE MATERIALS SELECTION

Alloy Designation	Base Element	Composition - %								Density lb/in ³	Thermal Expansion Coefficient, 10 ⁻⁶ per °F (70 - 2000°F)	
		Ta	Mo	Cb	W	Hf	Re	Zr	Ti			C
Rotors, Shafts & High-Stressed Parts:												
TZC (nonweldable)	Mo	-	Bal	-	-	-	-	0.15	-	0.12	0.367	3.3
TZM (nonweldable)	Mo	-	Bal	-	-	-	-	0.08	0.5	-	0.367	3.3
KDTZM-1175 (nonweldable)	Mo	-	Bal	-	-	-	-	0.12	0.61	0.035	0.367	3.3
ASTAR 811C (weldable)	Ta	Bal	-	-	8	0.7	1.56	-	-	0.025	0.6	3.9
FS-85 (weldable)	Cb	28	-	Bal	10	-	-	1.0	-	-	0.383	4.8
Cb-752 (weldable)	Cb	-	-	Bal	10	-	-	2.5	-	-	0.326	4.3*
Stator, Structural & Containment:												
T-111	Ta	Bal	-	-	8	2	-	-	-	-	0.604	3.9
Cb-1Zr	Cb	-	-	Bal	-	-	-	1.0	-	-	0.31	4.51

* Thermal Expansion for D-43 (10W-1Zr-Bal Cb) from 1000 -2500°F

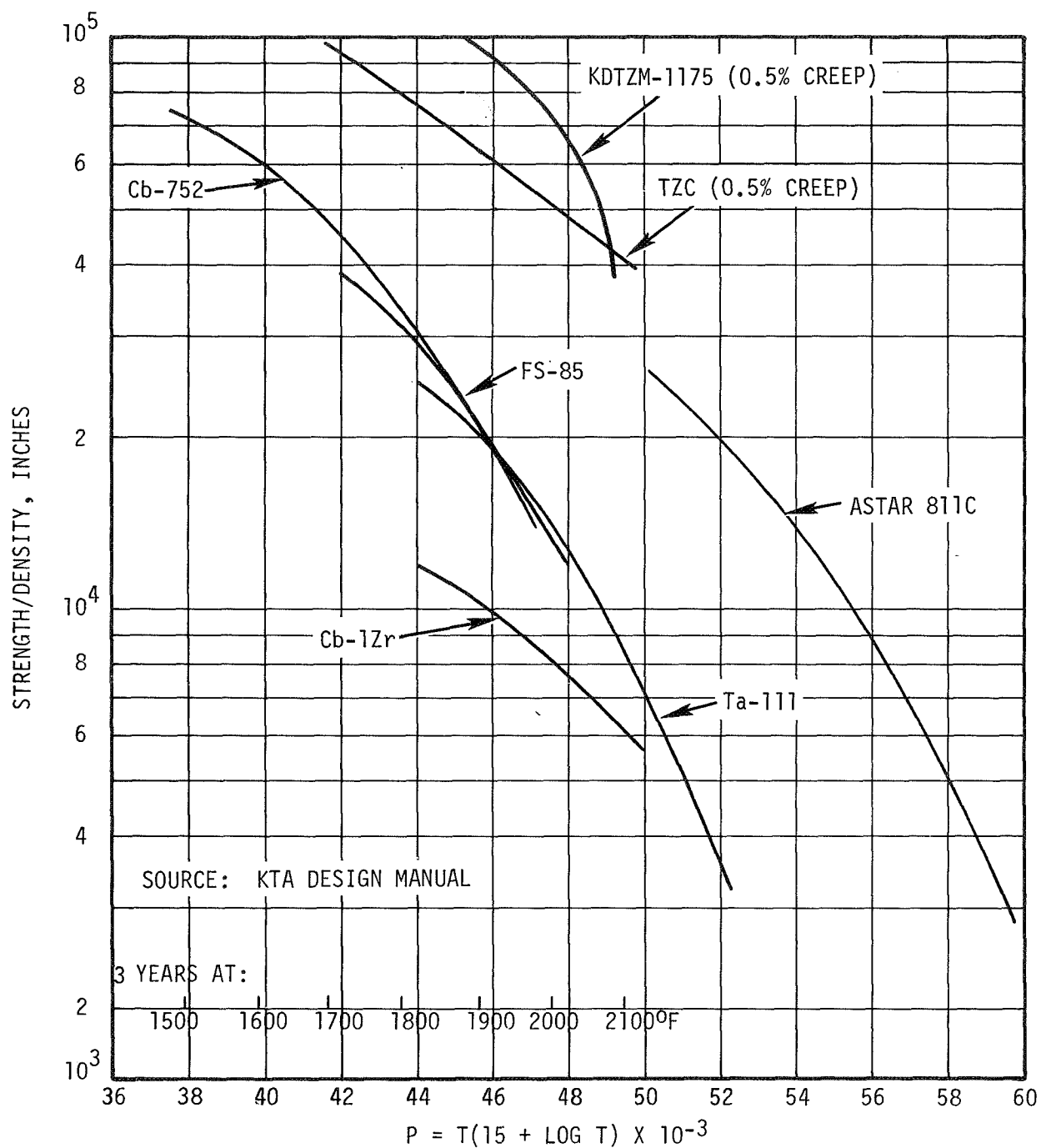


Figure 4.1-1. 1% Creep Strength/Density of Refractory Alloys.

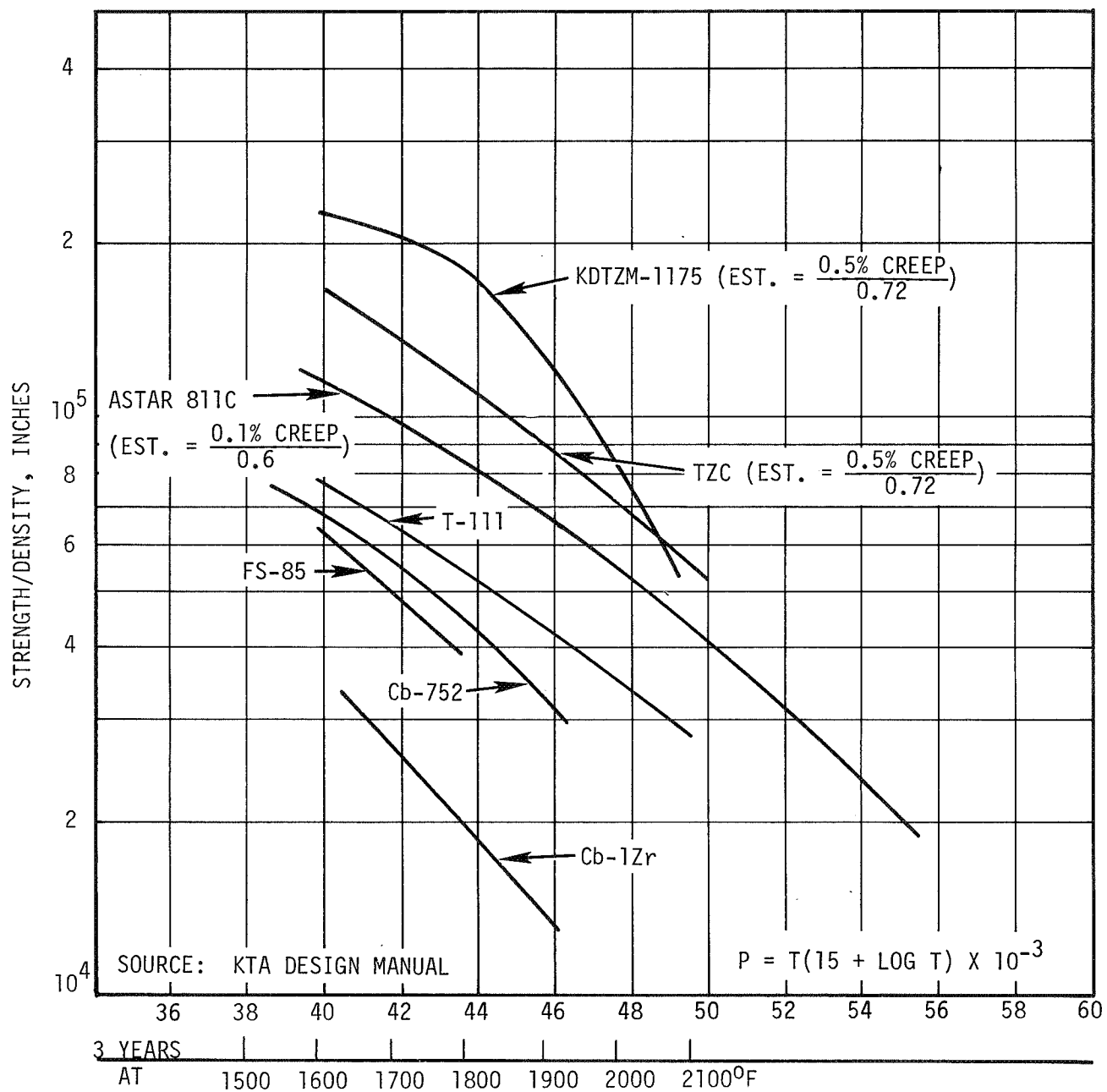


Figure 4.1-2. Rupture Strength/Density of Refractory Alloys.

Design data curves of the types shown in Figures 4.1-3 through 4.1-9 were prepared from various published sources including, principally, the NASA sponsored creep test program, Reference 4.1-1, and from some direct transfer of data from NASA or its subcontractors. In addition to the design data curves included herein, a complete set of design manual information on materials was prepared and sent to those principals within the contracted study program who had a need for such information; these involved persons within the sponsoring agency and at subcontractors' locations.

The above design data curves formed the principal basis for design study calculations and for the relative rating of various designs from different materials. The curves included strength data as a function of temperature, for ultimate and yield strength, and for the three-year life strengths at 0.5 or 1 percent and at rupture. In addition, (1) the transition between the elastic property limiting stresses and the creep or rupture limiting stresses are given for steady operating speed and for 120 percent overspeed conditions and (2) the three-year design stress curves based on creep and rupture are shown with their appropriate allowances from original strength levels for materials deviation and safety factor. For the refractory alloys the transition from elastic-limiting to creep-limiting properties appears to occur in the 1400-1600°F temperature range. Under overspeed conditions, the material must remain below its yield strength design stress limit; the requirement that the higher stress at 120 percent be within allowable stress requires that the wheel (or rotating component) must be designed to operate at a much lower stress at normal speeds so as not to exceed the design limit at the overspeed condition. The overspeed condition imposes the most severe elastic property limit on the component design at the lower

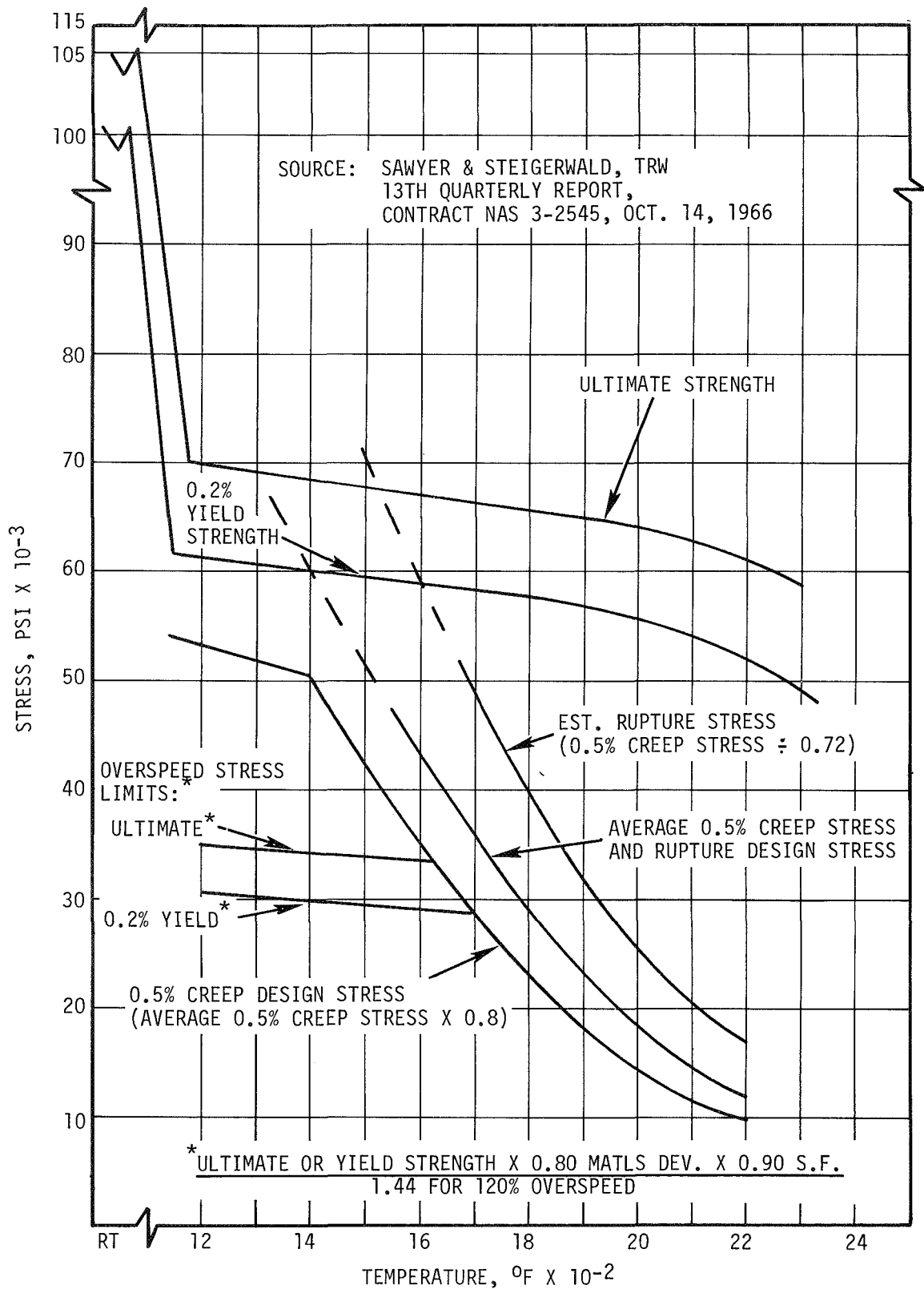


Figure 4.1-3. 3-Year Rupture, 3-Year Creep Strength, Tensile Strength for TZC.

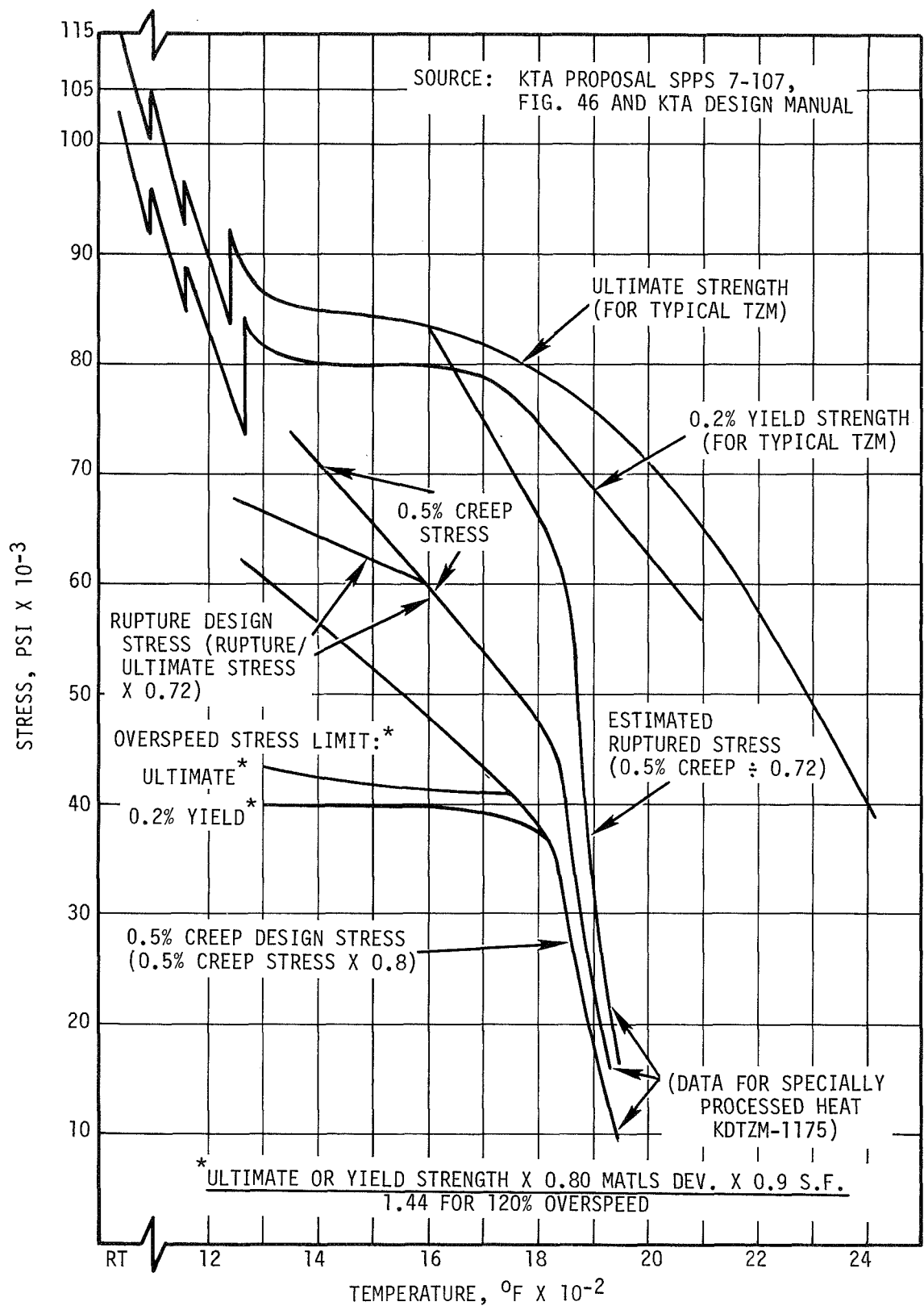


Figure 4.1-4. 3-Year Rupture and 3-Year Creep Strength for KDTZM-1175.

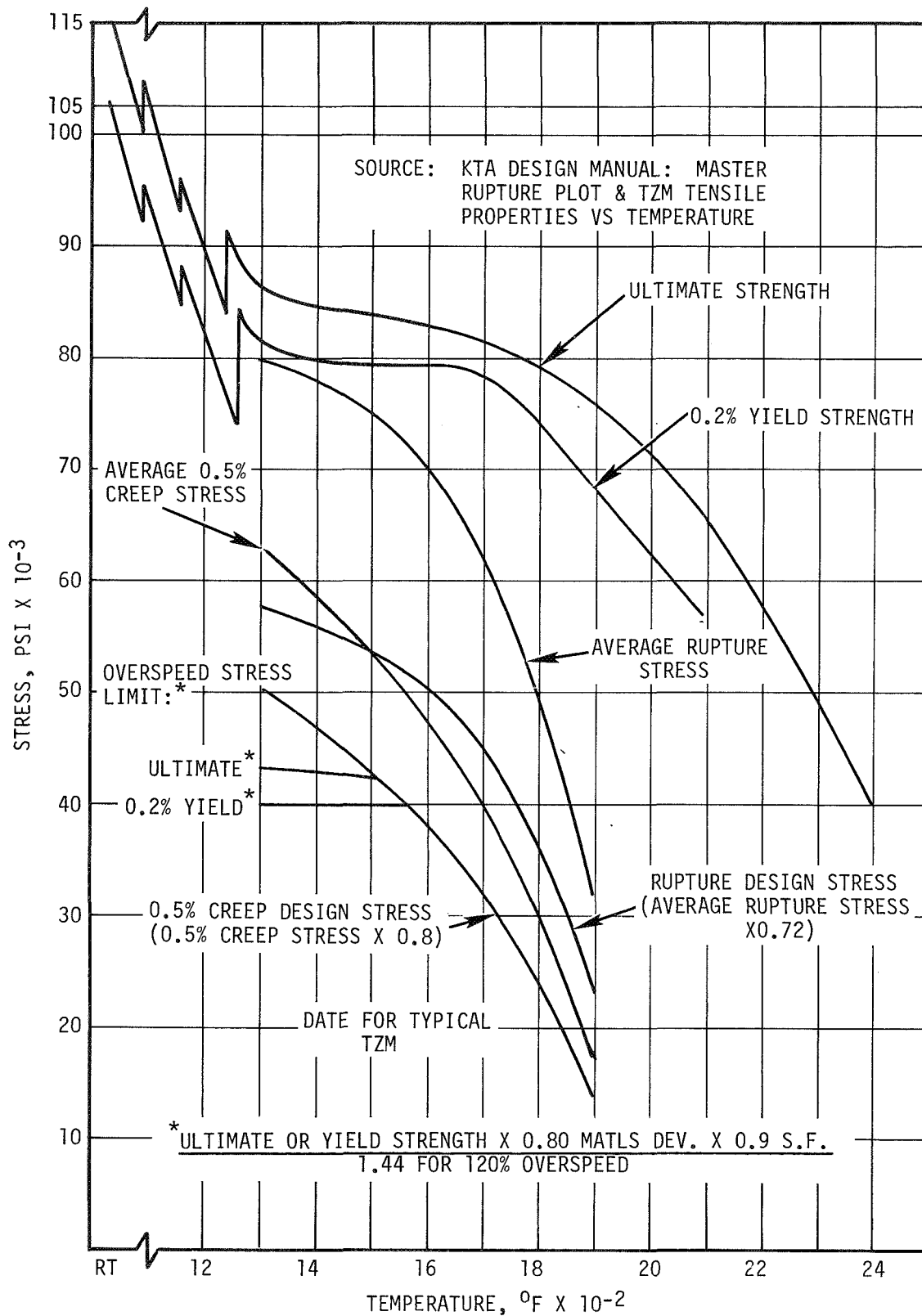


Figure 4.1-5. 3-Year Rupture, 3-Year Creep Strength, and Tensile Strength for TZM

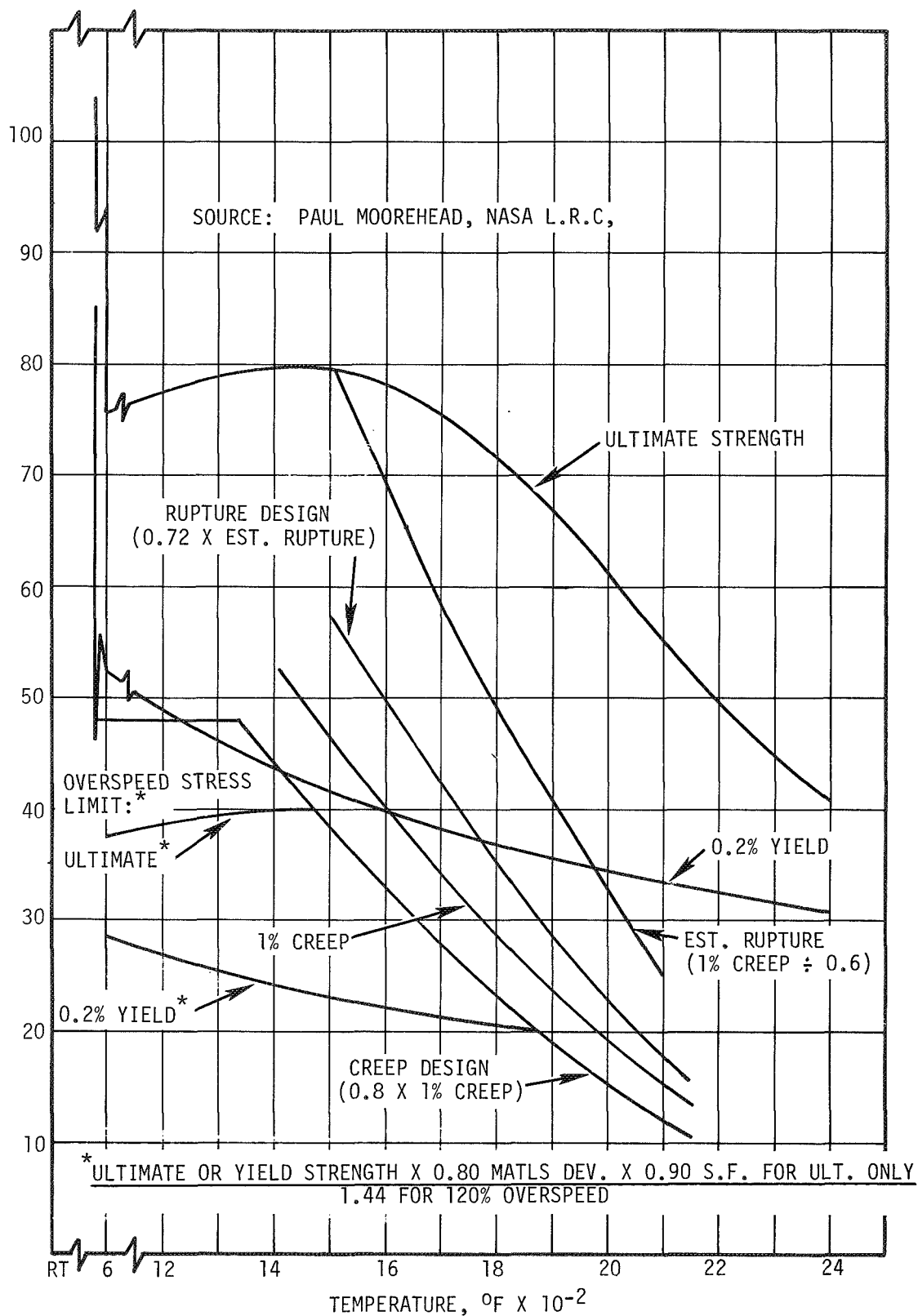


Figure 4.1-6. 3-Year Rupture, 3-Year Creep, and Tensile Strength for ASTAR 811C. (Annealed 1 Hour 3000 $^{\circ}\text{F}$)

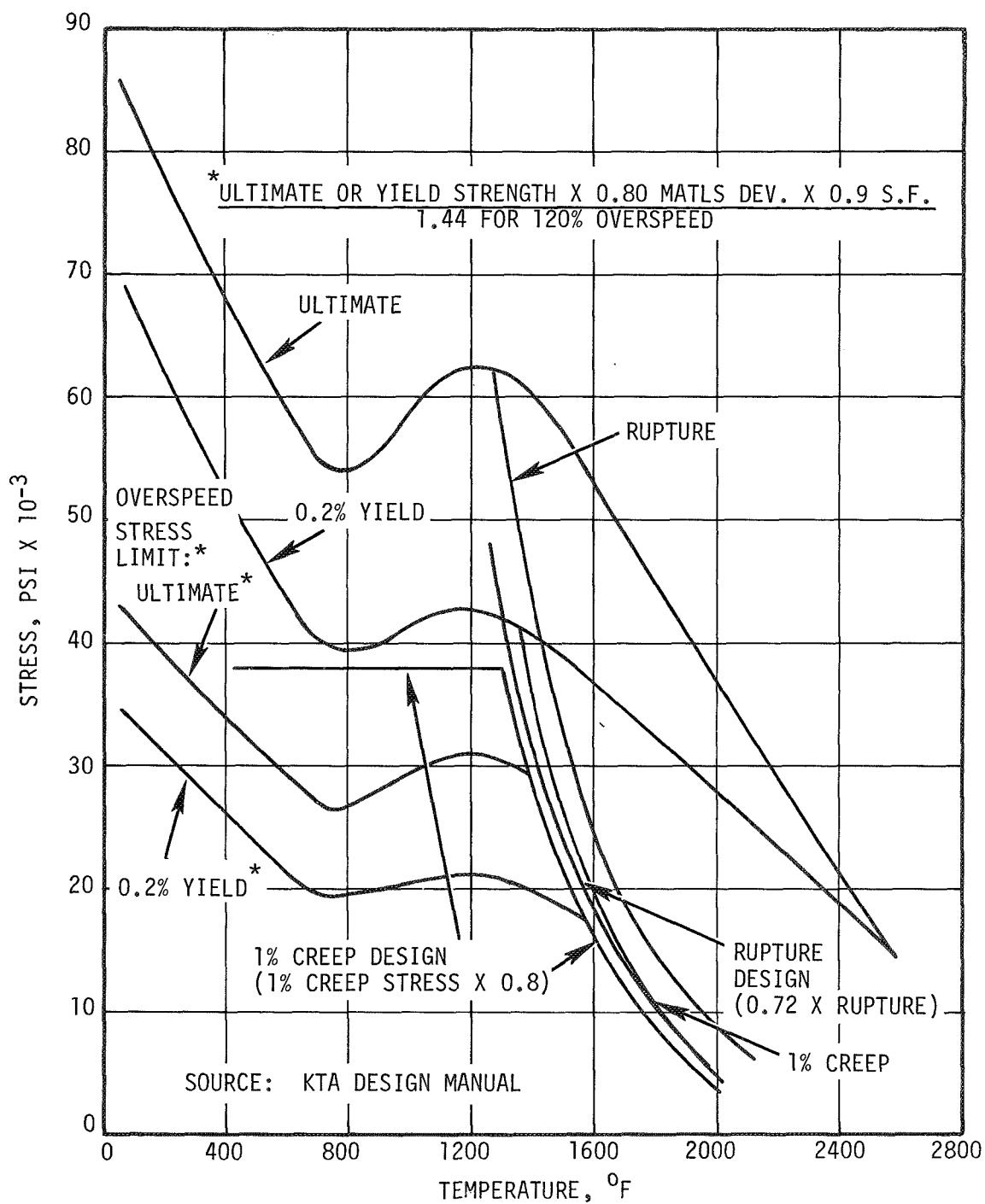


Figure 4.1-7. 3-Year Rupture, 3-Year Creep, and Tensile Strength for FS-85. (Recrystallized)

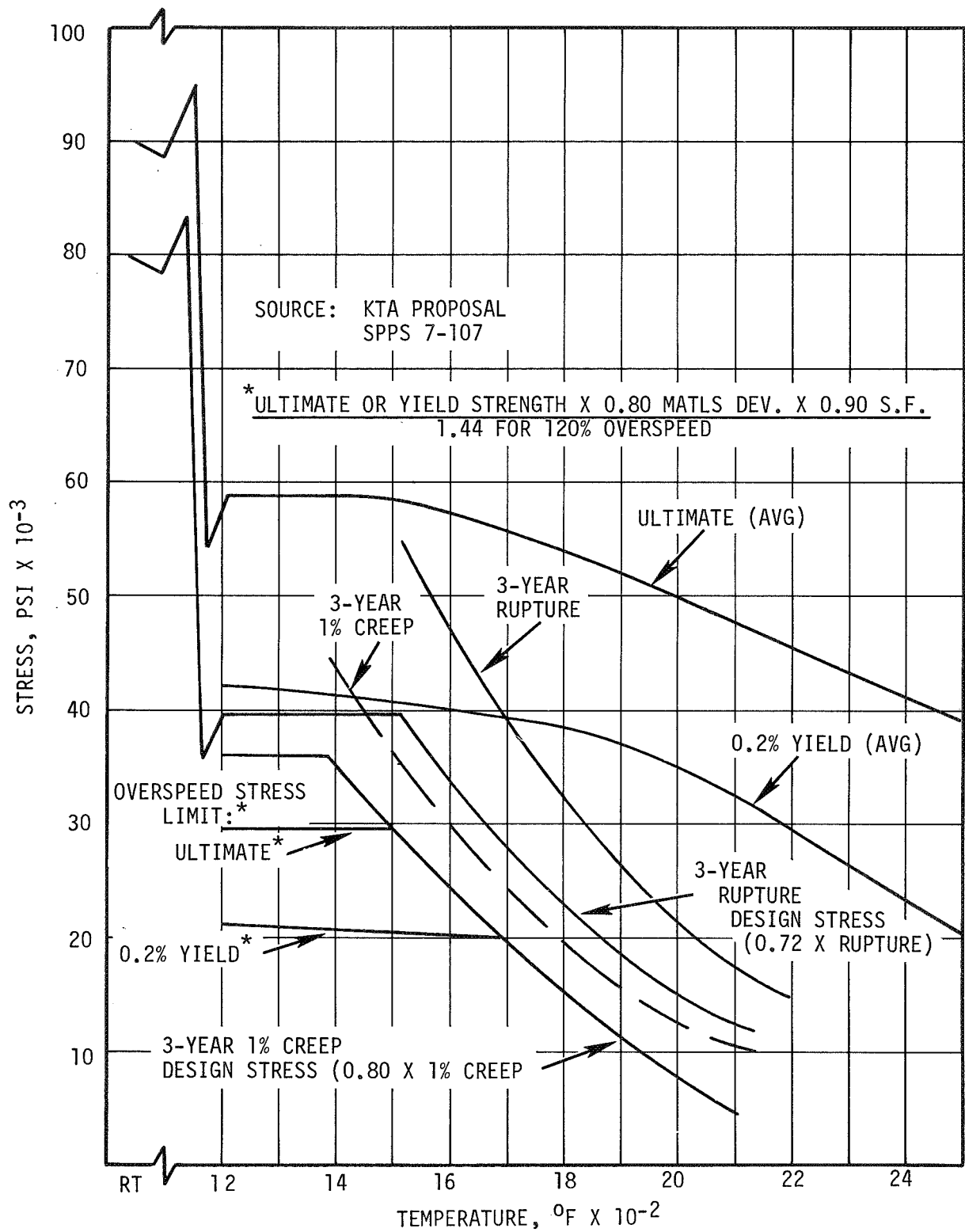


Figure.4.1-8. 3-Year Rupture, 3-Year Creep, and Tensile Strength for T-111. (Recrystallized)

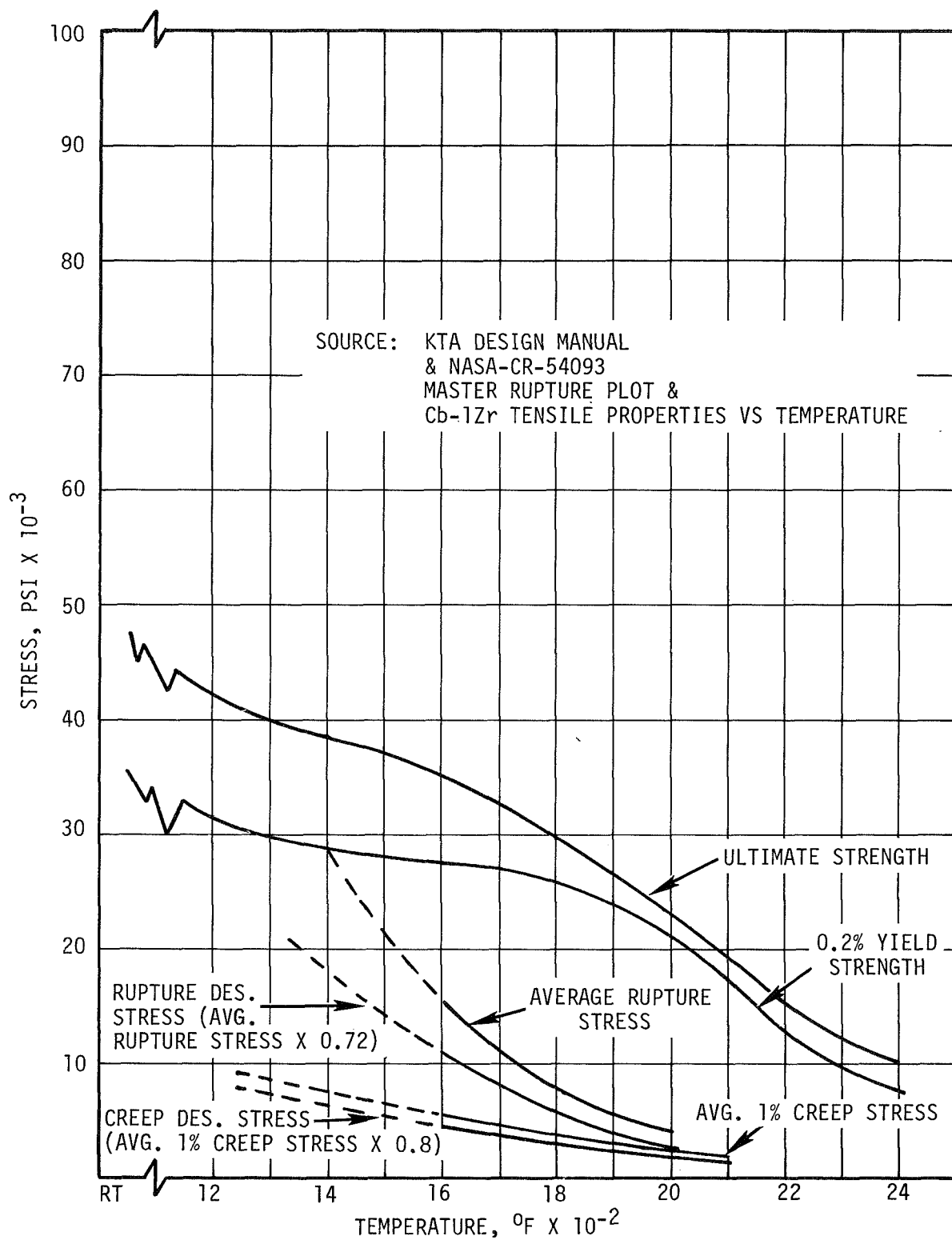


Figure 4.1-9. 3-Year Rupture, 3-Year Creep, and Tensile Strength for Cb-1Zr. (Recrystallized)

operating temperatures as observed from the above design curves.

4.2 DESIGN RELIABILITY

The available design data as extended by extrapolation and as extended by factors intended to derive strength for rupture or for other levels of creep from the initial 0.5 percent creep data is believed to be suitable for first-order design studies. These studies were intended to evaluate various designs, to select appropriate materials for each design and to make a first-order selection of a material-design combination which is evidently superior to its alternates. For this purpose, the present information was sufficient. Additional data is required before precise designs can be made from which components could be machined for reliable use in three-year operating life turbines.

In many cases the data is limited to only a few tests from a single heat (1) in which the processing history may not be fully understood or (2) in which the effects of either random processing variations or those caused by mill product size and shape are unknown. It has been necessary in most instances to extrapolate short-time creep data to long-time periods, to extend parameter curves by extrapolation to meet design data needs, and to estimate one strength property characteristic from another by percentage factors determined from similar property levels in another, more well-documented alloy. Quite frequently the parameter curves used to develop creep data below 2000°F are based upon tests performed at temperatures well above 2000°F. Finally, the tensile property data is often limited to only a few tests at room temperature and one or two tests at a single elevated temperature. Therefore, the data in some cases is relative rather than precise and is subject to the errors

introduced in making extensions in the original, limited data.

The need for limited, but sufficient, test information on the specific materials in component part mill produced sizes is emphasized. It is suggested that (1) tests be performed on material processed to meet the component part size requirements for the potassium vapor turbine, (2) limited tensile testing be done to develop room temperature to 2100°F yield strength, ultimate tensile strength and ductility curves, (3) the presence and temperature range of any ductile-to-brittle transitions be investigated, (4) anisotropic effects on ductility and impact be noted and their severity documented, (5) short-time creep rupture testing be performed in the temperature range 1400-2000°F and parameter curves be developed indicating the strength relationship between the various levels of creep and of complete rupture, (6) one or two long-time creep tests be initiated for purposes of eventual correlation of long-time test results with the short-time parameter curve test data; and (7) limited stress relaxation tests be conducted on TZC at the specific conditions required by the turbine design.

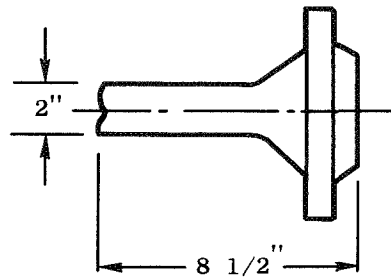
4.3 TURBINE FORGINGS

Because of the problems of welding or otherwise joining together the turbine wheels and turbine shafts, initial exploration was made of the feasibility of preparing integral forgings of the types shown in Figure 4.3-1. These included the integral wheel and shaft, a multistage, single-rotor forging, and a contoured wheel forging with integral inter-stage torque tubes.

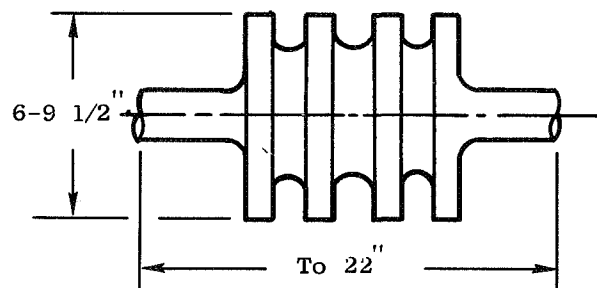
In those alloys, such as TZC and KDTZM which require extensive hot working operations for the purpose of developing mechanical properties,

Integral Forgings:

(A) Wheel & Shaft



(B) Multi-Stage Rotor



(C) Wheel & Torque Tube

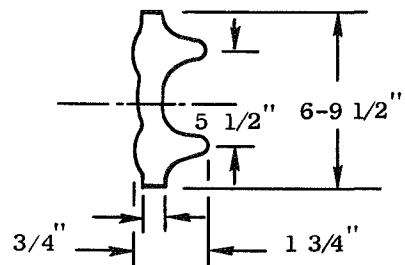
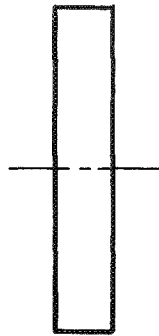


Figure 4.3-1. Integral Forgings Considered in Phase I.

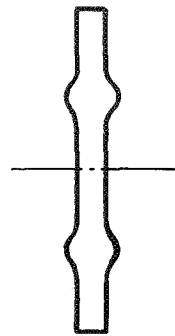
it was not feasible to achieve the degree of working in appropriate places in such complex forgings. The unusual shapes required in these forgings would have necessitated the development of special, closed-die forging practices and extensive trial and error development. In addition, the integral forgings would have required larger billet sizes which would not have permitted the high degree of working required in the final component. For the weldable alloys, which do not require a highly worked structure to develop their strength, such composite wheels would be unnecessary; they would better be joined in individual stages by welding.

The adverse effect of the necessity to develop well-worked structures in the TZM alloys, the desire to maintain individual component identification and replaceability in manufacturing, and the practical impossibility of making complex refractory alloy forgings of the larger size required in integral component forgings, combined to select the simple forging configurations shown in Figure 4.3-2. All wheel forgings would be made without their torque tubes as simple pancake forgings by upsetting billet stock to achieve the degree of working necessary. This process would utilize simple forging techniques without requirement for shaped dies. Should it be impossible to achieve the degree of cold work necessary in such forgings because of their final thickness requirement and limitations on starting billet diameters or sizes, a partially contoured forging would permit some additional work to be imparted to the forging and would provide some flow line improvement in the vicinity of the interstage torque tube connections as shown in Figure 4.3-2. The torque tubes themselves would be either machined from pancaked billet or, for preferred flow line characteristics would be machined from the walls of a forged cup. Upset forging of relatively large diameter forged or rolled bar

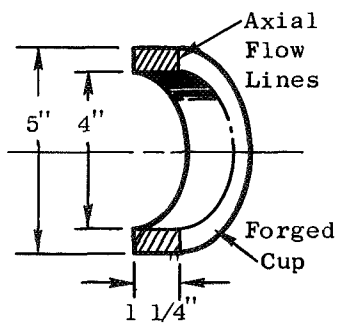
Wheel Pancakes



Partially Contoured Wheels



Torque Tubes



Shafts

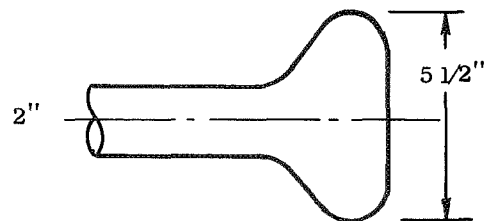


Figure 4.3-2. Typical Type of Forgings Recommended in Phase I.

stock would be used to make the flared turbine shafts and, at much smaller diameters, the TZC tie bolt head.

The survey of forging methods and current state of the art in forging technology indicated the following:

1. Ductile TZC bar stock has been prepared in diameters up to 2 inches but there has been no experience in the preparation of pancake forgings, contoured forgings, cup forgings, or bolt heading operations.
2. The TZM alloy has seen extensive application as simple pancake forgings, Reference 4.3-1, the mechanical properties of which have been quite extensively documented. The contour forging of molybdenum alloys has been accomplished in turbine blade form and in other shapes such as rocket nozzle preforms or nozzle throat inserts. However, it is doubtful if molybdenum forging experience exists which would obviate the need for development work on interstage connectors integral with turbine wheels. The upsetting operations necessary to make flared shafts and headed bolts are within the current state of the art; such operations are not expected to reduce the material properties in the TZM alloy.
3. The forging of reactive columbium-base alloys such as Cb-752 and FS-85 appears very feasible as indicated by Air Force sponsored work, Reference 4.3-2, in which glass-coated Cb-752, D-43, and Cb-132M were forged at 2400°F in both open and closed dies with contamination restricted to a depth of only 0.0046 inch. Because of (1) the expected forgeability of the weldable

alloys, (2) the ability of glass coatings to control surface contamination, and (3) the lack of any dependence of their strength on a worked structure, the forging of these alloys was not presumed to be a critical problem. The higher strength of ASTAR 811C alloy might adversely affect forgeability, however, preliminary forging trials would resolve this possible problem.

The basic need, having selected a material from which to make the major components, is the preparation of actual component forgings using optimized and reproducible processes and the complete documentation of the critical properties of such forgings for use in final product design. A more detailed presentation of the suggested approach is included in the Phase II report.

4.4 METALLURGICAL JOINING

The weld assembly of the structural portions of the turbine is also treated more fully in the Phase II report. In Phase I the option was still open of considering several possible methods of assembling the turbine rotor. These methods included mechanical joining, diffusion bonding, brazing, friction welding, and electron beam (EB) welding.

Mechanical joining was recommended for the nonweldable molybdenum alloys.

The weldable columbium or tantalum-base alloys were to be assembled by careful EB welding techniques; but, it was readily apparent that the ability to weld assemble a large number of finish machined rotors and to maintain accurate axial and radial dimensions in the finished welded assembly was in serious doubt. Also, the inability to readily replace

rotor components damaged in weld assembly, rotor balancing or final assembly offered serious disadvantages to any metallurgically joined rotor system and was a positive argument in favor of mechanical joining.

Diffusion bonding and brazing were most applicable to the non-weldable molybdenum alloys. Technology related to the diffusion bonding of refractory alloys has been established to the point where such an assembly method was feasible either by direct diffusion bonding techniques or by the braze diffusion sink method of bonding. In the latter method an appropriate braze alloy is placed at the joint interface; a lower melting phase forms and quickly diffuses into the parent refractory alloy during the bonding process, Reference 4.4-1. The advantages of the diffusion bonding and/or brazing method was the absence of any serious thermal stresses set up in the rotor assembly during the joining operation. Such low stress methods of joining would require large vacuum brazing furnaces and extensive fixturing and would introduce uncertainties regarding the quality of the diffusion bond or the integrity of the brazed joint, difficulties in inspecting the joints, and the risk of losing an entire rotor assembly in a single adverse aspect of the joining operation.

Friction welding was investigated briefly during the early part of Phase I. This method is currently being investigated for the assembly of INCO 718 compressor wheels in the TF-39 aircraft jet engine, Reference 4.4-2; friction welding of the refractory alloy potassium turbine would be a somewhat similar application. Molybdenum TZM and T-111 cylinders two inches in diameter X 0.200 inch in wall thickness were welded on a Caterpillar Model 100 Inertia Welder in a few simple welding experiments. The T-111 deformed, flowed, and welded without

cracking, but the TZM alloy, in two separate attempts, developed very fine longitudinal cracks and a continuous crack at the weld interface. Since the T-111 alloy, which was successfully joined by this method, could also be electron beam (EB) welded, the friction welding method offered no advantages. The necessity to provide a subsidence allowance in friction welding could result in a stackup error in any full rotor assembly. The friction welding method offered no advantages for the weldable alloys and failed to join the nonweldable molybdenum alloy.

Electron beam welding of the rotors offered the best system of joining the weldable alloy rotor wheels. Since the Phase I design study concluded that the molybdenum alloys were to be used for the rotors, no further thought was given to metallurgical rotor joining.

4.5 BEARING MATERIALS

The Phase I recommendations on bearing materials were based upon the various factors affecting selection of bearing materials as indicated in Table 4.5-1. Those favorable factors included;

- (1) A non-alloying tendency of mating bearing materials,
- (2) good surface stability in potassium,
- (3) stable dimensional characteristics,
- (4) matched thermal expansion with mating components
- (5) hard-soft materials combinations for rubbing accommodation of surface asperities,
- (6) high compressive strength and high elastic modulus to limit deformation or deflections,
- (7) good surface finish to avoid galling,

TABLE 4.5-1

FACTORS AFFECTING SELECTION OF BEARING MATERIALS

Tendency for Sliding Materials to
"Alloy"

Effect of Alkali Metal Environment
on Surface Chemistry

Dimensional Stability

Thermal Expansion

Thermal Conductivity

Hardness

Compressive Strength/Elastic Modulus

Surface Finish

Melting Point

Shear Strength

Wetting

Other Physical + Mechanical Properties
(Thermal and Mechanical Shock, Thermal
and Mechanical Fatigue)

Operating Conditions

Producibility

- (8) high melting point to avoid melting in the temporary absence of lubricant,
- (9) low surface shear strength for inherent surface lubricity,
- (10) ability to withstand some mechanical abuse, and
- (11) an ability to be reproduced, reliably on a commercial basis.

The suggested bearing materials combinations for use at the 800^oF and 1100^oF possible bearing temperatures are indicated in Table 4.5-2. The compositions of these materials and the extent of their evaluation are shown in Table 4.5-3. The selection of these materials was based upon the learning experienced in the NASA sponsored bearing materials program, Reference 4.5-1.

This bearing materials contract work provided an excellent background in understanding the characteristics of many bearing materials in the categories of thermal expansion, hardness, compressive strength, elastic modulus, wetting, thermal stability, corrosion, mass transfer in potassium, etc. The dry sliding friction and potassium friction tests added further information on a limited number of materials or combinations of materials at selected temperatures; and they provided a better understanding of the nature of friction and wear in liquid metal bearing materials. The data could not be expected to permit the high reliability of optimum bearing materials for various temperature ranges. It must be assumed that further friction, wear, and actual bearing tests will be necessary to select bearing materials in a highly reliable manner.

In spite of the fact that insufficient bearing and friction tests have been performed, the background of the bearing materials contract work made it possible to select materials which appeared to offer the

TABLE 4.5-2

SUGGESTED BEARING MATERIALS COMBINATIONS

Radial Bearings:

800°F	Carboloy 907 (WC-20TaC-6Co) Versus Carboloy 907 or K-139 (TiC-5Co) Versus Carboloy 907
1100°F	K-139 Versus K-602 (WC-10Ta-3Co) 90TiC-10Cb Versus 90TiC-10Cb 90TiC-10Cb Versus Lucalox

Thrust Bearings:

800°F	TiC+10Cb Inserts Brazed with AS537 in TZM Structure Versus Carboloy 907
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TABLE 4.5-3

COMPOSITION AND SCOPE OF EVALUATIONS OF SUGGESTED BEARING MATERIALS

Designation	Composition	Evaluated With	Types of Test		
			Individual Test*	Dry Sliding Friction	Liquid Potassium Friction
Carboloy 907	74WC-20TaC-6Co	Carboloy 907 Mo-TZM	x	x	x
E-139	95TiC-5Co	-	-	-	-
E-602	82W-10Ta-2.5Co-5.5C	-	-	-	-
Lucalox	99.8Al ₂ O ₃ -0.1MgO- 0.02SiO ₂ -0.02CeO ₂ - 0.02Fe ₂ O ₃	Lucalox	x	-	-
TiC+10Cb	83.6TiC-9.54 Cb- 5.85WC-0.73Co- 0.33Fe	TiC+10Cb Mo-TZM	x	x	x

* Such as hardness, stability, corrosion, thermal expansion, compressive strength, elastic modulus, etc.

most promise either at lower or higher temperatures and to predict, with some reasoned assurances, those combinations of bearing materials which offered promise for use in the KTA. Both reliability and performance were considered in making these selections using the information gained in the bearing materials program as a guide.

Although no liquid potassium friction tests were conducted with the Carboloy 907 composition, this material was selected as the most immediately available reliable low-temperature bearing material. The selection is justified based upon the results of tests with another WC cermet, Grade 7173, and successful operation of Carboloy 907 bearings in the General Electric full-scale liquid metal bearing test rig. Reference 4.5-2. Producibility and reliability were the over-riding factors in the selection of Carboloy 907. K-139 was chosen as a potentially reliable low-temperature bearing material based on successful evaluation of other cobalt bonded carbides (i.e., Carboloy 907) and the demonstrated superiority of titanium carbide with respect to chemical stability and resistance to mass transfer of carbon. Furthermore, the K-139 material offered the advantage of a dissimilar carbide (TiC) for greater resistance to welding and galling in contact with WC bearing materials. Again, the selection of K-139 over TiC + 10Cb for moderate temperature applications was based on producibility. Similarly, K-602 was suggested for higher bearing temperatures since the greater stability of WC in the presence of free tantalum in liquid potassium was demonstrated in a cobalt-free composition, K-601, under the bearing material contract. The addition of a minimal amount of cobalt made the composition more readily fabricable. The higher temperature capabilities of Lucalox and TiC + 10Cb were evident from the contract work. Although friction tests were only run on the latter, the

individual characteristics of each suggests their value as stable bearing material considerations in potassium.

The above valid reasons for suggesting the bearing materials are shown in Table 4.5-2 and are based in large measure upon the learning experienced in the NASA-sponsored bearing materials contract work. Specifically:

- (1) At 800°F, Carboloy 907, against itself, was selected since it is chemically stable, has high strength and hardness, is commercially available, has been evaluated in dry sliding friction, and has been evaluated as a full-size liquid metal bearing.
- (2) At 800°F, Carboloy 907, against a TiC-5Co cermet (K-139) offers the possibility of significant improvement due to the dissimilar nature of the materials combination; i.e., less tendency for "alloying;" also, both are commercially available, and the TiC stability and compatibility of the cobalt bonding have been demonstrated.
- (3) At 1100°F, where carbon transfer is more likely to occur, TiC cermets are favored; for the sake of using dissimilar materials the demonstrated carbide stability of WC-Ta compositions provides a material pair for the TiC cermet. (Cobalt additions to this cermet make it more readily fabricable and eliminate the need for extensive materials processing development.)
- (4) At 1100°F, a less readily available materials combination, TiC-10Cb against itself, offers demonstrated stability in

potassium and good friction and wear conditions as evidenced by work conducted under the bearing materials contract. However, additional processing development will be required for this composition, which makes it somewhat less desirable (in spite of its better stability) when compared to (3) above.

- (5) At 1100^oF, Lucalox against TiC + 10Cb offers potential improvement over (4) above by virtue of dissimilarity of compositions (less alloying tendency) and, possibly, by the formation of a thin stable potassium aluminate film on the surface of the Lucalox which should favor lubricity in the bearing.

The above bearing materials were suggested to offer several potential materials choices for the KTA bearings, trading low temperature, commercial available, etc., on the one hand for high temperatures, greater stability and need for process development on the other hand. In any case, additional bearing tests or friction and wear tests will be required to measure and improve the reliability and performance of both the materials themselves and the bearing designs before a final decision is made regarding actual KTA bearings.

4.6 RUBBING SEALS

The selection of nongalling materials combinations in KTA applications where rubbing may occur requires similar considerations as discussed above for journal and thrust bearings. Since close clearances are necessary, first to promote turbine efficiency in tip seals and in interstage seals and second, to maximize the effectiveness

of low leakage screw seals, the possibility of rubbing, metal transfer and seizing in such seals cannot be overlooked.

Refractory alloy creep must be considered in establishing the turbine blade tip and interstage seal clearances. Should a rub occur, however, it is necessary that the rub be accommodated without galling or seizing. This will require (1) deformation of the softer element in the seal by the harder element and (2) anti-galling characteristics inherent in dissimilar, hard-soft materials combinations. It was suggested at the end of Phase I that refractory alloy open-faced honeycomb could be used as the stationary portion of such seal systems in the hotter and dryer sections of the turbine where erosion by liquid metal droplets will not limit the effectiveness of the honeycomb. Such open-faced honeycomb shrouds of stainless steel were used in the 5000-hour test of a two-stage potassium vapor turbine, Reference 4.6-1. Where erosion is a problem, gross rubs may be accommodated by the upsetting of thin turbine blade squealer tips or soft interstage teeth, each rubbing against a hard, dissimilar material surface.

While the above suggestions appeared reasonable at the end of Phase I, it was evident that direct experience in the nature of gross interference rubbing in vacuum and in wet potassium vapor would be necessary in order to determine the possible magnitude of this problem and to evaluate the materials selections and design configurations which come to mind in attempting to produce compatible, nongalling, rubbing surfaces.

4.7 CORROSION AND EROSION

No new information was available at the end of Phase I regarding the corrosion and erosion of refractory alloy turbine components in wet potassium vapor at turbine operating experience.

On the basis of past experience, however, no significant corrosion problem should occur if (1) the refractory alloy raw materials are properly controlled to meet materials specifications, (2) proper welding techniques are used to avoid oxygen contamination, and (3) the oxygen concentration and other critical impurities in the potassium are maintained at reasonably low levels.

The erosion resistance of various materials under impacting liquid potassium droplets had, at the end of Phase I, given sufficient evidence to make certain an earlier estimate that significant droplet damage will occur in the last stage of the refractory alloy turbine. After three years of operation at 1280⁰F, 900 feet per second tip speed, and 18 percent moisture, it is expected that the damage will be such as to require the use of some form of moisture separation mechanism in order to achieve reliable life and efficient turbine operation. Since the precise magnitude of the damage is still uncertain, it will be necessary to conduct additional turbine testing in superalloy systems at high moisture contents to evaluate the effect of multiple droplet impact on the materials of turbine construction.

4.8 MATERIALS SUMMARY

It was concluded that the selection of materials and the immediately available materials property data were adequate for preliminary decisions affecting the selection of alternate design possibilities, but it was

evident that more complete property data should be procured on actual component part mill products processed in a planned and reproducible manner.

The initial selection of bearing materials was adequate for the Phase I stage of the design and additional materials were indicated which should possess improvements in reliability through either better bearing characteristics or through improved manufacturability.

Liquid metal corrosion was not expected to be a problem if proper common precautions are followed.

The precise extent of liquid droplet erosion to be expected in the life of a refractory alloy turbine is not predictable but some significant amount of damage must be expected and methods to minimize this damage must be considered.

A better appreciation is needed of the galling, metal transfer, and/or deformation accommodation which occurs under gross rubbing conditions in vacuum or wet potassium vapor in turbine-type tip seals and in interstage seals of various materials combinations and of various configurations.

4.9 REFERENCES

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- 4.3-1 Nichols, H.E., Fink, R.W., and Zimmerman, W.F., "Three-Stage Potassium Vapor Turbine - Fabrication and Assembly," Contract NASA CR-72501, General Electric Company, Cincinnati, Ohio.
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- 4.4-1 Freedman, A.H., and Mijus, E.B., "Research on Tantalum and Molybdenum Brazing Techniques," Contract AF 33(657)11227, Northrup Norair, Hawthorne, California, 1964.
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- 4.6-1 Nichols, H.E., and Fink, R.W., "Two-Stage Potassium Test Turbine - Mechanical Design and Development," Contract NAS 5-1143, Topical Report, NASA Contract No. NAS 5-1143, General Electric, Cincinnati, Ohio.

5. TURBINE MECHANICAL DESIGN

5.1 INTRODUCTION

Design Conditions

The turbine mechanical design problems are a composite of conditions imposed by the working fluid, potassium, the power requirement, the rotative speeds and the number of stages chosen. Special conditions are imposed by the bearing and seal areas which, in accordance with the Work Statement, either use potassium as a lubricant or synthetic oil. The potassium lubricant temperatures are in the range of 800° to 1100°F while those for synthetic oil are at 400°F.

A typical single spool, nine-stage turbine, cradled between two potassium lubricated bearings, with a thrust bearing at one end is shown in Figure 5.1-1. This design was submitted with the proposal for this contract and is, therefore, typical for the status of design at the beginning of this study. It is indicative for the problems of rotor and stator design and of vapor sealing in conjunction with a balance piston to minimize bearing thrust loads. High pressure, high temperature vapor enters the turbine at one end through an inlet scroll and leaves the turbine at the opposite end through an outlet scroll. The temperature profile plotted along the turbine from one end to the other is indicative of the local temperatures and the temperature gradients experienced.

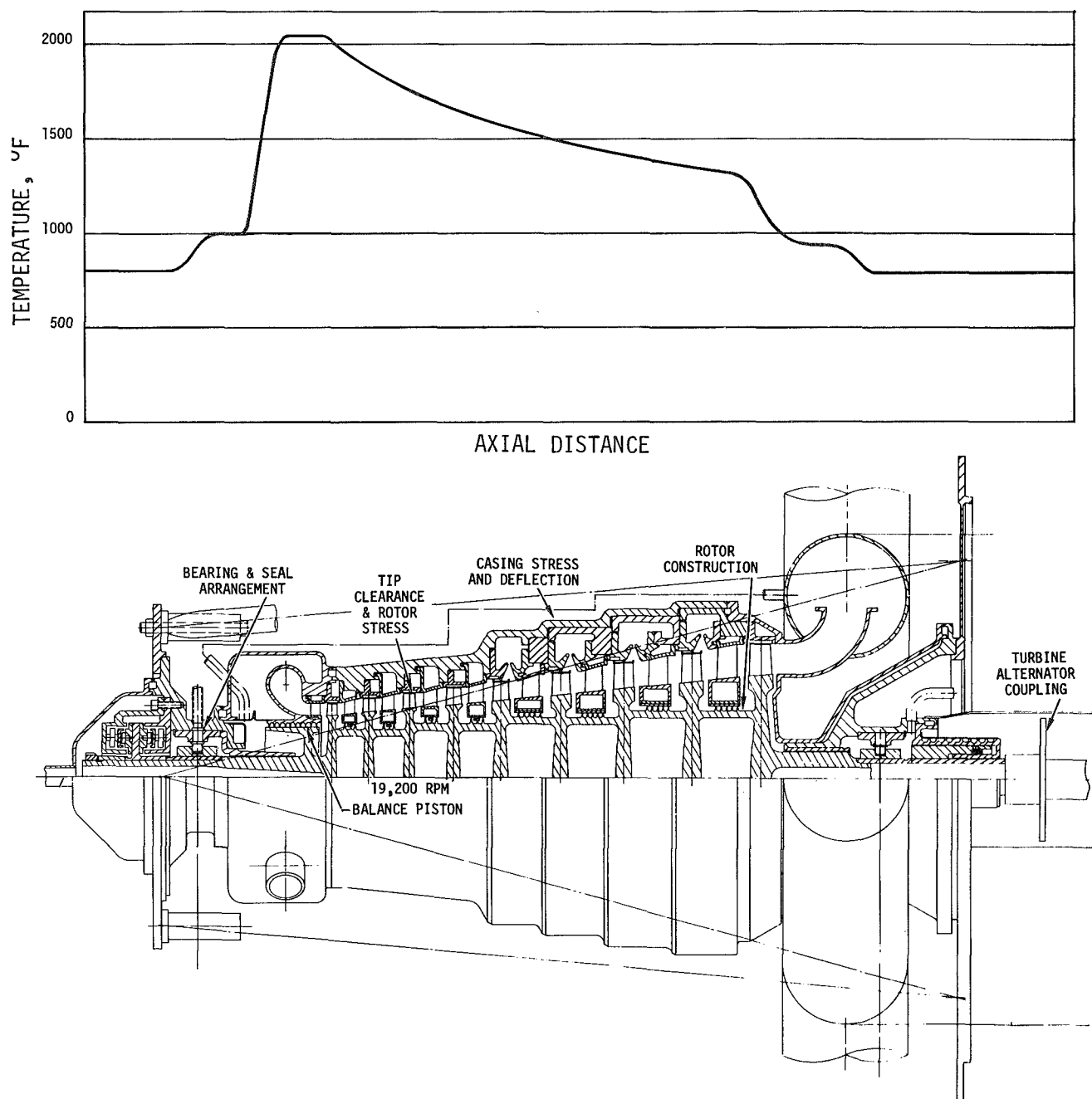


Figure 5.1-1. Turbine Mechanical Design.

Design Problems

The number of mechanical analyses employed in the design effort is fixed by the number of design problems.

The turbine rotor requires refined stress analysis for wheels, blades, blade connections, torque tubes and stub shafts. The hot portions of the rotor are creep limited, the lower temperature portions are yield stress limited. Casing and stator design pose problems of stiffness and rigidity rather than of stress limitations. Vibrations pose basic problems for blades and the rotor in general, but also for inlet and outlet scrolls as well. Large size gas ducts may be susceptible to vibrations which could lead to cracking and failure.

Differential thermal expansion can be an acute problem during turbine start-up or shut-down or any other transient condition. Most critical is the establishment of tip seal clearances consistent with creep rates to be expected and the clearances to be chosen in bearings and seals. Contact in any one of these areas can lead to failure and has to be duly avoided.

Choice of optimum materials is one of the prime design considerations and is especially important in this study to establish feasibility and comparative advantages of welded rotor designs versus bolted designs. Weight is of prime importance for any space type hardware. The observance of weight minimization has to be a continuous guideline in the design.

Design Approach

The technical discussions which follow concentrate on key design problems but do not attempt to be all complete as far as turbine hardware

design is concerned, consistent with the concept of a preliminary design study.

The first major design problem discussed is the high pressure seal which is typical for any cradled single spool turbine design.

The second major design problem deals with holding a tight tip clearance with a creep-strength limited rotor.

Following are studies of blade connections, blade vibrations, stage connections and thermal considerations, as applied to casing and stator design and the cold and hot clearances to be chosen. Bearing and seal design, flexible coupling, and the turbine configurations evolved are discussed in additional chapters.

5.2 HIGH PRESSURE SEAL

One of the basic turbine geometries considered was a single spool, in-line, straddle mounted rotor operating on potassium lubricated bearings with a lube inlet temperature of 800°F. A high pressure seal is needed to prevent the outflow of potassium vapor from the first stage wheel into the bearing and seal space surrounding the rotor stub shaft. The low temperature surfaces of the bearing cavity and the bearing seal housing will act as condensers and transmit large amounts of heat into the potassium lubricant. This unwanted heat transfer has to be minimized by effective, multistep labyrinth sealing and by insulating the exposed low temperature surfaces of the bearing and seal housing. It cannot, however, be avoided that high temperature vapor rushes into the liquid metal screw seal at sonic velocity through an annulus determined by seal clearance (~ 2 mil/inch diameter). Also, accumulation of condensed liquid around the shaft cannot be tolerated because of associated parasitic losses.

The design problems and the approach taken are illustrated in Figure 5.2-1. The high pressure seal shown is a step seal in conical arrangement, resulting in the lowest possible leakage rates under the design conditions given. The large diameter balance piston indicated in Figure 5.1-1 had to be abandoned for two reasons: Its leakage rate was excessive and so were the stress levels in the rotating cylinder. Therefore, the idea of compensating thrust was given up in favor of effective sealing.

The temperature and pressure downstream of the first stage stator blade row are 1997^oF and 151 psia, respectively. A portion of the shaft joins the region of the first stator exit to the forward bearing, where the temperature and pressure of the potassium liquid are 800^oF and 30 psia. Leakage of potassium vapor along this path can be controlled through the use of a labyrinth seal with small clearances, small diameters and a large number of teeth. However, since the flow through the labyrinth is essentially adiabatic, the vapor leaving the labyrinth in the region of the bearing is superheated to a temperature not very different from the 1997^oF at the stator exit. Thus, although a large axial distance has been used to reduce the pressure of the vapor, the temperature remains the same. Most of the temperature difference from 1997^o to 800^oF must occur over a short axial length of the stub shaft. This leads to high thermal gradients and stresses in the shaft.

The sharp temperature gradient can be alleviated to some extent by the seal arrangement shown in Figures 5.2-2 and 5.2-3. The seal is composed of five separate labyrinth seals, designated 1, 2, 3, 4, and 5, separated by four manifolds, designated a, b, c and d. The manifolds are alternately connected to downstream turbine stages having lower and higher pressures than the preceeding one. In this manner the temperature along

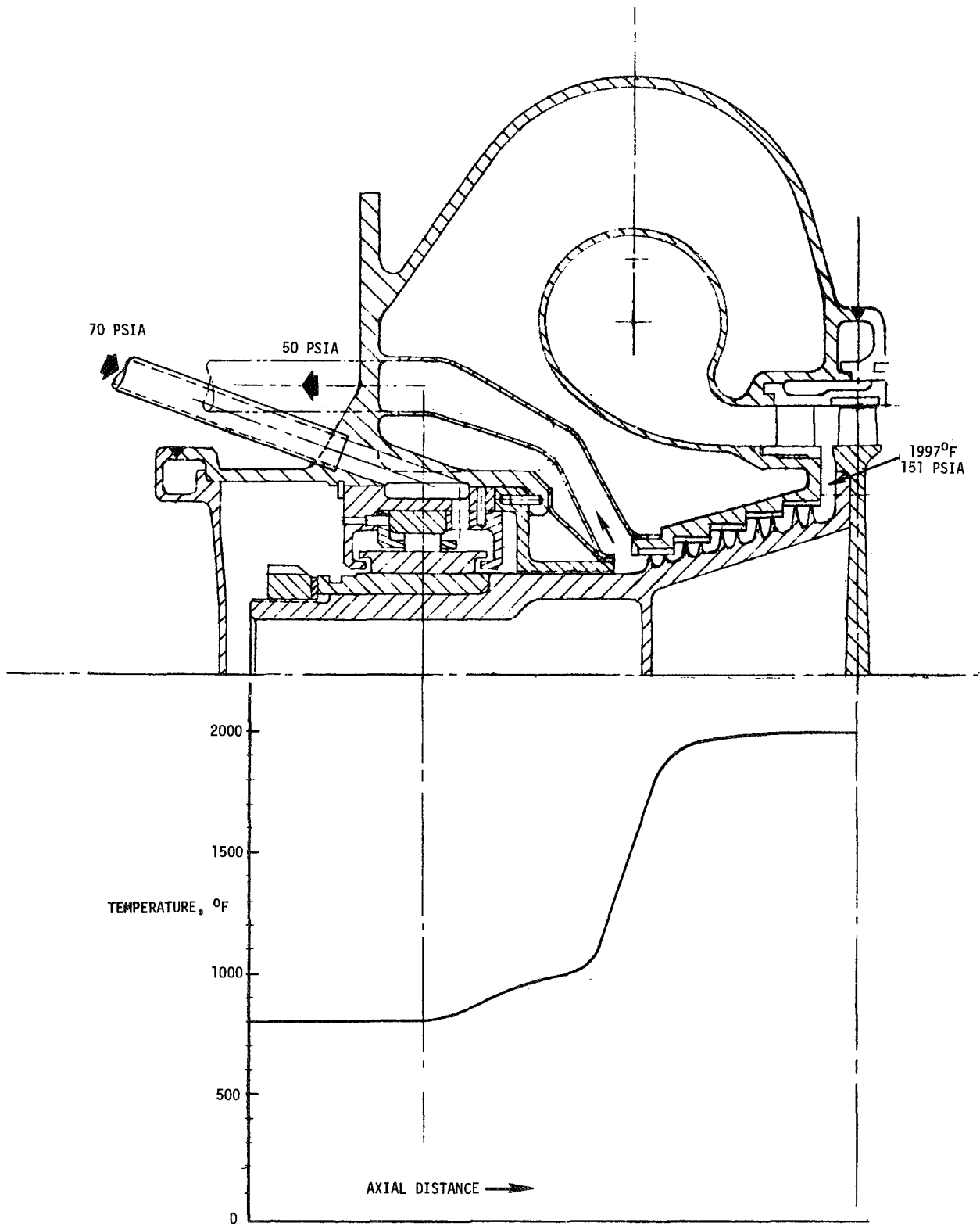
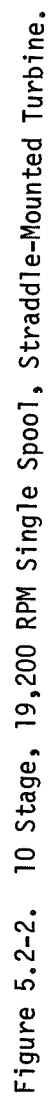


Figure 5.2-1. Forward Bearing and Seal. (Initial Design)



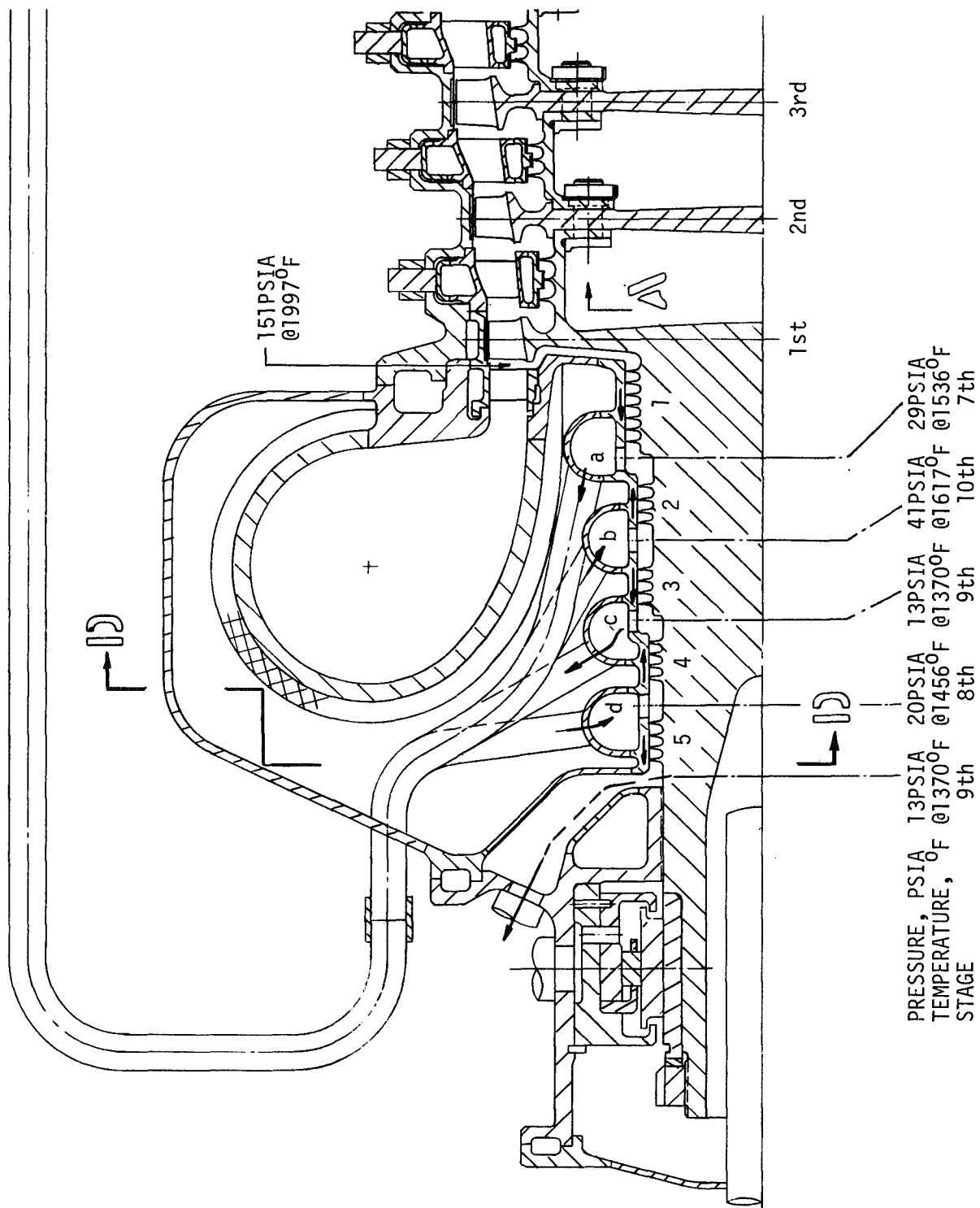


Figure 5.2-3. Multiple Labyrinth High-Pressure Forward Seal.

the shaft is made to drop more or less gradually from the first stage stator exit to the bearing.

Shown in Figure 3.2-5 of the Fluid Design Section is the labyrinth seal leakage flow rate against stage number for the multiple labyrinth seals. The dashed line represents the leakage of the individual stages. The solid line represents the total flow rate loss after each stage. In all labyrinth seals but the first, the leakage flow is available for work in the next stage. However, because of the multiple labyrinth seal described above, the leakage flow rate decreases the active flow rate through all stages except the last two. The power loss to the turbine arising from these leakages is shown in Figure 3.2-6. The total loss is 27.5 kw, which is approximately five percent of the net turbine shaft power. Approximately 17.4 kw of this loss can be directly attributed to the stage one seal.

Even though the thermal gradient has been reduced, some concern regarding thermal stresses is still warranted. The Transient Heat Transfer computer program is ideally suited to determine the temperature distribution at any point and time within the stub shaft. Before start-up, the entire system is assumed to be at a uniform temperature of 600^oF. At time $t = 0$, vapor and liquid at the temperatures shown are admitted to the shaft. Temperature plots at time $t = 2.5$ seconds, 20 seconds and 15 minutes have been made to illustrate the thermal gradients (Figures 5.2-4, -5, and -6). They appear to be highest in the mid-portion of the stub shaft during transient conditions and at steady state. In addition to the axial temperature variation, there is a maximum radial temperature variation of over 100^oF, with the inside diameter of the shaft at the lower temperature. The thermal stresses arising from this variation have been calculated using

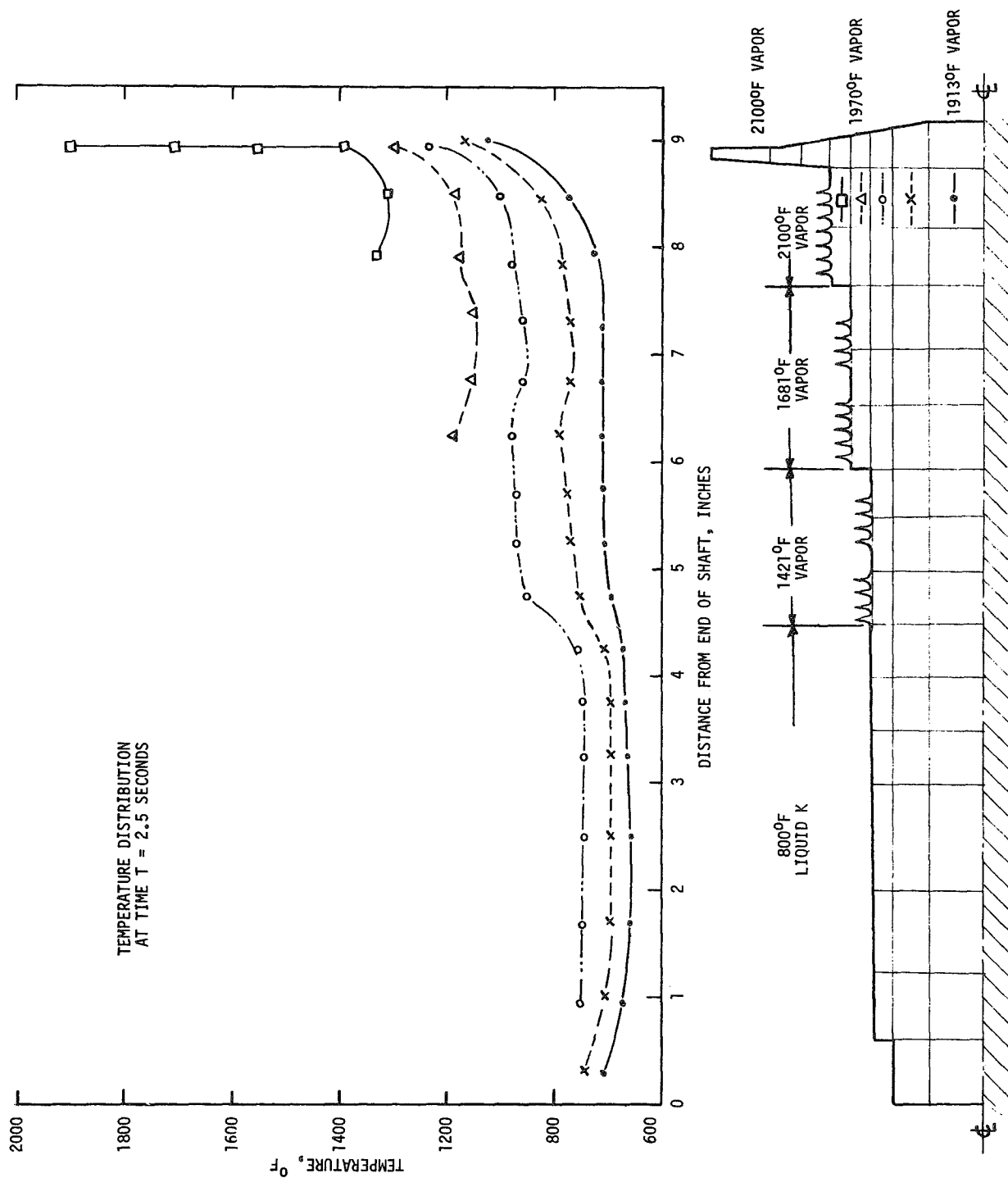


Figure 5.2-4. Forward Stub Shaft of 10 Stage, 19,200 RPM Rotor.

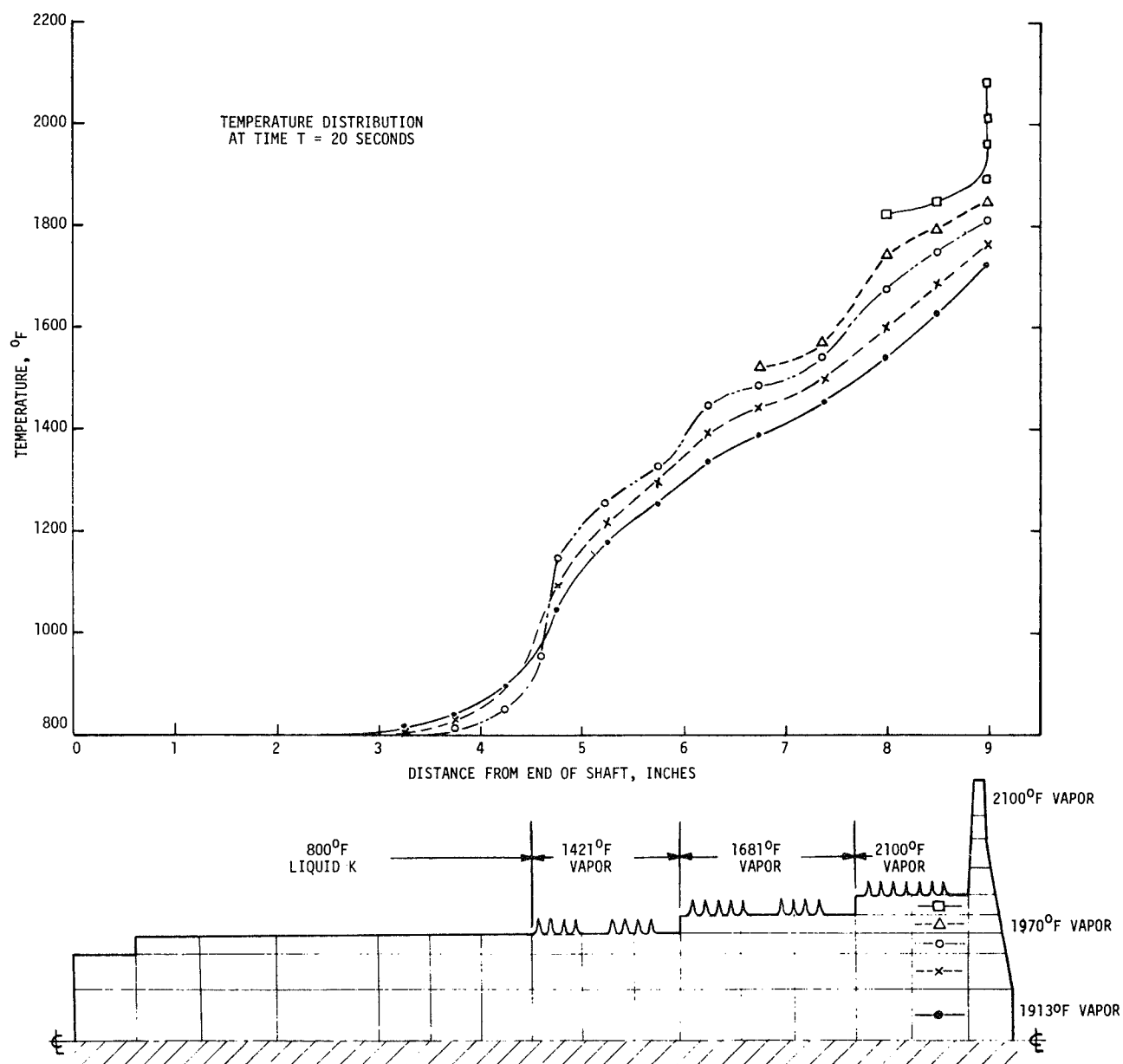


Figure 5.2-5. Forward Stub Shaft of 10 Stage, 19,200 RPM Rotor.

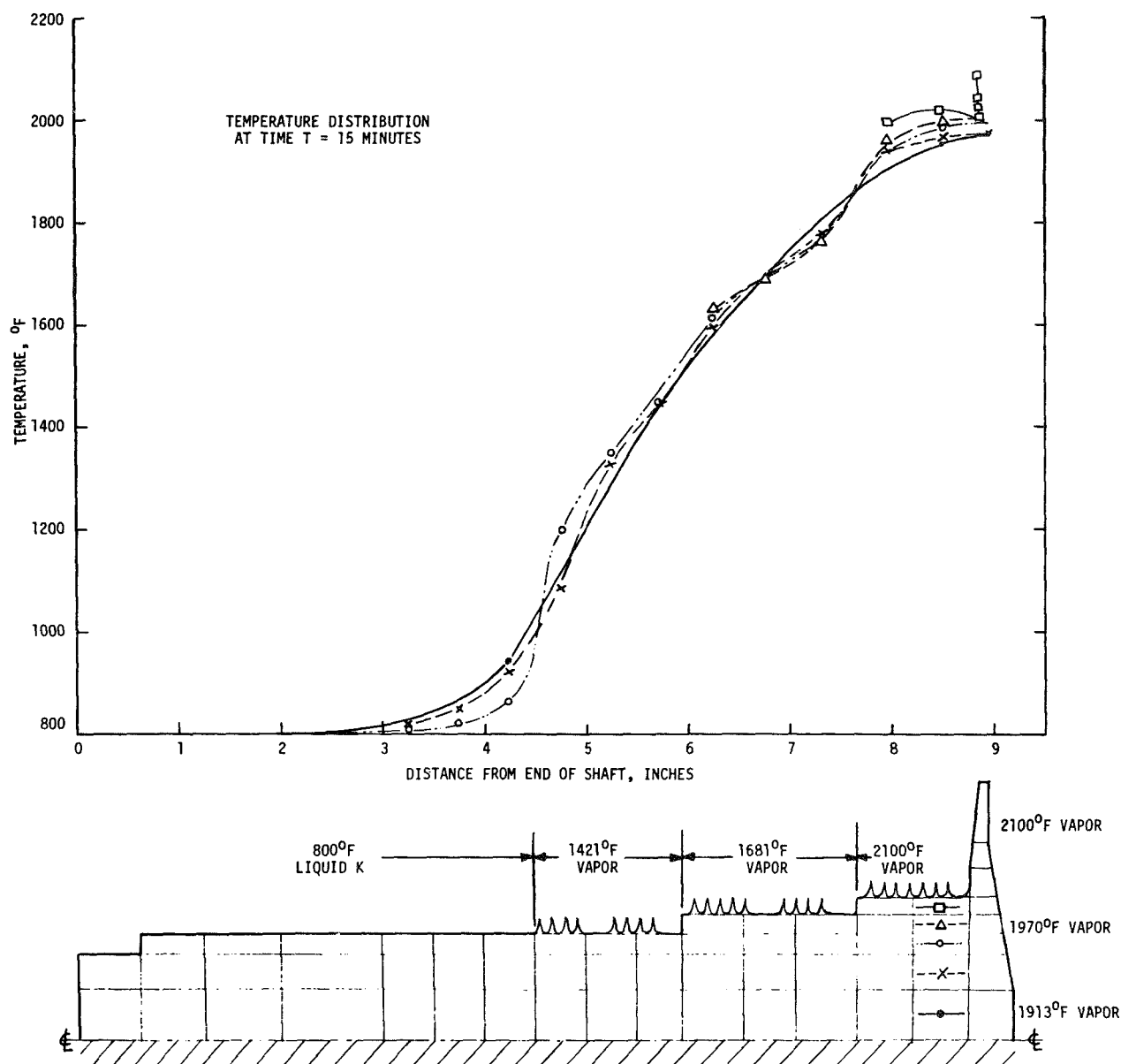


Figure 5.2-6. Forward Stub Shaft of 10 Stage, 19,200 RPM Rotor.

ROTOR finite element computer program. TZM material was assumed, with the calculated temperature distribution and 19,200 rpm rotative speed. Effective stresses of approximately 15,000 psi at the inner surface and 30,000 psi at the outer surface are predicted. These results are shown by Figure 5.2-7 for a stub shaft of 1-inch ID and 2.12 inches OD in the region 4-5 inches from the end of the shaft.

The conclusion is that the design is unattractive from four different viewpoints. The seal losses are high and constitute a large portion of the losses affecting turbine efficiency. The thermal stresses are uncomfortably high for a brittle refractory material. The bearing-rotor response proved that the straddle-mounted rotor behaved poorly. Finally, as discussed above in Section 4.3, the forging experience does not exist which is required to successfully produce the complex integral forging of the forward stub shaft and first stage wheel combination. The large billet size required would not permit the high degree of working necessary to develop adequate material properties for this highly stressed application. For these reasons, the GE design team abandoned the high pressure seal and pursued approaches where this high loss and risk item is avoided.

Turbines Without High Pressure Seal

Considerable experience with steam and vapor turbines which avoid the high pressure seal problems and losses is available within the General Electric Company. In steam turbine design, the single spool, split flow approach is used wherever possible. It not only avoids the high pressure seal but it also results in balanced axial forces, almost eliminating the need for a thrust bearing.

The second established approach is overhung turbines, which have been used for high pressure steam, but also in mercury commercial peaking power

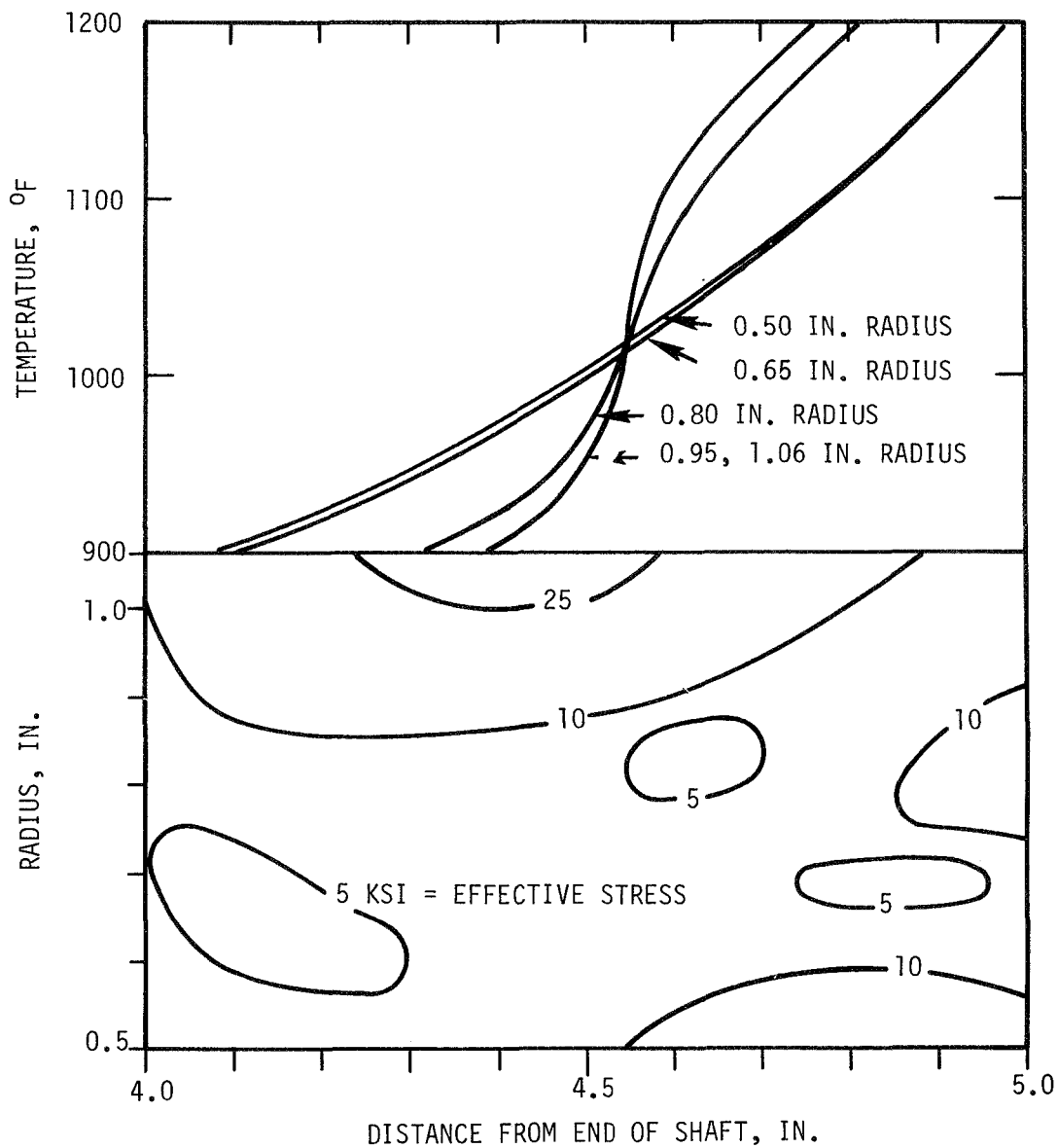


Figure 5.2-7. Effective Stresses in Region of Highest Thermal Gradients of Multiple Labyrinth Forward Stub Shaft Seal. Rotative Speed = 19,200 RPM.

plants where a six-stage mercury turbine with seven feet diameter at the last stage (developing 7500 kw power) has been overhung successfully.* The NSP design team, in designing two- and three-stage potassium test turbines, chose, after careful deliberation, the overhung design. The dynamic behavior of these overhung turbines is known to be excellent.** The overhung three-stage potassium test turbine is an extremely smooth operating turbine, based on design *** as well as operating experience.

Based on this positive experience, KTA turbine arrangements have been evolved which eliminate the high pressure seal. These turbine configurations are indicated on Figure 5.2-8, where

- (a) represents the single spool, in-line, straddle-mounted turbine of the proposal.
- (b) shows in principle a single spool, split flow turbine, straddle mounted between external bearings.
- (c) represents a single spool in-line turbine, with all stages overhung. For this turbine, both bearings are at the low pressure end with a single low pressure seal.

For the split spool turbines requested by the Contract Work Statement, similar approaches have been incorporated.

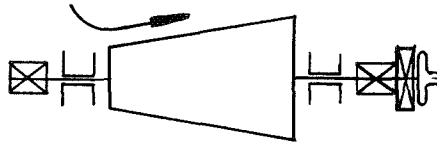
- (d) shows a split spool, counter-flow turbine, cradled between two external bearings with medium and low pressure seals. The high pressure seal losses are reused in the second spool. This configuration offers good balance of axial thrust.

* Ref. Mercury Unit Tops Pittsfield Plant, Published in "Power", March, 1950.

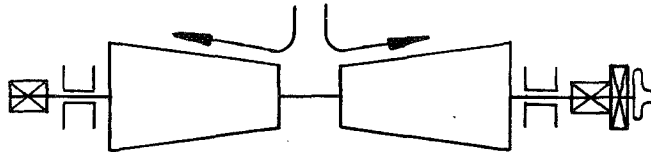
** Ref. 2 Stage Potassium Turbine, Final Report, Vol. II, p. 37, 38.
R67SD3006, Contract NAS 5-1143.

*** Three Stage Potassium Test Turbine, Final Design, Vol. II, Mechanical Design, p. 21, NASA-CR-72250.

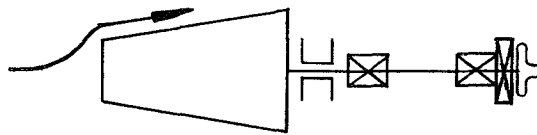
- A — ● SINGLE SPOOL, IN-LINE, STRADDLE-MOUNTED



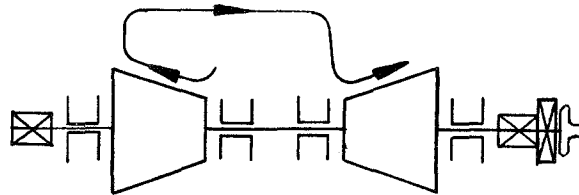
- B — SINGLE SPOOL, SPLIT FLOW, STRADDLE-MOUNTED



- C — ● SINGLE SPOOL, IN-LINE, OVERHUNG



- D — SPLIT SPOOL, COUNTER FLOW, HIGH PRESSURE SEAL BETWEEN SPOOLS



- E — SPLIT SPOOL, OVERHUNG + COUNTER FLOW, BEARING BETWEEN SPOOLS

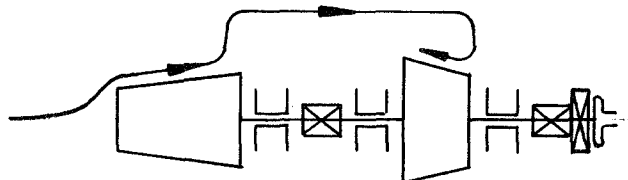


Figure 5.2-8. Turbine Geometries Considered.

(e) shows another split spool, counter flow design approach where one bearing is located between spools while the second bearing remains external. The first spool in this case is overhung, reducing sealing problems to medium and low pressure sealing. The arrangement uses counter flow and offers good balance of axial thrust.

When these turbine configurations were conceived, the biggest unknown was their predicted dynamic behavior. It was feared that certain of these designs, especially the all-overhung turbine might have excessive amplitudes at the first stage when operating at critical speeds. The chapter on bearing rotor response contains the answers to these questions. The big surprise was in finding out that the overhung turbine had a much better dynamic behavior than the cradled approaches because of damping and low temperature shaft rigidity. Consequently, the turbine configurations eliminating or reducing the high pressure seal problem received prime attention.

5.3 TIP CLEARANCE AND ROTOR STRESSES

A ten mil radial tip clearance between buckets and casing has been set as a reasonable and desirable design objective for turbines with blade lengths between 0.25-2.50 inches. A creep-limited rotor will require larger initial tip clearances in order to have only ten mils of clearance after three years of operation. The design must be based upon the lowest achievable creep rates in order to minimize this difference between initial and final tip clearance. Differential thermal expansion between rotor and casing during start-up, shut-down, or changes in power level will be an additional criterion in establishing the feasibility of the ten mil tip clearance.

Material Design Curves

Molybdenum, tantalum and columbium alloys are the only three types of materials considered. Figure 5.3-1 shows the allowable stress design curves for both TZM-molybdenum and TZC-molybdenum, the materials for the bolted rotor. At 1820°F, the TZM and TZC 0.1 percent creep curves intersect. This means that TZC should be used in the first two stages of the bolted rotor and TZM in the latter stages.

For the welded rotor, there must be a change from the FS-85 columbium alloy to ASTAR-811C tantalum alloy at 1600°F (Figure 5.3-2), based upon utilizing the material having the higher creep stress/density ratio. This requires that approximately the first five stages be made of ASTAR-811C.

The T-111 tantalum alloy is used in the casing of the turbine stages which have operating temperatures above 1800°F, with Cb-1Zr used for the remainder of the casing. The crossover temperature is chosen largely because it is the upper limit of our present operating experience with Cb-1Zr. T-111 and Cb-1Zr provide adequate strength in weldable refractory alloys without requiring the developmental problems of ASTAR-811C and FS-85.

Figures 5.3-1 and 5.3-2 have been developed from available material property information as follows:

For short-term 20 percent overspeed operation, the centrifugal stresses in a rotating part will increase by the square of the ratio of overspeed rotational speed to design speed:

$$\left[\frac{\omega_1}{\omega_o} \right]^2 = \left[\frac{1.20N_D}{N_D} \right]^2 = 1.44$$

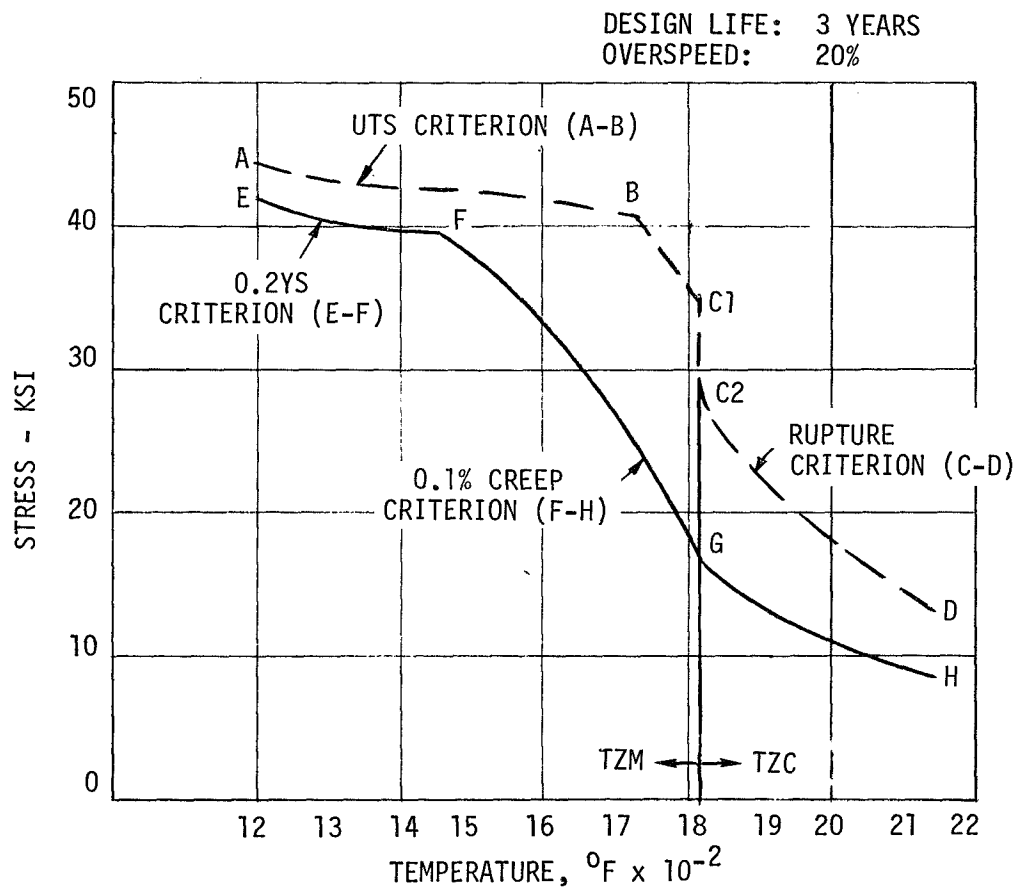


Figure 5.3-1. Allowable Stress Design Curves for TZC/TZM (Bolted) Rotors.

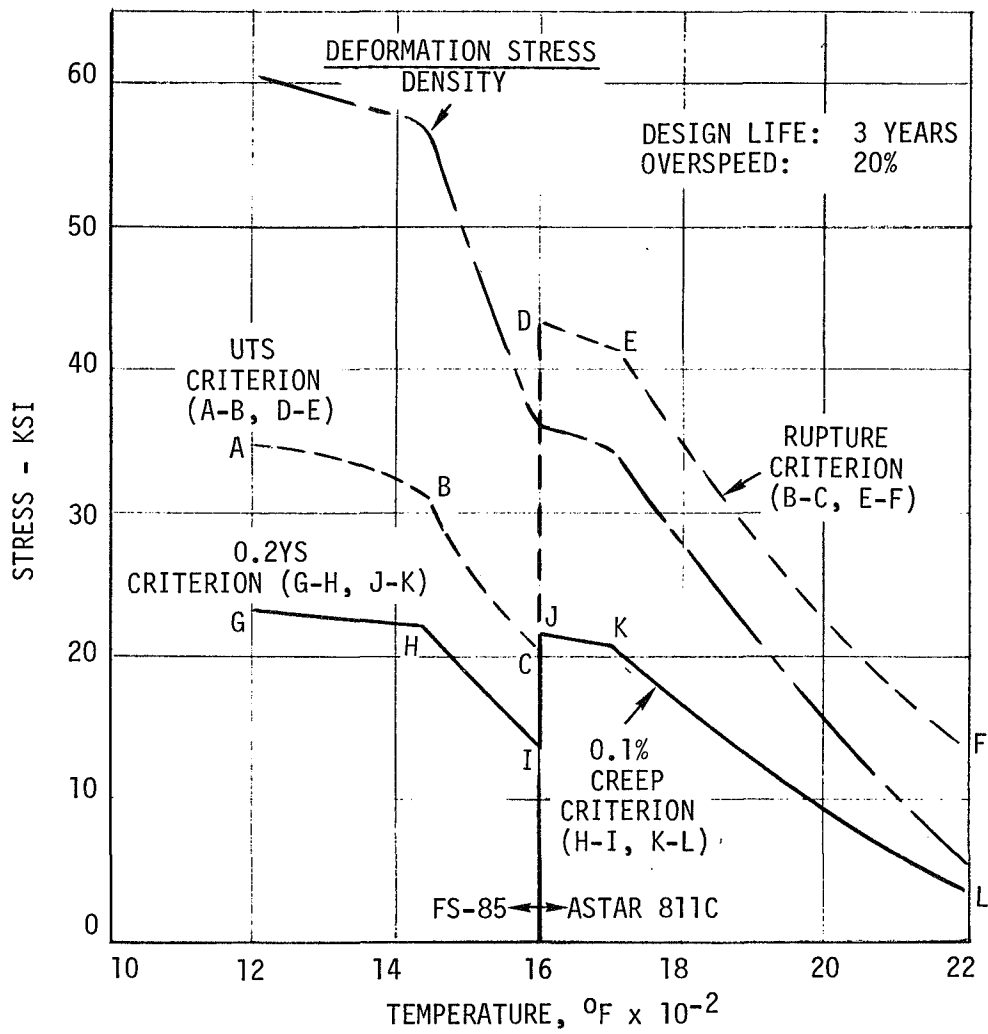


Figure 5.3-2. Allowable Stress Design Curves for ASTAR 811C/FS-85 (Welded) Rotors.

Therefore, the short-term material properties of average 0.2 percent yield stress (.2YS) and average ultimate tensile stress (UTS) should be decreased to 1/1.44 of their normal values.

Average material properties should be decreased to 80 percent of their normal value to account for deviations from average which might occur. This factor is assumed, in the absence of complete statistical data which would allow the calculation of 3σ standard deviations from average properties.

Since catastrophic failure is the penalty for exceeding the ultimate tensile stress of the material, an additional margin of safety is provided by allowing the design stresses to be no greater than 90 percent of the allowable stresses predicted by the above criteria.

In summary, the allowable stresses used to design rotating parts for short-term 20 percent overspeed are:

Basic average stress	.2YS	UTS
Overspeed factor	x 1/1.44	x 1/1.44
Material deviation factor	x 0.80	x 0.80
Margin of safety factor	<u> </u>	x <u>0.90</u>
	<u>0.80</u> (.2YS)	<u>0.72</u> (UTS)
	1.44	1.44
Allowable overspeed stress	= 0.56(.2YS)	= 0.50 (UTS)

Similarly, the three-year design stresses are based upon the known average creep stress (Cr_3) and average rupture stress (R_3). The overspeed factor is not required, because it is assumed that overspeed does not occur for three years, but the 80 percent material

deviation factor is applied to average creep and rupture stresses, and the 90 percent margin of safety factor is applied to the rupture stress.

Therefore, the allowable stresses used to design rotating parts for three year operation are:

Basic average stress	Cr_3	R_3
Material deviation factor	x 0.80	x 0.80
Margin of safety factor	_____	<u>x 0.90</u>
Allowable 3-year stress	0.80 Cr_3	0.72 R_3

Because of the wide range of operating temperatures which exist in most turbine designs considered, consideration must be given as to whether to use the allowable overspeed stresses or the allowable three-year stresses. The problem is resolved most easily by plotting the two types of stresses on the same chart, as functions of temperature, and using the minimum allowable stress which exists at a given temperature. Figure 5.3-1 presents this comparison for the rotor material combination of TZM and TZC.

The deformation criteria is developed first. The E-F portion of the curve is the allowable overspeed criteria 0.56 (.2YS), which is less than the allowable three-year 0.1 percent creep stress 0.80 (Cr_3) below 1460°F. Arc F-G shows that the allowable three-year 0.1 percent creep stress of TZM is less than the allowable overspeed yield stress at all temperatures above 1460°F. At 1820°F, the allowable three-year 0.1 percent creep stress of TZC is larger than that of TZM, so point G (1820°F) becomes the criteria for rotor material

selection. Arc G-H illustrates that the allowable three-year 0.1 percent creep stress of TZC is less than the allowable overspeed yield stress for temperatures of 1820° - 2150° F.

No consideration of stress/density relationship is required, because TZM and TZC have the same density of 0.369 lb/in.³

To summarize, where material deformation is to be the principal criteria:

Allowable deformation stress = minimum of 0.56 (.2YS) or
0.80 (Cr₃)

Material changeover point = temperature at which (0.80Cr₃)_{TZC}
> (0.80Cr₃)_{TZM}

The fracture criteria curve is developed, using the material changeover point of 1820°F established by the allowable deformation criteria. This is done because most design decisions are based upon the deformation criteria, with fracture criteria used relatively seldom but strongly considered when the allowable deformation stress must be exceeded.

Arc A-B of Figure 5.3-1 shows that the allowable overspeed ultimate tensile stress is lower than the allowable three-year rupture stress of TZM to 1750°F, above which the reverse is true. Arc C2-D shows that the allowable three-year rupture stress of TZC is less than its allowable overspeed UTS at 1820° - 2150° F.

Thus, the fracture criteria are:

Allowable fracture stress = minimum of 0.50 (UTS) or 0.72 (R₃)

Material changeover point = deformation changeover point

Similar curves are developed for the weldable material combination ASTAR-811C with FS-85. In this case, however, consideration must be given to the efficiency criterion of deformation stress/density, since ASTAR-811C (0.6 lb/in.³) weighs 1.6 times as much as FS-85 (0.383 lb/in.³). Using this criterion, FS-85 is used below 1600°F and ASTAR-811C is used above 1600°F. Figure 5.3-2 illustrates the allowable deformation and fracture criteria which were used to design weldable rotors.

Over arc A-B, 0.50 (UTS) is less than 0.72 (R_3) for FS-85, while the reverse is true over arc B-C (1450° - 1600°F). Over arc D-E, 0.50 (UTS) is less than 0.72 (R_3) for ASTAR-811C, while the reverse is true over arc E-F (1700° - 2200°F). These lines define the allowable fracture criteria.

Over arc G-H, 0.56 (.2YS) is less than 0.80 (Cr_3) for FS-85, while the reverse is true over arc H-I (1440° - 1600°F). Over arc J-K, 0.56 (.2YS) is less than 0.80 (Cr_3) for ASTAR-811C, while the reverse is true over arc K-L (1700° - 2200°F).

Wheel Design

The effective stress in the wheels is calculated according to the Distortion-Energy Theory of von Mises and Hencky (5.3-1):

$$\sigma_e = \frac{1}{\sqrt{2}} \sqrt{(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{xz}^2)}$$

Where the subscripts denote principal tensile or compressive stresses in the direction of the coordinate axes x, y, or z, or shear stresses such as τ_{xy} acting on the surface parallel to the x axis in the direction of the y axis.

The effective stress theory is very generally accepted as applicable to all materials except for very brittle ones, in which case the maximum principal stress is of most interest. The physical concept of effective stress is that the multiple principal stresses, when acting upon one material element, produce in that element the same physical effects which would occur when a test specimen of the same material is subjected to a tensile load of the same magnitude as the effective stress.

A basic mechanical design analysis method suitable for use in a computer program has been formulated. It allows the designer to analyze certain geometrical, stress and weight parameters of a given rotor with a minimum amount of input data. The results will be about 90 percent accurate.

There are many assumptions involved in this analysis, which can be understood and evaluated most easily if they are enumerated at the point where they are used. Basically, however, an "ideal" wheel of constant effective stress (no central hole) is defined by the centrifugal loads of the rotor blades, shrouds and rim as they are reacted by the wheel neck section stressed to an allowable deformation stress. The geometry, stress, and weight parameters are developed. Subsidiary calculations analyze "standard" torque tubes, "standard" inlet scrolls, and "standard" outlet scrolls. The burst stresses of the wheels and dovetail stresses are estimated. The analysis method is discussed in detail in Appendix 5.3.

The approach to turbine wheel analysis is summarized in Table 5.3-1. The classical analysis methods for constant-stress disks are utilized to define a wheel which has a stress level set by the

TABLE 5.3-1
TURBINE WHEEL ANALYSIS

- GENERAL APPROACH: TREAT AS CONSTANT-STRESS DISKS AND CORRECT FOR CENTRAL AND WEB-HOLES, THICKENED RIM, TORQUE TUBES, ETC.
- EARLY STAGES ARE CREEP-LIMITED; LATTER STAGES ARE YIELD-LIMITED (20 PERCENT OVERSPEED).
- DESPITE LOWER STRESSES, FIRST STAGES ARE DESIGN-LIMITING.
- HIGHER ROTATIVE SPEEDS ARE STRESS-LIMITING DESPITE THE SMALLER TURBINE GEOMETRY.

centrifugal loads acting on the neck section of the wheel. Corrections are made for the central hole required by a tiebolt or for the web holes required by bolted torque tubes and the resulting added stock required for reinforcement.

The early turbine stages are creep-limited, as the allowable stress material design curves showed. The latter stages are yield-limited according to the overspeed criteria previously discussed.

The early stages will have lower stresses because the pitch diameters and heights of the rotor buckets are small, but they still present the primary design problems because the allowable design stresses are low at their higher operating temperatures.

High rotative speeds present critical design problems because the centrifugal stresses increase with the square of the rotative speed, and despite the fact that the higher rotative speeds allow smaller turbine diameters.

Stress concentration factors (thickness multipliers) to be applied to ideal wheels are determined for the case of torque tubes bolted to the wheels. By increasing the ideal wheel thickness, the stresses are reduced to the allowable deformation stress of the material. The strain will be locally absorbed.

For the bolted torque tube case, eight bolts are always used, based upon the results of several iterations using multiples of four bolts. Their diameter is 0.3125 in. when the stage temperature exceeds 1820^oF (TZM stages) and 0.25 inch in the rest. The wheel thickness is increased in the plane of the bolts, based upon allowing the peak stresses around the bolt holes to reach the allowable

deformation stress of the wheel material, and assuming that stress concentration factors which exist in a plate with an infinite row of holes and in a biaxial stress field will apply to a rotating disk. Extra wheel thickness is added at smaller radii to support the added stock. It is further assumed that the torque tubes are all 0.100 inch thick. Torque tube radii and lengths are scaled or estimated from the appropriate drawing. It is assumed that torque tubes carry their own centrifugal loads (do not load the bolts).

Rotor weight and moment of inertia corrections are applied to the basic KTA mechanical analysis for bolted torque tube and curvic coupling/tiebolt cases.

For the tiebolt/curvic coupling case, it is assumed that all tiebolts are one-inch diameter. The "ideal" wheel centerline thickness is doubled for $0.5 < R < 0.7$ in order to maintain the wheel tangential stress at its "ideal" wheel value of σ_{TR} , while the wheel radial stress drops to zero at the bore. Extra thickness is added to the wheel between $0.7 < R < R_{TTI}$ (torque tube inner radius). It is assumed that satisfactory curvic coupling teeth can be fabricated. Torque tube radii and lengths are scaled or estimated from the appropriate drawing. The tiebolt is excluded from this analysis.

The stresses which are estimated by the basic mechanical analysis program for constant-stress "ideal" disks and for wheels with central tiebolt holes are compared to those calculated by the well-developed ELASTIC DISK computer program in Figure 5.3-3. The ELASTIC DISK program is used extensively in the analysis of General Electric gas turbine wheels. Sufficiently good agreement is found for both types of wheels that it may be concluded that the approximate analysis

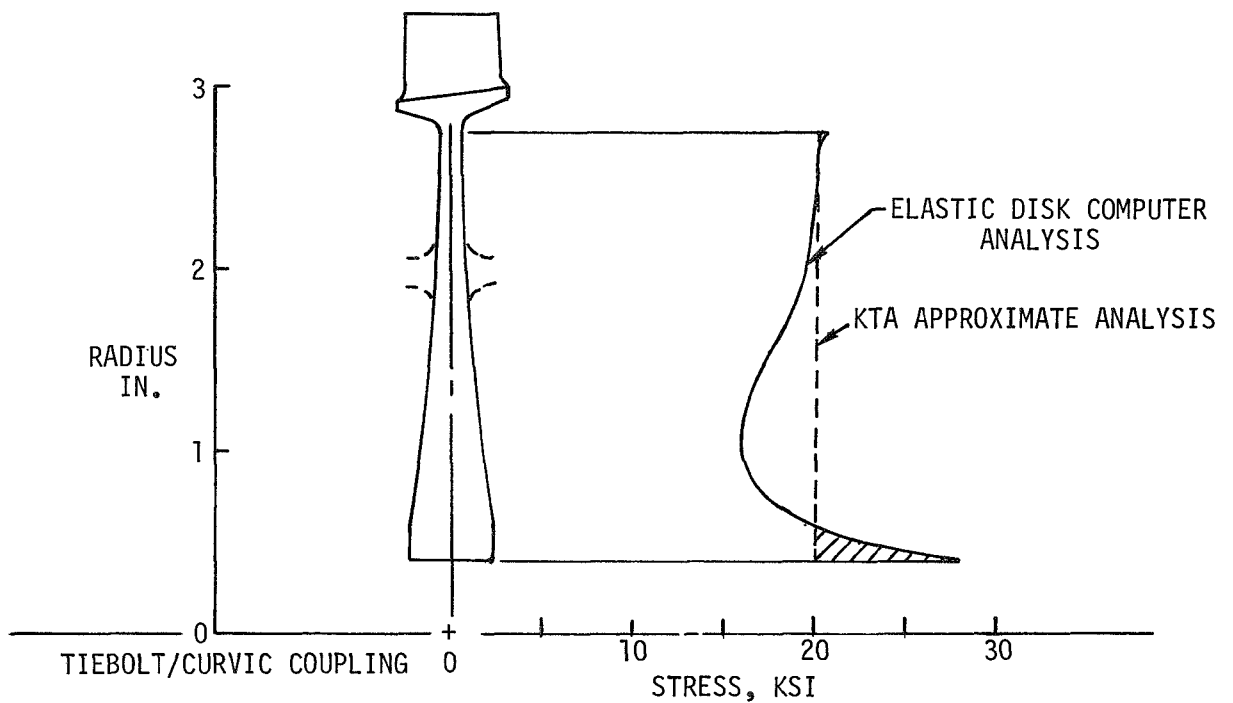
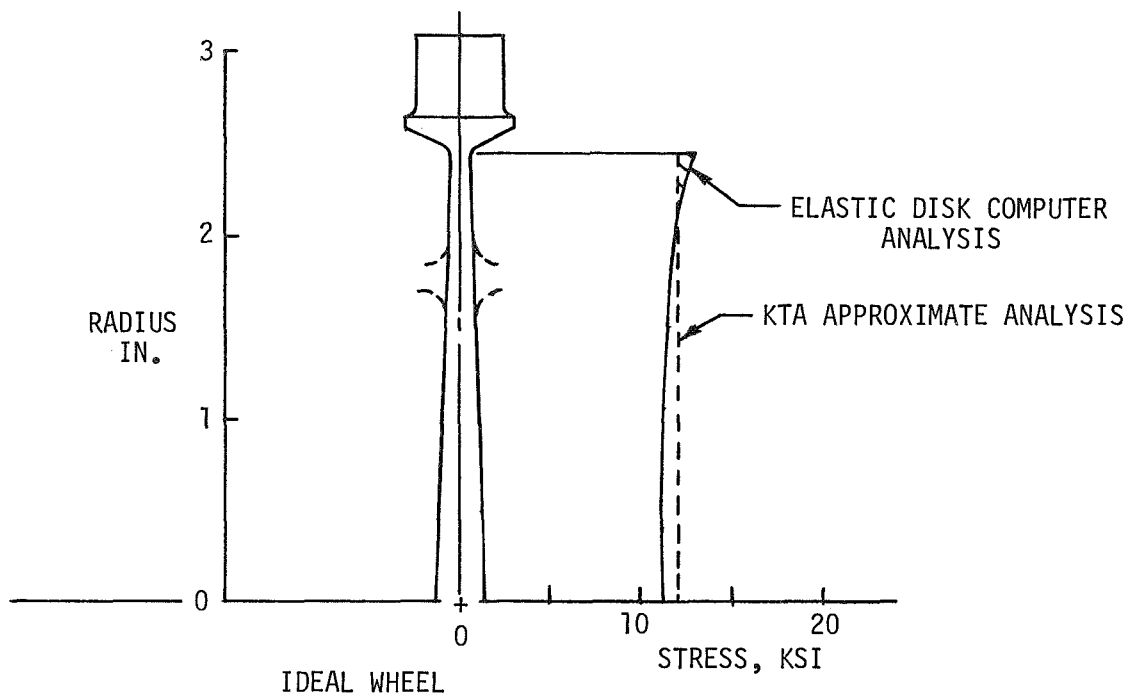


Figure 5.3-3. Comparison of KTA Mechanical Analysis Computer Program Results with "Elastic Disk" Analysis.

methods are satisfactory. The high bore stresses found in the wheel with central hole are unavoidable because the tangential stresses in this area tend toward infinity while the radial stresses drop to zero. More detailed design studies, which are beyond the scope of the general Phase I studies, will minimize the bore stresses. The practical effect of the high bore stresses is that stress redistribution will occur as the bore surface elements yield, thereby raising the wheel stresses outboard of the bore.

Stage one wheel creep rates are established by holding the hub thickness to a maximum thickness of about one inch if more satisfactory solutions are not found using the three year allowable 0.1 percent creep stress. Even though no central tiebolt hole and the related extra thickness for reinforcement is required of the ASTAR-811C material, its high density and low allowable stress in the early stages gave large centerline thicknesses. This was especially true at high rotative speeds. To find geometrically tenable solutions which did not greatly lengthen the vapor flow path, the maximum allowable hub thickness was set at one inch and a creep rate was calculated for the wheel stress giving this geometry. The creep rate was linearly interpolated from 0.1 percent and 1 percent creep stress data, and exceeded 1 percent for certain ASTAR-811C turbines. Solutions for TZC turbines were found at the 0.1 percent creep stress levels. Table 5.3-2 summarizes data which illustrates the necessity of reducing the hub thickness of the ASTAR-811C to avoid gross lengthening of the flow path. Decreasing the hub thicknesses to 1 inch, for example, produced wheel creep rates as high as 1.2 percent, which gave creep deformation as high as 25 mils at the airfoil roots.

TABLE 5.3-2

TYPICAL WHEEL STRESSES IN FIRST STAGES

Turbine No. (1)	Material	Initial Design			Redesign			
		$\sigma_{TR}^{(4)}$ ksi	$\sigma_{ac}^{(5)}$ ksi	To in.	To in.	σ_{TR} ksi	3 Yr Creep %	Creep Def. (7) Mils
3.0632	T ⁽²⁾	15.4	15.4	0.70	1.40 ⁽⁶⁾			
	A ⁽³⁾	14.7	14.7	3.20	1.00	21.7	1.2	25
3.0819	T	9.7	9.7	0.34	0.68			
	A	7.2	7.2	2.91	1.00	10.2	0.6	15
3.0824	T	10.5	10.5	0.49	0.98			
	A	8.5	8.5	3.37	1.00	12.8	0.8	17
1.0860	T	10.9	10.9	0.41	0.82			
4.0919	T	9.7	9.7	0.34	0.68			
	A	7.2	7.2	2.92	1.00	10.2	0.6	14
3.1019	T	9.7	9.7	0.34	0.68			
	A	6.6	6.6	4.04	1.00	10.2	0.7	17
3.1216	T	8.6	9.6	0.26	0.52			
	A	7.1	7.1	1.43	1.00	8.04	0.3	7
1.1219	T	9.7	9.7	0.36	0.72			
	A	7.3	7.3	2.84	1.00	10.3	0.6	17
1.1319	T	9.8	9.8	0.32	0.64			
	A	7.5	7.5	2.55	1.00	10.2	0.5	14

Notes: (1) Turbine number identification: 3.06 32

	Speed in krpm
	No. of stages
	Iteration No.

(2) TZC/TZM

(3) ASTAR-811C/FS-85

(4) Wheel tangential and radial stress

(5) Allowable deformation stress

(6) All TZC/TZM wheel hub thicknesses multiplied by two to reinforce hole required for tiebolt. This is not required for bolted torque tube design.

To illustrate the differences in the geometry of a typical ten-stage straddle-mounted turbine rotor when the two different rotor material combinations are used, Figures 5.3-4 and 5.3-5 were drawn to the same scale. The gross effects which the stage one ASTAR-811C wheel would have upon the flow path length, if the three year 0.1 percent creep stress were not exceeded, is apparent. The three inch, stage-one, hub thickness can be reduced to one inch by allowing the wheel stress to reach the level corresponding to 0.6 percent creep in three years. The thickness of the second stage is reduced to one inch maximum by designing to the 0.3 percent level. No changes are required in the remaining stages to achieve acceptable geometry with stresses corresponding to the allowable deformation stresses, which are equal to or less than the 0.1 percent creep level for three years of operation.

No manipulation of creep levels was required to achieve acceptable rotor geometry with the TZC/TZM material combination.

ASTAR-811C has definite maximum speed limitations which do not exist for the TZC material, because of the greater centrifugal stresses due to higher density and the lower allowable stresses at a given temperature. Using first stage operating conditions, the data of Figure 5.3-6 may be drawn. The 0.1 percent creep stress and 1 percent creep stress curves may be expressed as functions of speed because the stage one operating temperature varies with rotating speed and number of stages selected. They are general indicators - not exact values. The conclusion can be drawn, however, that to stay within the 1 inch centerline thickness and 1 percent creep stress criteria the operational speed should be limited to less than 24,000 rpm. Since the hub thickness limitation is not an absolute limit, the more basic criteria of blade root stress is presented. This shows that at 32,000 rpm the

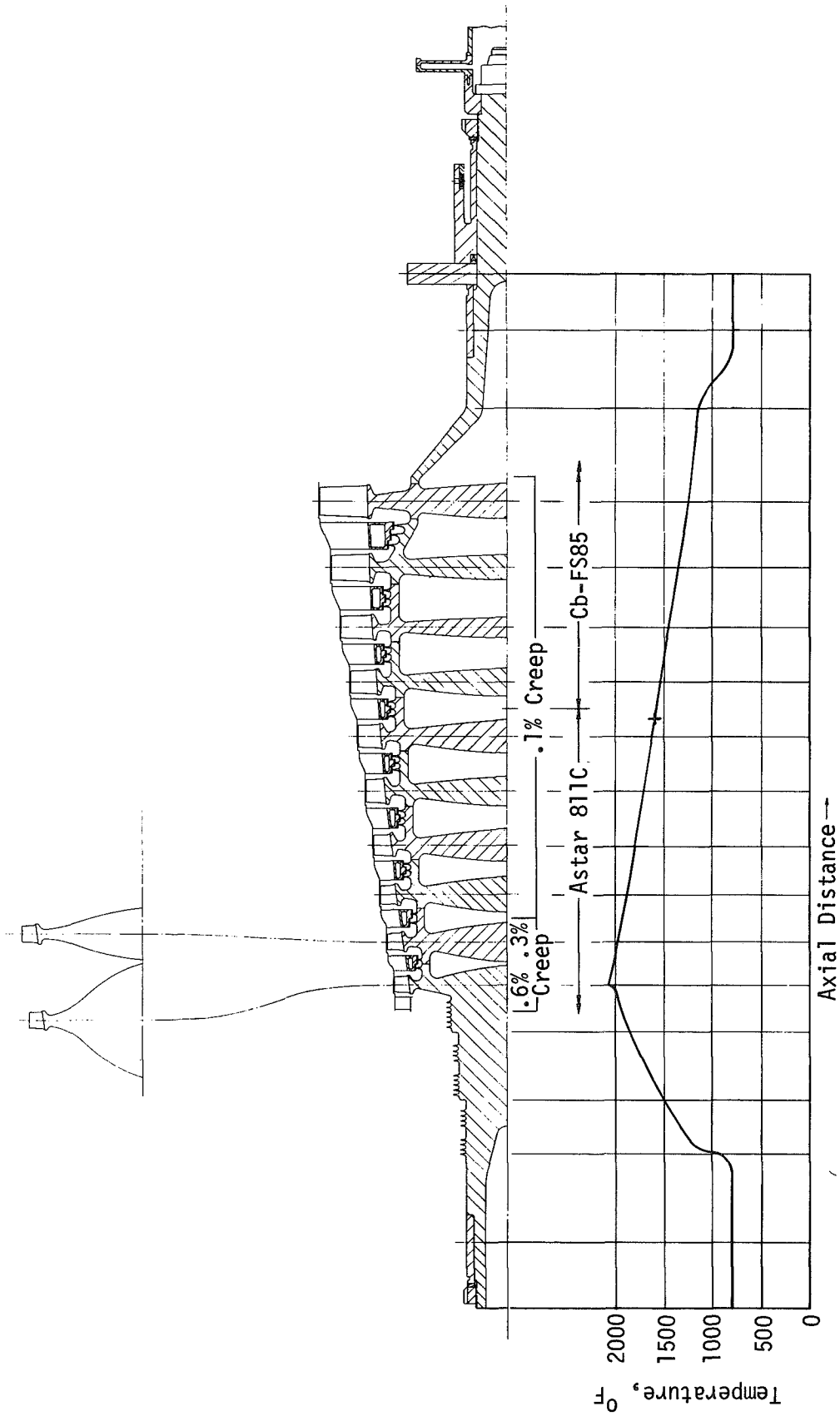


Figure 5.3-4. Effect of Designing to 0.1% Creep Stress in Stage 1 ASTAR Wheels.
(Welded Rotor)

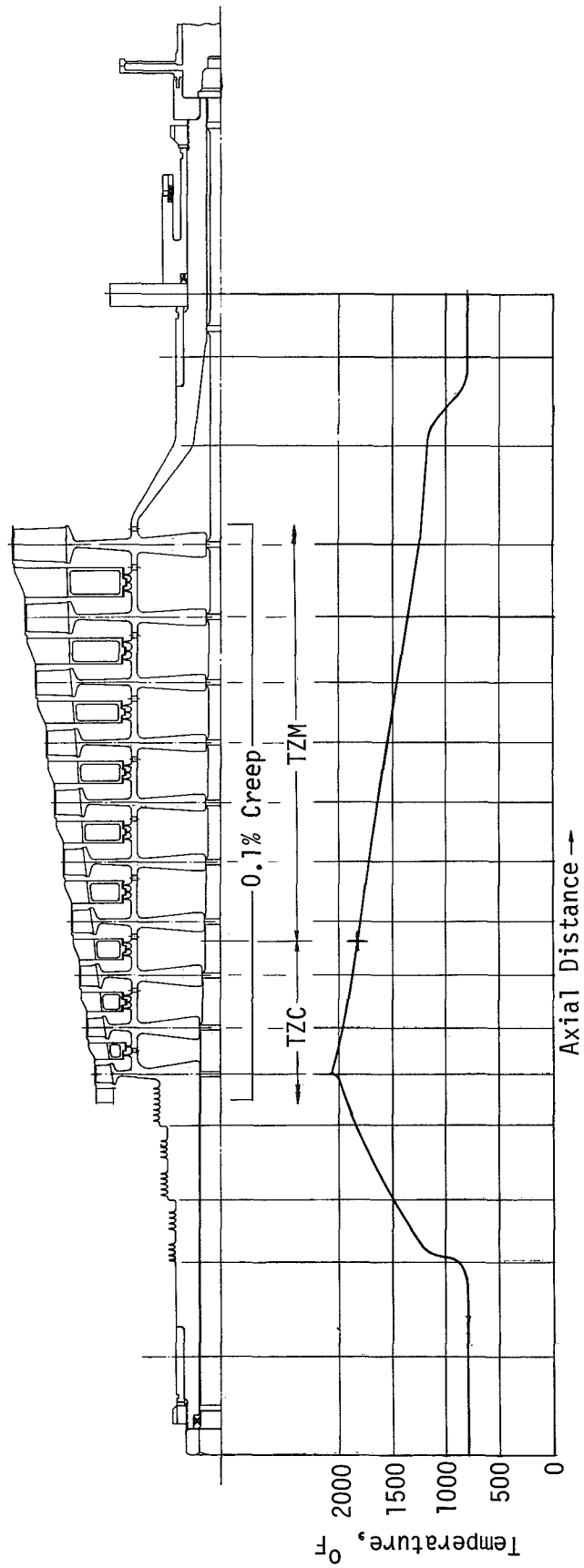


Figure 5.3-5. Effect of Designing to 0.1% Creep Stress in Stage 1 TZC Wheels.
(Bolted Rotor)

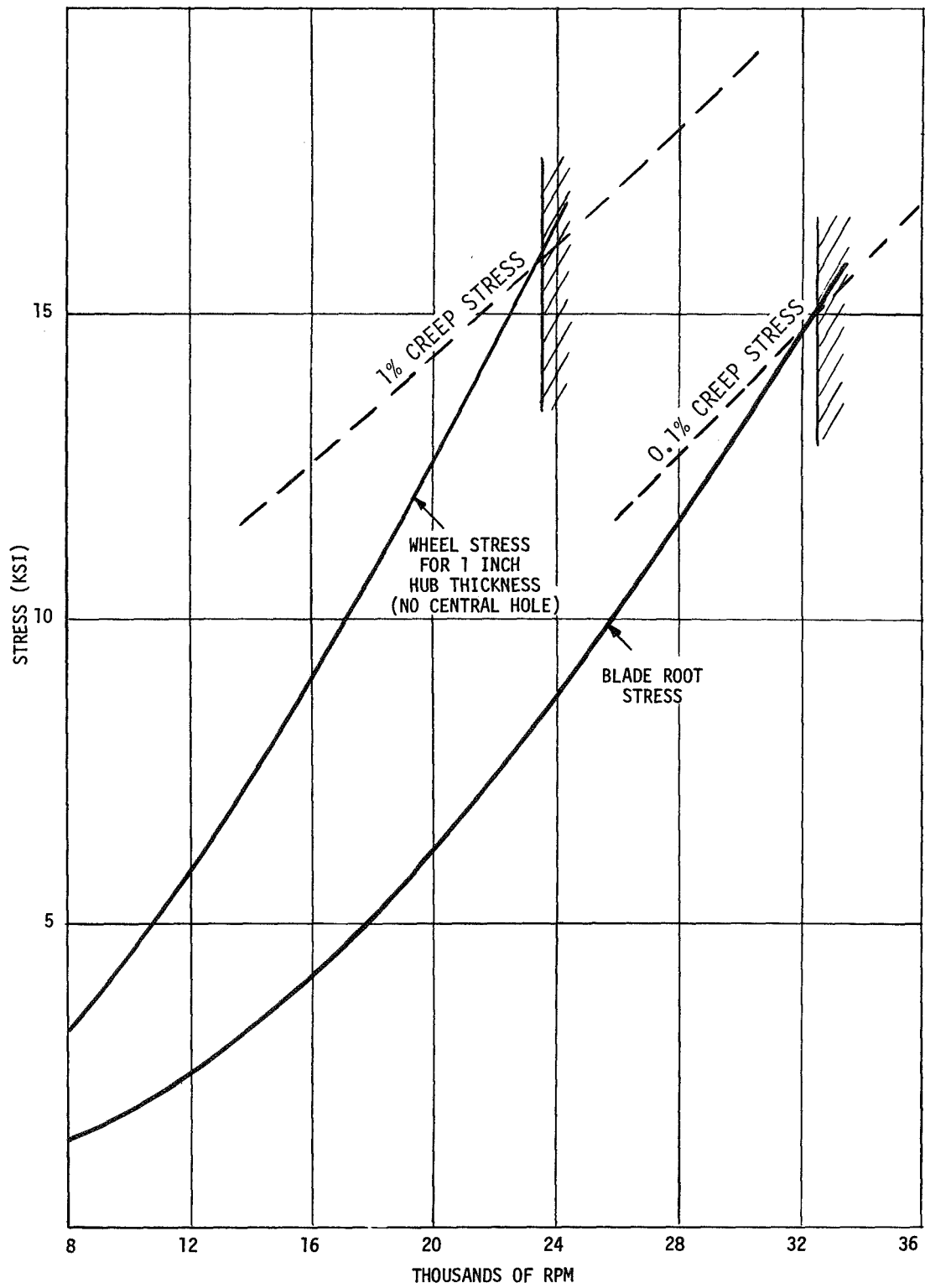


Figure 5.3-6. Limiting Operational Speeds for ASTAR 811C (Welded) Turbines.

blade root stresses reach the 0.1 percent level and this speed should not be exceeded.

For TZC material, with lower centrifugal stresses and higher allowable stresses, no apparent upper speed limitation exists within the range of rotative speeds considered, as shown by Figure 5.3-7. This is not a hard and fast rule, and every proposed turbine design must be considered on the basis of its unique operating parameters. The curve and the previous one merely illustrate in a general way the relatively greater ease of finding turbine design solutions with TZC material than with ASTAR-811C.

Also, as a matter of general interest, the effect of the 20 percent overspeed criterion was determined for the final stage of a typical nine-stage turbine operating at 19,200 rpm. FS-85 and TZM materials were used with solutions calculated at several values of overspeed. The curves of Figure 5.3-8 show that the FS-85 final stage of this turbine for 20 percent overspeed has a thickness nearly double that which is required for zero percent overspeed. However, the associated weight change amounts to only about four percent increase in the total weight of the rotor. For the TZM material, a 25 percent thickness increase occurred, which only amounted to about a one percent increase in the total rotor weight. The curves for neither material pass through the point of unity thickness ratio at zero percent overspeed because of the rule that wheel neck thickness not be allowed to fall below 0.100 inch for safety against material flaws and fabrication errors. The TZM wheel designed according to this rule required no additional thickness until the overspeed exceeded ten percent. The conclusion is that the imposition of the 20 percent overspeed rule does not invoke severe weight penalties on the turbine rotor.

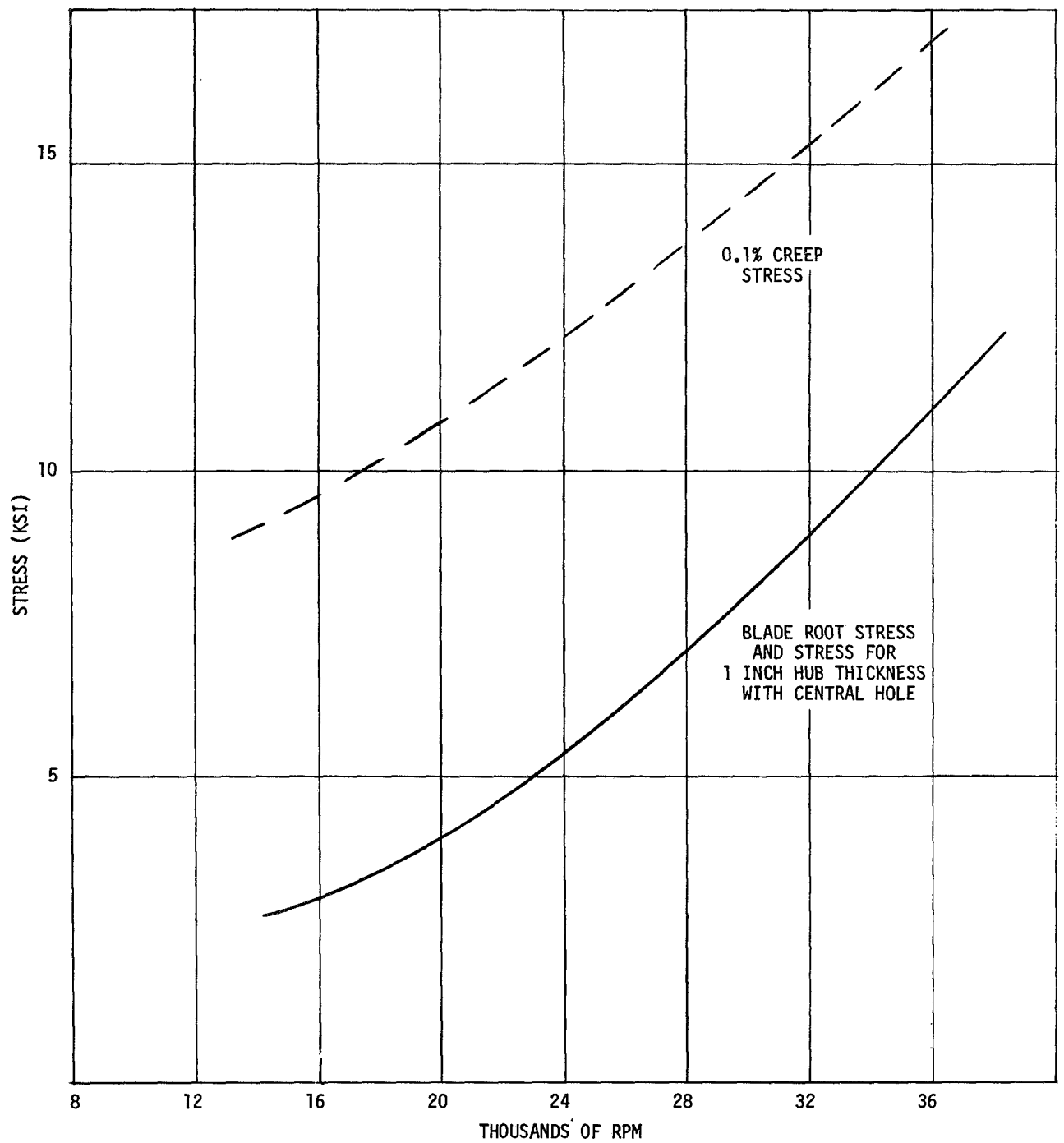


Figure 5.3-7. Limiting Operational Speed for TzC (Bolted) Turbines.

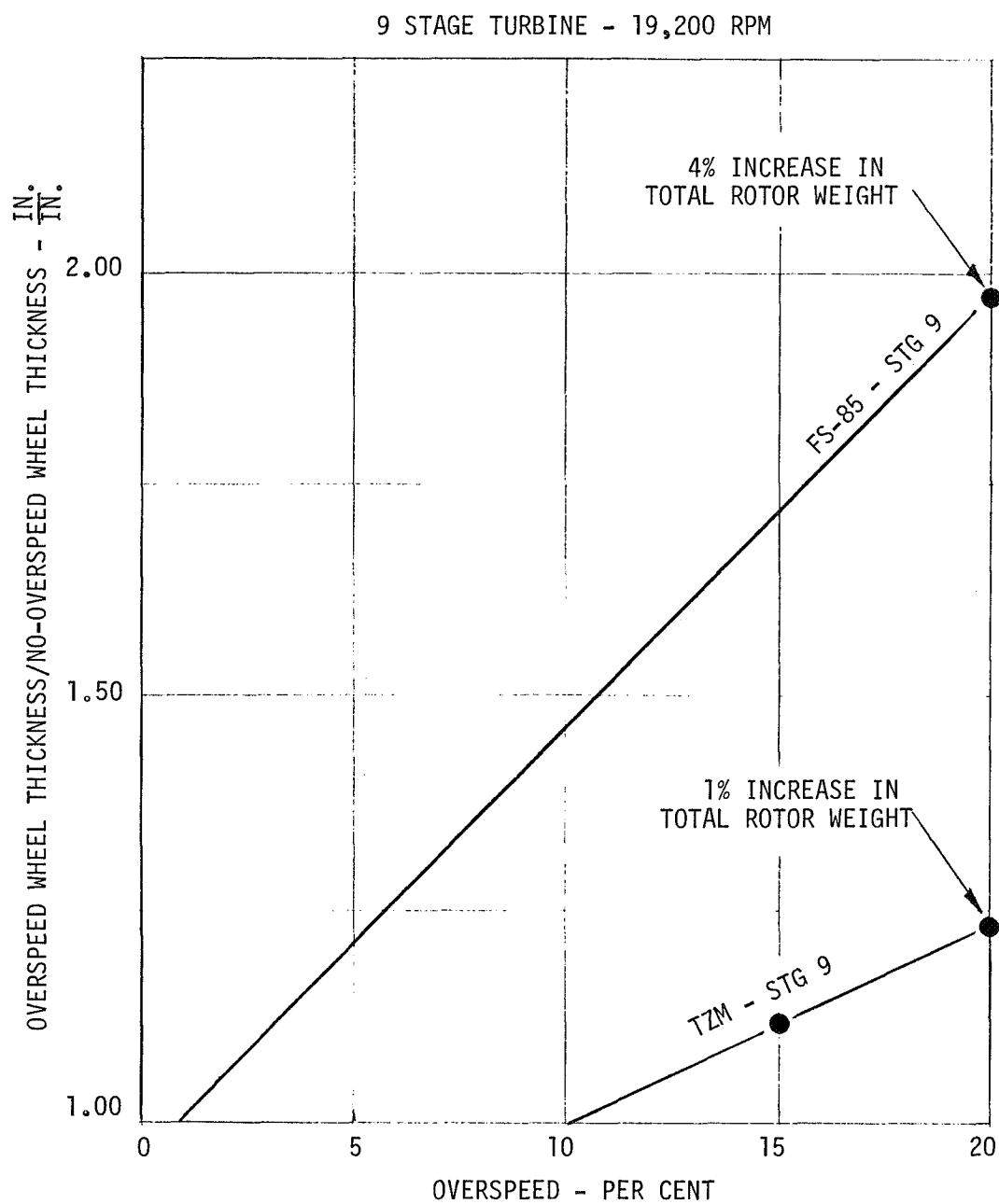


Figure 5.3-8. Effect of 20 Per Cent Overspeed on Last Stage Disk Hub Thickness.

Bucket Design

The exact analysis of stresses in the airfoils of turbine buckets is impossible in preliminary design studies. Only until the contour coordinates of the blades are fixed can the mass, area, moments of inertia, etc., be determined accurately. However, relatively accurate estimates of the principal bucket stresses may be made if the geometrical parameters of the three-stage turbine are used. The assumptions required are discussed in Appendix 5.3.

Basically, the parameter of airfoil root area/(axial chord)² is assumed, which allows the calculation of the airfoil root area when the axial chord is known. The airfoil volume may then be calculated from the flow path outer and inner diameters. A correction is applied to the weight calculation to account for the assumption of tapered blades. By assuming that the center of airfoil mass is at the center of the flow path, an airfoil centrifugal force is calculated. When the centrifugal force is divided by the root area and multiplied by a root stress concentration factor, the airfoil root stress is computed.

Generally, the airfoil root stresses are not major problems, but they do sometimes exceed the allowable deformation stresses where the allowable stresses are low because of high stage temperatures (early stages). Also, increased material density (ASTAR-811C), higher rotating speeds, and longer airfoils (final stages) produce higher centrifugal stresses which may exceed even the relatively high allowable deformation stresses of stages operating at lower temperatures.

This trend is shown in Table 5.3-3, wherein the airfoil root stresses are summarized for the initial, middle, and final stages of selected turbines. Those included in the table are generally the final iteration made for each combination of speed and number of stages, but this rule was sometimes dropped in favor of an earlier design which was analyzed for both TZC/TZM and ASTAR-811C/FS-85 material combinations. For the initial stages, the root stresses of TZC blades never reached the allowable deformation stresses by an appreciable margin. For ASTAR-811C, the margin was much closer or the allowable stresses were exceeded (3.0632, 3.0824, 1.1219). These facts lead to the conclusions of Figure 5.3-6 and 5.3-7. The middle stages of either material combination have no problems. The final stages of all TZC/TZM turbines were safe except for the peculiar turbine 1.1319, with 3.00 inch airfoil height. Several of the ASTAR 811C/FS-85 final stages (3.0632, 3.0824, 1.1319) had root stresses higher than the allowable deformation stresses.

The effects of high creep stresses in the wheels and buckets are illustrated by Figure 5.3-9, where overall creep stresses of 0.3 percent and 0.6 percent are assumed to apply to ASTAR-811C wheels and buckets, and 0.1 percent creep stress applies to TZC wheels and buckets. For a typical first stage tip radius of 3.37 inches, the 0.6 percent criterion requires that a radial tip clearance of 0.030 inch be provided at zero time (hot) in order to have the desired 0.010 inch clearance at the end of three years of operation. Similarly, the 0.3 percent criterion requires an initial radial tip clearance of 0.020 inch, and the 0.1 percent creep criterion requires that an initial radial tip clearance of 0.013 inch be provided. Power losses due to vapor leakage through the radial gaps will be proportional to the magnitude of the required

TABLE 5.3-3

TYPICAL AIRFOIL ROOT STRESSES

Turbine No. (1)	Material	First Stage		Middle Stage		Final Stage	
		σ_R (4) ksi	σ_{ac} (5) ksi	σ_R ksi	σ_{ac} ksi	σ_R ksi	σ_{ac} ksi
3.0632	T ⁽²⁾	8.9	15.4	17.9	32.1	24.3	40.9
	A ⁽³⁾	14.6	14.7	23.1	21.5	34.8	22.8
3.0819	T	3.9	9.7	5.6	19.2	17.5	41.4
	A	6.3	7.2	9.1	17.0	18.2	22.9
3.0824	T	5.4	10.5	8.8	27.6	23.3	41.1
	A	8.8	8.5	14.4	20.8	24.2	22.8
1.0860	T	7.3	10.9	12.3	29.7	28.9	39.9
4.0919	T	3.9	9.7	6.3	25.6	18.9	41.3
	A	6.3	7.2	10.4	20.2	19.6	22.9
3.1019	T	3.9	9.7	6.3	25.6	21.7	41.1
	A	6.3	6.6	10.4	20.2	22.5	22.9
3.1216	T	3.1	9.6	5.0	25.7	15.4	40.2
	A	5.0	7.1	8.1	20.3	16.0	22.5
1.1219	T	4.6	9.7	7.5	24.6	22.7	40.9
	A	7.6	7.3	12.3	19.7	20.7	22.8
1.1319	T	3.4	9.8	11.9	37.2	51.9	40.0
	A	5.6	7.5	12.4	17.7	53.9	23.0

Notes: (1) Turbine number identifies: 3. 06 32

Speed in krpm

No. of stages

Iteration No.

(2) TZC/TZM

(3) ASTAR-811C/FS-85

(4) Airfoil Root Stress

(5) Allowable Deformation Stress

radial gaps. Therefore, the TZM/TZC material combination is more desirable from this aspect.

More exact analysis of required tip clearances may be made when effective stress levels are known for each radial station in the wheel and airfoil. Creep deformations can then be calculated for finite radial elements and summed to give the total long-term deformation at the airfoil tip.

The slopes of the curves of Figure 5.3-9 are arbitrary, since they are functions of the variation of creep rate with time, which are presently undefined. If the creep rate is initially high, the radial gap will close toward 0.010 inch early in the operating life of the turbine and leakage will decrease correspondingly.

The only bending stresses which are applied to the airfoils are those due to the vapor forces of approximately two pounds per blade. Since the forces and the resulting stresses are so small, they were not analyzed for every turbine. For one typical turbine, however, these stresses were analyzed in order to ascertain orders of magnitude. Bucket heights defined by the vapor flow path dimensions were used, with airfoil moments of inertia deduced from the design of the three-stage potassium turbine.

The bending stresses calculated for the typical nine-stage, 19,200 rpm turbine chosen varied from 190 psi in stage one to 60 psi in stage five and 470 psi in stage nine. Thus, they are truly negligible in preliminary design studies even though they must be considered for a production design.

Critical frequency calculations of selected KTA turbine buckets were made to ascertain whether they will resonate in the vicinity of

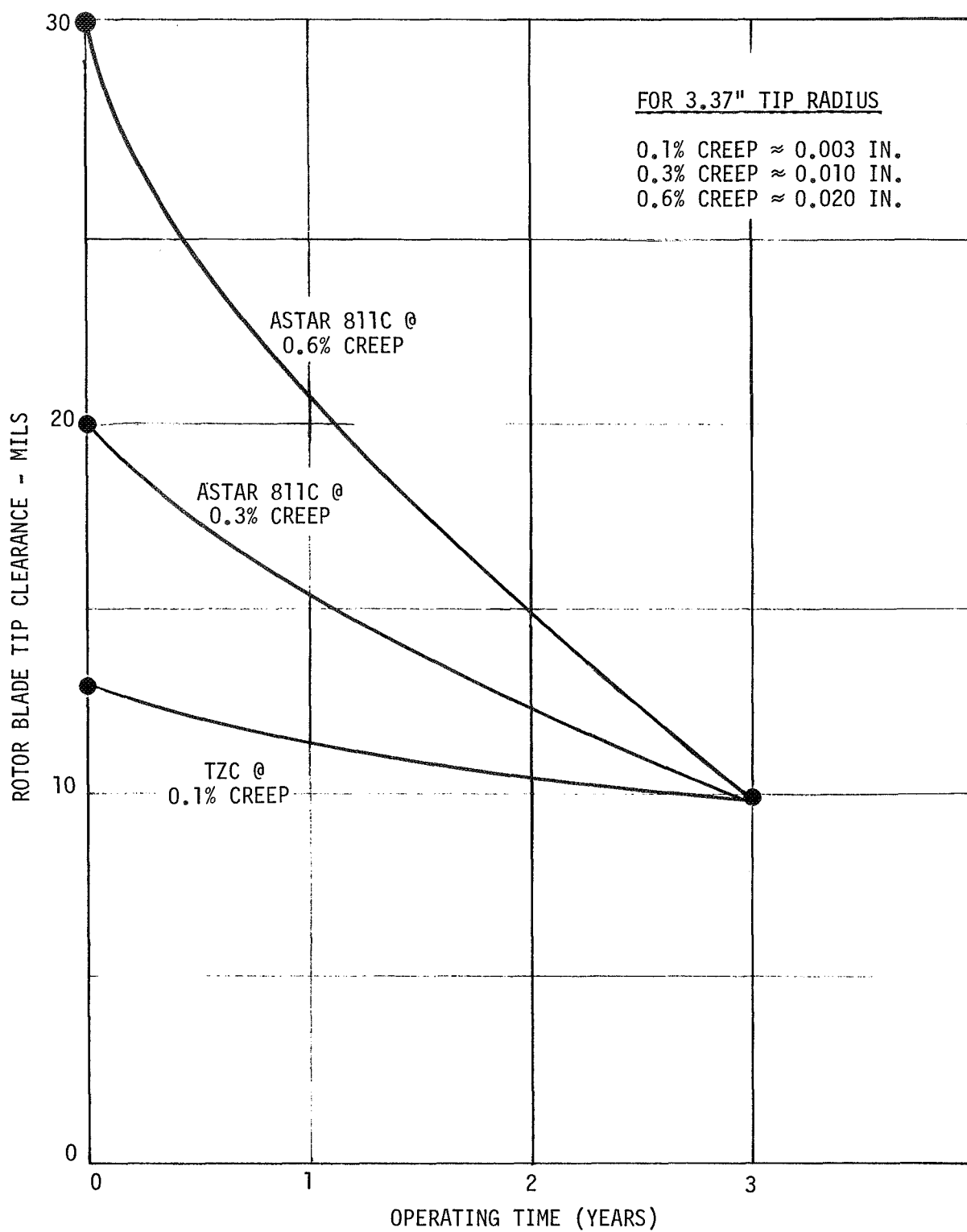


Figure 5.3-9. Rotor Tip Clearances for Stage 1 at Various Creep Rates.

the operating speed. Airfoil cross-sectional and stiffness properties were ratioed from detailed calculations made for buckets of the Three-Stage Potassium Test Turbine (NAS3-8520). The calculated natural frequencies of those buckets compared favorably with bench test results. This approach is summarized in Table 5.3-4. The formula which predicts the natural frequency of the buckets utilizes a frequency constant which forces this relationship to predict the bench-test measured frequencies. The excitation frequency equation assumes that the source of the excitation will be the nozzle vanes immediately upstream of the rotor buckets.

Figures 5.3-10 and 5.3-11 show the results of turbine blade critical speed calculations for four turbines of most interest. In order to visualize the proximity of the resonant condition, curves of excitation frequency are added. Resonant operation occurs only at the second blade critical frequency for stage four of 47R199284 and at the third blade critical frequency for stage five of 47R199284. All other stages of the turbines considered have considerable margins of safety, as can be seen by the distance separating the blade critical frequencies from the excitation frequencies. Therefore, in general, no blade resonant operation problems exist.

When close proximity of blade critical frequency and excitation frequency exist, there are at least two courses to follow - the number of nozzle vanes can be changed slightly, or their spacing may be staggered to produce more excitation frequencies of lower energy. The latter system has been successfully used on the three-stage potassium turbine and others.

TABLE 5.3-4
BLADE VIBRATION ANALYSIS

ANALYTICAL TECHNIQUE

RATIO BLADE CROSS-SECTIONAL AND STIFFNESS PROPERTIES FROM
DETAILED BLADES OF THREE-STAGE POTASSIUM TURBINE.

ASSUMPTIONS

NO. OF VANES = NO. OF BUCKETS

$$f_n = C_n \frac{r}{l^2} K_m \times 10^4$$

f_n = nTH NATURAL FREQUENCY, HZ
 C_n = FREQUENCY CONSTANT
 r = BLADE RAD. OF GYRATION
 l = BLADE HEIGHT
 K_m = MATERIAL CONSTANT

EXCITATION FREQUENCY

$$f_e = \frac{\text{RPM (NO. VANES)}}{60}$$

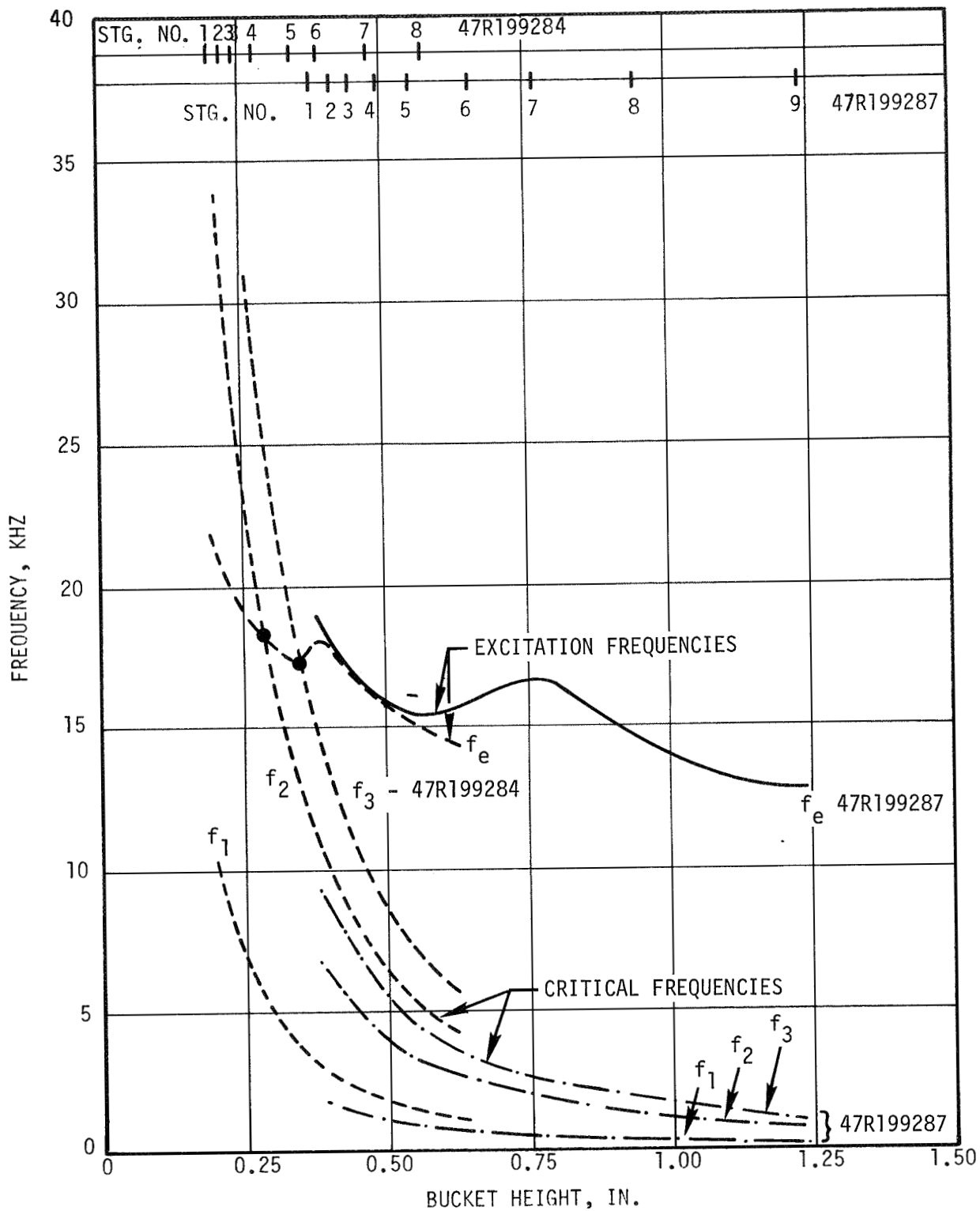


Figure 5.3-10. Rotor Bucket Critical Frequencies for KTA Turbines- 47R199284 (24,000 RPM, Single Spool, Split Flow, 8 Stage, TZC/TZM) and 47R199287 (19,200 RPM, Single Spool, Overhung, 9 Stage, ASTAR 811C/FS-85).

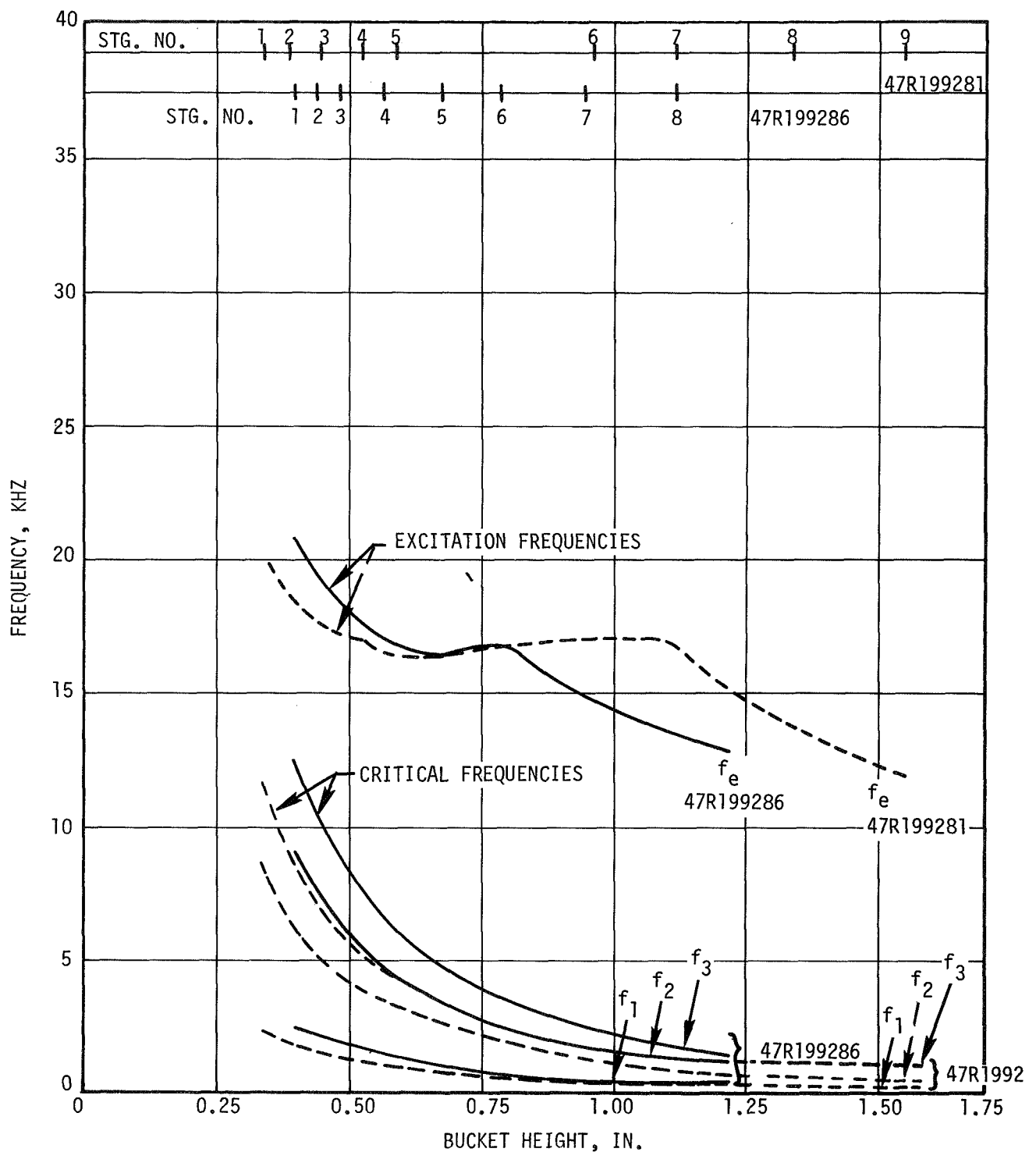


Figure 5.3-11. Rotor Bucket Critical Frequencies for KTA Turbines- 47R199281 (19,200 RPM, Split Spool, Counterflow, 5 + 4 Stage, ASTAR 811C/FS-85) and 47R199286 (24,000 RPM, Split Spool, Counterflow, 5 + 3 Stage, TZC/TZM).

Bucket Connections

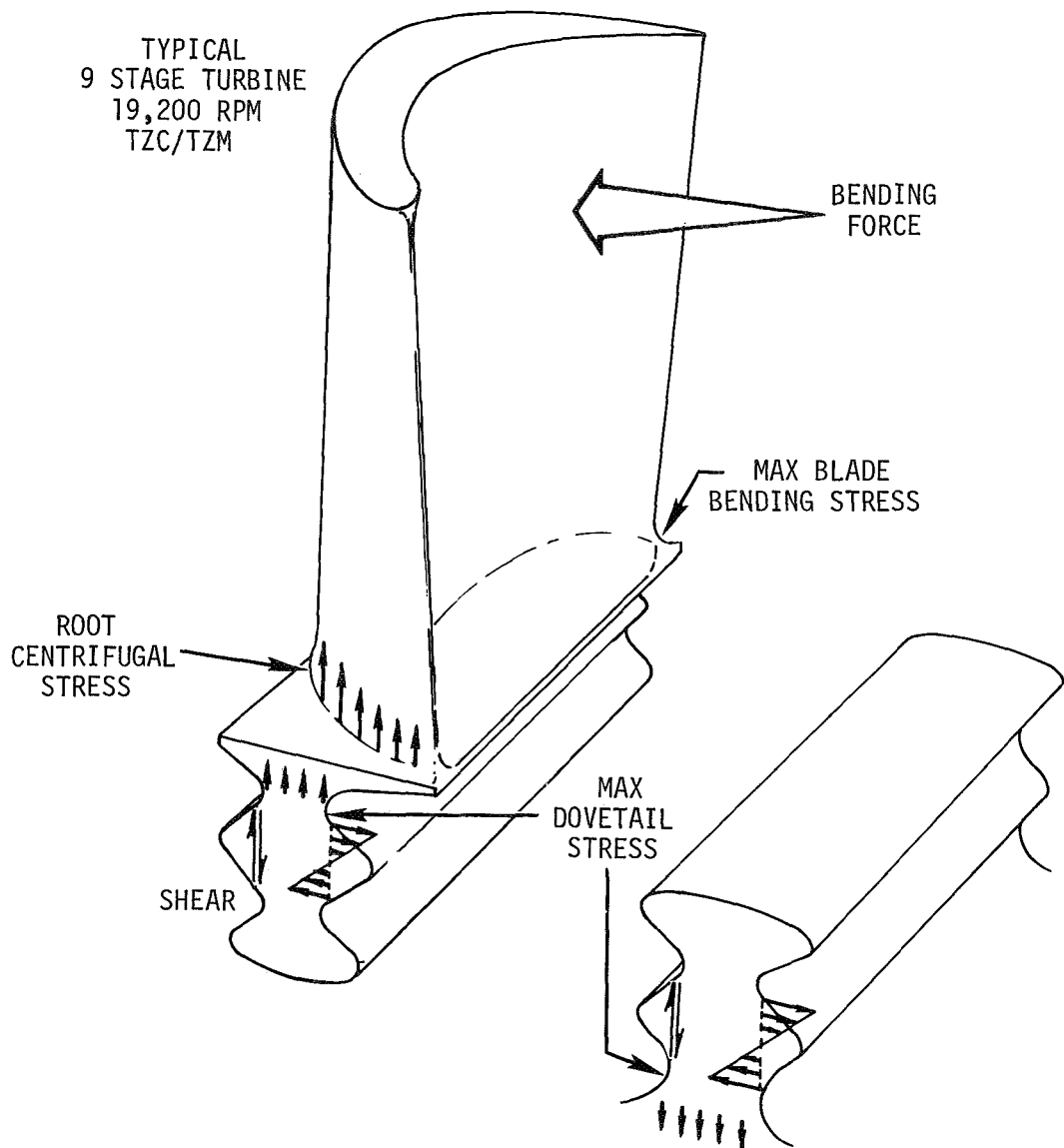
The major stresses in the buckets are in the dovetail region, and Figure 5.3-12 shows the general distribution of these stresses. Maximum stresses occur at the inner curvature of the top tang and in the turbine wheel, at the lowest neck section of the dovetail. These stresses are a combination of direct tensile stress, shear stress, and bending stress, considering the dovetail tang as a triangular beam. Analysis of the dovetail stresses is performed by a combination of graphical and analytical procedures. The dovetail is accurately described on a layout drawing where all dimensions and contact angles are determined. Then, based upon an analytical procedure (derived in part, from experimental stress analyses and employing techniques developed for, and used extensively in aircraft gas turbine design), the combined stresses are calculated. The wheel dovetail load is always greater than that of the bucket since the wheel carries not only the entire bucket, but also the stub section of the wheel between blades.

In order to simplify the analysis, stresses from the Three-Stage Potassium Test Turbine (NAS3-8520) were ratioed according to blade centrifugal force and geometry. A factor of safety is calculated for each of these stresses with respect to the allowable rupture stress, assuming that the stresses will redistribute as yield occurs:

$$\text{Factor of safety} = \frac{\text{Allowable rupture stress}}{\text{Combined stresses on blade dovetail top tang}}$$

$$\text{Factor of safety} = \frac{\text{Allowable rupture stress}}{\text{Combined stresses on wheel dovetail bottom tang}}$$

These criteria are used for judging the feasibility of using dovetails.



	STAGE 1	STAGE 5	STAGE 9
MAX DOVETAIL STRESS (PSI)	14,390	26,030	68,680
AIRFOIL ROOT CENTRIFUGAL STRESS (PSI)	3700	6750	17,900
BLADE ROOT BENDING STRESS (PSI)	190	60	470
BLADE HEIGHT (IN.)	0.38	0.56	1.24
ALLOWABLE FRACTURE STRESS (PSI)	15,900	41,230	43,810

Figure 5.3-12. Representative Dovetail Stresses.

In general, the last few stages of many turbines, especially those operating at high speed and of FS-85 material, have factors of safety less than one because of high centrifugal stresses. Dovetails are also ruled out in the early stages due to high temperature and size so small that minor discrepancies would produce catastrophic failure. They could, however, be used in intermediate stages. Shrouded buckets were eliminated from the study, since moisture extraction from rotor buckets was not considered. Since the use of dovetails is required only by the shrouded buckets, because of the complexity of fabricating integral shrouded buckets, no intensive efforts were made to analyze dovetail stresses in each turbine design considered beyond the initial basic mechanical design screening analysis. These conclusions are summarized in Table 5.3-5.

In conclusion it may be said that in all of the various turbine designs studied, all of the first two stages suffer because of the 2100⁰F inlet temperature. In the welded designs using ASTAR-811C, the 0.1 percent creep objective could not be met with reasonable wheel geometry. Instead, three year creep rates of approximately 0.6 percent in the first stage and 0.3 percent in the second stage had to be allowed for 19,200 rpm operation. A leakage penalty is imposed because of the larger initial tip clearances required. For 24,000 rpm or higher, the solutions get more difficult and the compromises less desirable.

The rotors using the molybdenum alloys TZC and TZM are lighter because of higher material strength. The hot stage solutions are easier because of lower centrifugal stresses, due to lower material density than ASTAR-811C. It was possible to find acceptable solutions at 0.1 percent creep in three years.

TABLE 5.3-5
DOVETAIL STRESS

- DOVETAIL STRESS

ANALYTICAL TECHNIQUE

RATIO STRESSES FROM THREE-STAGE POTASSIUM TURBINE DOVETAILS
ACCORDING TO BLADE CENTRIFUGAL FORCE AND GEOMETRY.

RESULTS

DOVETAILS CANNOT BE USED IN EARLY STAGES DUE TO HIGH TEMPERATURES
AND SIZE AND IN LAST STAGES DUE TO HIGH STRESS. THEY WILL WORK
IN INTERMEDIATE STAGES.

SINCE NOT NEEDED FOR MOISTURE-EXTRACTION SHROUDED BLADES, THEY
ARE ELIMINATED FROM THE STUDY.

- BLADE BENDING STRESS

TANGENTIAL FORCES OF ABOUT 2 LBS PER BLADE PRODUCE NEGLIGIBLE
ALTERNATING BENDING STRESSES OF A FEW HUNDRED PSI.

Even the lower airfoil root stresses are a problem in the ASTAR-811C/FS-85 rotors. The feasibility of using dovetail connections between the buckets and wheels is low except for certain intermediate turbine stages. Bucket vibration is not a major problem.

5.3-A APPENDIX

Basic Mechanical Design Analysis

The basic mechanical design analysis tool is a PANACEA computer program, which is one of a type designed to allow direct user programming of an advanced computer. It is analogous to a Data-Net program with respect to simplicity of programming instructions, but requires input by cards. The first set of a series of calculations is printed in sufficient detail to show every calculation step which was made, so that the system may be checked for error. Thus, for a multi-stage turbine, every detail of the calculations for the first stage may be easily verified.

The program is designed to operate with a minimum of fluid design input data.

For rotor analysis, the following information is required:

Rotor material density, ρ

Axial chord of buckets, c

Rotative speed, KRPM

Bucket root radius, R_R

Bucket tip radius, R_T

Bucket weight factor, ψ

Bucket solidity, $\bar{S}$

Bucket root area/chord factor, A_R/c^2

Shroud indicator

Dovetail indicator

Bucket factor of safety

Stage temperature, t

Material allowable stresses as functions of temperature

Rotor material Poisson's ratio, μ

Torque tube half-thickness, $H_{TT}/2$

The blade root stresses σ_R are expressed in terms of blade material density ρ , stage annulus area A_{an} , and rotative speed in thousands of rpm (KRPM):

$$\sigma_R = 4.2262 \rho A_{an} (\text{KRPM})^2 \quad \text{psi}$$

where the blade dimensions have been converted to terms of annulus area, a root stress concentration factor of 1.1 has been applied, and a tapered blade has been assumed which produces root stresses only 80 percent of those of an untapered blade. See the Nomenclature section of this Appendix for complete definition of terms.

The number of nozzle vanes N_n is estimated as a function of the vane pitchline radius R_R (assumed equal to the root radius of the following buckets), the axial chord c , and a solidity factor $\bar{S}$ of 2/3:

$$N_n = \frac{2\pi R_R}{c\bar{S}} = 9.4248 \frac{R_R}{c} \quad (-)$$

The number of rotor buckets N_B is calculated from the average aero-thermodynamic pitchline diameter $\frac{D_{P1} + D_{P2}}{2}$, the rotor axial chord c , and the aero-thermodynamic solidity factor $\bar{S}$:

$$N_B = \frac{\pi}{2} \frac{(D_{P1} + D_{P2})}{c\bar{S}} \quad (-)$$

The blade root area A_R is derived from an estimated ratio of root area to the square of the chord c :

$$A_R = \left(\frac{A_R}{c^2}\right) c^2 = 0.2363 c^2 \quad \text{in.}^2$$

The centrifugal force of one bucket ${}_1F_{CB}$ is expressed in terms of a weight multiplier of 1.05 (for fillets), the root stress σ_R , and the root area calculated above:

$${}_1F_{CB} = 1.05 \sigma_R (0.2363) c^2 = 0.24812 \sigma_R c^2 \quad 1b$$

And the centrifugal force of all buckets F_{CB} is the above multiplied by the number of buckets N_B :

$$F_{CB} = 0.24812 \sigma_R N_B c^2 \quad 1b$$

The centrifugal force F_{cs} of all shrouds of a stage with tip shrouds is estimated by using 25 percent of the weight of all buckets applied at the bucket tip radii R_T , which requires correcting the centrifugal force of the blades by dividing by their centroid radius

$$\frac{R_T + R_R}{2}:$$

$$F_{cs} = W_{SHR} \frac{R_T \omega^2}{g} = 0.25 F_{CB} R_T \left(\frac{2}{R_T + R_R}\right) = \frac{F_{CB}}{2} \left(\frac{R_T}{R_T + R_R}\right) \quad 1b$$

A wheel rim height H_R of 0.200 in. is used if the blades are integral with the wheel, or 1 in. if dovetails are present. Therefore, the wheel neck radius R_N is the bucket root radius R_R minus the wheel rim height H_R :

$$R_N = R_R - H_R \quad \text{in.}$$

The centrifugal force F_{CPDR} of the rim is estimated by using an empirical relationship derived from the geometrical parameters of the three-stage turbine, which accounts for the material removed by the contour of the wheel between the thin neck section and the ID of the rim:

$$F_{CPDR} = 46.808 \rho c (R_R^2 - R_N^2)(R_R + R_N)(KRPM)^2 \quad 1b$$

The subscript indicates that the weights of the blade platforms, dovetails, and rim are included.

The total centrifugal load on the wheel neck section F_N is the sum of the above plus that of the blades F_{CB} and shrouds F_{CS} :

$$F_N = F_{CPDR} + F_{CB} + F_{CS} \quad 1b$$

A correction is required when calculating wheel neck thickness T_N from the total centrifugal load on the neck F_N , because that portion of the rim-height H'_R which is not cut by dovetail slots is self-supporting.

$$T_N = \frac{F_N}{2\pi \sigma_{ac} (R_N + H'_R \mu)} - \frac{H'_R c(1-\mu)}{(R_N + H'_R \mu)} \quad \text{in.}$$

where σ_{ac} is the allowable deformation stress and μ is Poisson's ratio for the rotor material.

A lower limit of 0.100 in. is placed on the neck thickness T'_N to be used so that minor material flaws or fabrication irregularities do not result in catastrophic failure:

$$T_N > T'_N > 0.100 \quad \text{in.}$$

Since the neck stress is held constant throughout the wheel for an ideal disk, the wheel radial and tangential stresses σ_{TR} are equal at the value of allowable deformation stress σ_{ac} times the ratio of neck thickness used T'_N divided by the calculated thickness required T_N :

$$\sigma_{TR} = \sigma_{ac} \left(\frac{T'_N}{T_N} \right) \quad \text{psi}$$

The ideal wheel geometry is derived from the relationship of previously derived parameters. The hub thickness T_O and any other thickness T_x are functions of the wheel neck thickness T_N and exponential factors:

$$T_O = T_N e^{aR_N^2} \quad \text{in.}$$

$$T_x = T_O e^{-aR_x^2} \quad \text{in.}$$

$$\text{where } a = \frac{\rho \omega^2}{2g\sigma_{TR}} = \frac{14.190 \rho (\text{KRPM})^2}{\sigma_{TR}}, \quad R_N = \text{neck radius, } R_x$$

= any other radius, and e = base of natural logarithms.

The ideal disk weight W_{DISK} is calculated by integration of the thickness equation from the wheel centerline to the neck radius:

$$W_{DISK} = \frac{2\pi g \sigma_{TR} T'_N}{\omega^2} \left[e^{aR_N^2} - 1 \right] \quad \text{lb}$$

The rim weight W_{RIM} is the rim volume times its density ρ :

$$W_{RIM} = \pi \rho c \left[R_R^2 - R_N^2 \right] \quad \text{lb}$$

The weight of one bucket airfoil ${}_1W_B$ is its density ρ times its height H_B times its root area A_R times a volume factor of 0.85 for tapered blades:

$${}_1W_B = 0.85\rho H_B \left(\frac{A_R}{2}\right) c^2 \quad 1b$$

And the weight of all airfoils W_{BL} is the above times the number of buckets N_B :

$$W_{BL} = {}_1W_B N_B \quad 1b$$

The shroud weight W_{SHR} is assumed to be 25 percent of the airfoil weight:

$$W_{SHR} = 0.25 W_{BL} \quad 1b$$

The total wheel weight $W_{W\ TOT}$ is the sum of the disk, rim, bucket, and shroud weights:

$$W_{W\ TOT} = W_{DISK} + W_{RIM} + W_{BL} + W_{SHR} \quad 1b$$

Polar moments of inertia are calculated:

for the rim

$$I_{RIM} = \frac{W_{RIM}}{2} (R_R^2 + R_N^2) \quad 1b-in.^2$$

for the blades and shrouds

$$I_{BLSH} = (W_{BL} + W_{SHR}) \left(\frac{R_T + R_R}{2}\right)^2 \quad 1b-in.^2$$

for the disk

$$I_{DISK} = \frac{\pi\rho}{4} (T_O + T'_N) R_N^4 \quad 1b-in.^2$$

and the total is

$$I_{TOT} = I_{RIM} + I_{BLSH} + I_{DISK} \quad \text{lb-in.}^2$$

The wheel radius of gyration is:

$$k = \sqrt{\frac{I_{TOT}}{W_{TOT}}} \quad \text{in.}$$

The weight of one blade dovetail neck section ${}_1W_{DN}$ with geometry similar to that of the three-stage potassium turbine is a function of the weight of one rotor bucket ${}_1W_B$:

$${}_1W_{DN} = 0.07569 \quad {}_1W_B \quad \text{lb}$$

Which gives a centrifugal force ${}_1F_{CDN}$, assuming its mass is concentrated 0.2 in. below the bucket root:

$${}_1F_{CDN} = {}_1W_{DN} (R_R - 0.2) \frac{\omega^2}{g} \quad \text{lb}$$

The circumferential length L_{DN} available for the upper blade dovetail neck is estimated as 27.9 percent of the circumference of the wheel at 0.1 in. below the bucket root, based upon three-stage turbine experience:

$$L_{DN} = \frac{2\pi(R-0.1)(0.279)}{N_B} = \frac{1.75(R-0.1)}{N_B} \quad \text{in.}$$

Similarly to the reasons for a lower limit on wheel neck thickness, it is considered that dovetails are not feasible if $L_{DN} < 0.100$ in.

The dovetail neck will have an area ${}_1A_{DN}$ of 90 percent efficiency in carrying loads because of canting the dovetails in the wheels:

$${}_1A_{DN} = 0.9 \quad c \quad L_{DN}^2 \quad \text{in.}^2$$

The dovetail neck stress σ_{DN} is the sum of the blade, dovetail neck section, and shroud centrifugal forces divided by the neck effective area:

$$\sigma_{DN} = \frac{l_{CB}^F + l_{CDN}^F + l_{CS}^F}{0.9 \ c \ L_{DN}} \quad \text{psi}$$

But, according to the three-stage turbine dovetail stress analyses, the combined stress σ_{CBDT} at the blade dovetail top tang is 1.93 times the neck stress and the combined stress σ_{CWDB} at the wheel dovetail bottom tang is 2.38 times the neck stress:

$$\sigma_{CBDT} = 1.93 \ \sigma_{DN} \quad \text{psi}$$

$$\sigma_{CWDB} = 2.38 \ \sigma_{DN} \quad \text{psi}$$

These stresses are compared to the allowable fracture stress, assuming that stress redistribution will occur with yielding.

The area A_{HB} of an ideal disk which resists bursting is derived by integration of the wheel thickness formula, considering the uncut area of the wheel rim:

$$A_{HB} = T_o \sqrt{\frac{\pi}{a}} + 0.3 \ c - T_N \quad \text{in.}^2 \text{ if no dovetails present}$$

$$A_{HB} = T_o \sqrt{\frac{\pi}{a}} + 0.3 \ c - 0.1 \ T_N \quad \text{in.}^2 \text{ if dovetails present}$$

The centrifugal force F_{CH} of half of the wheel rotating about its own center of gravity is:

$$F_{CH} = \frac{W_{TOT} \ k \ \omega^2}{4g} \quad \text{lb}$$

When the centrifugal forces of the dovetails, buckets, and shrouds are added, the bursting stress σ_{HB} becomes:

$$\sigma_{HB} = \frac{F_{CH} + \frac{F_{CD} + F_{CD} + F_{CS}}{\pi}}{A_{HB}} \quad \text{psi}$$

since it is necessary to convert all centrifugal forces into their tangential components.

The bursting stress is compared to the allowable fracture stress.

The area moment of inertia I_{AW} of the wheel is calculated:

$$I_{AW} = \frac{\pi R_R^4}{4} \quad \text{in.}^4$$

The wheel shear area A_{SW} is:

$$A_{SW} = \pi (R_N + 0.3 H_R)^2 \quad \text{in.}^2$$

The wheel transverse moment of inertia I_{TW} is:

$$I_{TW} = \frac{W_{RIM}}{12} \left[c^2 + 3(R_N + 0.3H_R)^2 + 3R_N^2 \right] + \frac{W_{DISK}}{12} \left[\left(\frac{T_N + T_o}{2} \right)^2 + 3R_N^2 \right] \quad \text{lb-in.}^2$$

Torque tube weights and moments of inertia are calculated, as are several of the characteristic dimensions of the "standard" inlet and outlet scrolls and their weights.

The computer output is identified by a unique turbine number

such as

1	. 08	20	
			operating speed, krpm
			number of stages
			design iteration number

Nomenclature

The following nomenclature, arranged alphabetically by symbol, is used throughout the development of the KTA mechanical design computer program formulae.

<u>Symbol</u>	<u>Definition</u>	<u>Units</u>
a	Factor for computing ideal wheel thickness.	--
A_{AN}	Stage annulus area.	in. ²
$1 A_{DN}$	Area of one dovetail neck section.	in. ²
A_{HB}	Area resisting bursting of "ideal" wheel.	in. ²
A_R	Blade root area.	in. ²
$\frac{A_R}{2c}$	Blade root area/chord factor.	--
A_{SW}	Shear area of "ideal" wheel.	in. ²
c	Rotor bucket axial chord.	in.
D_{P1}	Inlet pitch diameter of rotor bucket	in.
D_{P2}	Outlet pitch diameter of rotor bucket	in.
e	2.71828, base of natural logarithms.	--
F_{CB}	Centrifugal force of all buckets of a rotor stage.	lb
$1 F_{CB}$	Centrifugal force of one bucket of a rotor stage.	lb
$1 F_{CDN}$	Centrifugal force of one dovetail neck section.	lb
F_{CPDR}	Centrifugal force of all platforms, dovetails, and rim of a wheel.	lb
F_{CS}	Centrifugal force of all shrouds of a wheel.	lb
F_N	Total centrifugal force on a wheel neck section.	lb

<u>Symbol</u>	<u>Definition</u>	<u>Units</u>
g	Gravitational constant, 386.4	in./sec^2
H_B	Rotor bucket height.	in.
H_R	Total wheel rim height.	in.
H'_R	Uncut wheel rim height.	in.
I_{AW}	Area moment of inertia of wheel.	in.^4
I_{BLSH}	Polar moment of inertia of rotor blades and shrouds.	lb-in.^2
I_{DISK}	Polar moment of inertia of "ideal" wheel disk section.	lb-in.^2
I_{RIM}	Polar moment of inertia of wheel rim section.	lb-in.^2
I_{TOT}	Total polar moment of inertia of an "ideal" wheel.	lb-in.^2
I_{TW}	Total transverse moment of inertia of "ideal" wheel.	lb-in.^2
k	Radius of gyration of an "ideal" wheel.	in.
KRPM	Rotative speed.	$\frac{\text{RPM}}{1000}$
L_{DN}	Wheel circumference available for dovetail neck.	in.
N_B	Number of rotor buckets.	--
N_n	Number of nozzle vanes.	--
R_N	Radius of the neck of a wheel.	in.
R_R	Rotor bucket root radius.	in.
R_T	Rotor bucket tip radius.	in.
$\bar{S}$	Blade solidity.	--
T_N	Calculated wheel neck section thickness.	in.
T'_N	Wheel neck section thickness used.	in.

<u>Symbol</u>	<u>Definition</u>	<u>Units</u>
T_O	"Ideal" wheel hub (ϕ) thickness.	in.
T_x	"Ideal" wheel thickness at any radius R_x .	in.
W_{BL}	Weight of all rotor buckets.	lb
$1 W_B$	Weight of one rotor bucket.	lb
W_{DISK}	Weight of an "ideal" wheel disk from $R = 0$ to $R = R_N$.	lb
$1 W_{DN}$	Weight of one dovetail neck section.	lb
W_{IS}	Weight of "standard" inlet scroll.	lb
$1 W_N$	Weight of one dovetail neck section.	lb
W_{RIM}	Weight of wheel rim.	lb
W_{SHR}	Weight of all rotor bucket shrouds.	lb
$W_{W TOT}$	Total "ideal" wheel weight.	lb
μ	Rotor material Poisson's ratio.	--
ρ	Rotor material density.	lb/in. ³
σ_{ac}	Allowable deformation stress, including over-speed criteria, at stage temperature t .	psi
σ_{ar}	Allowable fracture stress, including overspeed criteria, at stage temperature t .	psi
σ_{CBDT}	Combined stress at blade dovetail top tang.	psi
σ_{CWDB}	Combined stress at wheel dovetail bottom tang.	psi
σ_{DN}	Dovetail neck stress.	psi
σ_{HB}	Average burst stress in "ideal" wheel.	psi
σ_R	Rotor bucket root stress.	psi

<u>Symbol</u>	<u>Definition</u>	<u>Units</u>
σ_{TR}	Radial and tangential stress in an "ideal" wheel.	psi
ω	Rotative speed.	rad/sec

5.4 ROTOR STAGE CONNECTIONS

Four different methods for connecting the wheels were investigated, shown in Figure 5.4-1 - the bolted, the curvic coupling tie bolt, the welded and the integrally machined design. An evaluation of their principal advantages and disadvantages is shown.

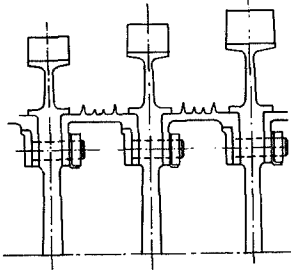
Bolted Rotor

This method is a standard technique in the assembly of jet engine rotors but, since the typical KTA rotor is much smaller, the assembly becomes more difficult. A flat on the bolt head prevents the bolt from rotating while a bent washer between wheel and nut secures the latter. In assembly, the bolt is strained close to the yield point in order to obtain as much preloading as possible.

Calculations have been made to determine the force with which the wheels are held together at start-up and after three years at operating temperature. At start-up, the short time properties of the material are used. Allowance has to be made for a possible temperature lag between bolt and wheel of 100°F, with the bolt being at the lower temperature.

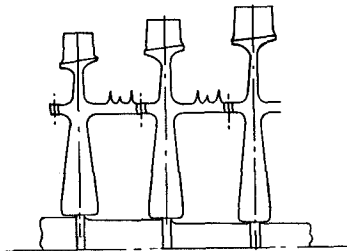
The stage one to stage two wheel connection is taken as an example (see Figure 5.4-2). From the turbine thermal analysis, the steady-state temperature in the bolt area is approximately 1850°F, and TZM bolts are used instead of TZC. This because TZM at that temperature is just as strong as TZC and bolt manufacturing techniques have been established. At 1850°F, 90 percent of the average 0.2 percent yield strength is 64,000 psi and the modulus of elasticity is 31.6×10^6 psi. Under these conditions, the bolt can be strained up to 0.2025 percent

● BOLTED WHEELS (MOLY ALLOYS)



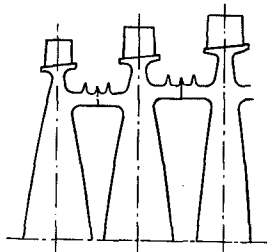
- + STANDARD TECHNIQUE
- CONCENTRATES RELAXATION IN ONE BOLT
- ASSEMBLY COMPLEXITY
- FASTENER INTEGRITY

● CURVIC COUPLING - CENTRAL TIE BOLT (MOLY ALLOYS)



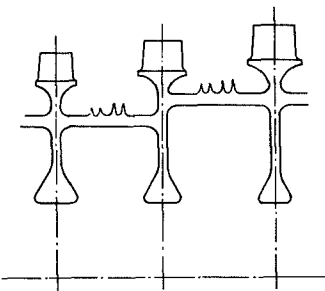
- + PROVEN APPROACH
- + RELATIVE SIMPLICITY
- LEAKAGE INTO INTERSTAGE CAVITY
- NEEDS HOLES IN DISKS

● WELDED ROTOR (Ta AND Cb ALLOYS)



- + HIGH MECHANICAL INTEGRITY
- + NO INTERNAL LEAKAGE
- + GEOMETRIC SIMPLICITY
- MANUFACTURING ACCURACY
- NEEDS DEVELOPMENT
- HEAVY ROTOR (HIGH DENSITY MATERIAL)

● INTEGRALLY MACHINED (MOLY ALLOYS OR Ta ALLOYS)



- + GEOMETRIC SIMPLICITY
- AVAILABILITY OF FORGINGS AND PROPER DISTORTION
- MANUFACTURABILITY
- LARGE HOLES IN DISKS - STRESS PROBLEMS

Figure 5.4-1. Rotor Wheel Attachment Methods.

without yielding. A 100°F temperature difference between the wheel and bolts (with similar expansion coefficients) amounts to a strain of 0.039 percent. Subtracting this from 0.2025 percent leaves an allowable initial pretension of 0.1635 percent. At 70°F, this corresponds to a stress of 76,000 psi. At 1850°F, 0.1635 percent strain corresponds to a bolt stress of 51,000 psi due to a drop in the elastic modulus with temperature.

Due to plastic deformation, this stress will not remain constant with time. Without stress-relaxation test data, it is difficult to determine the residual stresses in the bolt after three years at 1860°F, however, the following hypothesis can be used to obtain a rough estimate of the residual stresses:

If the bolt were under a constant stress of about 14,500 psi, the bolt would creep 0.1635 percent in three years at 1850°F (Figure 5.4-3). If the bolt creeps that much, all preload would disappear and the residual stresses would drop to zero. However, since all of the bolt preload cannot be lost, the residual stress after three years must be at some level below 14,500 psi. This level will be higher, the higher the pretension. The bolts holding the above two wheels together are relaxing under a fixed strain at 0.1635 percent. The stresses drop off from 51,000 psi very rapidly during first stage creep, then very slowly during second stage creep. In order to obtain some idea of the size and number of bolts required, it is estimated from Figure 5.4-4, that a stress of 10,000 psi remains after three years at 1850°F. The residual clamping force of eight 5/16-24 UNF or twelve 1/4-28 UNF bolts amounts to approximately 4000 pounds.

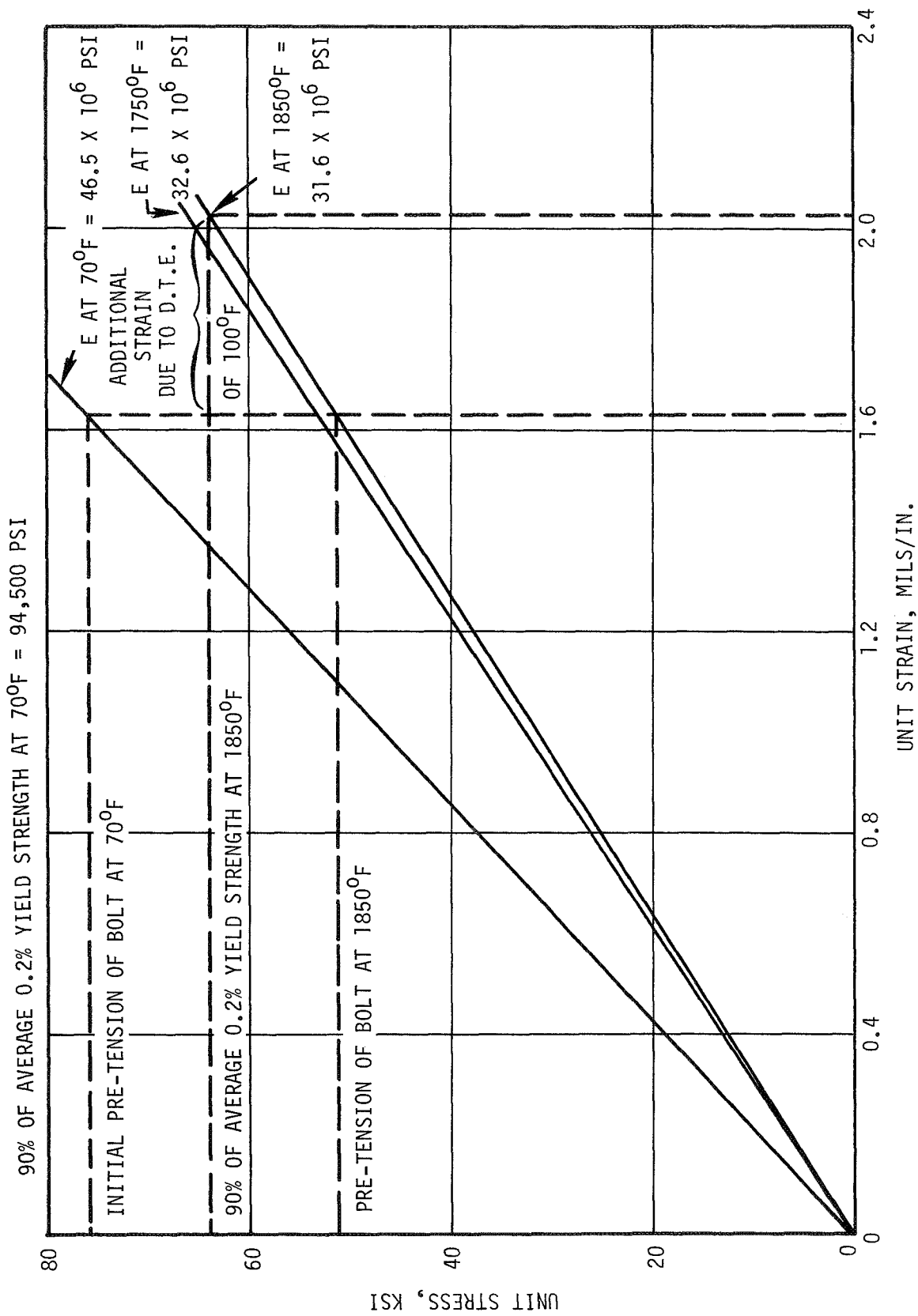


Figure 5.4-3. Analysis of T2M Bolts for Stage 2 Wheel Connection.

- MAJOR PROBLEM: BOLT RELAXATION IN EARLY STAGES ESPECIALLY IF TORQUE TUBES ARE IN TENSION
- USING EIGHT 5/16 DIA BOLTS, RESIDUAL CLAMPING FORCE AFTER 3 YEARS IS 4000 LBS.

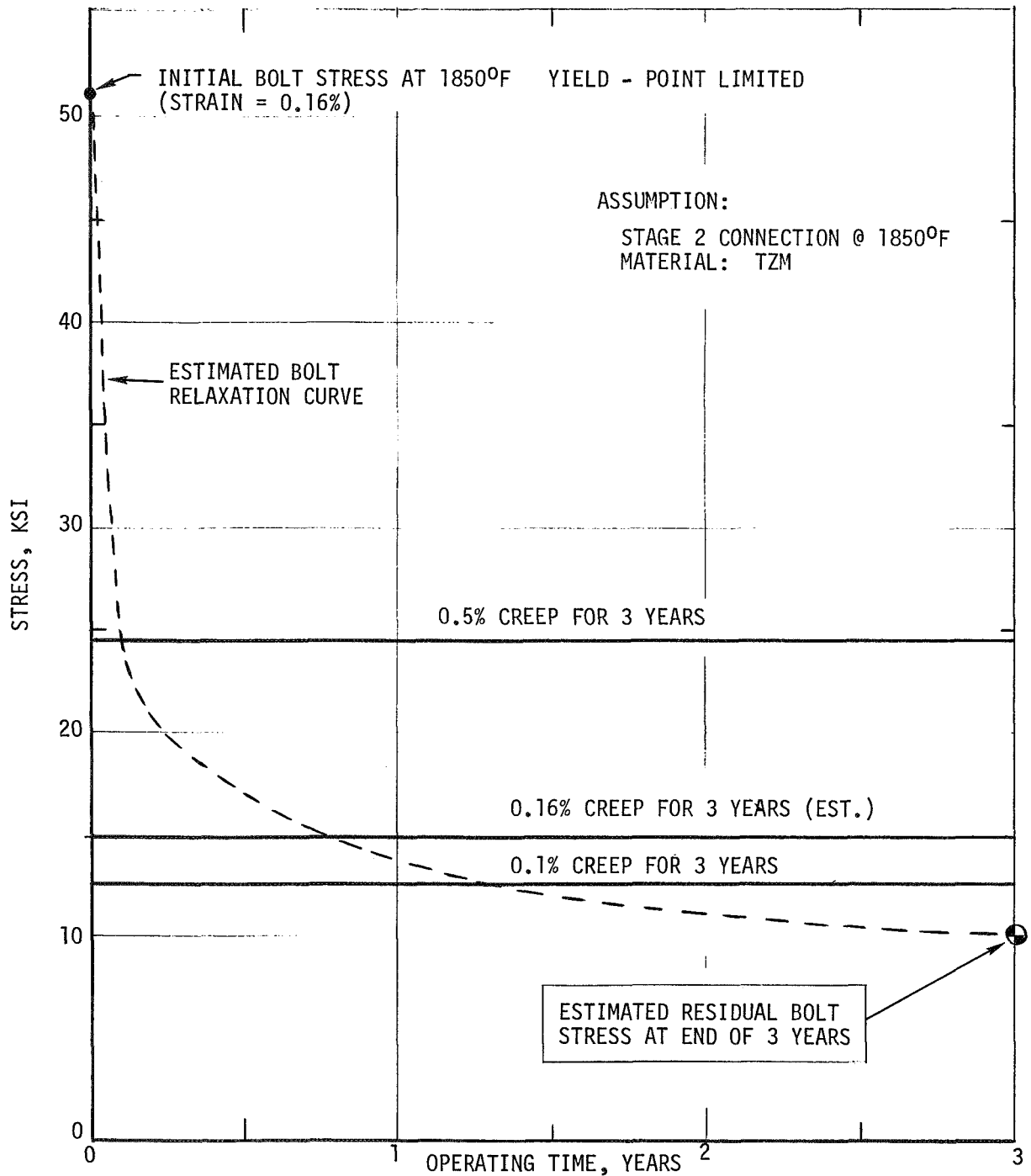


Figure 5.4-4. Rotor Bolting Analysis.

Connecting each stage by bolts appears to be quite feasible, especially after the second stage where lower temperatures also produce less stress relaxation and, therefore, higher residual stresses. Eight bolts, 1/4-28 UNF thread, would provide ample clamping force.

Curvic Coupling/Tie Bolt

The application of the curvic coupling/tie bolt combination as a turbine wheel connecting method has been very successful in the Two- and Three-Stage Potassium Test Turbine Programs (NAS5-1143 and NAS3-8520).

The curvic couplings are face gears which are ground into the torque tubes, and are capable of transmitting many times more torque than is required. They are excellent axial and radial alignment devices, if properly used at large enough diameters.

Figure 5.4-5 shows a typical rotor design using curvic coupling/tie bolt connection. Figure 5.4-6 shows a typical steady-state temperature distribution along the tie bolt of a 10-stage, cradled turbine. The tie bolt is made of TZM material and has varying diameters to provide the necessary strength at the hot section of the turbine, but tapering down towards the cold end to obtain the capacity for initial stretching (pretensioning). The latter is a provision to minimize loss of load during long-time operation. As shown in Figure 5.4-7, the stresses in the hot end of the tie bolt, at the stage one and two wheels, are well within the allowable stress envelope bounded by the 0.5 percent three-year creep strength curves

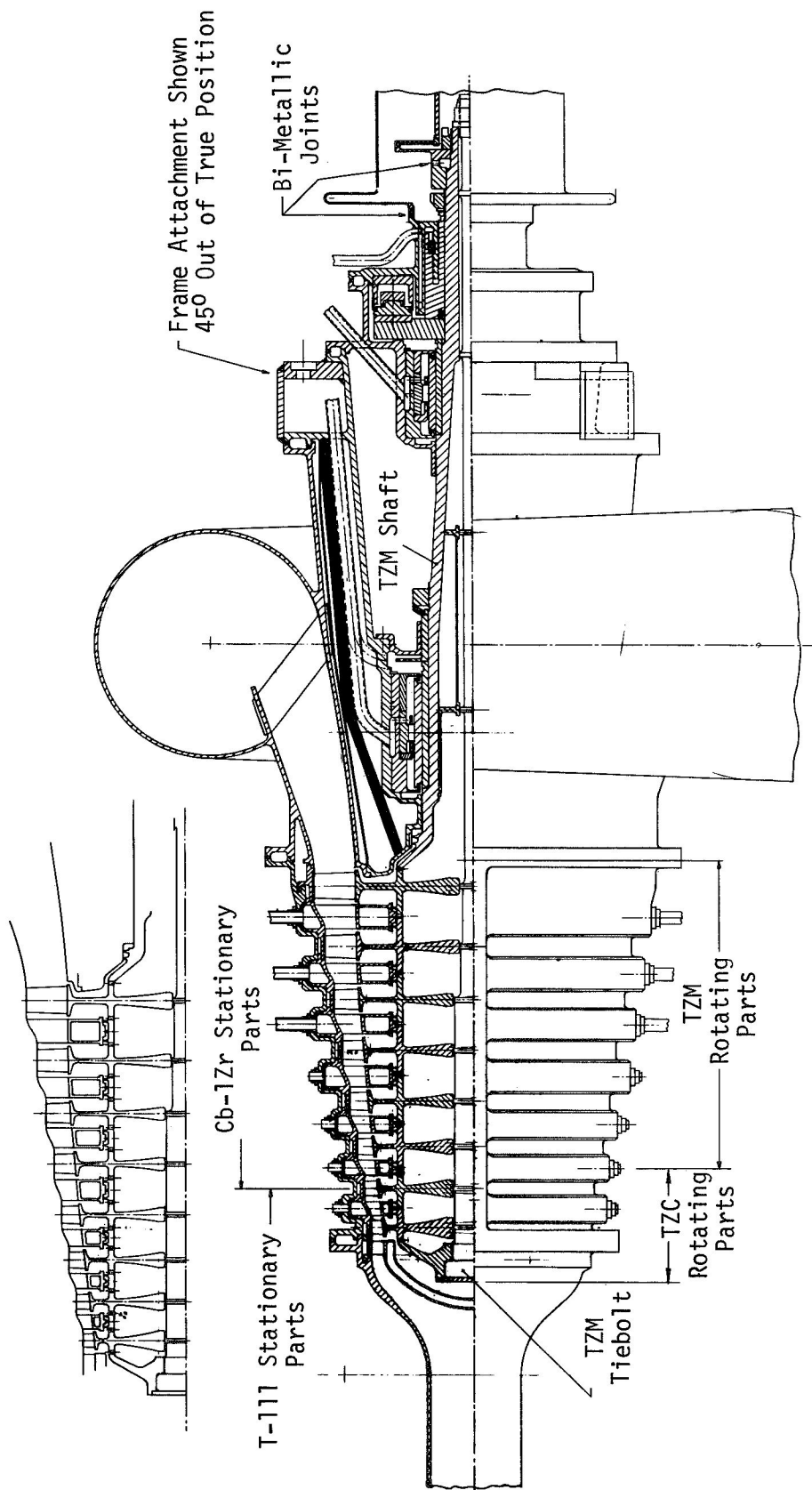


Figure 5.4-5. 8 Stage, 19,200 RPM Overhung, Curvic Coupling-Tie Bolt Rotor.

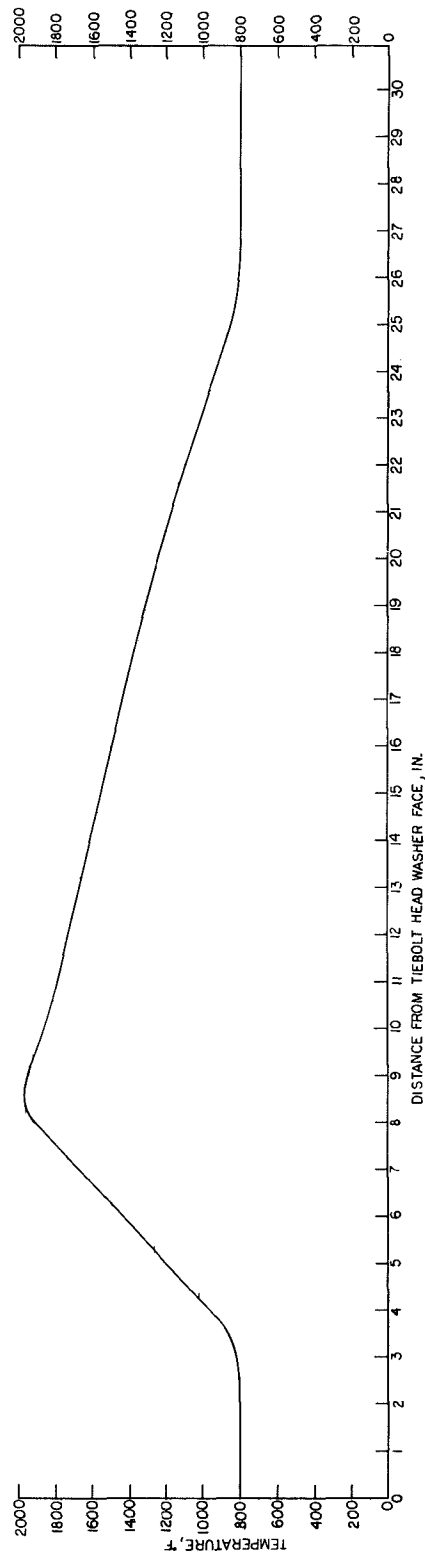
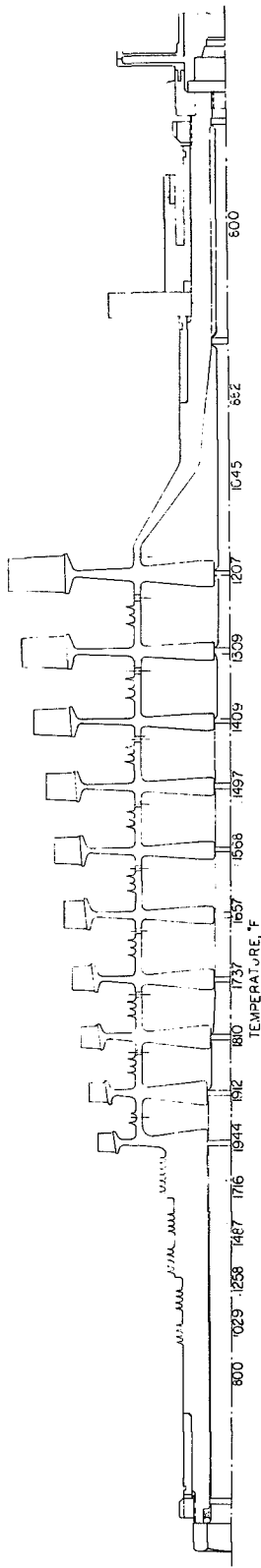


Figure 5.4-6. Tie Bolt Temperature Distribution for 10 Stage, 19,200 RPM Turbine.

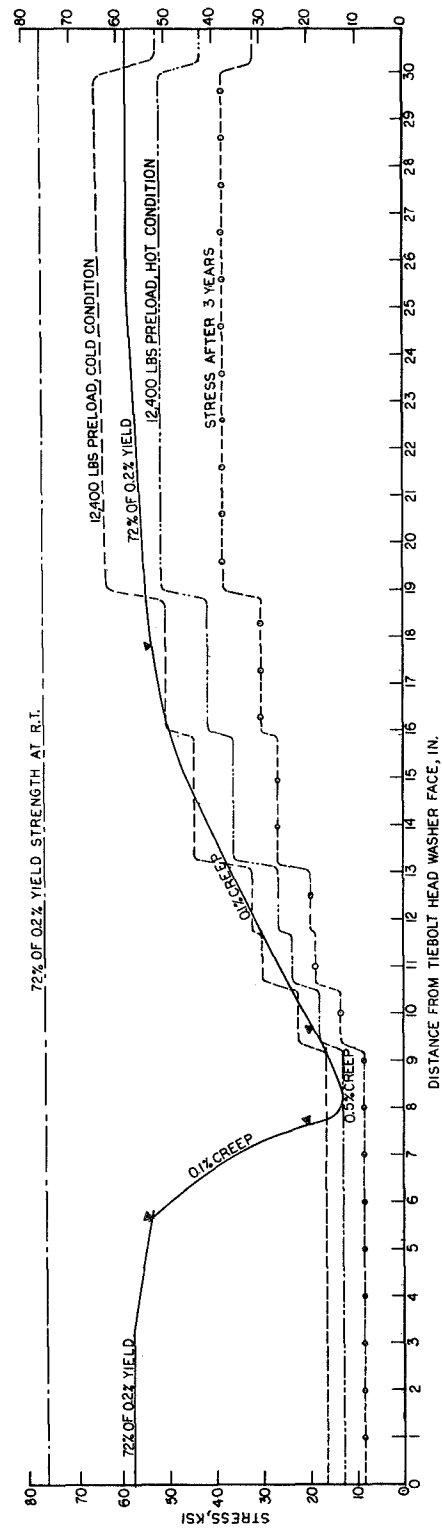
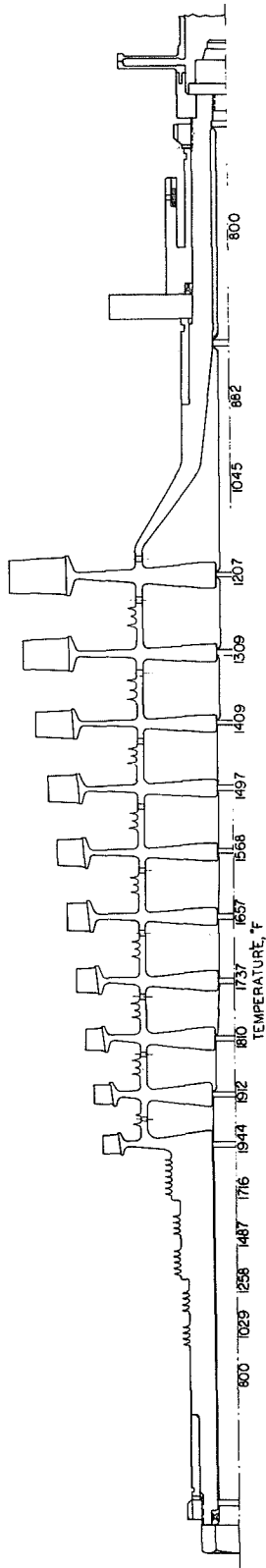


Figure 5.4-7. Tie Bolt Stress Distribution for 10 Stage, 19,200 RPM Turbine.

for TZM. All other stresses are well within the 0.1 percent, three-year creep strength or the 0.2 percent design yield strength curve.

Since tie bolt and wheels are both made of TZM or TZC, the temperature difference between tie bolt and wheels at the curvic coupling is small at steady-state conditions, differential thermal expansion is not a problem. Pretension will nevertheless be lost due to a decrease of the modulus of elasticity with increasing temperature. Calculations show that a pretension of 12,400 pounds on the cold tie bolt will drop to 10,000 pounds upon reaching operating temperature.

According to the computer thermal analysis of the turbine, the worst transient conditions occur immediately after start-up. The temperature distribution at time $t = 2.5$ seconds was chosen for a detailed analysis of wheel and tie bolt differential thermal expansion. The results show a three percent differential thermal growth of the tie bolt and consequently, a temporary 10 percent loss of load.

Assuming that the stresses in the tie bolt were equal to the allowable 0.5 or 0.1 percent three-year creep stress, all pretension would be lost after three years at constant stress. This is not the case with the turbine tie bolt. First of all, the stresses at any section of the bolt are well below the allowable 0.5 or 0.1 percent, three-year creep stress. Secondly, the bolt is under constant strain and not constant stress; this means the stresses drop as the bolt relaxes by plastic deformation. Due to lack of stress, relaxation data for moly-TZM, it is difficult to establish the residual stresses and consequently the residual load in the tie bolt after three years at operating temperature.

However, in order to obtain a rough estimate of the residual load, it is assumed that the tie bolt creeps 0.5 percent in a one-inch hot section and 0.1 percent in four cooler areas of two-inch total length. This would result in a permanent set of 0.007 inch and consequently a load loss of 2560 pounds, leaving a residual load of 7440 pounds.

Sufficient initial stretch in the long tie bolt permits taking up the slack incurred by local creep, thus maintaining a high load. From this standpoint, it is a better design than the wheel connection by individual small bolts.

Welded Rotor

Two weldable alloys were selected for the rotor. The highest strength weldable refractory alloy, the tantalum alloy ASTAR-811C, is employed in the first four stages. For the latter, cooler stages, the Columbium alloy FS-85 can be used. From the material design curve of Figure 5.3-2 (previously introduced) for ASTAR-811C/FS-85 rotors, it can be seen that 1600°F is the transition temperature for switching from ASTAR-811C to FS-85 on the basis of weight efficiency. Joining of the wheels could be accomplished either by electron beam welding or friction welding, the latter of which is a technique whose parameters have not been established for the above alloys. Once established, high mechanical integrity would be the result. Since no leakage path exists within the rotor, the overall efficiency of a turbine with welded rotor would be improved. Figure 5.4-8 shows a typical welded rotor design.

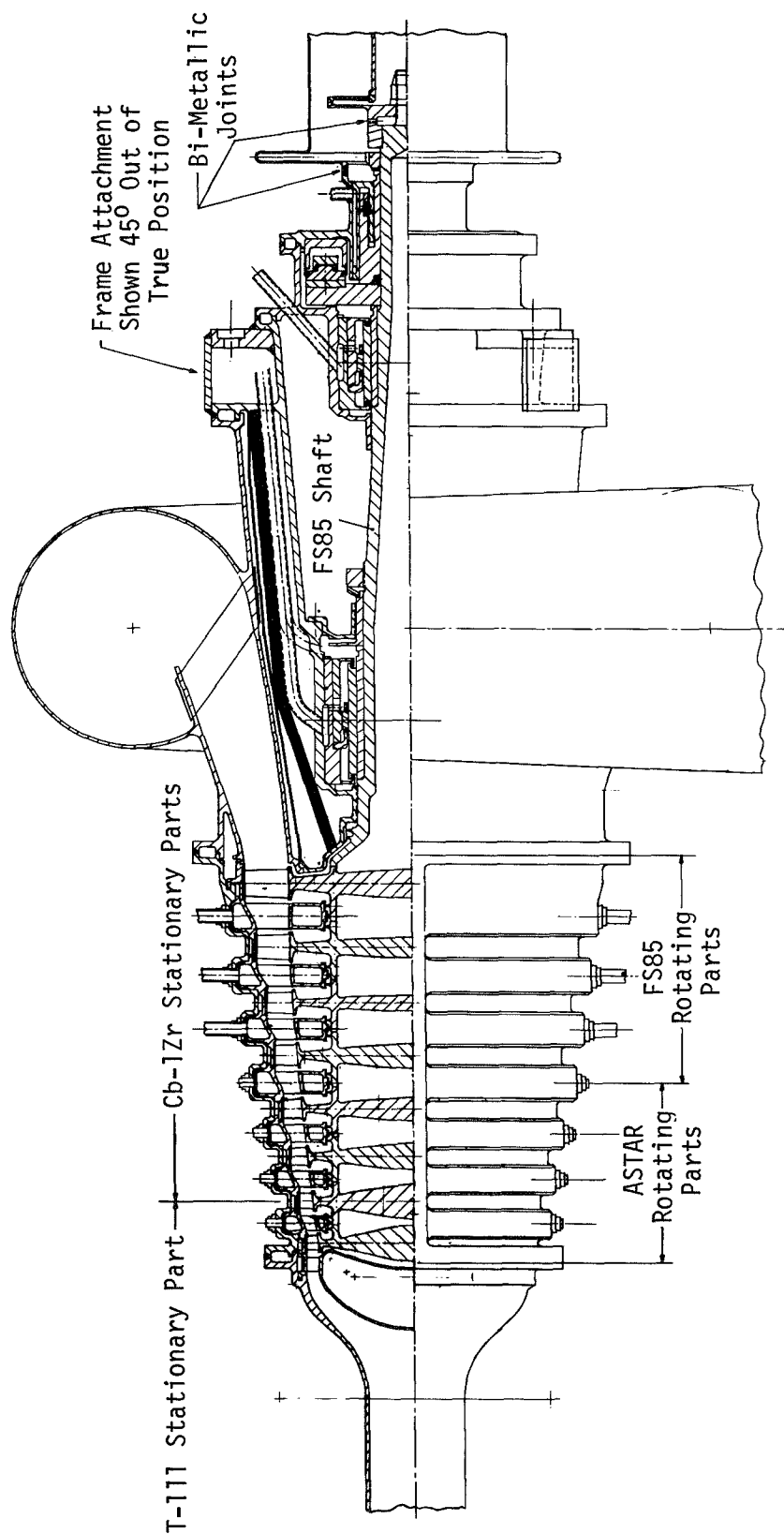


Figure 5.4-8. 8 Stage, 19,200 RPM, Overhung, Welded Rotor.

Integrally Machined Rotor

Since integrally machined rotors are now being successfully employed in the compressor design of General Electric TF-39 and T-58 jet engines, they have also been considered for use in the KTA turbine. It is certainly the most striking concept of rotor construction due to its geometric simplicity, rigidity, and absence of leakage path. Several major obstacles exist, however. As in the three previous stage connection methods, one must start with a molybdenum or tantalum alloy forging. The availability of the large forging which would be required for the integrally machined rotor has not yet been established. For ease of manufacture (boring) fairly large holes have to be cut into the center of the disks. A large hole in the center of a disk rotating at high speed and temperature represents a formidable stress problem. Integrally machined rotors are generally feasible in the relatively cool environment of the compressor portion of a jet engine, but not in a high temperature turbine with extreme creep limitations. As a result, this approach has not been further pursued.

Torque Tubes

Torque tubes are employed as connecting the separation members between the wheels. In the welded and curvic coupling/tie bolt rotor, the torque tubes are an integral part of the wheels. For the bolted rotor, the torque tube becomes a separate part. Since drawings are usually not available at the time the preliminary design analysis is started, it is necessary to estimate the torque tube dimensions if first approximations of geometry, weight, and moment of inertia information is desired. The several forces acting on the torque tube which have to be analyzed to obtain the total effective stress are summarized in Table 5.4-1:

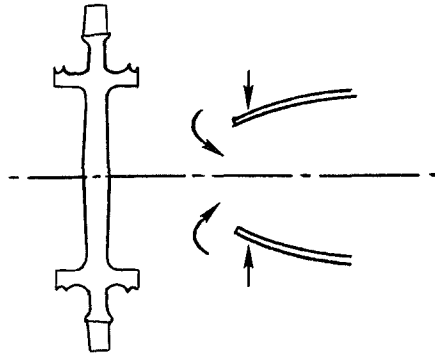
TABLE 5.4-1
ROTOR TORQUE TUBES

FORCES

- TORQUE (SMALL ON FIRST STAGES AND PROGRESSIVELY HIGHER IN SUBSEQUENT STAGES).
- AXIAL COMPRESSION OR TENSION (DUE TO STAGE-TO-STAGE PISTON LOADING). THIS INCREASES FRONT-TO-AFT IN DIRECTION OF FLOW.
- BENDING STRESS WHERE HOTTEST TORQUE TUBES ARE MIDWAY BETWEEN BEARINGS.
- CENTRIFUGAL FORCES DUE TO ROTATION.

ANALYSIS

TREAT TORQUE TUBES AS THIN-WALLED CYLINDERS CONSTRAINED BY ADJACENT DISKS.



MAJOR PROBLEM AREAS

FIRST TWO STAGES

1. Torque - Small on the first few stages but progressively higher in subsequent stages.
2. Axial compression or tension - Due to stage-to-stage piston loading. This force increases from front-to-aft in the direction of the flow. (Figure 5.4-9)
3. Bending Stress - Where hottest torque tubes are midway between bearings.
4. Centrifugal Forces - Due to rotation.

In the stress analysis, the torque tubes are treated as thin-walled cylinders constrained by adjacent disks. The following is the basic input to the computer:

All torque tubes are assumed to be one inch long.

Torque tubes must be capable of transmitting 150 percent of the torque required at 550 kw, with equal torque production by each stage of the turbine.

Torque tube thicknesses are assigned as a function of whether or not dovetails are present.

The computer output is printed in such a way that weight and moment of inertia corrections for actual dimensions are easily made. A factor of safety based upon the allowable deformation stress is defined:

$$\text{Factor of safety} = \frac{\text{Allowable deformation stress}}{\text{Effective stress}}$$

When the resultant factor of safety is greater than unity, the design should be quite satisfactory. If it is much less than unity, a

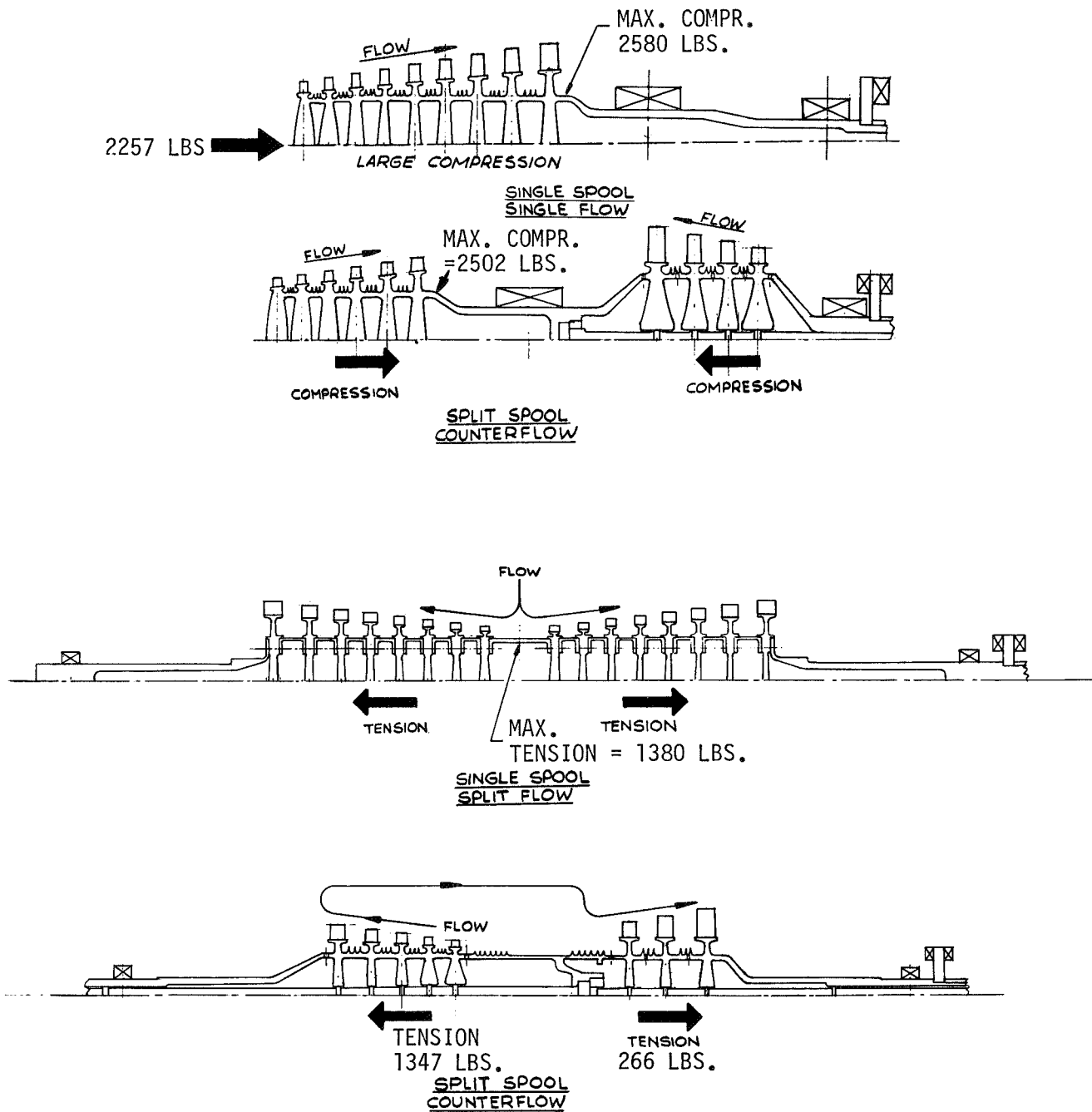


Figure 5.4-9. Torque Tube Axial Loads.

redesign should be made or the midpoint and endpoint effective stresses should be calculated in detail, considering fillet geometry, sense of stresses, etc.

The results showed that no major problems exist except for the first two stages of the turbines. The basic problem is the decreased material deformation strengths because of the elevated temperatures, which are unable to meet the load requirements. The major components of torque tube stresses are the centrifugal stresses and deformations. These deformations must be matched with the deformation of the wheels at the torque tube centerlines, when both the wheel centrifugal stresses and the torque tube loads are considered. Thus, the solution requires numerous iterations based upon an assumption of torque tube dimensions and exact knowledge of wheel stresses and deformations - beyond the scope of the preliminary mechanical analyses. The screening analyses, however, indicated that geometrically tenable solutions for torque tubes could be found.

The conclusions to be drawn from the rotor stage connection studies of the welded rotor, the bolted rotor, and the curvic coupling/tie bolt rotor are that all three designs appear to be feasible at 19,200 rpm, but it becomes difficult to identify the best. The bolted rotors are lighter because they are based on high strength molybdenum alloys. Their inherent problem is bolt relaxation, primarily at stages one and two. Their additional problem is internal leakage. The curvic coupling is wide open for vapor flow. Vapor will leak to lower temperature areas and will condense. This condensation inside the rotor is considered a cause for rotor distortion in steam turbine design and as a rule is not permitted. The two- and three-stage

potassium test turbines showed no ill effects stemming from internal leakage and condensation. For ten- and twelve-stage turbines, the seriousness of the internal leakage problem remains an unknown. The bolted and the curvic coupling/tie bolt designs would be preferable for gas turbine applications. In two-phase flow, however, the welded rotor should be preferred because it is leak tight. The weld materials also offer the advantage of better ductility and accordingly, advantages in forging. Concentric welding of turbine stages without undue distortion is considered feasible by using electron beam welding. It appears, however, that the welded rotors require more developmental effort than the bolted molybdenum designs. It seems premature at this moment to rule out one or the other. It is suggested to continue on the basis of interchangeable rotors and to make a selection only after hardware and test experience have been established.

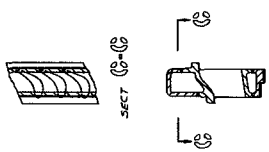
5.5 CASING AND STATOR

A drawing of the turbine casing is shown in Figure 5.5-1. It is horizontally split, and connected to the inlet and exhaust scrolls by welded flanges. Where the vapor temperature exceeds 1800°F, the casing material must be T-111 tantalum alloy, while below 1800°F the casing may be made of Cb-1Zr columbium alloy. The casing has deep annular recesses or convolutions for the nozzle diaphragms. Rigidity is provided by the longitudinal ribs and the horizontal weld flanges. This construction permits a ready changeover to moisture extraction from the nozzle vane trailing edge.

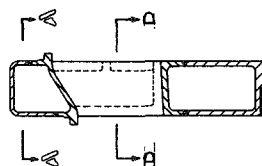
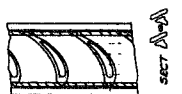
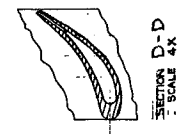
Stresses

Potassium vapor at elevated temperatures generates significant pressures within the turbine casing. Since the casing is made up of two horizontally split halves, the pressure forces are transmitted through the weld into the flange. In order to determine the outer fiber stresses due to bending, the flange is treated as a cantilever beam with an end load. To conserve weight, a beam of constant strength was chosen.

The casing dimensions for the 2 x 8 stage 24,000 rpm split flow turbine (Drawing 47R199284) were chosen to determine the total pressure forces. The effective flange lengths for this turbine amounts to 0.38 inches and the angle required for uniform strength is 30 degrees. The three year, one percent creep stress was chosen as the limiting stress.



NOZZLE DIA. ASSY DETAIL
2ND STAGE - SCALE 2X



NOZZLE DIA. ASSY DETAIL
8TH STAGE - SCALE 2X

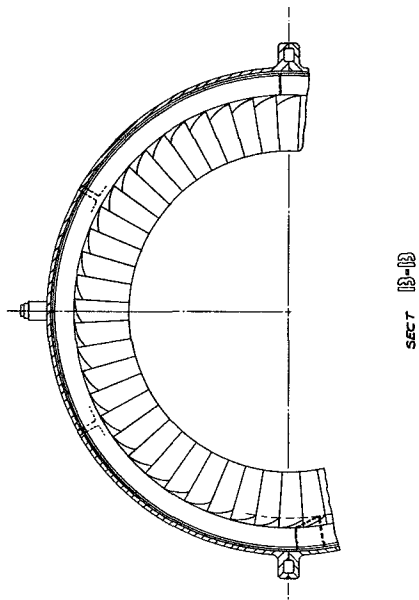
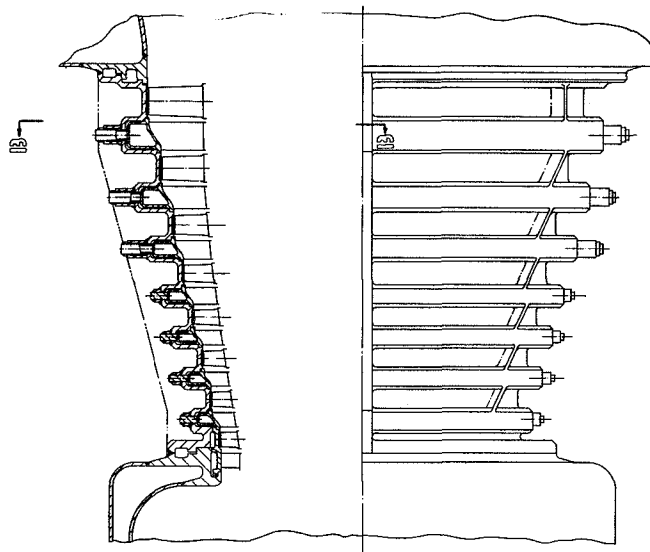


Figure 5.5-1. Turbine Casing Design.

Above the stage one turbine wheel the casing temperature is approximately 1920°F. The allowable three year, one percent creep stress is 15,000 psi and the corresponding flange thickness at the root is 0.312 inch. This is quite thick for flight-weight hardware, but also very conservative because the high stresses are only in the extreme fibers and much strengthening is obtained from the adjacent inlet scroll to which the casing is welded. At the second stage, the casing temperature of 1710°F permits the use of Cb-1Zr. The internal potassium vapor pressure is approximately 100 psi in this area and the allowable three year, one percent creep stress of Cb-1Zr at 1710°F is only 5,000 psi. This makes the flange 0.41-inch thick at the point where it joins to the casing half shell. Again, this is quite heavy but Cb-1Zr is only half as dense as T-111. The required flange thicknesses for all turbine stages are plotted on Figure 5.5-2.

As for the casing, the nozzle diaphragms are made of T-111 in the first two or three stages and Cb-1Zr in the latter stages. Figure 5.5-3 shows the construction details. The vanes are electrical discharge machined integrally with outer and inner bands. Both the outer and inner retaining rings are box-sections for maximum torsional rigidity. The retaining rings are joined to the nozzle diaphragm bands by either electron beam or tungsten inert gas welding. The principal fabrication problem to be resolved is whether the vanes can be successfully fabricated to their finished dimension and surface finish requirements by electrical discharge machining. Nozzle diaphragms of similar configuration have been made, but not with dimensions as small as the KTA will require. The working space between adjacent vanes may well cause a change of manufacturing plans. If

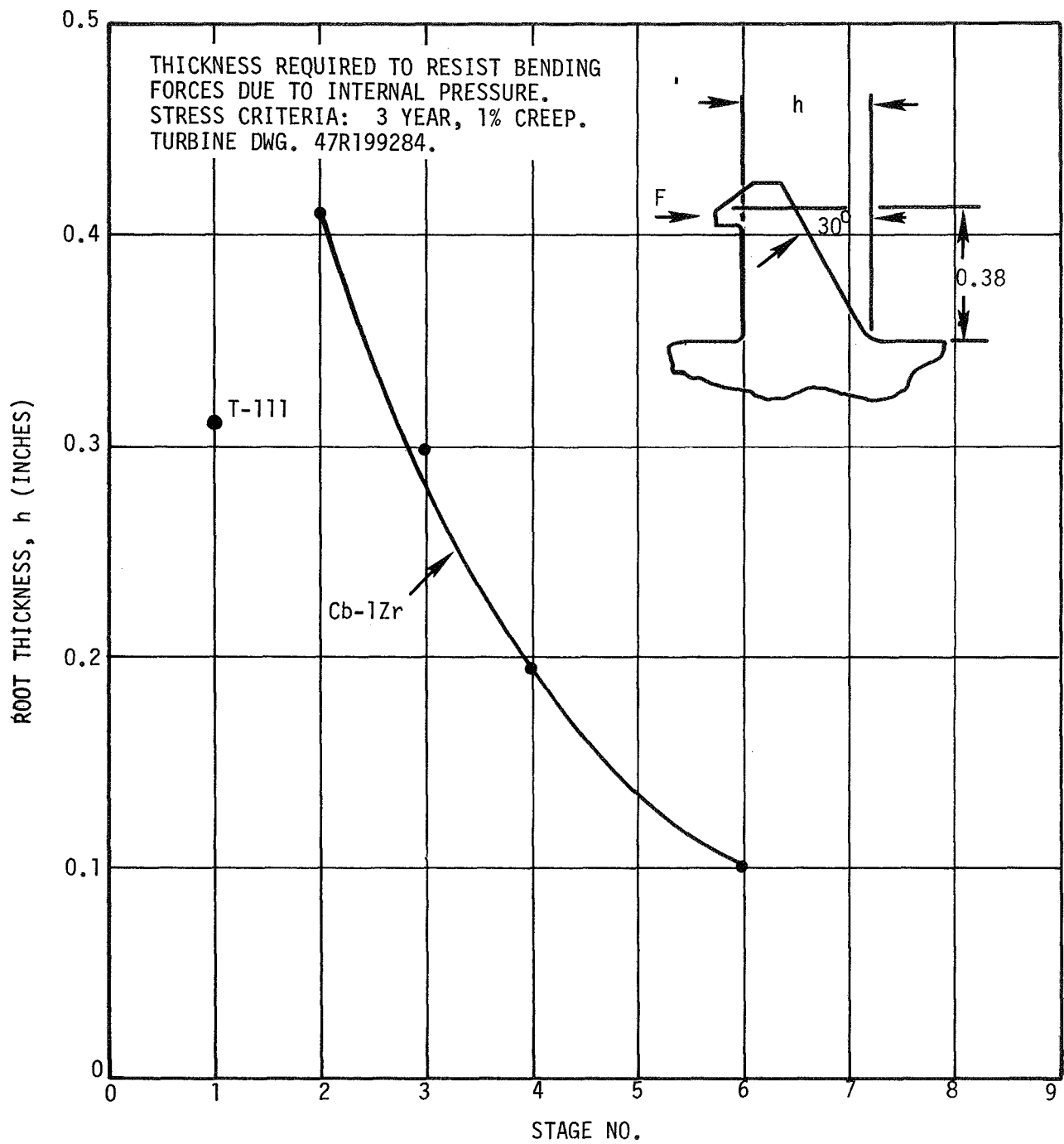
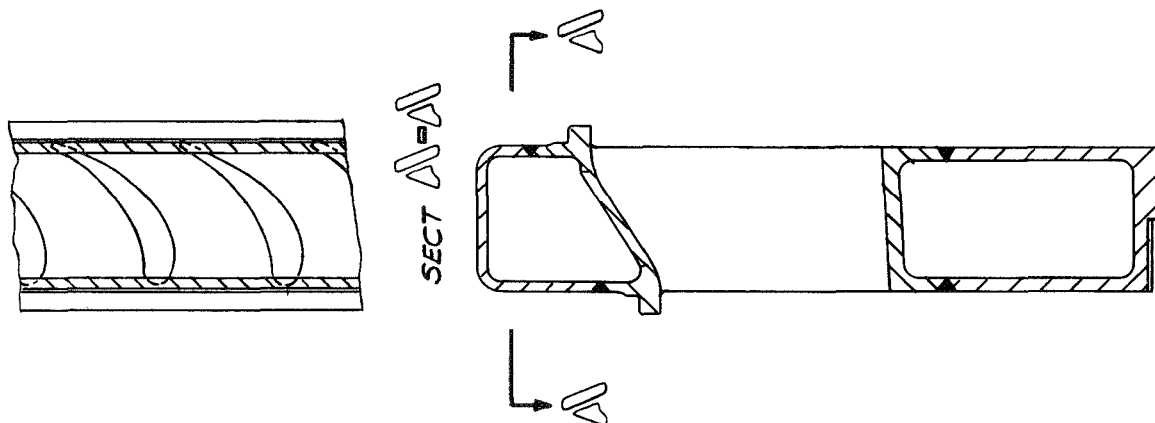


Figure 5.5-2. KTA Casing Horizontal Weld Flange.



- DESIGN CONSIDERATIONS:
- MATERIAL: T-111 & Cb-1Zr (CORRESPONDING TO CASING).
- VANES ARE EDM MACHINED INTEGRALLY WITH OUTER AND INNER BANDS (SMALL VANE SIZE).
- OUTER & INNER RETAINING RINGS ARE BOX-SECTIONS FOR MAXIMUM TORSIONAL RIGIDITY.

Figure 5.5-3. Nozzle Diaphragms.

so, the method presently used on the three-stage potassium test turbine could be utilized - vanes machined as separate pieces, fitted into contoured slots in the inner and outer shroud bands, and brazed into position. For the KTA, electron beam welding could be substituted. The nozzle diaphragms are positioned in the casing by electron beam welding a cylindrical plug which protrudes through the casing wall. The honeycomb rub strips of the interstage labyrinth seals are attached to the inner diameter of the diaphragm inner ring by electron beam welding.

The stresses in the diaphragms arise from two principal sources - temperature differentials and pressure differentials. For the present, it has been assumed that temperature differences within the nozzle diaphragms are minimal, and that the stresses which they would generate are negligible. The worst thermal stresses will be generated by casing restraint of the diaphragms if the thermal deformations of the casing are very different from those of the diaphragms. Close similarity is expected, however, because of the assumption of excellent insulation to control thermal losses and because of equal coefficients of thermal expansion for the casing and nozzle diaphragms (same material).

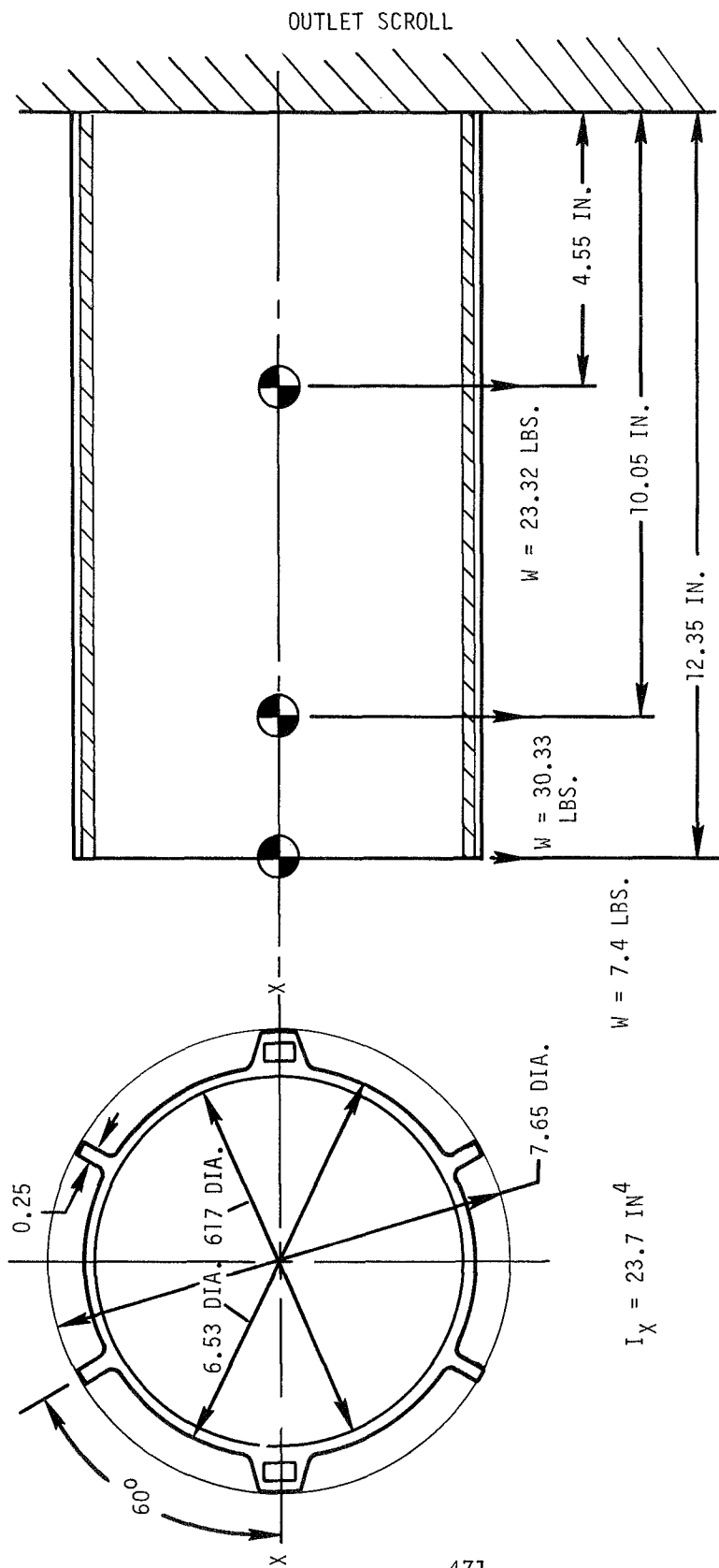
The pressure stresses originate with the difference between nozzle inlet and outlet static pressures, acting on the areas between interstage labyrinth seal teeth tips and flow path outer diameters, on both inlet and exhaust sides of the nozzles. The axial forces generated can be several hundred pounds, but are resisted by an appreciable total nozzle vane area. The exact analysis of the resulting stresses in the nozzle vanes requires a computerized approach, which in turn

requires detailed knowledge of vane geometry not available during preliminary studies. Experience with similar nozzle vanes of stainless steel which operate at the temperatures present in the latter stages of KTA turbines give confidence that no major stress difficulties exist which cannot be successfully resolved.

Deflections

Calculations were made to determine the approximate magnitude of the casing deflection when subjected to a lateral acceleration of 20 G's (644 ft/sec^2). It was assumed that the casing is cantilevered from the outlet scroll. Three lumped masses represent the weight of the inlet scroll, the T-111 casing portion at stage one and two, and the remaining Cb-1Zr casing portion to the exit scroll. In addition to the two horizontal weld flanges, four longitudinal stiffener ribs are assumed. These ribs increase the bending stiffness and prevent excessive axial creep of the convoluted casing. While the casing cross-section and wall thickness vary between inlet and outlet end, a constant cross-section similar to that of the inlet end was assumed throughout for simplification. All dimensions and lumped masses are shown on Figure 5.5-4.

The total deflection of the free end of the casing amounted to only 0.0011 inch. The actual deflection of the casing will be less, since the supported end has a larger cross-section, and therefore, a greater resistance to bending. A modulus of elasticity of 13.95×10^6 psi, corresponding to Cb-1Zr at 1350°F was used throughout.



CASING IS TREATED AS A CANTILEVER BEAM WITH END AND INTERMEDIATE LOADS

Figure 5.5-4. Analytical Model for Casing Deflection Study.

It is recognized that vibrations can become a troublesome problem in any part of the casing which has thin walls and large unsupported areas, including scrolls. It is believed that acceptable design solutions can be found at the expense of requiring increased weight. Therefore, vibration elimination studies must be made simultaneously with casing stress, deflection, and weight analyses. The primary goal for the structure must be minimum weight with acceptable margins of safety. Computerized methods of analysis exist within General Electric to attack these problems, such as the MASS, MULTI-SHELL, and VAST computer programs. However, within the time available for the preliminary studies it was necessary to concentrate most heavily upon the problems of rotating parts and delay the complete solution of casing problems until later. The casing wall thicknesses shown on drawings are estimates, and have not been optimized with respect to stiffness and weight.

5.6 SCROLLS

Inlet Scrolls

Formulas have been derived for making calculations of typical KTA inlet involute scroll stresses and wall thicknesses. The geometrical relationships shown on Figure 5.6-1 give reasonable geometry and use a minimum of parameters. These parameters are easily derived from the known first-stage parameters of bucket tip radius R_T and root radius R_R . The temperatures used are those calculated in the thermal analysis of a typical eight-stage turbine. The analysis methods developed are applicable to the shaded geometry, with additional

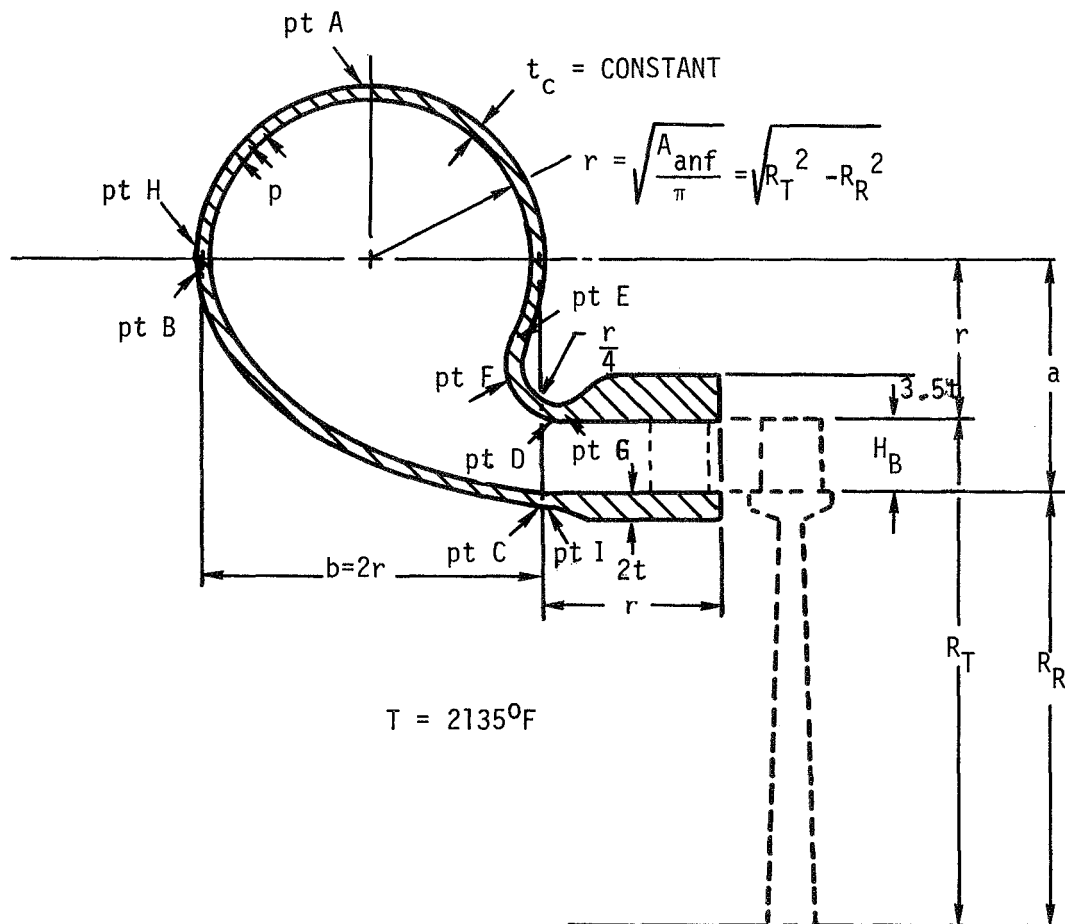


Figure 5.6-1. Analysis Model of "Standard" Involute Inlet Scroll.
(Maximum Area Section Shown).

computations required for the involute entry flanges. Only pressure stresses are calculated. A minimum factor of safety of $FS = 1.5$ is used, based upon the allowable three-year rupture design stress of T-111 at the temperatures shown.

The thicknesses required for each geometrical element of the structure are calculated, based upon achieving a 1.5 factor of safety and using the inlet vapor pressure. Then factors of safety are calculated for all points, assuming that the largest thickness $t_c = 0.182$ inch is used all around: $FS = \frac{0.182 (1.5)}{t}$ for 2.0632 turbine (six stages, 32,000 rpm) and $\sigma_e = \frac{N}{0.182}$

<u>Point</u>	<u>FS (-)</u>	<u>σ_e (psi)</u>
A	4.4	1140
B	4.7	1050
C	1.5	3340
D	16.1	350
E	3.4	1470
F	16.1	310
G	2.2	2550
H	3.8	1300
I	2.3	2190

where t is the required thickness at a point and N is the unique combination of geometrical parameters which describe the effective stress at a point. In the above case, the critical point is C, where the maximum effective stress and minimum factor of safety exists.

To these pressure stresses must be added thermal stresses (essentially non-existent because of low thermal gradients), stresses from structural loads (which do not exist until the inlet scroll is used as a structural member), and bending stresses due to differential strains. The latter can exist at points C (= I) and D (= G) because of the rapid changes in cross-sectional areas at those points. The large factor of safety of about nine which will exist at point D after stress-averaging occurs will minimize this problem at that location. The judicious use of proper contouring at point D will minimize the problem there. Also, the stresses in the involute decrease rapidly around the involute as smaller cross-sections occur. The model chosen is, therefore, considered to give a satisfactory approximation.

The factors of safety and stress levels which have been calculated for turbine 2.0632 are very similar to those which will occur in any other turbine where the assumed geometrical ratios are preserved.

There is no practical way to make all principal stresses be tensile (or compressive) because the inlet pressure must act on the "outer" surface of certain geometrical figures and the "inner" surfaces of others. It is also practically impossible to decrease the maximum wall thickness t_o required, because this thickness can only decrease to the thickness t_i of the inner cylinder, no matter how greatly the ellipse ratio of the elliptical torus section is changed. A severe weight and volume penalty would result if this is attempted.

Because of manufacturing complexity, it is considered impractical to have an inlet scroll with variable wall thicknesses to reduce weight.

An inlet scroll weight analysis is made for every stage of every turbine analyzed by the basic mechanical design analysis program. These figures are used for making weight estimates and for checking the more detailed weight analyses made of scroll designs which appear on KTA drawings. The detailed weight analyses are included in KTA weight figures to be reported later.

Exhaust Scroll

The geometrical relationships of Figure 5.6-2 are selected as those which give reasonable exhaust scroll geometry and use a minimum of parameters, which may be easily derived from the known last-stage parameters of blade tip radius R_T and root radius R_R . The temperatures used are those calculated in the thermal analysis of a typical eight-stage turbine. The analysis methods developed are applicable to the shaded geometry. Only pressure stresses are calculated. A minimum factor of safety of $FS = 1.5$ is used, based upon the allowable three-year rupture design stress of Cb-1Zr at the temperature shown.

The thicknesses required for each geometrical element of the structure are calculated, based upon achieving a 1.5 factor of safety and using the exhaust vapor pressure. Then factors of safety are calculated for all points, assuming that the largest thickness calculated is impractically small, and therefore, 0.080 inch wall thickness will be used to allow manufacture: $FS = \frac{0.080 (1.5)}{t}$ for 2.0632 turbine and $\sigma_e = \frac{N}{0.080}$

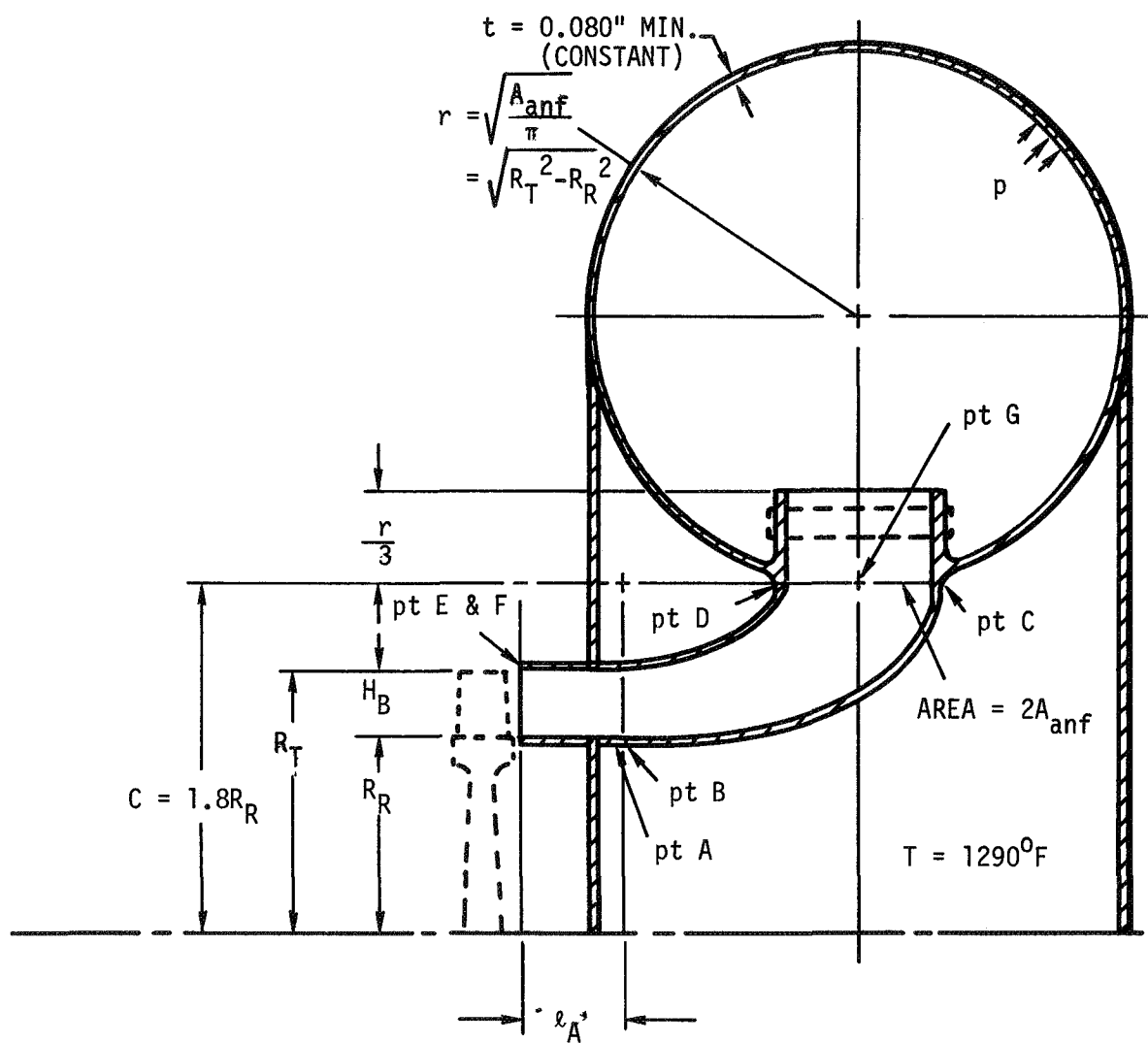


Figure 5.6-2. Analysis Model of "Standard" Constant - Area Exhaust Scroll.

<u>Point</u>	<u>FS (-)</u>	<u>σ_e (psi)</u>
A	73.6	204
B	41.0	370
C	184.6	80
D	550.5	30
E	47.1	350
F	51.5	320
G	41.2	360

where t and N are again the required thickness and the combination of geometrical parameters which uniquely describe the effective stress.

In the above case, the limitation of 0.080 inch minimum wall thickness produces extremely low effective stresses and extremely high factors of safety. This indicates that exhaust scrolls will generally not have great stress problems, even when needed at intermediate turbine stages.

To these pressure stresses must be added thermal stresses (essentially non-existent because of low thermal gradients), stresses from structural loads, and bending stresses due to differential strains. The latter two will be present at each end of the elliptical tori, but are assumed to be absorbed by judicious contouring of sections in these areas. The unbalanced force of exit pressure times exit flange area might be considered to be absorbed by other parts of the structure. However, if an expansion bellows is present this load is approximately 400 pounds (for 2.0632) will react on the circular torus. It will be compressive, and therefore subtractive, from the tensile loads

in the circular torus. It is assumed that this load is well-distributed over the large line of intersection between the circular torus and the outlet flange, which is required for acceptable fluid flow.

The factors of safety and stress levels which have been calculated for turbine 2.0632 are very similar to those which would occur in any other turbine where the assumed geometrical ratios are preserved.

An outlet scroll weight analysis is made for every stage of every turbine analyzed by the basic mechanical design analysis program, but detailed weight analyses of the scrolls which appear on the KTA drawings are the ones which are included in the system weight figures reported later.

5.7 THERMAL CONSIDERATIONS

The calculation of an accurate overall turbine temperature distribution is a necessary prerequisite for the analysis of thermal stresses, and for differential thermal expansion calculations to establish proper clearances. Also, allowable stresses and other material properties are based upon the calculated temperatures.

The governing considerations in the thermal analysis are that the stator and rotor thermal expansion coefficients are similar and low, since all turbine construction materials are refractory alloys (Figure 5.7-1). Therefore, the primary source of geometrical interferences and thermal stresses will be temperature differences. These will be worst during the transient start-up conditions.

The method of attack is to use an established computer program for analysis, assuming that all parts are preheated to 600°F and instantaneously subjected to full vapor temperature.

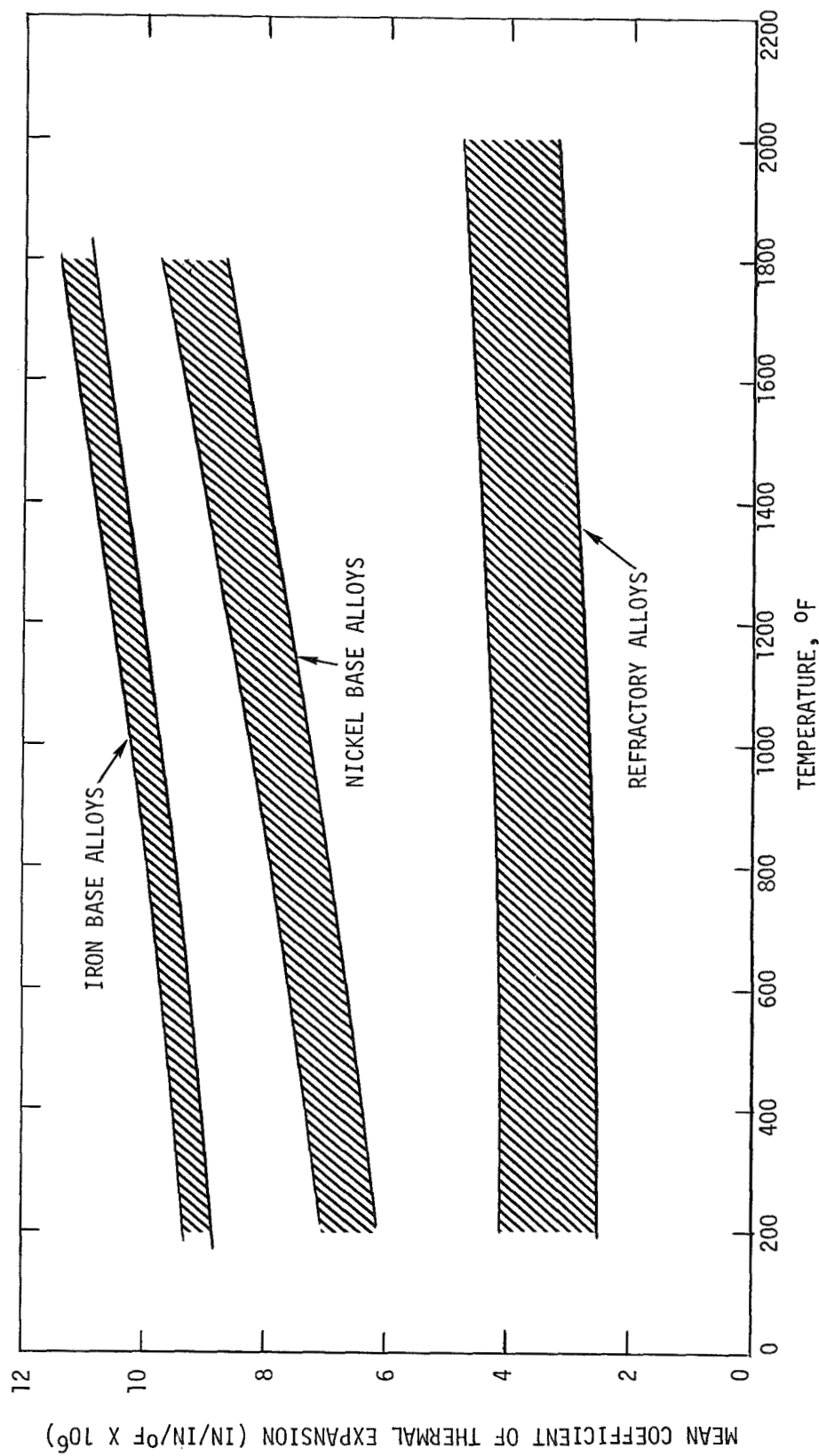


Figure 5.7-1. Mean Coefficient of Thermal Expansion as a Function of Temperature for Various Material Types.

Transient Response

Calculations of temperatures are performed using the Transient Heat Transfer computer program, which is based on a thermal-electrical analogy wherein the turbine is subdivided into a series of material blocks or "nodes". Kirchhoff's laws are written about each node with heat flow instead of current as parameter. Figure 5.7-2 shows the nodes pattern plots for both the rotor and the stator. Because of the symmetry of the turbine parts about the centerline of rotation, the nodal plot can be shown in a single plane. Each node is wedge-shaped, having minimum width at its inner radius and maximum width at its outer radius.

Drawing 263E156 of turbine 1.0820 (eight-stage, 20,000 rpm, single flow) was assumed to be a satisfactory and typical geometrical model. The vapor inlet temperature was assumed to be 2150°F, with 1300°F outlet temperature. Rotor wheels, the torque tube following each wheel and the casing adjacent to rotor blade tips were assumed to be exposed to potassium vapor at stage temperature. The casing adjacent to the tips of the nozzle blades were assumed to be exposed to potassium vapor at stage temperature. The casing adjacent to the tips of the nozzle blades were assumed to be exposed to potassium vapor at temperatures intermediate between the preceding and following stage temperatures. Trapped volumes in the casing and between turbine wheels, below torque tubes, were assigned potassium vapor temperatures below the temperature of the closest stage. Bearing areas were assumed to contain liquid potassium at 800°F. Insulation was assumed to surround the casing, with its sink temperature being 1200°F. Realistic thermal contact coefficients were assumed where

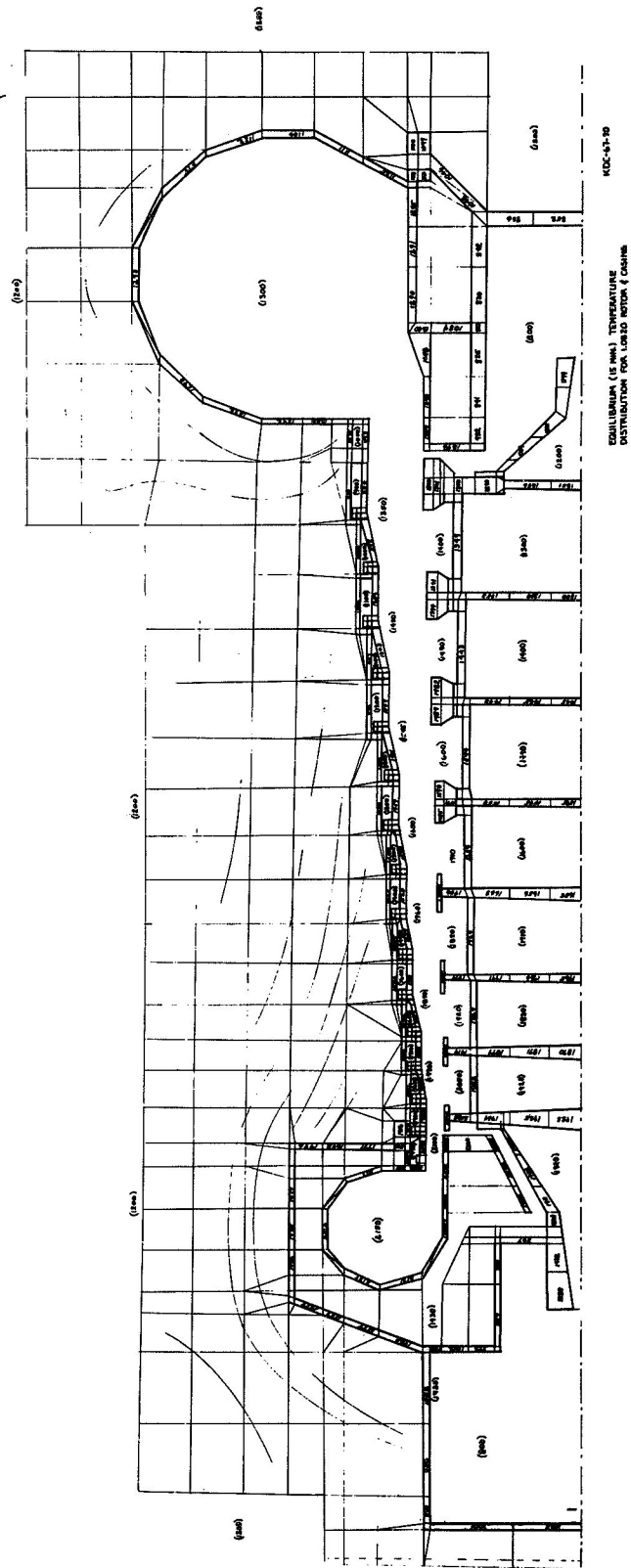


Figure 5.7-2. General Turbine Temperature Distribution.

potassium/metal, insulation/metal or metal/metal interfaces occurred. Preheating of all parts to 600°F was assumed. These thermal analysis input conditions and the material properties used are listed in Tables 5.7-1 and 5.7-2. Computed temperatures were printed at 2.5, 5, 7.5, 10, 20 and 30 seconds; and at 1, 2, 5, 10 and 15 minutes after application of the full values of the constant temperatures. Thermal equilibrium was sufficiently well established after 15 minutes to allow the assumption that the results at that point were the same as for infinite time. This is evident from a typical temperature vs. time plot shown in Figure 5.7-3.

Axial thermal expansion calculations based upon the equilibrium temperature distribution have been made. For every node along the paths from thrust bearing to first stage through the stator and rotor, the temperature calculated for the selected time, the coefficient of thermal expansion for that temperature, and the axial distance between node centerlines were used in a thermal expansion calculation. The result indicates that axial thermal expansion is not of major concern for the T-111 stator - TZM rotor combination. At zero seconds, when all parts have been preheated to 600°F, the worst differential is 0.003 inch at stage one. The worst transient differential at 2.5 seconds, is 0.010 inch at stage one. The worst equilibrium differential, at 900 seconds, is 0.009 inch at stage eight. In all cases, the casing growth exceeds the rotor growth.

When Cb-1Zr is substituted for T-111 for all stages aft of stage two, the coefficient of thermal expansion rises 16 percent and the thermal conductivity rises 18 percent (averages over the temperature range 1300° - 1800°F), while the density drops 49 percent. The net

TABLE 5.7-1

INPUT CONDITIONS FOR 1.0820 ROTOR/CASING
THERMAL ANALYSIS (BASIC CASE)

Materials - All static parts T-111, all rotating parts TZC or TZM.

All parts at 600°F at time = 0.

714 nodes used.

Contact coefficient = 100 for metallic interfaces, = 1000 for potassium contact,
 = 10 for insulation contact.

Temperatures constant from time = 0+ to 900 sec:

Lub K temperature	800°F
Inlet scroll temp.	2150
Outlet scroll temp.	1300
Fwd. labyrinth leak temp.	1930
Insulation OD temp.	1200

		Stage No.							
		1	2	3	4	5	6	7	8
Casing K chamber	a	1990°F	1820	1650	1490	1350	1250	1150	1000
Casing K chamber	b	1920	1710	1600	1400	1300	1200	1100	900
Stage temperature	c	2050	1920	1820	1710	1600	1490	1400	1300
Interstage temp.	d	1920	1820	1710	1600	1490	1400	1300	1200
Stator temp.	e	2100	1990	1870	1760	1650	1545	1450	1350

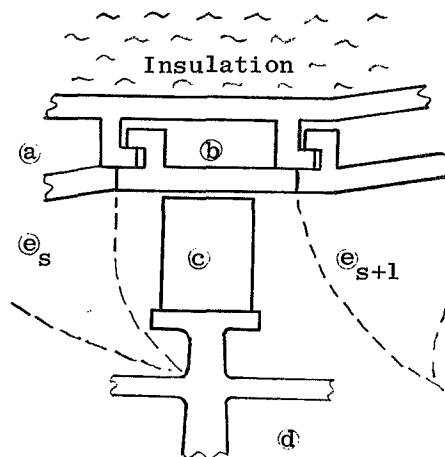


TABLE 5.7-2
MATERIAL PROPERTIES INPUT TABLE FOR THERMAL ANALYSIS

Code Name	Property	Units	Temperature, °F						
			70	400	800	1200	1600	2000	2400
TZC/TZM	Density	lb/ft ³	637.63						
	Thermal Conductivity	Btu/hr ft°F	79.0	76.0	71.7	67.1	62.0	57.0	51.3
	Specific Heat	Btu/lb°F	0.064	0.064	0.064	0.066	0.072	0.077	0.083
T111	Density	lb/ft ³	1043.7						
	Thermal Conductivity	Btu/hr ft°F		25.8	28.2	29.9	31.2	32.3	33.2
	Specific Heat	Btu/lb°F		0.034	0.035	0.036	0.038	0.038	0.039
Insulation	Density	lb/ft ³	0.332						
	Thermal Conductivity	Btu/hr ft°F		2.58	2.82	2.99	3.12	3.23	3.32
	Specific Heat	Btu/lb°F		0.034	0.035	0.036	0.038	0.038	0.039

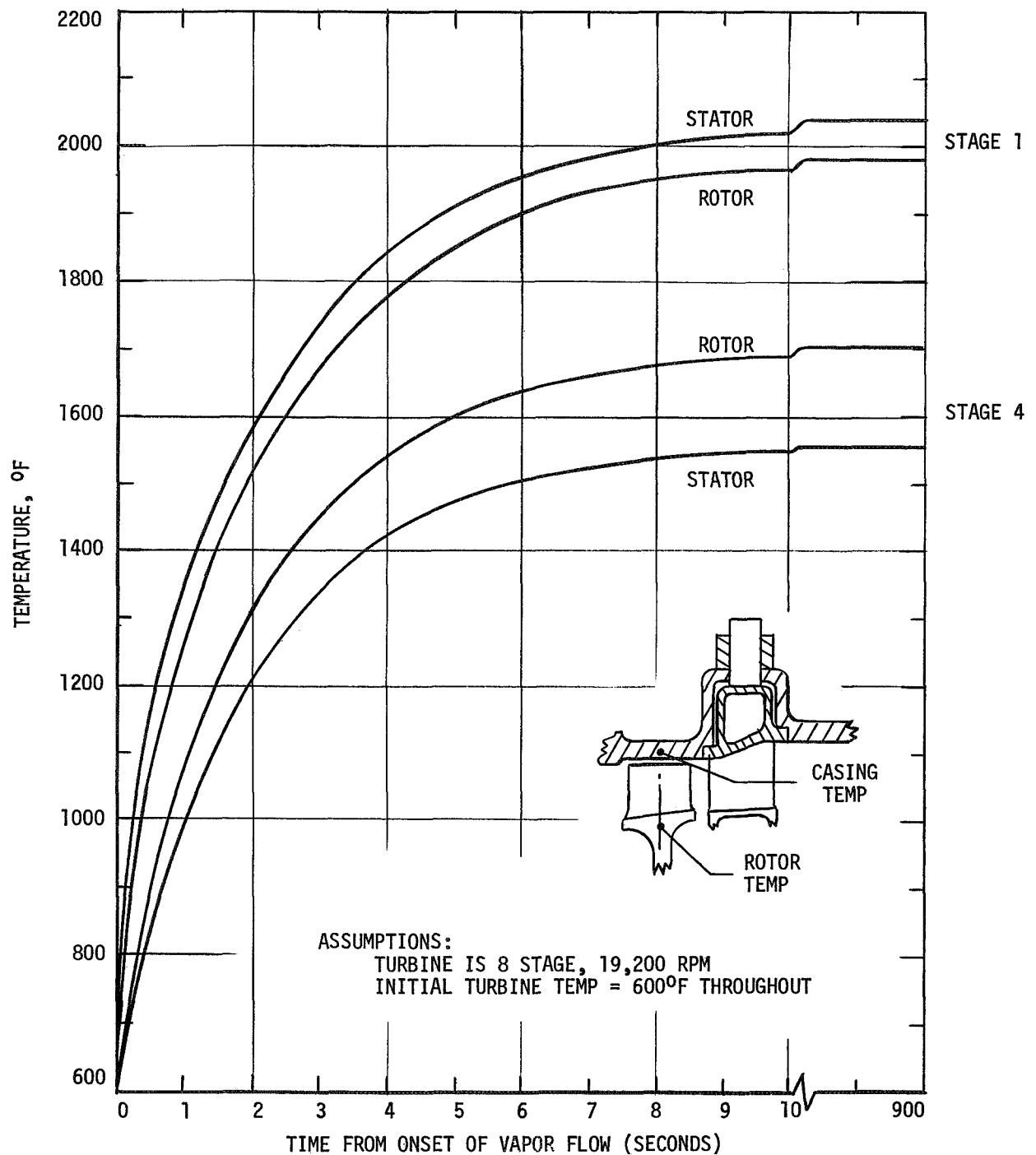


Figure 5.7-3. Transient Temperature Response of Typical Rotor and Stator Following Onset of Vapor Flow.

effect is estimated to produce worst differential thermal expansions of 0.003 inch at stage one at zero time (600°F), 0.012 inch at stage one at 2.5 seconds, and 0.011 inch at stage eight at equilibrium. The casing growth still exceeds the rotor growth.

When ASTAR-811C and FS-85 materials are substituted for TZC and TZM in the rotor, it is estimated that the change of material properties produce worst differential thermal expansions of 0.014 inch at stage one at 2.5 seconds and 0.007 inch at stage eight at equilibrium.

When the bearing lubricant temperatures are changed from 800°F to 1100°F, it is estimated that the worst differential thermal expansions will be 0.007 inch at stage one at 2.5 seconds and 0.006 inches at stage eight at equilibrium.

For an overhung rotor design such as drawing 263E182 (eight stages at 19,200 rpm), it may be assumed that the wheel temperatures remain unchanged, but the thrust bearing is moved 11 inches aft of its basic case position. The net expansion effect on the casing will be approximately the same as the basic case. The worst differential thermal expansions will be -0.014 inch at stage one at 2.5 seconds (rotor growth exceeds casing growth) and -0.015 inch at stage eight at equilibrium.

With the same overhung geometry and 1100°F bearing temperature, the worst differential thermal expansions are -0.029 inch at stage one at 2.5 seconds and -0.030 inch at stage eight at equilibrium.

For a turbine with oil-lubricated bearings, such as shown by drawing 47R199274, the thrust bearing moves 6 inches aft of its basic

case location and the lubricant temperature drops. The worst differential thermal expansions are estimated to be 0.008 inch at stage one at 2.5 seconds and 0.007 inch at stage eight at equilibrium.

The estimates for all the changes from the basic case analysis are based upon logical changes in material properties and upon valid laws of heat transfer, but cannot be considered as accurate as the basic case. It appears safe, however, to estimate that no more than 0.035 inch of axial mismatch between casing and rotor will occur under the transient (start-up) conditions. If this figure be approximately doubled to allow for manufacturing tolerance stackups and some margin of safety, 0.060 inch axial rotor-stator clearance will be satisfactory to prevent axial rubbing problems.

Figure 5.7-4 shows the zero time (600°F warmup), transient, and equilibrium temperature profiles for the casing rabbet nodes of stage three of the 1.0820 turbine. The added length dimensions at selected time increments show the order of magnitude of the differential radial growths were less than 0.004 inch and the direction of the differential causes the rabbets to separate instead of inducing prohibitively high compressive stresses. The magnitude of separation will not cause cocking or other geometrical difficulties. The slopes of the curves as they approach the 900-second point allows the conclusion that it represents the equilibrium temperature distribution. The differential thermal expansions of the rabbets reverse themselves from the transient situation, and less than 0.001 inch of interference occurs in most locations to cause compressive stresses. Actually, such differential growth would simply reduce the allowable manufacturing and assembly tolerances.

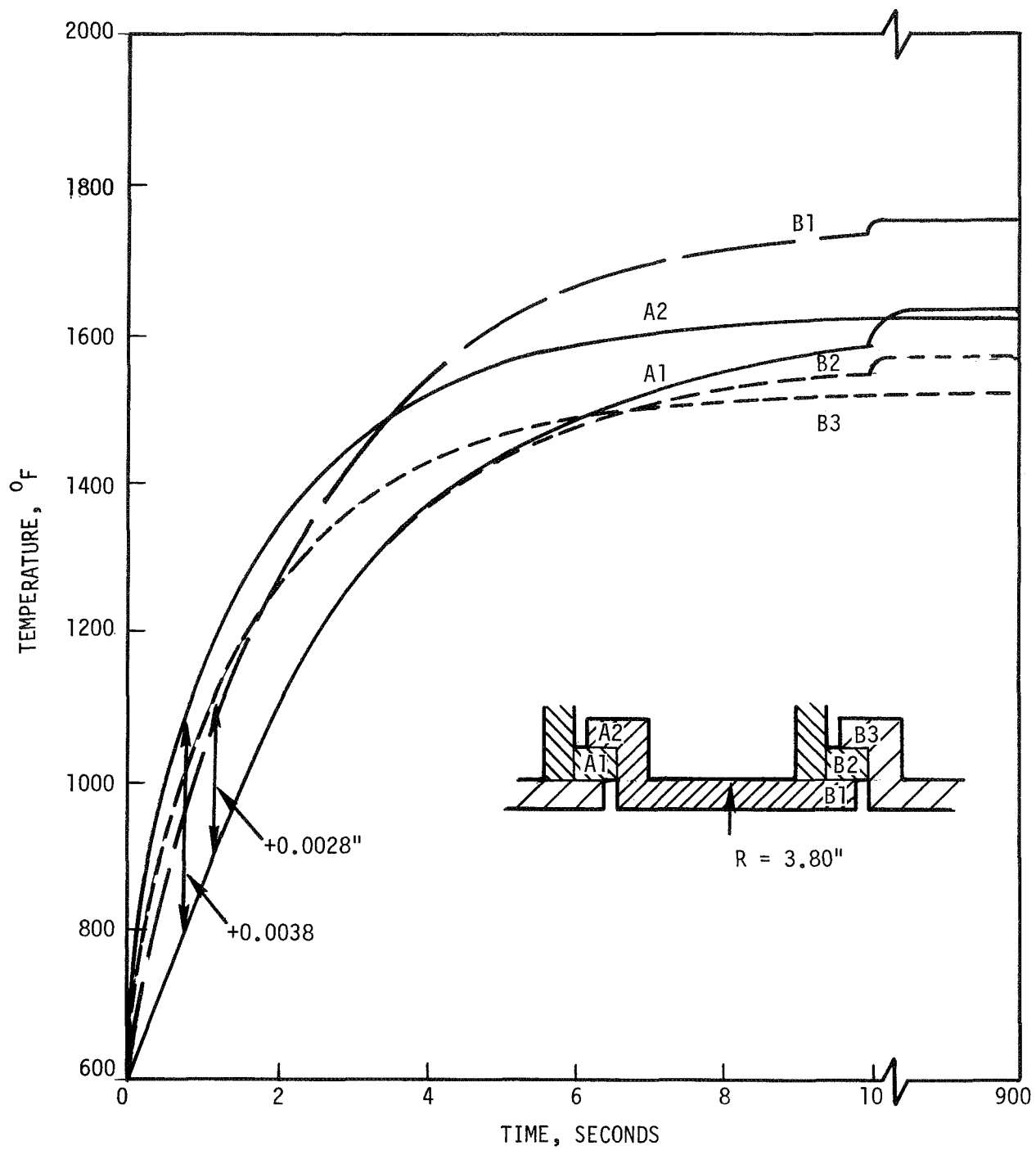


Figure 5.7-4. Casing Rabbet Temperatures for Stage Three of 1.0820 Turbine.

Tip Clearance

The temperature histories of the wheel nodes at the root of the blades and the casing nodes at the tips of the blades were studied for stages one, four, and eight of the eight-stage turbine. By assuming that no temperature differences occur when different rotor materials are substituted, the results of Table 5.7-3 are obtained. It is assumed that the casing does not creep, but the wheels and blades creep 0.1 percent in three years (except that the ASTAR-811C stage one wheel creeps 0.7 percent in three years).

Table 5.7-3 shows that the turbines may safely be designed to have a minimum rotor blade tip clearance of 0.010 inch after three years of operation. With a TZC/TZM rotor and T-111/Cb-1Zr casing, the clearances are 0.013 - 0.015 inch as soon as equilibrium temperatures are reached, and are 0.012 - 0.016 inch under the worst transient conditions, including 600°F warmup. Cold assembly tip clearances of 0.012 - 0.015 inch are required for the stages investigated.

With an ASTAR-811C/FS-85 rotor and T111/Cb-1Zr casing, the clearances are 0.014 - 0.030 as soon as equilibrium temperatures are reached, which imposes a leakage penalty at stage one because of the 0.7 percent creep required to hold the stage one hub thickness to about 1 - 1.5 inches. Under the worst transient conditions, including 600°F warmup, the tip clearances are 0.017 - 0.028 inch. Cold assembly tip clearances of 0.018 - 0.031 inch are required for the stages investigated.

TABLE 5.7-3

ROTOR BLADE TIP RADIAL THERMAL EXPANSION
FOR EIGHT STAGE, 20,000 RPM TURBINE

Parameter	Units	Stage		
		1	4	8
3 years (0.1% creep):				
Hot casing tip radius	in.	3.380	3.820	4.660
Hot tip clearance	in.	0.010	0.010	0.010
Hot blade tip radius	in.	3.370	3.810	4.650
(ASTAR-811C @ 0.7% creep)	in.	3.370	--	--
Hot blade height	in.	0.230	0.830	1.210
Hot wheel radius	in.	3.140	2.980	3.440
0 years (0% creep):				
Hot casing tip radius	in.	3.380	3.820	4.660
Hot tip clearance	in.	0.013	0.014	0.015
(ASTAR-811C)		(0.030)	--	--
Hot wheel tip radius	in.	3.367	3.806	4.645
(ASTAR-811C)	in.	(3.350)	--	--
Equilibrium:				
Casing tip temperature	°F	1988	1560	1104
Rotor blade temperature	°F	2051	1707	1301
Rotor wheel temperature	°F	2046	1708	1300
TZC Assembly:				
Casing material	--	T-111	Cb-1Zr	Cb-1Zr
Rotor material	--	TZC	TZM	TZM
Cold casing tip radius	in.	3.355	3.795	4.639
Cold tip clearance	in.	0.015	0.012	0.013
Cold rotor tip radius	in.	3.340	3.783	4.626
Cold blade height	in.	0.228	0.824	1.204
Cold wheel radius	in.	3.112	2.959	3.422
ASTAR Assembly:				
Casing material	--	T-111	Cb-1Zr	Cb-1Zr
Rotor material	--	ASTAR-811C	FS-85	FS-85
Cold casing tip radius	in.	3.355	3.795	4.639
Cold tip clearance	in.	0.031	0.018	0.020
Cold rotor tip clearance	in.	3.324	3.777	4.619
Cold blade height	in.	0.227	0.823	1.202
Cold wheel radius	in.	3.097	2.954	3.417
Worst transient ΔT :				
Casing tip temperature	°F	1605	1550	1105
Rotor blade temperature	°F	2051	1707	1301
Rotor wheel temperature	°F	1675	1700	1215
TZC Assembly worst transients:				
Hot tip radius	in.	3.375	3.820	4.660
Hot tip clearance	in.	0.012	0.014	0.016
Hot wheel tip radius	in.	3.363	3.806	4.644
ASTAR Assembly worst transients:				
Hot casing tip radius	in.	3.374	3.820	4.660
Hot tip clearance	in.	0.028	0.014	0.017
Hot wheel tip radius	in.	3.346	3.806	4.643

Interstage Seals

The temperature histories of the torque tube nodes on which the labyrinth seal teeth are located, and the casing nodes at the outer radius of the nozzle blades, were also studied. By assuming that no temperature differences occur when different rotor materials are substituted, the results of Table 5.7-4 are obtained. It is assumed that the casing does not creep, but all torque tubes creep 0.1 percent in three years. The computations are based upon the seal teeth being included as part of the nozzle height, because they would be expected to reach the same temperature as the nozzle blades when leakage flow occurs. It is also assumed that the nozzle blades are made from the same material as the casing.

Table 5.7-4 shows that the turbines may safely be designed to have a minimum torque tube labyrinth seal tooth tip clearance of 0.010 inch after three years of operation. Tip clearances of 0.012 and 0.013 will exist as soon as equilibrium temperatures exist. The assembly of TZC/TZM rotor and T-111/Cb-1Zr casing requires cold assembly teeth clearances of 0.013 inch, which remain the same under worst transient conditions, including 600°F warmup. The assembly of ASTAR-811C/FS-85 rotor and T-111/Cb-1Zr casing requires cold assembly teeth clearances of 0.013 and 0.017 inch, which become 0.012 and 0.013 inch under the worst transient conditions, including 600°F warmup. These conclusions are strictly valid only for the geometries and temperatures used, but are expected to be generally true for the range of geometries and temperatures under investigation.

It may be concluded from the thermal studies that, since all of the materials used in the turbine have coefficients of thermal

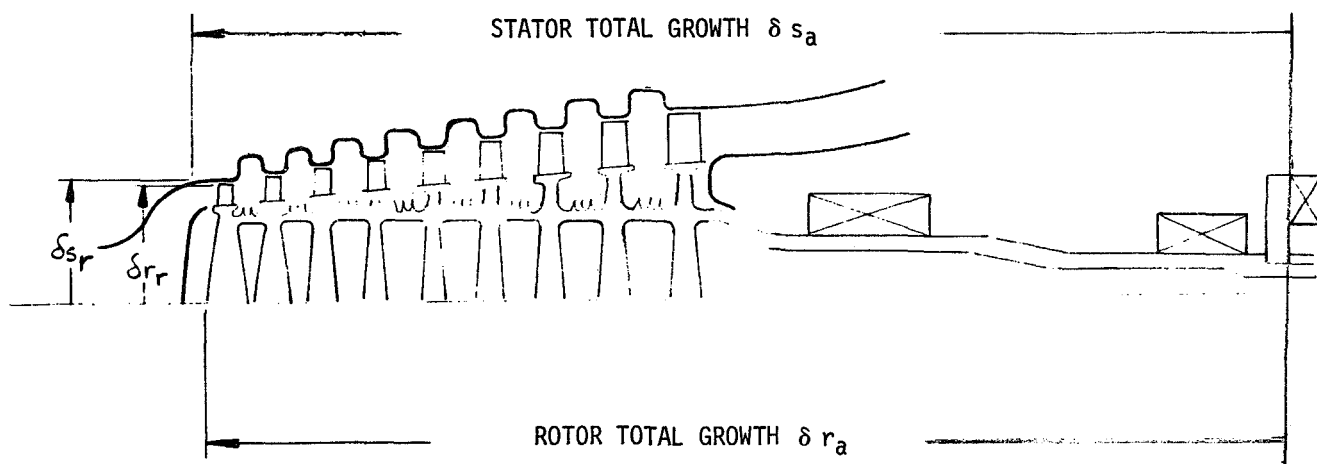
TABLE 5.7-4

TORQUE TUBE LABYRINTH SEAL TIP RADIAL THERMAL EXPANSION
FOR EIGHT STAGE, 20,000 RPM TURBINE

Parameter	Units	Stage	
		1	4
3 year (0.1% creep):			
Hot casing nozzle radius	in.	3.430	3.870
Hot nozzle height	in.	1.000	1.260
Hot tooth tip clearance	in.	0.010	0.010
Hot torque tube radius	in.	2.420	2.600
0 years (0% creep):			
Hot casing nozzle radius	in.	3.430	3.870
Hot nozzle height	in.	1.000	1.260
Hot tooth tip clearance	in.	0.012	0.013
Hot torque tube radius	in.	2.418	2.597
Equilibrium			
Cassing nozzle temperature	°F	1910	1500
Nozzle blade temperature	°F	1990	1650
Torque tube temperature	°F	1980	1650
TZC Assembly:			
Casing material	--	T-111	Cb-1Zr
Rotor material	--	TZC	TZM
Cold casing nozzle radius	in.	3.405	3.846
Cold nozzle height	in.	0.992	1.251
Cold tooth tip clearance	in.	0.013	0.013
Cold torque tube radius	in.	2.400	2.582
ASTAR Assembly:			
Casing material	--	T-111	Cb-1Zr
Rotor material	--	ASTAR-811C	FS-85
Cold casing nozzle radius	in.	3.405	3.846
Cold nozzle height	in.	0.992	1.251
Cold tooth tip clearance	in.	0.013	0.017
Cold torque tube radius	in.	2.400	2.578
Worst transient ΔT:			
Casing nozzle temperature	°F	1890	1490
Nozzle blade temperature	°F	1990	1650
Torque tube temperature	°F	1970	1640
TZC Assembly worst transients:			
Hot casing nozzle radius	in.	3.431	3.870
Hot nozzle height	in.	1.000	1.260
Hot teeth tip clearance	in.	0.013	0.013
Hot torque tube radius	in.	2.418	2.597
ASTAR Assembly worst transients:			
Hot casing nozzle radius	in.	3.431	3.870
Hot nozzle height	in.	1.000	1.260
Hot tooth tip clearance	in.	0.012	0.013
Hot torque tube radius	in.	2.419	2.597

expansion which are similar and low, both radial and axial rotor-stator differential thermal expansion is not a problem at steady state conditions.

Transient conditions, such as exist at start-up, are the greatest problem even though turbine preheating to 600⁰F has been assumed. In the calculations, it was also assumed that the turbine is instantaneously subjected to full vapor temperature. This is a very severe condition from a standpoint of thermal shock to the turbine wheels and blades. But even with these extreme assumptions it has been established that ten mil radial tip clearance can be safely employed. Axial tolerances between rotor and stator should be approximately 0.060 inch. These conclusions are illustrated by Figure 5.7-5.



RESULTS:

MAX $\Delta \delta$ AXIAL = 0.035 IN. (START-UP)

MAX $\Delta \delta$ RADIAL = 0.004 IN. (START-UP)

CONCLUSIONS:

0.010 IN. RADIAL CLEARANCE O.K.

USE 0.060 IN. AXIAL ROTOR-STATOR CLEARANCE

Figure 5.7-5. Differential Thermal Expansion Between Rotor and Stator.

5.8 BEARINGS

The successful development of high speed, high-temperature rotating machinery frequently depends as much on bearing design and on proper integration of the system from a rotor-bearing dynamics standpoint, as it does on the optimum design of its aerodynamic and electrical components. Frequently also, the mechanical design of the machinery can be simplified and its reliability improved by using the process fluid as the lubricant. This permits such advantages as reduction in thermal gradients along the rotor by operating the bearings closer to system temperature, elimination of complex oil seals, prevention of oil contamination of critical system components, and elimination of cumbersome and sometimes unreliable auxiliary lube oil system.

It must be recognized, however, when using process fluids such as liquid metals as lubricants, that they have low viscosity and lack the boundary lubrication property of conventional hydrocarbon lube oils. These factors lead to reduced load capacity and greater potential for surface damage at start-up and under conditions of inadvertent high speed runs. For these reasons, both potassium and organically lubricated bearings were considered in Phase I of the KTA preliminary design study.

The primary journal bearing selected for the KTA application was the tilting pad type because of its excellent stability characteristics, high stiffness and damping, inherent self alignment, and successful experience in turbulent lubricant film applications. The thrust bearing selected also was the tilting pad which gives high load capacity and, when used in conjunction with leveling links, is self-aligning. The back-up journal and thrust bearings selected were both of the shrouded

step configuration which have been successfully applied in other high speed turbomachine applications. (5.8-1, 5.8-2, 5.8-3, and 5.8-4).

Detailed performance calculations for these bearing types were made and are presented in the following sections of this report. These performance data are shown for both potassium and organic lubrication.

Organic Lubricant

Consideration has been given to the possibility that the KTA could be lubricated with a high temperature, radiation-resistant organic fluid. This fluid would be required to function as the bearing lubricant in the temperature range from radiator temperature to 600⁰F. The fluid would be subjected to radiation levels of about 10^7 rads of gamma energy. Neutron levels of about 10^{11} nvt of fast neutrons would also be involved, but it is reasonable to assume that the gamma dosage would cause the major damage (5.8-5). The desired life is on the order of 25,000 to 45,000 hours.

In order to satisfy these conditions, it is considered to be essential that the fluid be protected with an inert gas blanket and operated in a closed system. A large number of investigators have shown that the effective fluid life at high temperatures can be increased manyfold when oxidation effects are minimized. In addition, actual test data from (5.8-6) have shown that the resistance of oils to radiation damage can be increased by a factor of at least two when air is excluded. When all of these gains are totalled, it becomes apparent that the use of an inert gas environment could increase the effective life of the fluid by at least one order of magnitude.

The combined effects of temperature, radiation dosage and time pose a severe problem as far as fluid selection is concerned. In general, fluid degradation will be manifested by three types of physical change:

- . The production of volatile decomposition products
- . Changes in the bulk viscosity of the lubricant
- . The formation and precipitation of oil-insoluble sludges

Of these, changes in the bulk viscosity is of major immediate concern since this will have a direct effect on calculated bearing performance. This is also the intermediate step in the process of sludge formation.

Viscosity will almost always increase, thus causing increased power losses in the bearings.

At various times a number of candidate fluids, both synthetics and mineral oil derivatives, have been proposed as high-temperature lubricants. These include:

- . High phenyl content silicones
- . Esters
- . Fluorinated hydrocarbon derivatives
- . Polyphenyl ethers
- . Highly aromatic petroleum base stocks
- . Super-refined mineral oils
- . Synthetic hydrocarbons
- . Alkyl aromatic hydrocarbons
- . Halogenated silicones

Actually, many of the synthetics are much over-rated as far as thermal stability is concerned. This particularly is true of the esters.

Their chief advantage lies in their very favorable viscosity-temperature properties which permit operation over a wide temperature range. At high temperatures, above 450°F, many of the mineral oils are more stable than the esters, although they have limited low-temperature capability because of viscosity and pour point. Of the fluids listed above, the polyphenyl ethers are reputed to be the most thermally stable compounds, with decomposition temperatures in the range from about 700 to 850°F. Like the mineral oil derivatives which are stable at high temperatures, the polyphenyl ethers also have high pour points and relatively poor viscosity-temperature characteristics.

When the requirement for radiation stability is superimposed on the need for high thermal stability, the list of candidate fluids shrinks rapidly. Halogenated fluids, containing elements such as fluorine, bromine or iodine, are very poor choices for combined thermal and radiation exposure because of the danger of producing corrosive compounds. Even without radiation, most of these halogenated fluids can become corrosive at high temperatures, on the order of 500° to 600°F.

Although silicones are generally considered to have poor radiation resistance, high phenyl content silicones, particularly those also containing conjugated aromatic substituents have been found to be very tolerant to radiation because of the stabilizing effect of the aromatic groups (5.8-7). However, these gains in radiation resistance are made at the expense of other physical properties, thus negating the good qualities of this class of fluids.

All of the work which has been reported to date on radiation-resistant fluids (e.g. 5.8-5, 6, 8, 9 and 10) has shown that the most stable compounds are hydrocarbons containing aromatic structures. These include

the polyphenyl ethers, aromatic hydrocarbons such as terphenyl, derivatives of these aromatic hydrocarbons, and mineral oils with a high aromatic content. The super-refined mineral oils, prepared from aromatic base stocks, would probably be good candidates although these oils are not as thermally stable as the polyphenyl ethers. However, it is very doubtful if any quantity would be available commercially. Most of the effort on the super-refined oils has been concentrated on the paraffinic base oils because these have the best physical properties for wide temperature ranges (5.8-10).

A particularly good discussion of the problem of the radiation stability of petroleum oils can be found in the work reported in (5.8-6). The following data, abstracted from (5.8-11), summarize the effect of radiation and temperature on typical polyphenyl ether type fluids. Although it was not specified, these results were apparently obtained in air.

Comparative Radiation Damage of Polyphenyl Ether Type Fluids at 375°F

	Dosage (rads x 10 ⁹)	Radiation damage (% decomposition)			Radiation yield	
		Volatile	Residue	Total	Gt	Gr
Polyphenyls	1.78		2.44	2.46	6.77	6.72
Polyphenyl ethers	1.78	0.52	9.40	9.95	18.08	17.10
Aromatic C-ethers	1.78	0.67	17.61	18.33	33.51	32.05

^aGt = Total decomposition; Gr = polymer formation

These results were obtained with much higher dosages than would be encountered in this application. It was reported that these fluids

would be indefinitely stable to radiation exposure of 10^6 or perhaps 10^7 rads.

Since there is a direct connection between radiation and thermal stability, because of the similarity in the decomposition mechanisms, these types of fluids appear to offer the greatest promise for this application. There does not appear to be any good reason for extending this list to include other types of oils, especially those with obvious deficiencies, such as the halogenated compounds.

Based on the background and experience cited above, the best approach is to use those fluids, such as the polyphenyl ethers, for which test data and physical property data are already available. Property data on some of these fluids are given in Table 5.8-1. The fluids include:

1. Monsanto MCS 210 (mixed isomeric four-ring polyphenyl ether).
2. Shell VRT Fluid E (mixed isomeric five-ring polyphenyl ether, predominantly meta-linked).
3. Monsanto MCS 418 (polyaromatic fluid, structurally related to polyphenyl ethers).
4. Shell ET 378 (mixed isomeric four-ring polyphenyl ether).

The Shell VRT Fluid E (5P4E) was selected because of its low vapor pressure and high thermal stability.

It also was concluded to limit organic lubrication to 400°F inlet temperature to avoid decomposition.

TABLE 5.8-1

TYPICAL PHYSICAL PROPERTY DATA ON CANDIDATE FLUIDS

	Monsanto MCS (four ring poly- phenyl ether)	Shell VRT Fluid E (5 ring polyphenyl ether) (5P4E)	Monsanto MCS 418 (Aromatic ether, similar to poly- phenyl ethers)	Shell ET-378 (four ring polyphenyl ether)
Viscosity-Centistokes				
100 °F	63.2	355	25.2	68.82
210	6.0	13.1	4.13	6.17
400	1.31	2.1	0.81 (500 °F)	1.35
700	0.48	0.92	0.48	
Pour-Point	+10 °F	+35 °F	-20 °F	5 °F
Density g/ml				
100 °F	1.17	1.18	1.184	1.16
210	1.12	1.138		
400	1.04	1.058	1.101 (300 °F)	1.03
700	0.90	0.973 (600 °F)	0.926	0.89
Vapor Pressure mm Hg				
520 °F	10 mm	0.7 mm		4.8 mm (500 °F)
680 °F	100 mm	30 mm	140 mm (700 °F)	
Thermal Stability (Isoteniscope)				
840 °F		870 °F	Boils at 795 °F	< 800 °F
Specific Heat				
100 °F	0.36	0.39	0.347	0.36
300 °F	0.43	0.47	0.418	0.407
500 °F	0.50	0.58	0.488	0.453
700 °F	0.56			0.505
Thermal Conductivity BTU hr.sq.ft. °F/ft.				
100 °F	.0768	0.0783	0.0714	0.080
300 °F	.0702	0.0773	0.0690	
500 °F	.0642	.0762	0.0585	

Potassium Lubricant

In computations of potassium-lubricated bearing performance the density was obtained from (5.8-12) and the viscosity was obtained from (5.8-13). Shown in Figures 5.8-1 and 5.8-2 are these properties plotted against temperature. Information on conductivity and heat of vaporization may be found in (5.8-14 and -15).

Journal Bearings

The two basic types of fluid film bearings are:

- (a) The externally pressurized (hydrostatic) bearings, where the load capacity is generated principally by introducing high pressure lubricant into several pockets or recesses in the bearing through orifices or other restrictors. The load capacity of this type of bearing is, therefore, principally dependent on the available pressure difference between the high pressure lubricant source and the ambient pressure.
- (b) The self-acting (hydrodynamic) bearing where pressures are self-generated in the fluid film between the rotor and stator by shear of the thin, viscous, lubricant film. These self generated pressures serve to separate the rotor and stator elements. The load capacity of this type of bearing is, therefore, principally a function of the speed and lubricant viscosity. The lubricant supply to the bearing need not be pressurized, except to the extent required to overcome the losses in circulating the flow to lubricate and cool the bearings.

Calculation of the load capacity, as well as fluid film stiffness

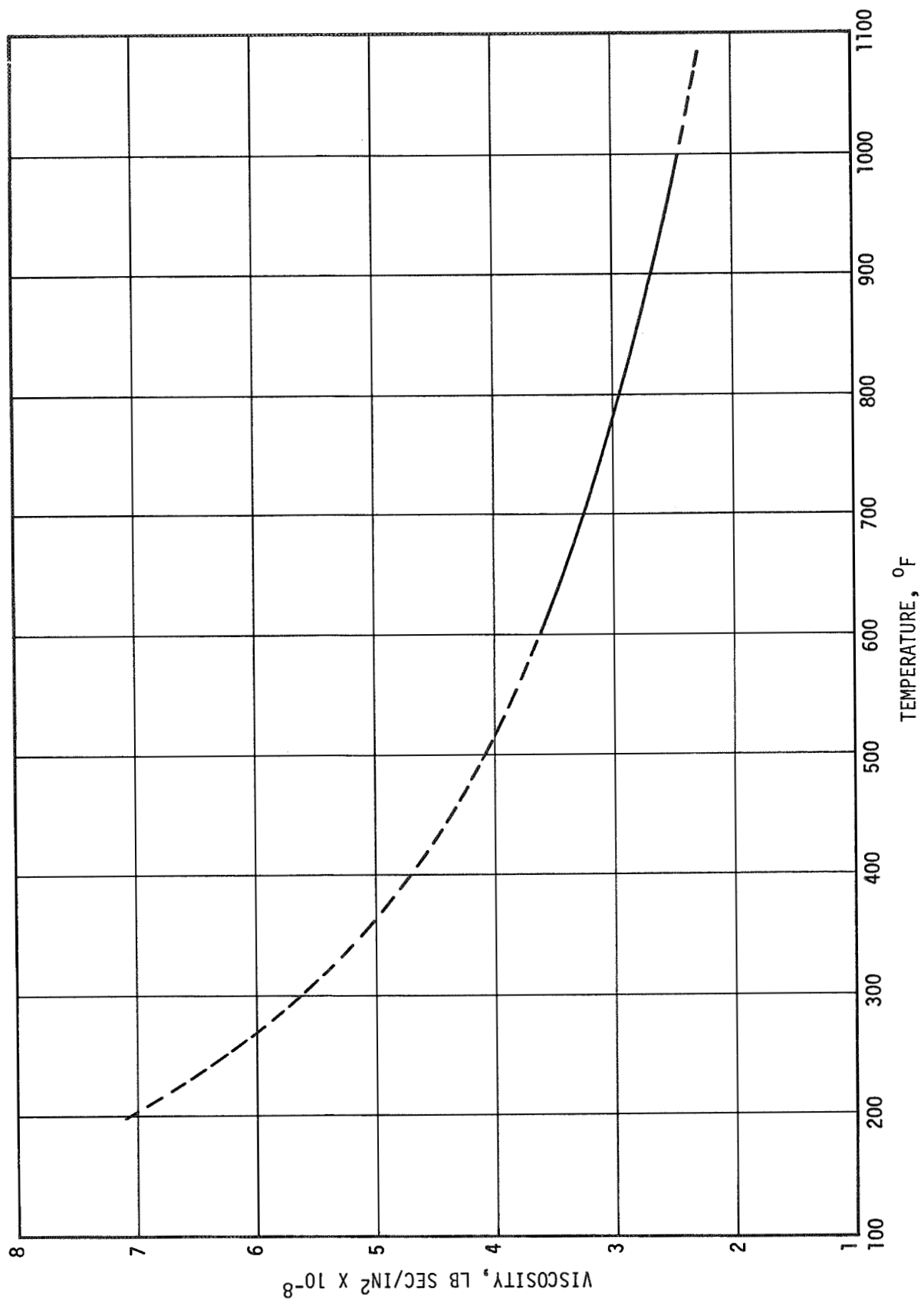


Figure 5.8-1. Liquid Potassium Absolute Viscosity. (Reference 5.8-9)

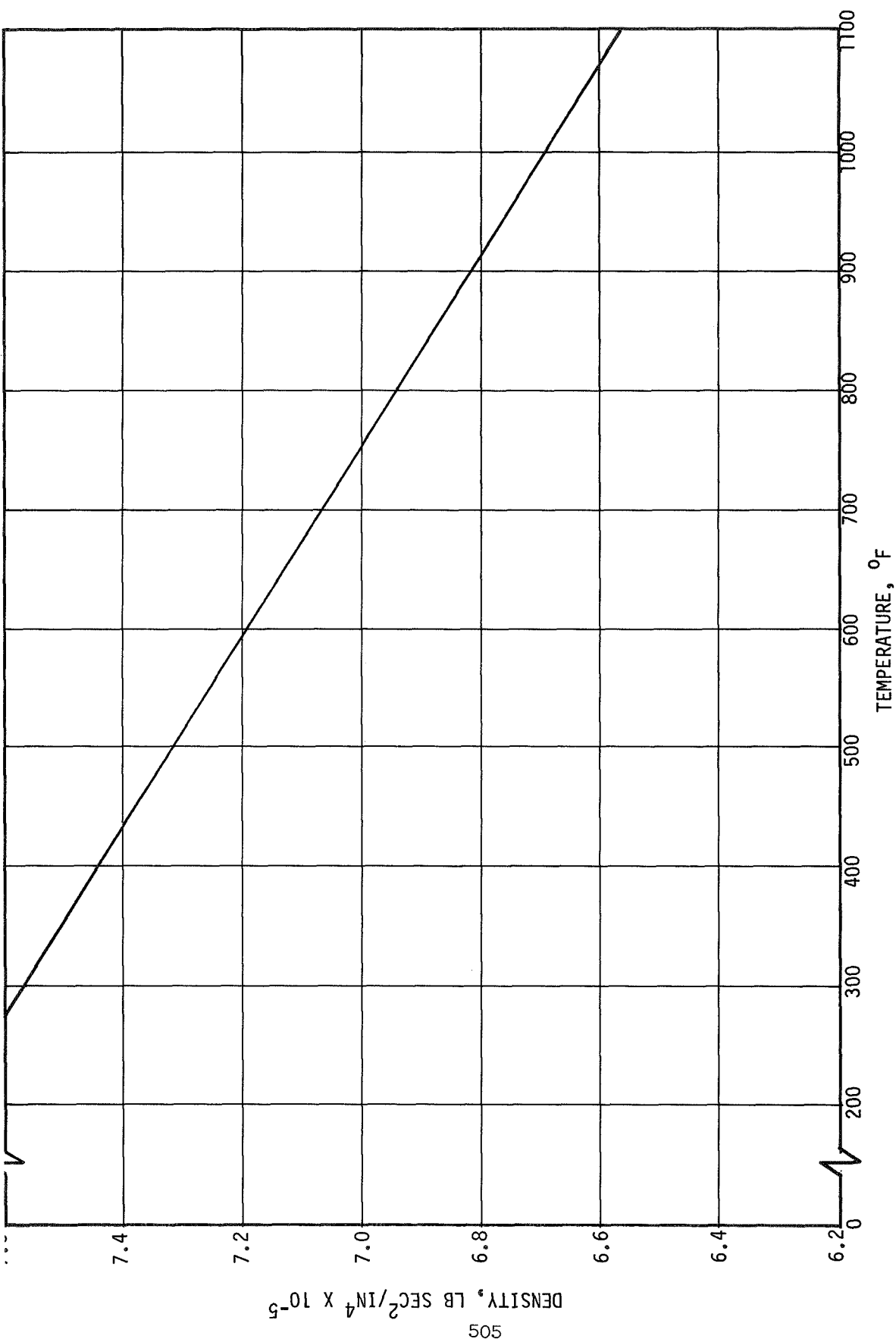


Figure 5.8-2. Liquid Potassium Density. (Reference 5.8-8)

and damping, for all the rotors in the present study showed that the self generated bearing pressures were entirely adequate so that the journal bearings could in all cases be of the self acting type. The self acting bearing type is preferred for space power units since it does not require a flow of high-pressure lubricant. The special provision of a source of high-pressure liquid would impose a significant penalty on the system. Accordingly, self acting journal bearings were selected for the KTA rotors.

Criteria

The self-acting bearings for space power rotating machinery have to satisfy the following criteria:

1. They must be inherently stable, even under zero net radial load, as in operation in gravity-free field. (This requirement rules out such bearings as the plain-circular bearing).
2. High load capacity, fluid film stiffness and fluid film damping are desirable for operation on a thick lubricant film, with margin for off design conditions, and for low vibration amplitudes under dynamic loads.
3. Self alignment provisions are desirable to tolerate thermally or otherwise induced misalignment. These are needed also so that when the dynamic rotor orbits are of a conical mode, the film thickness will remain essentially constant along the bearing length.
4. Low power loss, so as to minimize parasitic losses.

Tilting Pad Bearing

In view of these requirements, the tilting pad bearing which, properly designed, is inherently stable and seal aligning, is the principal bearing choice. The tilting pad bearing also has somewhat higher fluid film stiffness and damping than other self acting bearing types. This is illustrated by a comparison that was made of four stable bearing types, intended for a typical potassium turbine rotor. The comparison was made on the following basis:

Rotor speed = 20,000 RPM

Lubricant = Potassium at 950°F (viscosity = 2.34×10^{-8} lb. sec/in²
density = 0.026 lbs/in³)

Bearing diameter = 2"

Bearing length = 2"

Bearing diametral clearance = 0.002"

At the concentric shaft position (i.e. with zero net load on the radial bearings), the calculated bearing stiffness and damping were:

Bearing Type	Radial Bearing Stiffness (lbs/inch)	Radial Bearing Damping (lb. sec/inch)
Tilting pad with 0.5 preload coefficient	12.4×10^4	71.4
Shrouded step	9.2×10^4	38.4
Herringbone grooved	7.6×10^4	46.2
Three lobe with 0.5 preload coefficient	3.2×10^4	51.9

Furthermore, tilting pad bearings have been successfully operated in liquid alkali metals, so that there exists a body of laboratory and field experience with these bearings that generally substantiates their calculated performance. NaK lubricated bearings are, for example, used

in the NaK PMA pump of SNAP 8. Potassium lubricated tilting pad bearings have been tested to provide technology information under SNAP 50/SPUR. Also, Mechanical Technology Incorporated has conducted an AEC sponsored liquid metal bearings technology program in which a 50 lb. rotor supported on 2" diameter, NaK lubricated tilting pad bearings was operated over a speed range from 0 to 26,000 RPM. The steady load on the bearings was continuously varied from 0 to 80 psi, both with a balanced rotor and with 0.4 ounce-inches of unbalance. Operation was entirely stable and the film thickness and dynamic orbit measurements that were made showed generally close agreement with the theoretically predicted performance (5.8-16).

Number of Pads - A tilting pad bearing can be made up of three or more pads with three, four and five pad bearings being common in industry. Although a properly designed bearing with any number of pads can be successfully applied to the KTA, four pads have been selected here for the following reasons:

Start-up - The four-pad bearing, when loaded between pads has the highest load capacity of any configuration at very high eccentricity ratios (.92 - .97). For component ground testing where horizontal operation may be required, the four-pad bearing will, therefore, have the lowest lift-off speed. This is shown in Table 5.8-2 where the relative load capacities of tilting pad bearings configurations are compared. It will be noted in Table 5.8-2 that for more moderate eccentricity ratios (.4 and .85) that the three-pad bearing has somewhat higher load capacities. However, in all turbine and alternator configurations studied, the bearing loadings are quite small and the extra margin at low

TABLE 5.8-2

SUMMARY OF RELATIVE BEARING CHARACTERISTICS

Number of Pads	Direction of Steady State Load	Start-up Relative Load Capacity	Max Long Time Relative Load Capacity ($\epsilon = .85$)	Normal Relative Operating Load ($\epsilon = .4$)
3	Between Pads	.84	1	1.36
4		1	1	1
5		.95	.76	.67
6		.87	.61	.52
3	Into Pad	.78	.95	1
4		.71	.71	.71
5		.59	.49	.51
6		.50	.39	.42

eccentricity ratios is unnecessary.

Seal Clearance - Although the bearing clearance is defined by the pivot circle, the seal clearances must be set by the total distance that the shaft can move within the bearing. This motion is the distance between pivots which is defined as:

$$C_{\text{Seal}} = \frac{C_{\text{brg}}}{\cos \frac{360}{2n}}$$

Where C_{seal} and C_{brg} are the seal and bearing radial clearances, respectively, and n is the number of pads. This relationship is shown more graphically in Figure 5.8-3, where it can be seen that the larger the number of pads, the smaller is the required seal clearance. There is a sizable reduction in clearance between three and four pads but as the number is increased beyond four, the reduction gets progressively smaller and does not warrant the extra complexity of additional pads.

Experience - Most of the successful liquid metal lubricated bearings including the SNAP-8 NaK PMA, the high temperature NaK bearing test machine at Mechanical Technology, Incorporated, and the potassium bearing experiments at General Electric have been on four pad bearings.

Alternate Journal Bearings

As was indicated earlier, one of the most important requirements of the KTA bearings is that of stability or freedom from half frequency whirl. The stability problem in high-speed liquid-metal-lubricated machinery where the lubricant film is almost always turbulent has plagued the development of space power systems for years. For this

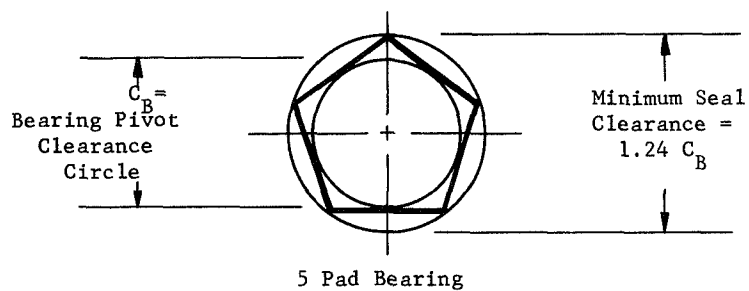
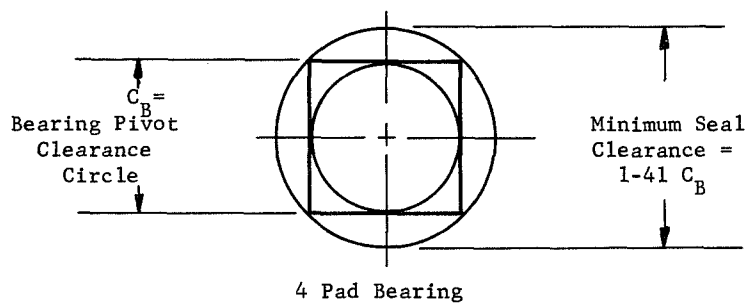
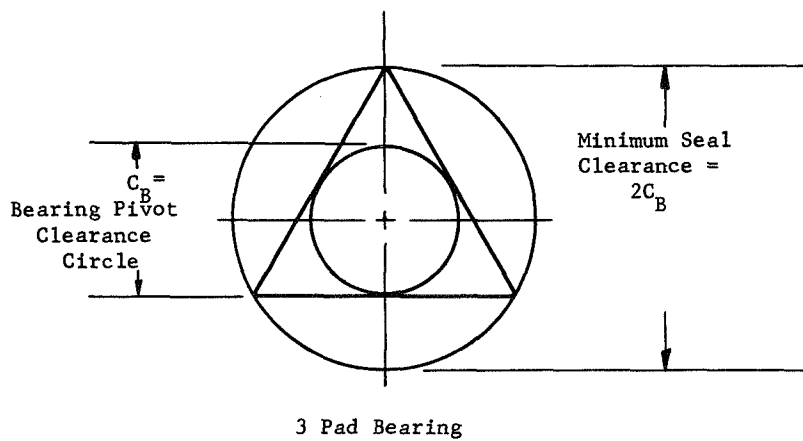


Figure 5.8-3. Comparison of Required Seal Clearances.

reason the tilting-pad bearing has emerged as the standard while the search for a less complicated stable bearing type which does not possess the potential pivot-wear problem has continued.

The three types mentioned earlier have the greatest potential for high stability limits and have been also compared for stability under the same conditions as indicated above. At the zero eccentricity position the calculated critical masses were:

<u>Bearing Type</u>	<u>Critical Mass (lbs)</u>
Shrouded Step	109
Herringbone Grooved	93
Three Lobe with 0.5 preload coefficient	39

The critical mass is defined as the rotor mass associated with the bearing under which stability can be assured. For example, a rotor weighing 218 pounds (2×109) and having a uniform mass distribution between two shrouded step support bearings could be expected to operate stably up to 20,000 RPM. If supported on three lobe bearings, the maximum rotor weight would only be 78 pounds (2×39).

Of these three bearing types, the shrouded-step bearing was chosen as the alternate bearing type because, in addition to its higher stability limit, it does not possess the inherent cooling problem of the herringbone-grooved bearing. (If the herringbone-grooved bearing is to operate hydrodynamically by taking the lubricant from a flooded cavity surrounding it, some fluid must be bled from the center plane to remove the viscous losses). Furthermore, the shrouded step bearing has been successfully applied to a large high-speed, water-lubricated (which has similar lubrication properties to potassium) turbo-

compressor which has potentially greater stability problems than does the KTA.

As was indicated before, it is strongly recommended that the bearings be self-aligning to accommodate manufacturing, thermal, and structural flexibility misalignments as well as misalignments resulting from rotor conical whirl modes. A method commonly utilized to accomplish this with solid geometry journal bearings is the tangential spoke flexure, which is shown in Figure 5.8-4. The spoked mount provides for high radial mounting stiffness necessary for rotor dynamics and at the same time gives low angular mounting stiffness required for self alignment. The tangential spoke arrangement permits utilizing materials of different thermal expansion coefficient (such as cemented carbide bearing, a high-strength alloy spoke, and an austenitic stainless housing) since most of the differential growth with temperature can be accommodated by static rotating of the bearing.

Performance

The KTA aerodynamic and electrical designs considered in Phase I covered a speed range from 16,000 RPM to 32,000 RPM and these designs necessarily required several different bearing sizes. To aid in sizing bearings, evaluating losses, and calculating dynamic response of these several configurations a series of bearing performance curves were prepared and were used as standard reference design data for Phase I. These curves defined load capacity, power loss, stiffness and damping as a function of size at 16,000 RPM, 20,000 RPM, 24,000 RPM, and 32,000 RPM for typical potassium lubricated tilting pad journal bearings. Only the curves for 24,000 RPM 5P4E organically lubricated

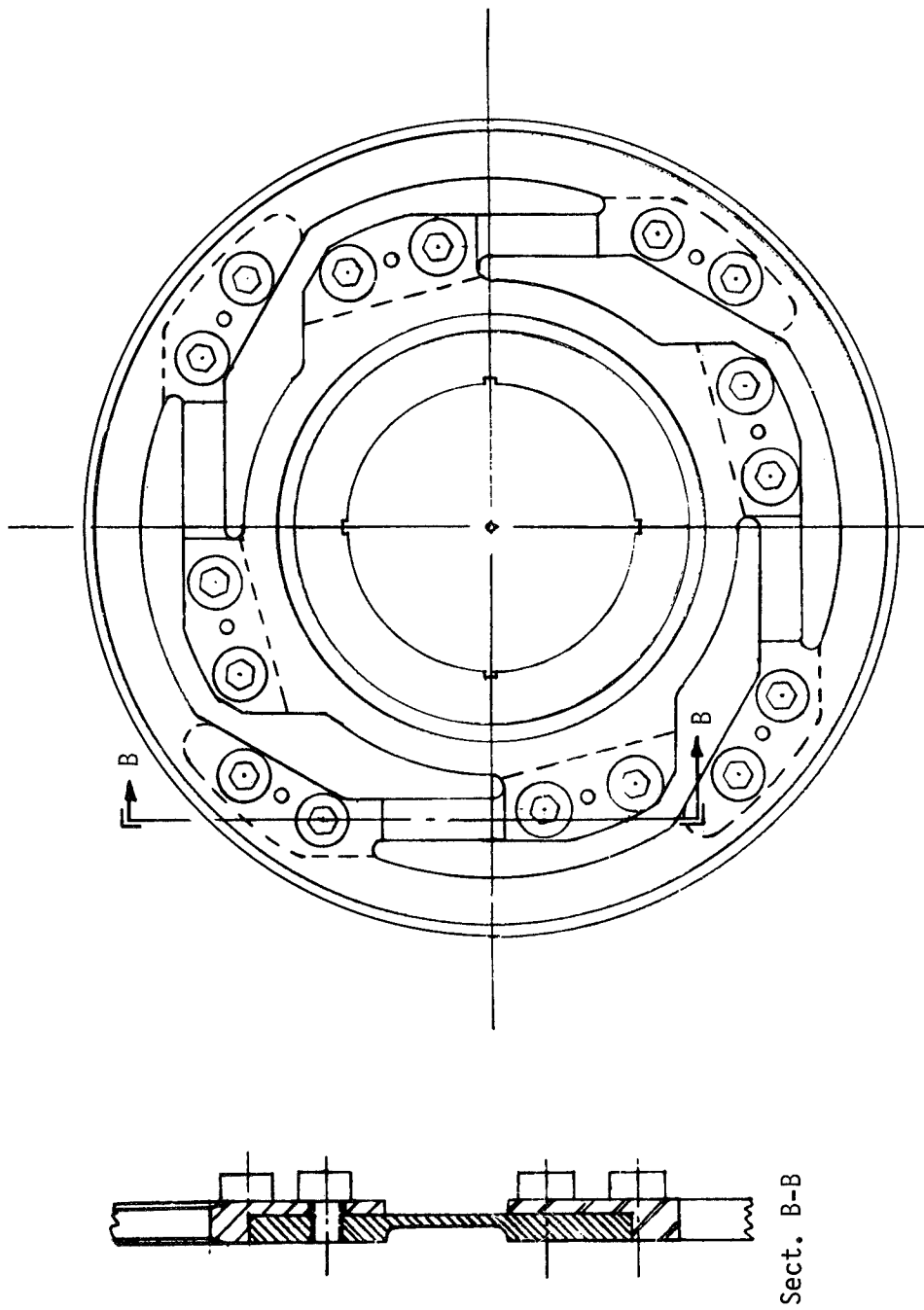


Figure 5.8-4. Self Aligning Step Journal Bearing.

operation (Figures 5.8-5 to -8) and 20,000 RPM potassium lubricated operation (Figures 5.8-9 to -12) are presented here since these are the speeds selected as optimum during Phase I. (Although the potassium lubricated KTA design speed is 19,200 RPM, the 20,000 RPM data is considered close enough for the purposes of the Phase I study). This performance data is based upon the turbulent lubrication theory developed at Mechanical Technology, Incorporated which has been experimentally verified at high speeds utilizing NaK as the lubricant (5.8-12).

Size

The minimum bearing size was established by the following criteria:

Under the force of gravity and with an unbalance in inches equal to 4 times the rotor weight divided by the design speed, the bearing eccentricity ratio could not exceed 0.5 with the load directed into the pad.

With such a criteria, the maximum eccentricity ratio that will be experienced under the specification conditions of 3.93g acceleration plus 3g shock is 0.84. This is shown on the curve of reciprocal Sommerfeld number versus eccentricity ratio on Figure 5.8-13 for the load applied between two pads and directly into a pad, and neglects the additional load support derived from squeeze film forces generated under conditions of shock and vibration. In general, all bearings exceed the minimum size dictated by this criterion for reasons of either required shaft or bearing stiffness dictated by dynamic considerations, as will be discussed later.

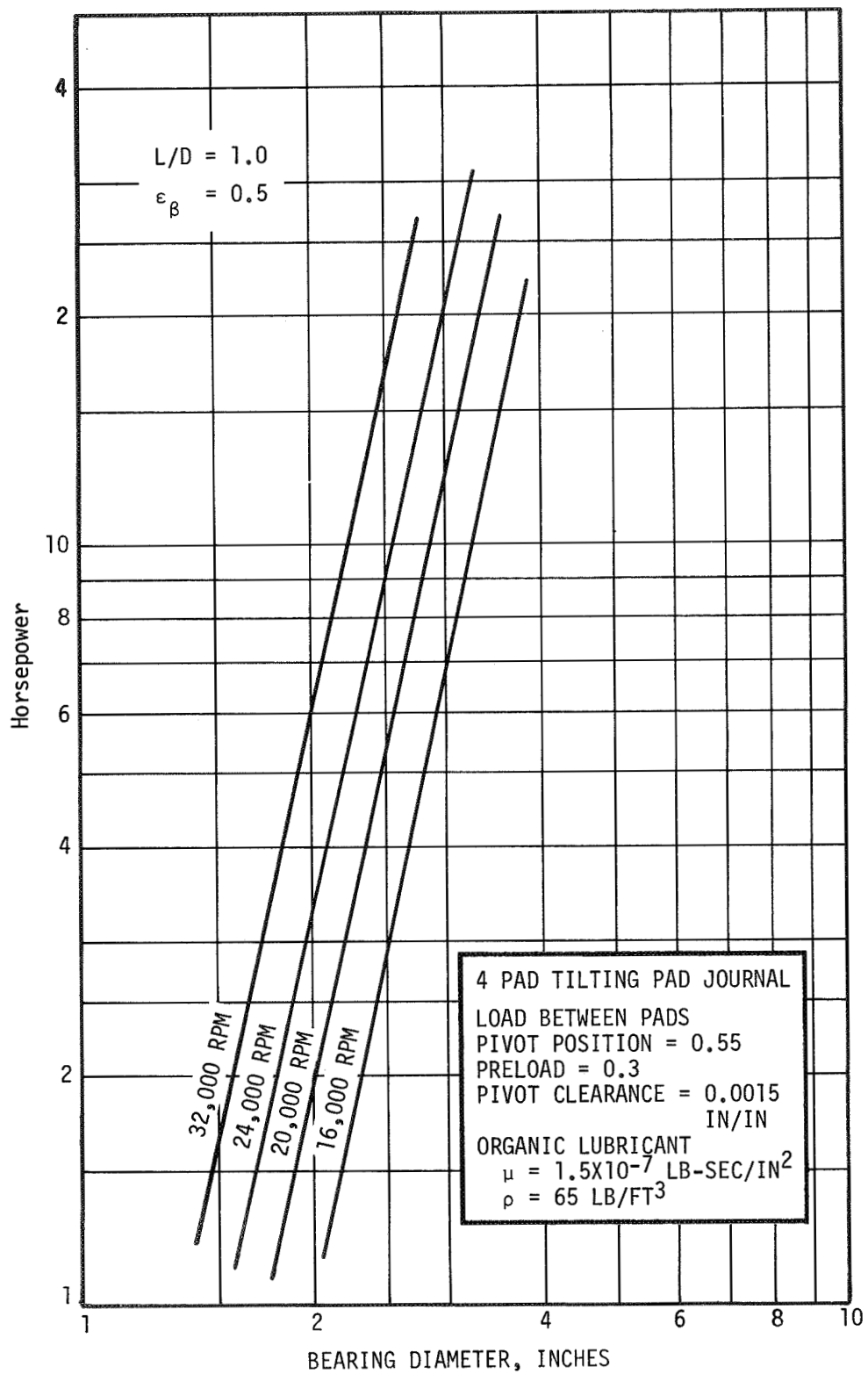
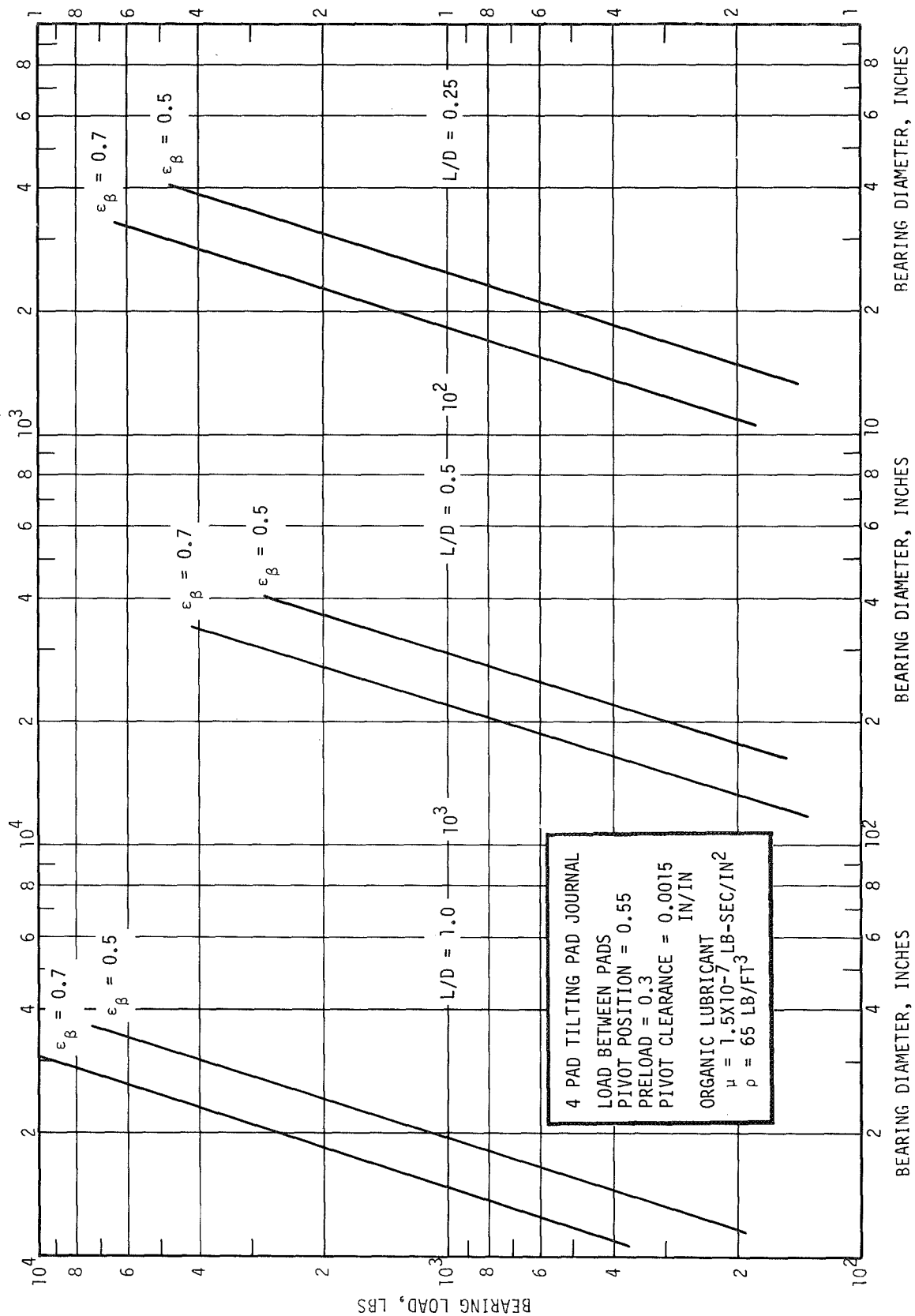


Figure 5.8-5. Bearing Power Loss With Organic Lubricant.



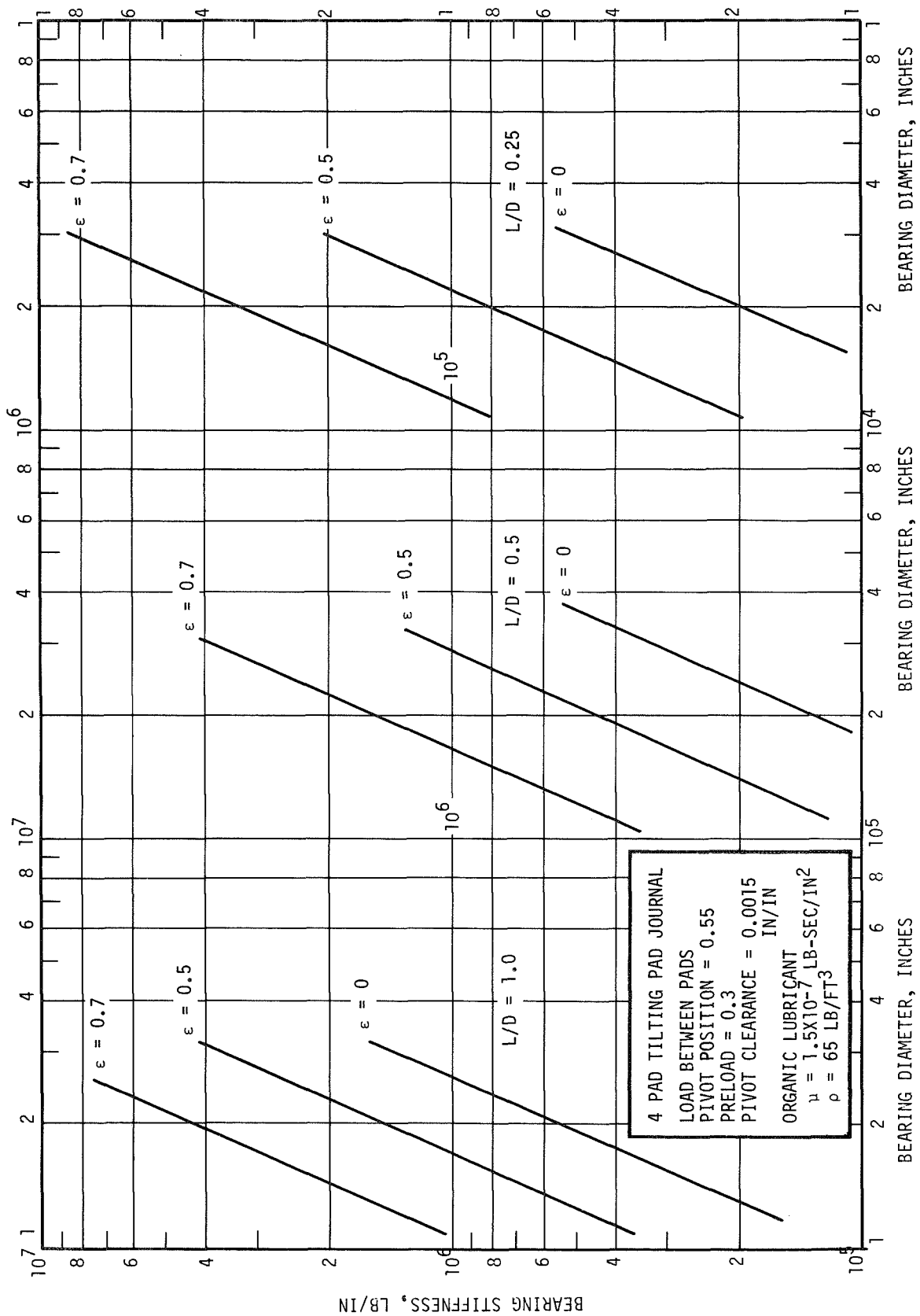


Figure 5.8-7. Bearing Stiffness at 24,000 RPM With Organic Lubricant.

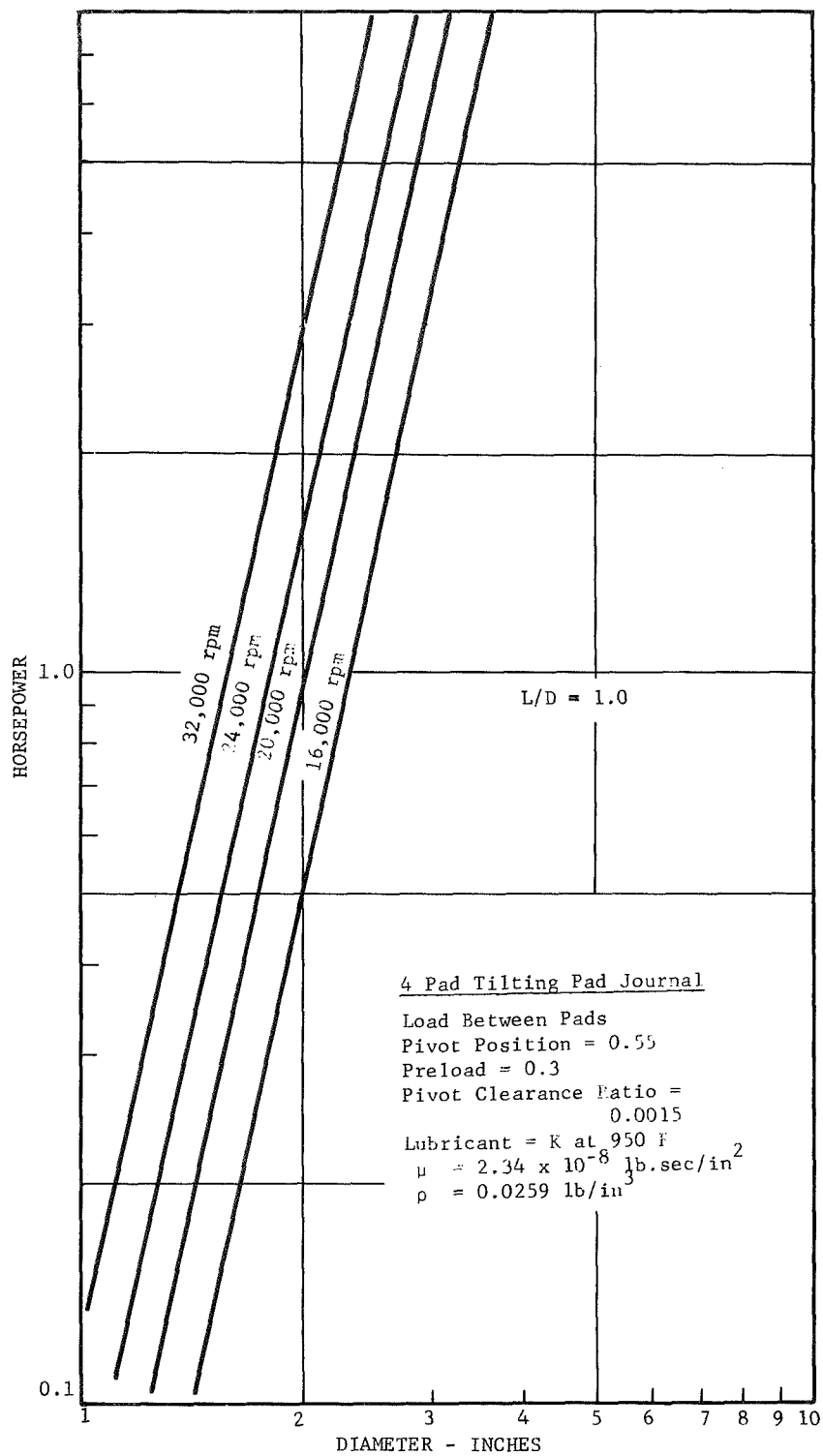


Figure 5.8-9. Bearing Power Loss with Potassium.

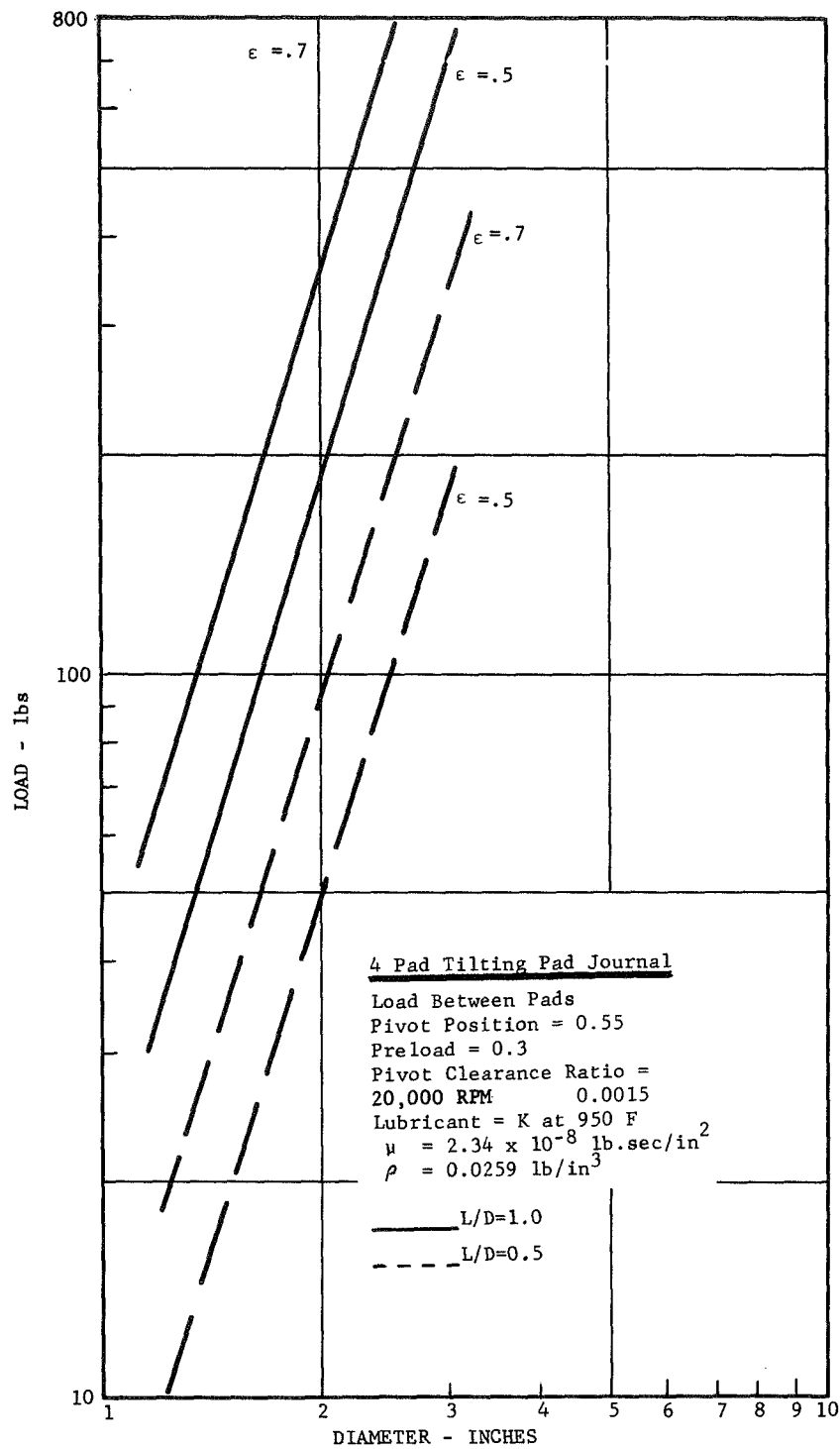


Figure 5.8-10. Bearing Stiffness with Potassium.

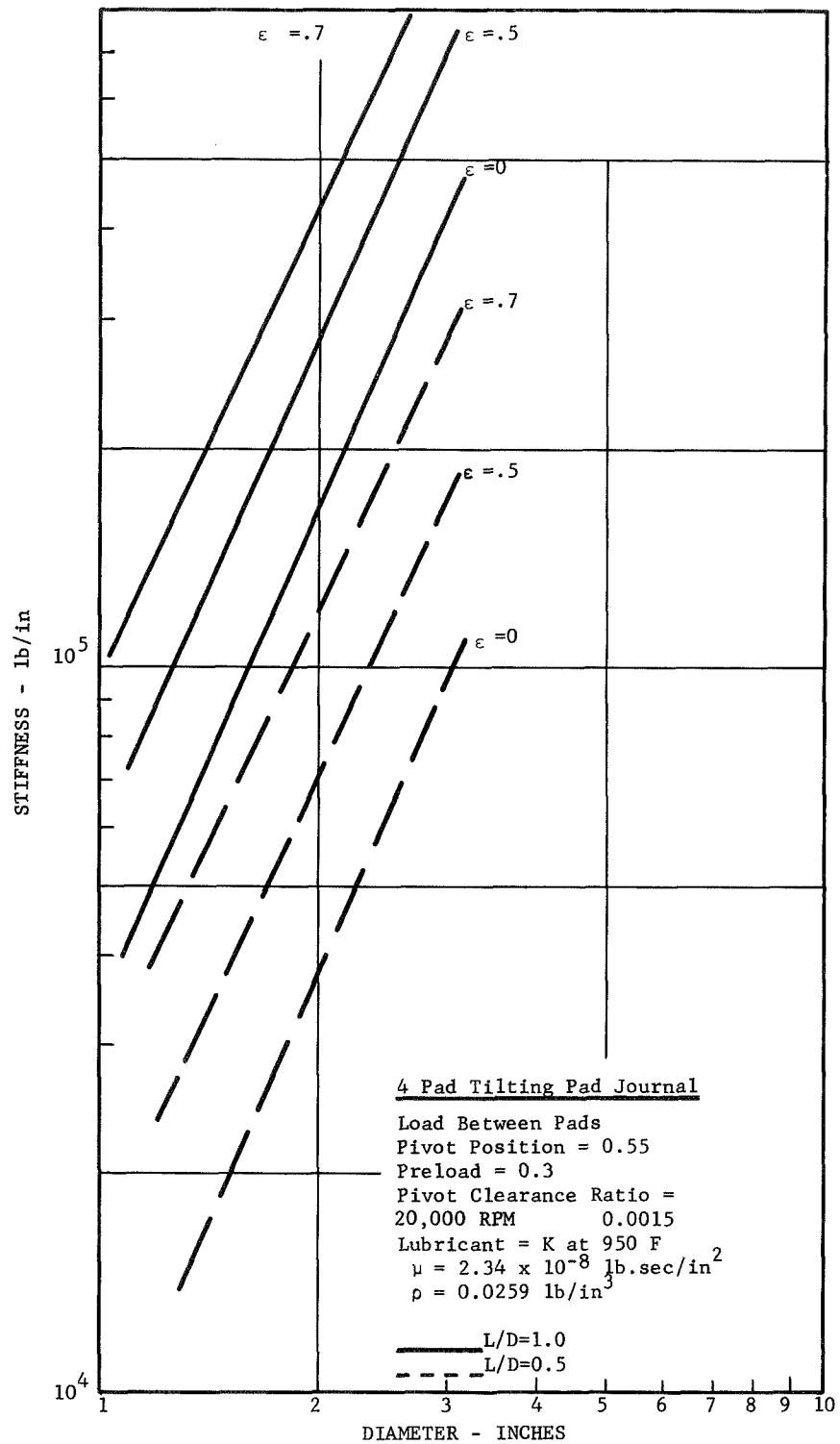


Figure 5.8-11. Bearing Stiffness with Potassium.

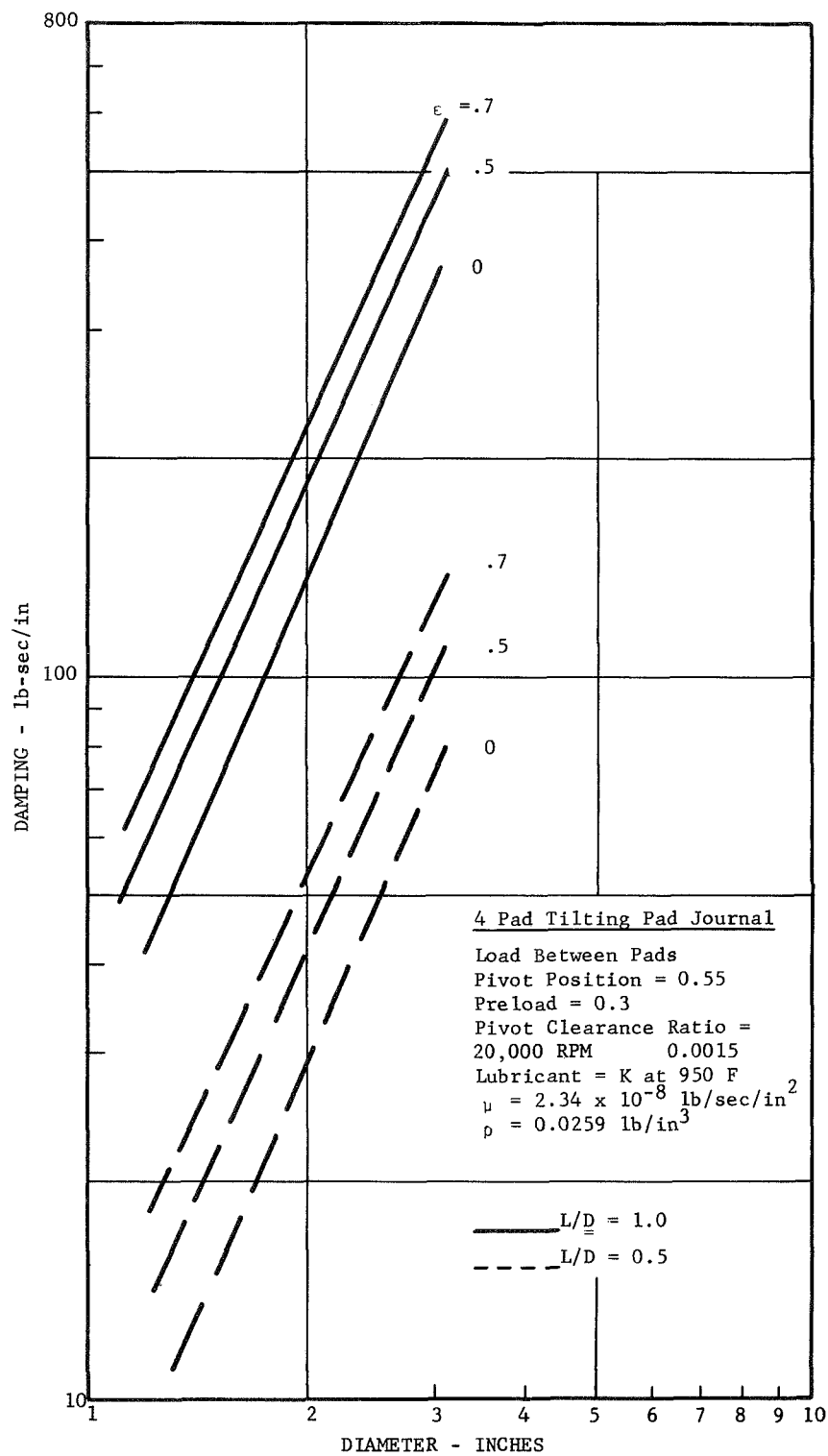


Figure 5.8-12. Bearing Damping with Potassium.

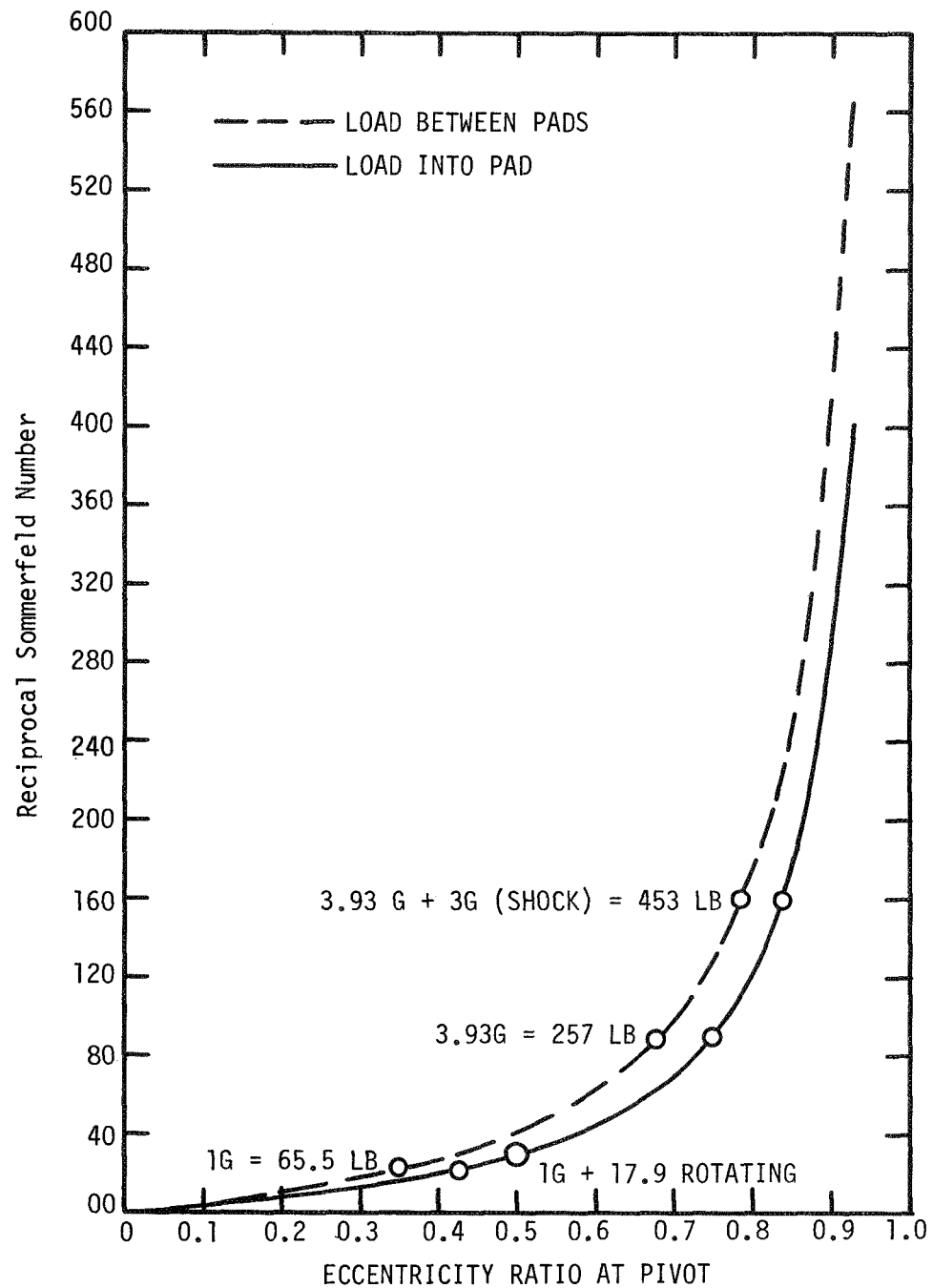


Figure 5.8-13. Static Load Versus Deflection. 4-Pad, Tilting-Pad Journal Bearing; Reynolds No., 10,000, Preload, 0.3; Length/Diameter, 1.0.

Thrust Bearings

Since the coupling between the turbine and alternator is not capable of supporting machine thrust forces, two thrust bearings are required in the KTA: one in the turbine and one in the alternator. The thrust loads resulting from aerodynamic and 3.93g forces are moderately high (in the range of 1000 to 2000 lbs) and it is desirable to keep the thrust bearings as small as possible to minimize the parasitic losses. For this reason the steady state thickness has been kept in the order of .001 inches and the film thickness under conditions of acceleration and shock allowed to approach .0007 inches which, like the journal bearings, necessitates self-alignment features.

The tilting pad bearing has been selected as the primary KTA thrust bearing because of its exceptionally high load capacity and, when mounted on leveling links, is self-aligning. The leveling link design which was developed by Kingsbury has been an industry standard for years and lends itself well to the KTA application. In Figure 5.8-14 is shown a typical tilting pad thrust bearing on leveling links in one of the turbines studied in Phase I.

As the alternate bearing, the shrouded step bearing is being recommended. This is based upon the results of previous studies at Mechanical Technology, Incorporated where high load capacity and low power loss was required in thrust bearings operating in the turbulent regime. (The shrouded-step bearing gives superior load capacity over the common Rayleigh-step and tapered-land types because of the reduced end leakage afforded by the side lands or shrouds.) This bearing has also been successfully applied to high speed water lubricated machine and computer programs are available at Mechanical Technology, Incorpo-

rated to calculate its performance in the turbulent regime.

To make a solid geometry thrust bearing self aligning, it is either necessary to soft mount it on some form of flexure or to mount it freely in a spherical seat. Flexure mounting was discarded early in Phase I because the thrust loads are of such a high magnitude as to result in excessive static deflections of the thrust bearing stator if the mount were to have the angular compliance necessary to insure self aligning capability. The spherical seal mount which incorporates a hydrostatic fluid film between the spherical members is shown schematically on Figure 5.8-15. The hydrostatic supply is abstracted from the step bearing by taking a small amount of fluid from the bearing at each step.

The spherical seal would be designed with the same number of pockets as shrouded pads. Fluid abstracted from the region of the step termination would be conducted through a cross feed passage and through a flow-control restrictor to the corresponding hydrostatic pad on the spherical seat. Since the spherical seat can operate at a much smaller film thickness than the active thrust bearing, for example, 0.002 inches to 0.0003 inches, the flow requirements for pressurization are very small and the influence of the fluid abstracted from the step on the active thrust bearing will thereby be minimized.

A restraining tab, wire, or pin is essential in this design to prevent rotation of the thrust plate. This type of bearing concept applied to one of the turbine concepts is shown on Figure 5.8-16.

Performance

As in the case of the journal bearings, thrust bearing performance

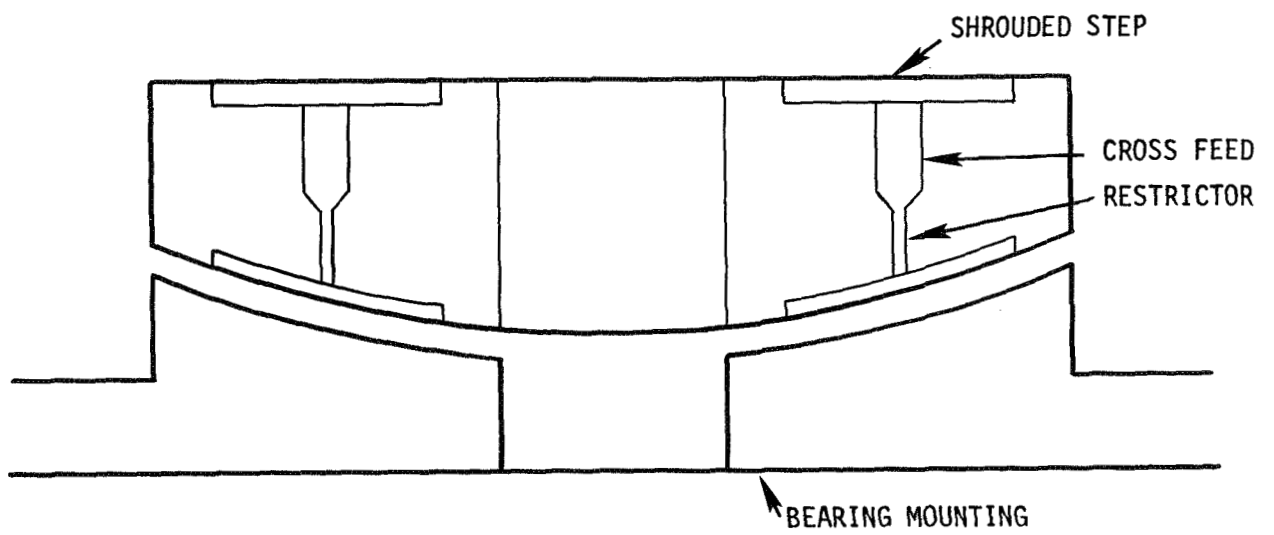


Figure 5.8-15. Schematic Diagram of Spherical Seat Mount for Shrouded Step Thrust Bearing.

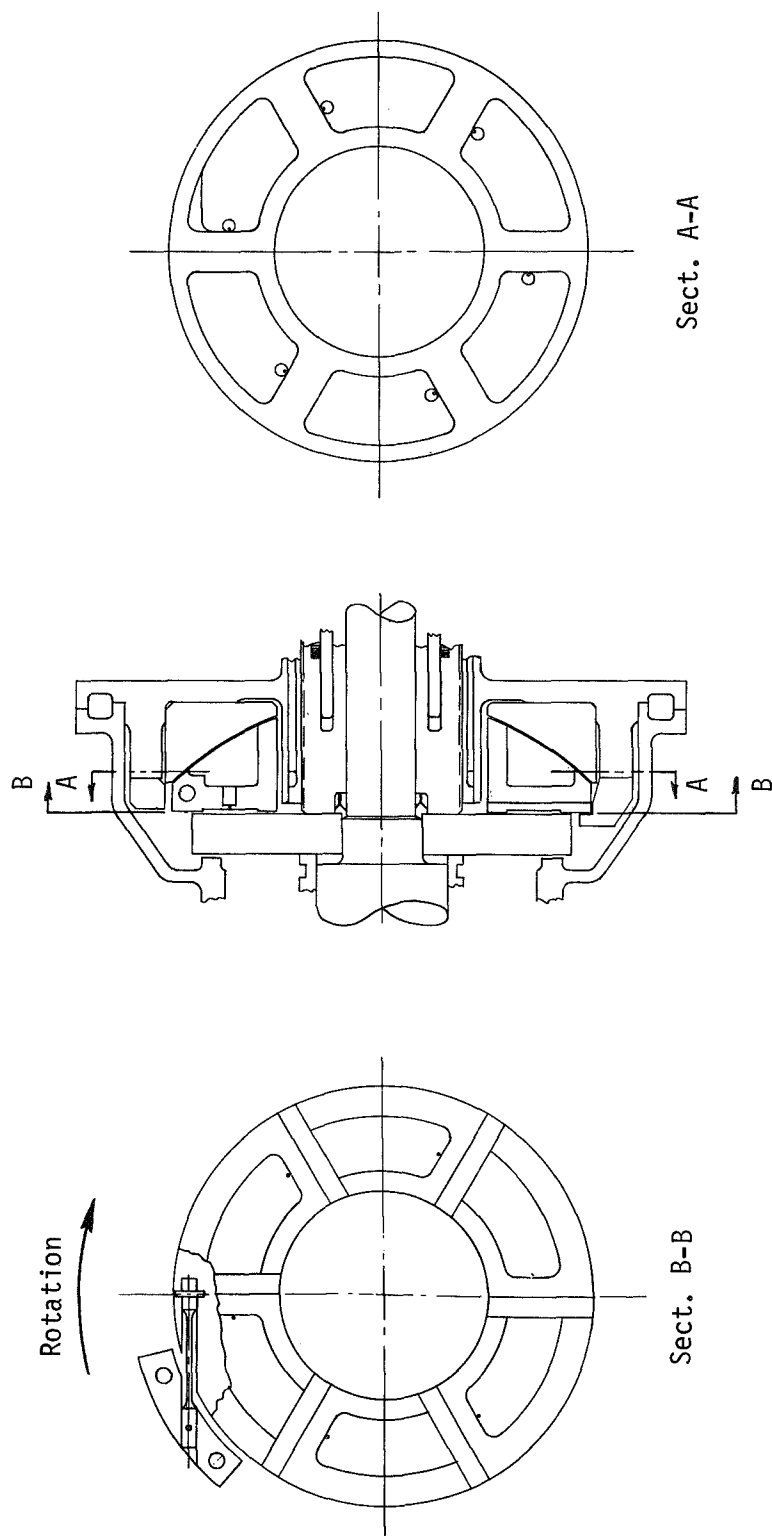


Figure 5.8-16. Self Aligning Step Type Thrust Bearing.

curves for load capacity, film thickness, and power loss as a function of diameter were made at speeds of 16,000 RPM, 20,000 RPM, 24,000 RPM, and 32,000 RPM to aid in sizing for the various machine configurations considered in Phase I. Only the curves for 24,000 RPM oil lubricated operation (Figures 5.8-17 and 5.8-18) and 19,200 RPM potassium lubricated operation (Figures 5.8-19 and 5.8-20) are presented here since these are the speeds selected as optimum during Phase I. The potassium lubricated thrust bearing performance was calculated based upon the design charts presented by Raimondi and Boyd (5.8-17) with approximate corrections for turbulence based upon the linearized turbulent lubrication theory of Ng and Pan (5.8-16). The oil lubricated thrust bearing curves are for the shrouded step design which has very similar performance characteristics to the tilting pad. The shrouded step performance was computer solved by utilizing a Mechanical Technology, Incorporated bearing design program.

Summary and Conclusions

During Phase I, both oil and potassium lubricated bearings were considered. Several organic fluids were evaluated and it was concluded that of the organics known to have good temperature stability and radiation resistance, the five ring polyphenyl ether (5P4E) was the best candidate KTA lubricant.

The performance calculations indicated that self-acting hydrodynamic bearings had more than ample load capacity for all rotor configurations studied. In no case was a potassium lubricated bearing found to be marginal in performance and in most cases, their power consumption was less than its oil counterpart.

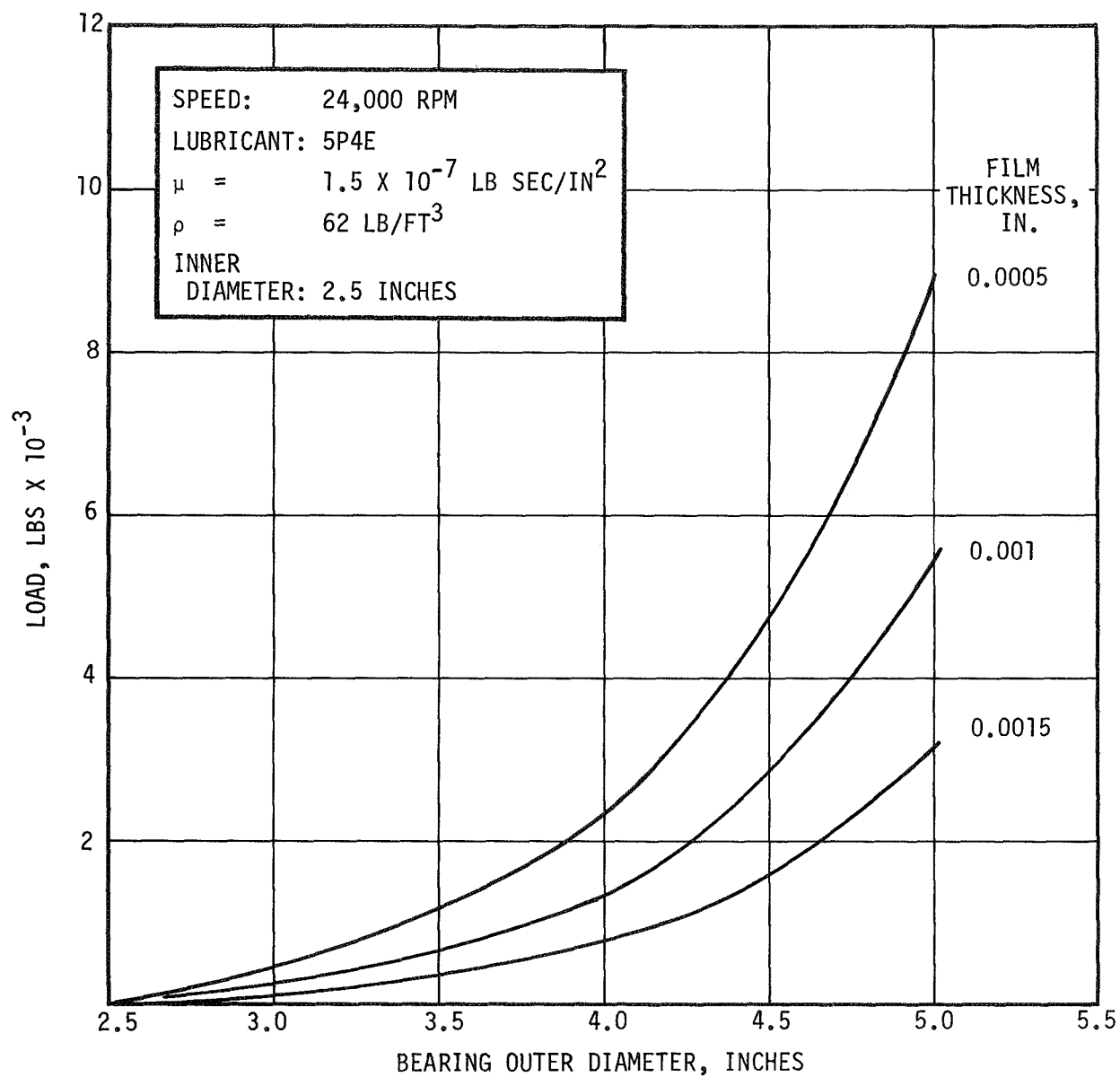


Figure 5.8-17. Shrouded Step Thrust Bearing Load.

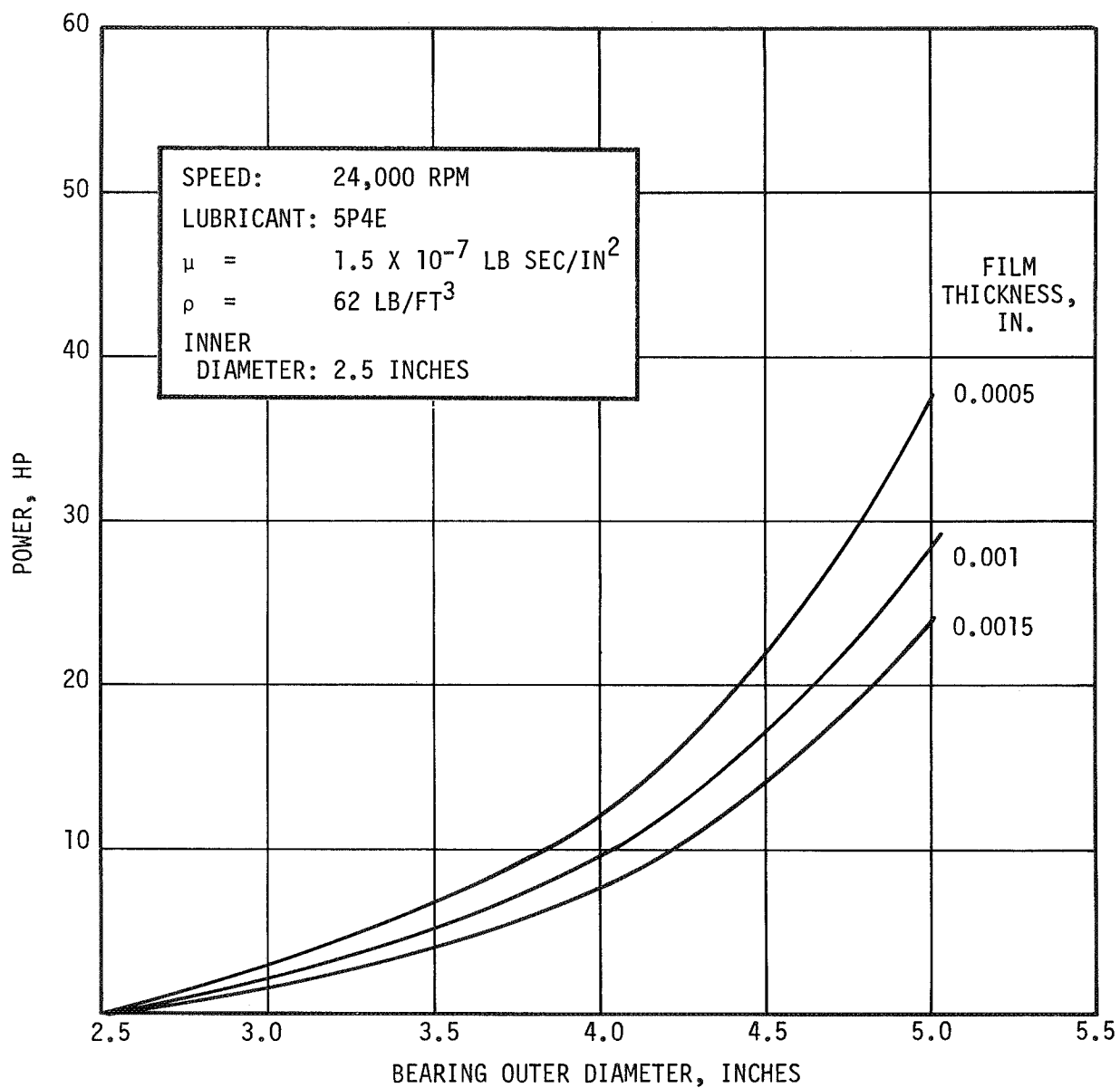


Figure 5.8-18. Shrouded Step Thrust Bearing Power.

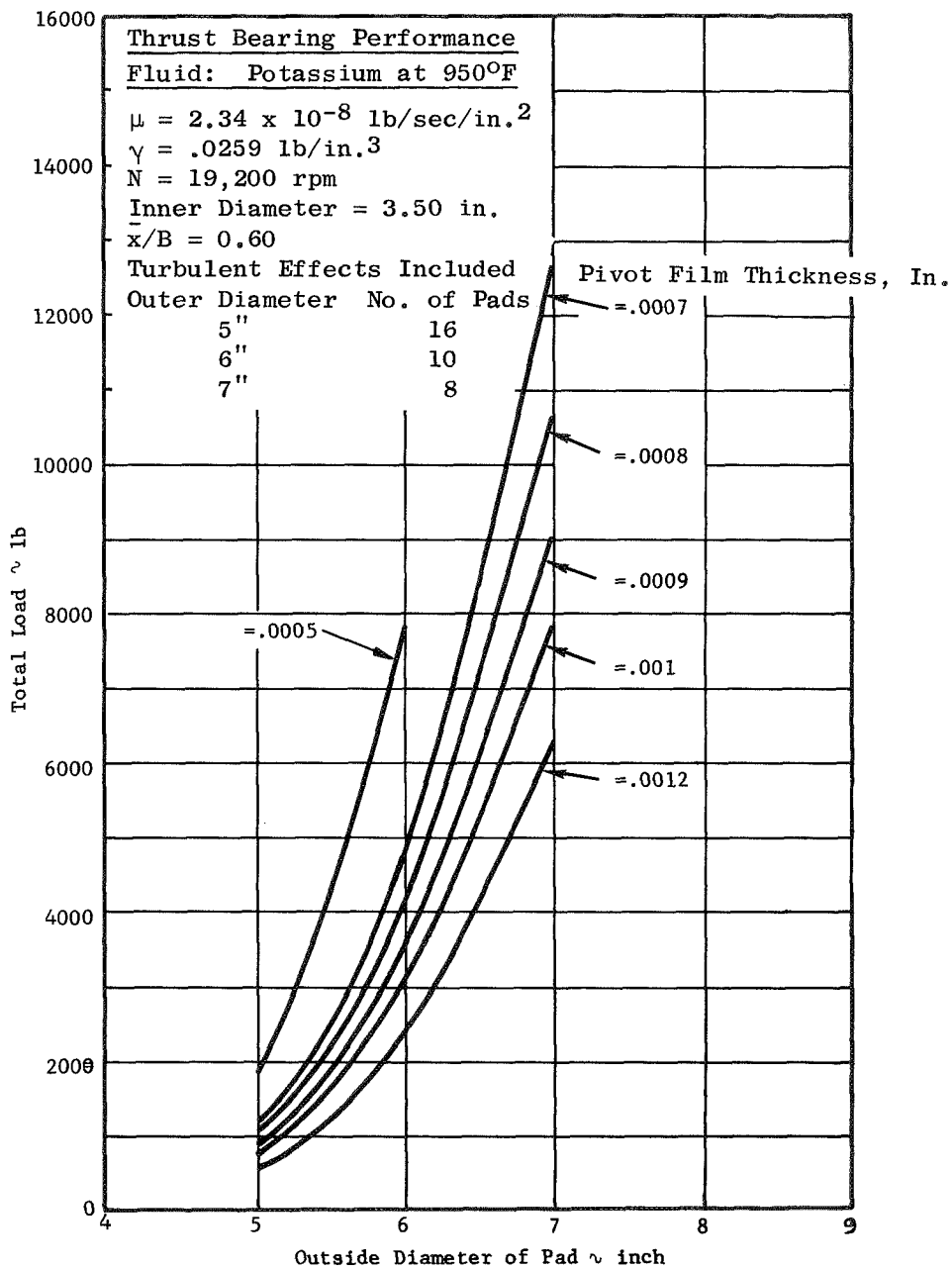


Figure 5.8-19. Tilting Pad Thrust Bearing Load.

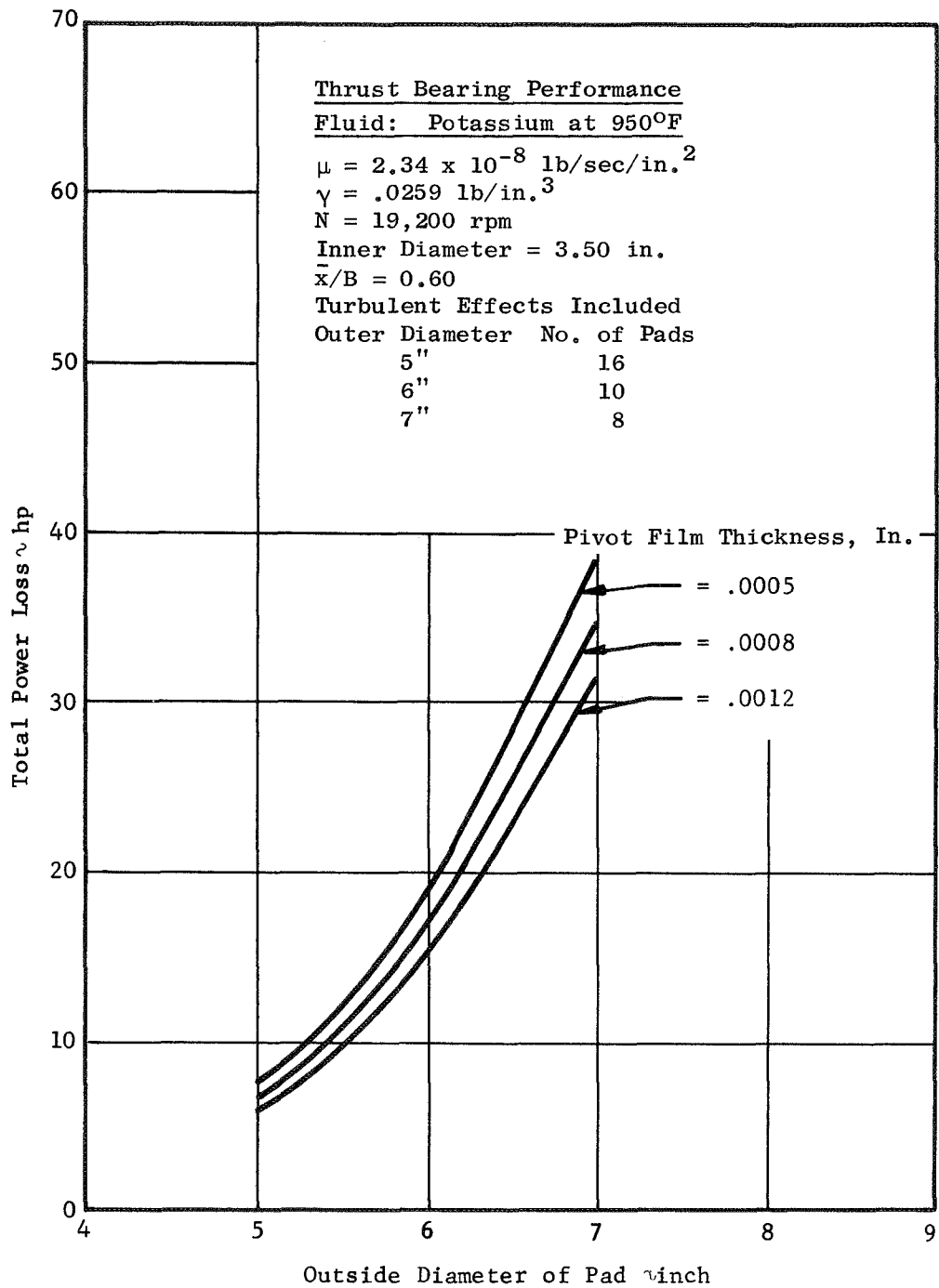


Figure 5.8-20. Tilting Pad Thrust Bearing Power.

Tilting pad bearings (both journal and thrust) were selected as the primary KTA bearing because of their high load capacity, stiffness, and damping; self-aligning capability; and background of successful operation in liquid metal and other turbulent lubrication applications. Furthermore, if preloaded, the tilting pad journal bearing is inherently stable. For this reason a preload coefficient of 0.3 was used in all cases.

Conservative journal bearing designs were achieved by insuring that the eccentricity ratio at design speed and under 1g plus assembly unbalance did not exceed 0.5. For thrust bearings, the minimum film thickness at design speed and under 1g plus aerodynamic forces (in the case of turbines) was at least .001 inches. This provides a good margin against off design conditions, including the 3.93g acceleration plus 3g shock requirement.

For the secondary, or back-up bearing, the shrouded step journal and thrust bearing configuration was selected. The shrouded step journal bearing has one of the highest stability limits of the solid geometry types and both these journal and thrust bearing types have been successfully applied in highly turbulent applications.

5.9 SEALS, LIQUID

A number of liquid seals are required on the shaft of the turbo-alternator. On the potassium lubricated machines liquid seals are required between bearings and the vapor used in the flow path of the turbines. Other seals are required between the bearings and the housing containing the coupling between alternator and turbine and to isolate the alternator rotor cavity from the adjacent bearings. In the organically lubricated machines liquid seals are required between the organically lubricated bearings and the potassium vapor in the turbine. In addition, the bearings must be isolated from the housing carrying the coupling and from the alternator rotor cavity.

The potassium lubricant for the bearings must be pressurized so as to be (about 10 psi) above the pressure of the potassium vapor in adjacent flow path areas and to suppress cavitation. In the partially overhung counterflow machine the potassium vapor pressure in the flow path adjacent to the aft bearing is 38.4 psia. On the opposite side of this bearing is the housing enclosing the coupling which must run at vacuum conditions to assure no running in liquid. This represents the most severe seal pressure difference. In the organically lubricated machines an additional problem is the maintaining of an absolute separation between the potassium vapor working fluid and the liquid organic lubricant. As is the case for the potassium lubricant, the bearings must be pressurized while part of the seal is exposed to vacuum conditions.

Screw Seals

Because of the high rotative speeds (19,200 to 24,000 rpm) and long life (3 to 5 years) contemplated, rubbing face seals are not

practical. As a result, sealing must be brought about by dynamic action. The screw seal has been selected for this application because at the same rotative speed it will consume the least power to sustain a given pressure difference of any dynamic seal known. The pressure difference sustained is proportional to the axial length and is a function of rotative speed, seal geometry and the fluid properties (density and viscosity). Shown in Figure 5.9-1 is the variation of required seal length for a pressure difference of 10 psi as a function of rotative speed for two fluid temperatures, 800 and 1100°F. The results were obtained analytically by making use of the experimental correlation in Reference 5.9-1. The data of Figure 5.9-1 were used to size potassium seals for the various KTA configurations. The data presented in Figure 5.9-2 were obtained in a similar manner and were used to size seals for the organic lubricant.

The liquid seals must contain the bearing lubricant completely during starting of the KTA. Thus, they were sized for full pressure differential at 40 percent of design speed. This is a stringent requirement and makes the seals considerably longer than required for design-point operation. No penalty in power consumption however, is paid for the longer length of the seals because they will only be wetted to the length required to sustain the applied pressure differential. Power consumed is proportional to the wetted length.

Zero Leakage Seal

In those liquid seals in which one side is exposed to vacuum, viz., the housing containing the coupling or the alternator rotor cavity, provisions must be made for minimizing the leakage of both

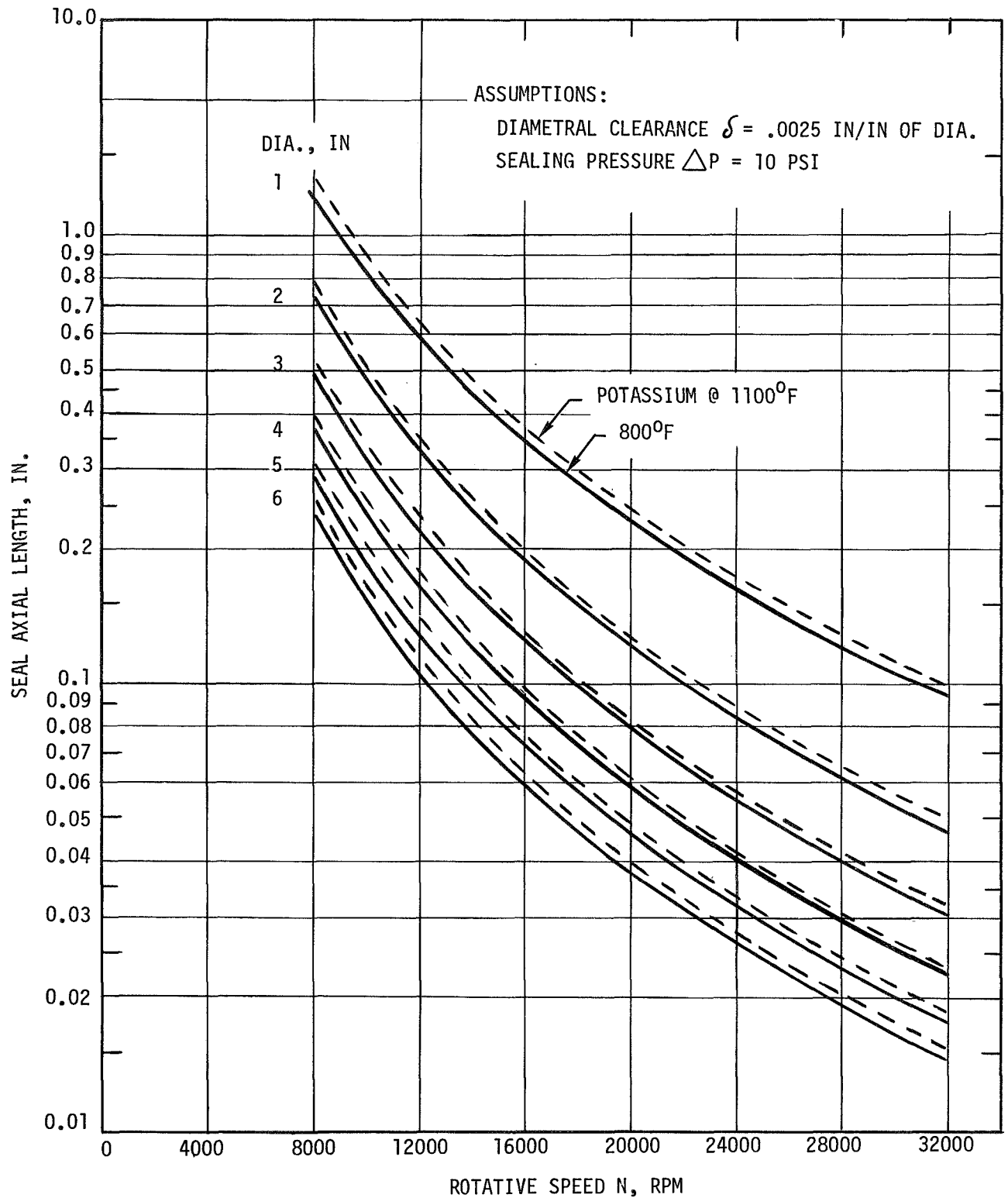


Figure 5.9-1. Screw Seal Length Versus Rotative Speed for Various Seal Diameters.

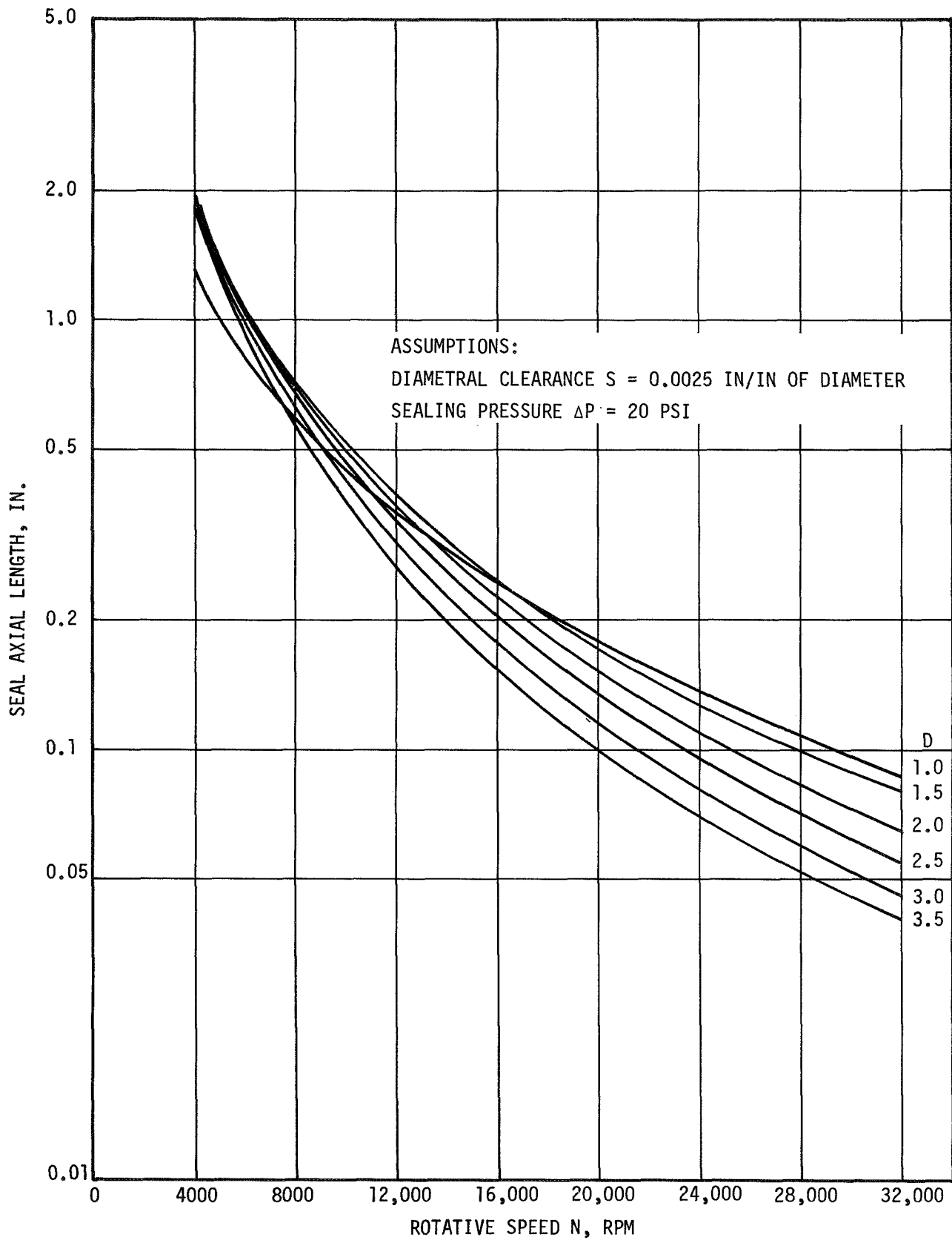


Figure 5.9-2. Screen Seal Length Versus Rotative Speed for Various Seal Diameters.

vapor and liquid. The zero-leakage concept is shown in Figure 5.9-3. The pressure differential across the seal is maintained through the use of a screw seal (pumping to the left) as shown. At some moderately high speed the interface between liquid and vapor breaks down, permitting the intermittent leakage of liquid from the seal. The centrifugal separator prevents this because liquid and vapor escaping to the right are turned to the left and come in contact with the inner diameter of the centrifugal separator. This liquid is returned to the screw seal. The molecular pump causes an added resistance to the escape of vapor by pumping to the left. The molecular pump is similar to the screw seal. However, since it operates in the transition region between continuum and molecular flow, the thread is made much coarser and several thread flights are used. Added control of the vapor escaping the zero-leakage seal can be obtained by cooling the molecular pump to the lowest possible temperature, reducing the vapor pressure of the gas to a small fraction of one psia.

Spiral Groove Face Seal

In order to extend the zero leakage feature of the screw seal molecular pump combination down to zero rpm, lift-off face seals, such as those used in the SNAP-8 program, have been suggested for use in the KTA. Since the added complexity is great and the reliability low, the next best sealing method makes use of permanently engaged bellows face seals as shown in Figure 5.9-4. The design of these seals presents formidable problems because of high liquid temperatures and extremely low leakage-rate allowance. In addition, a high degree of reliability must be maintained for a period of three years of

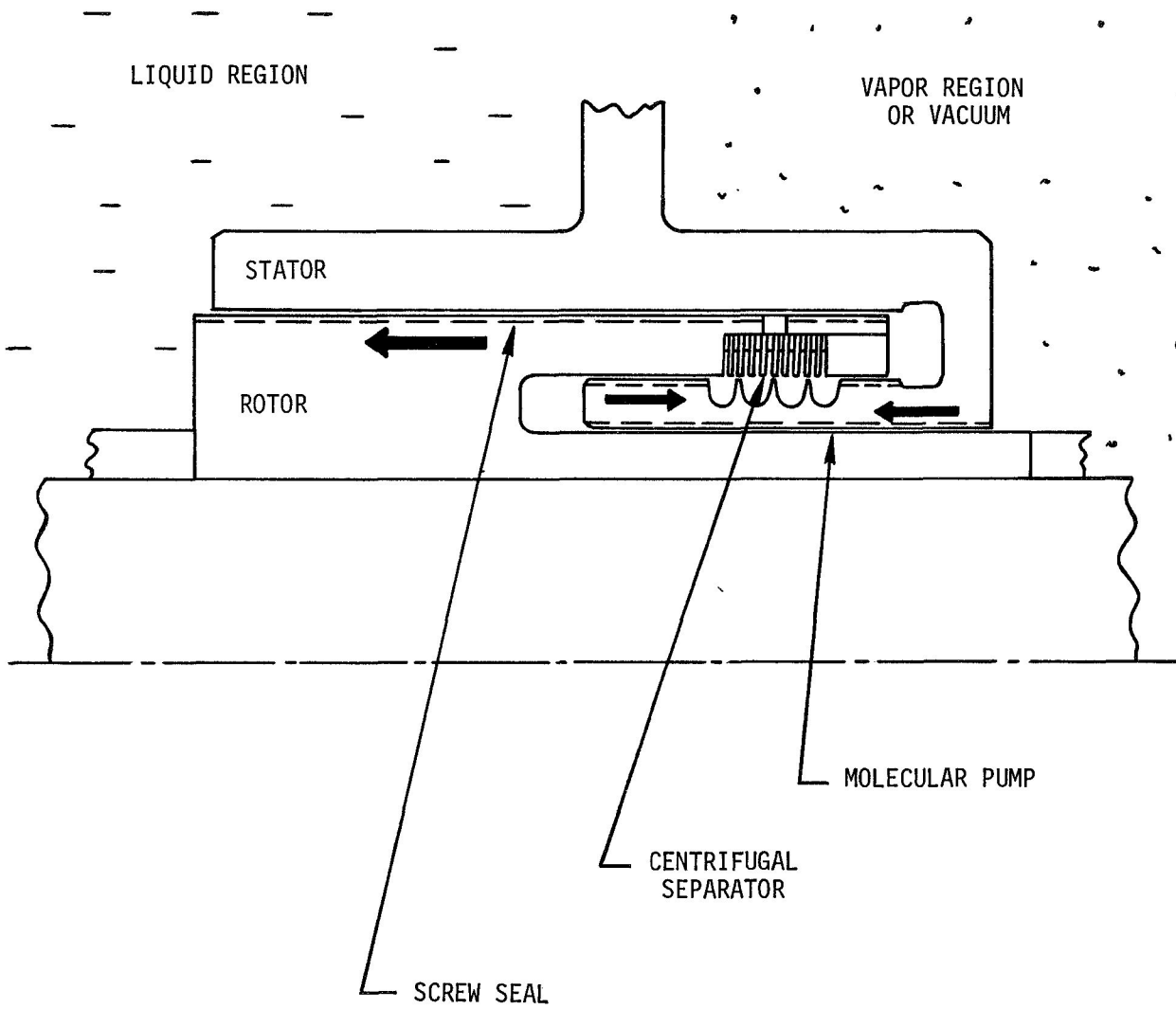
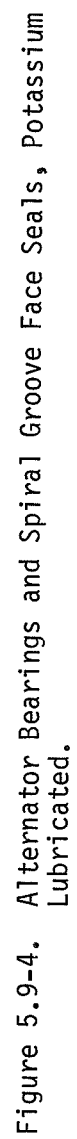


Figure 5.9-3. Zero-Leakage Seal Concept.



unattended operation. For this purpose, the seal must operate with separation between the sliding surfaces. Sliding contact of solid surfaces would only be permissible at startup and shutdown. By incorporating the visco seal principle in the form of a spiral groove into the face seal, sealing surface separation and low-leakage rates can be obtained simultaneously. The sealing-surface separation at operating speeds will ensure long seal life, and the pumping effect of the spiral-groove geometry will provide low leakage rates. Some testing has already been performed in sodium to 1000⁰ F and a sliding velocity of 79 feet per second (NASA TN D-3942).

Most of the potassium leaking past the seal will be slung off the face seal rubbing ring into an involute outlet leading to the injection pump at the alternator anti-drive end. Although the seal has inherent operating advantages, the manufacturing tolerances of the seal components in regard to runout and flatness are very small (in the order of 25-50 micro-inches).

Sealing Between Oil and Potassium

An effective seal between oil and potassium has been demonstrated in the two-stage potassium turbine test of General Electric, Figure 5.9-5. In this test, after a learning period of the initial 100 test hours, failure-free operation of the seal was obtained through 5000 hours of endurance testing. The seal in the two-stage turbine is composed of a dynamic seal used as the pressure dam and an argon buffer seal incorporated for positive separation of fluids. The pressure differential capability of the dynamic seal was 28 psi at 50 percent speed. This pressure differential was required during

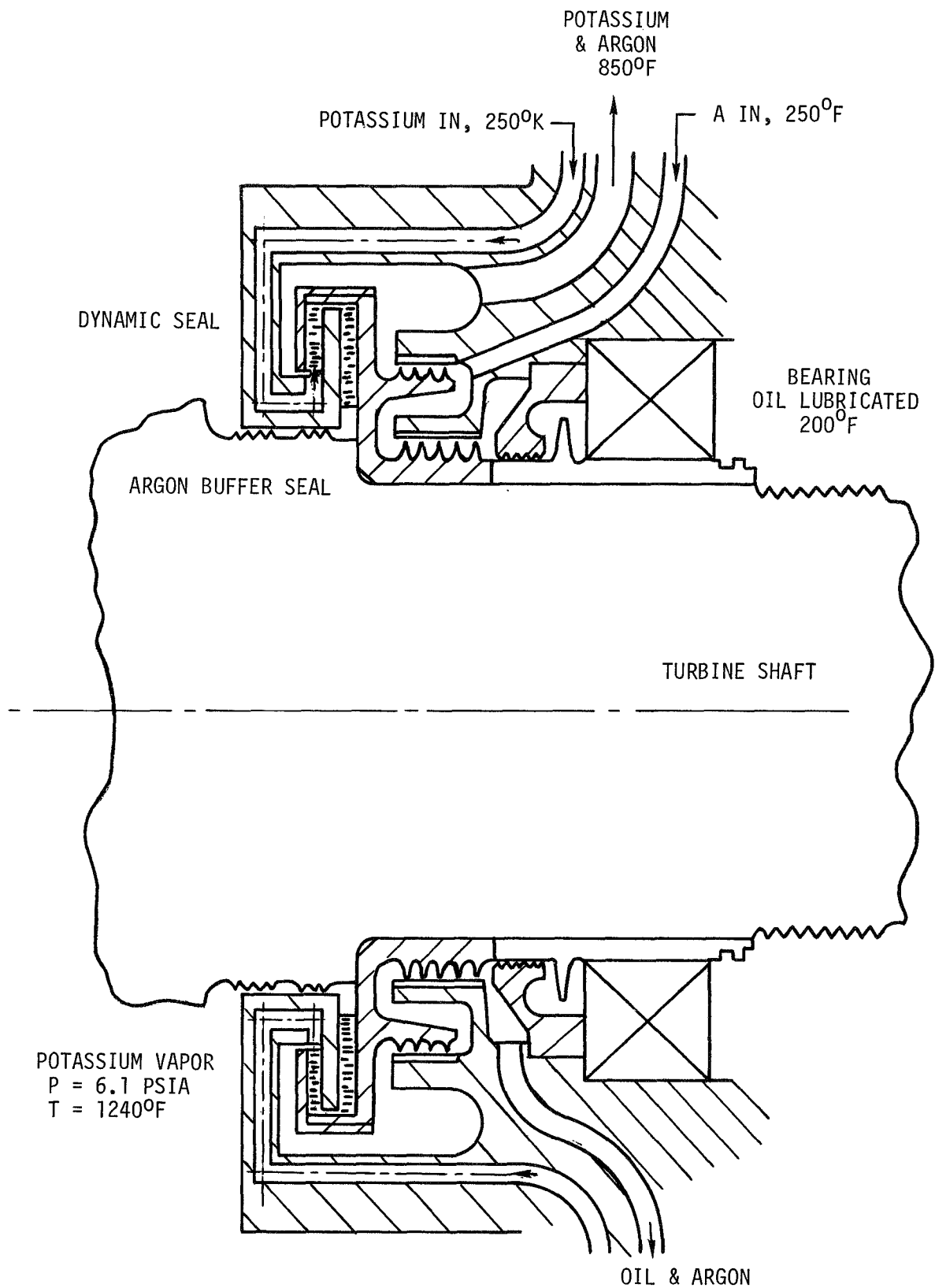


Figure 5.9-5. Potassium to Oil Seal of 2 Stage Potassium Test Turbine.

startup, in evacuating the potassium loop before initiation of boiling. The pressure differential capability of the buffer seal is only in the order of ± 2 psi. As long as argon flows to both sides, oil as well as potassium, a mixing of fluids is prevented. By plugging any outlet lines for argon-oil and argon-potassium mixtures, the buffer seal can be made ineffective. It is concluded that the buffer seal can only be used in conjunction with a dynamic seal providing the pressure dam.

The buffer seal creates a mixture of argon and oil on one side and a mixture of argon and potassium on the other side. The General Electric turbine facility is provided with means to reclaim the argon from both mixtures by separating out oils as well as potassium. Cooling, gravity and micro-filters are incorporated for this purpose. The filters are arranged in duplex and do require cleaning maintenance during operation, such as once in 1000 hours. The separation of oil and argon is considerably more difficult than the separation of potassium and argon. In fact, after some time of operation it was concluded that residual oil in the reclaimed argon is a source of loop contamination. Therefore, reclamation of the oil infested argon was abandoned.

The argon reclamation system constitutes a major part of the test facility consisting of air cooled heat exchangers, separating tanks, filters, the associated piping and a large size argon compressor. Argon reclamation in space without gravity and without low temperature coolants is very difficult. To dispose of the argon used in buffer seals becomes impractical when considering three to five years operating time and the number of seals involved, even if these flows were minimized. Employment of argon buffer seals implies provisions

for extraction of argon from the loop. In the General Electric turbine test facility these provisions were available. In fact, argon was extracted continuously out of the two-stage turbine facility condenser in conjunction with large amounts of potassium vapor. In any liquid-metal test facility exposed to the atmosphere, the use of argon as a cover gas is advantageous. In a refractory metal loop, however, there is no incentive and no reason to use argon within the loop. Inert gas is usually used in the dump tanks of these loops to control the liquid-metal level within the loop. In a space system any interface between inert gas and liquid metal has to be avoided, because otherwise, inert gas by solution may get into the loop and collect in the condenser area fouling the condenser.

Vacuum Seal

For all these reasons, the incorporation of argon buffer seals in a space power system has to be abandoned, leaving the use of vacuum between oil and potassium seals as the only safeguard against mixing. A seal system of this kind is pictured in Figure 5.9-6. On the potassium as well as the oil side, zero leakage seals are used which prevent the leakage of liquid but will not prevent the leakage of vapor at a vapor pressure corresponding to the temperature level maintained within the seal. In potassium, 600^oF corresponds to 0.6 mm Hg vapor pressure. To obtain the same low vapor pressure on the organic lubricant side, a temperature level of 200 to 400^oF will have to be maintained in the seal depending on the oil used. The potassium vapor and the oil vapor cavities are separated by a minimum clearance seal. The effectiveness of this seal may be improved by helical

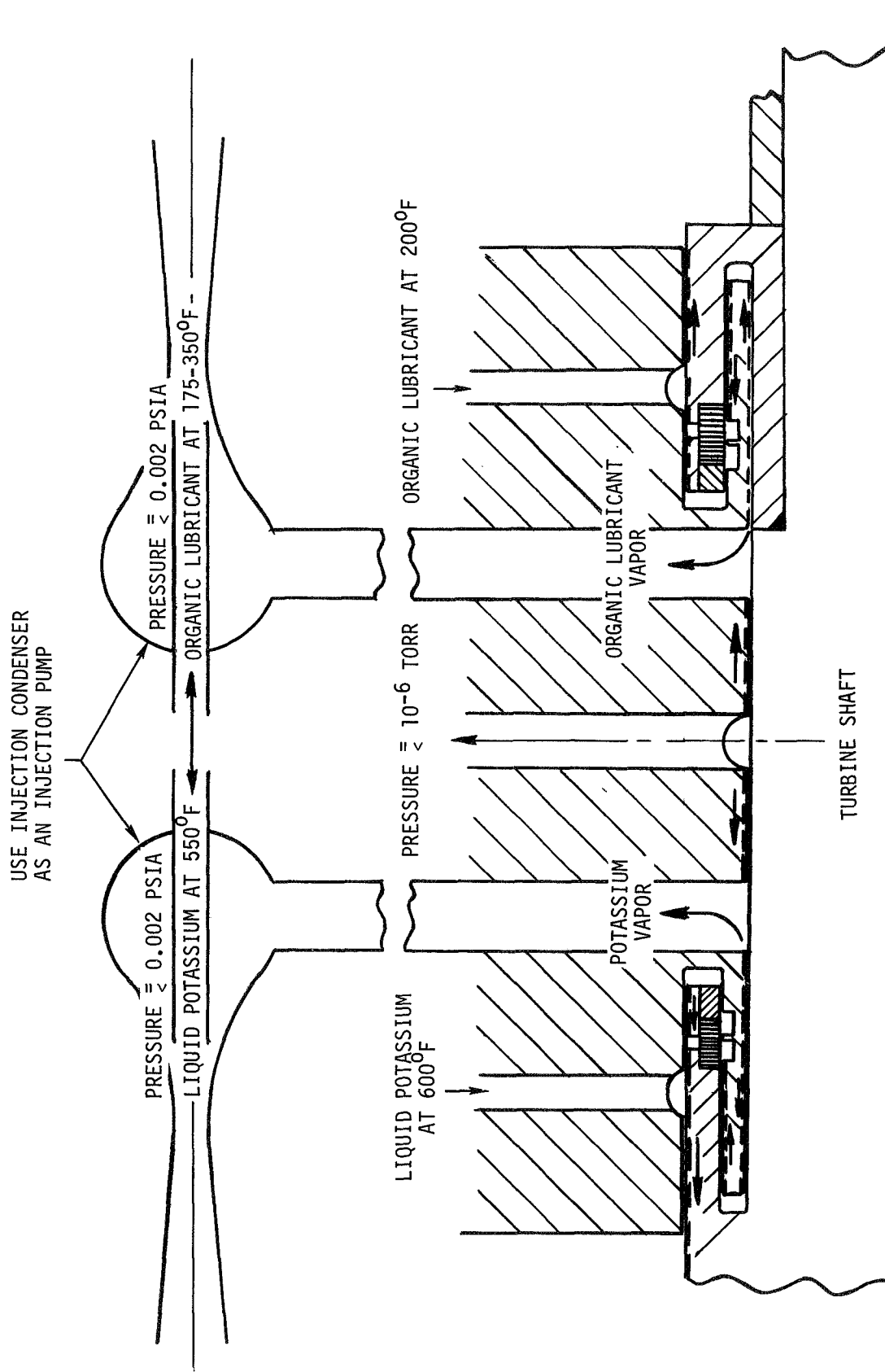


Figure 5.9-6. Vacuum Seal Between Organically Lubricated Bearings and Potassium Turbine.

grooves acting as molecular pumps. This seal will minimize the exchange of vapor from one cavity to the other to a level low enough not to create a problem. The seal, however, will not provide hermetic sealing under any off-design operating condition. To prevent any transport of vapors from the organic lubricant loop to the potassium loop and vice versa, it is possible to subdivide this molecular seal and bleed the center portion to space (or to a vacuum chamber). The two vapor cavities are scavenged by injection pumps using cooled liquids. For the potassium injection pump, 550°F appears practical. On the organic lubricant side, 175 to 350°F, depending on the oil used, are reasonable values. In principle, a seal system has been established which has a good chance of successfully keeping the two fluids separate. The big mechanical design task remains to reduce this principle to acceptable, practical dimensions. In the turbine two such seals are required for two organically lubricated bearings. It is obvious from the sketch that this kind of seal has a profound effect on bearing span and consequently on vibrational behavior of the rotor. It is questionable whether a practical design can be arrived at.

5.10 BEARING AND SEAL DESIGNS

The design considerations for turbine and alternator bearing and seal designs are different; also the design considerations for bearings and seals running in potassium are different from those for running in an organic lubricant. Separate designs were made for each application.

Alternator Bearing and Seal Designs for Potassium Lubricant

Shown in Figure 5.10-1 is a design of the drive-end alternator bearings and seals for operation in potassium. Included in the design are a four pad, pivoted-pad journal bearing, a double-acting, pivoted-pad thrust bearing with leveling links and two zero-leakage seals. The pivoted-pad bearing has a ball and socket pivot configuration. The pivoted-pad thrust bearing has eight pads. The zero-leakage seal on the left restricts the flow of potassium vapor into the housing for the turbine-alternator coupling and the one on the right maintains a low pressure (0.004 psia) in the alternator rotor cavity.

The alternator shaft is made from H-11 tool steel and the housing from Hastelloy C. The bearing journal and pads are made from sintered titanium carbide with a 10 percent Columbium binder while the bearing-pivot support ring is made of TZM molybdenum alloy. The bearing lubricant (liquid potassium) is supplied at 800°F at which temperature the H-11 shaft has a thermal coefficient of expansion of 7.2×10^{-6} in/in°F while the titanium carbide journal and pads have a value of 3.7×10^{-6} in/in°F. Compensation for these widely differing thermal growth rates is obtained by mounting the journal to the shaft by precision-ground conical joints having their apexes on the shaft centerline at the bearing center. A similar problem and the same

19,200 RPM

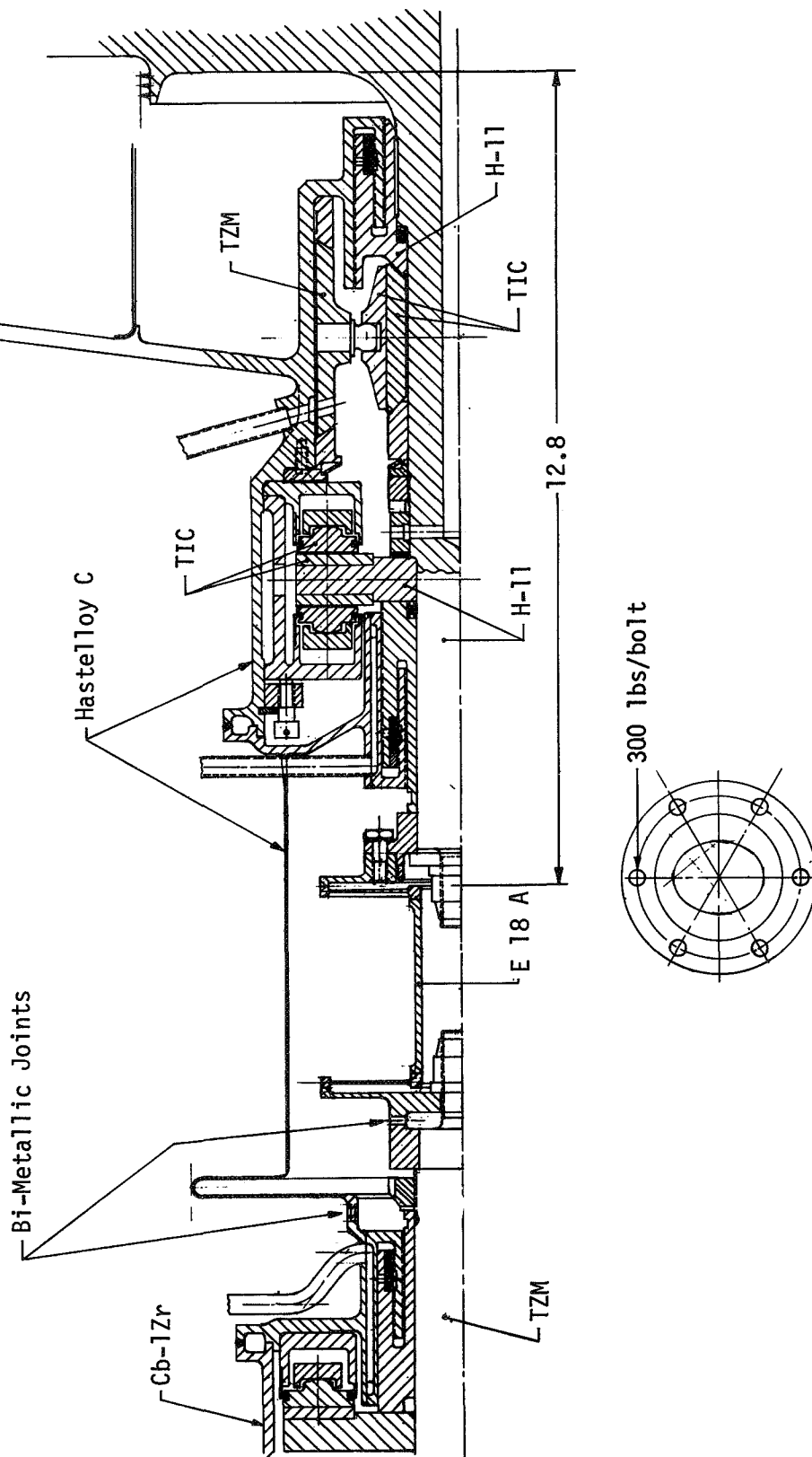


Figure 5.10-1. Alternator Bearings and Zero - Leakage Seals, Potassium Lubricated.

solution is associated with the pivoted-support ring and the adjacent housing. Spring loading is used to keep the conical joints in intimate contact. In order to maintain precisely the position of the pad pivots, the pivot-support ring is monolithic. Like the journal bearing, both of the rubbing surfaces of the thrust bearings, stator and rotor, are Columbium-bound titanium carbide.

Shown in Figures 5.10-2 and 5.10-3 are alternate designs showing an interchange of thrust-bearing and journal-bearing location. In addition, zero leakage seals are shown with involute outlets vented to the injection pump instead of liquid traps using centrifugal separators.

Turbine Bearing and Seal Design for Potassium Lubrication

Shown in Figure 5.10-4 is a design of the aft turbine bearing. From left to right, the design incorporates a liquid screw seal; a four-pad, pivoted-pad bearing; a shrouded-step thrust bearing with a shrouded-step bumper bearing and a zero-leakage seal. In the design shown, a stud protruding from the up side of the pad acts as the pivot. A cage around the stud positions the pad. For ease of assembly, a pad retainer assembly loosely holds the pad in position. X-bar seals prevent leakage of potassium vapor from the turbine into the coupling and potassium liquid from the bearing housing into the coupling housing. In order to minimize length, the thrust bearing is located radially adjacent to the zero leakage seal. The potassium lubricant is introduced near the pivot at 800^oF. Because the vapor in the lower-pressure adjacent turbine inlet scroll is at about 1600^oF, radiation shielding is employed around the bearing housing.

19,200 RPM

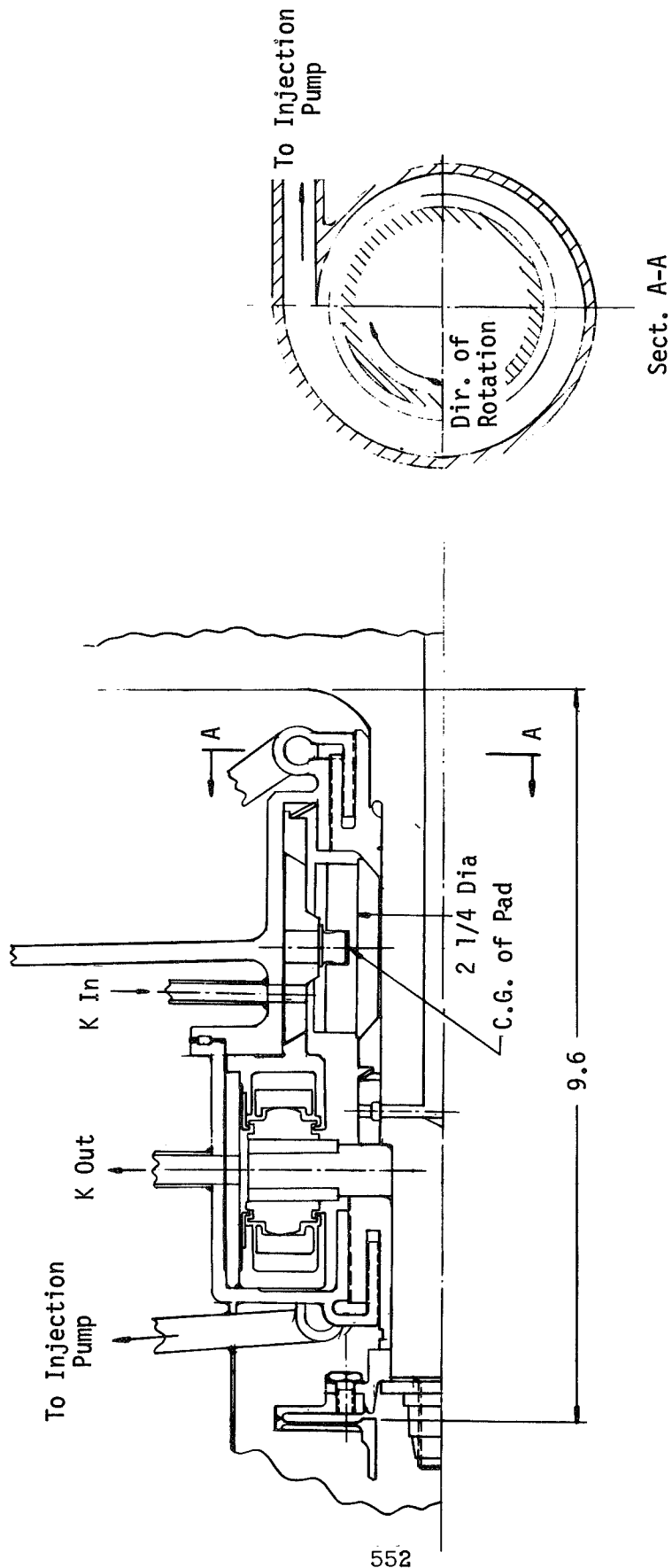


Figure 5.10-2. Alternator Bearings and Screw Seals (Involute Outlets), Potassium Lubricated, Thrust Bearing Outboard.

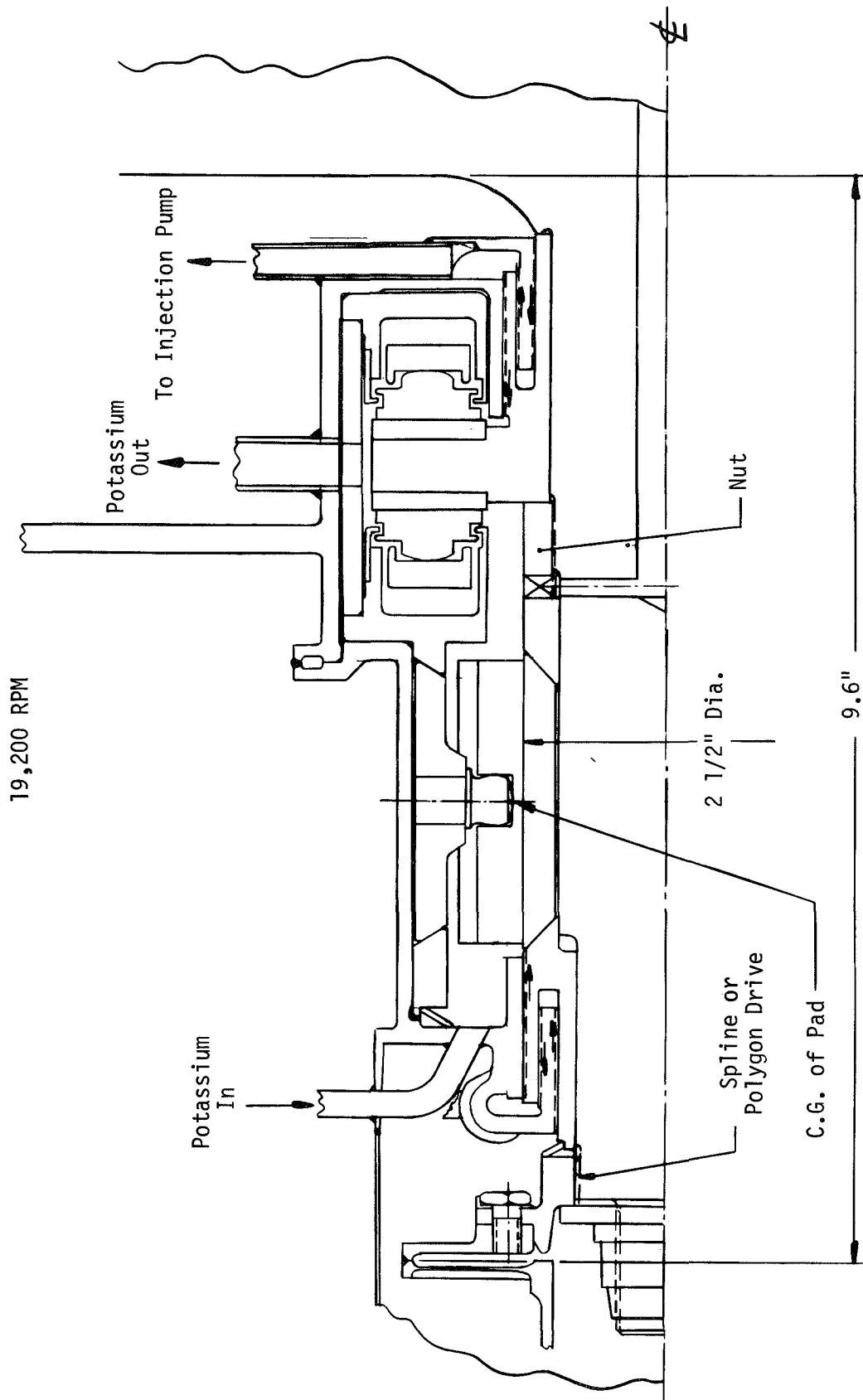


Figure 5.10-3. Alternator Bearings and Screw Seals (Involute Outlets), Potassium Lubricated, Thrust Bearing Inboard.

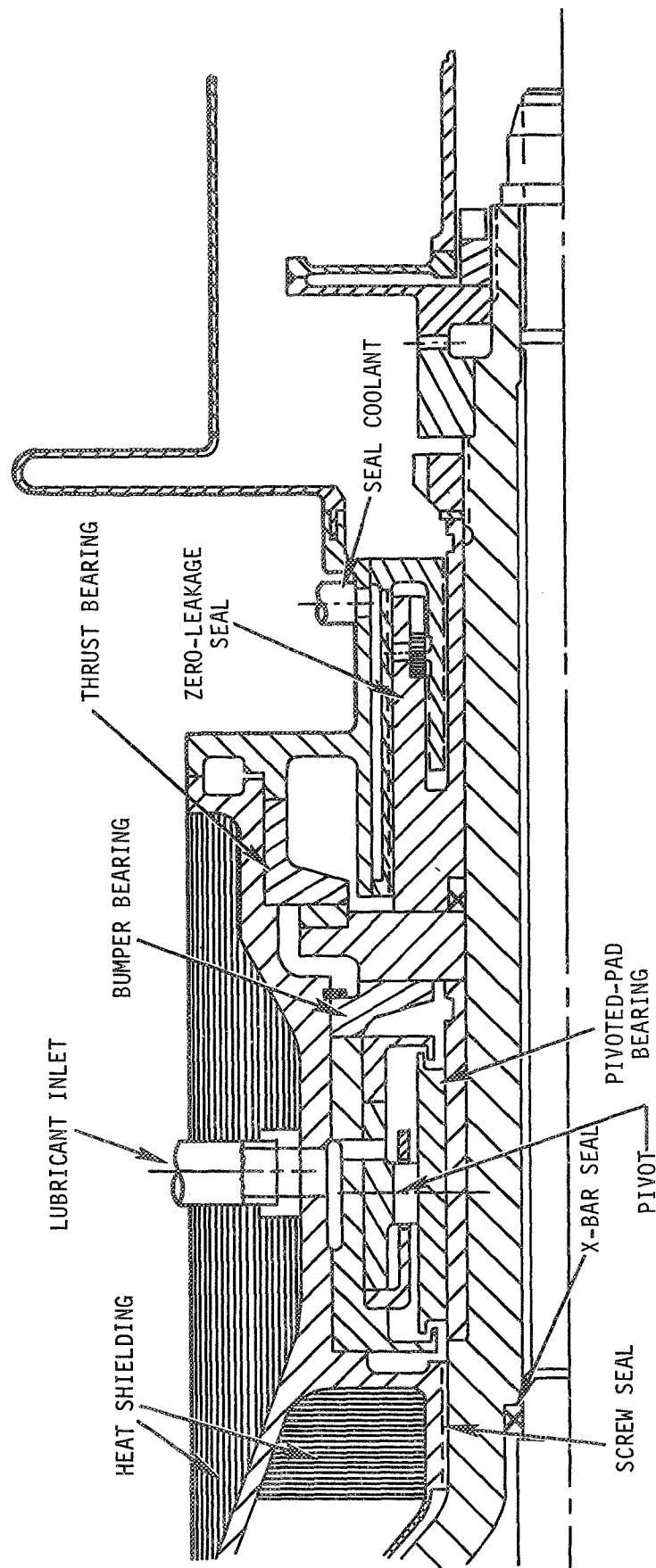


Figure 5.10-4. Turbine Bearing and Seal Design for Potassium Lubrication.

The turbine shaft and bearing housing are made of TZM molybdenum and Columbium + one percent zirconium alloys, respectively. All of the rubbing surfaces of both the journal and thrust bearing are made of Columbium-bound titanium carbide. Fortunately at 800°F the average coefficient of thermal expansion for the titanium carbide (3.7×10^{-6} in/in.°F) is approximately the same as for the TZM molybdenum (2.8×10^{-6} in/in.°F) and Columbium + one percent zirconium (4.3×10^{-6} in/in.°F) alloys. As a result the bearing clearance can be held within required tolerances without having to resort to the conical joints used in the alternator.

Turbine Bearing and Seal Design for Organic Lubricant

Shown in Figure 5.10-5 is the aft bearing and seal design for the turbine using the organic lubricant. From left to right the design comprises a potassium zero-leakage seal, molecular pumps for potassium and organic-lubricant vapors, a vent to vacuum (space), a zero-leakage seal for organic lubricant, a four-pad, pivoted-pad bearing, a shrouded step thrust bearing with shrouded-step bumper bearing and another zero-leakage seal to minimize the amount of organic-lubricant vapor which escapes into the coupling housing. The shaft and bearing housing are made from TZM molybdenum and Columbium + one percent zirconium alloys, respectively. The rubbing parts of both journal and thrust bearing are made from titanium carbide. Because the organic lubricant is supplied at 400°F and the coefficients of thermal expansion for the rubbing parts, the shaft and housing differ little, no significant problem of differential thermal expansion exists, simplifying the design.

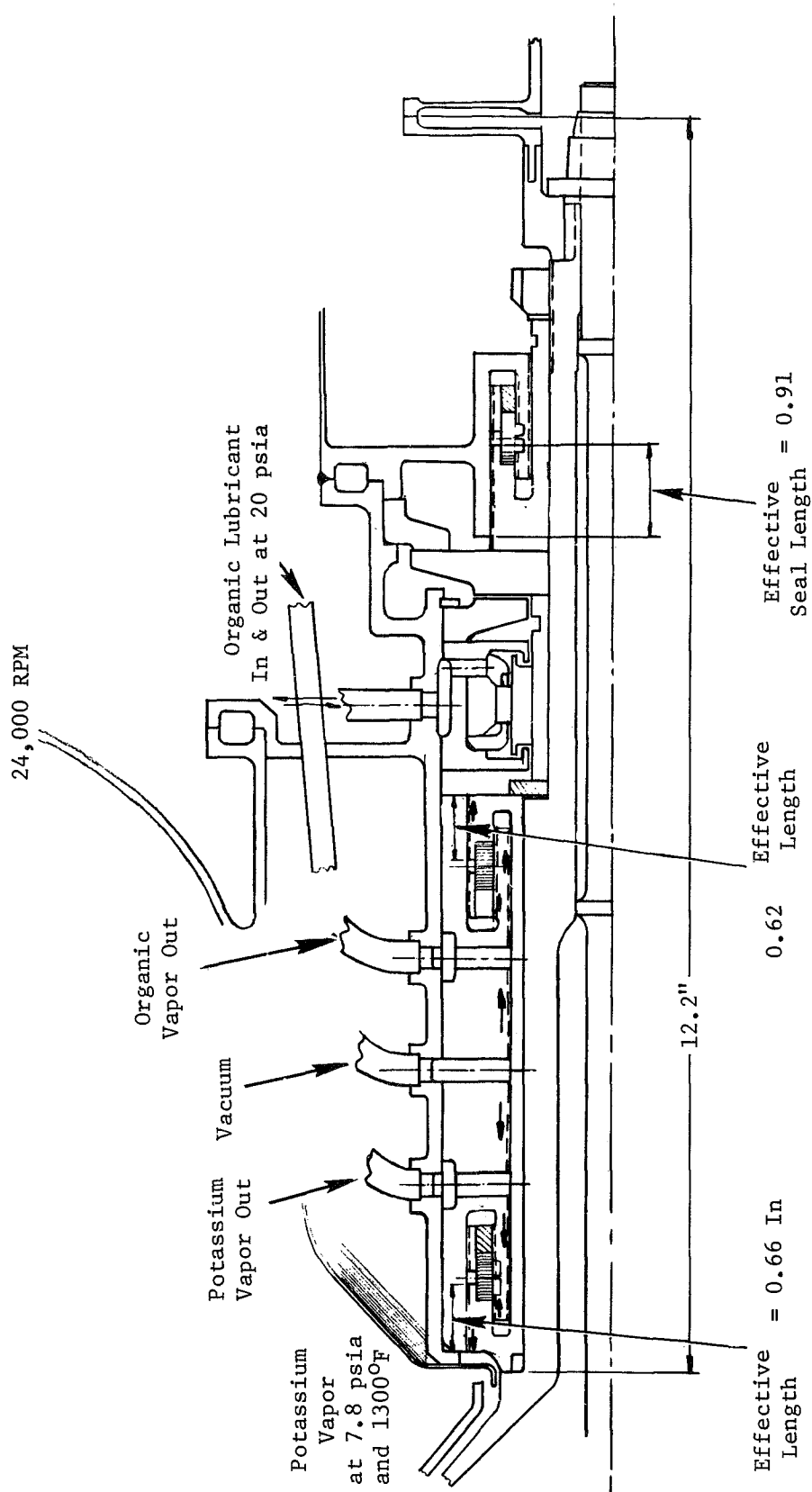


Figure 5.10-5. Turbine Aft Bearings and Zero-Leakage-to-Vacuum Seals, Organically Lubricated.

Alternator Bearing and Seal Design for Organic Lubricant

Shown in Figure 5.10-6 is the design of the bearings and seals for the drive-end of the alternator on organically lubricated bearings. The double-acting pivoted-pad thrust bearing is outboard of the pivoted-pad journal bearings. On either side of the bearings, there are zero-leakage seals. The one on the left minimizes lubricant vapor in the coupling housing and the one on the right performs the same function for the alternator rotor cavity. The journal bearing pads have a ball and socket pivot configuration and are configured so that the center of gravity of the pad is at the pivot point, reducing pad inertia effects during high speed operation. The thrust bearing has leveling links, permitting perfect alignment between pad and runner at all times.

The materials to be used in the organic lubricated alternator bearing and seal design permit a drastic reduction, if not elimination, of differential thermal expansion problems typical for potassium lubrication. Hardened steel, such as H-11, may be chosen for bearing and seal sleeves. Stellite 6B may be an alternate choice. The bearing pads may be silverplated, while hard inserts of Carboloy would be used at the pivot points. Hastelloy-C may be used for stator rings and casing.

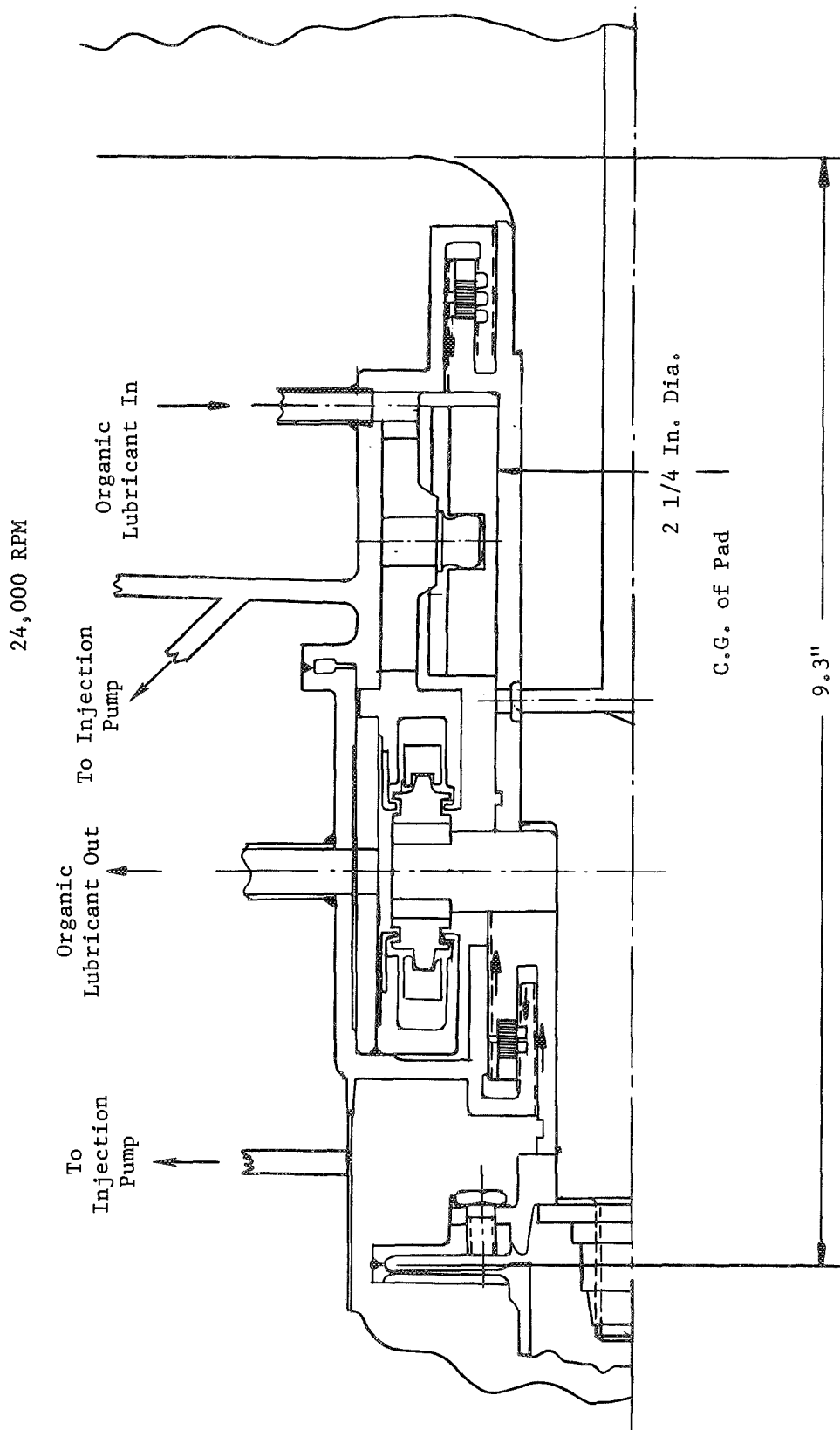


Figure 5.10-6. Alternator Bearings and Zero-Leakage Seals, Organically Lubricated.

5.11 TURBINE-ALTERNATOR COUPLING

Objectives

The coupling connecting the turbine rotor with the alternator rotor must be capable of transmitting 150 percent of the required torque at the design speed for three years, with an acceptable margin of safety. It must be capable of operating in the vacuum of space, unattended, at 800°F, and in a radiation environment. In addition, the coupling must be able to sustain a combination of axial, parallel offset and angular misalignment between turbine and alternator rotors. An axial misalignment of up to ± 0.040 inch could result from differential thermal expansion between the two stators and rotors. Rotor parallel offset in the order of 0.006 inch is expected as a result of manufacturing tolerances and rotor conical whirl.

A total of eight general types of high-speed, high power couplings have been surveyed and compared, resulting in the selection of the contoured-diaphragm coupling because of its unique combination of operating characteristics. These qualify it for space service in preference to other commercially available candidate couplings such as axial spring, laminated disk, universal joint, ball-race, gear, chain, or elastomeric couplings. Other new coupling designs may be postulated, but they would have to be developed, whereas many flexible diaphragm couplings are in service on flight equipment where reliability is the overriding criterion.

The reasons for choosing the flexible diaphragm coupling include:
(5.11-1):

- Ability to operate with up to two degrees of misalignment at

loads of the order of 0.1 HP/rpm and at speeds above 7000 rpm.

- No lubrication required.
- No backlash or relative circumferential movement between components.
- Constant velocity ratio.
- Accommodates axial, angular, and parallel misalignments.
- High temperature capability (above 1200°F, depending on material).
- High torsional spring rate.
- Low bending moment.
- Ability to operate in the space vacuum.
- Capable of up to 60,000 rpm.
- Capable of up to 5 HP/rpm.
- Capable of up to 8 degrees of parallel misalignment.
- Capable of up to 0.10 inch of parallel misalignment.
- Capable of up to 0.20 inch of axial movement.

(The last five statements cannot usually be met simultaneously.)

The use of thin parallel-sided disks, or diaphragms, as flexing elements is not new - the literature shows they were used for locomotive couplings in 1922. Contoured disks have been used as turbine wheels to provide uniform stress distribution. A flexible membrane coupling for the three-stage potassium turbine drive train (NAS3-10606) was designed and manufactured by the Bendix Corporation of Utica, New York. No problems were encountered during its operation.

The contoured profile developed by Bendix is a hyperbolic profile which gives constant torsional shear stress throughout the profile. This is achieved by making the thickness inversely proportional to the square of the radius. For a given allowable material shear stress, this profile achieves the smallest amount of material which must be

flexed, and the thinnest cross-section at any radius. The flexible diaphragm resulting from this profile design has the greatest permissible bending angle consistent with material torsional shear and bending fatigue strengths. It also has a lower bending spring rate, and is lighter weight than any other profile shape.

Although a wide range of diaphragm sizes is allowable, provided special tooling is made, an analysis of the characteristics of some existing geometries was made. This procedure has the practical advantage of reducing a large possible range of parameters to the values which have been found desirable by the manufacturer's experience and testing. If the existing geometries are found to be insufficient, a special geometry could be required. The diaphragm geometries analyzed are listed in Table 5.11-1.

Design

It is assumed that the coupling temperature does not exceed the bearing lubricant temperatures (400°F for oil, 800°F for potassium). E-18V maraging steel is satisfactory to 800°F, and the following properties will be used in subsequent calculations (if satisfactory at 800°F, the diaphragms will also be satisfactory at 400°F).

<u>Material</u>	<u>E-18V</u>	<u>Units</u>
Operating temperature	800	°F
Density	0.290	lb/in. ³
Poisson's ratio	0.3	--
Coefficient of thermal expansion (RT-900F)	5.6x10 ⁻⁶	in./in.°F

TABLE 5.11-1

CHARACTERISTIC PROPERTIES OF SELECTED
FLEXIBLE-DIAPHRAGM COUPLINGS
(E-18V at 800°F)

Parameter	Units	Diaphragm No.					
		1	2	3	4	5	6
Nom. flange OD	in.	4.411	4.412	5.342	5.563	4.751	4.412
Rad. to min. thk.	in.	2.050	2.050	2.254	2.371	2.200	2.050
Rad. to max. thk.	in.	0.820	0.939	1.168	1.250	1.100	0.978
Flange inner radius	in.	0.500	0.750	0.877	1.000	0.846	0.750
Min. thickness	in.	0.0082	0.0092	0.0086	0.0102	0.0110	0.0113
Wt. per diaphragm	lb	0.15	0.15	0.44	0.35	0.17	0.12
Radius ratio	--	0.4000	0.4580	0.5180	0.5250	0.5000	0.4770
Thickness ratio	--	0.0040	0.0045	0.0038	0.0043	0.0050	0.0055
Bending stress rate	$\frac{\text{psi}}{\text{deg}}$	47790	56690	52230	66520	66680	70940
Buckling torque	lb-in.	8260	11750	9400	17610	19910	21445
Buckling factor of safety	--	2.85	4.05	3.24	6.07	6.86	7.39
Shear stress rate	$\frac{\text{psi}}{\text{lb-in.}}$	4.6183	4.1163	3.6425	2.7755	2.9893	3.3514
Axial stress rate	$\frac{\text{psi}}{\text{mil}}$	1730	2040	1690	2040	2220	2550
Bending spring rate	$\frac{\text{lb-in.}}{\text{deg}}$	39	59	53	100	108	111
Axial spring rate	$\frac{\text{lb}}{\text{in.}}$	917	1390	1010	1730	2170	2590
Torsional spring rate	$\frac{\text{lb-in.}}{\text{deg}}$	62980	78440	99180	147330	116520	99180
Effective bending stress	psi	12140	17380	15420	20400	20860	23060
Torsional shear stress	psi	13410	11950	10570	8060	8678	9729
Combined stress	psi	25430	26460	23440	24440	25400	28210
Combined stress F.S.	--	1.97	1.89	2.13	2.05	1.97	1.77
20-g shaft force	lb	9.92	6.87	5.86	5.13	6.07	6.78
20-g add. stress	psi	18710	10160	9810	6050	6210	6680
Add. stress F.S.	--	8.0	14.8	15.3	24.8	24.2	22.5
Max. speed	KRPM	45.3	45.3	37.4	36.0	42.1	45.3
Max. speed F.S.	--	1.42	1.42	1.17	1.13	1.32	1.42

<u>Material</u>	<u>E-18V</u>	<u>Units</u>
Ultimate tensile strength	221,000	psi
0.2% yield strength	209,000	psi
Reduction in area	56	%
Tensile modulus of elasticity	26×10^6	psi
Modulus of rigidity	10×10^6	psi

The design of the diaphragm is based upon four major stress criteria and several secondary criteria (5.11-2). The primary life criteria for a flexible-diaphragm coupling under the influence of both axial and angular misalignments is the magnitude of the effective bending stress which results. If the frequency of application of the axial stress is greater than ten percent of the frequency of application of the bending stress (generally 1/rev), the two are directly additive. The resulting stresses become intolerably high for any significant amount of misalignment, and this criteria simply becomes a caution against operating close to an axial resonant frequency.

When the axial stress application frequency is less than ten percent of the bending stress frequency, effective bending stress ES_b is calculated from the bending stress S_b , the axial stress S_a , and the allowable ultimate tensile strength UTS:

$$ES_b = \frac{S_b}{1 - \frac{S_a}{UTS}} \quad \text{psi}$$

This criteria is analogous to a Goodman diagram, where steady-state and alternating stresses are superimposed. For Bendix's standard material (E-18V) the "continuous-duty" rating of ES_b is 50,000 psi, as shown by Figure 5.11-1. This continuous-duty rating is believed to be conservatively low.

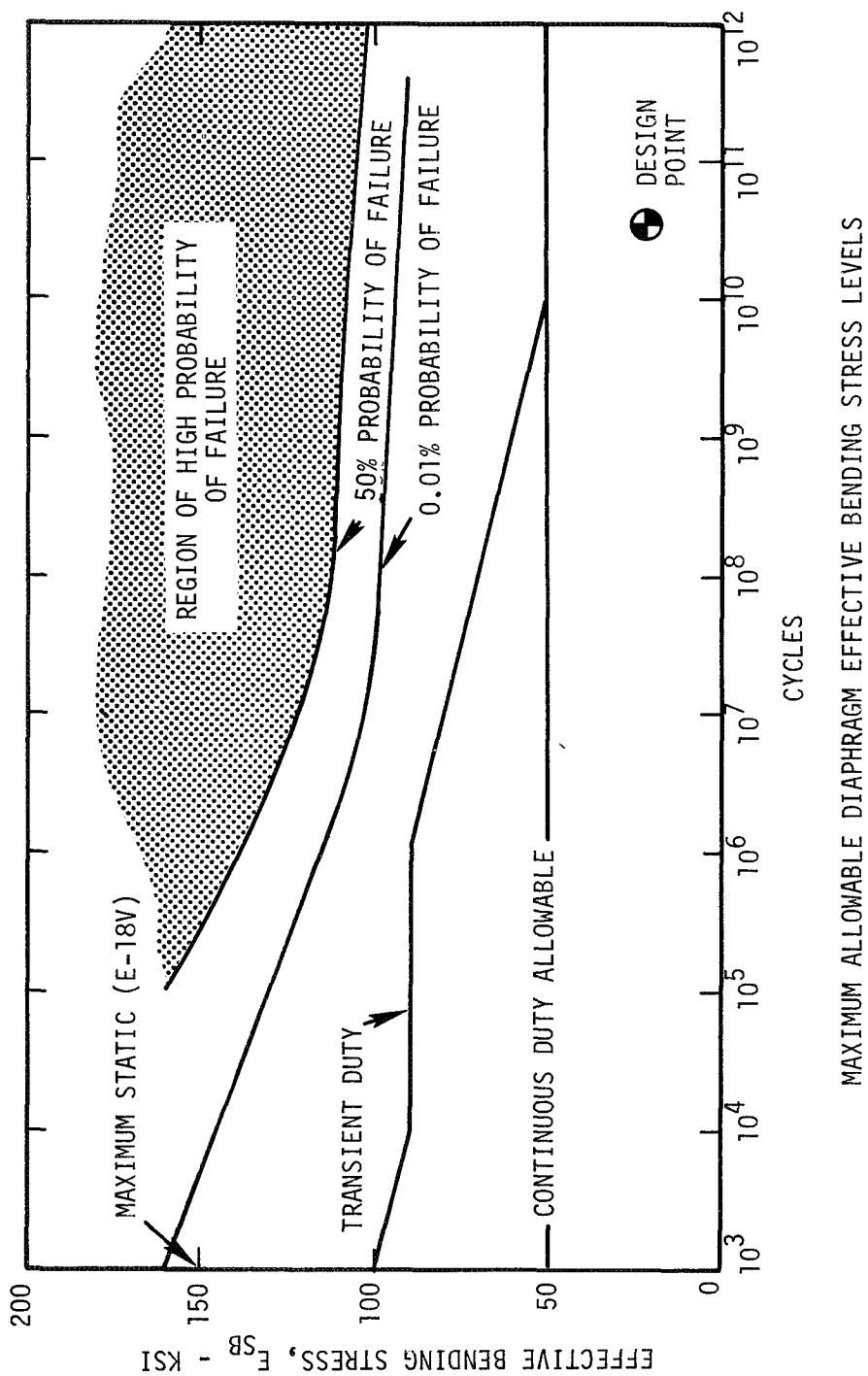


Figure 5.11-1. Normal Allowable Stresses for Flexible Diaphragm Couplings.

To obtain value for ES_b , assume that, at operating conditions, the total axial misalignment (distance between turbine and alternator shafts minus no-load coupling length) is ± 0.040 inch, and the total angular misalignment $2\theta = 0.5$ degrees. These dimensions were used to generate the calculated parameters of Table 5.11-1.

Diaphragm no. 3 is selected as the best compromise between the torque and misalignment requirements for speeds above 19,000 rpm, where 150 percent of the required torque is less than 3500 lb-in. The "rated" torque is only 30 percent of the allowable, and if 150 percent of the actual required torques exceed this value, the use of more than 30 percent of the allowable torque would be acceptable. At 16,000 rpm, where 150 percent of the required torque is 4400 lb-in., diaphragm no. 2 or 4 would be chosen.

Figure 5.11-2 shows the maximum allowable combined misalignments for an effective bending stress of 50,000 psi or an effective combined stress $ES_c = 40,000$ psi for diaphragm no. 3. This effective stress accounts for the effective bending stress ES_b caused by axial and angular misalignments as well as the torsional shear stress S_s caused by transmitting 550 kw at 16,000 rpm (worst expected S_s).

For comparative purposes, similar information is presented for a coupling of the same dimensions made of Rene' 41, the recommended fabrication material for temperatures greater than 1000°F.

The effects of differential thermal expansion through the loop of coupling, turbine shaft, turbine casing, interstage structure, alternator casing, and alternator shaft was studied. With an interstage structure material of a refractory alloy such as Cb-1Zr, the thermal expansions can be absorbed satisfactorily by selectively preloading the

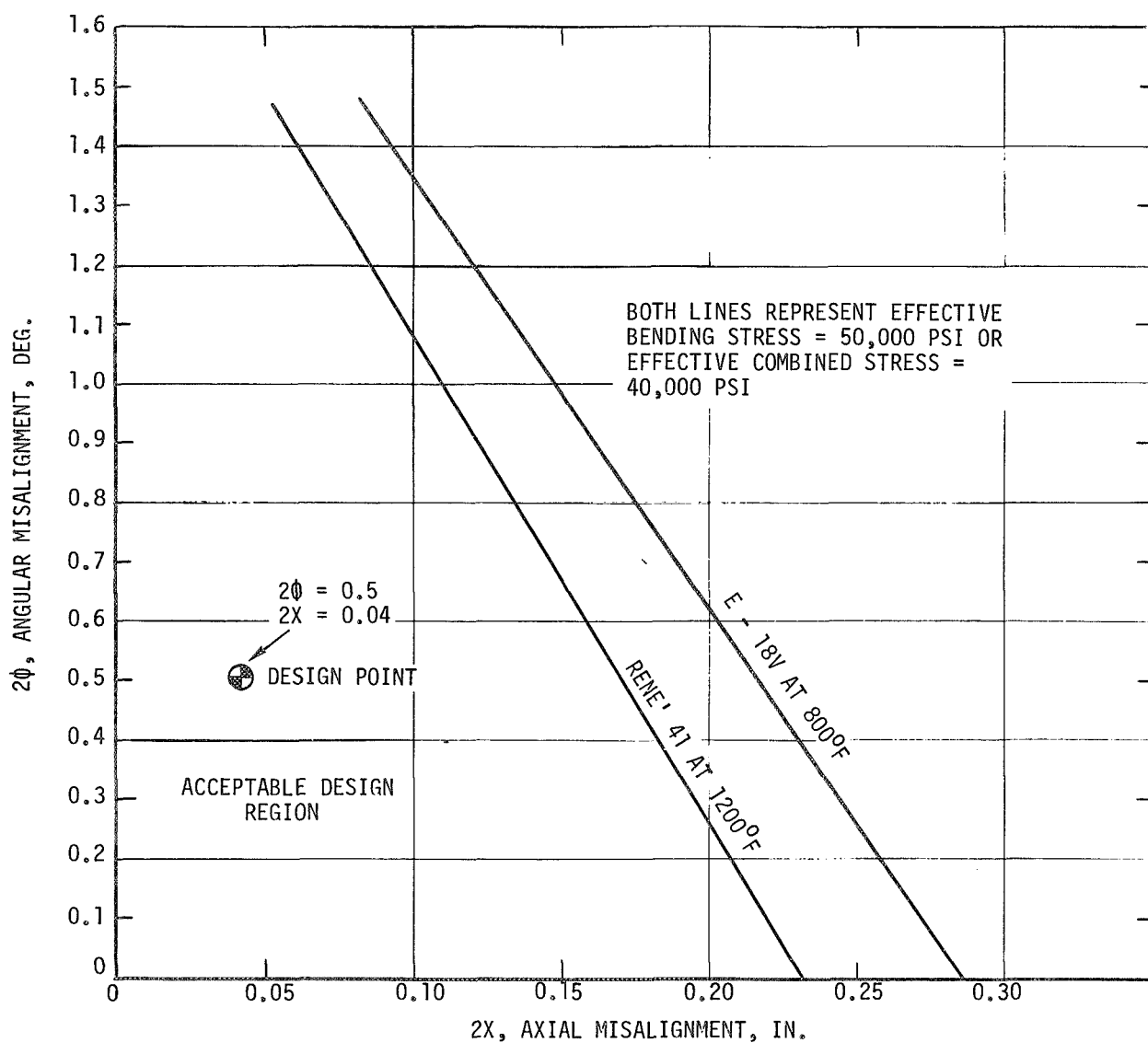


Figure 5.11-2. Allowable Combined Angular and Axial Misalignments for Flexible Diaphragm Coupling.

diaphragm membranes at assembly in the direction which minimizes overall differential thermal expansion at operating temperature.

The coupling design details are illustrated by Figure 5.11-3. The flexible diaphragm coupling is joined to the TZM turbine shaft by a curvic coupling of the "half-barrel" design. This allows differential thermal expansion to occur between the TZM and the E-18V coupling material. The TZM ring which contains the curvic teeth is retained on the turbine shaft with mating elliptical surfaces, in order to avoid keyway stress concentrations in the brittle TZM. Curvic tooth contact between the shaft and flexible diaphragm coupling is assured by a nut on the end of the shaft.

The flexible diaphragm coupling assembly is made by electron beam welding the outer diaphragm flanges to the end flanges and the inner diaphragm flanges to the floating shaft. These welds are made, inspected, and tested at the vendor's plant.

The joint between the coupling and alternator shaft is made by multiple small bolts. This method is required for accessibility. A shim could be incorporated into the bolted joint to preload the diaphragms in the direction required to assure minimum axial misalignment at operating temperatures.

The conclusion to be drawn from the coupling study is that a proved, reliable design exists which incorporates two flexible diaphragms, can safely absorb the stresses due to misalignments between the coupling and alternator, and can operate in the space vacuum.

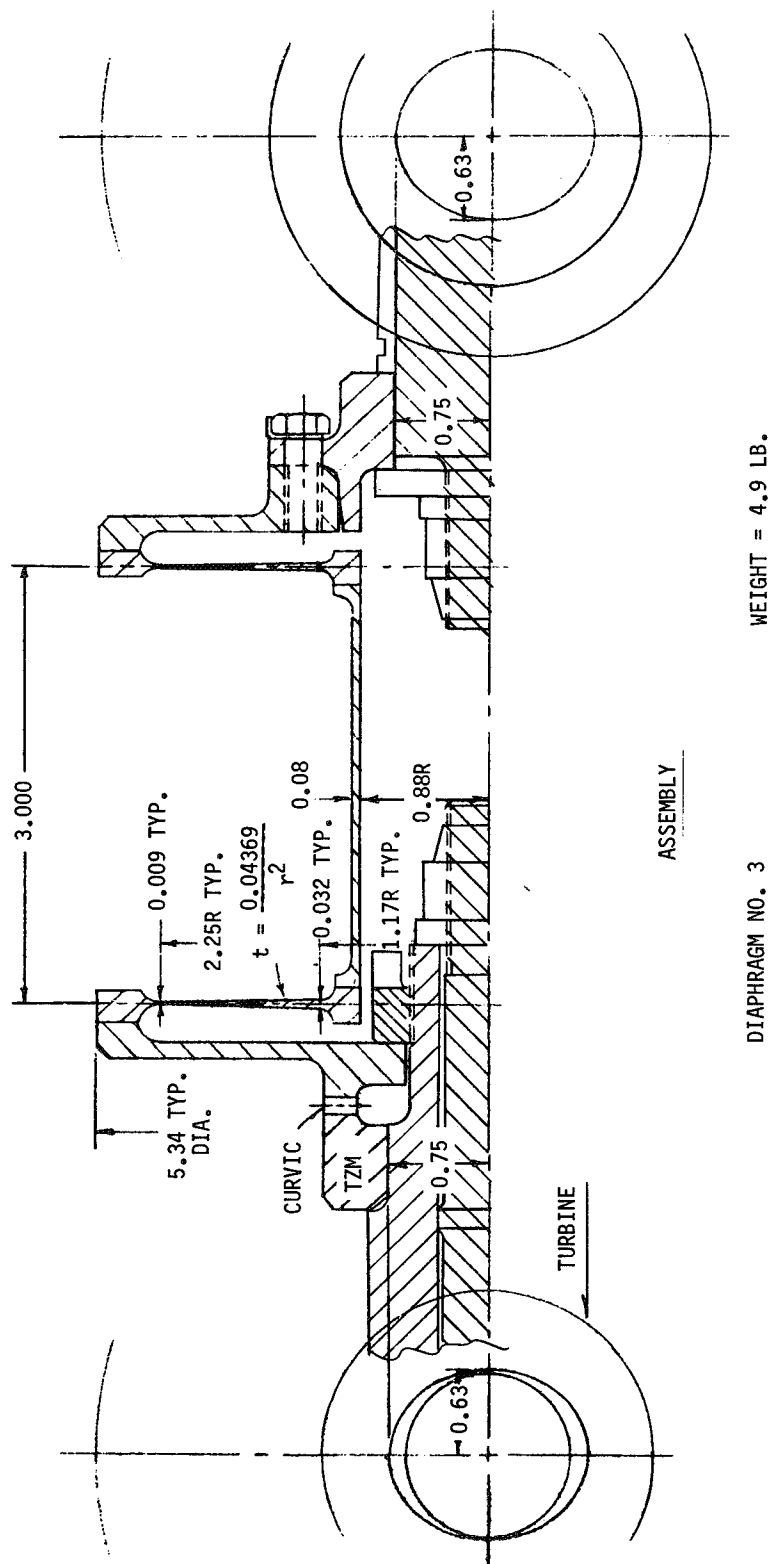


Figure 5.11-3. Flexible Diaphragm Coupling Connecting KTA Turbine to Alternator.

5.12 TURBINE CONFIGURATIONS

Design Studies

A total of 41 different design and study drawings were prepared during the Phase I investigation. These are listed in Table 5.12-1. They cover both component and configuration studies.

Selected Configuration

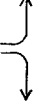
The work statement requests two turbines on potassium bearings and two turbines on oil bearings. In every case, a split spool turbine configuration is requested, which permits interspool condensate extraction. For the second configuration, a single spool turbine is invited. In selecting the turbines for oil lubrication, the main criterion was to have oil bearings and seals not exposed to any high temperature potassium vapors. Consequently, for the single spool turbine, the split flow arrangement, supported on two bearings which are both exposed to the same lowest potassium vapor temperatures at turbine outlet, was chosen (Table 5.12-2 and Figure 5.12-1). Similarly for the split spool turbine, a design which again exposes the bearing and seal areas to low and modest turbine temperatures only, was chosen (Figure 5.12-2). For the potassium lubricated turbine, a nine-stage overhung turbine appears to be the most suitable contender for a single spool arrangement (Figure 5.12-3). It minimizes temperature gradients in the bearing and seal areas and it is outstanding as far as bearing rotor response is concerned. Mechanically, it is the most attractive arrangement. It results in minimum inlet ducting and special freedom in outlet ducting permitting gradual diffusion.

TABLE 5.12-1
KTA DESIGN STUDIES

Dwg.	Iteration No. No. of Stages KRPM	Split Spool	Single Spool	Overhung	Single Flow (Inline)	Split Flow	Counterflow	Lubrication	Welded	Curvic	Bolted	Comments
263E179	0.1020	6 + 4										Casing Study
-180	0.0820		X		X							Casing Study
-182	1.0819		X	X	X			K	X			See -280
-183	1.0819		X	X	X			K		X		See -278
-184	1.0819		X	X	X			K			X	
-185	1.1019		X		X			K		X		Flowpath Only
-186	2.1019		X			X		K	X			
-187	1.1019		X		X			K	X			
-188	4.091	6 + 3					X	K	X	3		
-189	1.1019		X		X			K			X	
-191	3.0632		X		X					X		Wheels Only
-192	3.0824		X		X			Oil		X		Wheels Only
-193	3.1024		X		X			Oil		X		Wheels Only
-194	3.1024		X		X			Oil	X			Wheels Only
-195	4.0919	6 + 3					X	K		X		
-196	3.1216		X		X					X		Wheels Only
-197	2.1019		X			X		K			X	Wheels Only
-300	19							K				Alt. Rotor
-301	5.0919		X	X	X			K			X	
47R199268	0.1020		X		X							Casing Study
-269	0.1020		X			X			X			
-270	0.0820	6 + 2					X		X	X		
-271	3.0919	5 + 4		5			X	K	X	4		See -281
-272	4.0919	6 + 3					X					Separator Study
-274	4.0824		X			X		Oil			X	See -284
-275	5.0824	5 + 3					X	Oil		X		See -286
-276	19							K				Alternator
-277	24							Oil				Alternator
-278	1.0819		X	X	X			K		X		
-279	3.0824		X	X	X			Oil			X	
-280	1.0819		X	X	X			K	X			
-281	3.0919	5 + 4		5			X	K	X	4		
-284	4.0824		X			X		Oil			X	
-285	0.0919		X	X	X			K	X			2-Brg Arrangement
-286	5.0824	5 + 3				X		Oil		X		
-287	1.0919		X	X	X			K	X			Exhaust Scroll
-288												
-289	1.0919		X	X	X			K	X			
-290	3.1019	6 + 4		6			X	K	X	4		
-291												Exhaust Scroll

TABLE 5.12-2

TURBOALTERNATOR ARRANGEMENTS

Turbine Speed and Type of Lubricant	Drawing Number	Rotor Type	Flow Type	No. of Wheels	Wheel Conn.	Wheel Material	Remarks	Alternator	KTA Drawing Number
19,200 rpm K Bearings	263E287	Single Spool Overhung	Single	9	Welded	ASTAR and FS-85	All stages over- hung. Condensate extraction from stator optional.	Drawing No. 47R199276 733 KVA 1600 HZ, 480 V 800°F K Cooled	47R199289
	47R199290	Split Spool	Counter 	10	6 Welded Overhung + 4 Curvic Couplings	ASTAR, FS-85 and TZM	Centrifugal separators. Bearing in middle. First 6 stages overhung.		47R199290
24,000 rpm Oil Bearings	47R199274	Single Spool	Split 	2x8=16	Bolted	TZC and TZM	Condensate extraction from stator optional.	Drawing No. 47R199277 733 KVA 1600 HZ, 480 V 400°F Oil Cooled	47R199284
	47R199275	Split Spool	Counter 	8	5 + 3 Curvic Coupling	TZC and TZM	Centrifugal separators. Labyrinth seal in middle. Bearings and seals both ends.		47R199286

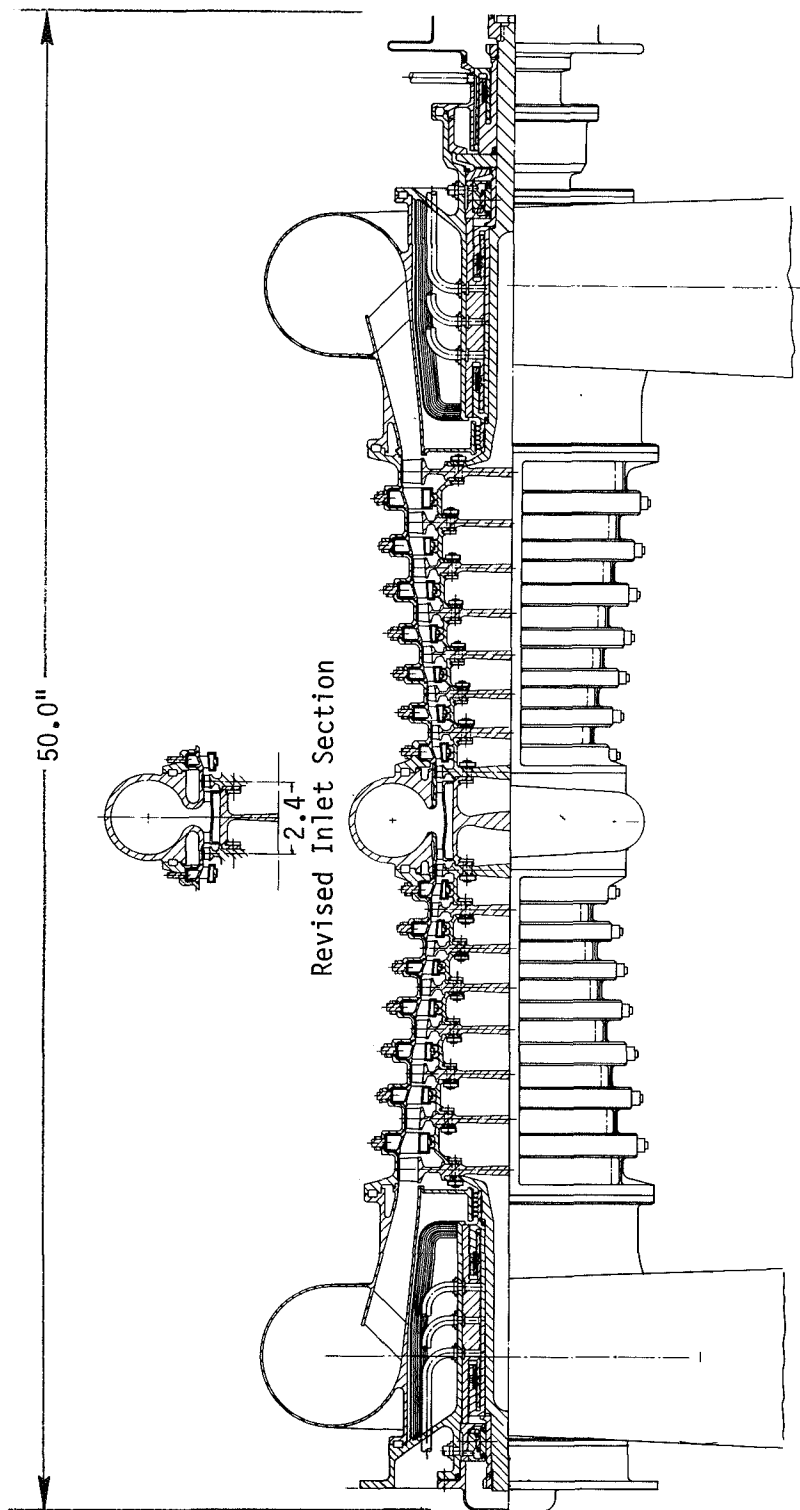


Figure 5.12-1. 8 Stage, 24,000 RPM, Single Spool, Split Flow Straddle-Mounted, on Oil Bearings.
(Dwg. No. 47R199274)

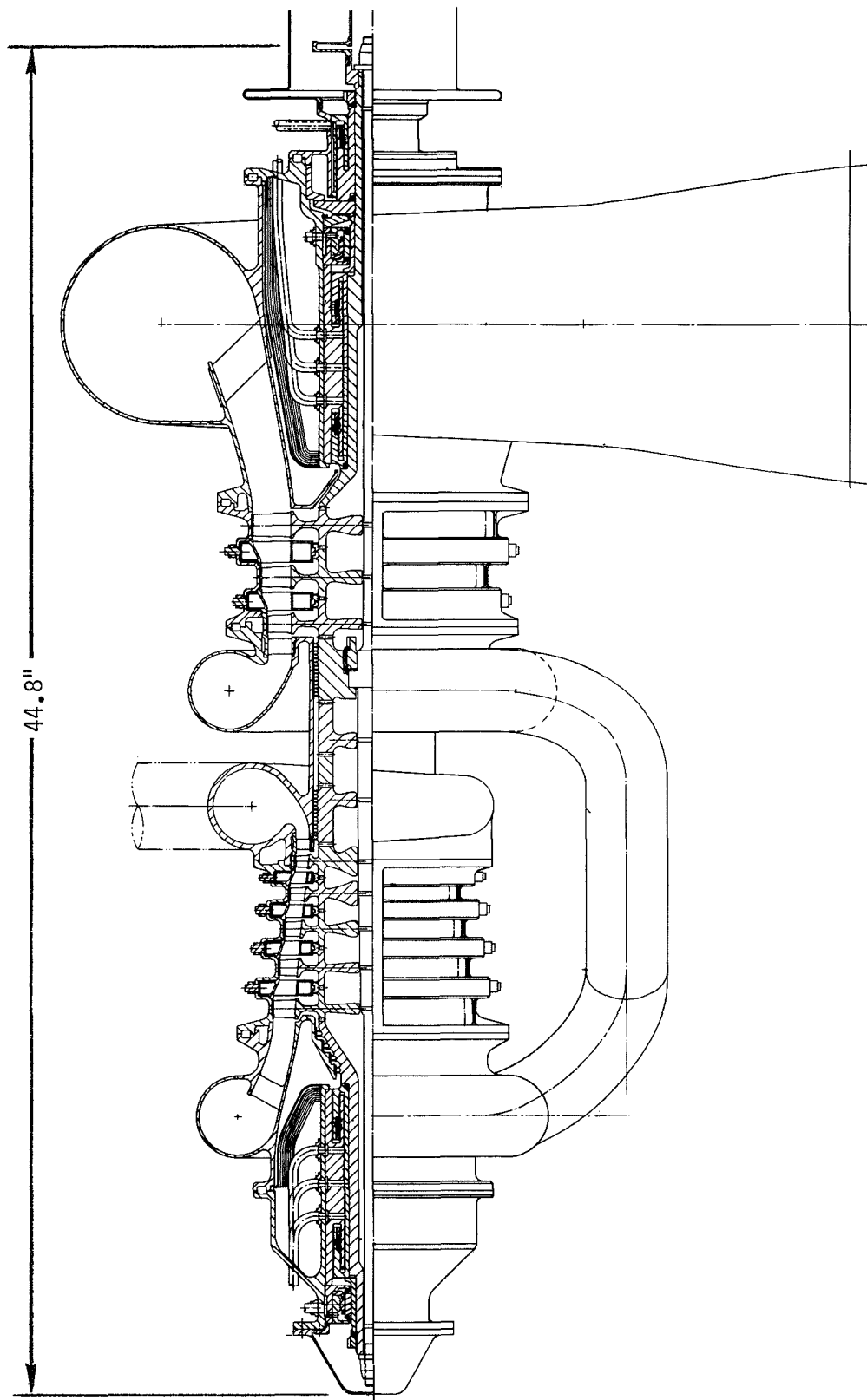


Figure 5.12-2. 8 Stage, 24,000 RPM, Split Spool Counterflow, on Oil Bearings. (Dwg. No. 47R199275)

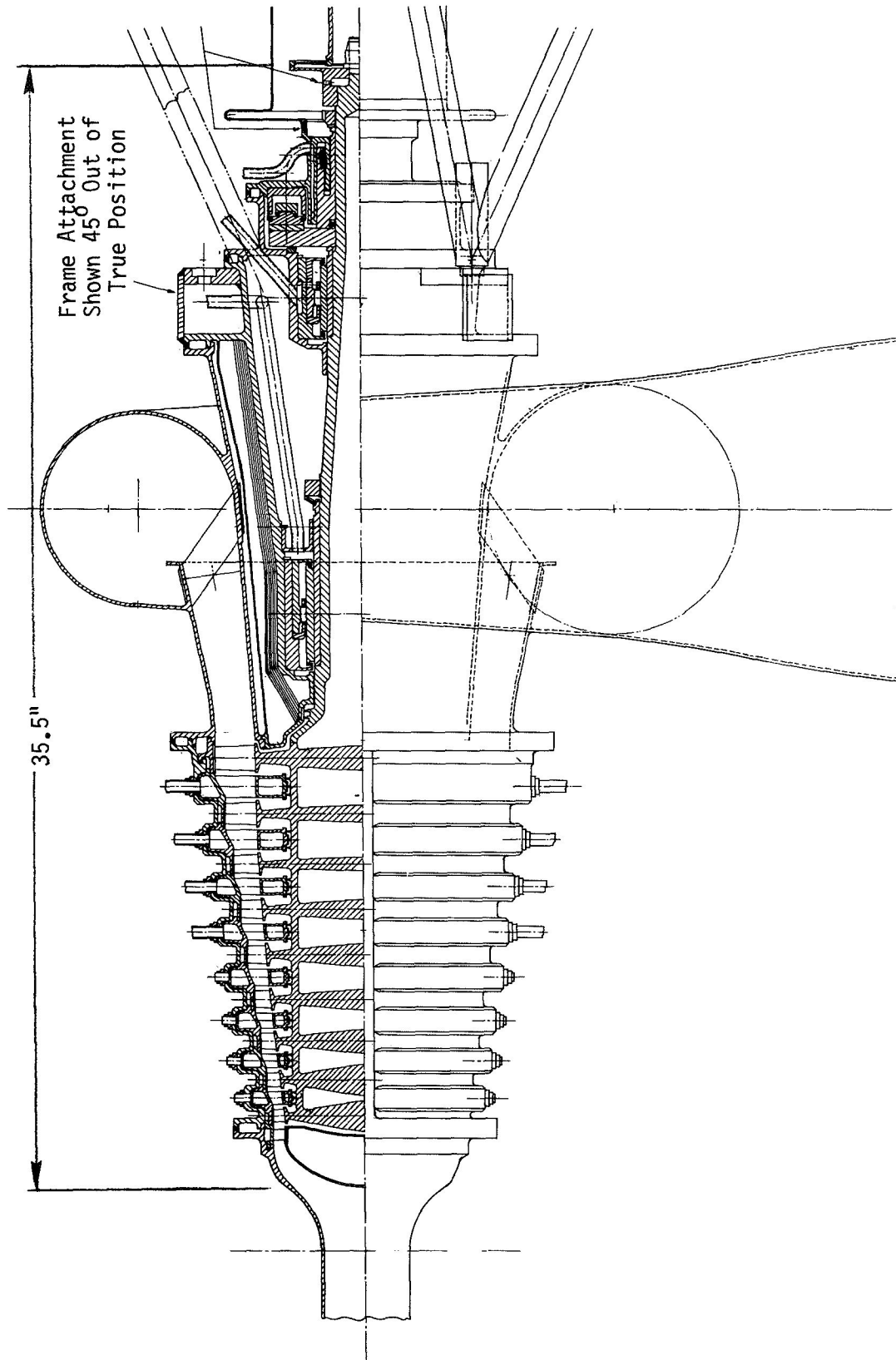


Figure 5.12-3. 9 Stage, 19,200 RPM, Single Spool Overhung, Welded Rotor on K Bearings.
(Dwg. No. 263E287)

For rotor construction, the first choice is the integrally welded rotor using ASTAR-811C and columbium FS-85 as the materials. It is believed to be the most stable design and the only one which eliminates internal vapor leakage. This rotor, however, can be replaced by either a bolted rotor or a curvic coupling/tie bolt rotor using high strength TZC and TZM molybdenum alloy wheels and torque tubes. The turbine stator is designed so that interstage condensate extraction from the stator can be employed if desirable. It is realized that this condensate extraction constitutes a development problem and that its effectiveness is not established at this moment. The same statement, however, is true for other condensate removal devices.

In selecting the split spool turbine for potassium lubricated bearings, it was considered once more that the high pressure seal adjacent to a low temperature bearing and seal area constitutes the biggest single problem, resulting in high loss, steep temperature gradients and unwanted heat flux into the bearing and seal area. It can best be resolved by elimination. Consequently, a 6 + 4 stage split spool design has been chosen where the first six stages are overhung, eliminating the high pressure seal (Figure 5.12-4). The design conditions for the interspool bearing are acceptable. Those for the second bearing are close to optimum. This turbine excels in efficiency. Its axial thrust is cancelled to quite some extent.

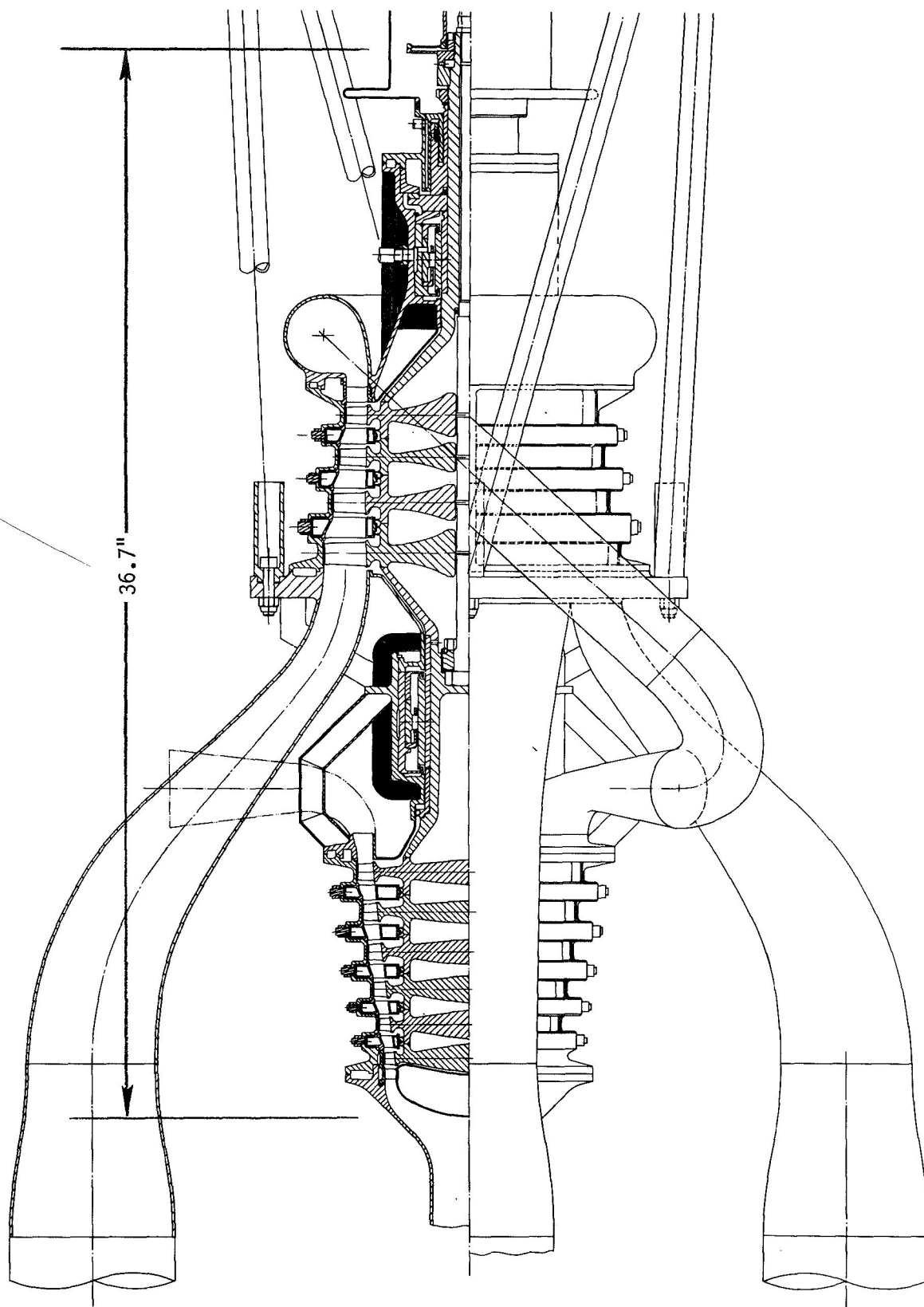


Figure 5.12-4. 10 Stage, 19,200 RPM, Split Spool (6 Overhung and 4 Counterflow) on K Bearings.
(Dwg. No. 47R199290)

5.13 DESIGN EVALUATION

Turbine Weights

Weights of the turbine wheels and buckets were calculated by the basic mechanical design analysis program. The weights of "standard" inlet and outlet scrolls which were also calculated by the program were generally not used, except for checking the hand-calculated weights of ducts and scrolls shown on drawings. The weights of all other rotor and stator parts were calculated by reducing the complex geometries to simple geometric sections and making hand calculations of weight. The calculations extended to all components appearing on a drawing, and no further.

Table 5.13-1 summarizes the weights of the major turbine components of the four recommended configurations. The turbines with a single spool, overhung rotor are lighter than the other configurations. The weights of the rotors are relatively small fractions of the total weights, and greater margins of safety in the rotating parts could be achieved at small overall weight penalty. The high weights of the static parts are estimates, since the thickness of all components have not been optimized.

Comparison of Selected Parameters

Certain selected parameters of interest for the four recommended turbines are summarized in Table 5.13-2. The minimum factors of safety indicate the following:

- The 47R199284 turbine is the only one requiring special care to avoid resonance of blades at design speed.

TABLE 5.13-1
TURBINE WEIGHT EVALUATION FOR SELECTED TURBINES

Configuration		Rotor	Stator	Inlet Scroll	Exit Scroll	Moisture Separator	Total
• Single Spool - 9 Stage Overhung (Interchangeable Rotors) K brgs 19200 rpm	{ Bolted Curvic Welded }	78 90 109	201 201 201	16 16 16	65 65 65	0 0 0	360 372 391
• Split Spool (6 Overhung + 4 Straddle-Mounted) K brgs 19200 rpm		152	255	14	57	42	520
• Single Spool - Split Flow (8 Stages) Oil brgs 24000 rpm		84	232	32	84	0	432
• Split Spool (5 Overhung + 3 Straddle-Mounted) Oil brgs 24000 rpm		70	205	41	77	32	425

TABLE 5.13-2

COMPARISON OF SELECTED PARAMETERS

Parameter	8 Stage 24000 rpm Split Flow Bolted 47R199284	8 Stage 24000 rpm 5 + 3 Counterflow Curvic 47R199286	9 Stage 19200 rpm Overhung Welded 47R199289	10 Stage 19200 rpm 6 + 4 Counterflow Wld/Cv. 47R199290
Factors of Safety (Min.)				
Blade Excitation Freq./Bucket Critical Freq.	0.97	1.67	2.05	2.05
Allowable Deformation Stress/Bucket Root Stress	3.47	1.79	1.23	1.01
Allowable Fracture Stress/Wheel Burst Stress	1.11	1.05	1.35	1.31
Allowable Deformation Stress/Wheel Stress	1.00	1.00	0.73	0.64
3-yr Wheel Creep Stress, % - Stg 1/Stg 2	0.1/0.1	0.1/0.1	0.5/0.15	0.7/0.3
Dovetails Allowed in Stages	5, 6, 7	0	5	5
Bearing Lubricant Temperature, °F	400	400	800	800
Fwd. Bearing/Seal Temperature, °F	1250	2150	1250	1625
Fwd. Bearing/Seal Pressure, psia	7	90	7	65
Blade Tip Radial Clearance Req'd, in.	0.012	0.012	0.024	0.027
Stator/Rotor Axial Clearance Req'd, in.	0.020	0.015	0.020	0.020
Stage 1 Blade Height, in.	0.199	0.398	0.382	0.399
Stage 1 Tip Diameter, in.	4.973	5.172	5.753	5.770
Last Stage Blade Height, in.	0.626	1.220	1.238	1.419
Last Stage Tip Diameter, in.	7.311	7.904	9.259	9.894

- The 47R199289 and 47R199290 turbines require more careful analysis of blade root stresses.
- The ratios of allowable rupture stress/wheel burst stress are nominally satisfactory, but significant increases in these values (for the final two or three stages of the turbines) could be provided by minor weight additions. This course is recommended.
- The first two stages of the ASTAR-811C/FS-85 turbines 47R199289 and 47R199280 must exceed the allowable three-year 0.1 percent creep stress of the material. In order to hold the maximum hub thickness to approximately one inch, three-year creep rates of 0.5 and 0.7 percent and reached in the first stages. Efficiency losses will occur because of leakage through the increased initial tip clearances required.
- Dovetails are generally not allowed in any of the turbines except for certain intermediate stages.

Turbine Mechanical Design Conclusions

1. The high-pressure seal can and should be avoided, for the following reasons:
 - High vapor seal losses of about nine kw for a stepped seal and 17 kw for a stepped seal plus vapor exchange.
 - Difficulty of sufficiently insulating the bearing and seal area to prevent large heat input to the lubricant by condensation.
 - Thermal stresses of about 25,000 psi due to thermal gradients as steep as 1500°F/inch.
 - Probable casing distortion due to unsymmetrical scroll inlet.

- Undesired extension of high temperature rotor structure.
- High temperature end of rotor has low modulus of elasticity, affecting bearing/rotor response.
- High temperature vapor seals are prone to creep and seizure.
- Tie bolt creates leakage problem between high pressure (~ 100 psia), high temperature ($\sim 2000^{\circ}\text{F}$) vapor and bearing cavity (~ 30 psia and $800^{\circ} - 1100^{\circ}\text{F}$).
- Difficulty of forging first stage wheel and stub shaft integral.

2. A ten mil radial tip clearance is feasible on all stages except stages one and two of welded ASTAR 811C/FS-85 rotors.

3. Both welded and bolted rotors are feasible at 19,200 rpm. Bolted rotors are feasible at speeds of 24,000 rpm and above, but welded rotors are not.

4. With both welded and bolted turbines, the 2150°F inlet temperature is feasible if 20 percent overspeed is short-time (yield limited). If 1000 hours at 20 percent overspeed is required, the inlet temperature must be dropped to 2100°F or lower.

5. Blade dovetails are not generally feasible.

6. Bolted rotors are limited in stages one and two by bolt relaxation and creep.

7. Casings are stress-limited in the first two stages.

8. Differential thermal expansion between stator and rotor is not a major problem.

9. The flexible diaphragm coupling will tolerate the space environment and radial misalignments anticipated in the KTA turbomachinery.

Certain key problems have appeared which require attention in the next phase of investigations. The bolted molybdenum alloy turbine has a stage one wheel-plus torque tube structure which appears highly questionable from a forging and three-dimensional strength point of view. Additional studies and stress analyses must be performed to salvage this design. Better stress-relaxation data for the molybdenum alloy bolts is also highly desirable. Although concentric EB welding of turbine stages without undue distortion appears feasible, it constitutes a development which should be practically attacked before investing additional design effort. The design predictions made for stages one and two are based on poorly established, unreliable creep data, which should be scrutinized before entering into Phase II. Condensate removal is considered essential for wet vapor turbines with a three-year life expectancy. Removal from the stator, possibly in conjunction with external condensate removal, is strongly suggested in preference to removal from the rotor.

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6. TURBINE PERFORMANCE

6-1. TURBINE DESIGN CONFIGURATIONS

A turbine with over hung stages is attractive because such a design eliminates the need for a bearing and seal at the high-temperature, high-pressure end. The characteristics of a nine-stage overhung turbine are shown in Figures 6.1-1 and 6.1-2. The high stage loadings (average, 1.99) are due to the low wheel speeds (440 to 670 ft/sec) required for reasonable blade heights in the first stages. A drawing of this turbine design in Figure 6.1-3 illustrates the attractive simplicity of this configuration. Disadvantages are lower efficiency (78.5 percent) due to the stage loading, high axial thrust (2580 lb) and high moisture levels (16.3 percent entering last stage).

An alternate solution which avoids the problems of high temperature bearings is shown in Figure 6.1-4, with stage characteristics shown in Figures 6.1-5 and 6.1-6. This is called a split flow configuration and is basically an eight-stage, 24,000 rpm turbine design in which half of the flow passes through each of two turbines arranged back to back with a common inlet at the center of the bearing span. This configuration has the advantage of balanced thrust, but has very small blade heights (0.20 inch in first stage) which reduce turbine efficiency to 76.8 percent because of tip clearance losses. High stage loading (average, 1.94) also contributes to the low efficiency.

9 STAGE, 19,200 RPM, SINGLE SPOOL OVERHUNG DESIGN

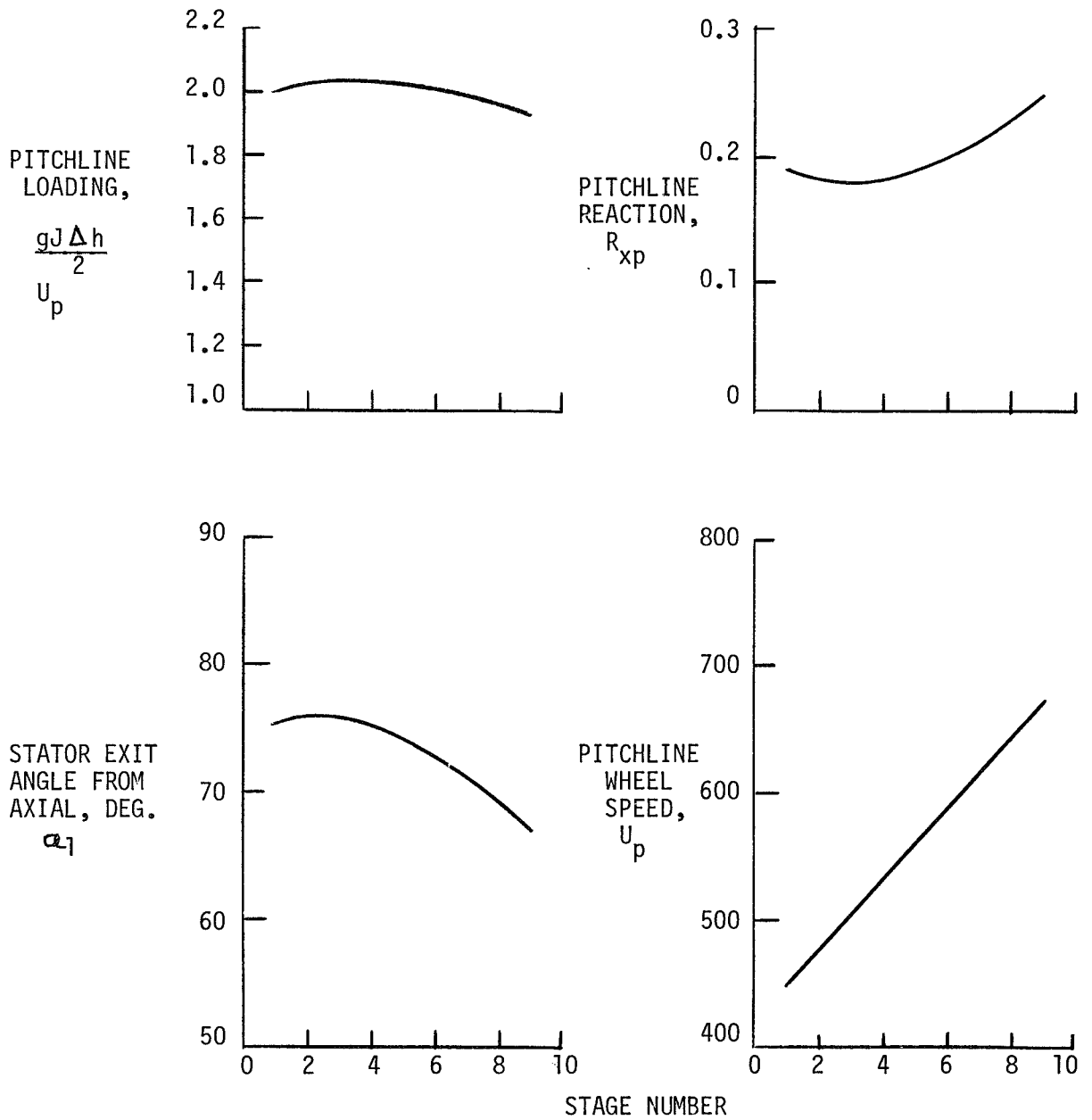


Figure 6.1-1. Stage Characteristics.

9 STAGE, 19,200 RPM, SINGLE SPOOL OVERHUNG DESIGN

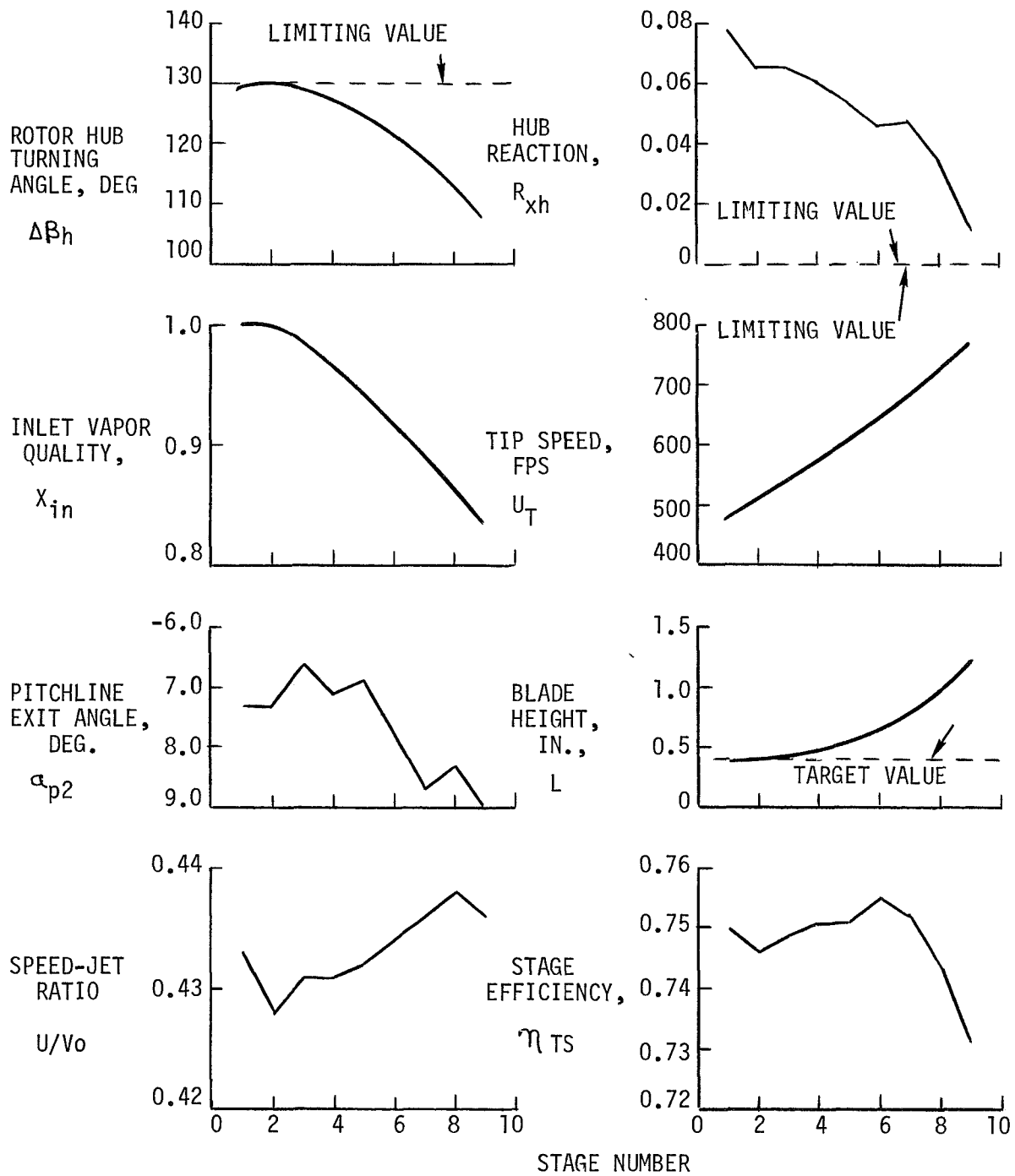
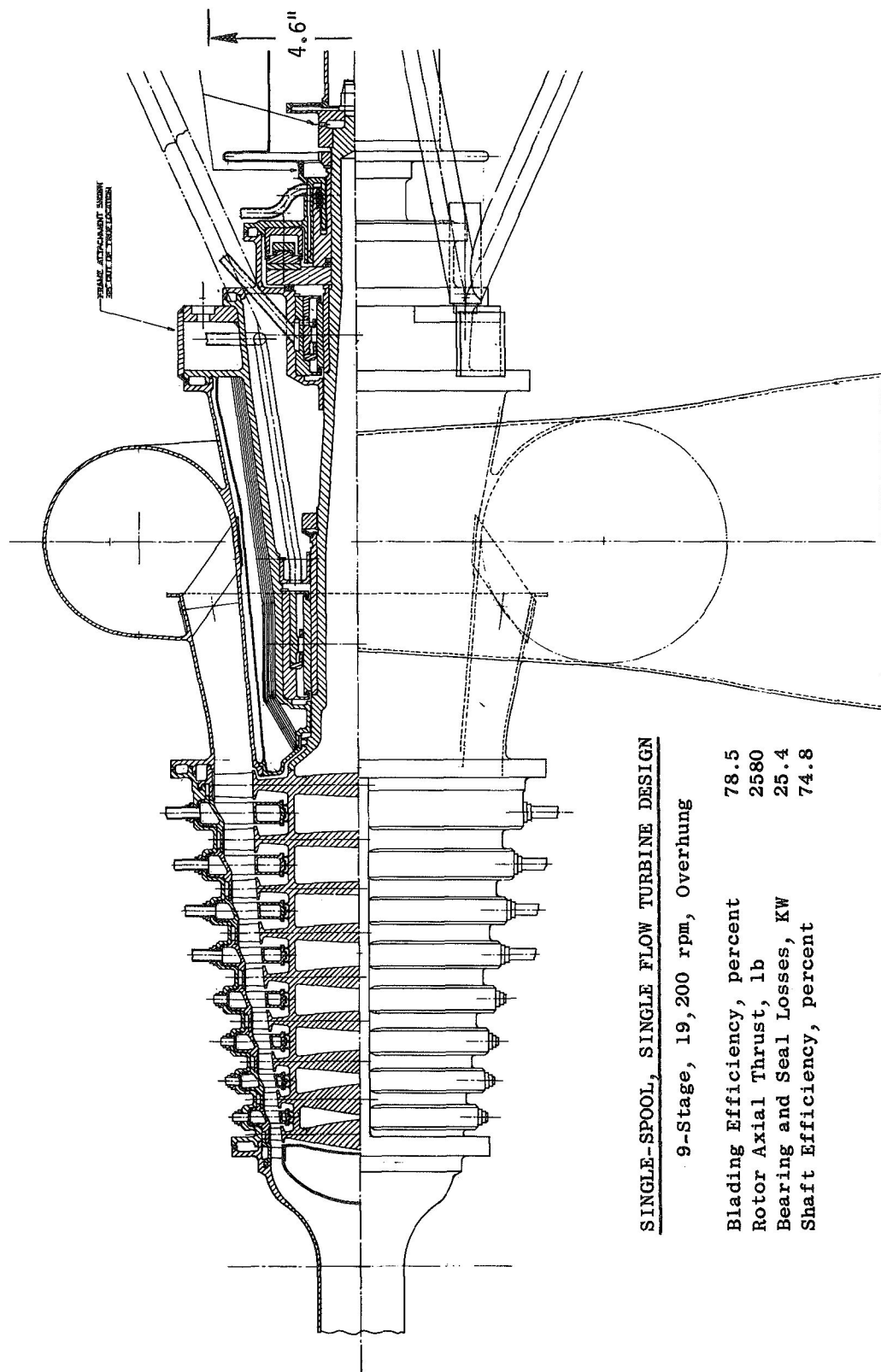


Figure 6.1-2. Stage Parameters.

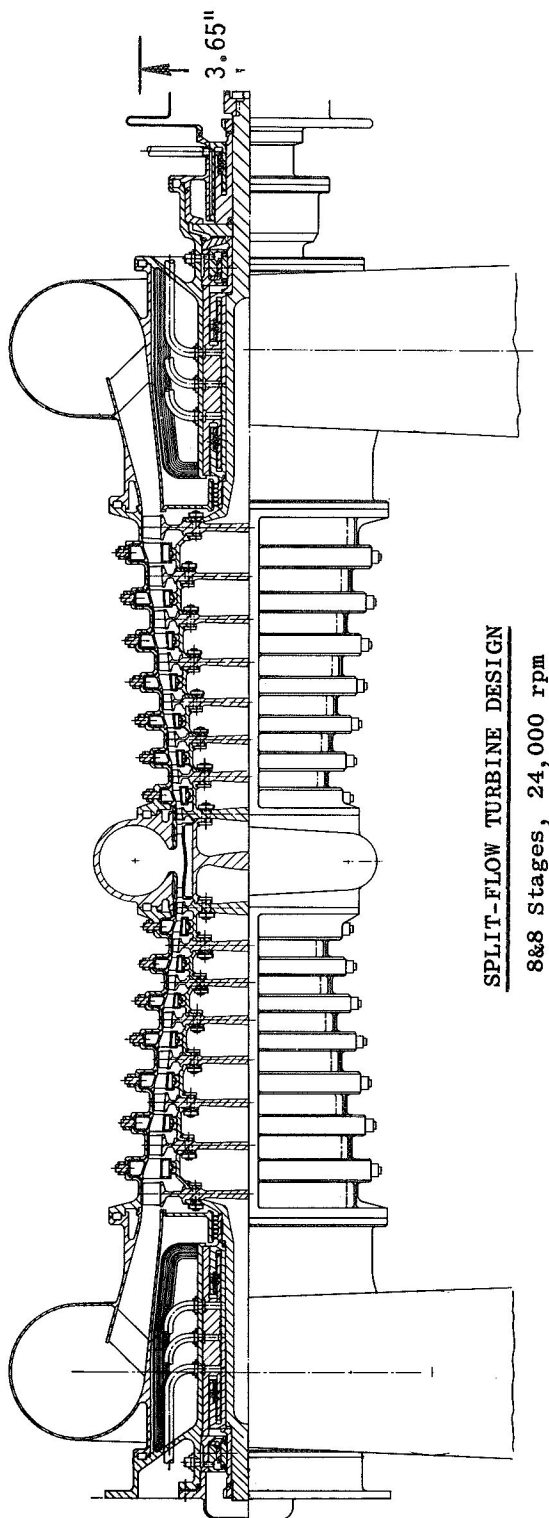


SINGLE-SPOOL, SINGLE FLOW TURBINE DESIGN

9-Stage, 19,200 rpm, Overhung

Blading Efficiency, percent	78.5
Rotor Axial Thrust, lb	2580
Bearing and Seal Losses, KW	25.4
Shaft Efficiency, percent	74.8

Figure 6.1-3. Single-Spool, Single-Flow Turbine Design.



SPLIT-FLOW TURBINE DESIGN

8&8 Stages, 24,000 rpm

Blading Efficiency, percent	76.8
Rotor Axial Thrust, lb	0
Bearing and Seal Losses, KW	26.6
Shaft Efficiency, percent	73.0

Figure 6.1-4. Split-Flow Turbine Design.

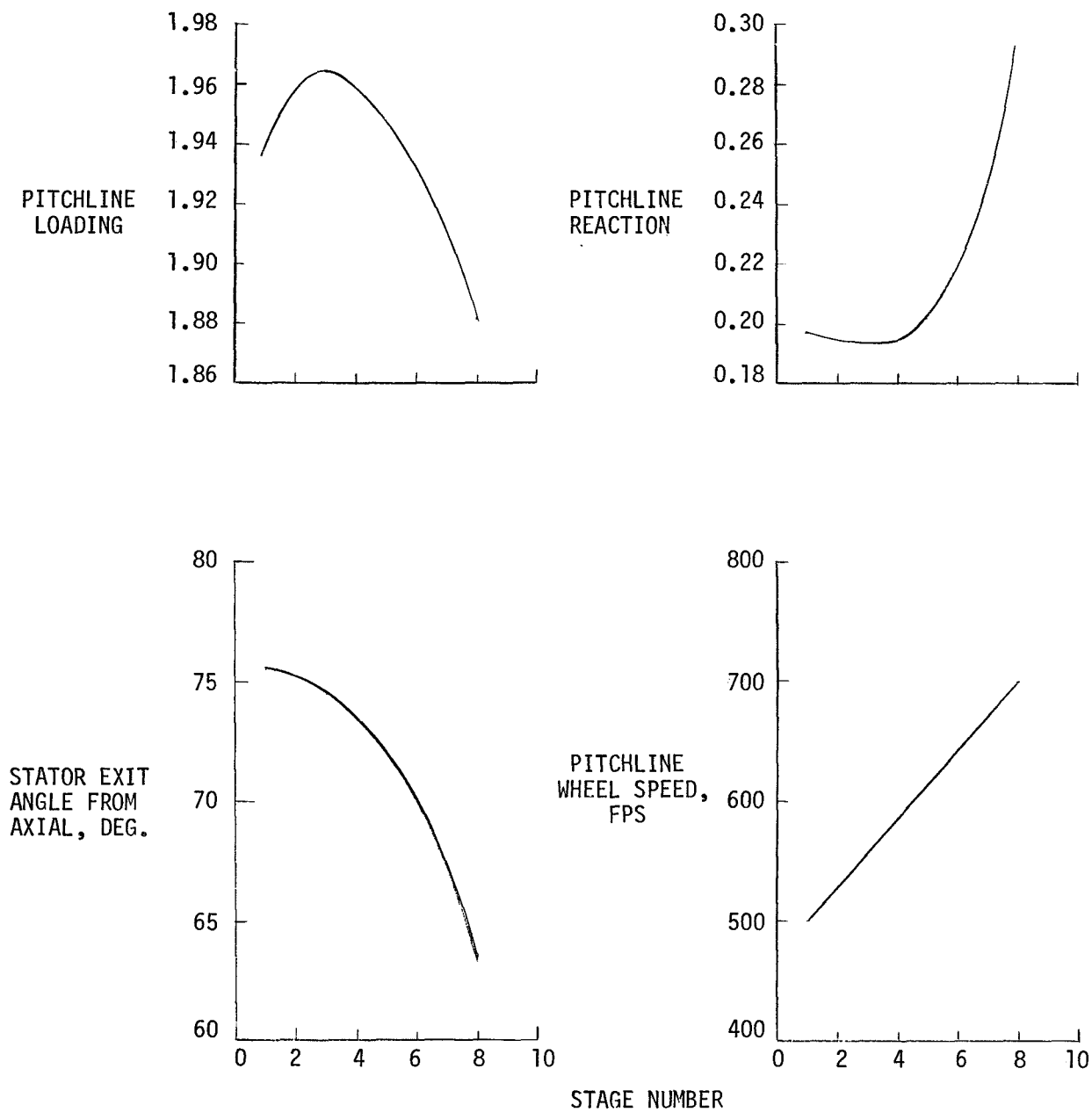


Figure 6.1-5. Split-Flow, Straddle Mounted Turbine-Stage Characteristics
Eight Plus Eight Stage, 24,000 rpm.

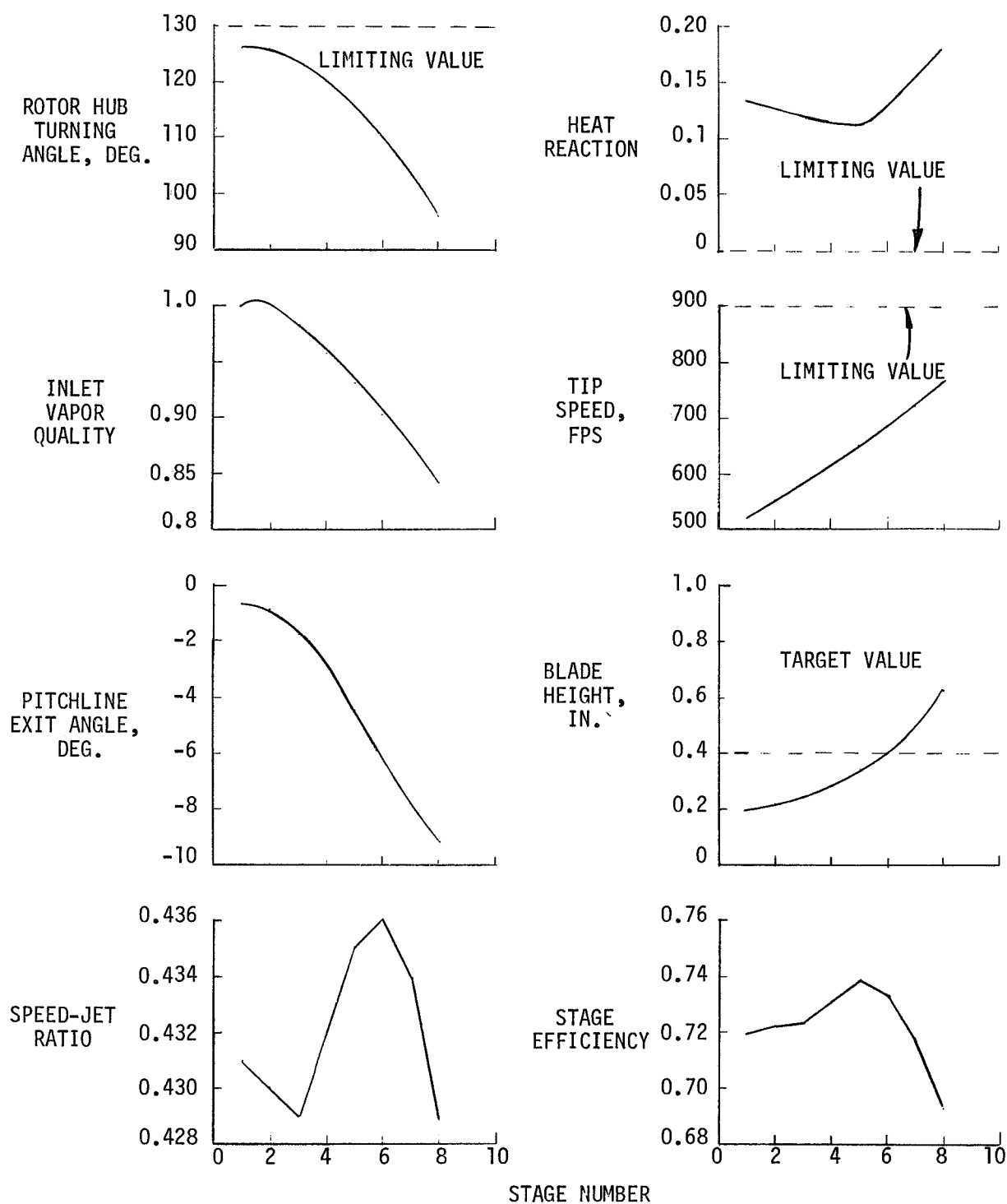


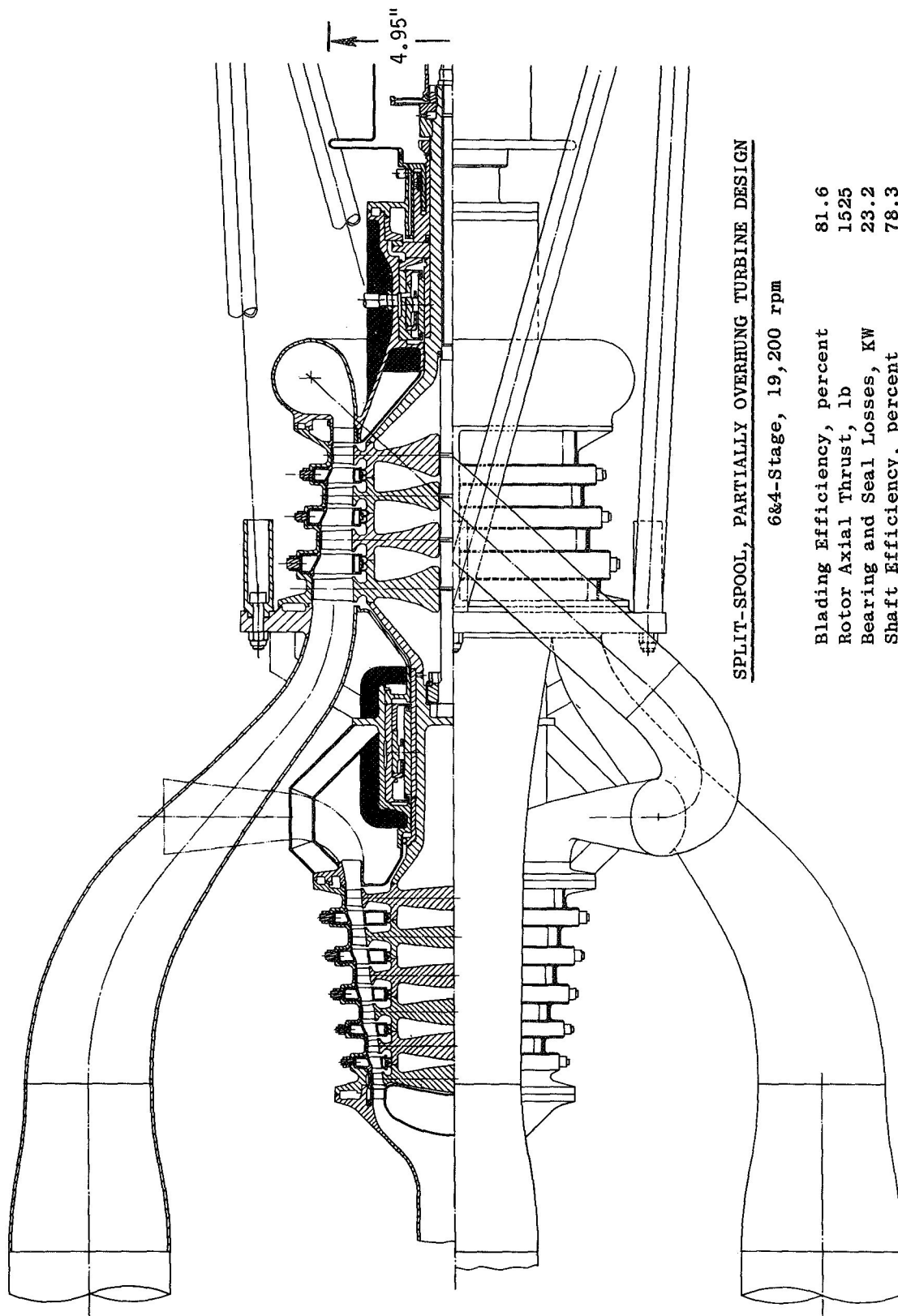
Figure 6.1-6. Split-Flow, Straddle Mounted Turbine-Stage Parameters
Eight Plus Eight Stage, 24,000 rpm.

A turbine configuration which appears to be a good compromise is the split spool, partially overhung design shown in Figure 6.1-7, with stage characteristics shown in Figures 6.1-8 and 6.1-9. In this arrangement, the first six stages are overhung and the last four stages are cradled between two journal bearings. The flow out of the high pressure turbine is ducted to an external moisture separator where most (80 percent) of the condensed liquid is removed before the flow enters the low pressure turbine. Because the wheel speeds of the last four stages are not limited by small blade heights, reasonable stage loadings and good turbine efficiency can be obtained. The average stage loading is 1.75 and the blading efficiency is 81.6 percent for this design. The labyrinth seal loss is only 8.9 KW. The main disadvantage of this design is the rather complex ducting and forward bearing structure.

A split spool, straddle mounted design for 24,000 rpm has five high pressure stages and three low pressure stages, shown in Figure 6.1-10, with stage characteristics shown in Figures 6.1-11 and 6.1-12. This design has higher stage loadings (average, 1.95), greater labyrinth seal leakage losses (21.4 KW) and lower turbine efficiency (77.9 percent) than the previous design.

6.2 TURBINE INLETS

Among the various turbine designs which were made, two kinds of inlets are shown, namely, involute scrolls and axial inlets. Based upon tests (Ref. 6.2-1) of a similar inlet, the involute scrolls were assumed to have a loss of 0.25 of the inlet velocity head, and the axial inlets were assumed to have no appreciable loss. A typical



SPLIT-SPOOL, PARTIALLY OVERHUNG TURBINE DESIGN

6&4-Stage, 19,200 rpm

Blading Efficiency, percent	81.6
Rotor Axial Thrust, lb	1525
Bearing and Seal Losses, KW	23.2
Shaft Efficiency, percent	78.3

Figure 6.1-7. Split-Spool, Partially Overhung Turbine Design.

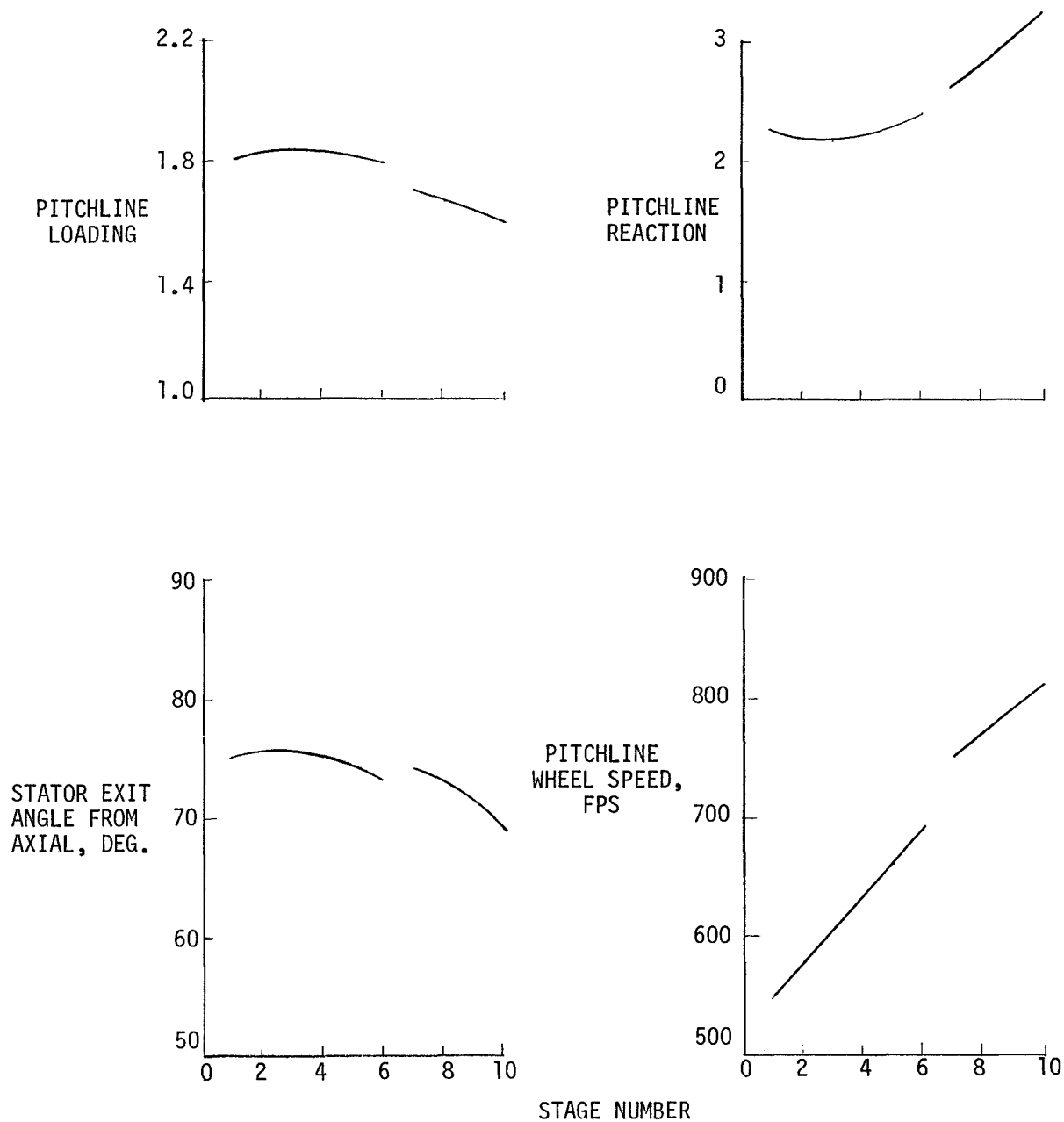


Figure 6.1-8. Split Spool, Partially Overhung Turbine-Stage Characteristics
Six Plus Four Stage, 19,200 rpm.

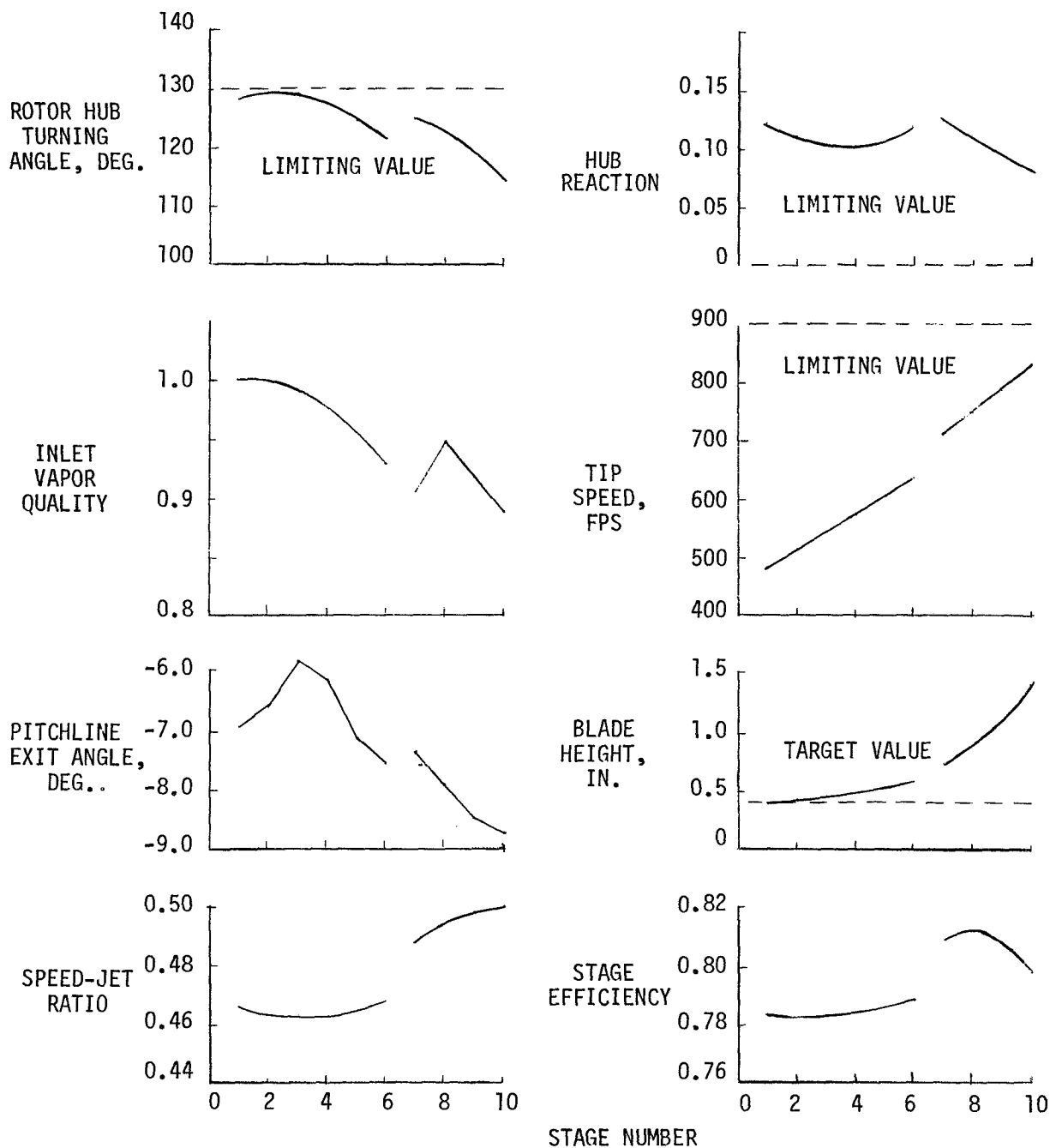


Figure 6.1-9. Split Spool, Partially Overhung Turbine-Stage Parameters
Six Plus Four Stage, 19,200 rpm.

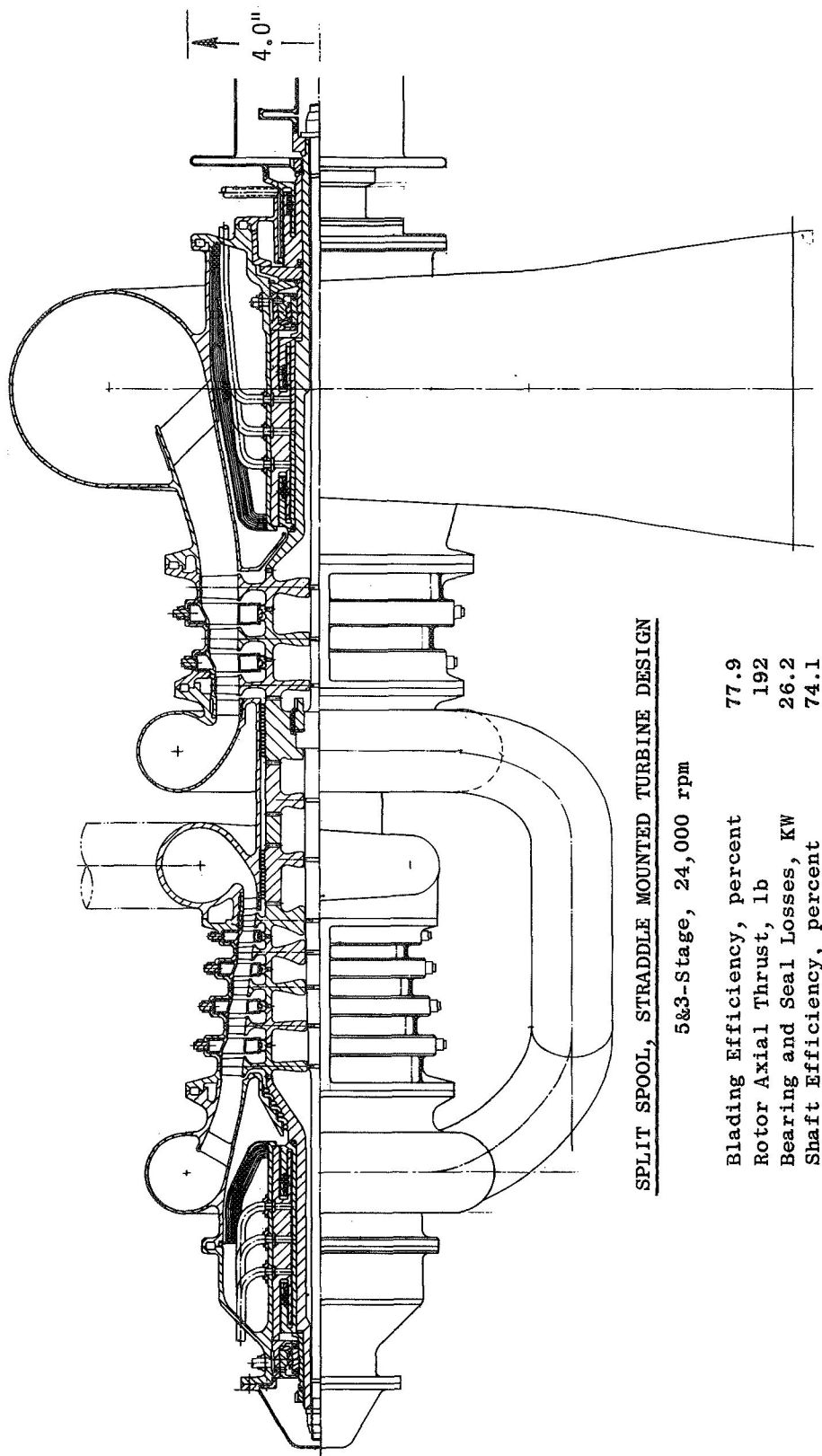


Figure 6.1-10. Split-Spool, Straddle-Mounted Turbine Design.

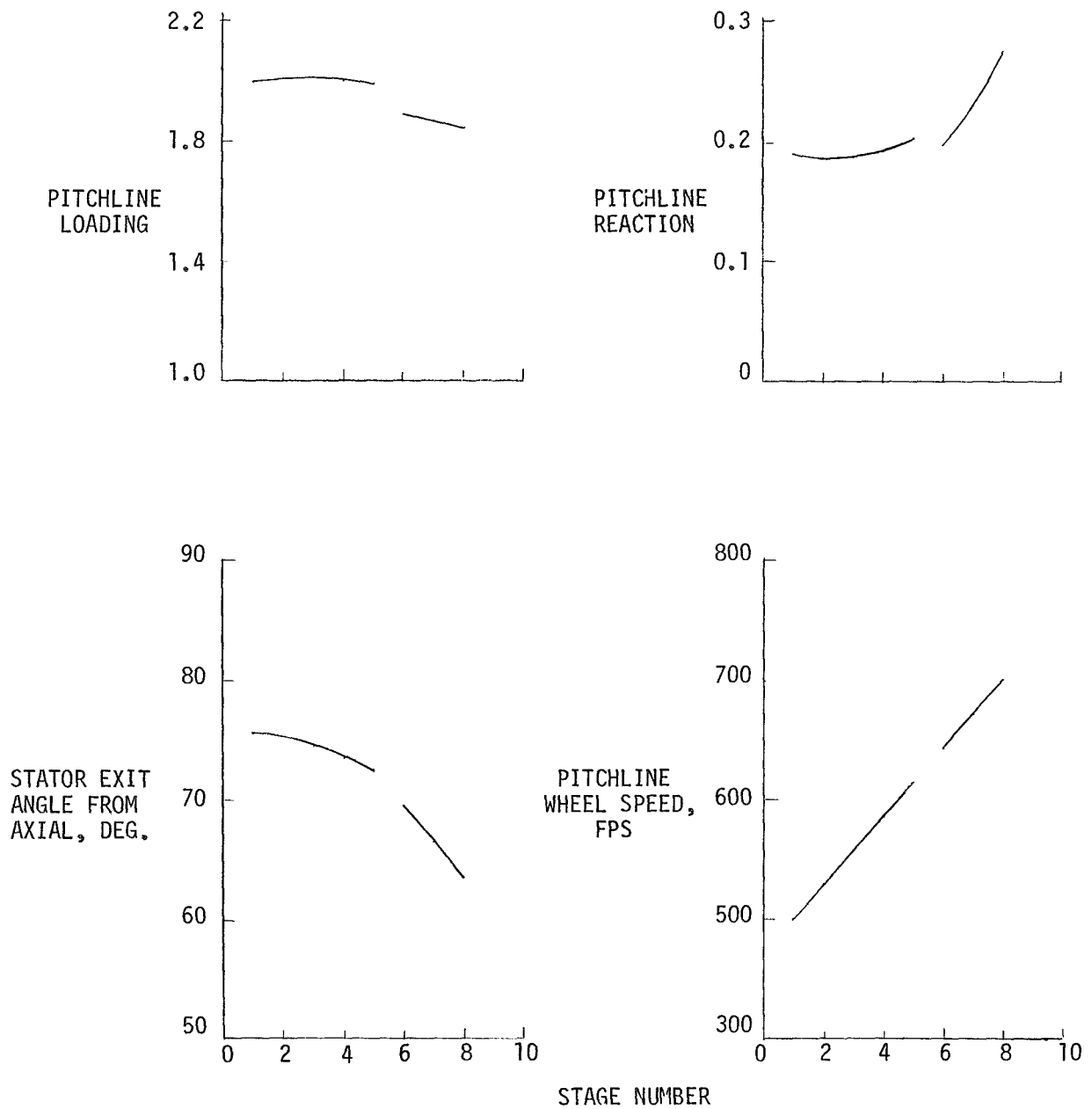


Figure 6.1-11. Split Spool, Straddle Mounted Turbine-Stage Characteristics
Five Plus Three Stage, 24,000 rpm.

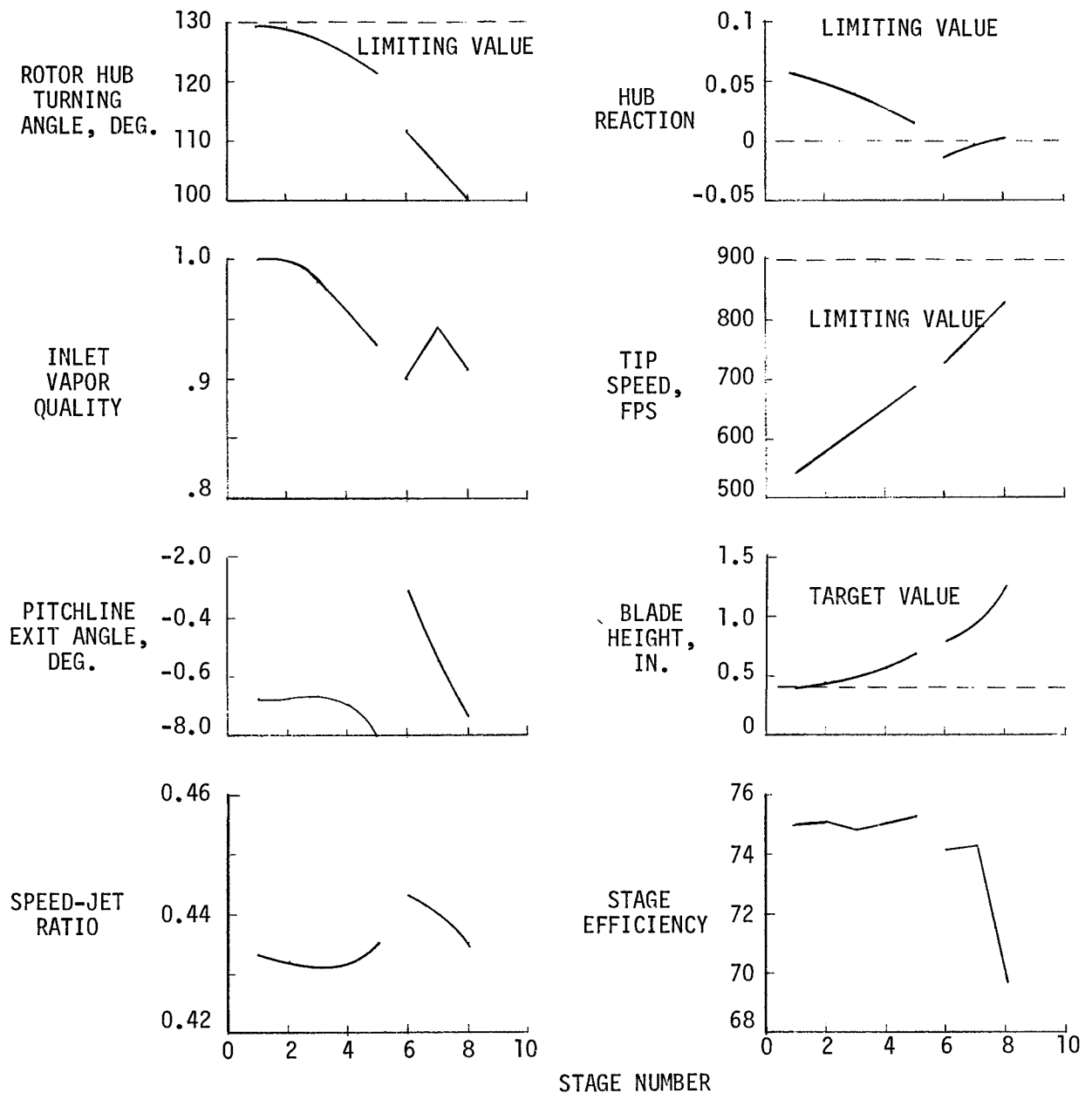


Figure 6.1-12. Split Spool, Straddle Mounted Turbine-Stage Parameters
Five Plus Three Stage, 24,000 rpm.

involute scroll is shown in Figure 6.2-1. The flow enters the turbine in the tangential direction. The inlet shown is for the high-pressure turbine. For the low-pressure turbine, the flow enters the turbine tangentially in four separate pipes but the configuration is otherwise similar to Figure 6.2-1. A typical axial inlet is shown in Figure 6.2-2. The inlet losses for the various turbine designs are summarized in Table 6.2-1.

6.3. TURBINE OUTLETS

The turbine outlet must collect the turbine exit flow and duct it to the condenser. Shown in Figure 6.3-1 is the variation in turbine exit-axial velocity with number of stages and rotative speed. All velocities shown are in excess of a reasonable duct velocity (200 to 300 ft/sec). Shown in Figure 6.3-2 is the variation of overall turbine efficiency with turbine exit axial velocity and total-pressure-loss coefficient for a turbine with total-to-total efficiency of 79.5 percent. The lowest curve is for a one exit head loss (no diffuser). Three points in efficiency are lost in this way at 700 ft/sec. The curves for 0.8 and 0.5 relate efficiency to rotor exit velocity with a diffusing scroll having the loss coefficient shown. In Figure 6.3-3 is shown similar results for some reasonable diffuser velocity ratios. The diffuser losses were taken from Henry, et. al. (Ref. 6.3-1). In spite of the loss due to diffusion, the overall efficiency of the turbine is higher with as much diffusion as is possible short, of course, of diffuser separation. From test data on exhaust hoods, a head loss of 50 percent was established as reasonable.

Shown in Figures 6.3-4 and 6.3-5 are two scroll configurations

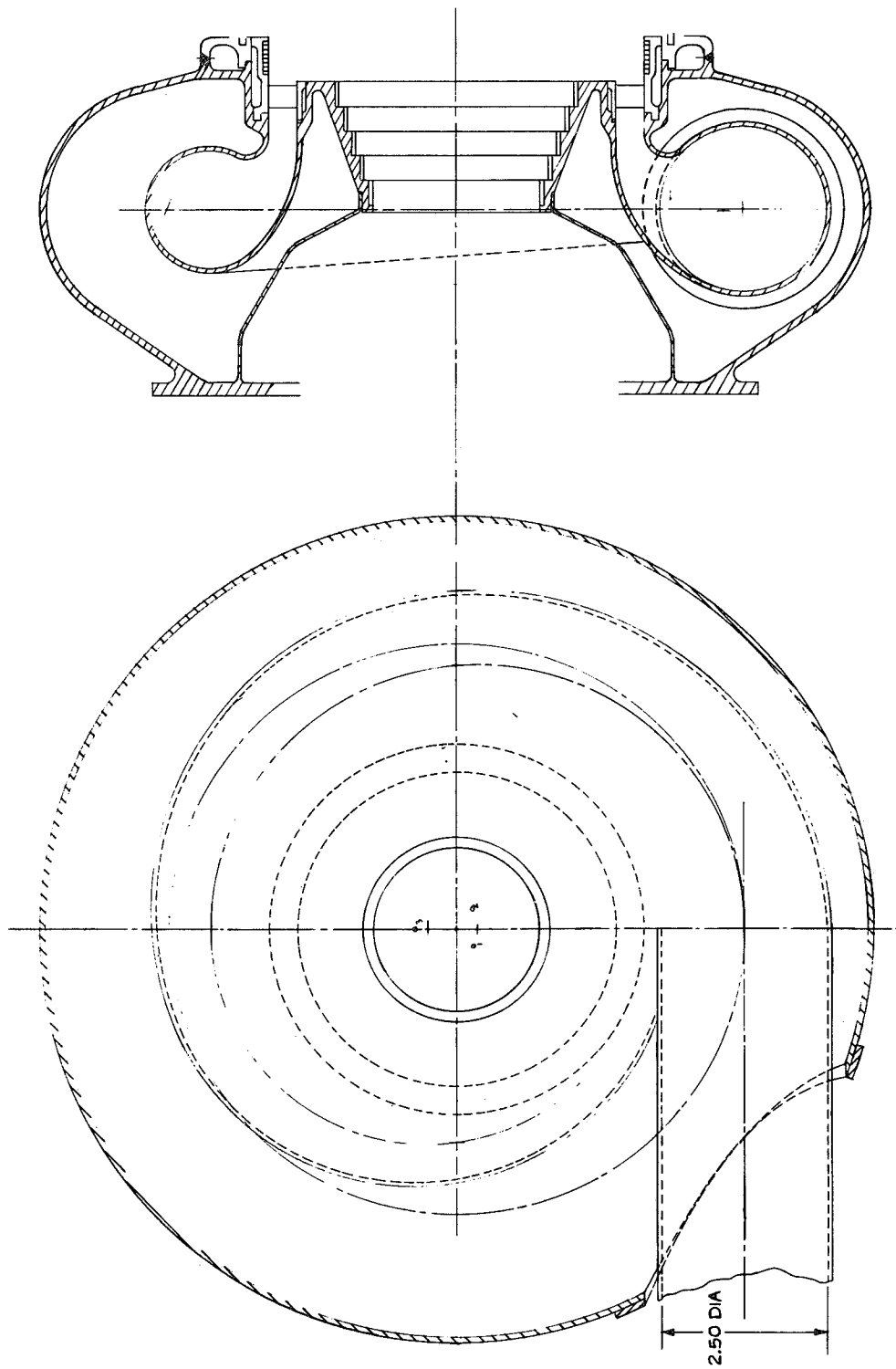


Figure 6.2-1. Involute Inlet Scroll.

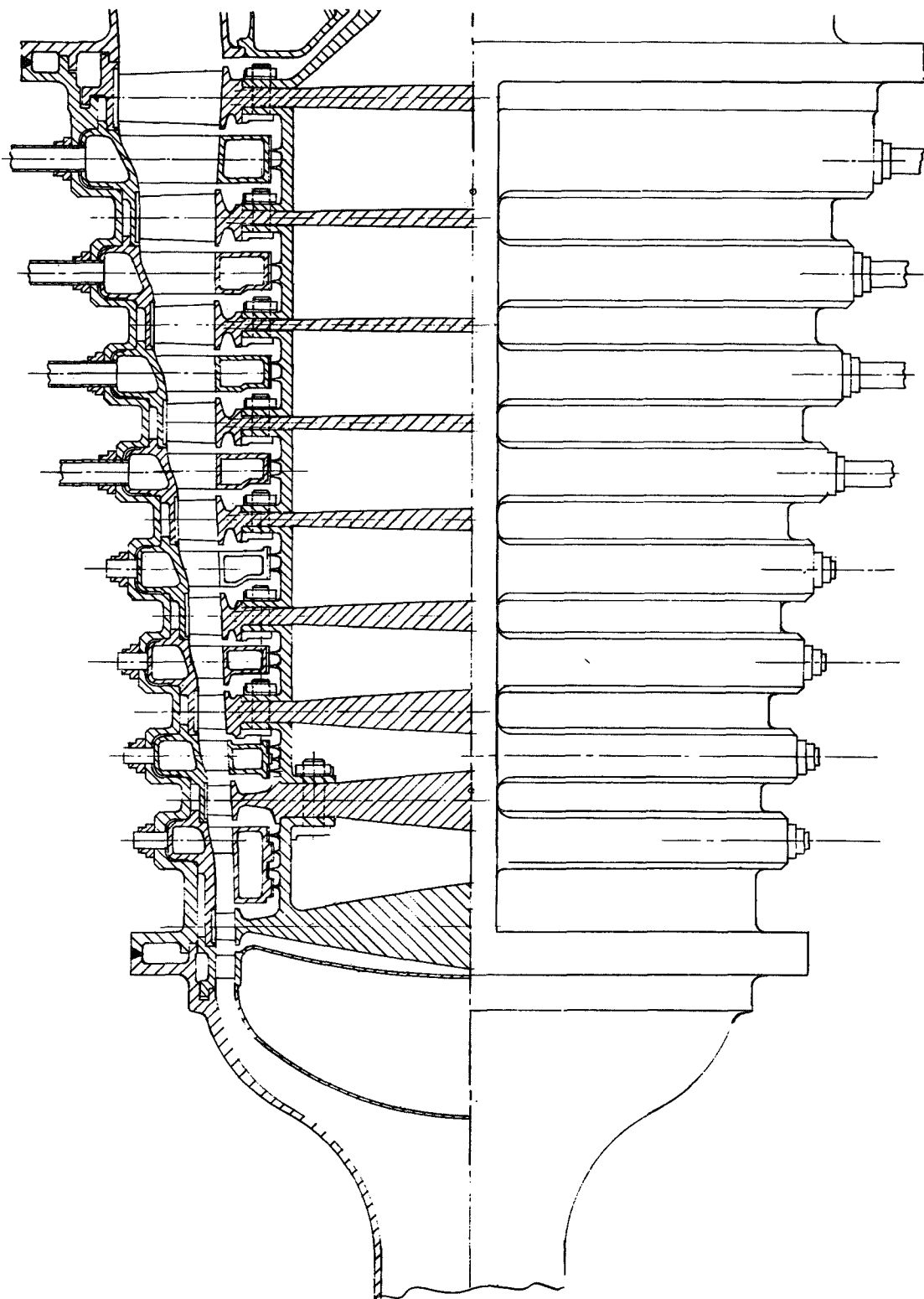


Figure 6.2-2. Axial Inlet.

TABLE 6.2-1

SUMMARY OF INLET LOSSES

<u>TYPE INLET</u>	<u>TURBINE DESIGN</u>	<u>INLET PRESSURE, PSIA</u>	<u>APPROACH VELOCITY, FPS</u>	<u>TOTAL PRESSURE LOSS COEFFICIENT</u>	<u>PRESSURE LOSS, PSI</u>
AXIAL	SINGLE-SPOOL, OVERHUNG	165	250	0	0
HIGH PRESSURE AXIAL	SPLIT SPOOL, PARTIALLY OVERHUNG	165	250	0	0
LOW PRESSURE INVOLUTE	SPLIT SPOOL, PARTIALLY OVERHUNG	38.4	250	0.25	.14
INVOLUTE	SPLIT FLOW	165	250	0.25	.47
HIGH PRESSURE INVOLUTE	SPLIT SPOOL, STRADDLE MOUNTED	165	250	0.25	.47
LOW PRESSURE INVOLUTE	SPLIT SPOOL, STRADDLE MOUNTED	32.8	250	0.25	.12

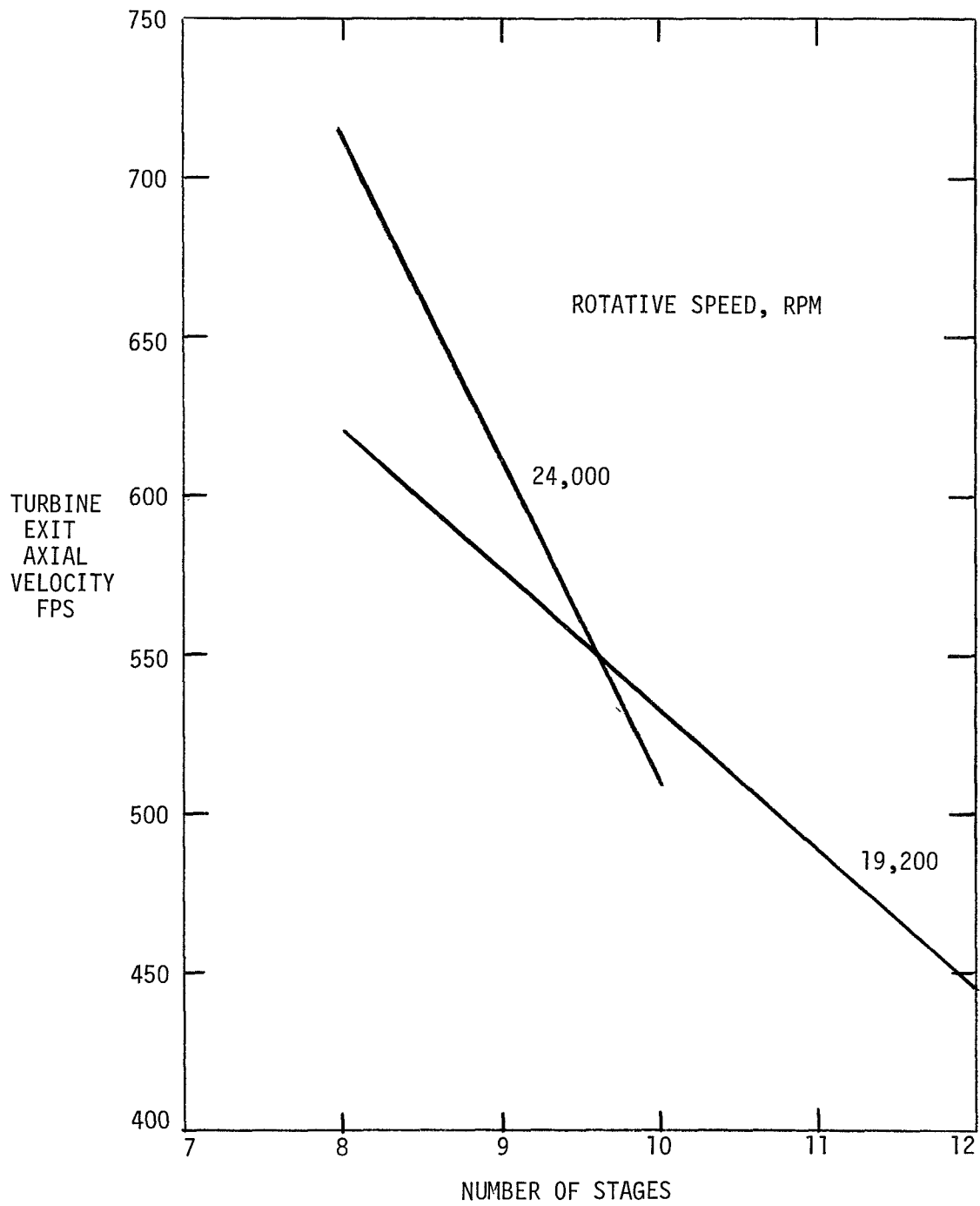


Figure 6.3-1. Turbine Exit Velocity Variation.

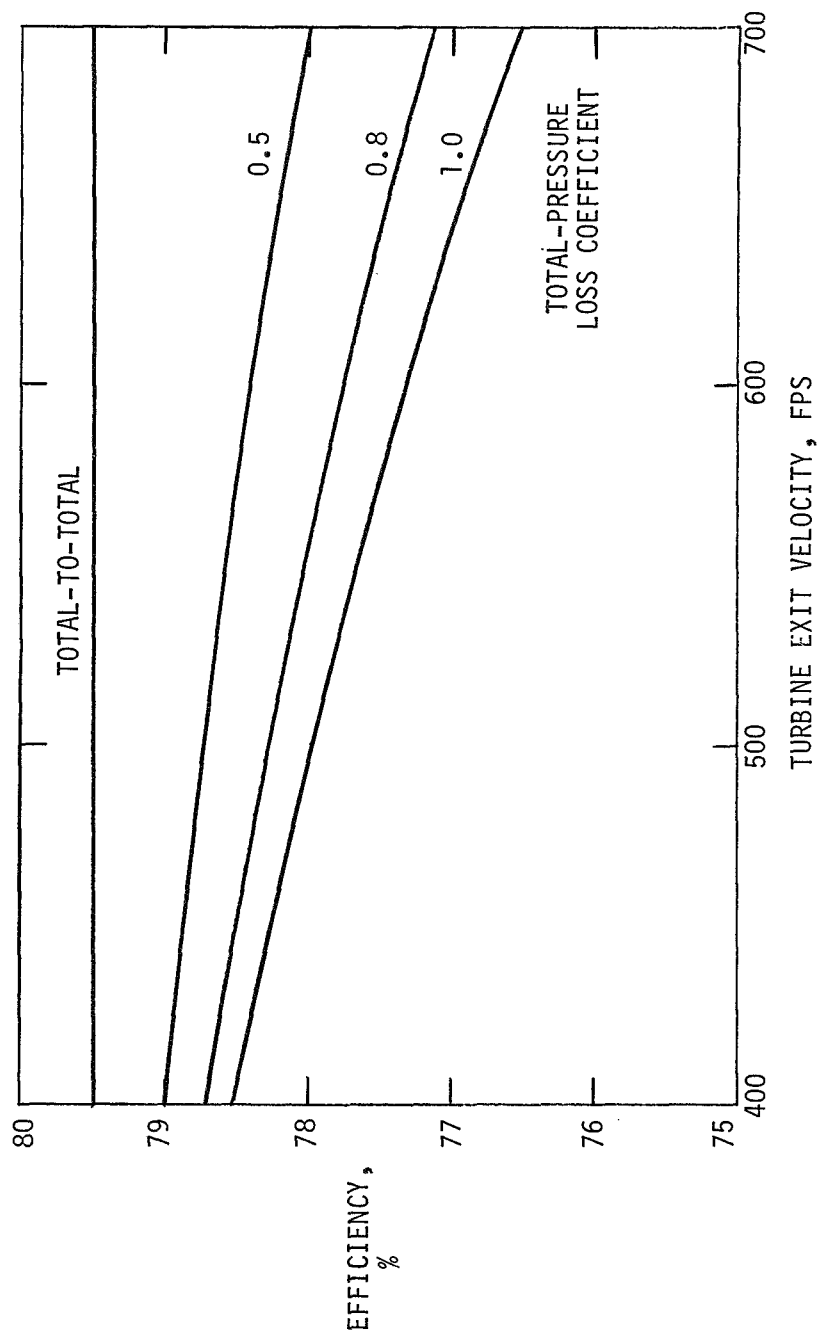


Figure 6.3-2. Estimated Scroll Losses.

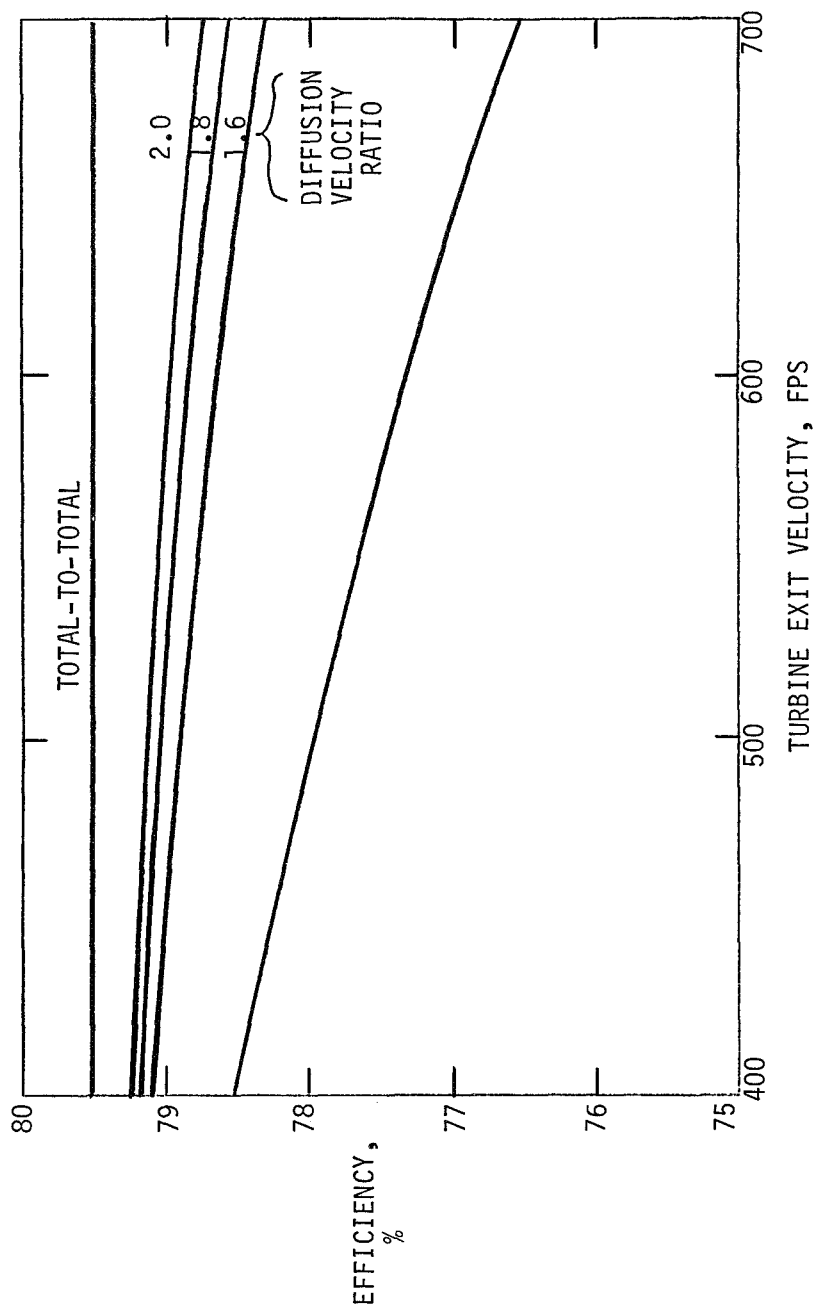


Figure 6.3-3. Effect of Diffusion on Scroll Losses.

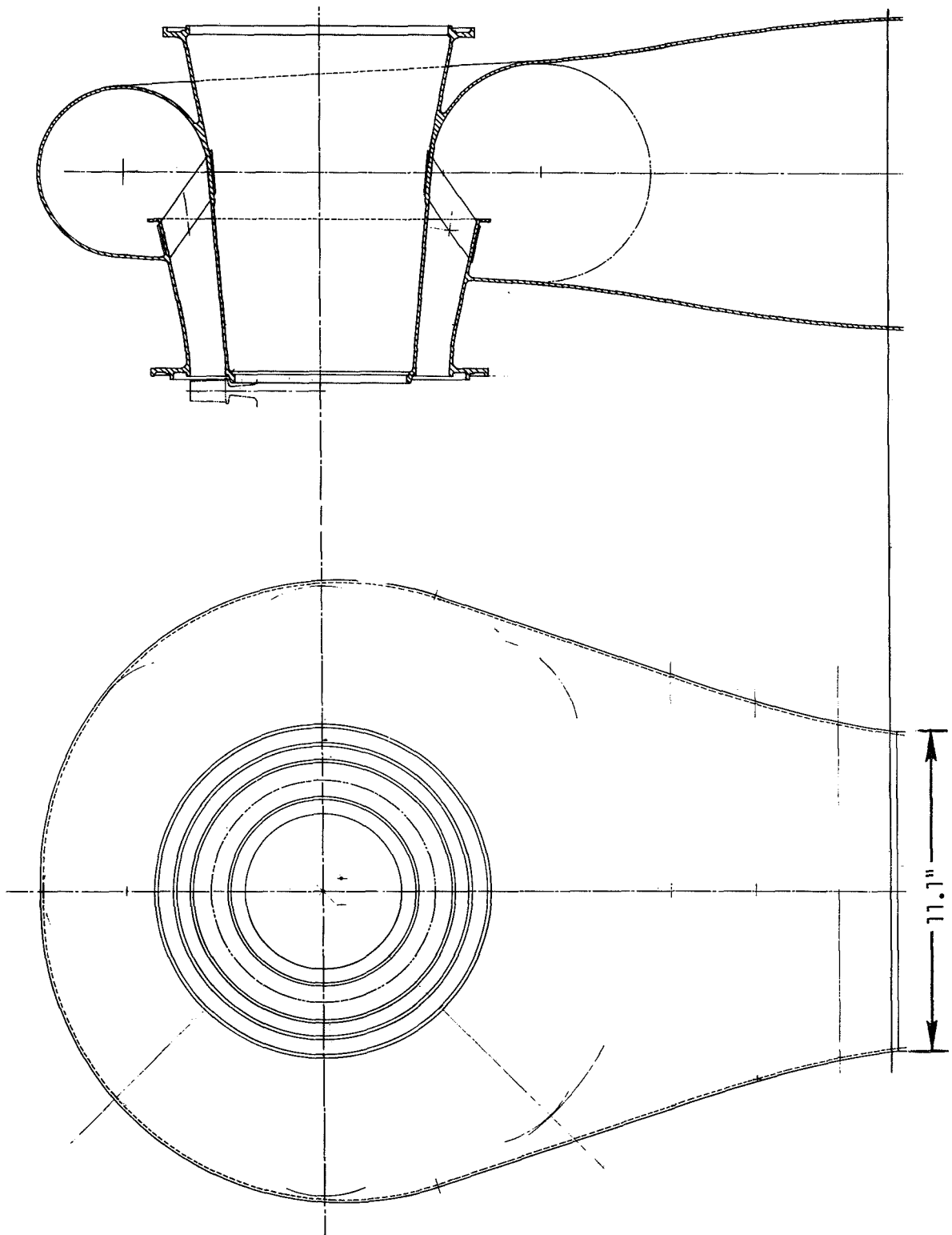


Figure 6.3-4. Turbine Exit Scroll (Dump).

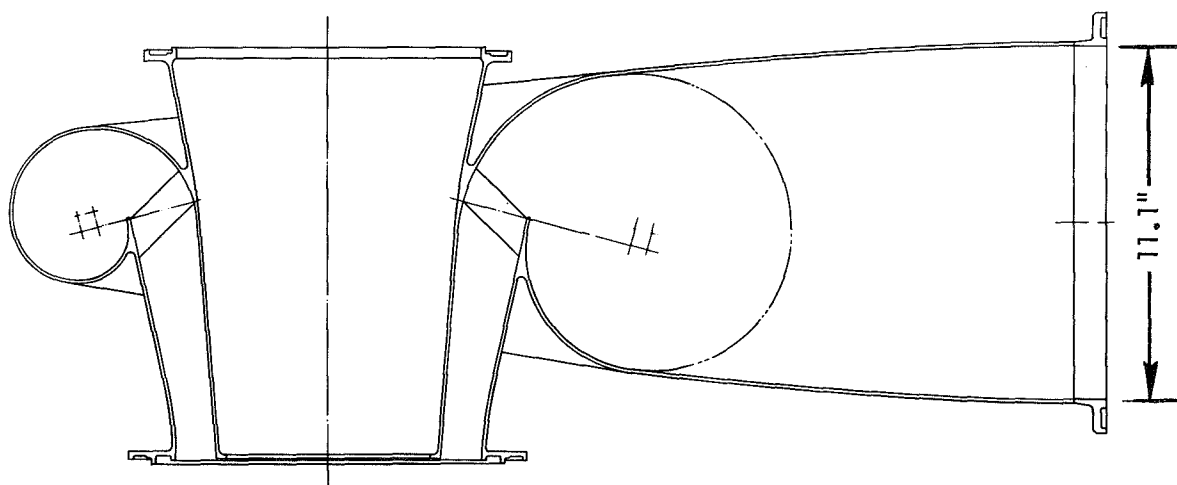
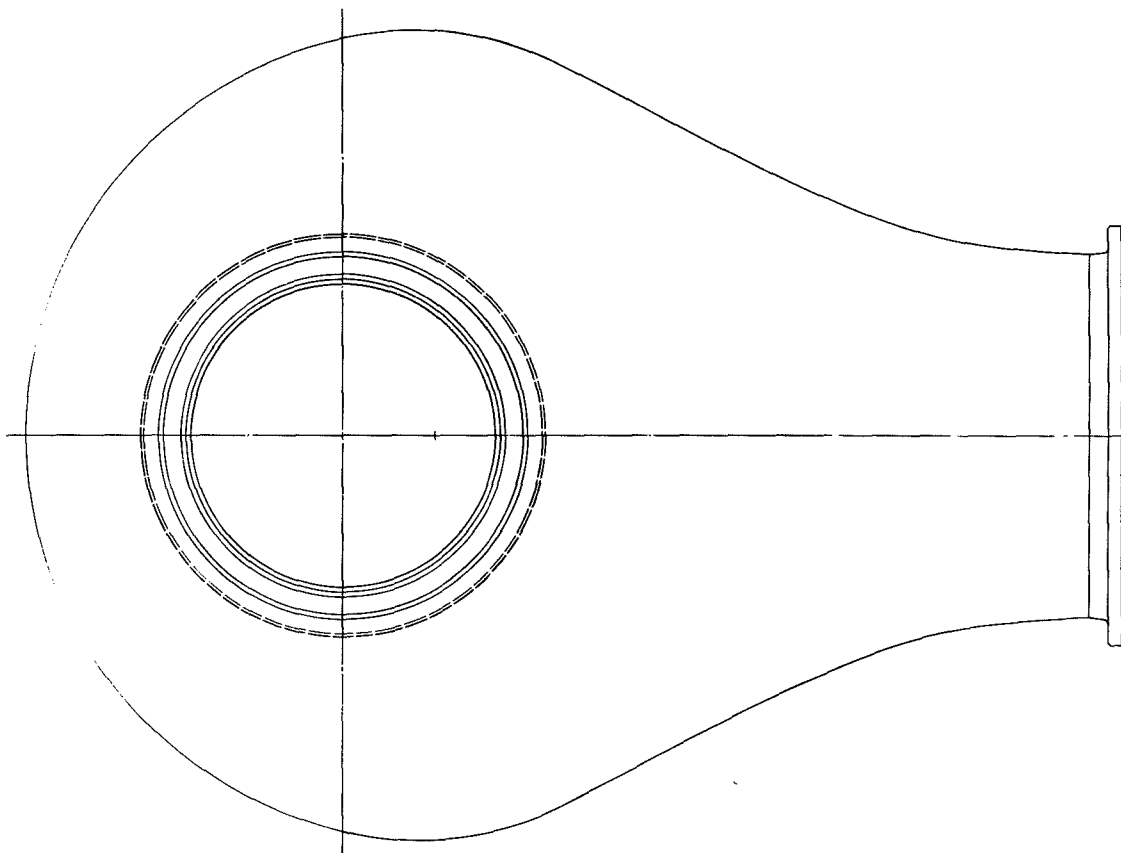


Figure 6.3-5. Spiral Turbine Exit Scroll.

for turbine exit, both of which have an exit velocity of 190 ft/sec, a reasonable condenser inlet velocity. The scroll of Figure 6.3-4 diffuses to about 360 ft/sec in an annular diffuser and then dumps into the scroll, losing most of the remaining dynamic head. The aft portion of the annular diffuser forms a baffle to break up the vortex forward (after dumping). The scroll of Figure 6.3-5 has a similar annular diffuser but attempts to avoid a direct dump. Each scroll has struts across the annular diffuser to carry turbine casing loads through the scroll. These struts are placed near the diffuser exit so as to postpone any tendency toward separation as long as possible. The struts have an airfoil cross section and are provided with 10 or 20 degrees of camber, so that the flow flows smoothly down both sides of the scroll. Because of the large turbine exit volume flow, both scrolls require a sizeable volume.

Shown in Figure 6.3-6 is a quadfurcated outlet for the split spool turbine. Such an outlet was required to minimize the distance between the low pressure and high pressure turbines. Shown in Figure 6.3-7 is a single outlet scroll for a similar split-spool turbine. A considerable increase in the distance between the high and low pressure turbines is required to accommodate this scroll. Shown in Figure 6.3-8 is the quadfurcated outlet applied to the nine-stage overhung turbine. It results in a very compact arrangement when the condensers are closely coupled to the turbine exit. The losses associated with the exits of the various turbines are summarized in Table 6.3-1.

6.4 EROSION AND MOISTURE EXTRACTION

Historically central-power-station-steam-turbine stages operating

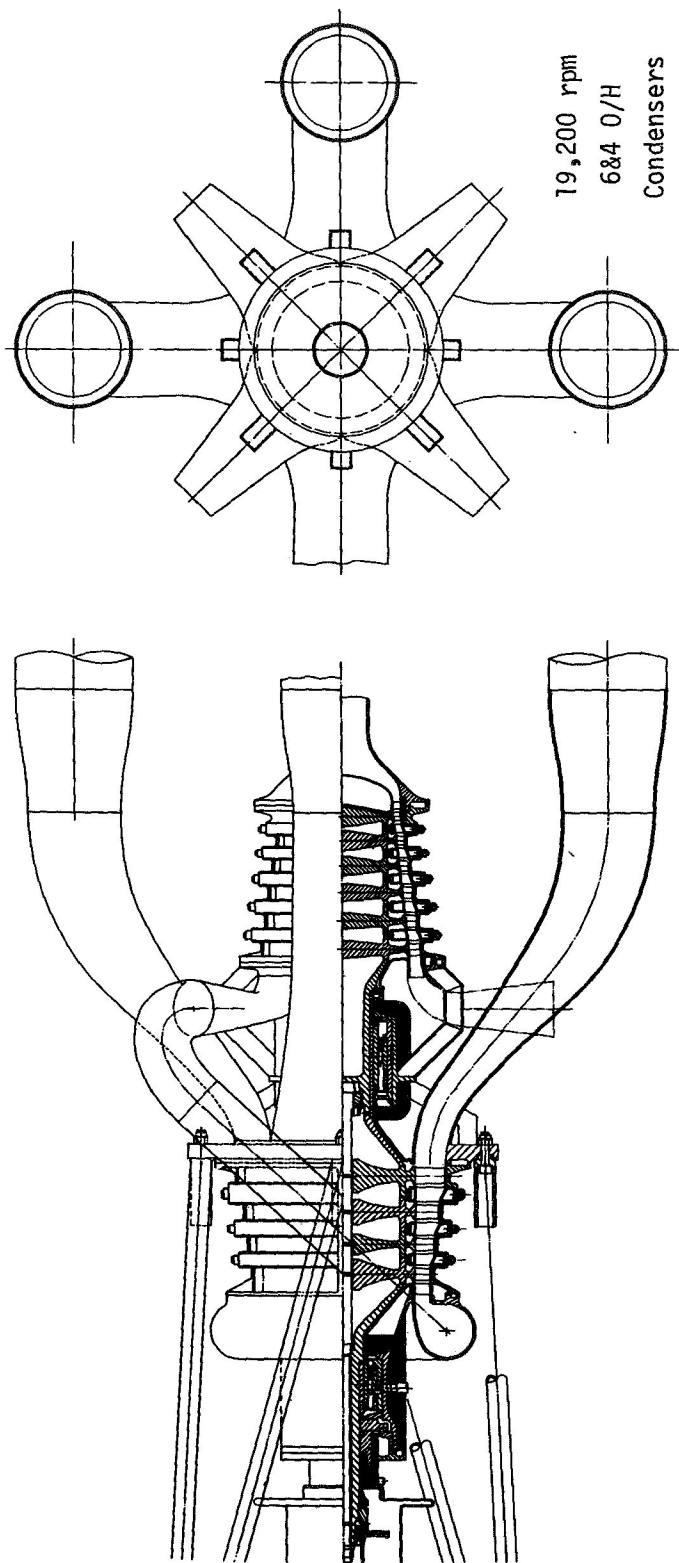


Figure 6.3-6. Quadfurcated Exits for Split Spool Turbine.

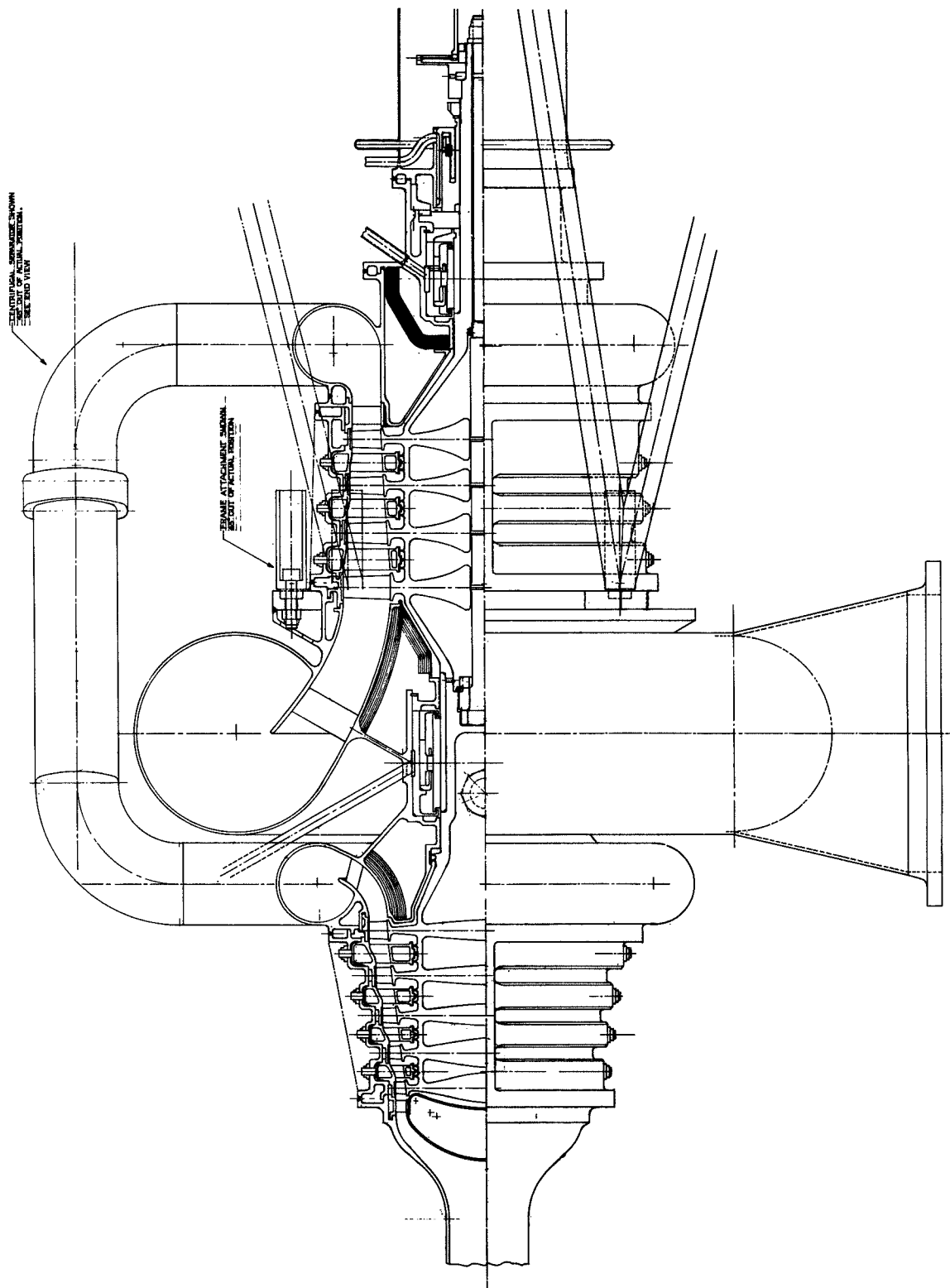


Figure 6.3-7. Straddle-Mounted Turbine Exit Scrolls.

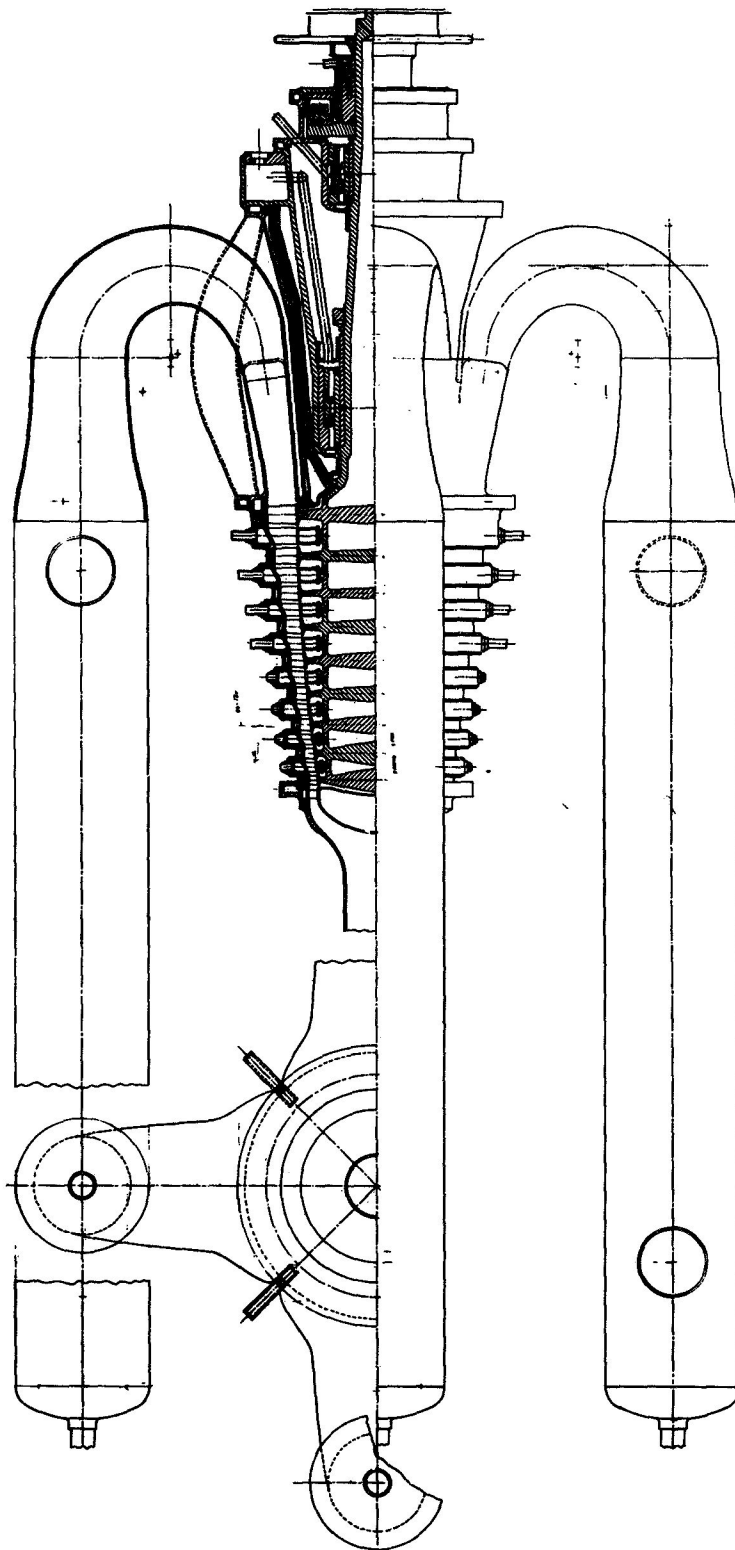


Figure 6.3-8. Quadfurcated Turbine Exit.

TABLE 6.3-1

SUMMARY OF EXIT LOSSES

<u>TYPE EXIT</u>	<u>TURBINE DESIGN</u>	<u>APPROACH VELOCITY, FPS</u>	<u>DIFFERENTIAL PRESSURE PRESSURE</u>	<u>PRESSURE LOSS, PSI</u>
SCROLL	SINGLE-SPOOL, OVERHUNG	578	.041	.30
QUADFURCATED	SINGLE-SPOOL, OVERHUNG	578	.021	.15
HIGH PRESSURE SCROLL	SPLIT-SPOOL, PARTIALLY OVERHUNG	331	.013	.50
HIGH PRESSURE QUADFURCATED	SPLIT-SPOOL, PARTIALLY OVERHUNG	331	.005	.21
LOW PRESSURE SCROLL	SPLIT-SPOOL, PARTIALLY OVERHUNG	463	.027	.19
LOW PRESSURE QUADFURCATED	SPLIT-SPOOL, PARTIALLY OVERHUNG	463	.012	.09
SCROLL	SPLIT FLOW	709	.060	.46
HIGH PRESSURE SCROLL	SPLIT-SPOOL, STRADDLE MOUNTED	397	.019	.62
LOW PRESSURE SCROLL	SPLIT-SPOOL, STRADDLE MOUNTED	698	.059	.45

in the wet region have been damaged by impact erosion. The erosion measured in metal loss per unit time can be shown to increase with the stage inlet moisture content and the tip speed of the rotor blades. Shown in Figure 6.4-1 on the coordinates of tip speed versus last stage inlet moisture fraction are plotted the conditions for all of the turbines studied in Phase I. The symbols for the four turbines selected for the four KTA designs are shown filled in. The split-spool turbines all lie in a column on the left-hand side (low-moisture-fraction portion) of the figure and the remainder lie in a column on the right-hand side (high-moisture-fraction portion) of the figure. The reason for this is that the split-flow turbines are provided with a vortex separator between the high- and low-pressure turbines. The selected split-spool turbines for the four KTA configurations have values at last stage moisture fraction of 9.2 and 11.1 percent while the single-spool turbines have values of 15.9 and 16.3 percent. The split-flow turbines have higher tip speeds, around 830 ft/sec, compared to about 770 for the single-spool turbines which are compensated for by the aforementioned lower moisture-fraction values for the split-flow turbines.

Shown in Figure 6.4-2 is the cumulative moisture present at the various stage inlets for a nine-stage turbine. Three cases are shown, namely, no extraction, internal extraction (effectiveness, 25 percent) and external extraction (effectiveness, 80 percent). With no extraction, the nine-stage turbine encounters a last stage inlet moisture fraction of about 16 percent whereas with either external extraction (vortex separator) or internal extraction (of as yet undefined configuration) the moisture fraction is reduced to about 9 or 9.5 percent with

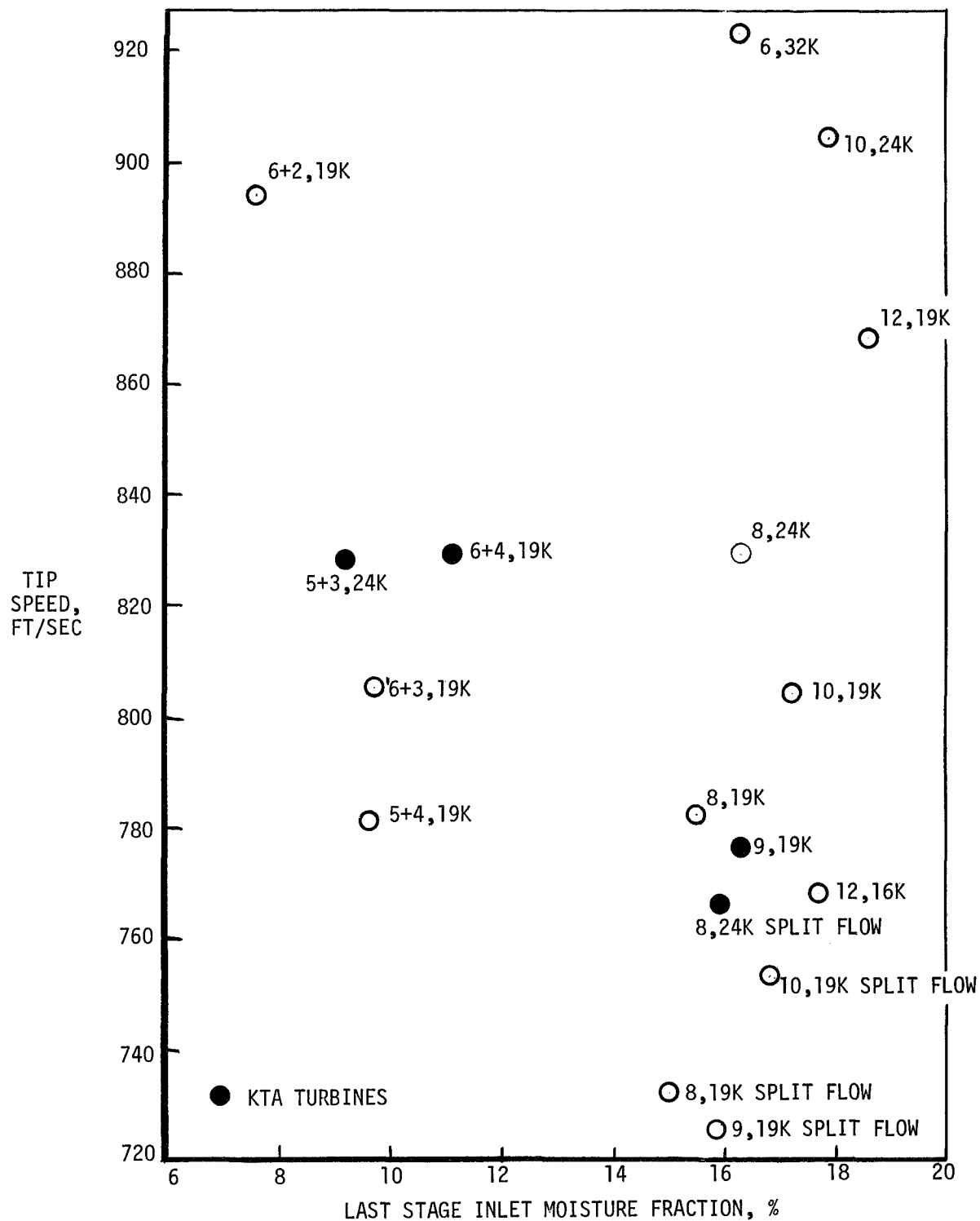


Figure 6.4-1. Severity of Erosive Conditions.

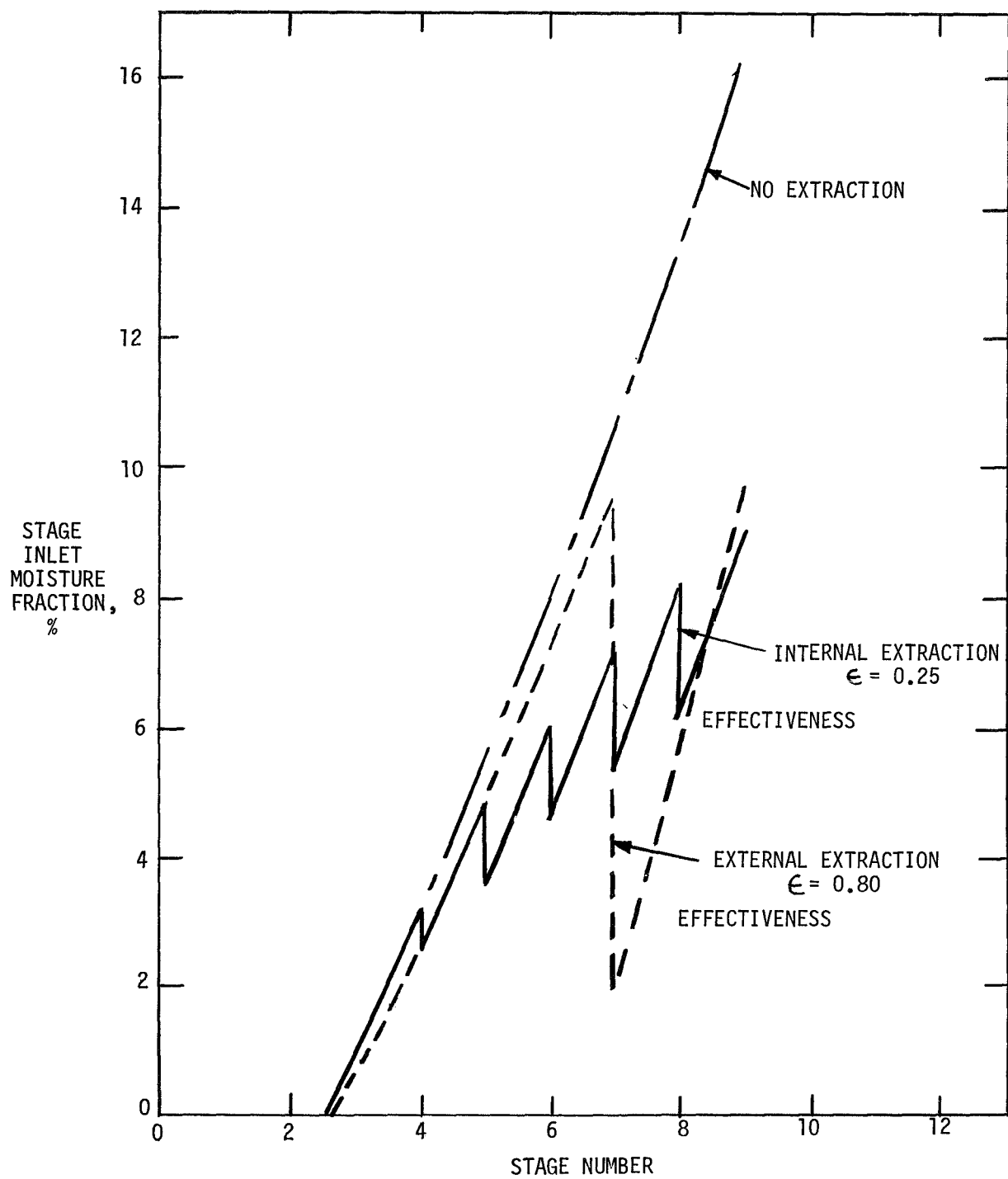


Figure 6.4-2. Cumulative Stage Moisture Present.

the estimated effectiveness values used. Although it cannot yet be stated at which level of inlet moisture fraction erosion will be negligible in a potassium vapor turbine, test data for the selected blade material (TZM) operating in potassium vapor with an inlet moisture fraction of about 9.5 percent will be obtained on the three-stage potassium vapor turbine under Contract NAS 3-10606. Since it is doubtful whether a turbine blade will run three to five years with a stage-inlet moisture fraction of 16 percent, several types of extraction devices will now be discussed.

Shown in Figure 6.4-3 is the application of rotor-blade trailing-edge moisture extraction to the ten-stage single-flow turbine on stages six through ten. Some of the moisture entering a given stage is deposited on the stator blades, forming large-size, slow-moving drops in the stator wake. Because of their low absolute velocity, the droplets impinge near the leading edge of the blade on the convex (suction) side. Some of the liquid contained in the drops splashes back into the vapor stream. The remainder remains on the blades where it is acted upon by an approximately 50,000 G force, causing near radial travel along the blade surface. The liquid accumulates under the rotating tip shroud and is dragged to the blade trailing edge by shear forces caused by the high-velocity vapor flow around the blades. At the trailing edge, the drops are slung off into the specially configured extraction device. The flow of liquid through the extraction device is aided by about one percent of vapor bled with the liquid to a lower pressure region.

Although such devices have been developed by the General Electric Company to the point where optimum dimensions exist for the devices

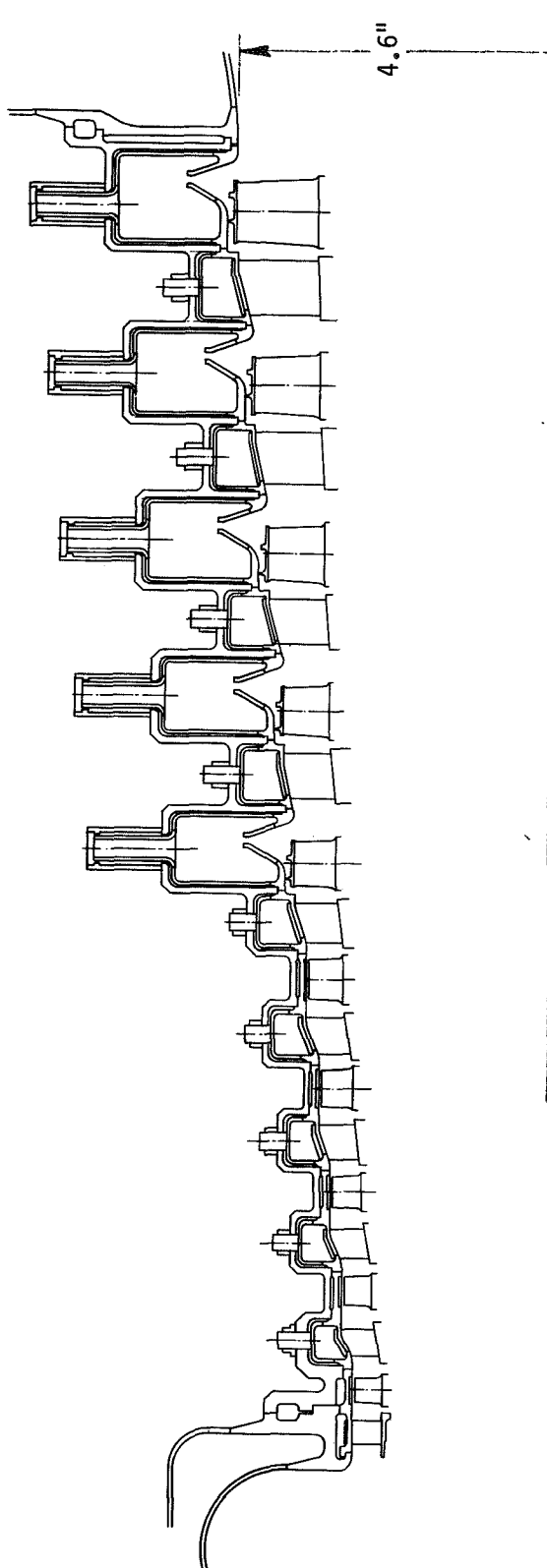


Figure 6.4-3. Rotor-Blade Internal Moisture Extraction.

and although they have been used with more or less success on a large number of central-station power plants, these types of devices have certain drawbacks. In the first place, the extraction devices complicate the KTA turbine casing because of their large radial extent beyond the flow path. In addition, the gap in each device where the liquid enters from the rotor disrupts the casing, making it necessary to carry casing loads around the extraction devices. Another disadvantage is that once the moisture enters a stage it can damage the rotor blades because it cannot be extracted without first impacting upon the rotor blades.

An alternative interstage moisture extraction device is shown in Figure 6.4-4. In this device, the moisture is removed from the stator rather than the rotor. As is apparent in the figure, the casing is not interrupted. In addition, since the moisture is removed from the stator, the moisture removed cannot do damage on the rotor. The details of the device are shown in Figure 6.4-5. The trailing edge of the blade is slotted and vented by pipes (shown in Figure 6.4-4) to a low pressure region. The pressure drop through the slot causes a small flow of vapor through the slot which entrains the liquid dragged to the trailing edge by the vapor flow around the blade. This device is just beginning to find use in central-station steam turbines. Cascade test data on such devices in air-water mixtures are reported in the Phase II portion of the report. Since the effectiveness of both rotor-blade and stator-blade extraction depend to a large extent upon the moisture deposited on the stator blade, there seems little reason to believe the stator extraction device would be any less effective than a rotor blade device on the same stage. Since it has fewer un-

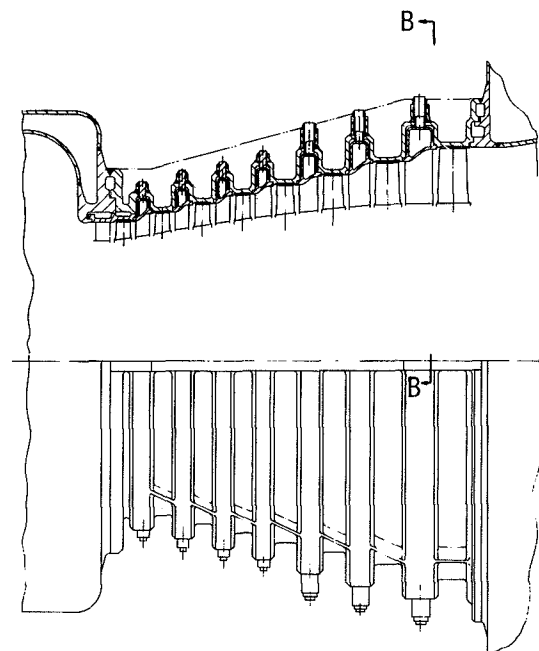
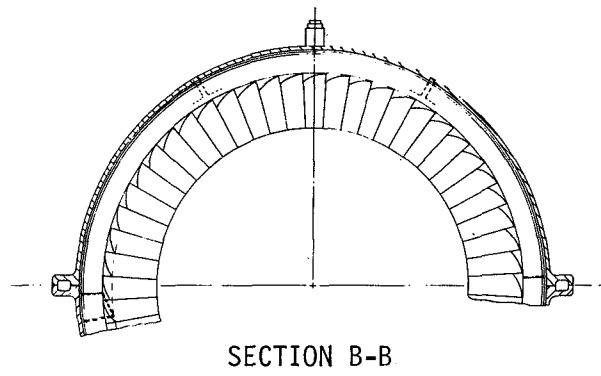


Figure 6.4-4. Internal Moisture Extraction From Stator.

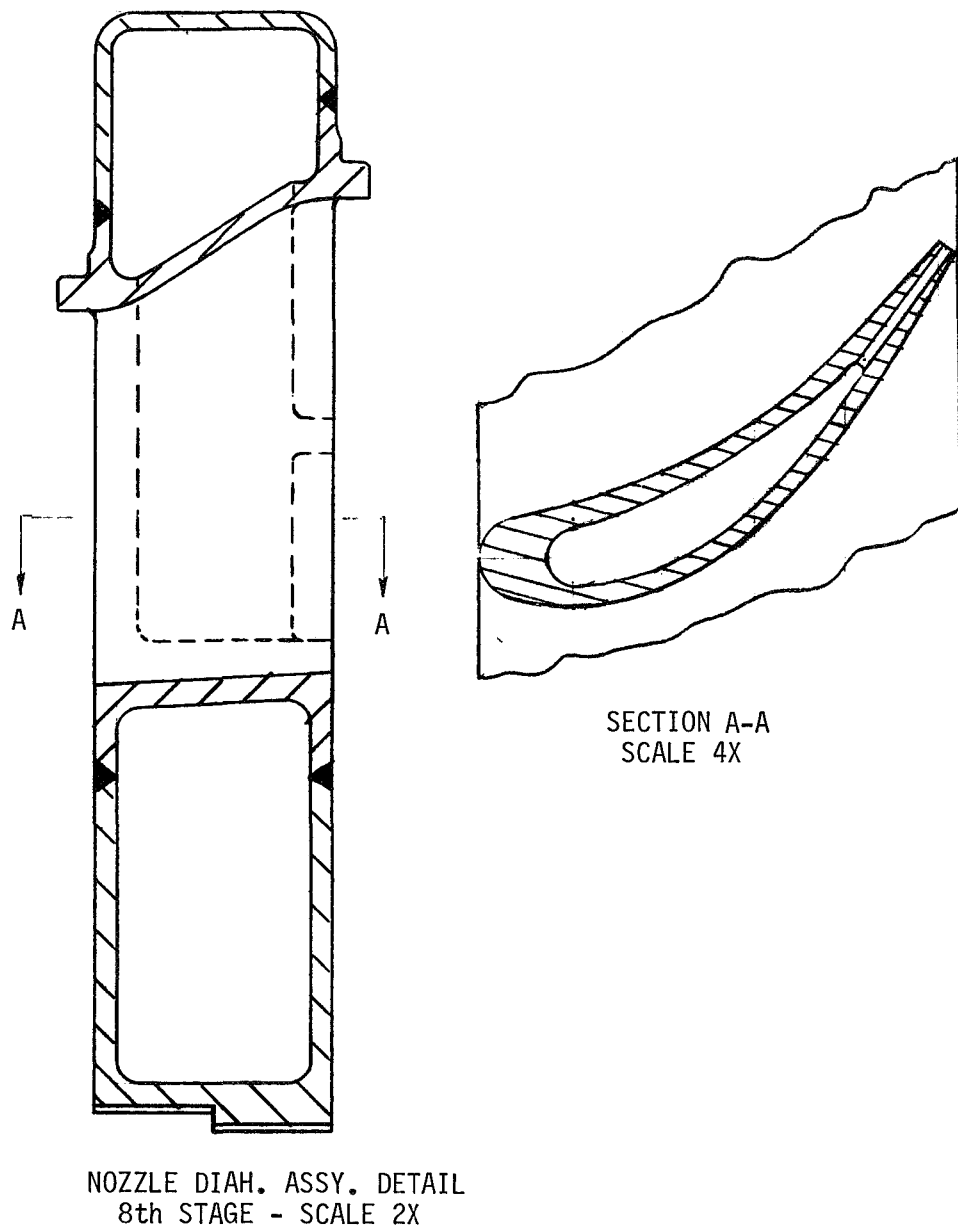


Figure 6.4-5. Details of Stator Moisture Extraction.

desirable features and at least the same effectiveness, the stator moisture extraction device seems more desirable than the rotor extraction device.

As indicated in Figure 6.4-2, depending upon the effectiveness achievable with various extraction devices, the use of an external extraction device (a vortex separator) is competitive with the use of internal extraction devices on several stages. A vortex separator for the split-flow turbines has the same internal diameter as the pipe carrying the vapor from the high to the low pressure turbine. A set of vanes at the inlet causes the wet vapor to swirl, thus throwing the liquid drops to the wall. The liquid, with a small amount of vapor, is extracted through a specially configured slot in the wall of the separator. A second set of vanes removes the swirl to minimize the pressure drop. A value of efficiency (effectiveness) of 80 percent was assumed in the KTA study.

6.5 TURBINE FLUID DYNAMIC EVALUATION

A turbine efficiency comparison of selected turbine designs for 19200 rpm, the rotative speed preferred for the alternator, is shown in Table 6.5-1. The net blading efficiency includes scroll pressure losses but does not include parasitic power losses due to bearings and seals. The highest efficiency design was not selected for the KTA because it was based on high wheel speeds and had high stresses. The next highest efficiency was for the ten-stage, split spool, partially overhung turbine and this design is the preferred one for the KTA system. The nine-stage overhung turbine design was also selected for further KTA evaluation because of its inherent simplicity, even though

TABLE 6.5-1

BLADING EFFICIENCY - 19, 200 RPM TURBINES

TYPE TURBINE DESIGN	NO. OF STAGES	TOTAL-TO-TOTAL EFFICIENCY	EXIT AXIAL VELOCITY,	NET BLADING ⁽¹⁾ EFFICIENCY
			FPS	
SINGLE-SPOOL, SINGLE FLOW	10	.818	532	.810
SINGLE-SPOOL, SINGLE FLOW, OVERHUNG	8	.775	583	.766
* SINGLE-SPOOL, SINGLE FLOW, OVERHUNG	9	.795	578	.785
SINGLE-SPOOL, SINGLE FLOW, OVERHUNG, HIGH EFFICIENCY	9	.832	402	.830
SPLIT FLOW	10 & 10	.804	531	.797
SPLIT SPOOL, STRADDLE MOUNTED	6 & 3	.811	516	.800
SPLIT SPOOL, PARTIALLY OVERHUNG	5 & 4	.786	422	.783
* SPLIT SPOOL, PARTIALLY OVERHUNG	6 & 4	.821	463	.816

* CHOSEN FOR KTA

(1) ADJUSTED FOR SCROLL LOSSES

its efficiency was not the best.

A similar comparison of three turbines designed for 24000 rpm, the selected rotative speed for an oil lubricated system, is shown in Table 6.5-2. An overhung turbine design is not feasible because a long seal is required between the oil bearing and the potassium working fluid and this increases the overhang intolerably. The other two designs were selected for evaluation in KTA systems.

The parasitic bearing and seal losses for the 19200 rpm turbines are listed in Table 6.5-3. The overhung turbines have lower labyrinth seal losses because they do not have the seal ahead of the first stage which is required on straddle mounted designs. The split flow design has relatively high labyrinth seal losses because it is essentially two small turbines, with seal losses in each. The journal bearing losses are small for straddle mounted designs and greater for overhung designs. The split spool, partially overhung turbines have the smallest total bearing and seal losses. For the 24000 rpm turbine design, the total losses in Table 6.5-4 are almost identical with some differences in the various components.

Taking the parasitic bearing and seal losses into account, the turbine shaft efficiencies are shown in Tables 6.5-5 and 6.5-6. The split spool, partially overhung ten stage turbine has a greater margin in performance over its competitors at 19200 rpm because it had lower parasitic losses.

TABLE 6.5-2

BLADING EFFICIENCY - 24,000 RPM TURBINES

<u>TYPE TURBINE DESIGN</u>	<u>NO. OF STAGES</u>	<u>TOTAL-TO-TOTAL EFFICIENCY</u>	<u>EXIT AXIAL VELOCITY, FPS</u>	<u>NET BLADING EFFICIENCY (1)</u>
SINGLE SPOOL, SINGLE FLOW OVERHUNG	8	.797	715	.780
* SPLIT FLOW	8	.783	709	.768
* SPLIT SPOOL, STRADDLE MOUNTED	8	.793	698	.779

* CHOSEN FOR KTA

(1) ADJUSTED FOR SCROLL LOSSES

TABLE 6.5-3

TURBINE BEARING AND SEAL LOSSES - 19,200 RPM TURBINES

TYPE TURBINE DESIGN	NO. OF STAGES	THRUST, LBS	LOSSES, KW				TOTAL
			LABYRINTH SEAL	JOURNAL BEARING	THRUST BEARING	SCREW SEAL	
SINGLE-SPOOL, SINGLE FLOW	10	1590	26.0	1.4	6.4	0.5	34.3
SINGLE-SPOOL, SINGLE FLOW, OVERHUNG	8	2532	9.0	5.0	9.4	1.0	24.4
SINGLE-SPOOL, SINGLE FLOW, OVERHUNG	9	2580	9.4	5.0	10.0	1.0	25.4
SINGLE-SPOOL, SINGLE FLOW, OVERHUNG, HIGH EFFICIENCY	9	--	--	--	--	--	--
SPLIT FLOW	10 & 10	0	19.4	1.4	5.8	0.4	27.0
SPLIT-SPOOL, STRADDLE MOUNTED	6 & 3	240	22.1	1.3	2.9	0.5	26.8
SPLIT-SPOOL, PARTIALLY OVERHUNG	5 & 4	1663	7.4	4.9	7.9	1.0	21.2
SPLIT-SPOOL, PARTIALLY OVERHUNG	6 & 4	1525	8.9	5.2	8.1	1.0	23.2

TABLE 6.5-4

TURBINE BEARING AND SEAL LOSSES - 24,000 RPM TURBINE

<u>TYPE TURBINE DESIGN</u>	<u>NO. OF STAGES</u>	<u>THRUST, LBS</u>	<u>LOSSES, KW</u>				<u>TOTAL</u>
			<u>LABYRINTH SEAL</u>	<u>JOURNAL BEARING</u>	<u>THRUST BEARING</u>	<u>SCREW SEAL</u>	
SINGLE-SPOOL, SINGLE FLOW OVERHUNG	8	2098	9.1	7.8	8.4	1.3	26.6
SPLIT FLOW	8	0	17.2	4.2	3.9	1.3	26.6
SPLIT-SPOOL, STRADDLE MOUNTED	8	192	21.4	0.7	2.8	1.3	26.2

TABLE 6.5-5

TURBINE SHAFT EFFICIENCY - 19,200 RPM TURBINES

<u>TYPE TURBINE DESIGN</u>	<u>NO. OF STAGES</u>	<u>NET BLADING BEARING AND SEAL EFFICIENCY</u>	<u>LOSSES, KW</u>	<u>SHAFT EFFICIENCY</u>
SINGLE-SPOOL, SINGLE FLOW	10	.810	34.3	.760
SINGLE-SPOOL, SINGLE FLOW, OVERHUNG	8	.766	24.4	.730
SINGLE-SPOOL, SINGLE FLOW, OVERHUNG	9	.785	25.4	.748
SINGLE-SPOOL, SINGLE FLOW, OVERHUNG, HIGH EFFICIENCY	9	.830	--	--
SPLIT FLOW	10 & 10	.797	27.0	.757
SPLIT-SPOOL, STRADDLE MOUNTED	6 & 3	.800	26.8	.761
SPLIT-SPOOL, PARTIALLY OVERHUNG	5 & 4	.783	21.2	.752
SPLIT-SPOOL, PARTIALLY OVERHUNG	6 & 4	.816	23.2	.783

TABLE 6.5-6

TURBINE SHAFT EFFICIENCY - 24,000 RPM TURBINES

<u>TYPE TURBINE DESIGN</u>	<u>NO. OF STAGES</u>	<u>NET BLADING EFFICIENCY</u>	<u>BEARING AND SEAL LOSSES, KW</u>	<u>SHAFT EFFICIENCY</u>
SINGLE-SPOOL, SINGLE FLOW OVERHUNG	8	.780	26.6	.742
SPLIT FLOW	8	.768	26.6	.730
SPLIT-SPOOL, STRADDLE MOUNTED	8	.779	26.2	.741

6.6 REFERENCES

- 6.2-1. R.W. Bassett: "Pressure Loss Tests of a Model of a Turbine Volute." National Research Council of Canada MET 38, August 1961.
- 6.3-1. J.R. Henry, C.C. Wood and S.W. Wilbur: "Summary of Subsonic Diffuser Data", NACA RM L 56F05, October 1956.

7.0 BEARING/ROTOR RESPONSE OF TURBINES AND ALTERNATORS

Rotor dynamic analyses were performed on most of the turbines of interest and on several variations of alternator arrangements. The more significant results of the most attractive machine concepts (both potassium and oil lubricated) are discussed in this section of the report. These analyses included both undamped critical speed and rotor-bearing response to unbalance which rigorously included bearing damping.

Generally, the potassium lubricated turbines were, from a dynamic standpoint, superior to the oil lubricated turbines because they were not troubled by the rotor weakening effects of a long complex seal arrangement. Also, turbines with overhung stages behaved better than straddle mounted arrangements because their non-uniform mass distribution resulted in a desirable location of vibration nodes which made it easier to realize the full damping potential of the bearings.

It was more difficult to achieve the same degree of mild vibrational behavior with the alternators than was possible with the turbines because of the heavy rotor masses that had to be supported on relatively compliant stub shafts. The work accomplished in Phase I, however, did point the way toward the refinements that would be made in Phase II which would significantly improve the alternator dynamic behavior.

7.1 MECHANISM OF VIBRATIONS

The primary exciting force for synchronous vibration, which is usually the most troublesome high speed machine vibrational problem, is rotor unbalance.* Because of the influence these analyses had on the designs of the machines, the background and significance of the analyses will be discussed briefly.

The critical speed of a rotor is a resonant condition in which either the rotor mass-bearing spring stiffness combination is resonant or the shaft itself having both mass and elasticity is resonant. The rotating speed corresponding to this resonance is called the critical speed. At this speed, amplitudes of vibration, excited by rotating unbalance, can become very large. If poorly damped, such large amplitudes of rotor vibration lead to seal rubs, bearing rubs, and in extreme cases, failure of the shaft. The usual way of representing the shaft critical speeds is to plot them as a function of bearing or support stiffness as is typically shown in Figure 7.1-1. The third critical is not influenced by the bearings at low values of bearing stiffness as it is a bending critical controlled entirely by shaft elasticity. (With zero bearing stiffness, the third critical is the classical "Free-Free" beam bending mode). At the right side of the map where the bearing stiffnesses are high the critical speed curves level off and the critical speeds are independent of bearing stiffness

* The common problem of fractional frequency vibration in the KTA components has been minimized by the use of tilting pad journal bearings which are free of hydrodynamic instabilities causing fractional frequency whirl. The possibility of fractional frequency instabilities resulting from out-of-phase aerodynamic forces are not regarded as likely, and since beyond the scope of this study, have not been considered.

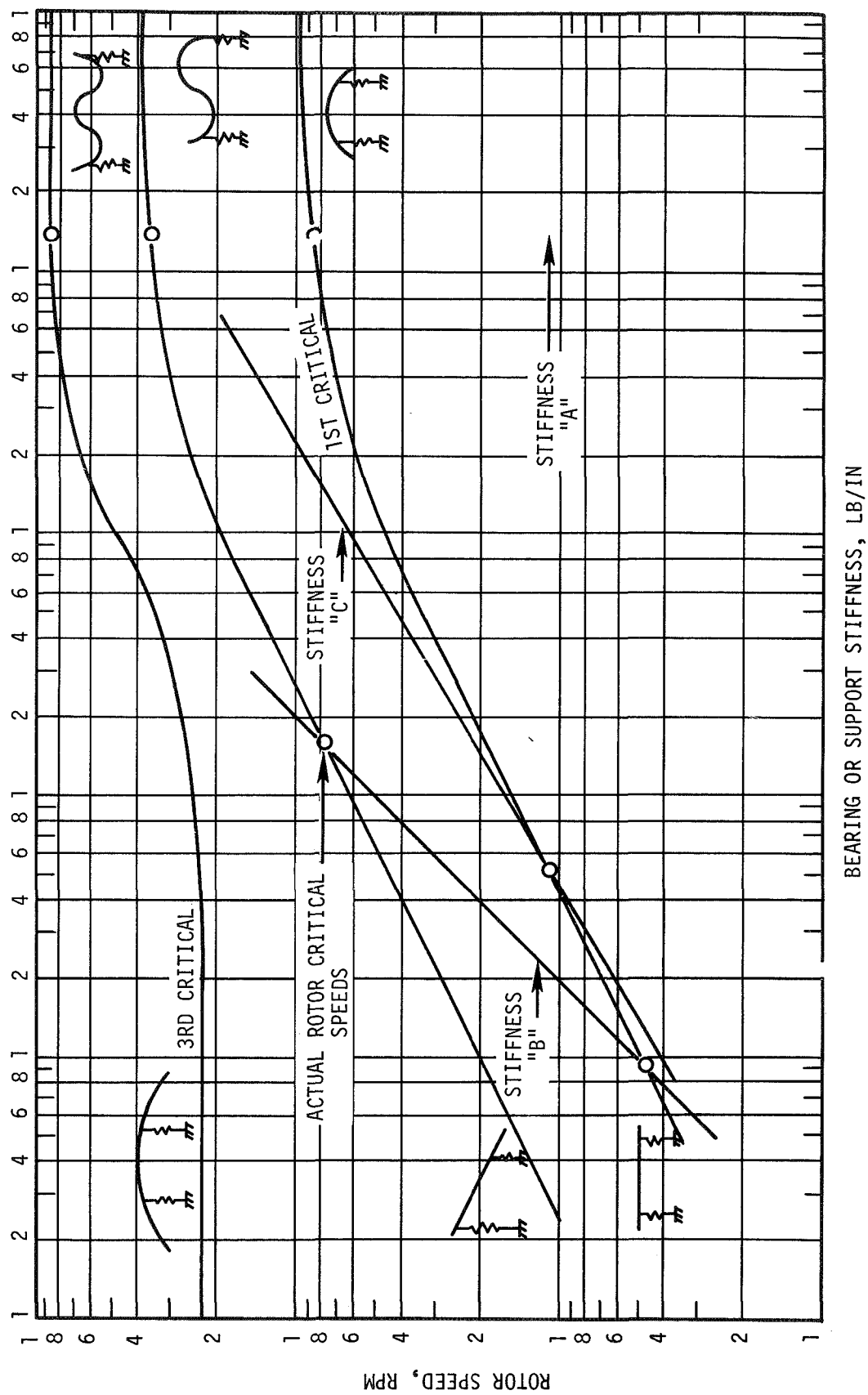


Figure 7.1-1. Typical Critical Speed Map.

and depend only upon the flexibility of the rotor. These are also seen by the diagrams associated with each critical on Figure 7.1-1.

For the purpose of discussion, three typical bearing stiffness curves have been plotted on Figure 7.1-1 to illustrate the effect of different bearing types on the vibrational behavior of the same rotor. Curve "A" represents a typical rolling element bearing, "B" a hydrodynamic laminar gas or liquid bearing and "C" a hydrodynamic turbulent liquid bearing. Machines mounted on ball or other rolling element bearings have very pronounced and usually violent critical speeds. The reason for this is clearly shown on Figure 7.1-1, since the rolling element stiffness is usually quite high and independent of speed. The first critical is a bending critical with the shaft nodal points located at or very near the bearing. Since the rolling element bearing possesses little or no damping, large amplitudes of vibration at resonance can be expected. For this reason, most ball bearing machines are designed to operate below the first critical or have the ability to accelerate through them very quickly to avoid the dangerous resonance condition.

At the other extreme are hydrodynamic laminar bearings which may be either gas or liquid lubricated. These bearing types are generally softer than rolling element bearings and, because their load capacity and stiffness are a function of the rate of fluid film shear, the stiffness is a function of speed. Rotors mounted on laminar hydrodynamic bearings usually experience critical speeds at relatively low values of rotative speed and in the range where the resonance is the type where the rotor mass is bounding on the compliant bearing film. These are commonly called "Rigid Body" criticals because little or no

shaft bending occurs. The vibration amplitudes associated with the first two rigid body critical speeds are usually small and well controlled for three basic reasons. First, they occur at low speeds where the centrifugal force resulting from unbalance is small (this driving force is a function of speed squared) and, secondly, since little or no shaft bending occurs, the mass shift which produces large centrifugal forces is minimized. Finally, and of great importance to KTA type of machinery is the fact that most of the vibrational motion can be made to take place in the bearings which, because of the squeezing motion of the fluid film, results in large damping forces. These damping forces can greatly attenuate the resonance and in some cases completely obliterate any evidence of a theoretical critical speed. This important aspect will be discussed in more detail later.

Hydrodynamic bearings in which the lubricant flow is turbulent, as is the case with the KTA, have a stiffness-speed characteristic much like a laminar bearing except that the stiffness increases with speed at a much higher rate. This is because of the increased rate of fluid film shear with increasing turbulence. Because of this, the bearing stiffness curve tends to be more parallel to the critical speed curve (see Figure 7.1-1), and the rotor has a much wider range of speeds in which it is in close proximity to the first two rigid body speeds. This, however, usually presents little problem because of the large damping afforded by the turbulent bearing.

As has been indicated above, the problem of operating through the first two critical speeds in the area where the shaft is relatively rigid is usually a mild one. Operating through the first two criticals in the area of high bearing stiffness where the shaft is bent is a

problem because the mass shift resulting from the bent shaft greatly increases the rotating load and because the bearings are located very close to the nodal points. Since the nodal point is by definition the point of zero amplitude, little or no damping can be developed in the bearings by squeeze action and large magnification results. Operation in the vicinity of, and through the third critical, which as pointed out above is always a bending critical, is practical but requires judicious bearing placement and bearings with good damping performance. The region of the third critical is usually avoided in gas bearing machinery because insufficient damping is afforded by the gas film, but oil and potassium bearings because of the higher viscosity, can be designed to produce the required damping forces. It should be noted, however, that the third critical, like the first and second in the area of high bearing stiffness, must be avoided because of the aforementioned nodal point problem.

The critical speed analysis, which is a computer solved problem where the rotor is treated as a series of discrete masses and springs following the method of Prohl, Reference 7.1-1, does not consider damping. Obviously, damping must be rigorously included in any vibrational analysis to fully understand the rotor dynamic behavior. This was accomplished on the KTA components discussed in this report following the procedure described in Reference 7.1-2. It should be pointed out that this procedure, together with the bearing dynamic stiffness and damping coefficients used in the analysis, has been experimentally verified on two different high speed rotors, one utilizing a low viscosity oil and the other using NaK as the bearing lubricant (References 7.1-3 and 7.1-4).

The critical speed analysis and resulting critical speed map and mode shape diagrams are, however, very useful tools to the designer in appraising potential vibration problems, establishing shaft size and stiffness requirements, and setting bearing locations. In addition, they often help in the physical understanding of the unbalance responses which include damping, insofar as identifying which of the system vibration modes are participating and what design changes may be made to reduce unbalance amplitudes.

7.2 TURBINES

Critical speed analyses were performed on several potassium and oil lubricated turbine designs. These were of various aerodynamic and mechanical configurations including single spool-single flow, single spool-split flow, and split spool of various stage numbers ranging from eight to ten. Also, the effect on critical speed of various wheel torque tube joint designs were analyzed. These included welded, bolted, and curvic coupling-tiebolt designs. Depending upon the joint configuration, various materials including ASTAR 811C, FS-85, TZM and TZC were incorporated into the designs. The most promising mechanical design of each aerodynamic configuration was subjected to a rotor response analysis in which actual vibration amplitudes driven by rotor unbalance was calculated. The results of the most promising designs studied in Phase I are presented here. These designs were selected based upon the best dynamic behavior or greatest potential for improvement and upon aerodynamic and other mechanical considerations which have previously been covered in this report.

The designs selected for the most detailed rotor dynamic analysis during Phase I included the following potassium lubricated turbines:

- (1) Ten stage, single flow, straddle-mounted, with curvic coupling tiebolt torque tube (GE Drawing No. 263E185) Figure 7.2-1.
- (2) Nine stage, single flow, overhung, with welded rotor (GE Drawing No. 47R199287) Figure 7.2-2.
- (3) Nine stage, split spool*, five overhung stages, with the overhung stages welded and the straddle-mounted stages curvic coupling - tiebolt (GE Drawing No. 47R199271) Figure 7.2-3.

And the following oil lubricated turbines:

- (1) Eight stage, split flow, straddle-mounted with a bolted torque tube (GE Drawing No. 47R199274) Figure 5.12-1.
- (2) Eight stage, split spool, straddle-mounted, with curvic coupling tiebolt (GE Drawing No. 47R199275) Figure 5.12-2.

For the purpose of the Phase I screening study, the following assumptions in the dynamic analyses were made:

- (1) The bearing sizes were those indicated on the referenced drawings. The bearings were initially sized to satisfy the loading criteria established in Chapter 5.8. In some

*The Phase I selected split spool aerodynamic design presented earlier was a ten stage with six overhung stages. Scheduling did not permit a rotor dynamic analysis of the ten stage configuration but the results of other dynamic analyses led to the conclusion that the six overhung stage version could easily be made to have a very good dynamic behavior.

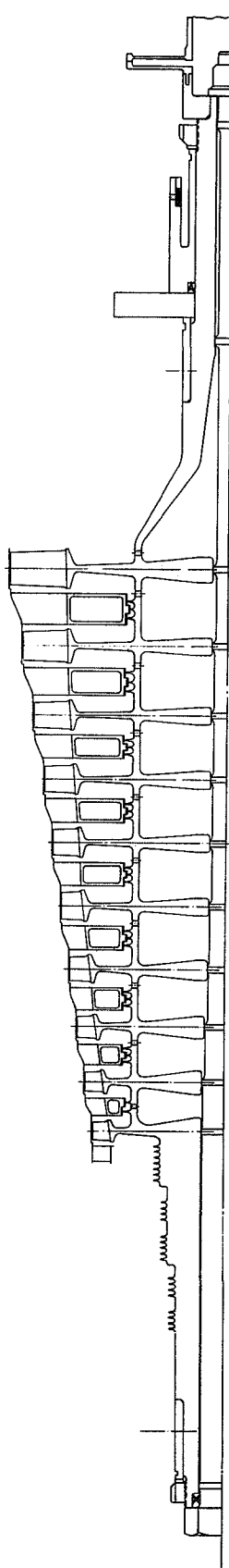


Figure 7.2-1. Ten Stage, Single Flow, Straddle Mounted Turbine with Curvic Coupling and Tiebolt, Potassium Bearings.
(Dwg. No. 263E185)

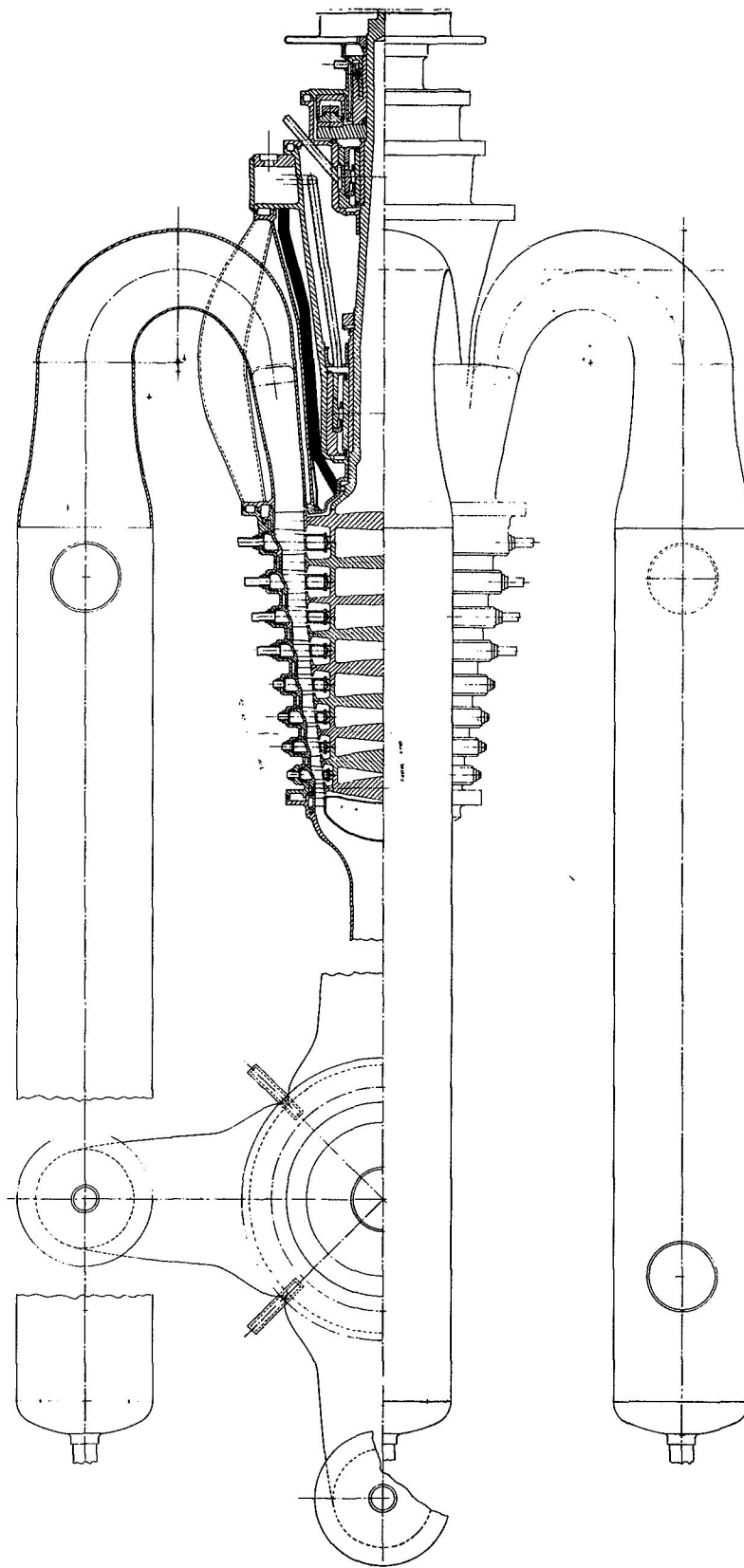


Figure 7.2-2. Nine Stage, Single Flow, Overhung Turbine with Welded Rotor, Potassium Bearings. (Dwg. No. 47R199287)

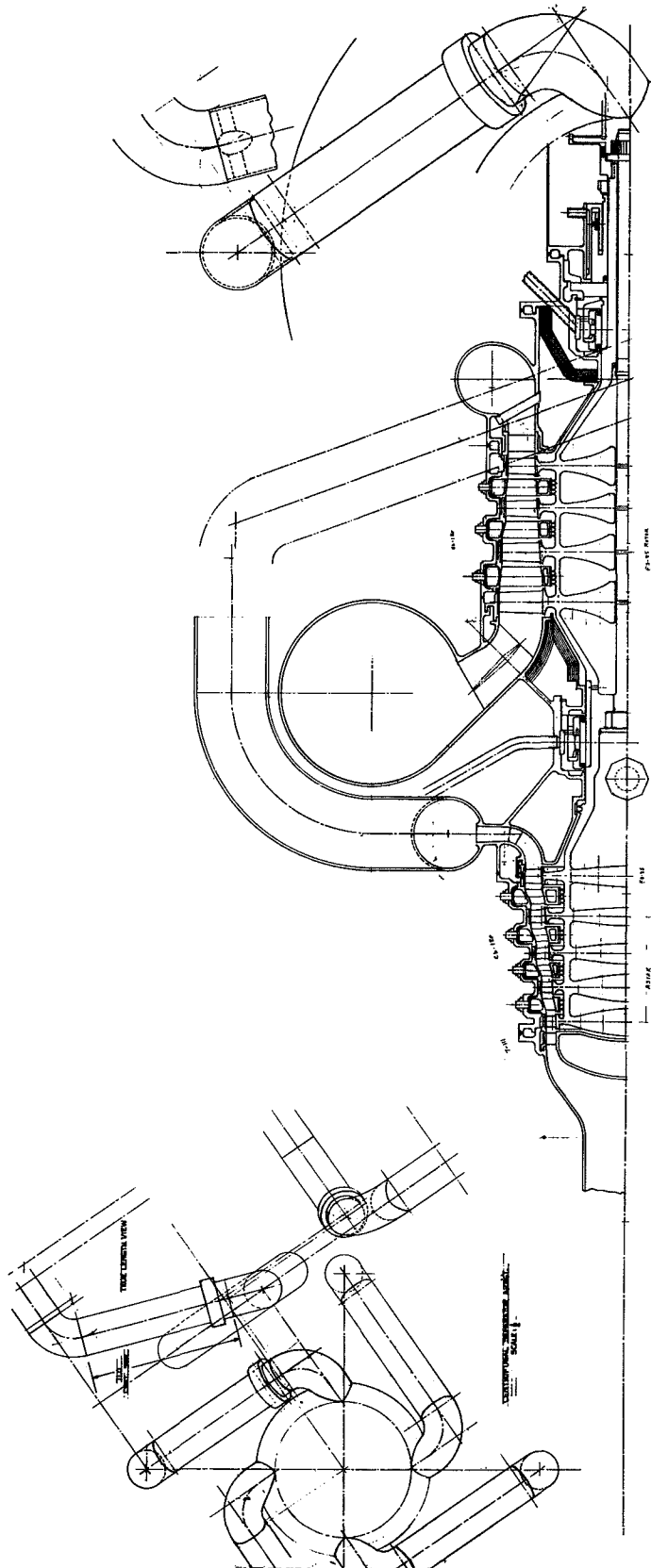


Figure 7.2-3. Nine Stage, Split Spool Partially Overhung Turbine with Potassium Bearings. (Dwg. No. 47R199271)

cases the bearings were increased in size to improve the rotor dynamic characteristics or for other mechanical design considerations. Generally, bearings with a length to diameter ratio of unity were used because this length gives a high ratio of damping to critical damping which results in good vibration attenuation.

- (2) The dynamic characteristics (stiffness and damping) were calculated assuming that the bearings were four pad tilting pad type with an assembled clearance ratio of .0015 in/in and with a preload coefficient of 0.3. Vertical operation was assumed ($\epsilon = 0$) and depending upon the machine, the physical properties of potassium at 950⁰F or 5P4E at 400⁰F was used as the lubricant.
- (3) A rotating unbalance of .010 in-oz was applied in each of two planes. These planes corresponded to the location of the spool end stages and the location of the unbalance in the two planes was oriented 180⁰ out of phase with each other. This orientation of unbalance produces the largest couple and amplifies the conical vibration mode which tended to be the most troublesome one for most of the machine configurations studied. The value of unbalance (.010 in-oz per plane) corresponds roughly to the commonly accepted balancing criteria, Reference 7.1-2.

$$\frac{4W}{N} = \text{in-oz}$$

Where: W = Rotor weight in pounds
N = RPM

Potassium Lubricated Turbines

The critical speed maps of the potassium lubricated turbines are shown on Figures 7.2-4, 7.2-5, and 7.2-6. Examination of Figures 7.2-4 and 7.2-5 shows that the ten stage straddle mounted turbine and the nine stage overhung turbines operate at design speed (19,200 rpm) very comfortably between the second and third critical speeds with the second undamped critical speeds occurring at approximately 12,000 rpm and 7,500 rpm respectively for the two turbines.

The results of the response to unbalance loads for these two turbines with .020 in-oz total unbalance are shown on Figures 7.2-7 and 7.2-8. The speed at which the critical speed analysis predicted the second critical is shown on both figures and the absence of any amplitude peaks at these speeds is typical of a highly damped rotor system. It will be noticed that the straddle mounted turbine exhibits increasing amplitudes as the speed is increased over the second critical (see Figure 7.2-7) whereas the overhung version does not exhibit this characteristic. The reason for this is that the straddle mounted turbine has a relatively uniform mass distribution with each bearing about equally loaded. This results in a very strong conical mode of vibration at speeds in excess of the second critical with the nodal point lying between bearings. With out of phase unbalance applied near the rotor ends, this conical mode is accentuated. The nine stage turbine, on the other hand, has a very non-uniform mass distribution which causes the nodal point to fall outside of the confines of the bearing and a less pronounced conical mode. As a result, out-of-phase unbalance does not accentuate the second critical mode to the degree that it does in straddle-mounted arrangement.

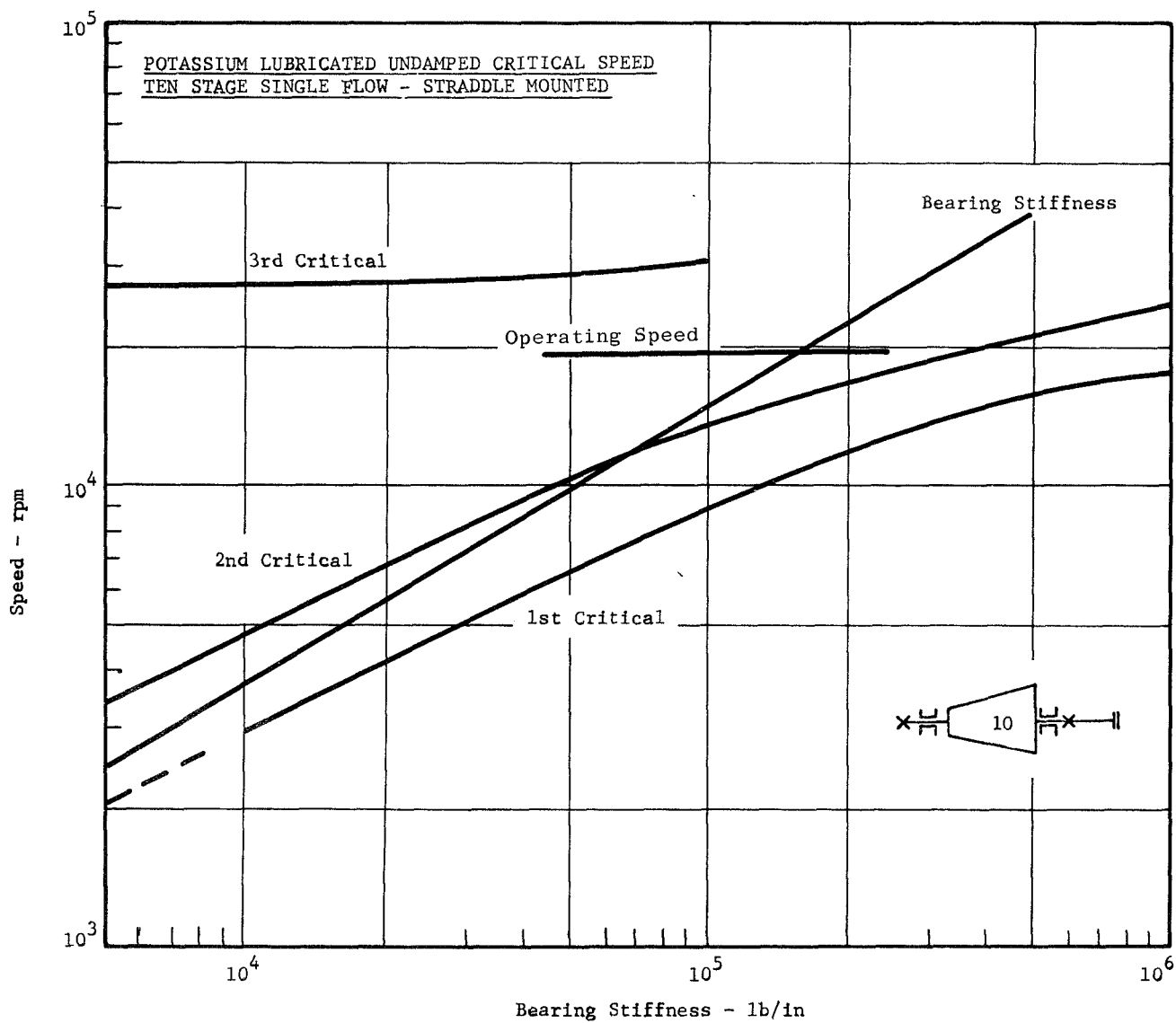


Figure 7.2-4. Potassium Lubricated Undamped Critical Speed Ten Stage Single Flow - Straddle Mounted.

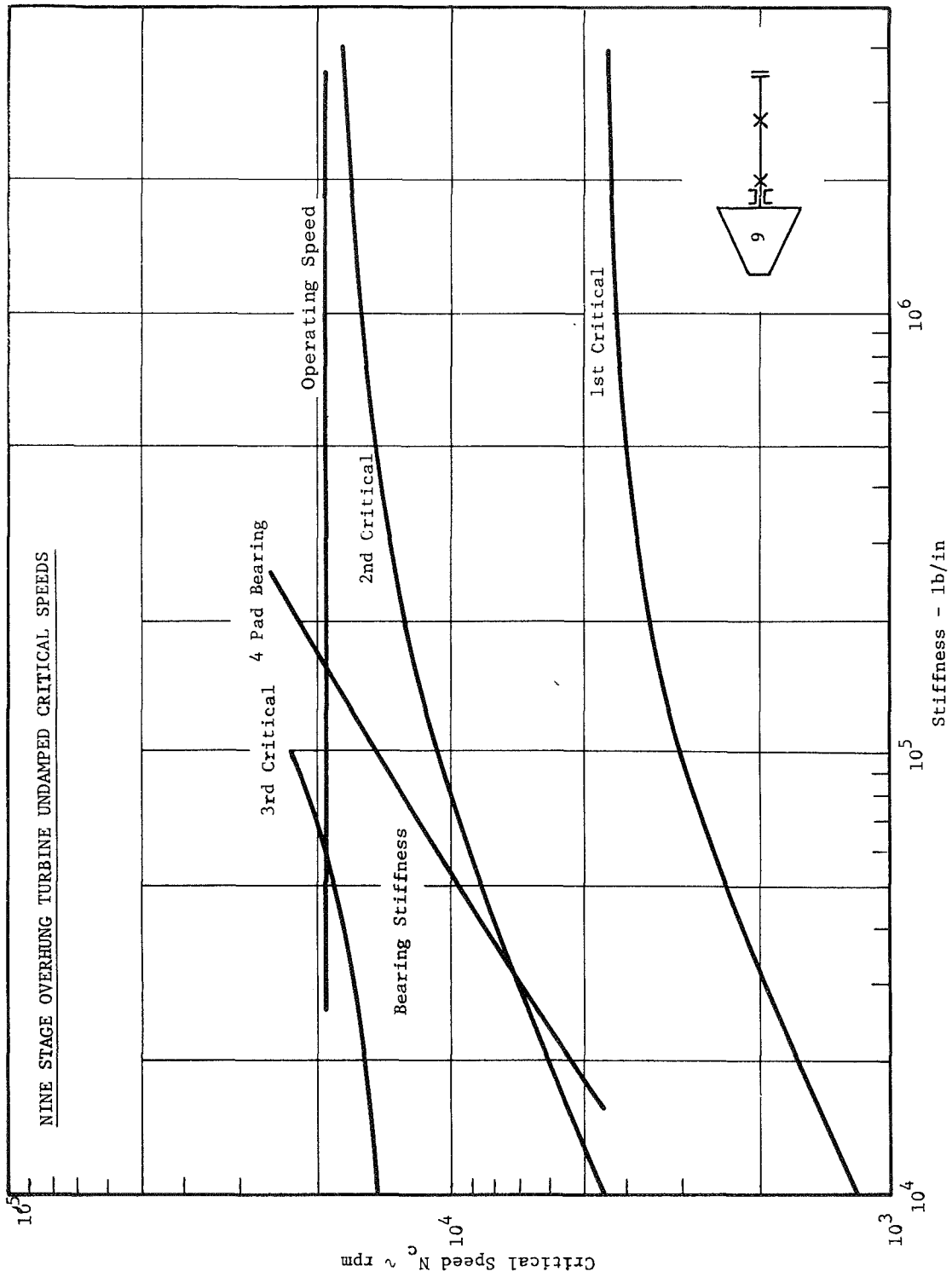


Figure 7.2-5. Nine Stage Overhung Turbine Undamped Critical Speeds.

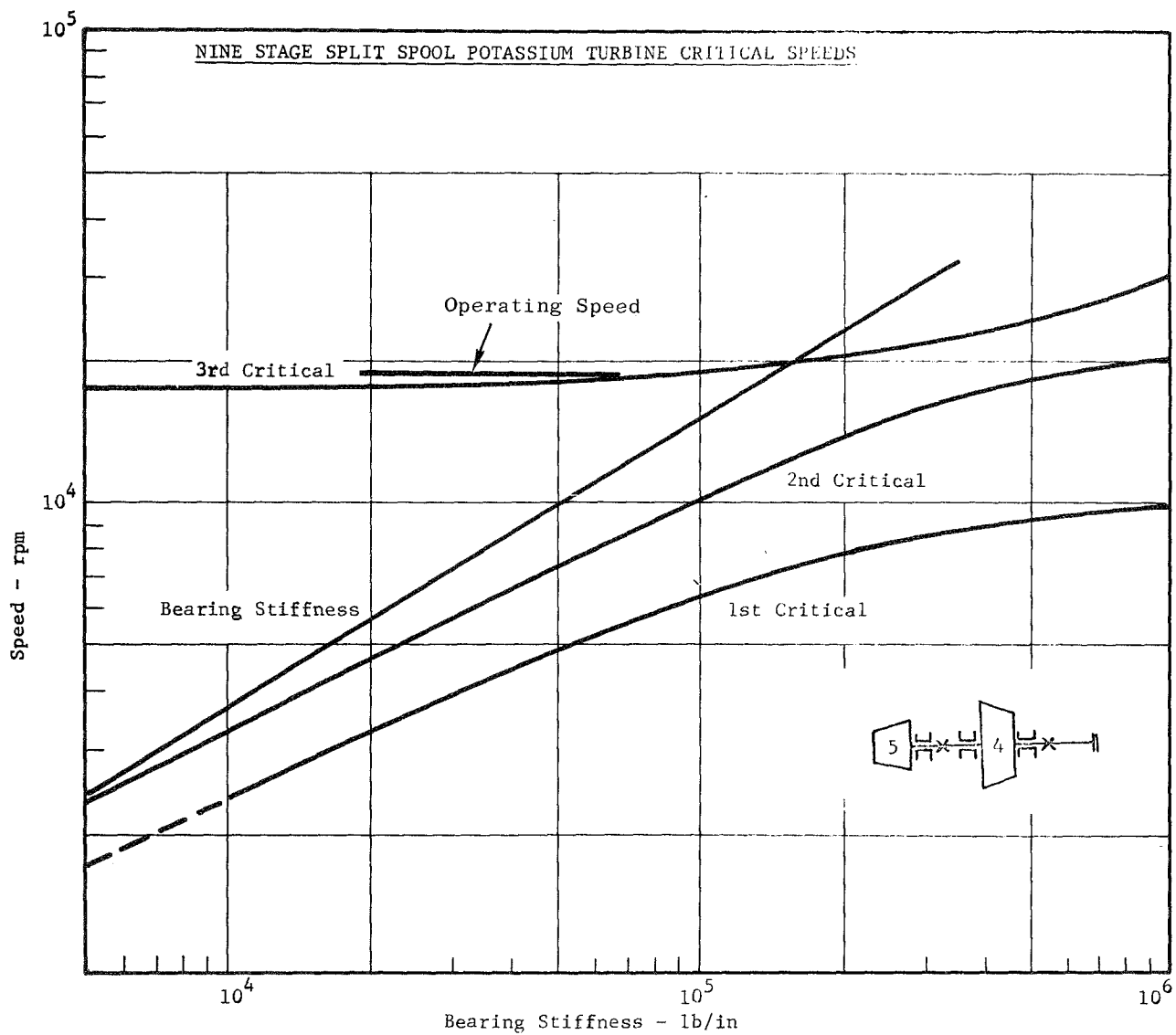


Figure 7.2-6. Nine Stage Split Spool Potassium Turbine Critical Speeds.

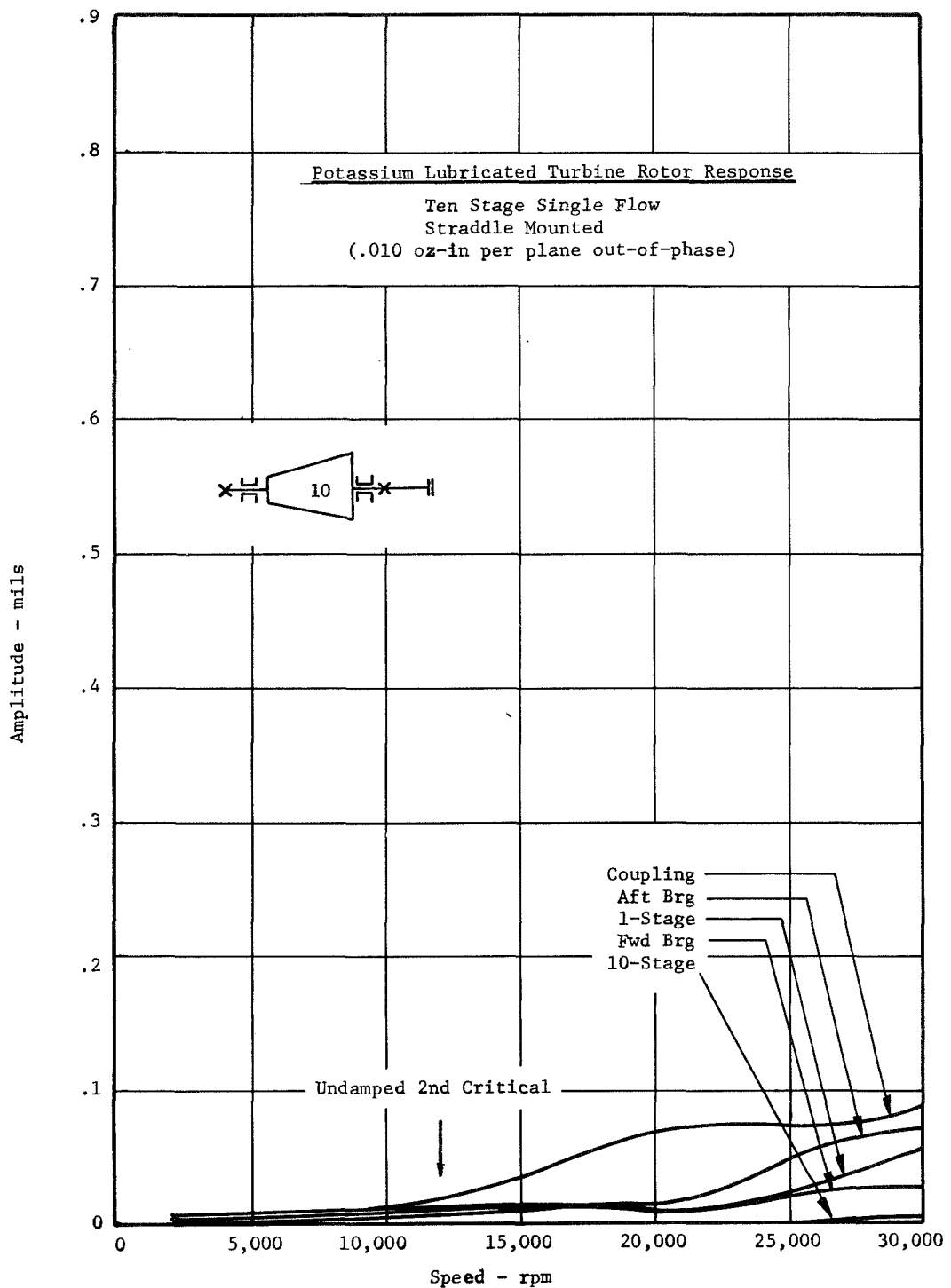


Figure 7.2-7. Potassium Lubricated Turbine Rotor Response.

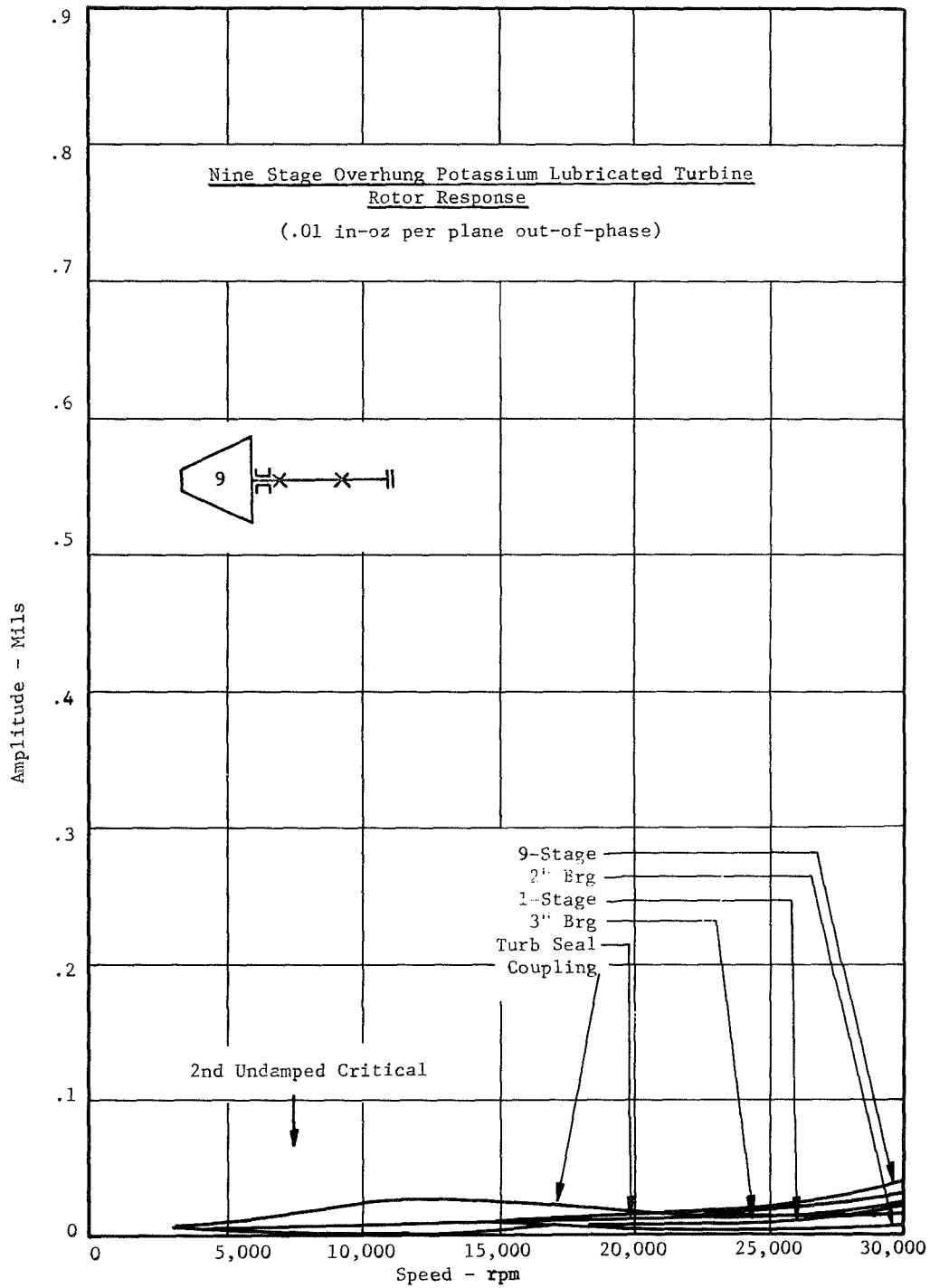


Figure 7.2-8. Nine Stage Overhung Potassium Lubricated Turbine Rotor Response.

Although it would appear on the surface that the straddle-mounted turbine arrangement would, from a dynamic standpoint, behave better than an overhung arrangement, there are factors other than that previously mentioned which tend to favor the approach where at least some of the stages are overhung. Assuming that at least one large diameter bearing can be incorporated as was done in the non-straddle-mounted turbines studied, the center of the shaft can be made very stiff. The absence of a bearing at the inlet end of the shaft leads to a situation where that end of the shaft is effectively very much stiffer in the overhung case than the straddle-mounted case. This is because the first few stage torque tubes, which are the most compliant because of their high temperature and small diameter, are required to carry no bearing reaction in the overhung case and because the overhung versions required no high pressure seal which also introduces significant compliance into the straddle-mounted turbines. Furthermore, in those cases where operation is close to the third critical, it is much more difficult to remove the bearings from the nodes to improve system damping in the straddle-mounted case because of the inherent relative weakness (compliance) of the rotor stub shafts.

Referring to the critical speed map of the nine stage split spool turbine shown on Figure 7.2-6, it will be noted that the calculated third undamped critical speed occurs at approximately 20,000 rpm. The amplitudes of vibration resulting from the two .010 in-oz phase unbalance forces are, however, quite small as is shown on Figure 7.2-9 and are actually somewhat smaller at speeds in excess of 20,000 rpm than the ten stage straddle-mounted turbine. Although the amplitude of vibration at the coupling end of the shaft does show a pronouncedment in the vicinity of the third critical, the absence of a sharp

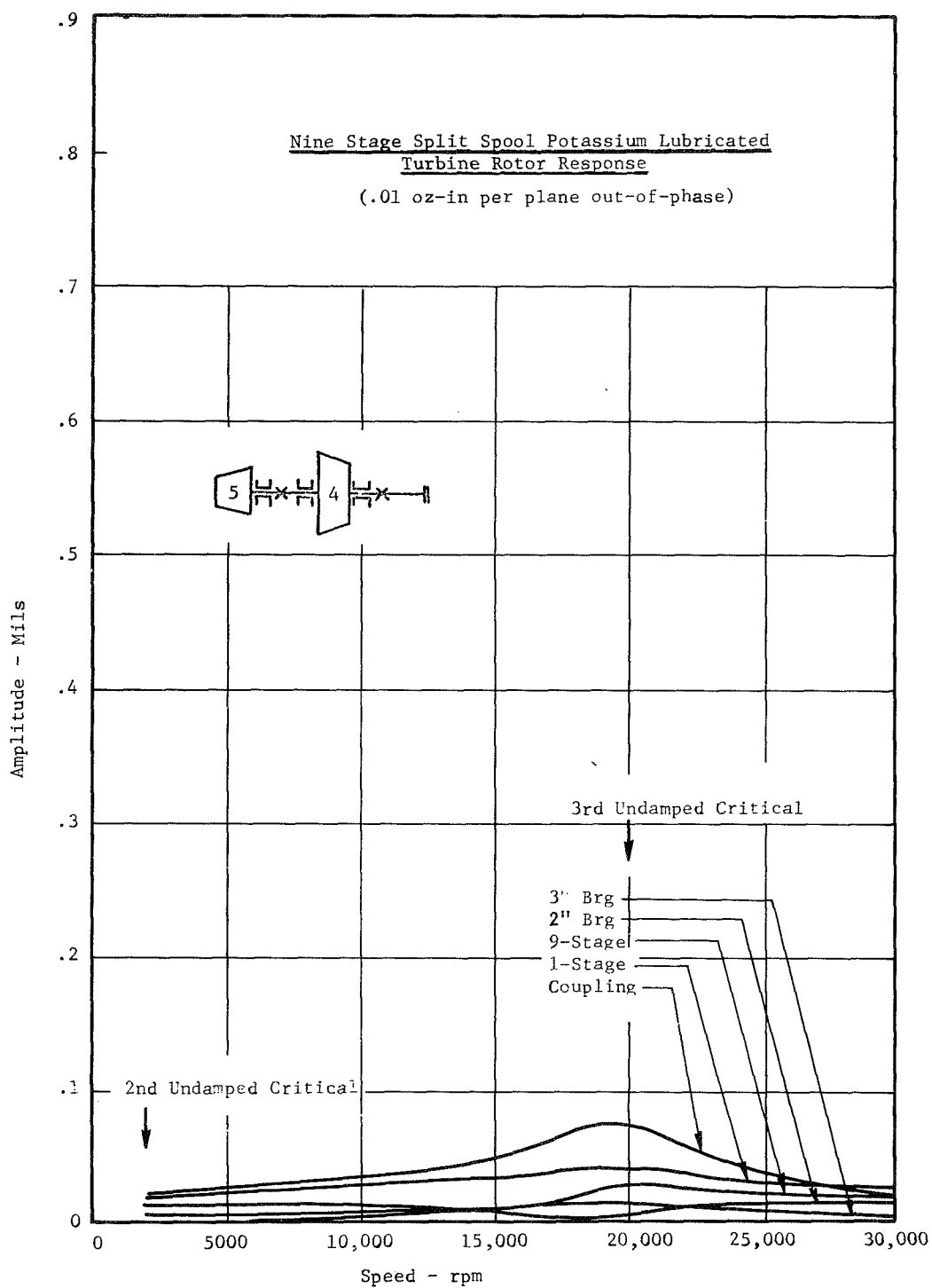


Figure 7.2-9. Nine Stage Split Spool Potassium Lubricated Turbine Rotor Response.

peak demonstrates the significance of bearing damping. The third critical speed mode shape for this split spool turbine is shown on Figure 7.2-10 and it shows quite clearly what can be done to further reduce the vibration amplitudes in the vicinity of the third critical. It will be noted that one of the nodal points falls extremely close to the coupling end bearing and because of the inherent absence of radial motion at that point, bearing damping which results from radial squeeze motions is minimal. This absence of motion at the 2.0 inch bearing can also be seen on the response plot on Figure 7.2-9. Almost all of the system damping in this case results from the 3.0 inch diameter bearing which is far displaced from both nodes. By simply moving the two-inch bearing to the right (toward the coupling), much more system damping would result with little or no change in the frequency of the undamped critical speed (it can be seen from Figure 7.2-6 that bearing stiffness up to about 5×10^5 lb/in has little effect on the third critical, thus bearing location also has little effect).

Oil Lubricated Turbines

The two oil lubricated turbines selected on the basis of the Phase I aerodynamic and alternator studies were the split flow and split spool arrangements. Both of these have eight stages with a design speed of 24,000 rpm and are straddle-mounted. Straddle-mounting of these aerodynamic arrangements is preferred because it puts the bearings and seals at the lowest temperature areas of the rotor, which of course, is the most desirable bearing location when using temperature limited lubricants.

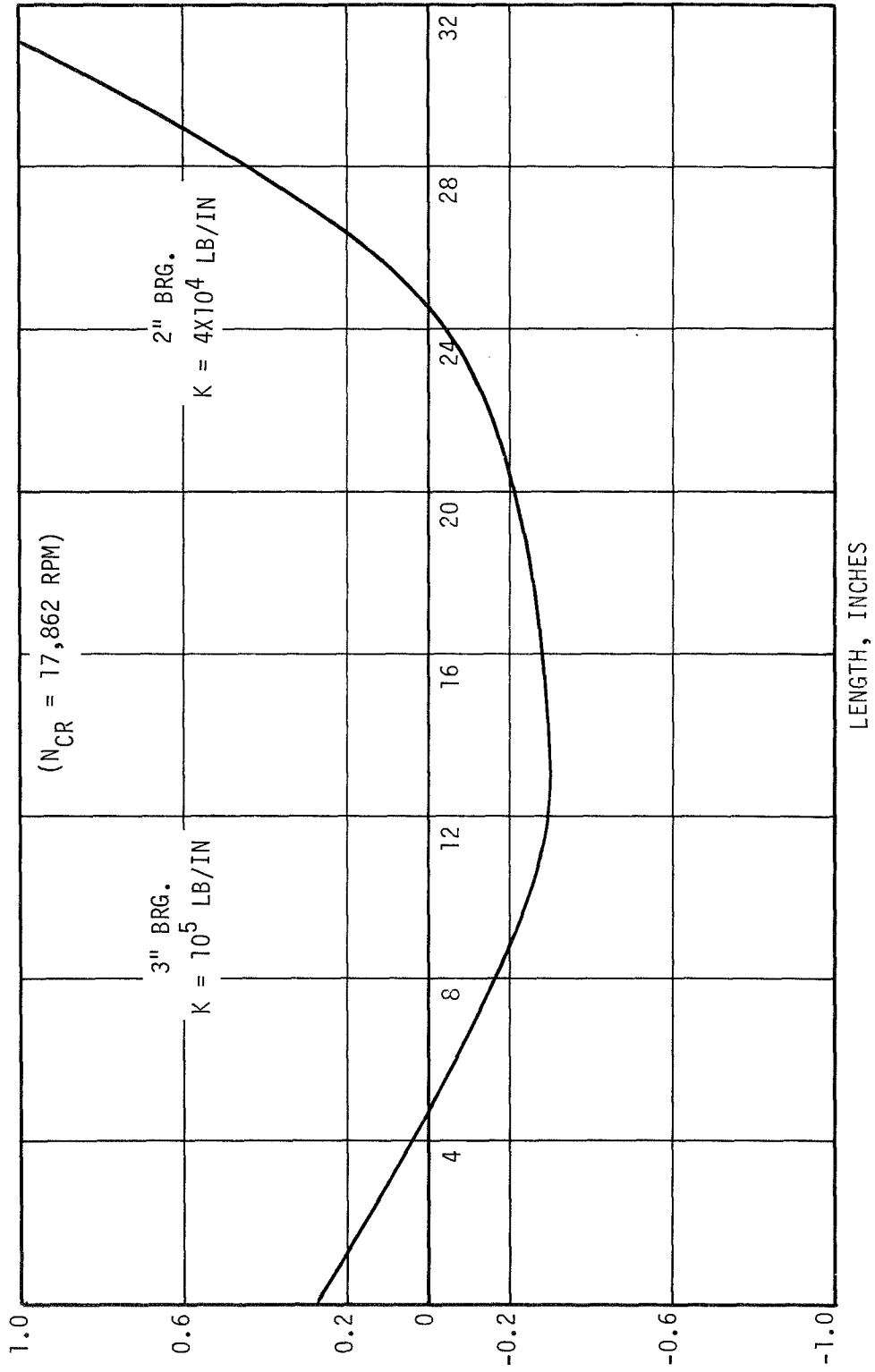


Figure 7.2-10. Nine Stage Split Spool Turbine 3rd Critical Speed Mode Shape.

The oil lubricated turbines have a much larger bearing power consumption than do the potassium lubricated turbines (assuming comparable bearing sizes) because of the formers higher speed and lubricant viscosity (see Figure 5.8-3 and 5.8-7). Furthermore, seal losses are higher for the same reason and increased even further because of the need for double seals to separate the organic lubricant from the potassium. It was, therefore, desirable to maintain the smallest possible bearing diameter in the oil lubricated machines. The bearings for these turbines were set with a 1.5 inch diameter and a length to diameter ratio of 0.5 which results in more than adequate load capacity (see Figure 5.8-4.)

Considering first the split flow turbine whose critical speed map is shown on Figure 7.2-11 design speed operation is well above the third critical speed. However, at design speed, the third and fourth criticals are well separated (approximately 15,000 rpm and 32,000 rpm respectively) and safe operation at 24,000 rpm could be expected. The problem, however, is in accelerating through the third critical (which, as has been discussed before, is a bending critical) if the full damping potential of the bearings cannot be realized. The results of the response analysis shown on Figure 7.2-12 clearly shows the problem in getting through the third critical and the mild vibrational amplitudes that can be expected at 24,000 rpm. The cause of the third critical speed problem is shown on the mode plot in Figure 7.2-13. Here it can be seen that one of the nodes corresponds almost exactly with the coupling end bearing which explains the undamped nature of this critical.

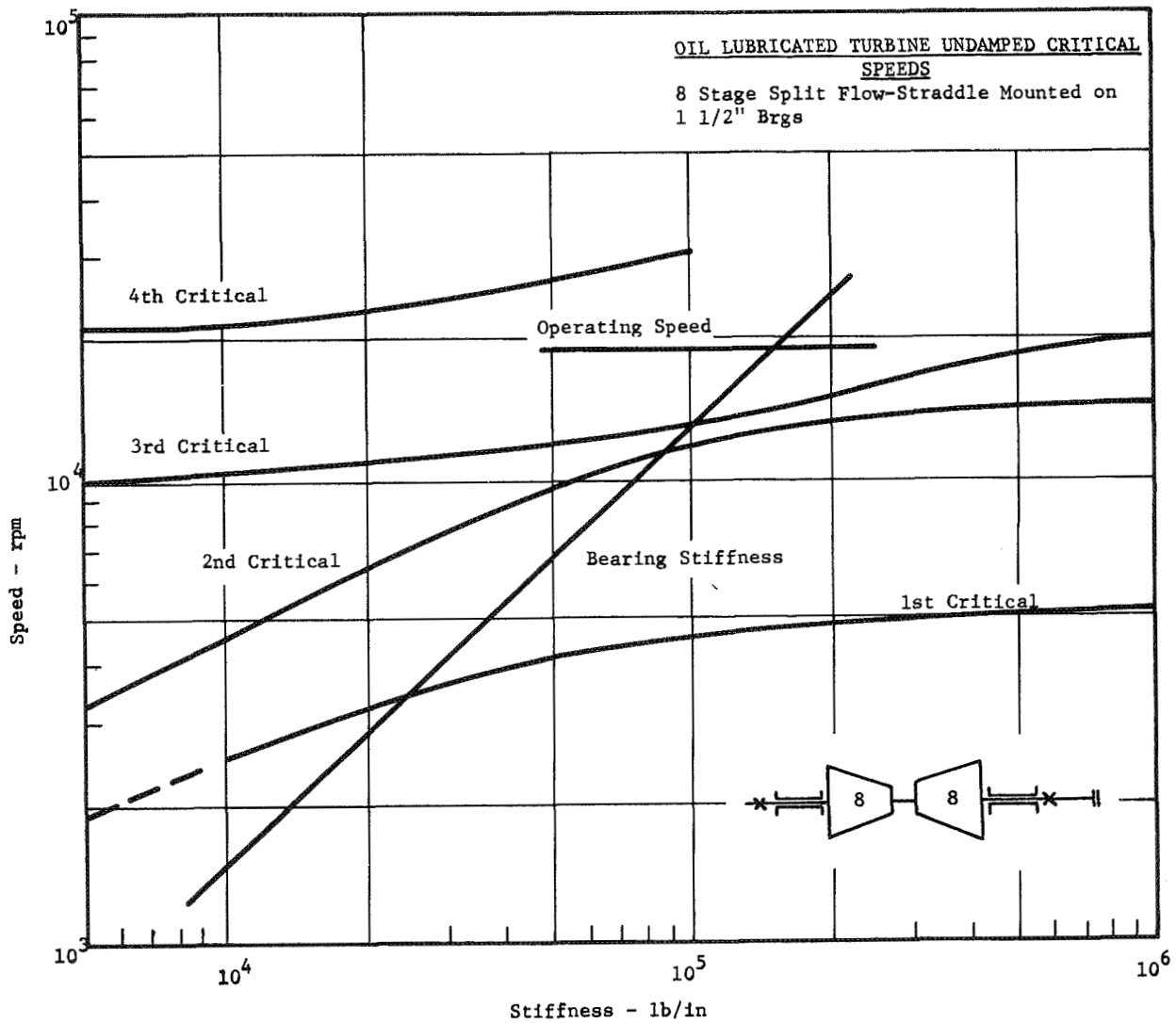


Figure 7.2-11. Oil Lubricated Turbine Undamped Critical Speeds.

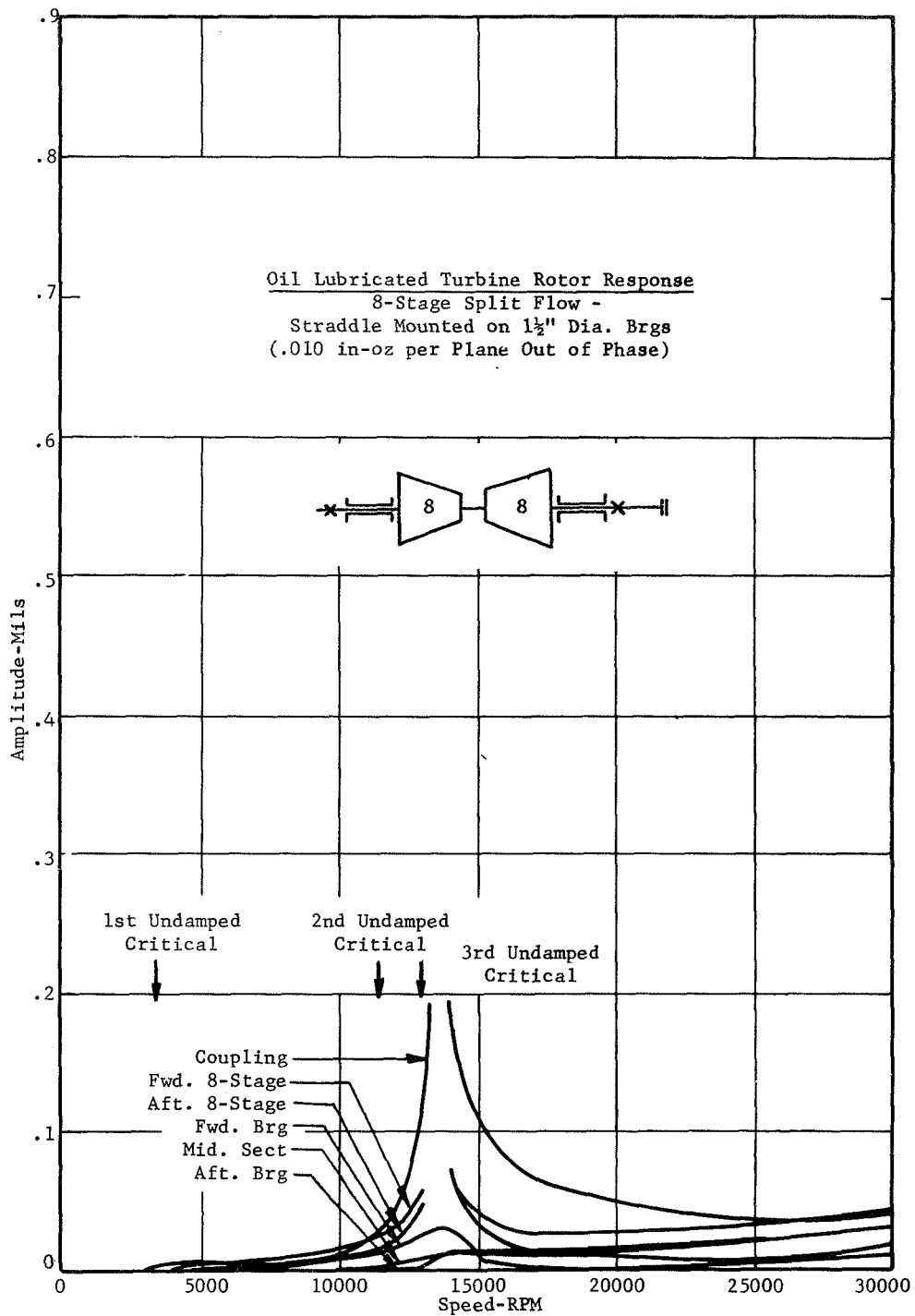


Figure 7.2-12. Oil Lubricated Turbine Rotor Response.

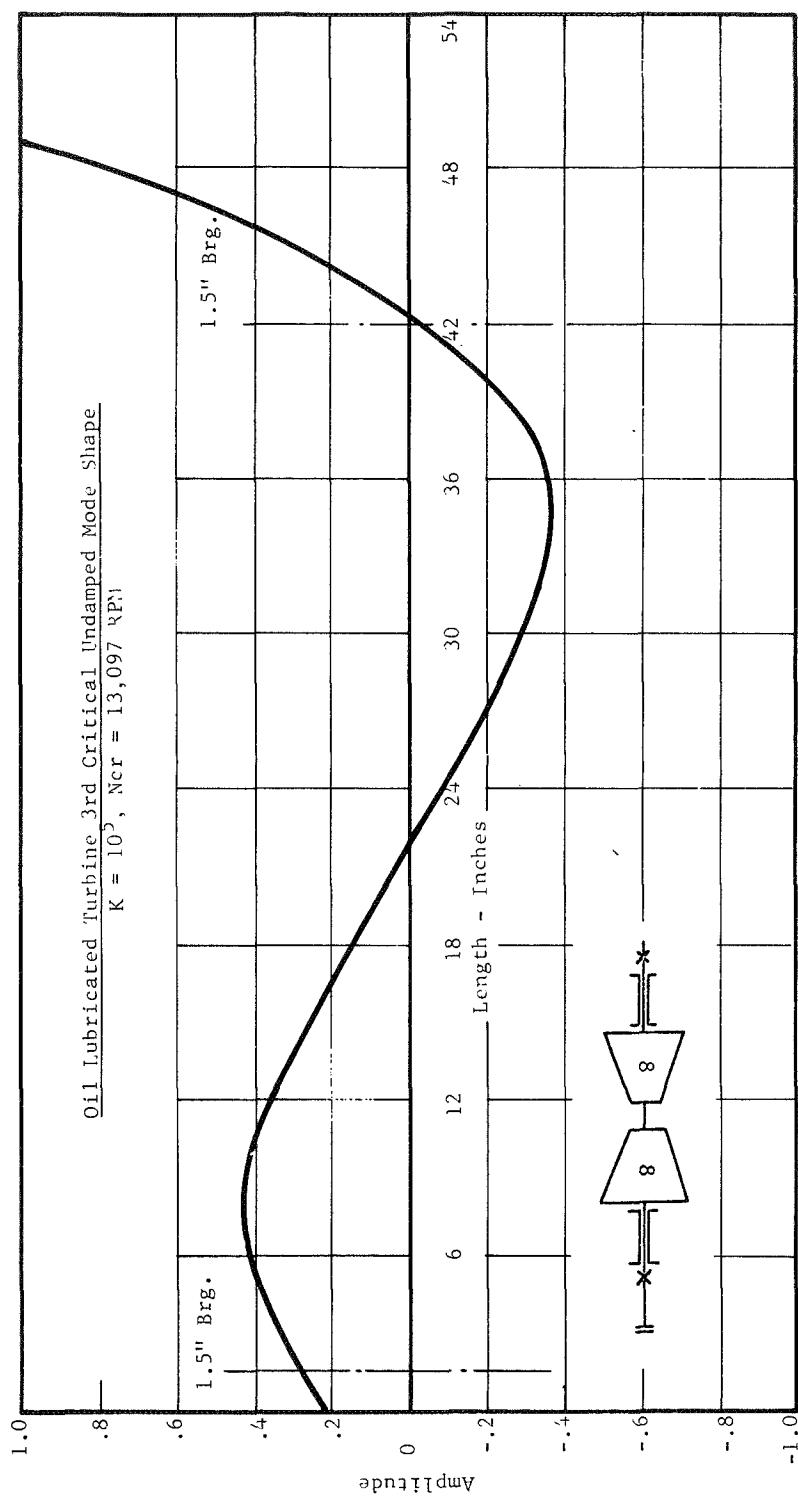


Figure 7.2-13. Oil Lubricated Turbine 3rd Critical Undamped Mode Shape.

Because of the large amount of space taken by the oil-potassium seals, it is not easy to move the bearing away from this nodal point. However, the node can be moved by changing the relative stiffness of the stub to turbine portions of the shaft to shift the location of the points of maximum deflection. This, of course, will also affect the frequency of the resonance. The bearing size was increased on the split flow turbine from 1.5 inches in diameter to 2.25 inches to stiffen the stub shaft and bring the node closer to the coupling. (Examination of Figure 7.2-13 shows that the weakest shaft section is the stub shaft in the area of 36 inches). The critical speed map of the revised stub shaft is shown on Figure 7.2-14 and shows the expected increase in critical speed. The response plot on Figure 7.2-15 indicates the improvement in that the amplification at resonance has been reduced to a point where the rotor design could be considered feasible. If this turbine arrangement were to be selected for refinement during Phase II, further improvements could be made by further adjustment of the bearing location, length, and seal design.

The critical speed map for the split spool oil lubricated turbine (also on 1.5 inch bearings) is shown on Figure 7.2-16 and is not significantly different from the split flow turbine. The third critical speed mode shape shown on Figure 7.2-17 indicated that the node is sufficiently far displaced from the coupling bearing to expect a reasonably well damped resonance in the area between 15,000 rpm and 20,000 rpm. This was confirmed by the rotor response analysis results shown on Figure 7.2-18.

Although it is apparent that, like the split flow arrangement, further improvements could be made in the vibrational behavior of the

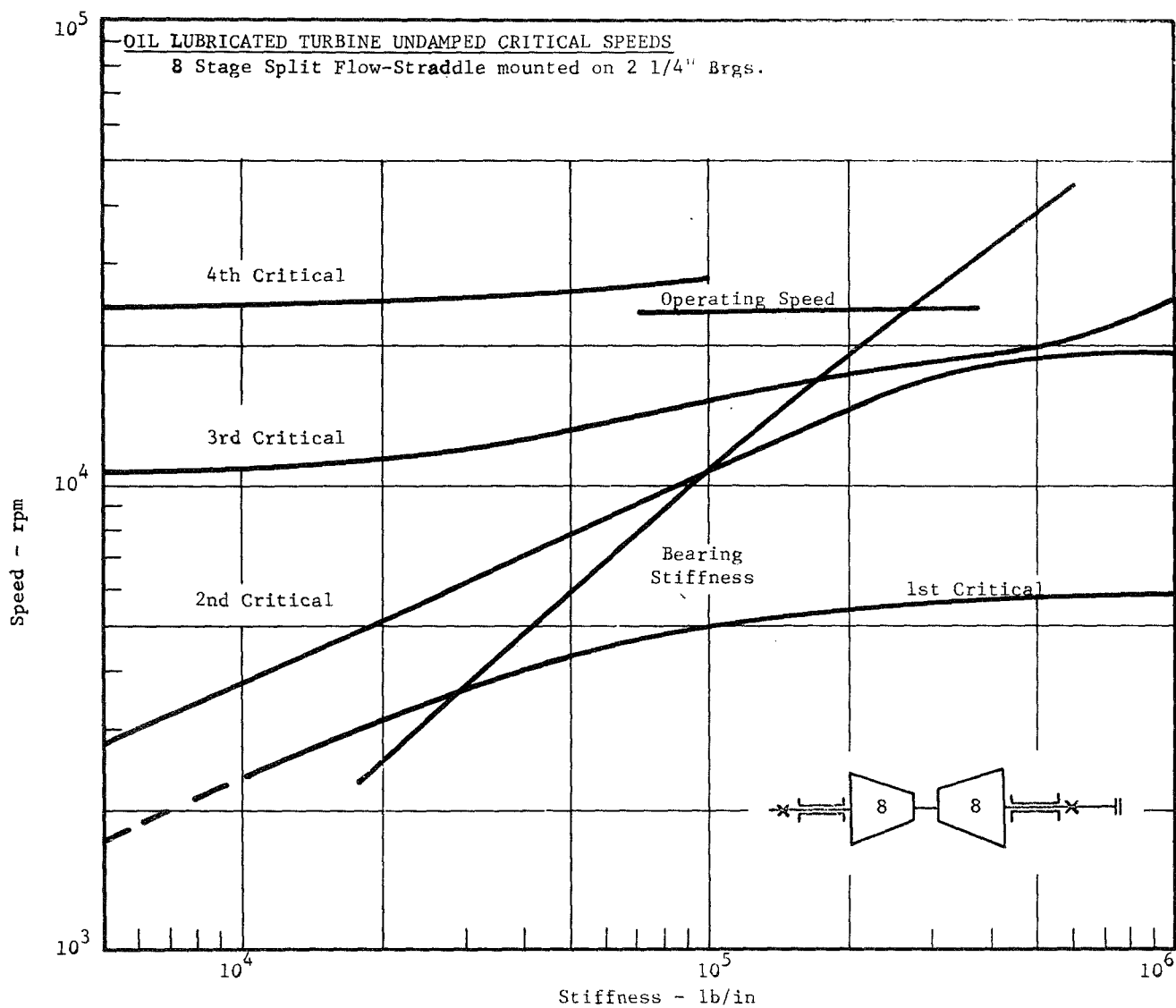


Figure 7.2-14. Oil Lubricated Turbine Undamped Critical Speeds.

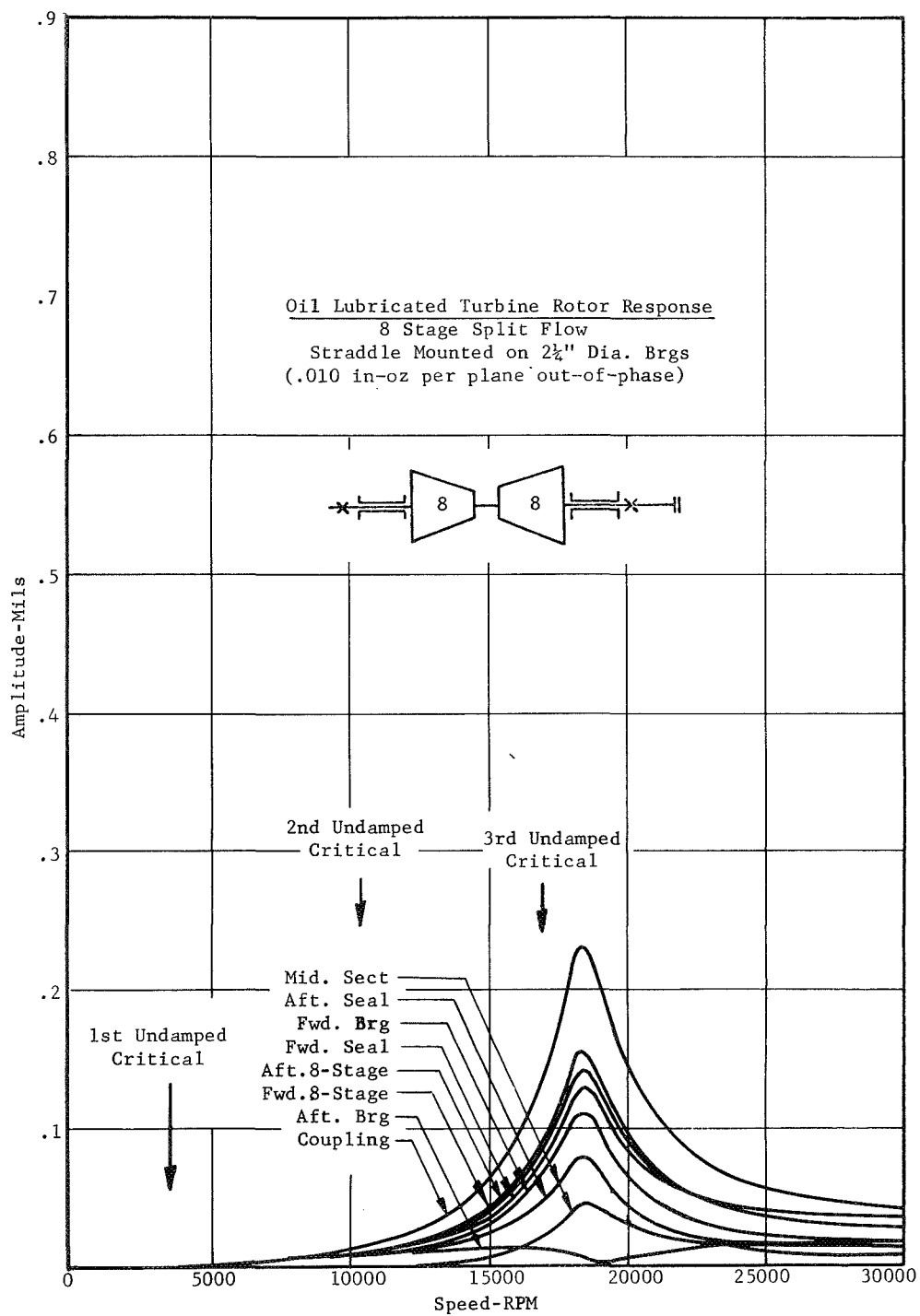


Figure 7.2-15. Oil Lubricated Turbine Rotor Response.

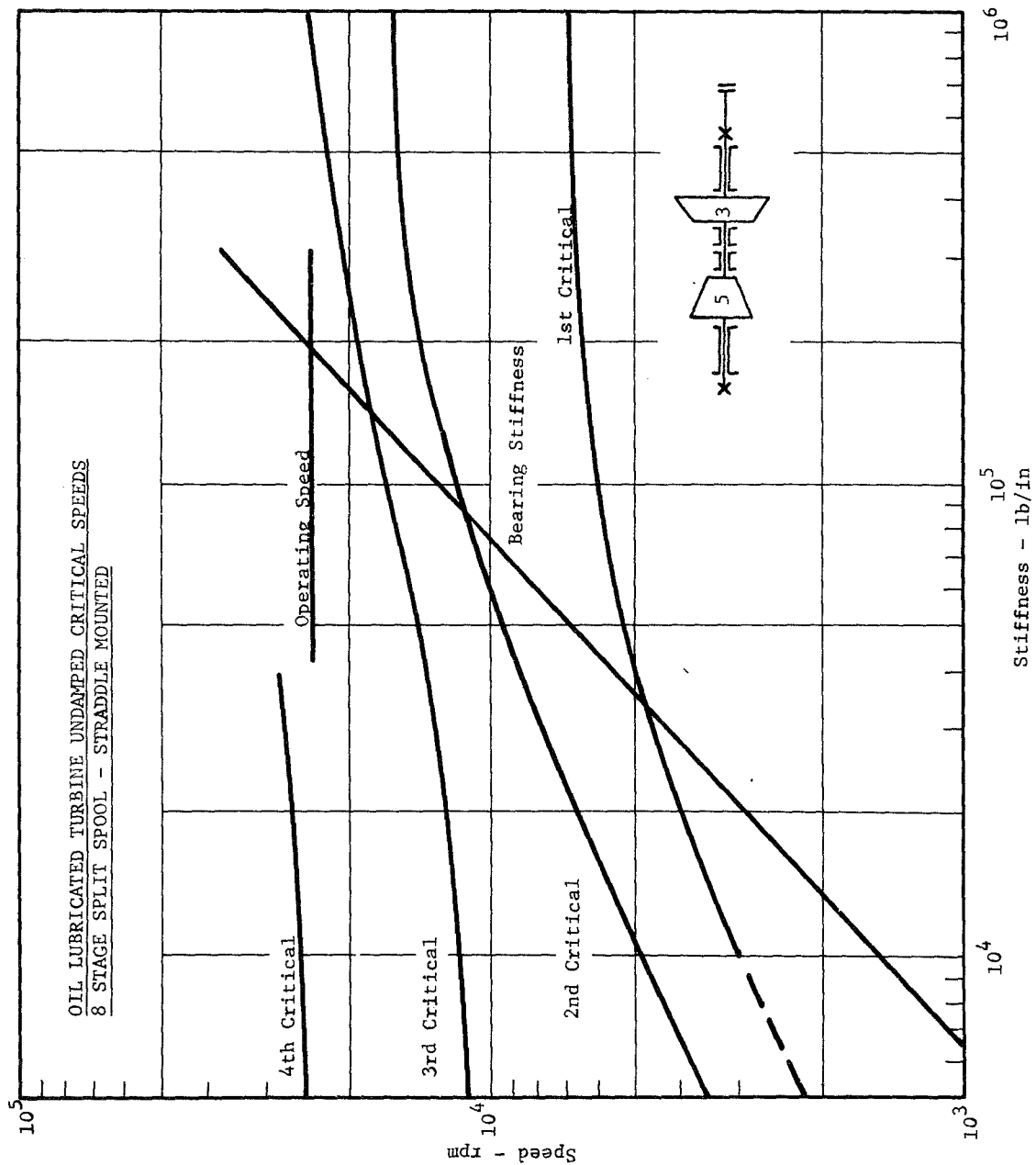


Figure 7.2-16. Oil Lubricate Turbine Undamped Critical Speeds, 8 Stage Split Spool - Straddle Mounted.

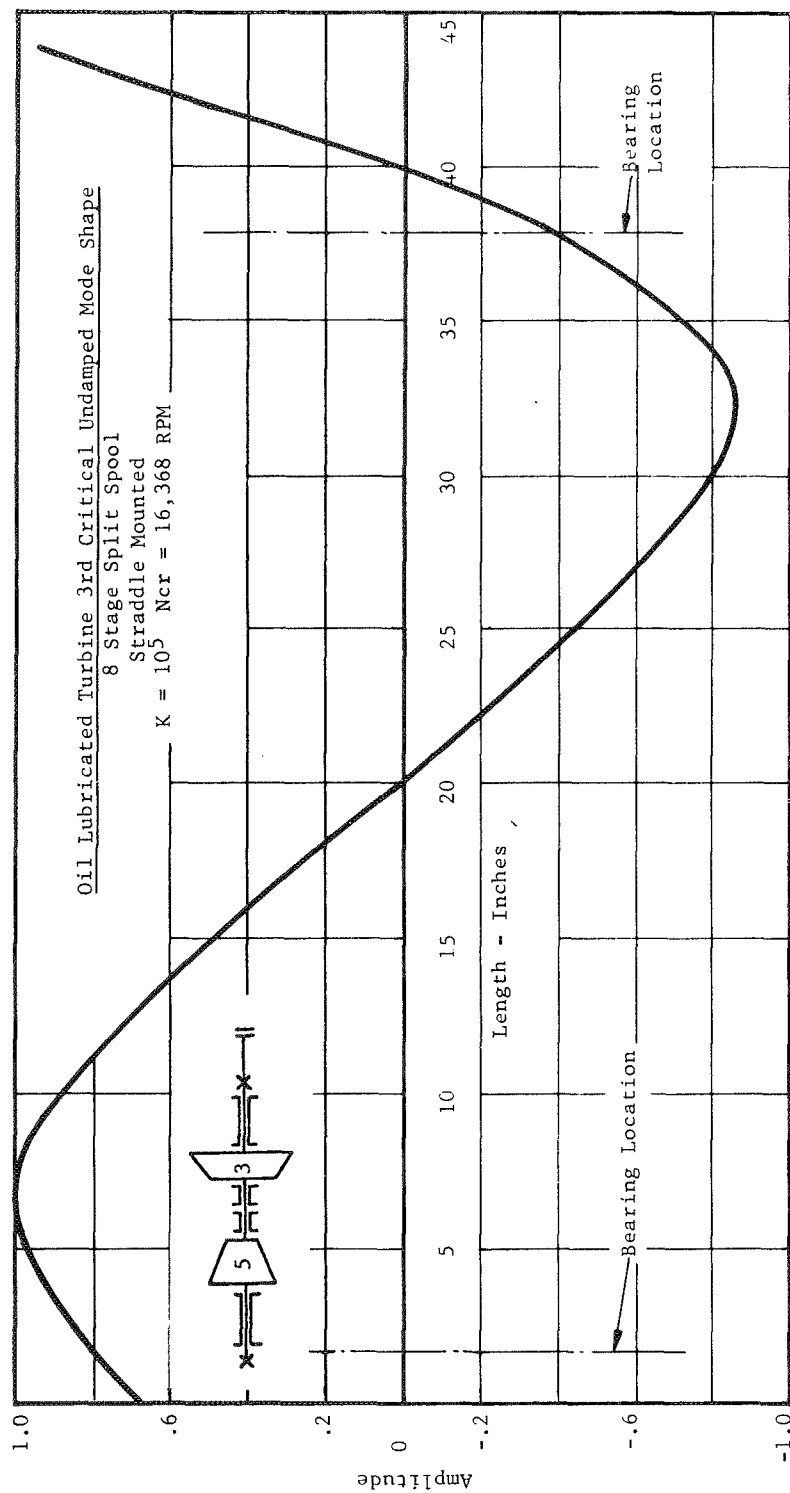


Figure 7.2-17. Oil Lubricated Turbine 3rd Critical Undamped Mode Shape.

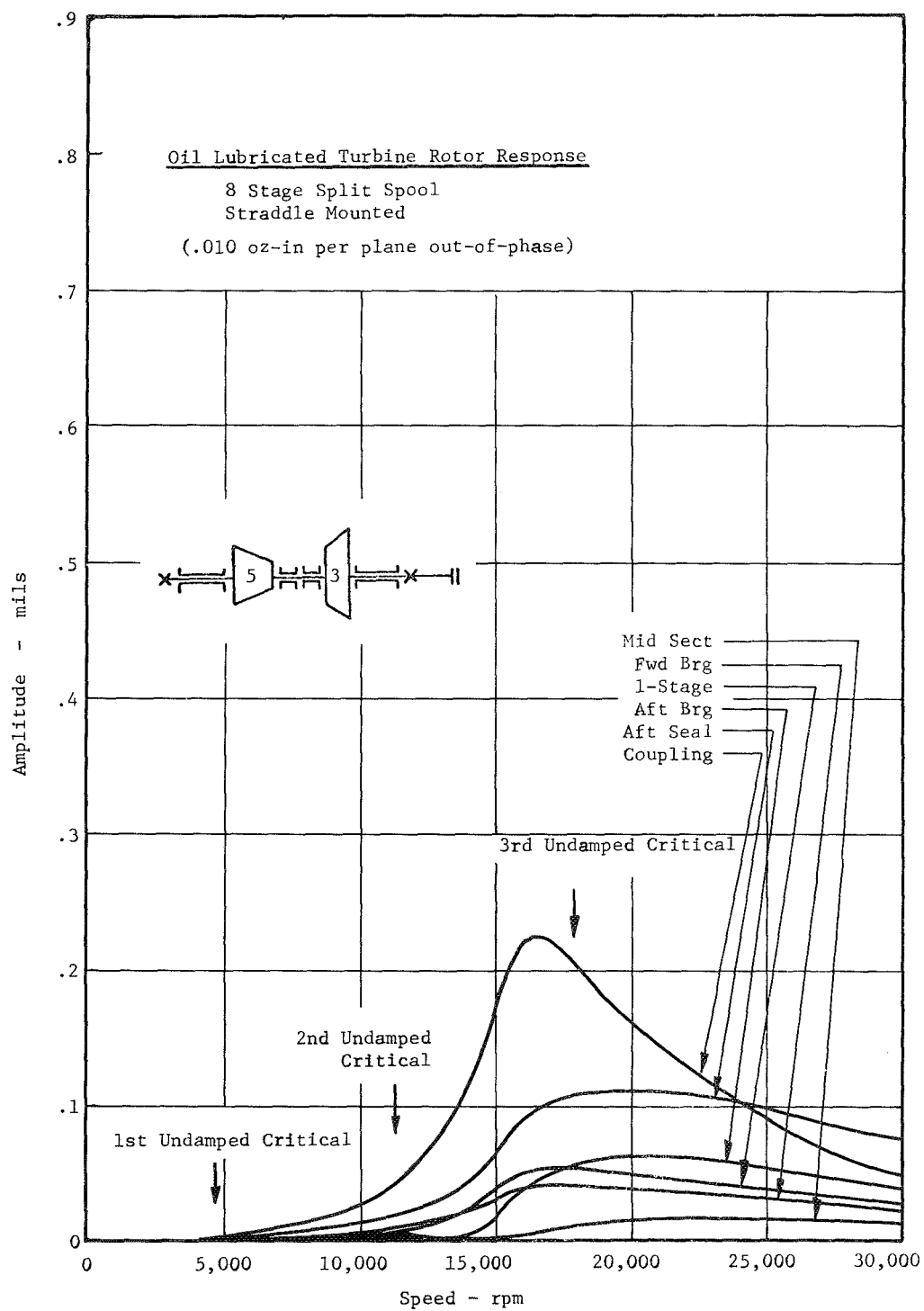


Figure 7.2-18. Oil Lubricated Turbine Rotor Response.

split spool turbine; these were also deferred to Phase II since the analysis work accomplished in Phase I indicated that the arrangement, although not optimum, was feasible.

7.3 ALTERNATORS

All of the alternator arrangements (both potassium and oil lubricated) studied in Phase I were basically straddle-mounted. During the study, variations in stub shaft configuration (length and diameter) and in the thrust bearing location were made in an effort to force the design speed operation to be below the third critical speed. Unlike the non-straddle-mounted turbines, it was found that the third critical almost always caused a problem with the alternators (which are straddle-mounted) because the high degree of stub shaft compliance resulted in small amplitudes of vibration at the bearings which makes it impossible to realize the damping potential of the bearings.

The same assumptions were used in the analysis of the vibrational response of the alternators as was used on the turbines except that the value of unbalance used was .04 in-oz per plane on the potassium cooled alternators and .03 in-oz per plane on the oil cooled alternators. These unbalances are consistent with the aforementioned criteria of $\text{in-oz} = 4W/N$

Potassium Cooled Alternators

The five basic variations in the stub shaft configurations of the potassium cooled alternator that were studied included:

- (1) The configuration shown on Figure 7.3-1 (GE Drawing No. 47R199276) which has 2.25 inch diameter bearings and had the

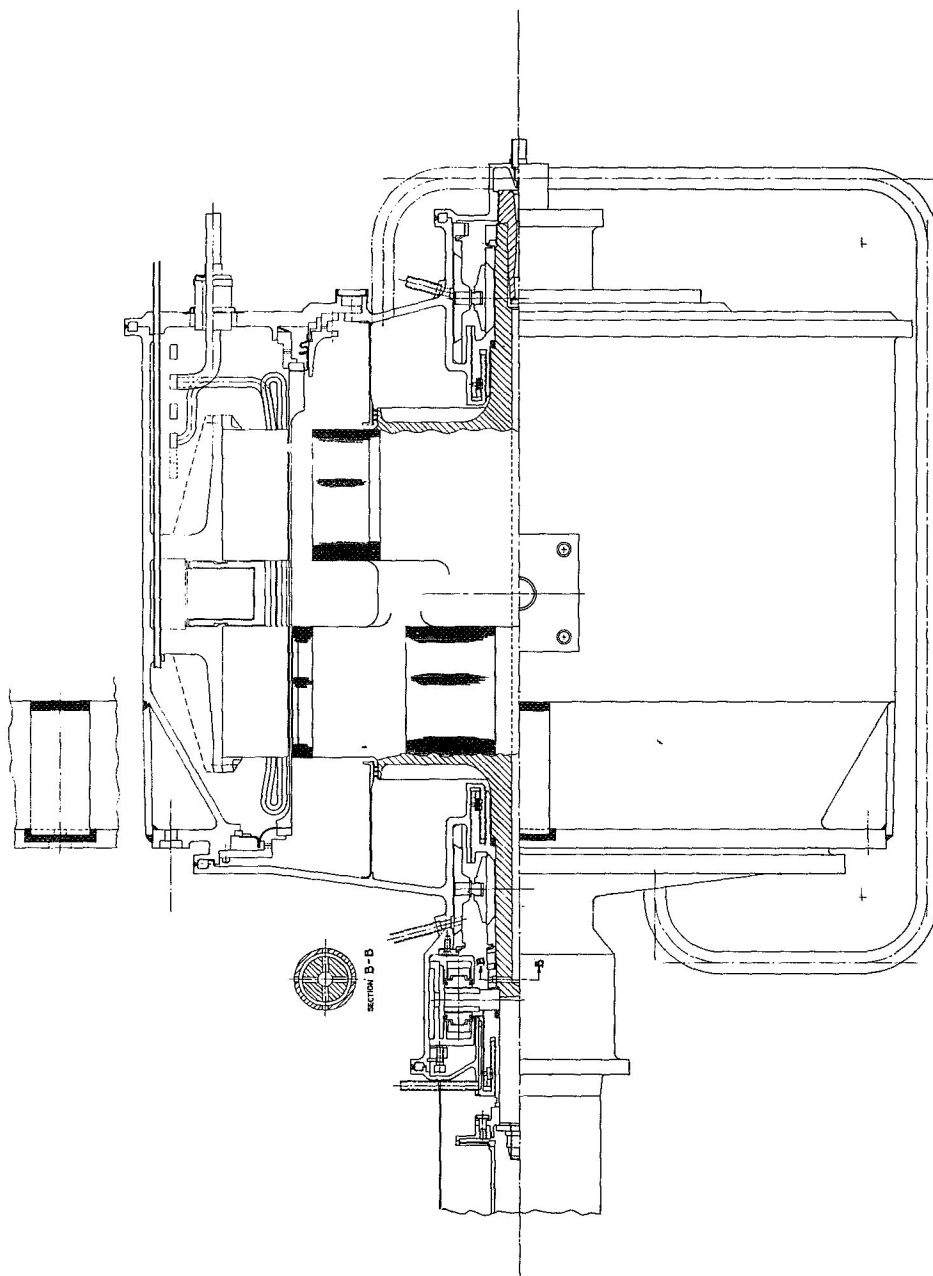


Figure 7.3-1. Alternator on Potassium Bearings - Configuration 1.
(Dwg. No. 47R199276)

thrust bearing located between the coupling and the coupling end journal bearing.

- (2) The same as (1) but with the bearing diameter increased to 3.0 inches and the stub shafts shortened by a more compact seal-bearing packaging. Figure 7.3-2.
- (3) The same as (2) except with the thrust bearing located on the end of the rotor opposite to the coupling.
- (4) The same as (3) but with 3.5 inch diameter bearings.
- (5) The same as (2) and (3) but with the thrust bearing located between the alternator rotor and the coupling end journal.

The critical speeds and schematic diagrams of these five configurations are shown on Figures 7.3-3 to 7.3-7. Except for the first configuration, all have third critical speeds above the design speed of 19,200 rpm and have fairly predictable responses to unbalance loads. The critical speed map for the alternator with 2.25 inch diameter bearings (Configuration 1) indicates undamped second and third critical speeds at approximately 3,000 rpm and 13,000 rpm. The response plot on Figure 7.3-8, however, shows a resonance peak at about 9,000 RPM or near midway between the two calculated undamped criticals. It is not uncommon to find the maximum amplitude frequency occurring midway between criticals and this phenomenon has been both experimentally and analytically verified (reference 7.3-3). Some insight into the physical reason behind this can be gained by referring to mode plot on Figure 7.3-9. Here it can be seen that both the second and third criticals tend to be conical in nature and have relatively large amplitudes at the bearings. One would expect under this condition to

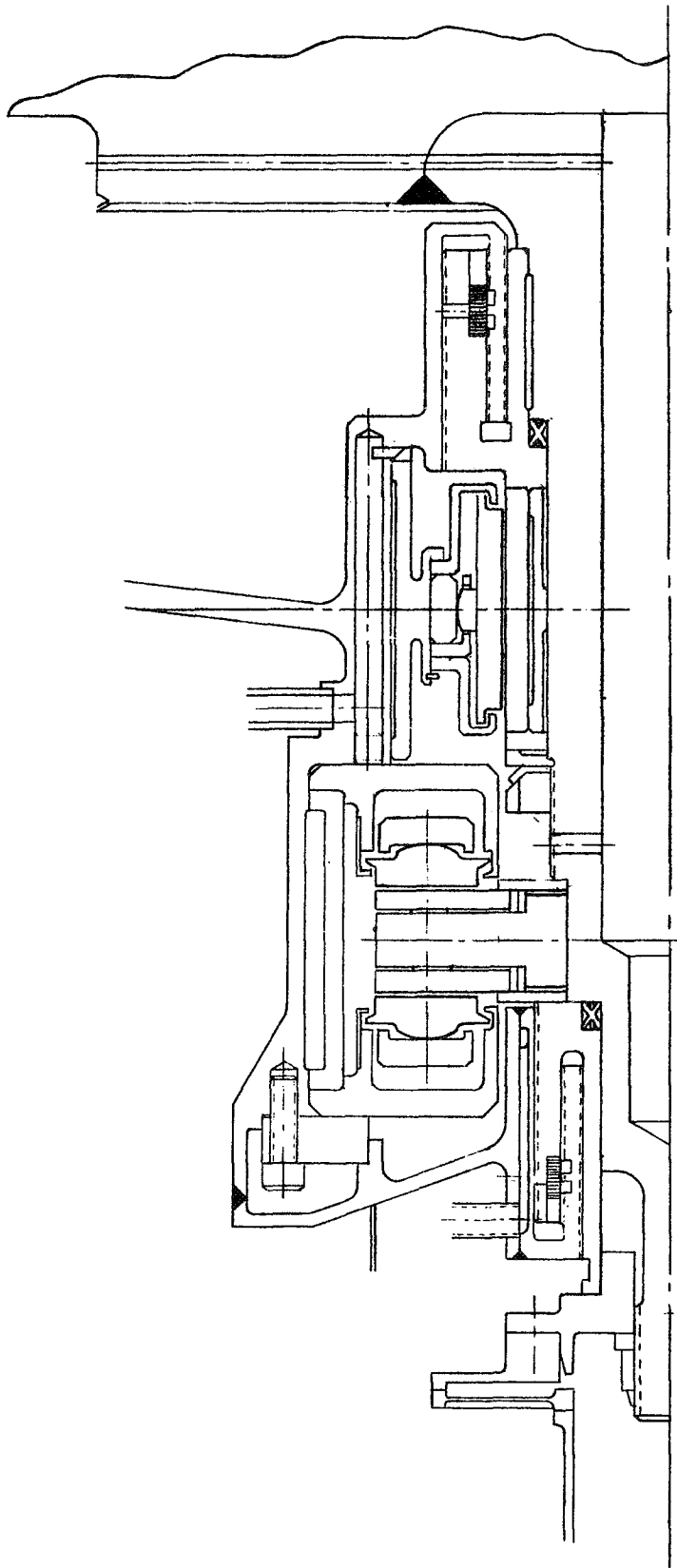


Figure 7.3-2. Alternator on Potassium Bearings - Configuration 2.
(SKC 2998)

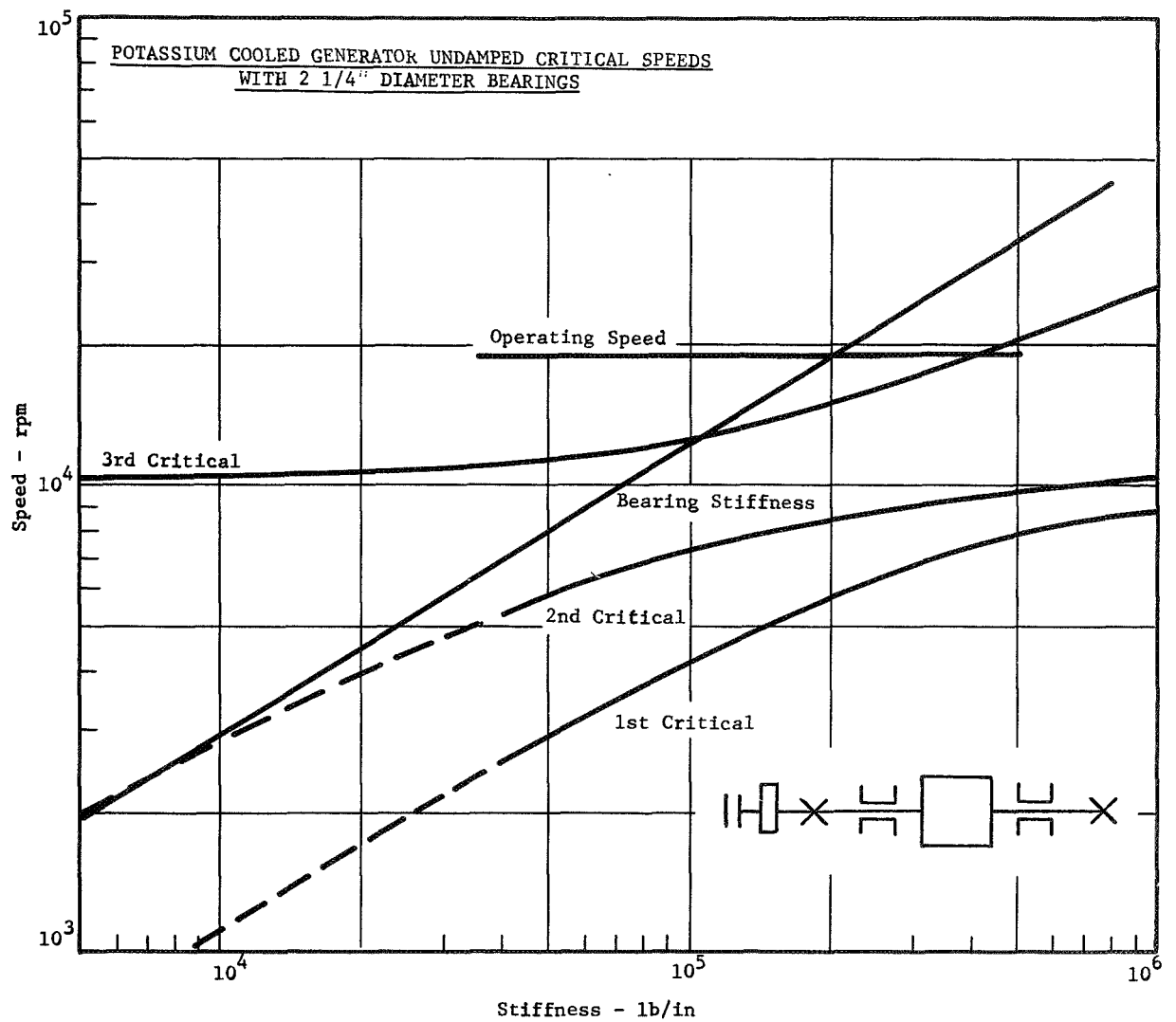


Figure 7.3-3. Potassium Cooled Generator Undamped Critical Speeds with 2 1/4" Diameter Bearings. (Configuration 1)

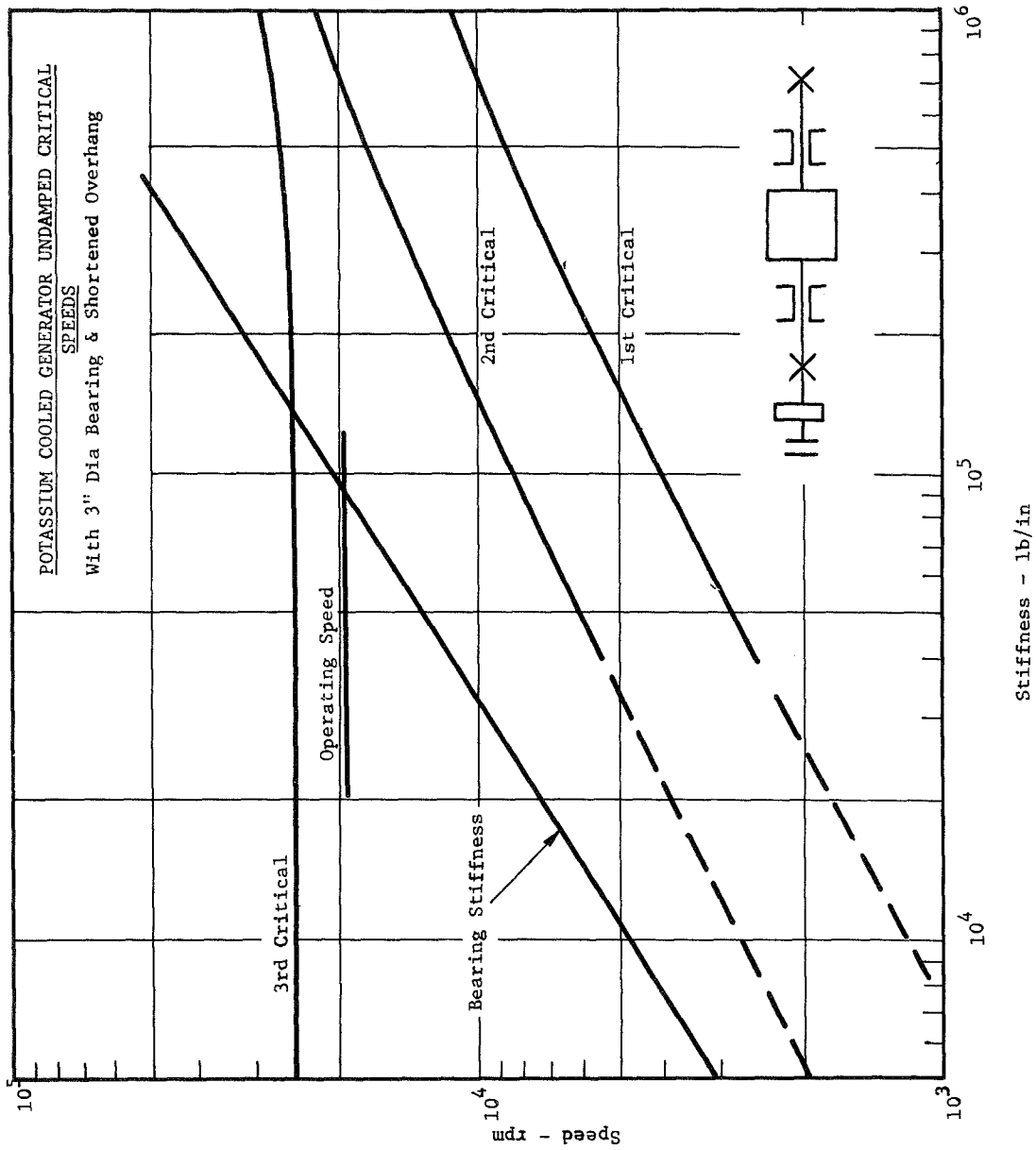


Figure 7.3-4. Potassium Cooled Generator Undamped Critical Speeds.
(Configuration 2)

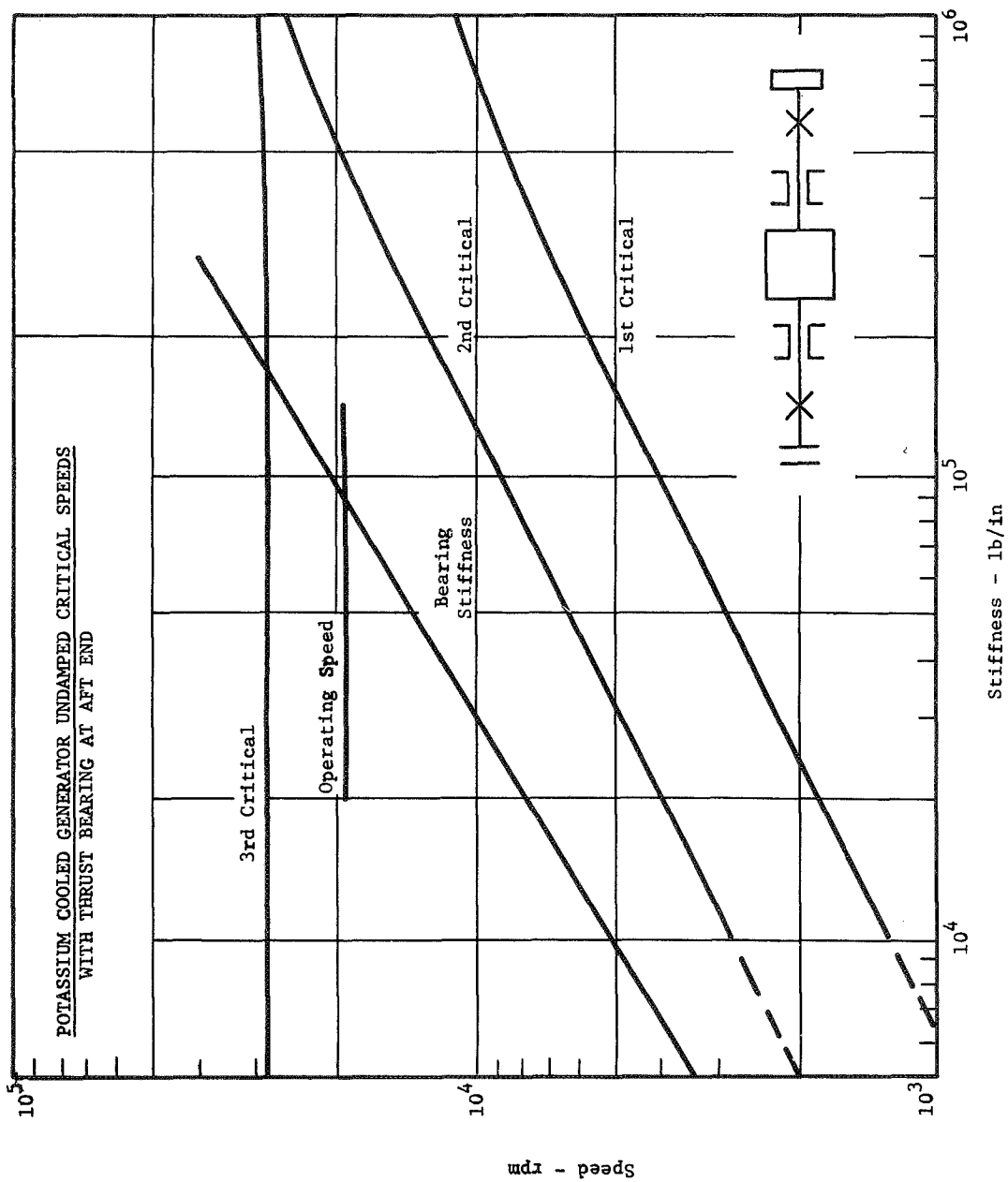


Figure 7.3-5. Potassium Cooled Generator Undamped Critical Speeds with Thrust Bearing at Aft End. (Configuration 3)

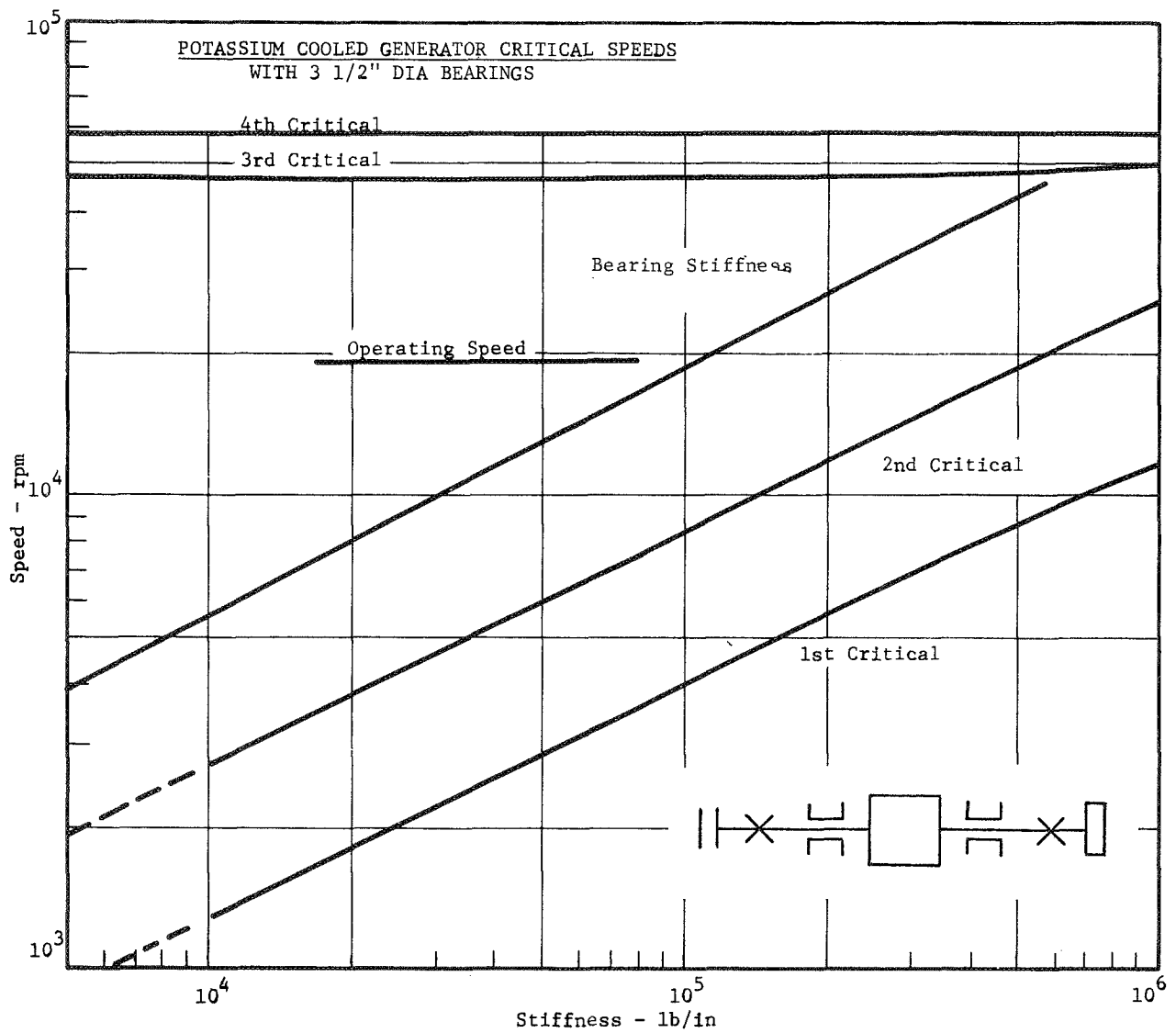


Figure 7.3-6. Potassium Cooled Generator Critical Speeds with 3 1/2" Diameter Bearings. (Configuration 4)

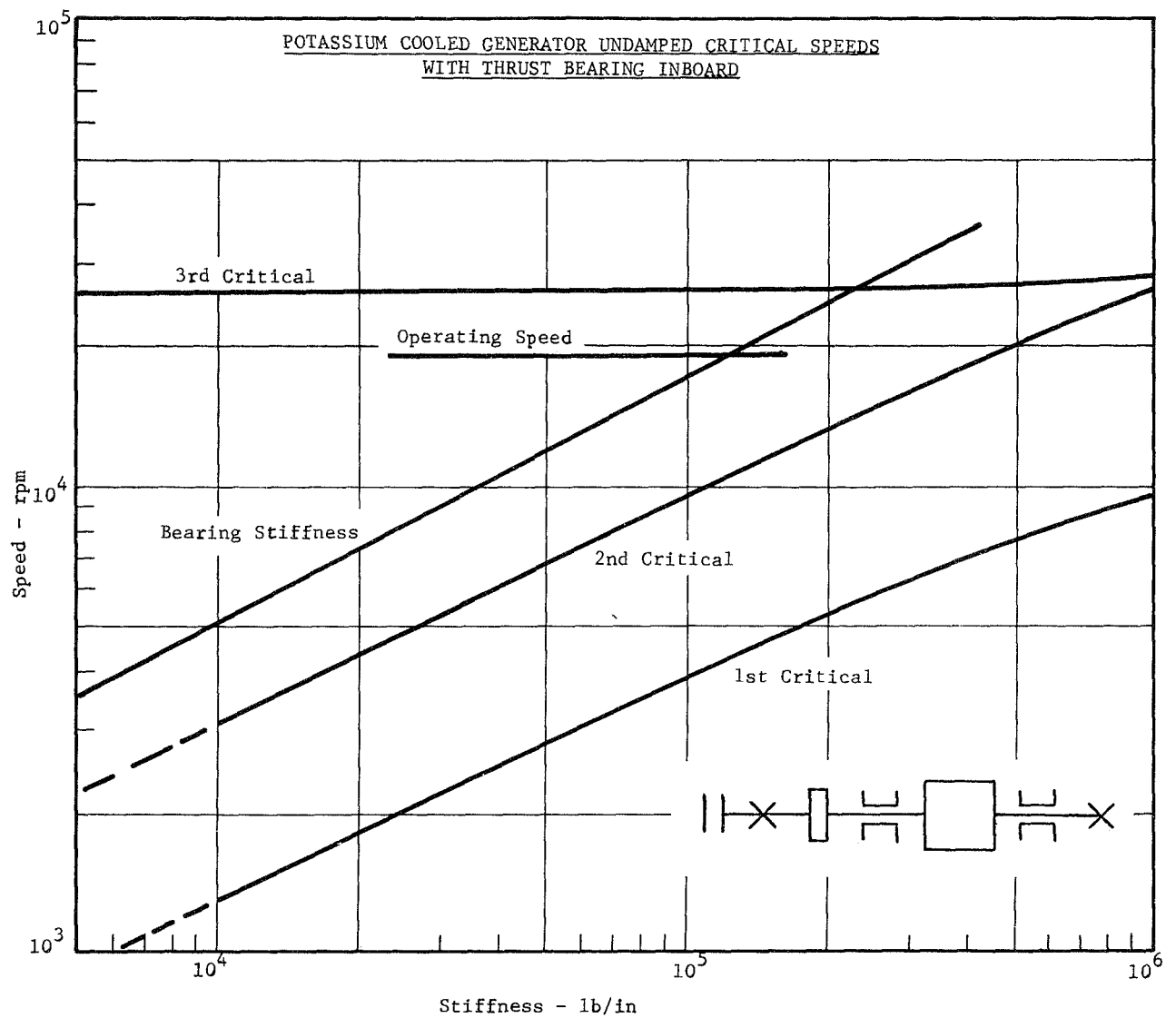


Figure 7.3-7. Potassium Cooled Generator Undamped Critical Speeds with Thrust Bearing Inboard. (Configuration 5)

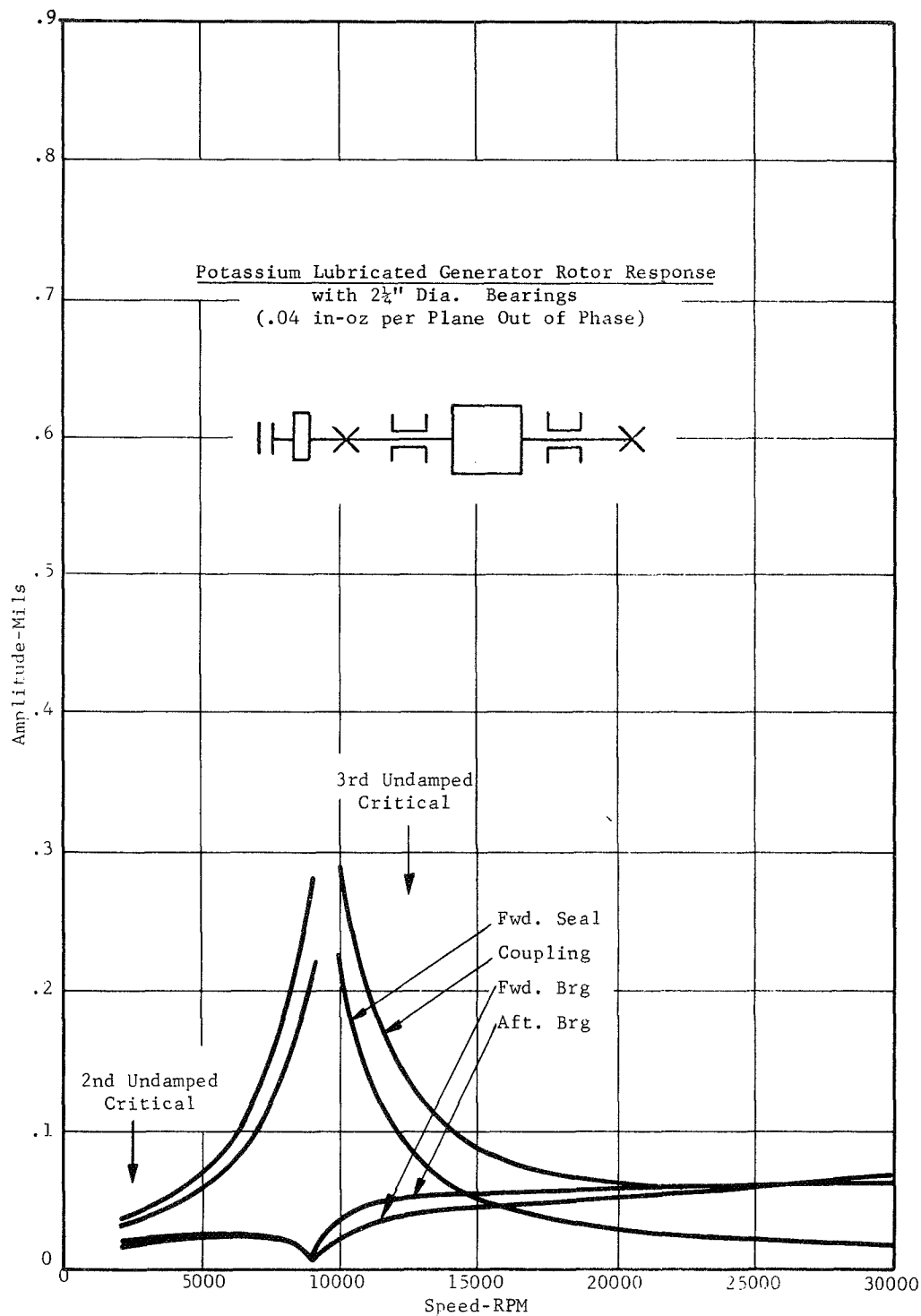


Figure 7.3-8. Potassium Lubricated Generator Rotor Response with 2 1/4" Diameter Bearings. (Configuration 1)

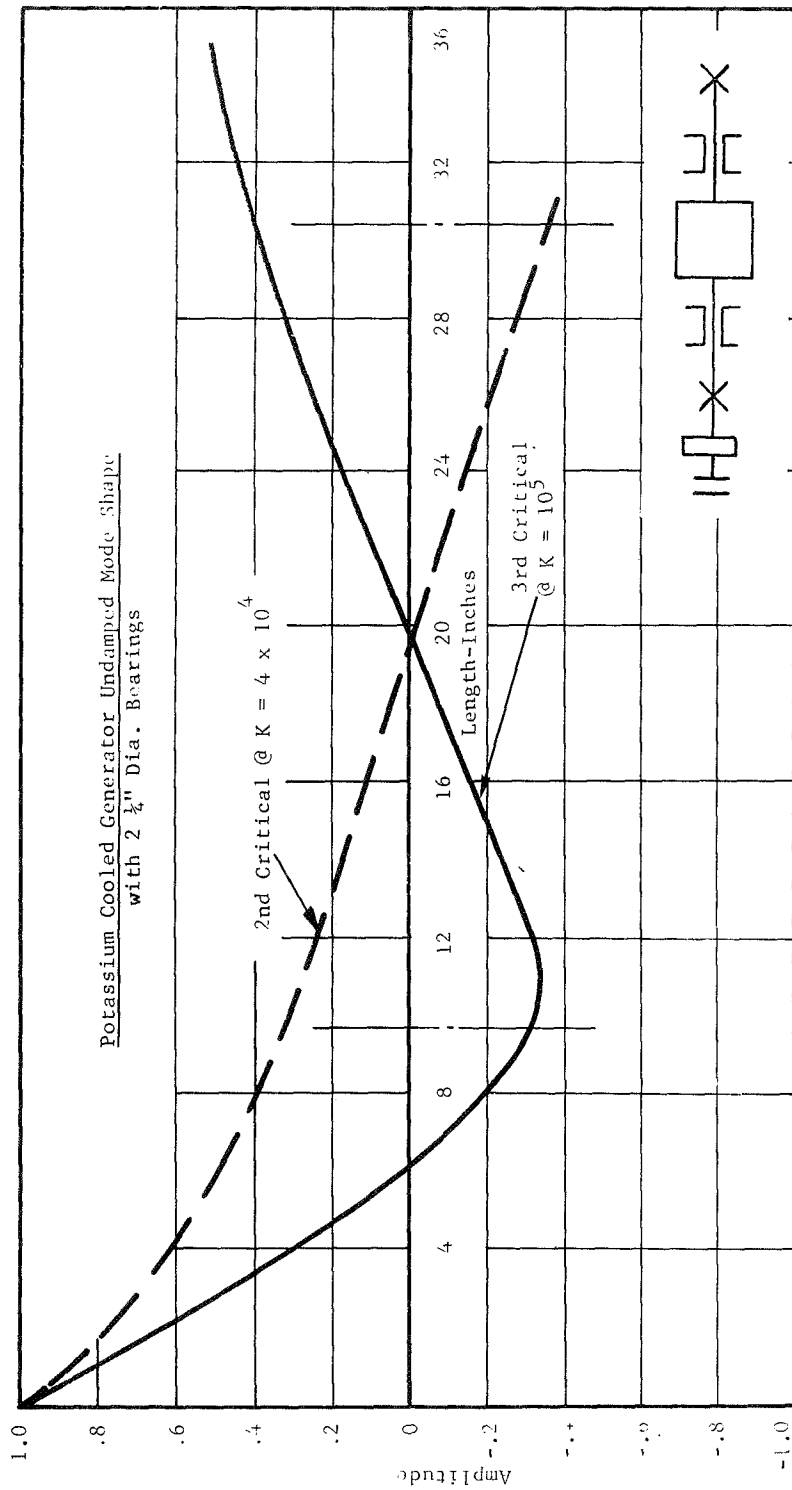


Figure 7.3-9. Potassium Cooled Generator Undamped Mode Shape with 2 1/4" Diameter Bearings. (Configuration 1)

experience good attenuation at these resonances to the point where they may be obliterated, as is evident from the results of the response analysis. However, in the frequency range between these two criticals, the mode shape must change from one to the other and it is evident that the coupling always is the location of maximum amplitude. Therefore, at some frequency (speed) between the two criticals, the portion of the rotor between bearings will lie nearly parallel and close to the centerline. In other words, much of the rotor will be near nodal, having very small vibration amplitudes. Such a mode shape at 9,000 rpm with a bearing stiffness of 4×10^5 lb/in. is shown in Figure 7.3-10. Although this is not exactly the mode shape at 9,000 rpm with a bearing stiffness of 6×10^4 lb/in, it serves to typify the vibrational mode that must occur at about the speed that the maximum coupling amplitudes were calculated. With such a mode shape, the bearing damping forces become very small and although not a critical speed in the classical sense, large amplitudes of vibration at the coupling result.

The remaining configurations (2 to 5) were attempts to both raise the third critical and to remove the nodes from the areas near the bearings so that sufficient damping could be obtained to attenuate the effects of the third critical speed. As can be seen from Figures 7.3-4 to -7, all the third criticals for these configurations are above design speed but the response plots (Figures 7.3-11 to -14) for all except configuration 4 show very large amplification at the third critical. These amplitude peaks are both too close to the design

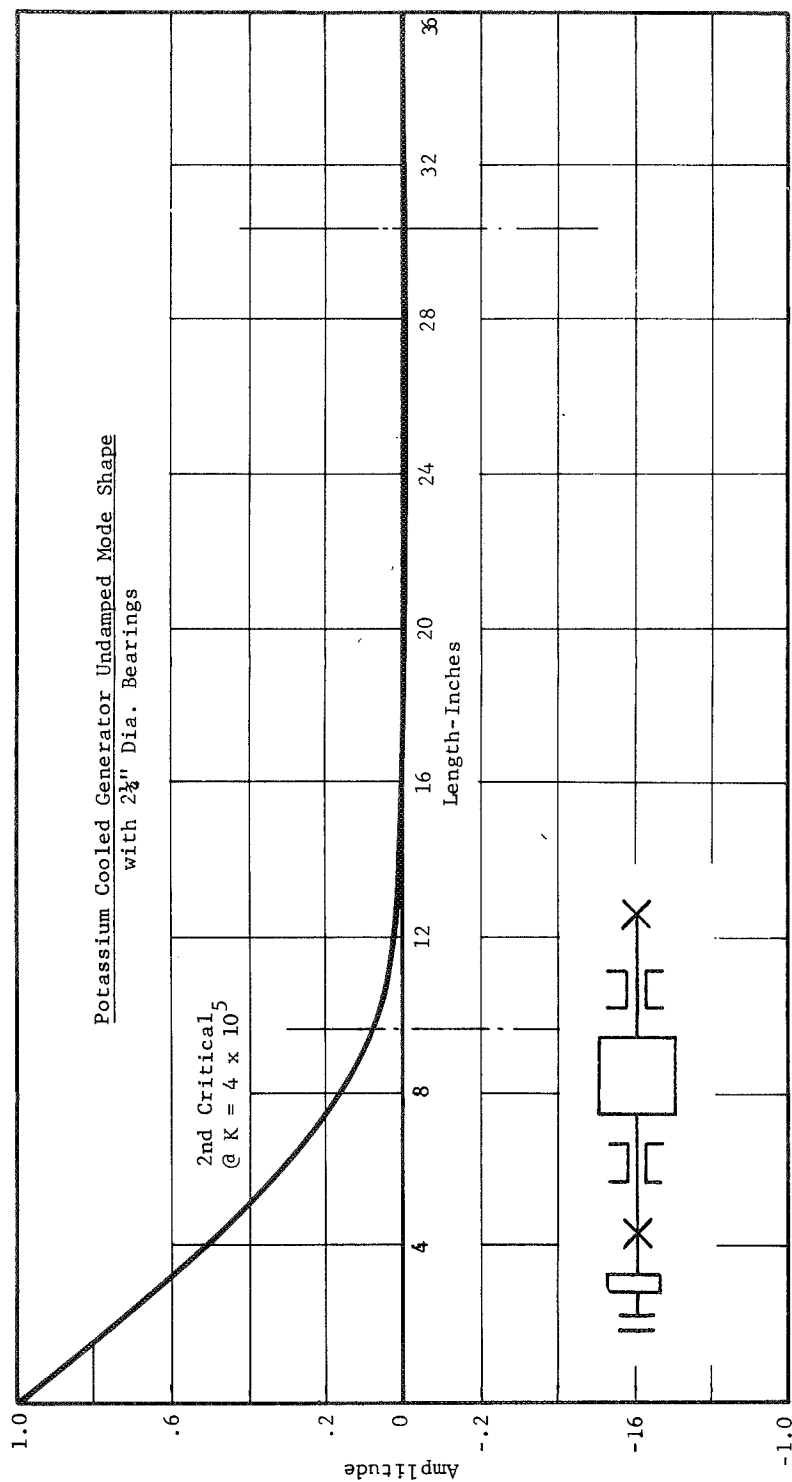


Figure 7.3-10. Potassium Cooled Generator Undamped Mode Shape with 2 1/4" Diameter Bearings. (Configuration 1)

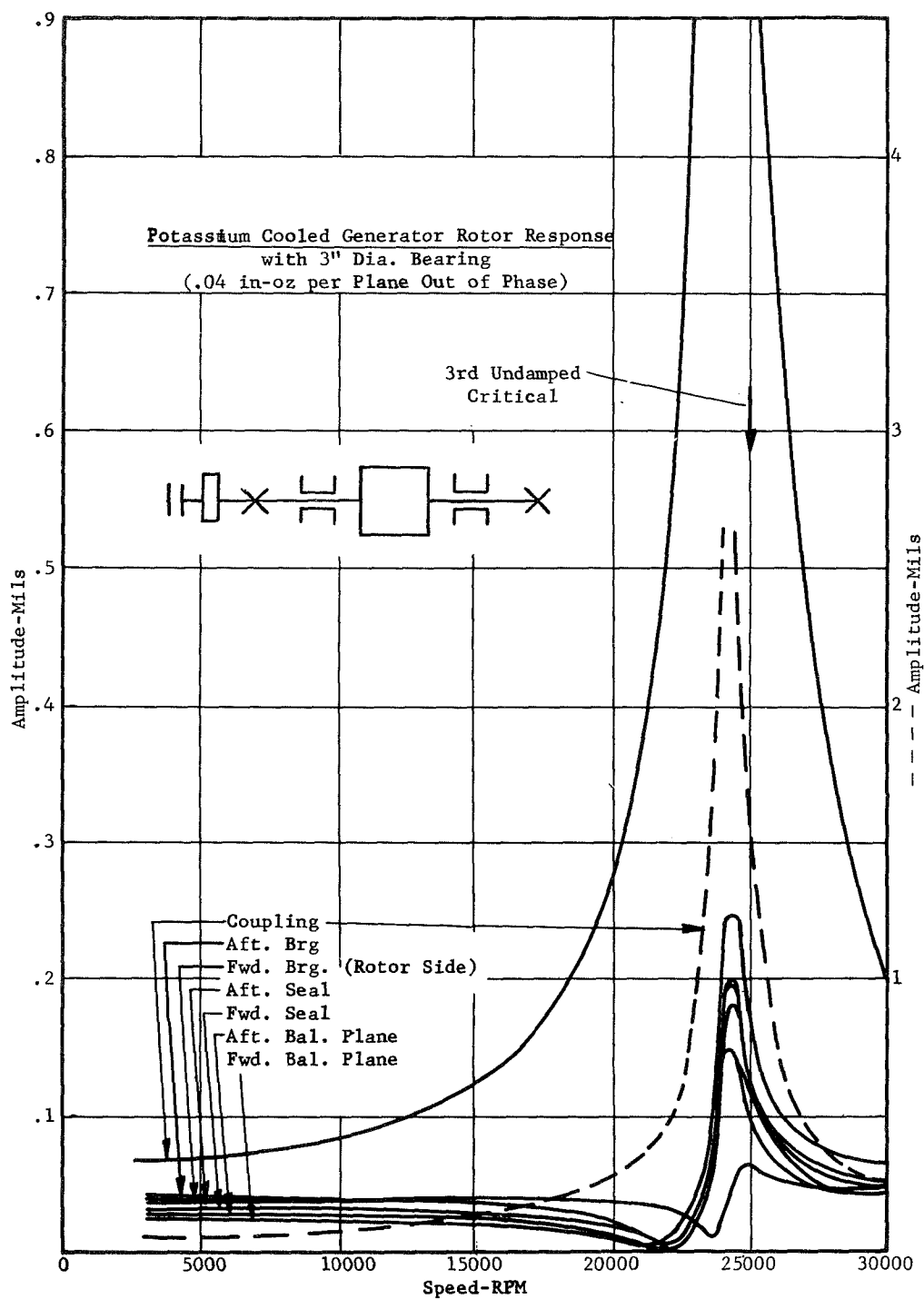


Figure 7.3-11. Potassium Cooled Generator Rotor Response with 3" Diameter Bearing. (Configuration 2)

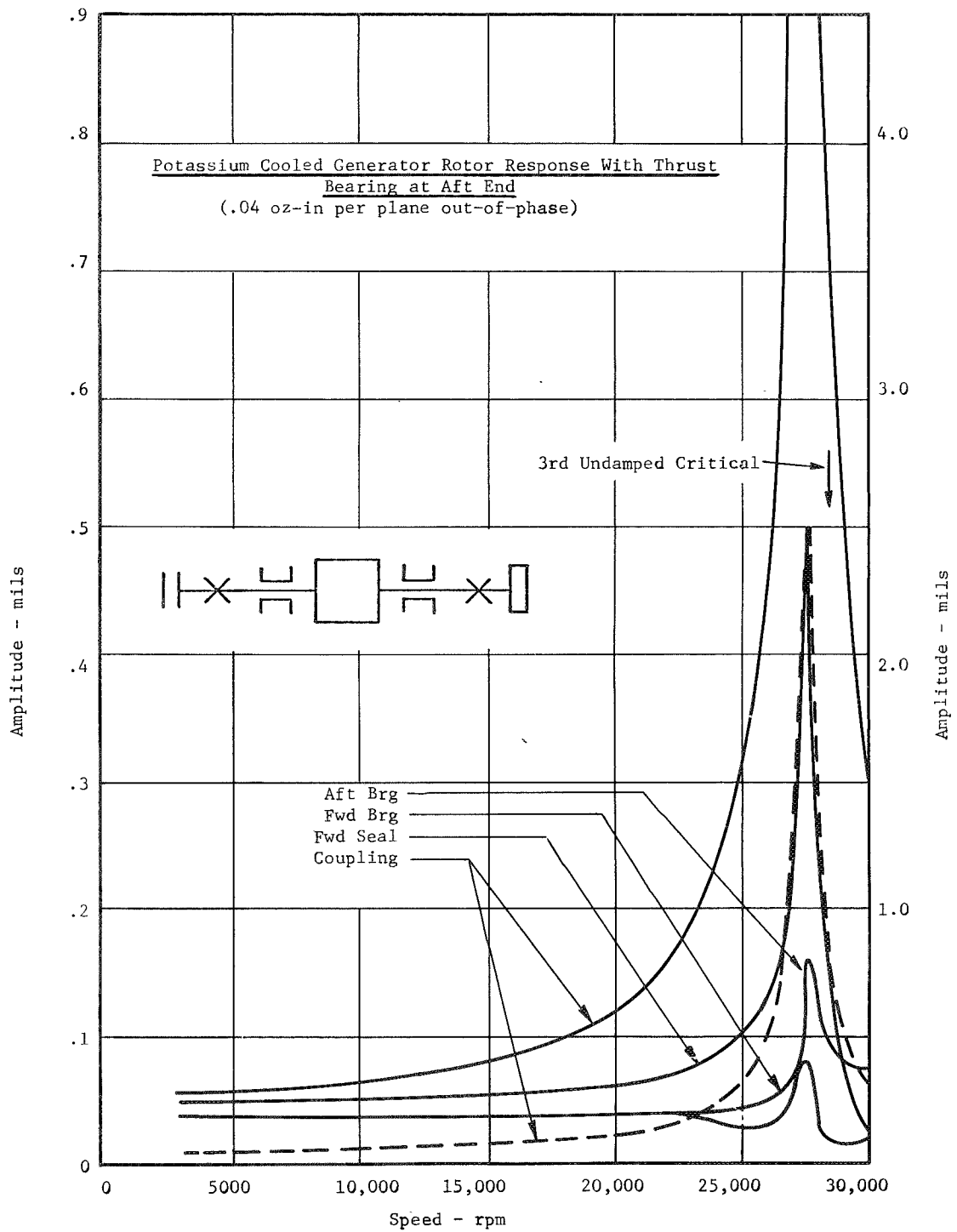


Figure 7.3-12. Potassium Cooled Generator Rotor Response with Thrust Bearing at Aft End. (Configuration 3)

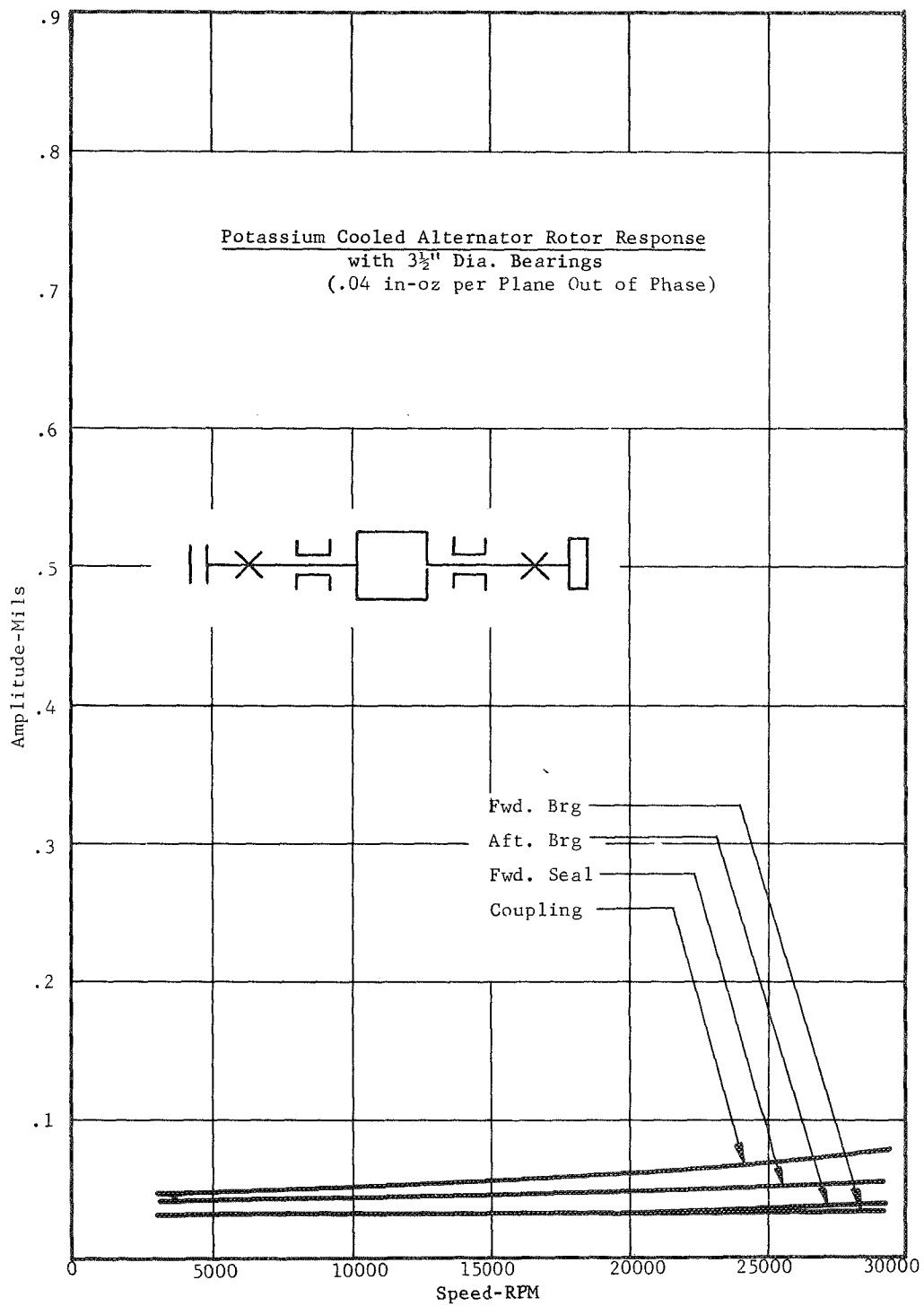


Figure 7.3-13. Potassium Cooled Alternator Rotor Response with 3 1/4" Diameter Bearings. (Configuration 4)

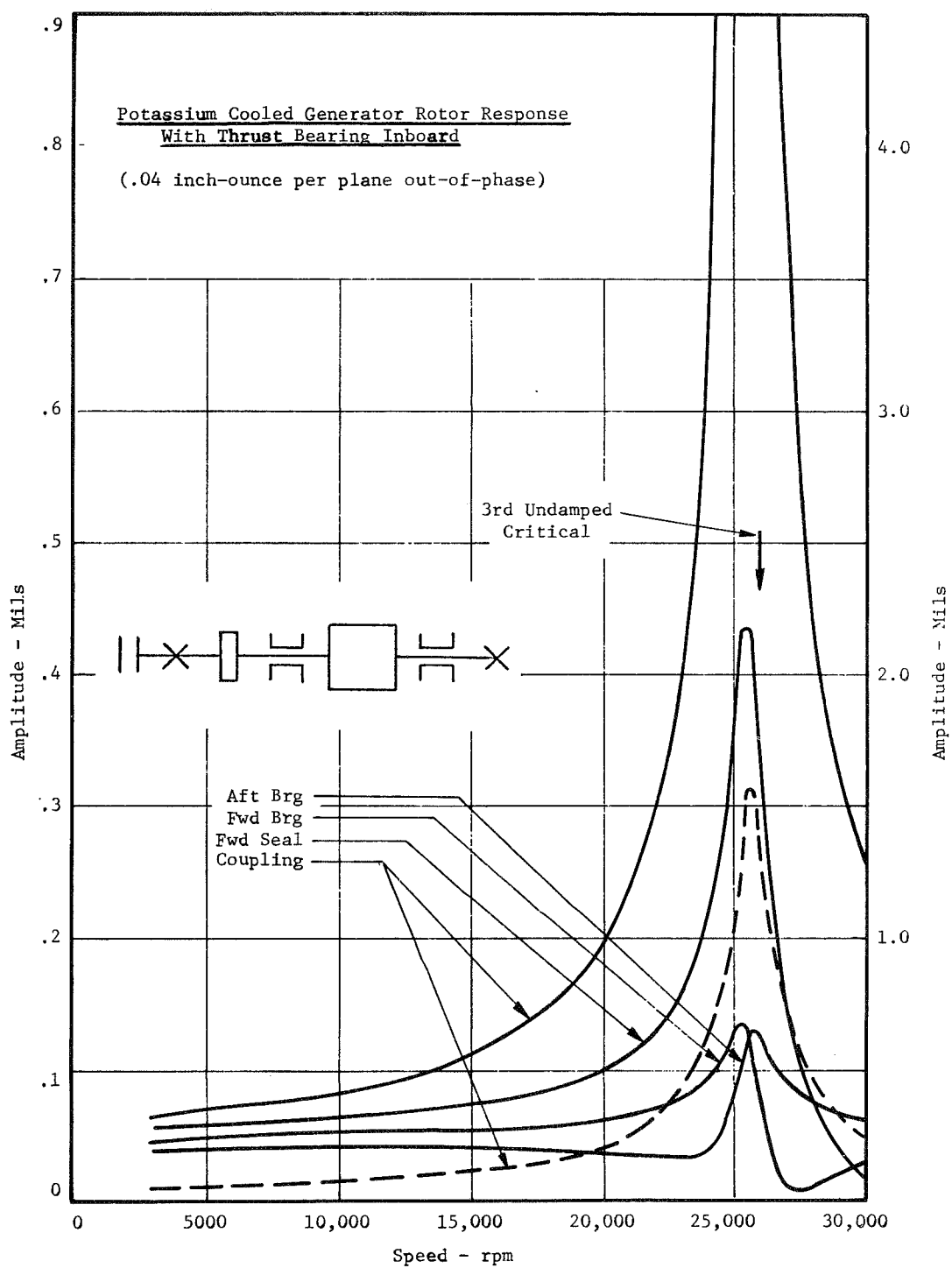


Figure 7.3-14. Potassium Cooled Generator Rotor Response with Thrust Bearing Inboard. (Configuration 5)

overspeed condition (24,000 rpm) and are so steep in nature as to be a potential problem. The reason for these large amplifications at the third resonance is again shown on the typical mode plots on Figure 7.3-15 and -16 which indicate the proximity of the nodes to the bearings.

Only configuration 4, which was a brute force approach in raising the third critical to well beyond the operating range, shows much promise in adequately controlling the rotor vibration amplitudes. The use of 3.5 inch diameter bearings here permitted sufficient stiffening of the stub shafts to raise the third critical to close to 50,000 rpm. Although the mode shape at this critical is no different in nature from those previously discussed, the potential damping is not required because of the large separation between the operating speed and this troublesome critical.

Oil Cooled Alternators

The oil lubricated alternator which weighed only about three quarters that of the potassium cooled alternator (because of its higher design speed) also showed similar vibrational problems. For the purposes of Phase I, two stub shaft designs were considered. One had 1.5 inch diameter bearings and its critical speed map is shown on Figure 7.3-17. In this case the second critical speed was the most troublesome as is shown on the response plot on Figure 7.3-18. The reason for this is that the extremely weak stub shafts caused the second critical to be bending in nature with a mode shape very similar to that shown in Figure 7.3-10. The third critical mode shape was like that of potassium cooled alternator shown on Figure 7.3-7 which, of course, resulted in almost complete attenuation of this critical.

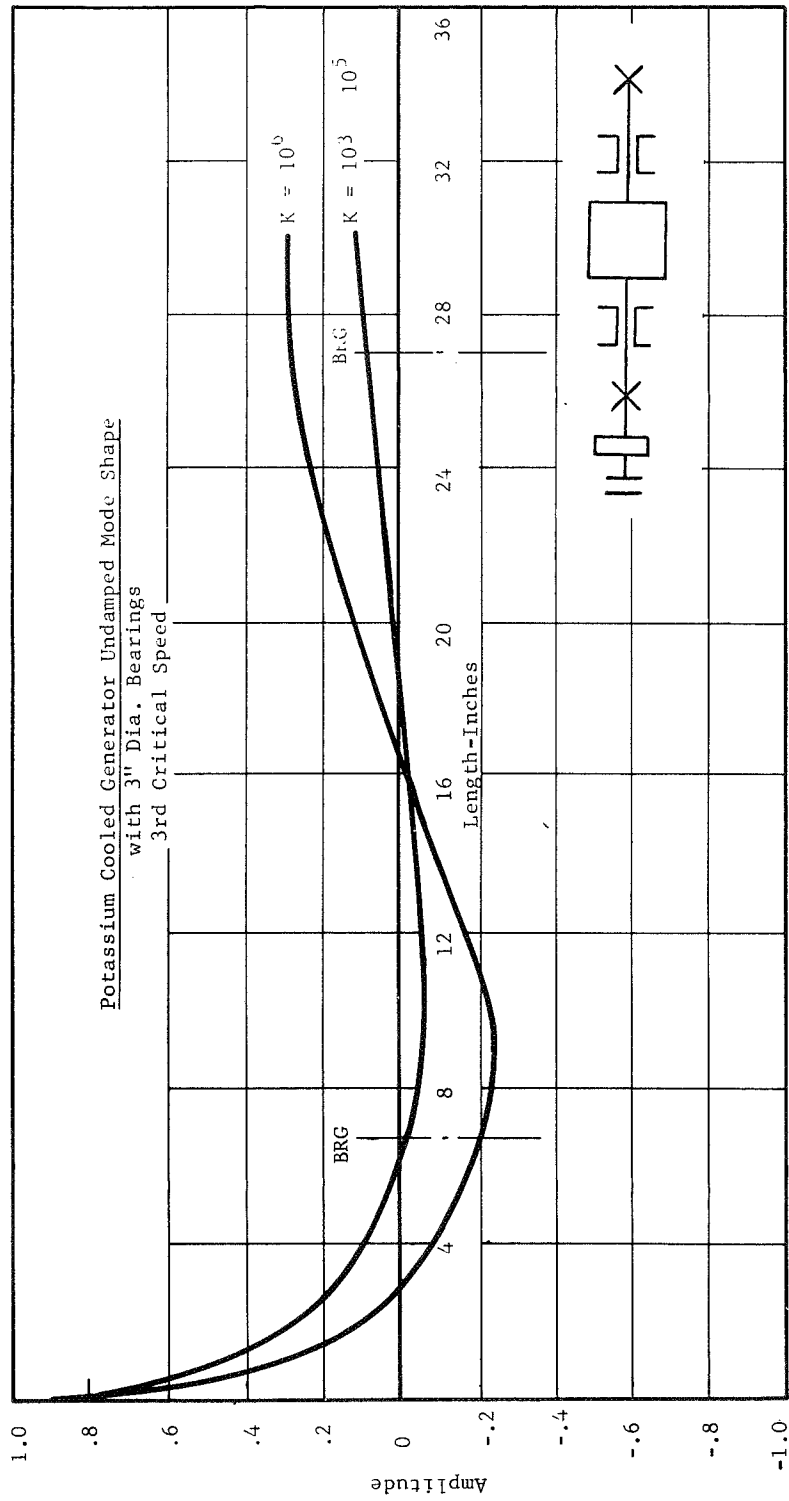


Figure 7.3-15. Potassium Cooled Generator Undamped Mode Shape with 3" Diameter Bearings. (Configuration 2)

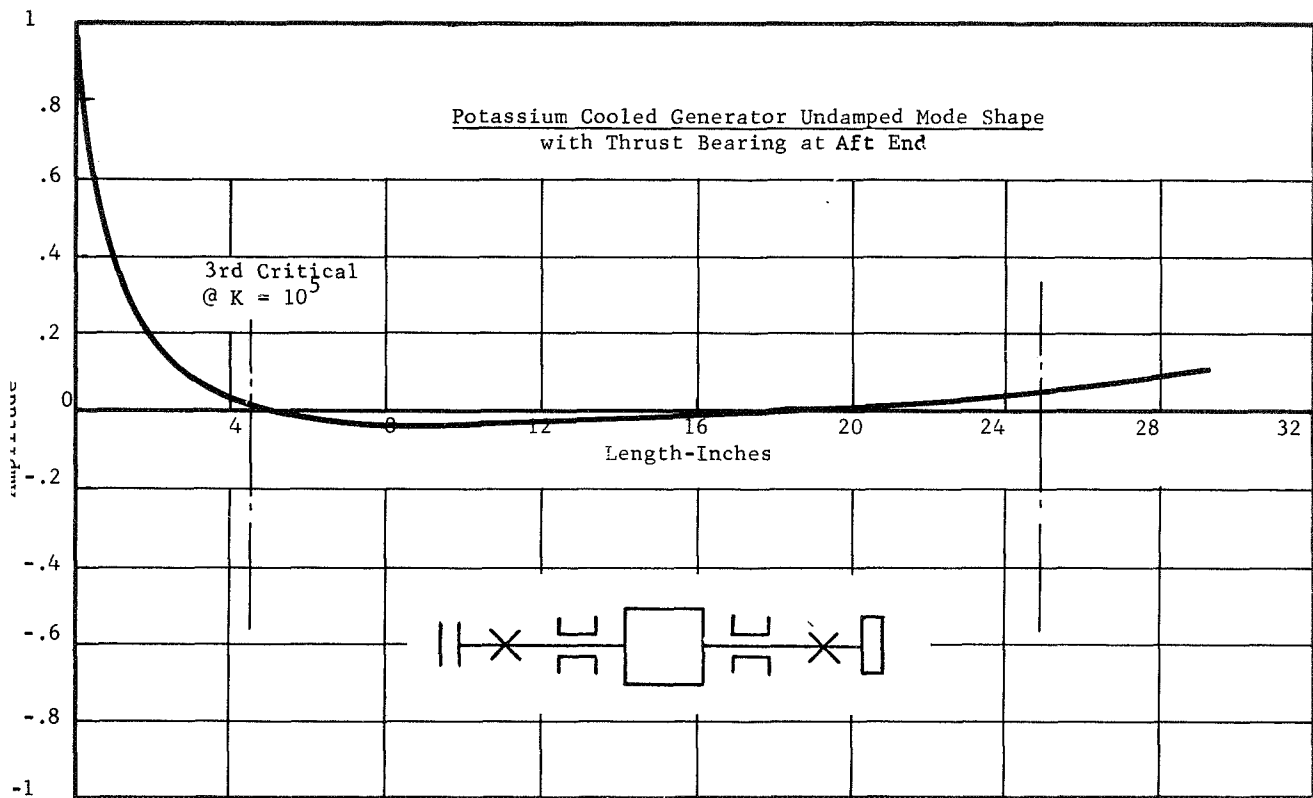


Figure 7.3-16. Potassium Cooled Generator Undamped Mode Shape with Thrust Bearing at Aft End. (Configuration 3)

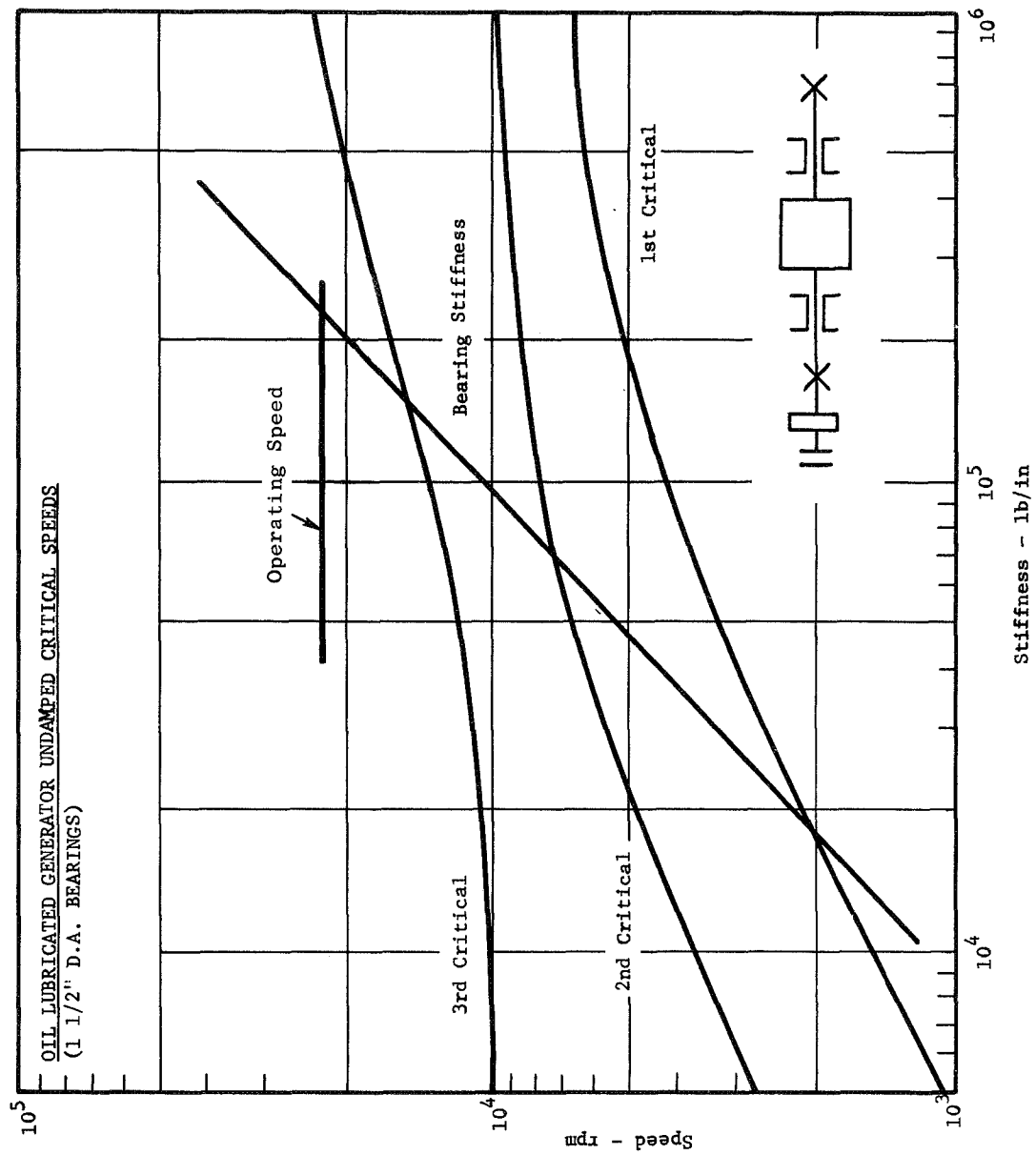


Figure 7.3-17. Oil Lubricated Generator Undamped Critical Speeds.
(1 1/2" Diameter Bearings)

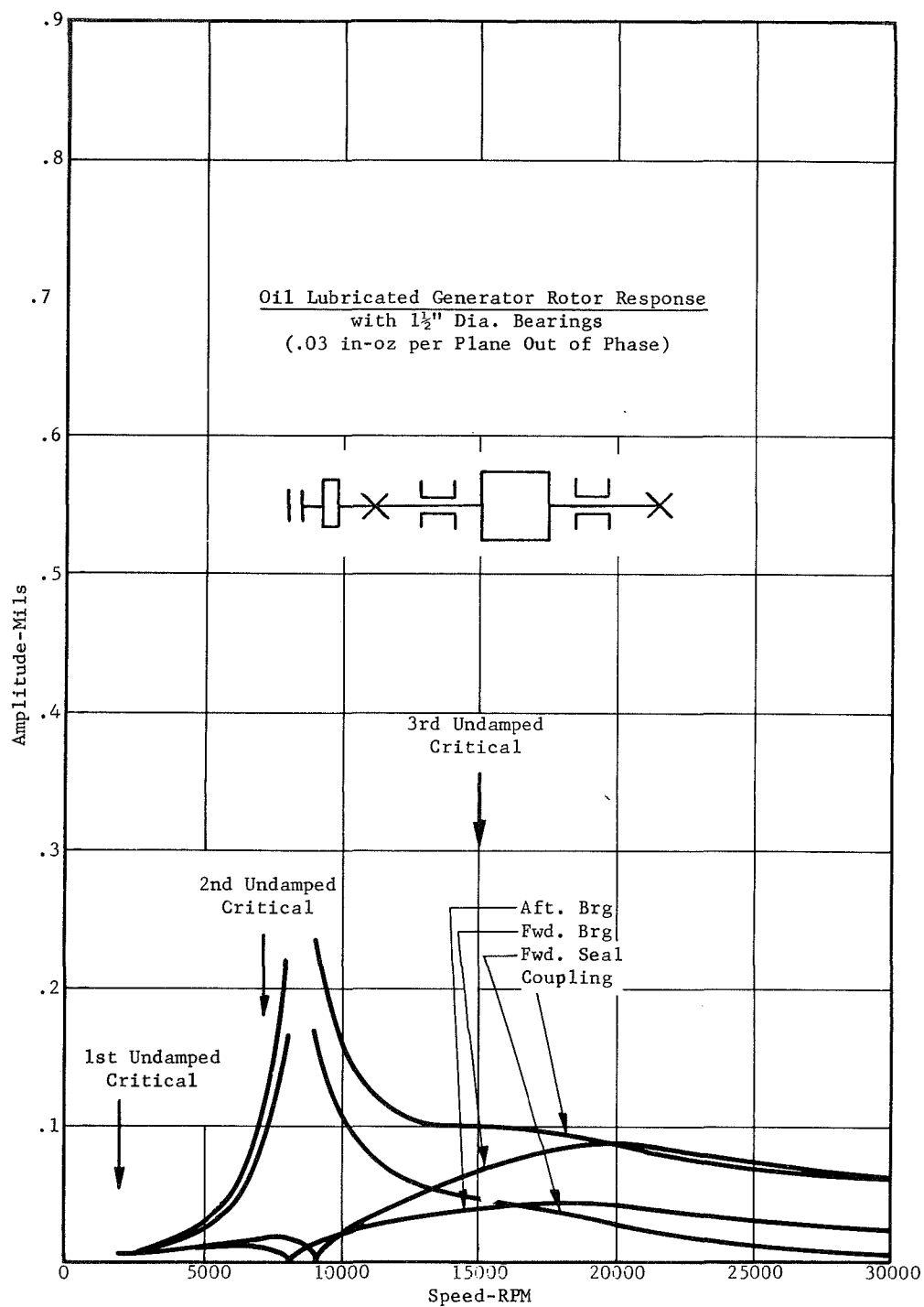


Figure 7.3-18. Oil Lubricated Generator Rotor Response with 1 1/4" Diameter Bearings.

In an effort to alleviate the problems of driving through a poorly damped second critical speed, the stub shafts were stiffened by increasing the bearing diameter to 2.25 inches which resulted in pushing the third undamped critical to 24,000 rpm (see Figure 7.3-19). The results of the response analyses shown on Figure 7.3-20 shows the favorable improvement on the second critical, but not surprisingly, resulted in a situation very similar to the potassium cooled alternator configuration 1.

7.4 SUMMARY AND CONCLUSIONS

The rotor dynamics analyses conducted during Phase I of the KTA design study were preliminary in nature in that none of the machine concepts were optimized from a dynamics standpoint and the effects of such variables as balance location and orientation, housing flexibility, bearing tolerance and force fields in excess of zero were considered. However, all of the rotors were considered on the same basis and the results have given considerable insight into the best potential arrangements for the KTA components, have considerably aided in the selection of the most promising concepts, and have provided the direction that is necessary to improve the designs in Phase II.

The potassium lubricated turbines generally showed smaller amplitudes of vibration than did the oil lubricated turbines primarily because the oil turbines were troubled with very compliant stub shafts as a result of the long complex oil seals at the bearings. Although the oil lubricated turbines had maximum amplitudes of vibration approaching 0.25 mils within the design speed range to 24,000 rpm as compared to amplitudes in the potassium lubricated turbines which were,

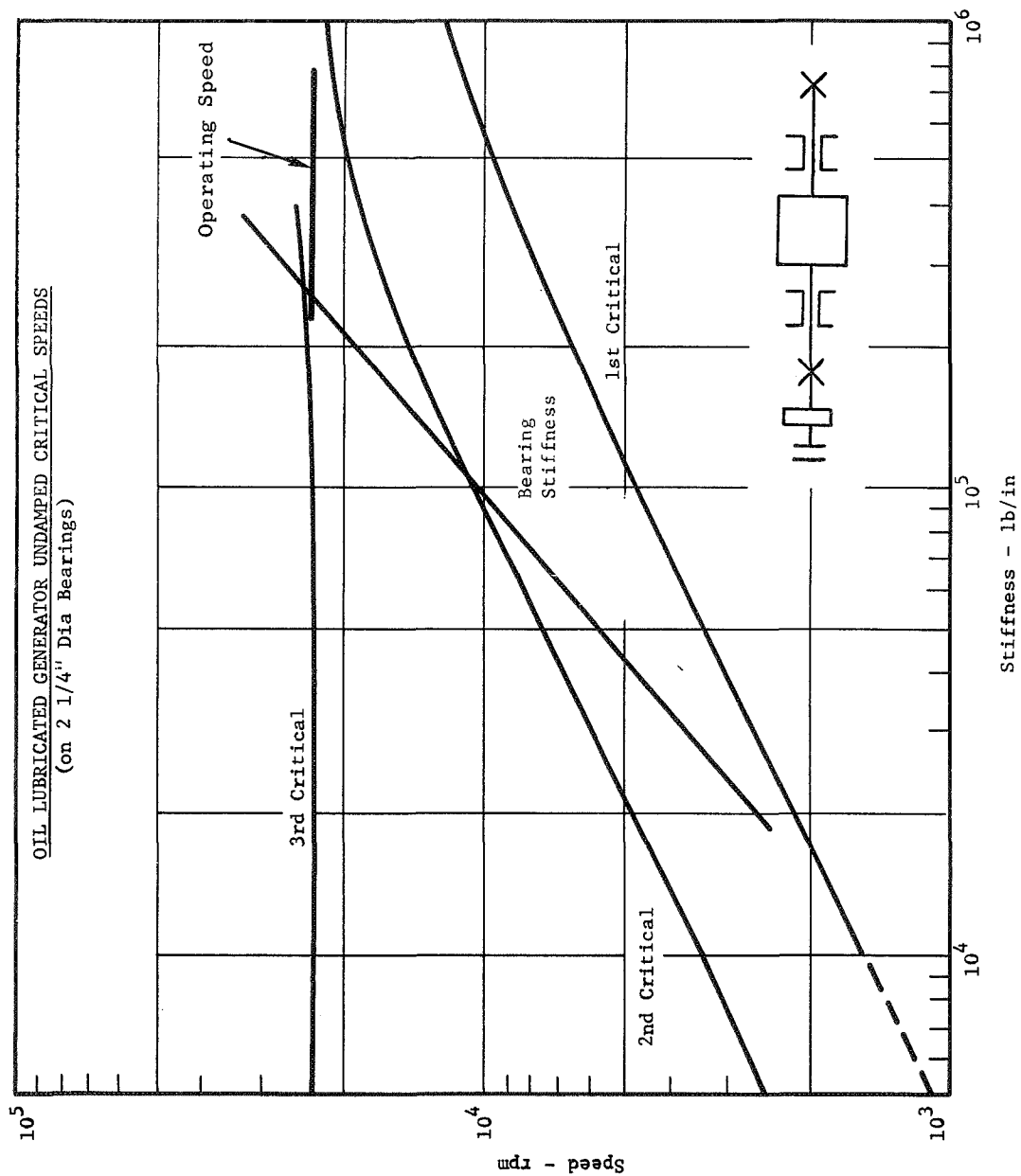


Figure 7.3-19. Oil Lubricated Generator Undamped Critical Speeds.

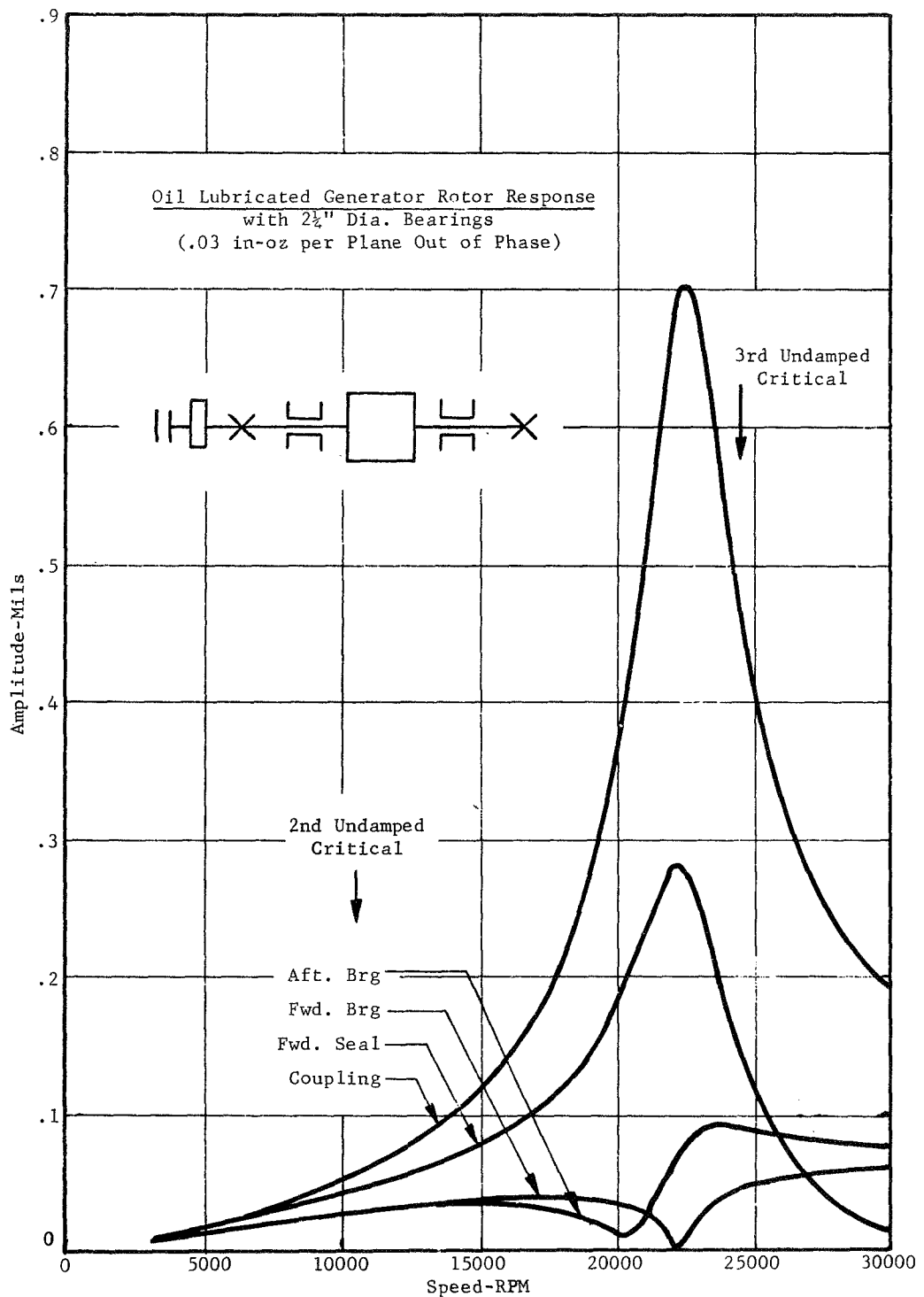


Figure 7.3-20. Oil Lubricated Generator Rotor Response with 2 1/4" Diameter Bearings.

in all cases, less than 0.1 mils, the oil lubricated turbines are still, from a dynamics standpoint, feasible. Maximum amplitudes of 0.25 mils, in itself, are not unreasonable and furthermore, the way to reduce the resonance peaks is clear. This can readily be accomplished by increased bearing and stub shaft size and possible modification of bearing location with respect to the nodes. Both of these changes may, however, compromise seal performance and power consumption.

The potassium lubricated turbines that had some stages overhung tended to have smaller vibration amplitudes than did the straddle-mounted arrangements. For example, the maximum amplitude noted with the nine stage overhung arrangement was approximately 0.03 mils as compared to the ten stage straddle-mounted arrangement which had maximum amplitudes of about 0.07 mils at 20,000 rpm. This phenomenon can be explained by the desirable nodal location of the rotors studied that have non-uniform mass distributions. Generally speaking, these rotors had mode shapes at the more troublesome bending criticals which had large displacements near the bearings and small displacements at the heavier portions of the rotor which resulted in effective use of the bearing damping potential and minimization of the centrifugal force driving potential of the heavy rotor sections.

Both the oil and potassium lubricated alternator rotors had very large vibration amplitudes compared to the turbines. The maximum amplitudes were always at the drive coupling and were usually of the order of 1 to 3 mils. Resonance peaks tended to be poorly damped and were usually within the speed range of interest. In all cases the problem with the alternators stems from the relatively weak stub

shafts which are required to support a heavy and concentrated rotor mass. Minor modifications of the stub shafts had little beneficial effect and it must be concluded that a sizable increase in stub shaft section modulus will be required to eliminate the alternator resonance problem. This was demonstrated with the potassium cooled alternator configuration No. 4 in which 3.5 inch diameter bearings were incorporated.

7.5 REFERENCES

- 7.1-1. Prohl, M.A., "A General Method for Calculating Critical Speeds of Flexible Rotors," Journal of Applied Mechanics, Vol. 67, 1945, pp. A142-A148.
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8. TURBOALTERNATOR INTEGRATION

Design studies of alternators, turbines, couplings, bearings and seals have been presented above. The selection of two single-spool and two split-spool KTA designs, one each for organically lubricated and potassium lubricated bearings was made on the basis of the aforesaid component design studies. Each of these four designs was provided with four journal bearings. The selected lubricant supply temperatures were 400^oF for the organic lubricant and 800^oF for potassium. The use of four bearings facilitates the development of the turbine and alternator separately. A flexible-diaphragm coupling requiring no lubrication is provided to join the two components. The housing around this coupling in all cases is isolated from the bearings by two screw seals having liquid traps and molecular pumps (also screws). The housing itself is evacuated by means of the injection pump which also evacuates the alternator rotor cavity and supplies internal coolant to the alternator rotor. On all of these designs internal condensate removal through stator blade trailing edge slots is possible without modification of the casing.

In all cases the four-bearing KTA designs included a light, rigid, tubular frame made from the same material as the alternator casing, titanium for the organically lubricated bearings and Hastelloy-C for the potassium lubricated bearings. The titanium results in a 30 percent weight saving for the same strength compared to steel. The

Hastelloy-C has a coefficient of thermal expansion not too different from that of the turbine materials. The frames are designed to operate at lubricant temperature (400°F for titanium and 800°F for Hastelloy-C) by having thermal radiation shielding outside the frame and the lubricant circulating inside.

8.1 ORGANICALLY LUBRICATED SINGLE-SPOOL KTA

In selecting the turbines for organic lubrication the main criterion was not to have the bearings and seals near any high-temperature potassium vapor. Consequently, selected for the single-spool turbine was the eight-stage split-flow arrangement, straddle-mounted between two bearings which are both exposed to potassium vapor at turbine outlet temperature. Shown in Figure 8.1-1 is a drawing of this KTA design which has a rotative speed of 24,000 rpm. Because of the split-flow arrangement of the turbine, no high-pressure seal is required. The symmetric bearing and seal arrangement near the turbine exits results in low temperature gradients in the shaft. There is ample space in this design for adequate turbine exit diffusers and scrolls. The modest tip speed (766 ft/sec) of the turbine is beneficial to resistance to erosion. The use of steel in the shaft, bearings and bearing housings avoids the problem of differential thermal expansion. Due to the high speed (24,000 rpm) the rotor of the alternator is 120 pounds lighter than the lower speed (19,200 rpm) potassium lubricated KTA. The titanium casing saves 280 pounds of alternator weight compared to a steel casing.

This KTA design does have a significant number of disadvantages. The seal between potassium and the organic lubricant presents a high

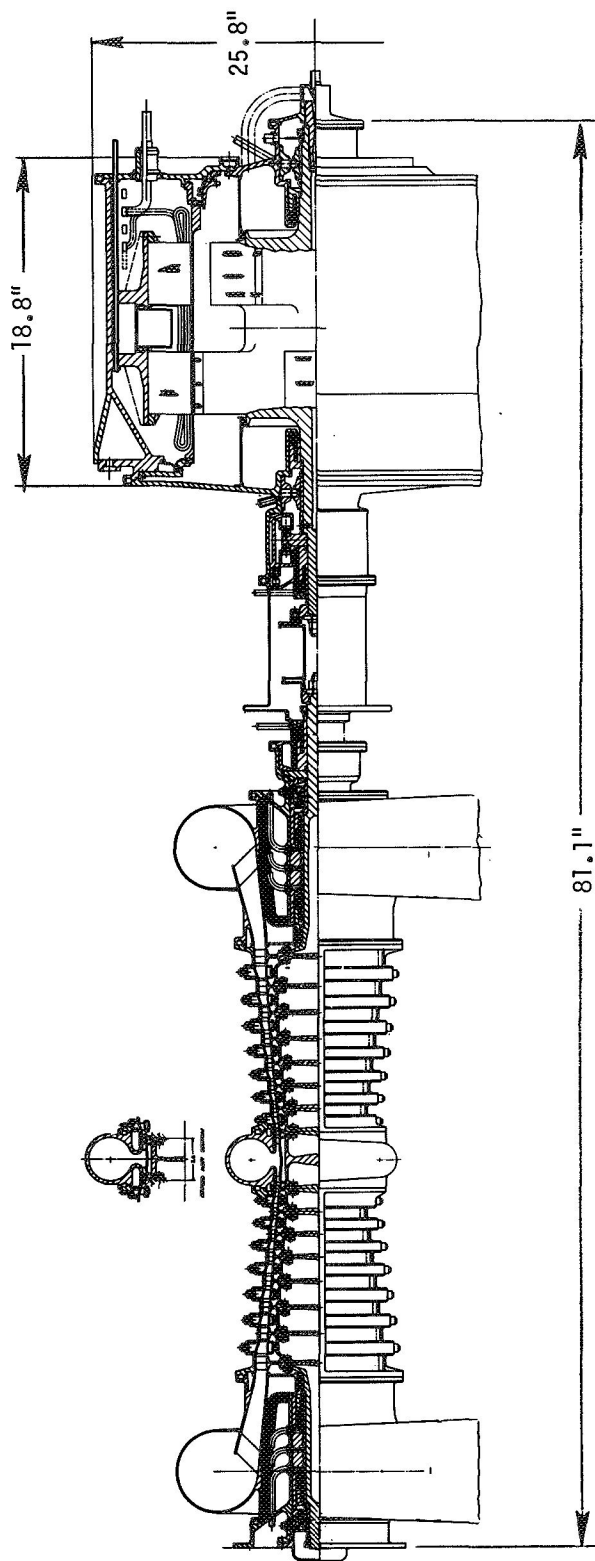


Figure 8.1-1. KTA - 8 Stage, 24,000 rpm Split-Flow Turbine. Turbine and Alternator on Organic Bearings, 400°F.

risk: it must be vented to space precluding hermetic sealing of the KTA. In addition, the maximum bending moment in the turbine rotor is at its center where it is exposed to turbine inlet temperature, resulting in poorer rotor response than other designs. Torque-tube stresses in this region approach the allowable values. The turbine is complex because it contains sixteen stages while only getting the performance of an eight-stage turbine. Although not shown in the figure, this KTA has a complicated frame weighing (137 lb) more than any other design between turbine and alternator.

8.2 ORGANICALLY LUBRICATED SPLIT-SPOOL KTA

For the 24,000 rpm, split-spool turbine design with organic lubrication, the KTA design which exposes the bearing and seal areas to the lowest potassium temperatures was selected. This design, which has a five-three split in the number of stages, is presented in Figure 8.2-1. The forward bearing is adjacent to potassium vapor at about 1560°F instead of turbine inlet temperature and the aft bearing is adjacent to 1280°F vapor, resulting in acceptable shaft thermal gradients. There is ample space available for all scrolls and ducts including the high-volume-flow exit scroll. Because of the split-spool arrangement, an external separator can be fitted, reducing the blade erosion potential. A higher tip speed (828 ft/sec) than for the previous single-spool design is justified because of the external separator.

This KTA design has many of the same disadvantages as the organic split-flow design, including those associated with the seals between potassium vapor and the organic lubricant, those connected with the

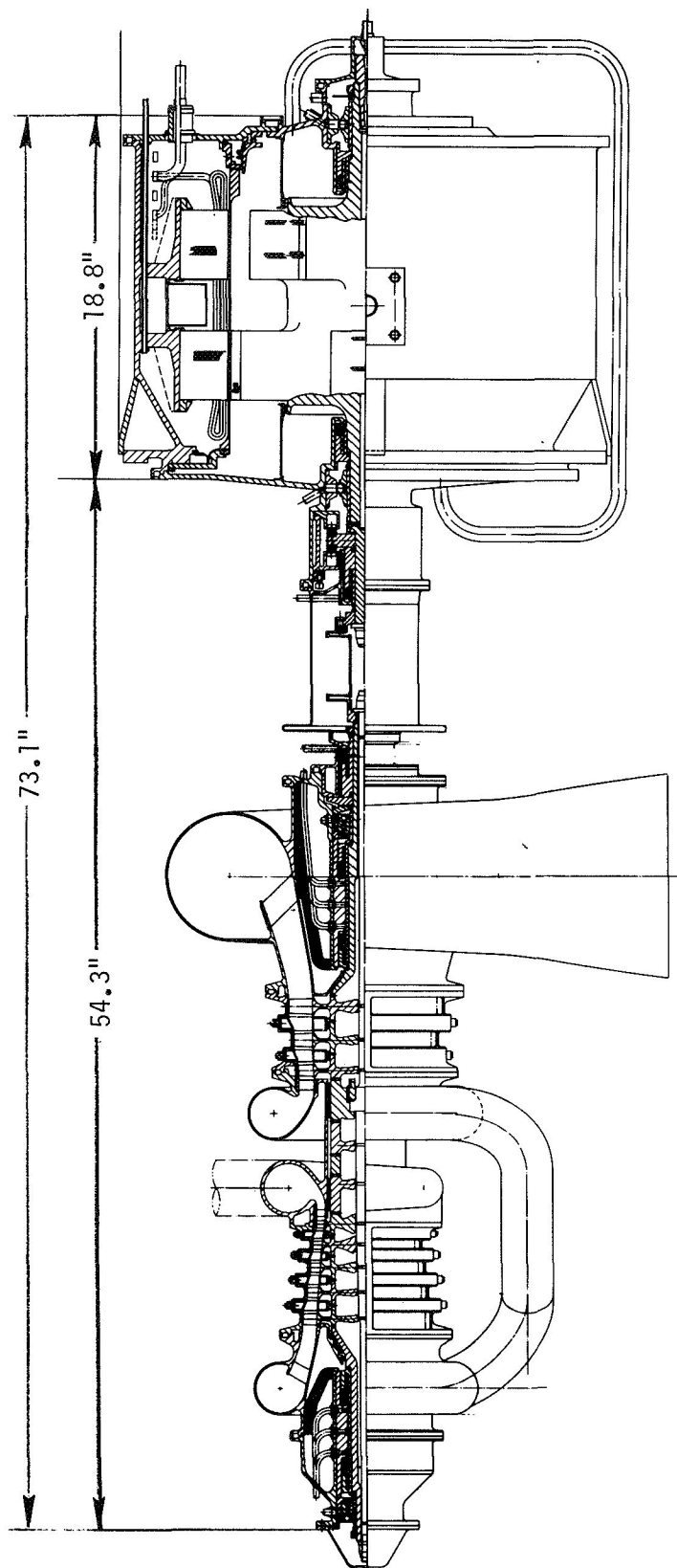


Figure 8.2-1. KTA - 8 Stage, 24,000 rpm Split-Spool (5+3) Counterflow Turbine. Turbine and Alternator on Organic Bearings, 400°F.

center-section torque tube being exposed to vapor at turbine inlet temperature and a complex frame between turbine and alternator. In addition, the losses associated with the high-pressure vapor seal in the turbine are high due to the large diameter required for rotor rigidity. Also, the curvic coupling provides a vapor leakage path between spools. Finally, complex external ducting is required for accommodating the external separator.

8.3 POTASSIUM LUBRICATED SINGLE-SPOOL KTA

For the potassium-lubricated turboalternator, a nine-stage overhung turbine was selected as the most suitable contender for a single-spool arrangement. This design, which has a rotative speed of 19,200 rpm, is shown in Figure 8.3-1. This arrangement eliminates the high-pressure seal. It minimizes temperature gradients in the bearing and seal areas and it is outstanding as far as rotor bearing response is concerned. Mechanically, it is the most attractive arrangement. It results in minimum inlet ducting and special freedom in outlet ducting permitting a long and straight diffuser. The frame connecting the turbine with the alternator is rigid and minimal. Three interchangeable rotors are offered with this turbine. The first choice is the integrally welded rotor using ASTAR 811C and columbium FS-85 as the materials. It is believed to be the most stable design and the only one which eliminates internal vapor leakage. This machine is hermetically sealed. The rotor, however, can be replaced by either a bolted rotor or a curvic coupling tiebolt rotor using high-strength TZC and TZM molybdenum alloy wheels, torque tubes and bolts. The turbine stator is designed so that interstage condensate extraction

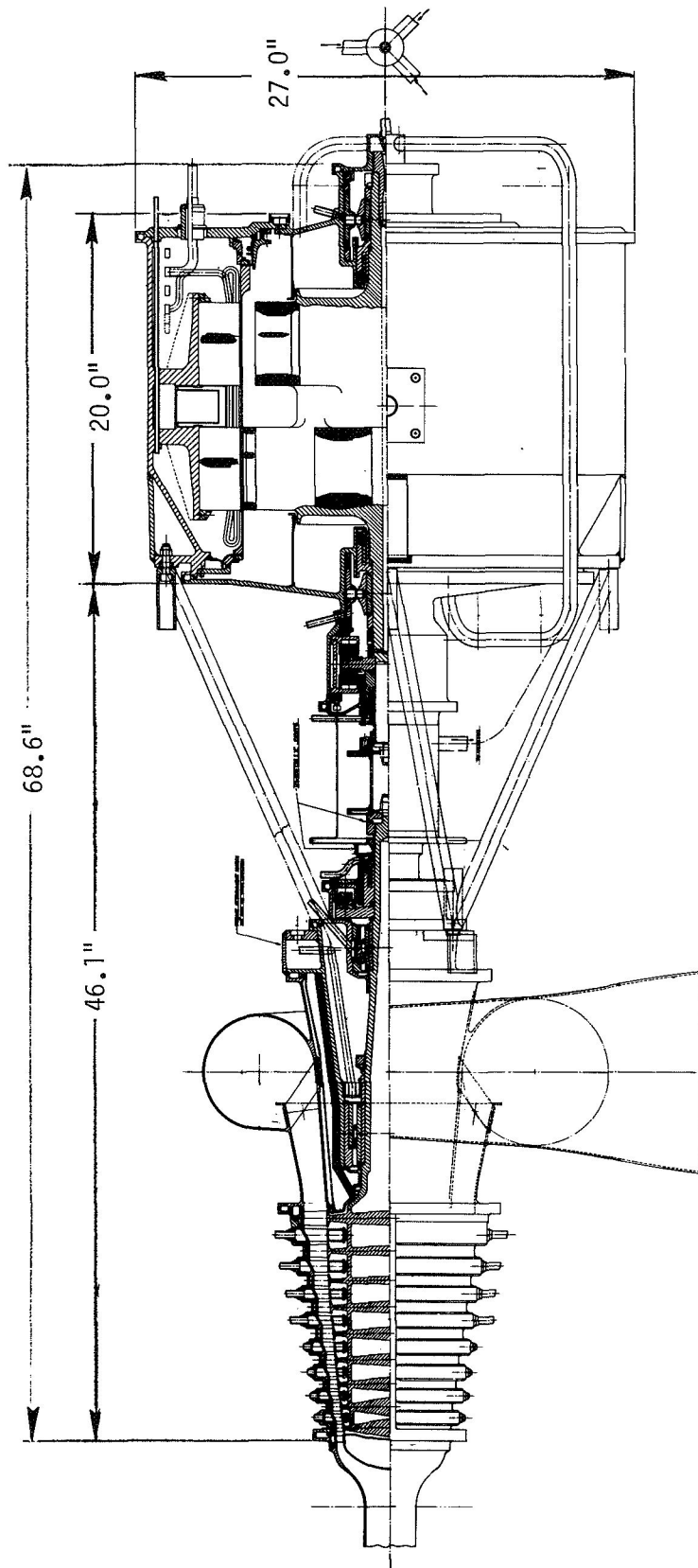


Figure 8.3-1. KTA - 9 Stage, 19,200 rpm Overhung Turbine. Turbine and Alternator on Potassium Bearings, 800°F.

from the stator can be employed if desirable. It is realized that this type of condensate extraction constitutes a development problem. However, the maximum tip speed (776 ft/sec) is modest considering erosion.

The disadvantages of this design are few. Although internal condensate removal can be provided no external removal is possible. The present configuration of the alternator has marginal rotor-bearing response.

8.4 POTASSIUM-LUBRICATED SPLIT-SPOOL KTA

In selecting the split-spool turbine for potassium-lubricated bearings, the fact that the high-pressure seal adjacent to a low-temperature bearing and seal area constitutes the biggest single problem, resulting in high loss, steep temperature gradients and unwanted heat transfer into the bearing and seal area can best be resolved by the elimination of the high-pressure seal. Consequently, a split-spool design was chosen in which the first six stages were overhung, eliminating the high-pressure seal, and four stages were straddle-mounted. This design which has a rotative speed of 19,200 rpm is shown in Figure 8.4-1. The temperature environment (1600°F) for the interspool bearing is acceptable; that for the aft bearing is close to optimum (1280°F). This turbine excels in efficiency. Its axial thrust is cancelled to quite some extent by using the counterflow arrangement. Interspool moisture separation is provided by four vortex separators. A higher tip speed (828 ft/sec) is justified for this design with respect to erosion because of the external separator. The low-exhaust preferably flows in smoothly expanding ducts, directly

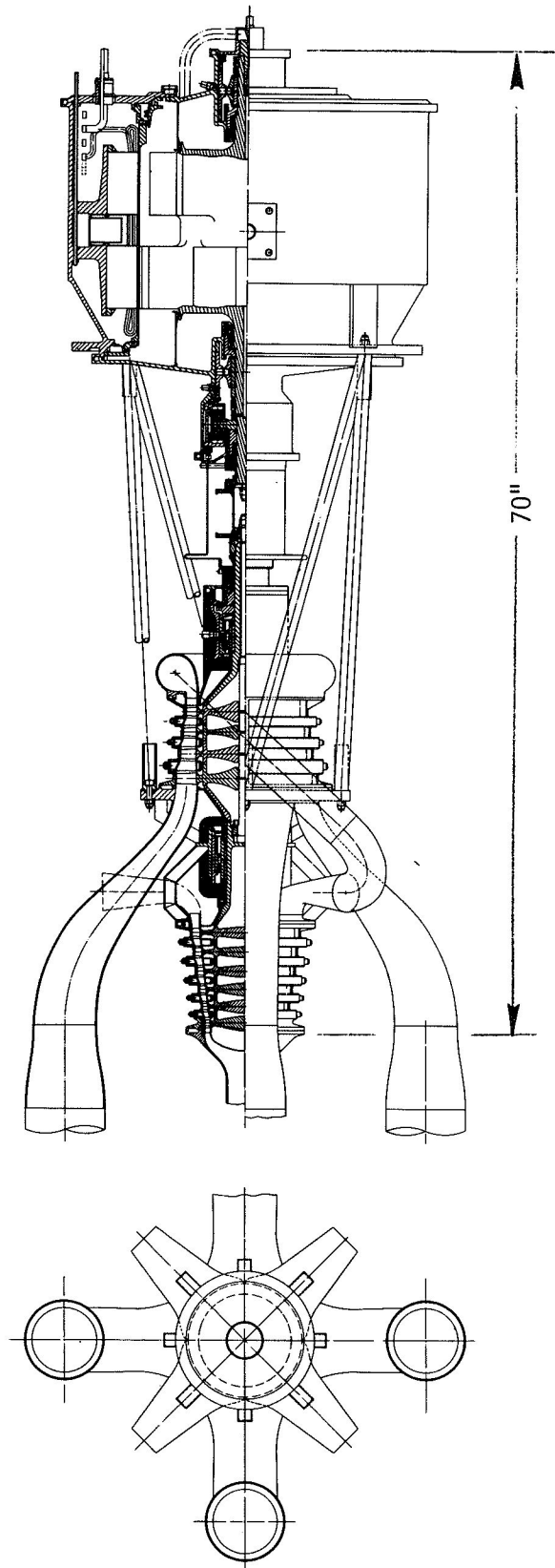


Figure 8.4-1. KTA - 10 Stage, 19,200 rpm Turbine (6 Overhung + 4 Counterflow). Turbine and Alternator on Potassium Bearings, 800°F.

into the four condensers. This turbine, like the single-spool potassium lubricated turbine, is connected with the alternator by a light-weight frame consisting of eight tubular struts and eight bolts.

One disadvantage of this design is that the forward turbine bearing support is heavy and complex. Because of the location of the forward turbine bearing between the two spools, a curvic coupling-tiebolt connection of the two spools is required to permit the bearing-pivot support ring to be monolithic. The turbine for this design contains two more internal seals than the single-spool overhung design, resulting in more complexity. The complex external ducting required for the four external separators is further complicated by a lack of space around the forward bearing for scrolls. The problem was solved by providing four outlets and inlets in this region.

8.5 POTASSIUM-LUBRICATED TWO-BEARING KTA

It has been believed originally that the rotor-bearing response of the KTA would be hardly different from its components, turbine and alternator. It has been found, however, that the addition of a coupling considerably affects the overall rotor-bearing response. In addition, severe dynamic response problems were experienced with the cradled alternator rotor, especially at its coupling end. For this reason, a two-bearing KTA has been studied, combining the nine-stage overhung turbine with the alternator on potassium-lubricated bearings. This design which has a rotative speed of 19,200 rpm is shown in Figure 8.5-1. In this approach two bearings, two zero leakage seals, the coupling and the frame have been eliminated. They have been traded for larger bearing and seal diameters, more risky and higher-stressed

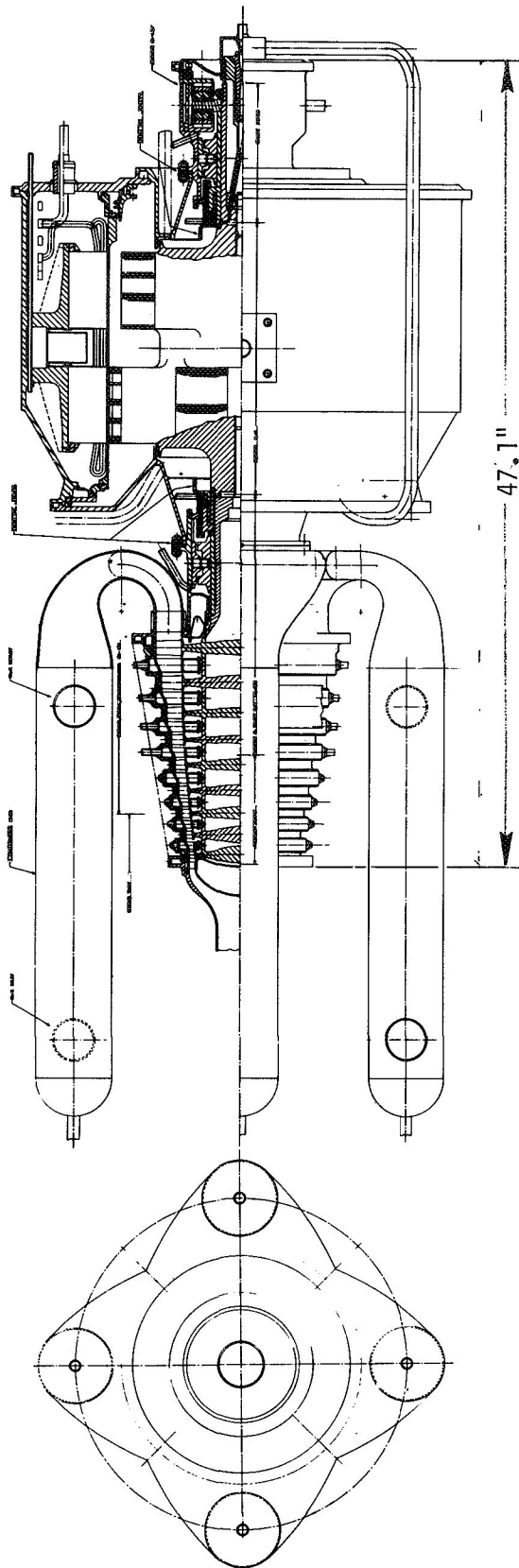


Figure 8.5-1. KTA - 9 Stage, 19,200 rpm Overhung Turbine. Turbine and Alternator on Potassium Bearings. (2-Bearing Design), 800°F.

bi-metallic joints and less freedom in exhaust ducting, precluding an exit scroll.

All refractory-metal bearings, including shaft and housing are a feature of this design. The rotor-bearing response will be shown below to be excellent for this design. As is the case for the four-bearing, single-spool, potassium-lubricated KTA there is no need for a high-pressure vapor seal in the turbine. And, as in other machines, internal condensate removal from the stators is optional.

Some disadvantages of this design include a highly stressed bi-metallic shaft connection, inaccessible bearings and seals, and no external condensate extraction. In addition, it is difficult to assemble the turbine and alternator because of having only two bearings. Also, separate development of turbine and alternator requires additional shafts and bearings and sealed casings.

9.0 TURBOALTERNATOR BEARING/ROTOR RESPONSE

In addition to the bearing rotor response analyses accomplished on individual turbines and alternators, analyses on coupled turbo-alternators were also completed. The results of these analyses serve to indicate the effect of the coupling and establish the vibrational interplay between the two rotors.

The same assumptions regarding bearings, dynamic coefficients, and amount and location of unbalance were used in analyzing the vibrational behavior of the KTA's as was used in analyzing the individual components (as was described in the preceeding chapter). The coupling was simulated in the response analysis by treating the diaphragms as short, small diameter beams having the same angular stiffness as the diaphragms described in Chapter 7. The shaft section between diaphragms was treated as conventional shafting using its exact mass and section modulus properties.

The following four-bearing KTA's were selected for bearing rotor response vibration analysis in Phase I:

<u>Drawing No.</u>	<u>Coolant</u>	<u>Turbine Description</u>	<u>Alternator Config.</u>	<u>Fig. No.</u>
47R199281	K	9 Stage Split Spool	No. 2 Similar to	8.4-1
	K	9 Stage Overhung	No. 2	8.3-1
47R199284	Oil	8 Stage Split Flow	2.25 Inch Brgs.	8.1-1

The results of the analyses of the several KTA rotor response analyses are shown on Figures 9.0-1 to 9.0-6. For each KTA configuration, the response of the turbine and alternator portions are shown on separate figures and can be compared directly with the individual components discussed in the preceeding chapter.

Examination of these curves indicate that the vibrational behavior of the KTA components are pretty much the same coupled together as when they were analyzed independently. The coupling does, however, appear to effectively stiffen the alternator shaft as can be seen by the apparent increasing of the speed at which resonance (3rd critical) occurs in the coupled alternators. The larger alternator coupling maximum amplitudes evident in all cases of the coupled components is caused by the higher speed at which resonance occurs (centrifugal forces are larger) and by the increased centrifugal force at the coupling end caused by the addition of the mass of the coupling between diaphragms. (Only half of this mass was included in the analysis of the independent components.)

The potassium lubricated turbines show only small differences in vibrational behavior when coupled to the alternators up to the speed of alternator resonance. As would be expected, this alternator resonance is felt in the turbine, but the coupling does a reasonably effective job of isolating the two as can be seen by the relatively small turbine vibration amplitudes (compared to alternator amplitudes) in the vicinity of alternator resonant speeds. The oil lubricated turbine appears to be almost unaffected even at alternator resonance. This phenomenon has been tentatively attributed to coincidental combination of desirable mode shape and unbalance orientation between

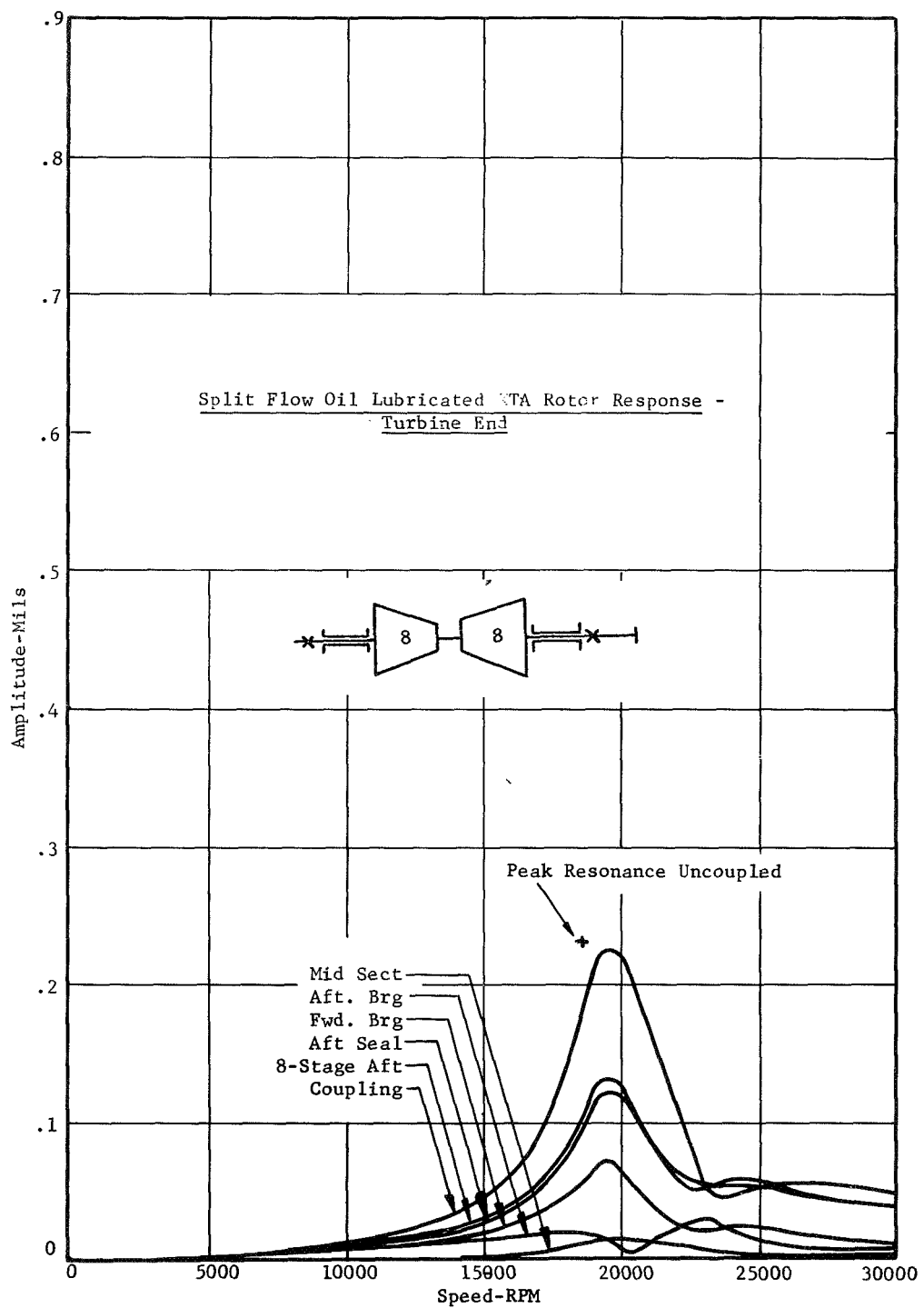


Figure 9.0-1. Split Flow Oil Lubricated KTA Rotor Response - Turbine End.

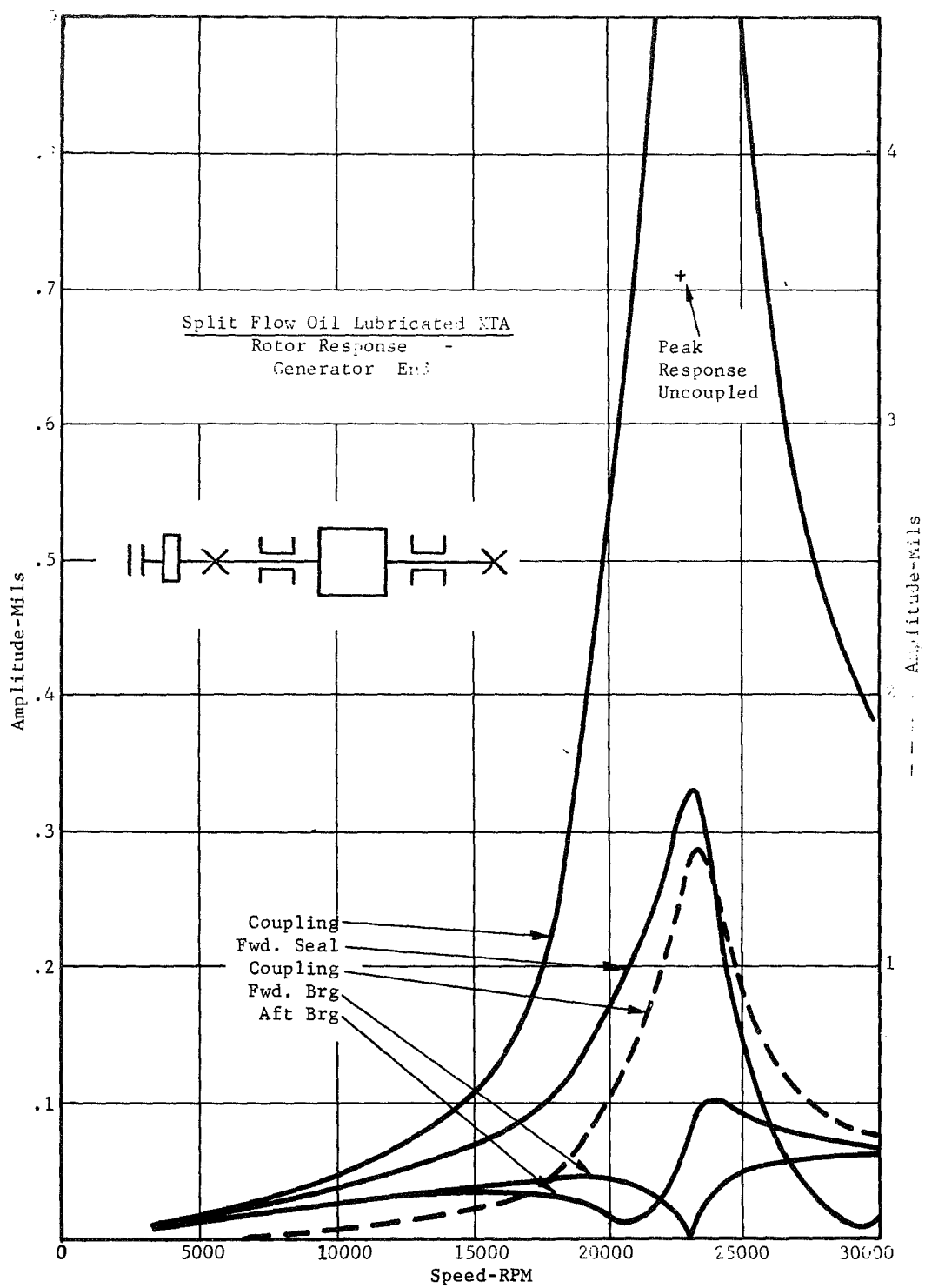


Figure 9.0-2. Split Flow Oil Lubricated KTA Rotor Response - Generator End.

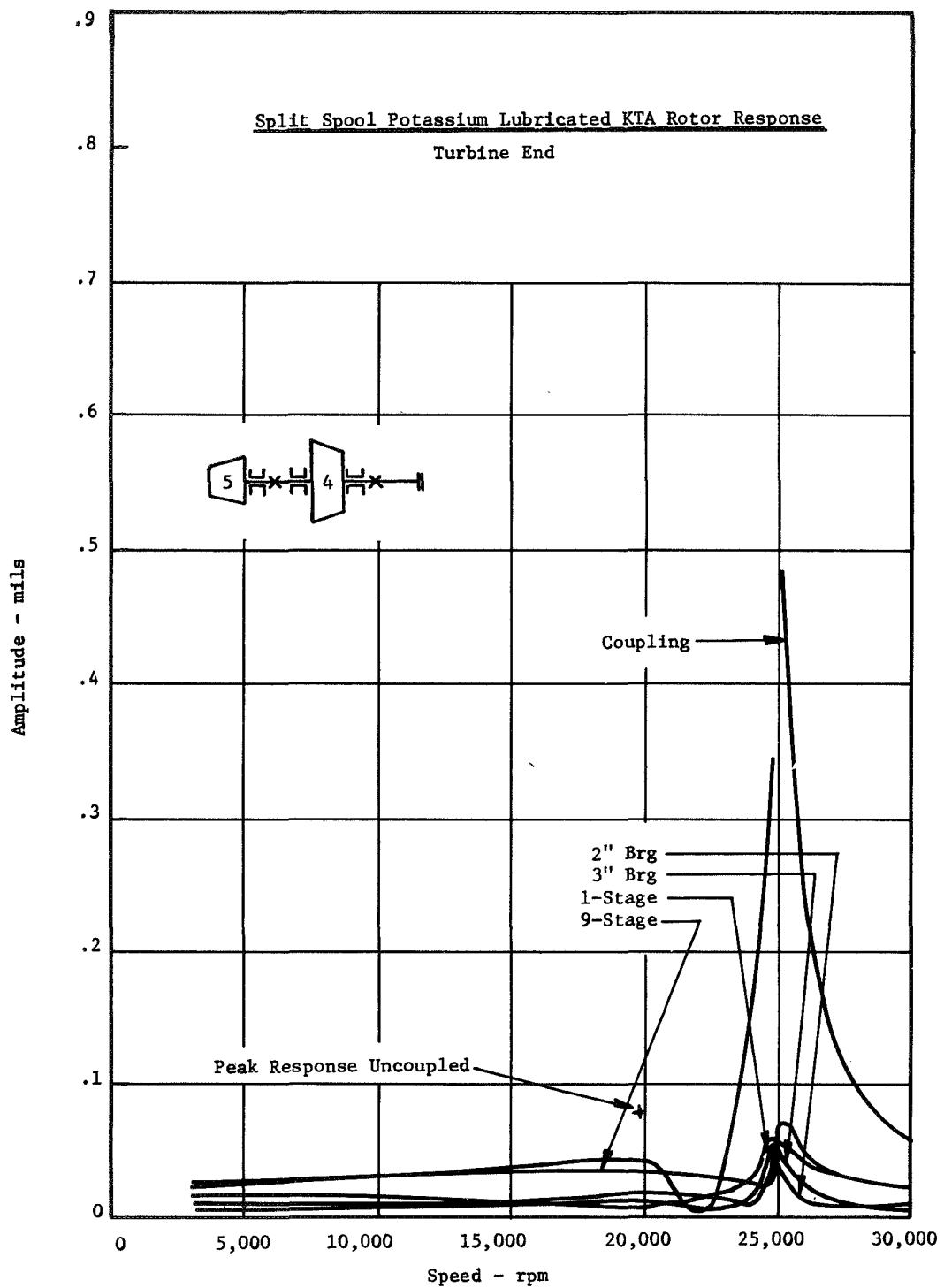


Figure 9.0-3. Split Spool Potassium Lubricated KTA Rotor Response - Turbine End.

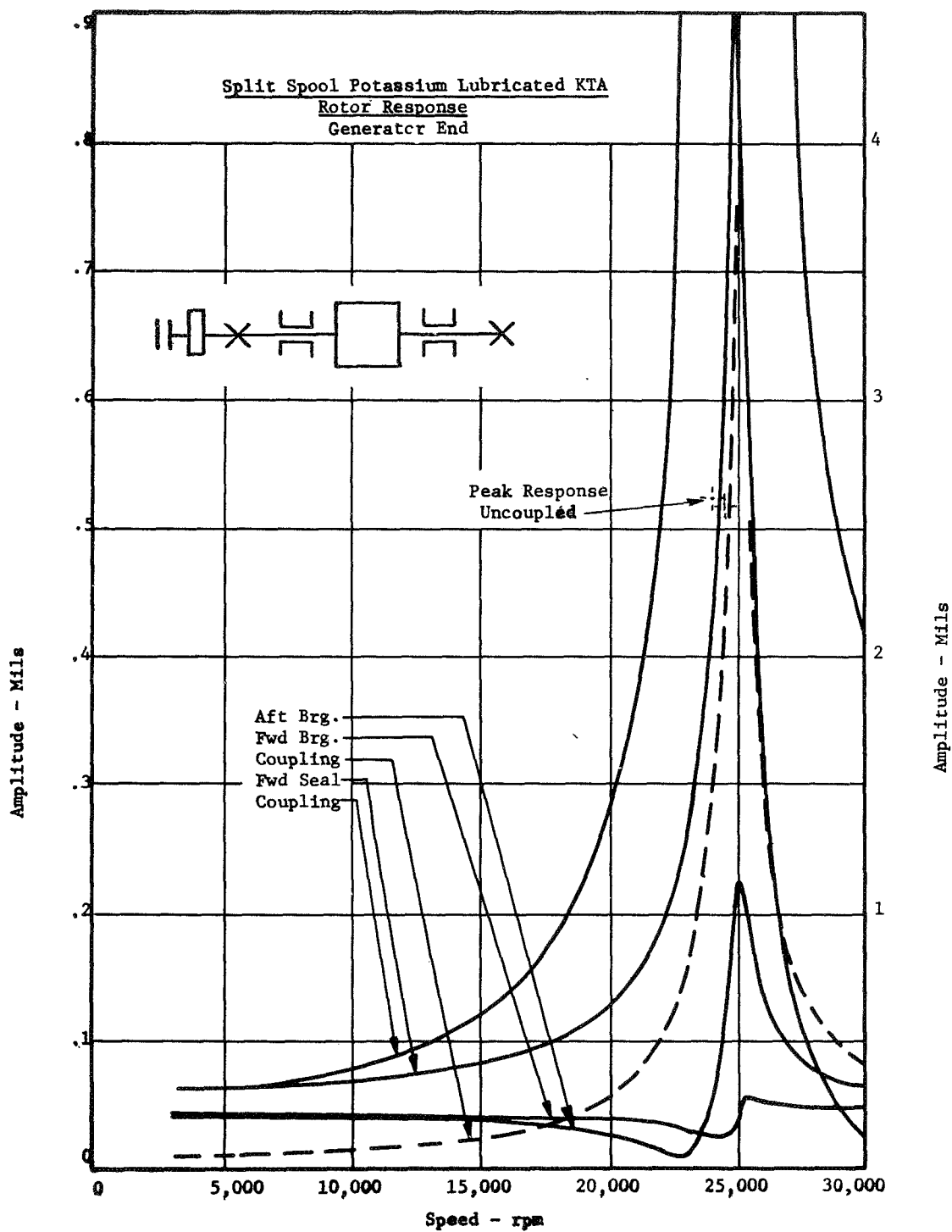


Figure 9.0-4. Split Spool Potassium Lubricated KTA Rotor Response - Generator End.

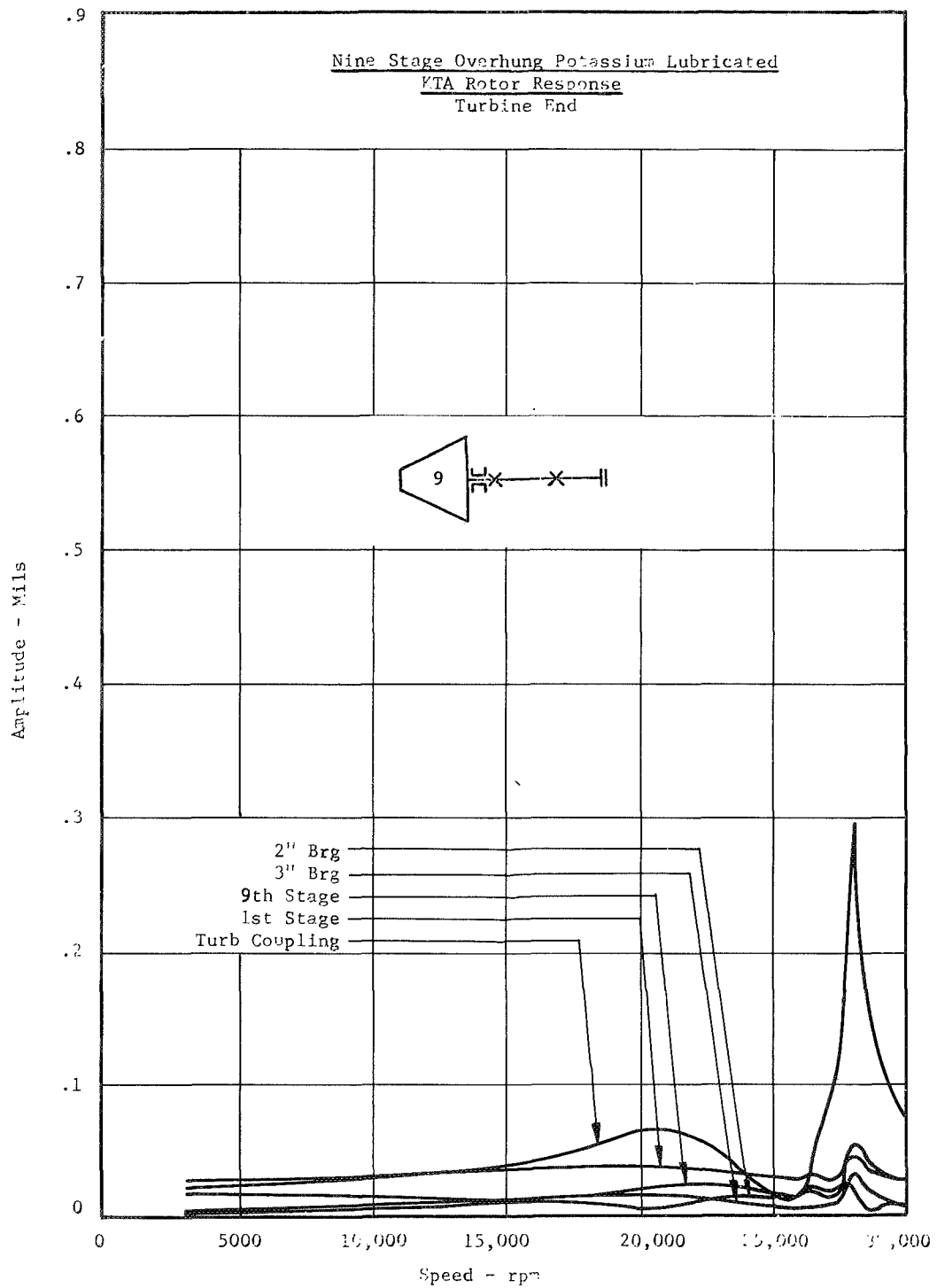
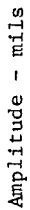


Figure 9.0-5. Nine Stage Overhung Potassium Lubricated KTA Rotor Response - Turbine End.



Amplitude - mils

the two shafts which makes the turbine coupling insensitive to alternator coupling excitations. Should this KTA be selected for refinement during Phase II of the study, this unusual characteristic will be further studied.

It has been noted several times that the alternator has an inherent problem with large vibration amplitudes near the bending criticals and that similar problems were encountered on several of the straddle mounted turbine arrangements studied early in Phase I. Furthermore, most of the machines studied with non-uniform mass distributions showed very mild vibration amplitudes at the point of the calculated undamped critical speeds. As a result of these several machine arrangement studies, it was postulated that a two-bearing KTA with an overhung turbine could be designed which would solve the alternator vibration problem that is inherent with small stub shafts. Such a machine also could show a significant reduction in parasitic losses because it requires only half the bearings and seals required in the KTA's evaluated so far.

A preliminary two-bearing KTA (Drawing No. 47R199285), Figure 8.5-1, was subjected to critical speed and rotor response analyses and the results are shown on Figures 9.0-7 to 9.0-9. The third undamped critical speed is approximately 21,000 rpm and has its nodes at about the center of the turbine and alternator masses (see Figure 9.0-8) as would be expected for this type of arrangement. For the rotor response calculation the orientation of unbalance at the planes of the turbine inlet and alternator rotor ends was such as to accentuate this mode (the unbalance orientation is indicated in Figure 9.0-9). The rotor amplitudes resulting from unbalance forces

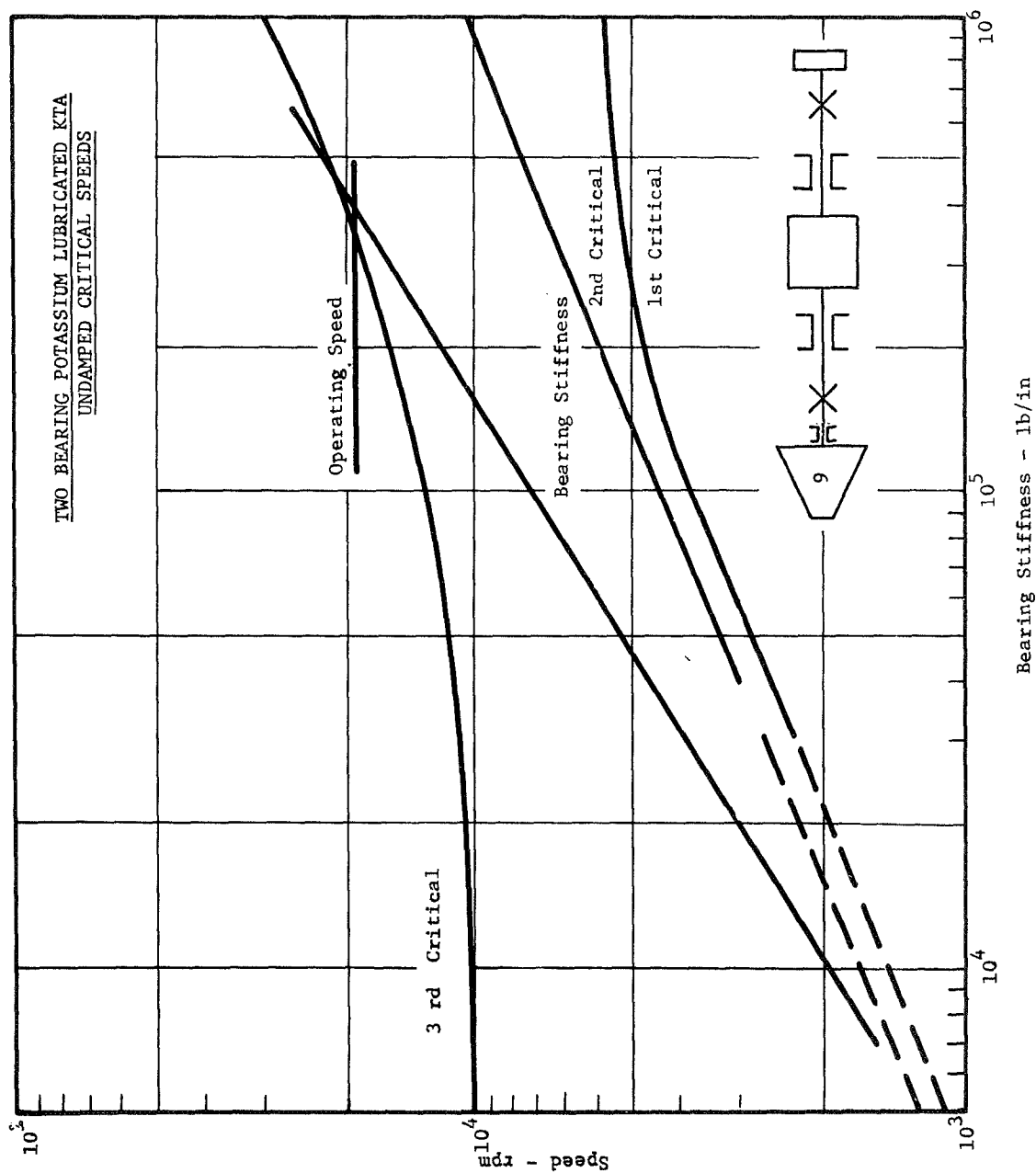


Figure 9.0-7. Two Bearing Potassium Lubricated KTA Undamped Critical Speeds.

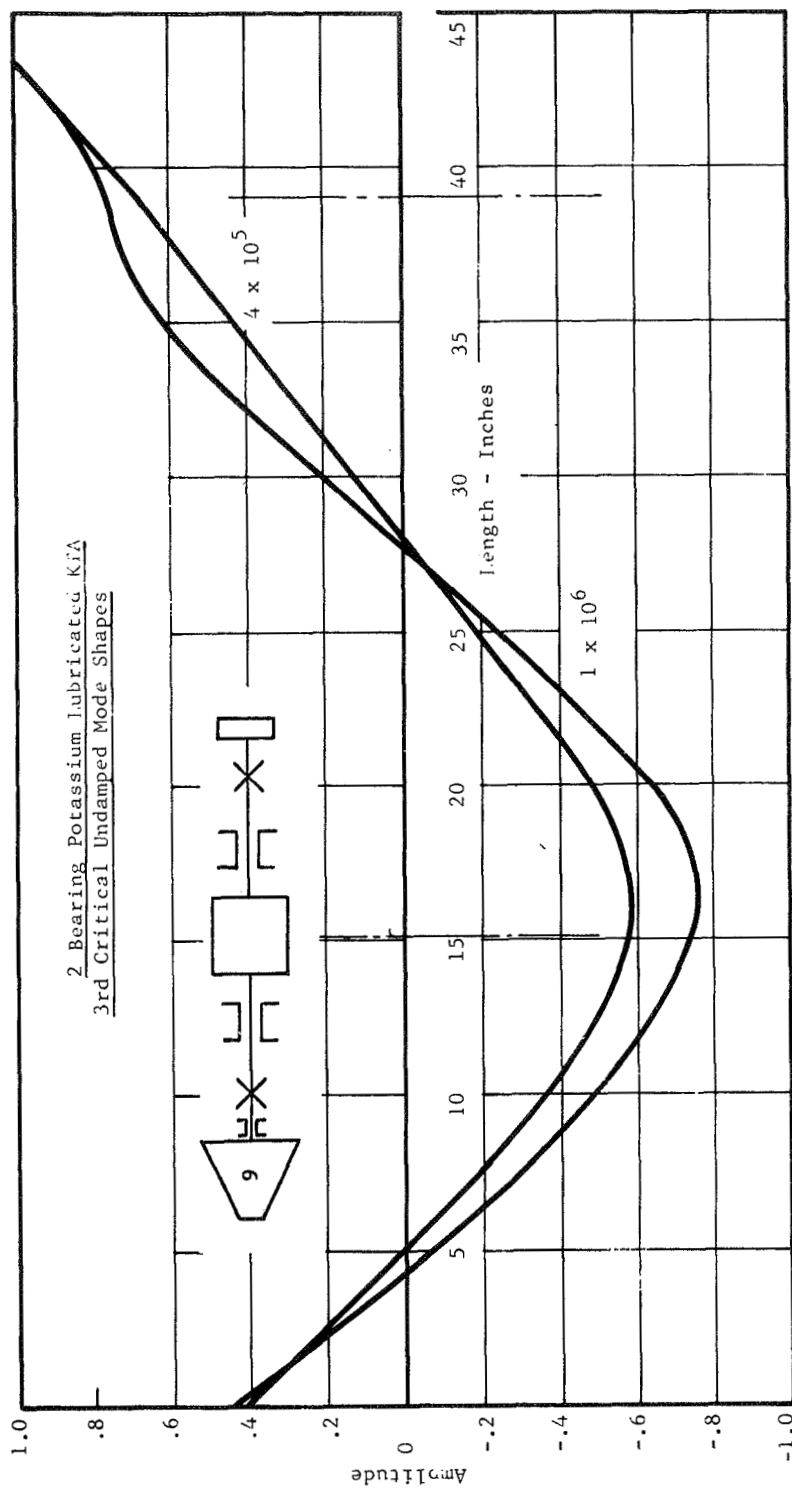


Figure 9.0-8. 2 Bearing Potassium Lubricated KTA - 3rd Critical Undamped Mode Shapes.

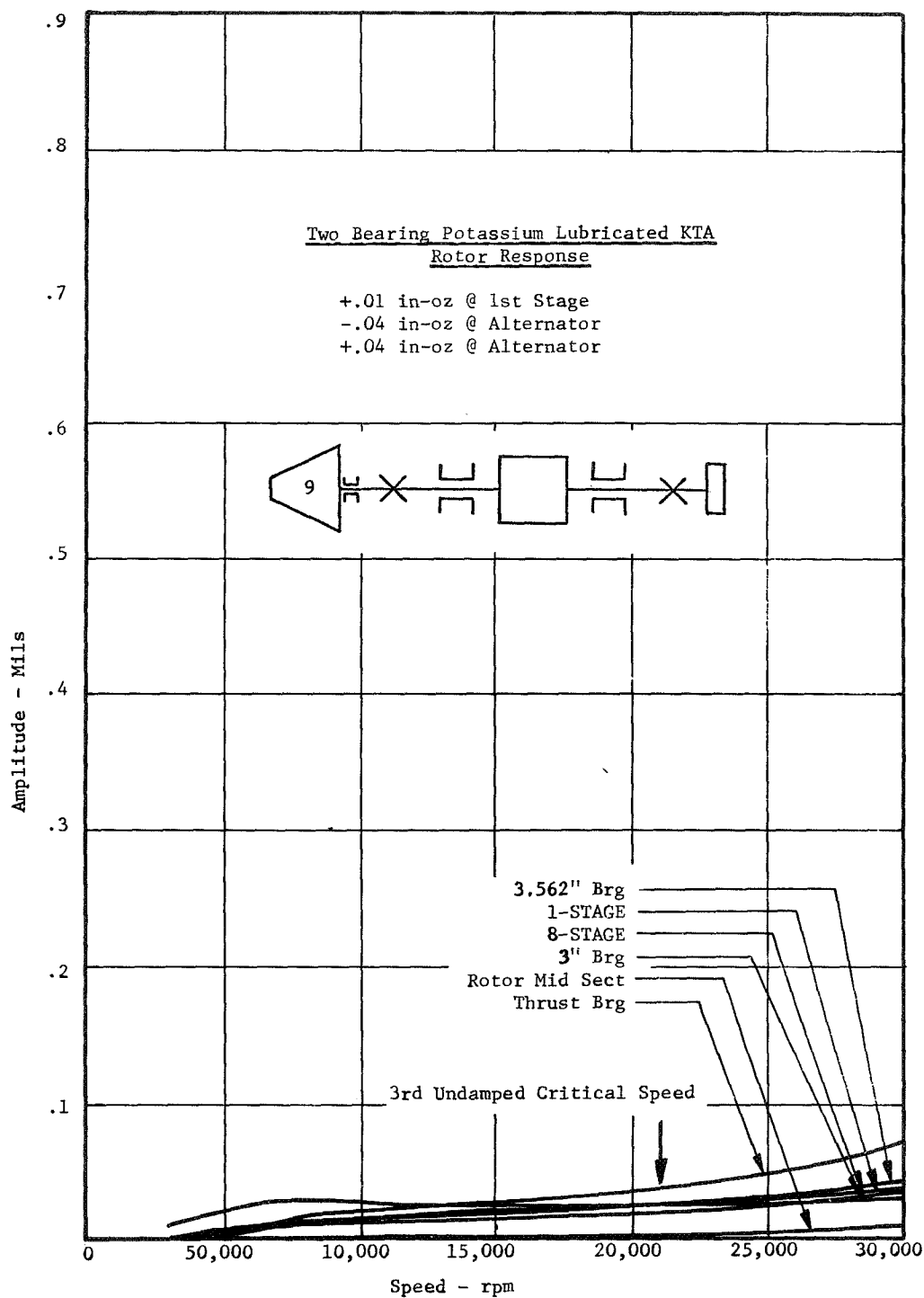


Figure 9.0-9. Two Bearing Potassium Lubricated KTA Rotor Response.

shown on Figures 9.0-9 indicate no evidence of a resonance which again clearly demonstrates the effectiveness of bearing damping when the bearings are far displaced from the nodes.

9.1 SUMMARY AND CONCLUSIONS

The analysis of the coupled KTA's demonstrated the effective isolation afforded by the diaphragm coupling. When coupled together the turbines and alternators behaved dynamically about the same as they did independently. That is resonant frequencies and amplitudes were of the same order. As would be expected, the alternator resonances, which were relatively large, were seen in the turbines but were greatly attenuated by the coupling.

The two-bearing KTA studied shows promise in that it has a very mild vibrational behavior when compared on the same basis as the four-bearing coupled KTA's. Although this KTA had an undamped critical speed at 21,000 rpm, the response analysis showed no resonance peaks and the maximum amplitude of vibration anywhere on the shaft never exceeded 0.07 mils at speeds to 30,000 rpm.

10. TURBOALTERNATOR EVALUATION

By contract the KTA designs are to be evaluated by the following measured of excellence, in descending order of priority:

- High reliability
- Ease of development
- Maximum energy conversion efficiency
- Minimum weight

10.1 RELIABILITY

Shown in Table 10.1-1 are a number of items which significantly affect the reliability of the KTA. Shown also is comparative information regarding these items for the four selected KTA designs and in addition a two-bearing KTA design. Item 1, the maximum bearing temperature gradient, is the difference between the nominal bearing temperature and the adjacent potassium vapor temperature in the turbine. For potassium and organically lubricated bearings the lubricant was supplied at 800 and 400°F, respectively. From the standpoint of temperature gradient, which relates directly to thermal stress, the overhung turbine with potassium bearings is the best with a gradient of 480°F. For the organically lubricated machines the split flow has the smaller gradient, 880°F.

The temperature of the shaft at the point of maximum bending

TABLE 10.1-1

KTA RELIABILITY COMPARISON

NUMBER OF BEARINGS	4	4	4	4	4
LUBRICANT	POTASSIUM	POTASSIUM	ORGANIC	ORGANIC	POTASSIUM
LUBRICANT TEMPERATURE, °F	800	800	400	400	800
NUMBER OF STAGES	9	6 & 4	8	5 & 3	9
SPEED, RPM	19,200	19,200	24,000	24,000	19,200
TURBINE TYPE	OVERHUNG	COUNTERFLOW	SPLITFLOW	COUNTERFLOW	OVERHUNG
1. MAXIMUM BEARING TEMP. GRADIENT, °F	480	801	880	1163	480
2. SHAFT TEMPERATURE AT MAXIMUM BENDING, °F	1280	1601	1955	1955	1280
3. TURBINE ROTOR RESPONSE	S	S	M	S	S
4. ALTERNATOR ROTOR RESPONSE	M	M	M	M	S
5. HERMETICALLY SEALED?	Yes	Yes	No	No	Yes
6. EXTERNAL EXTRACTION	No	Yes	No	Yes	No
7. LAST STAGE INLET MOISTURE, %	16.3	11.1	15.9	9.2	16.3
8. LAST STAGE TIP SPEED, FPS	776	829	766	828	776
9. STATOR BIMETALLIC JOINT DIAMETER, IN.	3.6	3.85	3.6	3.7	9.8, 7.3
10. ROTOR BIMETALLIC JOINT DIAMETER, IN.	2.3	2.3	1.85	1.85	4.7
11. NUMBER OF DZL SEALS	4	4	8	8	2
12. NUMBER OF SCREW SEALS	4	4	5	5	2
13. FRAME WEIGHT, LB	9	22	137	115	0
14. RANK - NOT CONSIDERING EROSION	1	2	5	4	3
15. RANK - CONSIDERING EROSION	4	2	3	1	(4)

S = SATISFACTORY

M = MARGINAL

moment, item 2, affects the level of the modulus of elasticity and, therefore, the stiffness of the rotor. In both of the organic turbines the inlet vapor is introduced in the middle of a straddle-mounted shaft, resulting in the maximum temperature, 1995°F, at that point. In the potassium counterflow turbine the high-pressure (high-temperature) turbine is overhung so that an intermediate temperature, 1601°F, occurs at the forward bearing where the bending moment is largest. Only in the overhung potassium turbines, four-bearing or two-bearing KTA, does the turbine exit temperature, 1280°F, occur at the point of maximum bending moment.

Items 3 and 4 are turbine and alternator rotor response. With respect to the turbines, all had satisfactory rotor response characteristics except the split-flow organic turbine. This machine with a small-diameter hot section in the middle of a straddle mounted rotor had marginal response. Both the organic and the potassium alternators had marginal response characteristics, because the vibrational nodes occurred at or near the bearings. As is indicated in Phase II, the rotor response of the alternators can be made satisfactory by moving the thrust bearing to a position between the drive-end bearing and the rotor, minimizing seal lengths, minimizing rotor weight and using larger diameter journal bearings.

Hermetical sealing (item 5) of the KTA is only possible for the potassium lubricated machines. For the organically lubricated machines there must be a vent to space in the seal which separates the lubricant from working fluid.

External extraction (item 6) is possible only for the counterflow turbines. This permits lowering the last stage moisture fraction

(item 7) to 11.1 and 9.2 percent for the potassium and organic counter-flow machines, respectively. The overhung and split-flow turbines without interstage condensate removal have last stage moisture fractions of 16.3 and 15.9 percent, respectively. The level of last stage moisture fraction has implications in possible droplet impact erosion. Erosion increases with moisture present and tip speed for a given blade material. The two counterflow turbines, which have lower last-stage moisture fraction values, have somewhat higher last stage tip speeds (item 8) than the overhung and split-flow turbines which have higher values of moisture fraction. If condensate removal from stators is incorporated in all stages exceeding 4% moisture, the moisture levels can be limited to about 10 percent, assuming 25% extraction effectiveness.

Bimetallic joints (items 9, 10) are required in both the turbine rotors and stators in going from refractory materials in the turbine to non-refractory materials in the alternator. Hardware development programs in NSP indicate that the reliability of bimetallic joints decreases with the diameter of the joint. Of the two types, those in the stator are the most difficult to develop because a metallurgical bond is required compared to a mechanical joint in the rotor. Table 10.1-1 indicates little difference among the rotor or stator bimetallic joints for the four selected KTA designs. However, for the two-bearing KTA, sizable bimetallic joints are needed in both the rotor and stator compared to those in the four selected KTA designs.

Since seals on the rotating machinery have historically been trouble spots, the number of both dynamic zero leakage (DZL) and screw seals are tabulated in Table 10.1-1 (items 11, 12). The organi-

cally lubricated KTA designs each have a total of 13 such seals compared to 8 for the potassium lubricated machines. The two-bearing machine is best in this regard since it only has 4.

The weights of the frames (item 13) for each KTA are shown in Table 10.1-1. The heavier weights are generally associated with those designs in which it is most difficult to attain good alignment between the turbine and alternator under all modes of operation. Best is the two-bearing machine with no frame, closely followed by the overhung potassium machine at 9 pounds. The worst design in this respect is the split-flow organic machine having a frame weight of 137 pounds.

Not considering erosion the KTA designs are ranked for reliability in order of preference (item 14). The first choice is the overhung potassium KTA because it has low temperature gradients near the bearings, low temperature at the point of maximum bending moment, it is hermetically sealed, the stator bimetallic joint is of small diameter, it does not have an excessive number of seals and the frame is of minimum size. All of the potassium KTA designs including the two-bearing design are preferred over the organic designs because of the complicated seals required to keep the lubricant and working fluid separated.

With regard to erosion, the moisture entering the stage in question was considered to be the most important factor. Thus in item 15 the ranking is in the opposite order of the last stage inlet moisture. This makes the split spool machines the preferred ones.

10.2 EASE OF DEVELOPMENT

Shown in Table 10.2-1 are a number of items which significantly affect development ease. All selected four-bearing KTA designs permit separate component development (item 1). This is not true of the two-bearing machines. As indicated before, the high-pressure vapor seal (item 2) is a source of vapor leakage and thermal stress. The problems are accentuated by high-pressure (high-temperature). Developmental work will be required in this area, especially if the highest pressures listed are used. From the aspect of development of the high-pressure vapor seal, the designs in which the turbine discharge is adjacent to a bearing are to be preferred. This occurs for the overhung potassium and the split-flow organic machines. The problem associated with the organic counterflow is the worst since the high-pressure vapor seal is in the middle of the shaft and not adjacent to a bearing. The bending moment is maximum in this location, making it mandatory that the seal diameter be made large.

The requirement for a seal between organic lubricant and turbine working fluid-potassium (item 3) makes the organic machines quite undesirable from a developmental standpoint. The seal is very complicated and requires a vent to space.

With regard to turbine rotor construction (item 4), the rotor for the overhung potassium-lubricated turbine can be a single weldment. This is also true for the two-bearing machine. Alternatively, these two designs can be bolted. The potassium counterflow turbine could have the high- and low-pressure turbines made in the form of weldments, but because of assembly requirements curvic couplings held with a

TABLE 10.2-1

COMPARISON OF DEVELOPMENT EASE FOR KTA CONFIGURATIONS

NUMBER OF BEARINGS	4	4	4	4	2
LUBRICANT	POTASSIUM	POTASSIUM	ORGANIC	ORGANIC	POTASSIUM
LUBRICANT TEMPERATURE, °F	800	800	400	400	800
NUMBER OF STAGES	9	6 & 4	8	5 & 3	9
SPEED, RPM	19,200	19,200	24,000	24,000	19,200
TURBINE TYPE	OVERHUNG	COUNTERFLOW	SPLITFLOW	COUNTERFLOW	OVERHUNG
1. SEPARATE COMPONENT DEVELOPMENT?	Yes	Yes	Yes	Yes	No
2. HIGH PRESSURE VAPOR SEAL PRESSURE, PSIA	7.8	38.4	7.8	134.4	7.8
3. ORGANIC TO POTASSIUM SEAL?	No	No	Yes	Yes	No
4. TURBINE ROTOR CON- STRUCTION	WELDED OR BOLTED	WELDED PLUS CURVIC	NOT WELDED	NOT WELDED	WELDED OR BOLTED
5. BEARING THERMAL EXPANSION PROBLEM?	Yes	Yes	No	No	No
6. EXIT DUCTING	AMPLE ROOM	NO SCROLL PERMITTED	AMPLE ROOM DOUBLE	AMPLE ROOM	LITTLE ROOM
7. CROSSOVER DUCTS?	No	Yes	No	Yes	No
8. RANK	1	2	4	5	3

tiebolt would be required to assembly the entire rotor. Because of the high stresses caused by the high rotative speed (24,000 rpm), the organic turbines could not use the weaker weldable alloy ASTAR-811C. As a result a mechanical connection had to be resorted to. The three potassium-lubricated turbines are preferred because either welding or bolting are possible.

In the potassium-lubricated machines, tungsten carbide is required for the journals and pads of the bearings in the alternator. The shaft material, H-11, which is magnetic, has a coefficient of thermal expansion which is about twice that of the tungsten carbide. Thus the pad-support ring of each journal bearing as well as the journal must be assembled by machining concentric conical faces on the tungsten carbide and mating steel parts to preclude difficulties due to differential thermal expansion. This type of conical connection must be classed as a development problem at the present time. The organic machines avoid this difficulty because bearing materials having thermal expansion coefficients about the same as H-11 can be used.

Because of the large exit vapor volume flow which is directly tied to the flow rate and vapor density, there is difficulty in finding room for an exit scroll on the counterflow potassium turbine. A quadfurcated exit is recommended for this turbine design. This exit system is a natural when four condensers are used. It permits minimum duct losses and provides rigid casing connections between the low and high-pressure turbines. Room for conventional scrolls is provided for the remaining designs.

Crossover ducts (item 7) are required for the two counterflow designs. While these require some development, they permit the use

of inter-spool condensate separators.

From a developmental point of view the overhung, potassium-lubricated bearing machine requires the least development, followed closely by the counterflow machine with potassium lubricated bearings. The two-bearing, overhung machine ranks third because the turbine and alternator cannot be developed separately.

10.3 CONVERSION EFFICIENCY

The performance of the four selected KTA designs is summarized in Table 10.3-1. Alternator internal power losses amount to 35 to 36 KW in all cases. This includes all electrical losses and rotor drag. The alternator on potassium bearings has about 5 KW larger parasitic loss (bearings and screw seals) than with organic bearings. This is primarily due to a larger journal bearing loss for the alternator on potassium bearings. Thus the potassium and organically lubricated alternators have overall efficiencies within one percentage point of each other. The turbine parasitic power losses vary little from machine to machine - 23.2 to 26.6 KW. The large labyrinth seal losses in the organic machines are compensated for by large thrust bearing losses in the potassium machines. With regard to turbine blading losses, the split-flow organic machine has the highest value due to only eight stages and only minimal blade height. The decrease in these losses from the organic counterflow (split spool) through the potassium overhung to the potassium counterflow (split spool) machines is primarily due to an increase in the number of stages. The powerplant conversion efficiency follows the trend of turbine overall efficiency. The highest conversion efficiency, 69.6 percent, is

TABLE 10.3-1

KTA PERFORMANCE SUMMARY

LUBRICANT	POTASSIUM	POTASSIUM	ORGANIC	ORGANIC
LUBRICANT TEMPERATURE, °F	800	800	400	400
NUMBER OF STAGES	9	6 & 4	8	5 & 3
SPEED, RPM	19,200	19,200	24,000	24,000
TURBINE TYPE	OVERHUNG	COUNTERFLOW	SPLIT FLOW	COUNTERFLOW
POWER OUTPUT, KW	450	450	450	450
ALTERNATOR INTERNAL LOSSES, KW	(35.3)	(35.3)	(35.8)	(35.8)
ALTERNATOR INTERNAL EFFICIENCY, %	92.7	92.7	92.5	92.5
ALTERNATOR PARASITIC LOSSES, KW	(19.4)	(19.4)	(14.2)	(14.2)
ALTERNATOR OVERALL EFFICIENCY, %	89.2	89.2	90.0	90.0
TURBINE PARASITIC LOSSES, KW	(25.4)	(23.2)	(26.6)	(26.2)
TURBINE BLADING EFFICIENCY, %	78.5	81.6	76.8	77.9
TURBINE BLADING LOSSES, KW	(144.9)	(119.7)	(159.4)	(149.8)
TURBINE OVERALL EFFICIENCY, %	74.8	78.2	73.0	74.1
CONVERSION EFFICIENCY, %	66.7	69.6	65.6	66.6

achieved by the potassium counterflow machine, closely followed by the overhung potassium machine, having a conversion efficiency of 66.7 percent. For the organic machines the highest conversion efficiency, 66.6 percent, was attained by the counterflow machine compared to 65.6 percent for the split flow machine. The entire range of these efficiencies is 4 percentage points.

Shown in Table 10.3-2 is a performance comparison of the two- and four-bearing KTA designs both having the same nine-stage overhung turbine. The entire difference is brought about by the reduction in the bearing and seal (parasitic) losses in the two-bearing machine. The total bearing and seal loss for the four-bearing machine is 44.8 KW compared to 34.2 for the two-bearing machine. The result is an improvement in conversion efficiency of 1.4 percentage points.

10.4 WEIGHTS

With regard to rotor weight (Table 10.4-1), the higher-speed, (24,000 rpm) organic machines have smaller values, as is to be expected, than the lower-speed (19,200 rpm) potassium machines. In the alternator the static-part weight of the higher-speed organic machine is lower by about 240 pounds than for the lower speed potassium machine. In the turbine the static-part weight is significantly smaller for the overhung potassium design. Reduction in the weight of the potassium overhung turbine compared to the counterflow turbine makes the KTA based upon the overhung design about 140 pounds lighter than the potassium counterflow machine. The advantage of the organic machines over the lighter potassium machine is about 200 pounds and it is due primarily to the saving in weight in the alternator which can be achieved at

TABLE 10.3-2

KTA PERFORMANCE COMPARISON BETWEEN 4- AND 2-BEARING DESIGNS

CONFIGURATION	4-BEARING	2-BEARING
LUBRICANT	POTASSIUM	POTASSIUM
LUBRICANT TEMPERATURE, °F	800	800
NUMBER OF STAGES	9	9
SPEED, RPM	19,200	19,200
TURBINE TYPE	OVERHUNG	OVERHUNG
POWER OUTPUT, KW	450	450
ALTERNATOR INTERNAL LOSSES, KW	(35.3)	(35.3)
ALTERNATOR INTERNAL EFFICIENCY, %	92.7	92.7
ALTERNATOR PARASITIC LOSSES, KW	(19.4)	(17.1)*
ALTERNATOR OVERALL EFFICIENCY, %	89.2	89.5
TURBINE PARASITIC LOSSES, KW	(25.4)	(17.1)*
TURBINE BLADING EFFICIENCY, %	78.5	78.5
TURBINE BLADING LOSSES, KW	(144.9)	(142.9)
TURBINE OVERALL EFFICIENCY, %	74.8	75.8
CONVERSION EFFICIENCY, %	66.7	68.1

* ONE HALF THE TOTAL

TABLE 10.4-1

KTA WEIGHT SUMMARY

LUBRICANT	POTASSIUM	POTASSIUM	ORGANIC	ORGANIC
LUBRICANT TEMPERATURE, °F	800	800	400	400
NUMBER OF STAGES	9	6 & 4	8	5 & 3
SPEED, RPM	19,200	19,200	24,000	24,000
TURBINE TYPE	OVERHUNG	COUNTERFLOW	SPLIT FLOW	COUNTERFLOW
TURBINE ROTOR WEIGHT, LB	109	152	84	70
STATIC PARTS WEIGHT, LB	282	368	348	355
ALTERNATOR ROTOR WEIGHT, LB	383	383	256	256
STATIC PARTS WEIGHT, LB	890	890	642	642
FRAME WEIGHT, LB	9	22	137	115
TURBINE WEIGHT, LB	391	520	432	425
ALTERNATOR WEIGHT, LB	1273	1273	898	898
KTA WEIGHT, LB	1673	1815	1467	1438

the higher rotative speed. This higher speed can be achieved in the alternator because of the lower coolant temperature chosen for the alternator and it is not an achievement which was brought about by the choice of coolant.

Shown in Table 10.4-2 is a weight comparison between the four- and two-bearing overhung potassium KTA designs. About 90 pounds can be saved by reducing the number of bearings to two. The turbine and frame weight drops about 110 pounds but the alternator weight increases about 20 pounds. The two-bearing machine has a weight about mid-way between the lighter weights for the potassium and organic four-bearing machines.

TABLE 10.4-2

KTA WEIGHT COMPARISON BETWEEN 4- AND 2- BEARING DESIGNS

CONFIGURATION	4-BEARING	2-BEARING
LUBRICANT	POTASSIUM	POTASSIUM
LUBRICANT TEMPERATURE, °F	800	800
NUMBER OF STAGES	9	9
SPEED, RPM	19,200	19,200
TURBINE TYPE	OVERHUNG	OVERHUNG
TURBINE ROTOR WEIGHT, LB	109	91
STATIC PARTS WEIGHT, LB	282	200
ALTERNATOR ROTOR WEIGHT, LB	383	390
STATIC PARTS WEIGHT	890	904
FRAME WEIGHT	9	0
TURBINE WEIGHT	391	291
ALTERNATOR WEIGHT	1273	1294
KTA WEIGHT	1673	1585

11. CONCLUSIONS AND RECOMMENDATIONS

After the completion of the Phase I design study it is of interest to summarize the major design conclusions with regard to component design conditions and overall arrangements. Recommendations are made wherever appropriate. The relationship of weights and efficiencies obtained justifies a critical review.

11.1 ALTERNATOR

A critical review of alternator design results indicates the following:

- Using an 800°F coolant temperature an alternator can be designed with a small safety margin for a rotative speed of 19,200 rpm.
- The feasibility of the 24,000 rpm rotor is dependent upon the reduction of the coolant temperature significantly below 800°F.
- The 0.004 psia vapor pressure required in the rotor cavity to limit windage losses is achievable if the coolant temperature is reduced significantly below 800°F.
- A titanium casing maintained at 600°F is 150 lbs. lighter than a Hastelloy C casing for the 19,200 rpm design.
- The reduction of the alternator size from 550 kw_e to 450 kw_e makes it compatible with the turbine output, reduces alternator weight 10 percent and improves rotor-bearing response.

The selection of a 600°F potassium coolant temperature would:

- o Make a 24,000 rpm rotor feasible
- o Establish low windage losses
- o Permit using a Titanium casing

11.2 TURBINE

Inlet Temperature

Critically reviewing the turbine designs, one has to admit that due to the 2100°F turbine inlet temperature, practically all stages one and two suffer in one way or the other. In the welded designs using Astar 811C, the 0.1 percent creep objective could not be met. Instead, three year creep rates of 0.3 and 0.6 percent had to be admitted for 19,200 rpm operation. For 24,000 rpm no solution could be established.- The molybdenum alloy turbine, in case of the bolted design, requires an integral stage one wheel and torque tube structure, which poses a problem from a forging and three-dimensional strength point of view. Bolts in the first stage could not be permitted. - The curvic coupling tiebolt design becomes questionable in the area of stages one and two, because of permanent deformation of the hub cap and tiebolt. When it comes to stator design, T-111 has to be used for turbine inlet and stages one and two. From there on, Cb-1Zr is entirely satisfactory.

Turbine Outlet Conditions

The turbine outlet is almost as limiting as the turbine inlet. Lower outlet pressures result in increasing volume flows, tip speeds and stress levels. The turbine becomes specifically heavier, as does the exhaust ducting. A limiting factor will be turbine erosion.

It is believed that the present turbine outlet conditions are already critically low.

Turbine Rotor Design

In the turbine designs presented, three different rotor types were incorporated: the bolted rotor, the curvic coupling tiebolt rotor, and the welded rotor. All three designs appear to be feasible at 19,200 rpm, and it is difficult to identify the best. The bolted rotors are lighter because they are based on high-strength molybdenum alloys. Their inherent problem is bolt relaxation primarily at stages one and two. No bolts are permissible in the first-stage wheel, resulting in a difficult forging operation to make the torque tube between the first- and second-stage wheels integral with the first-stage wheel. In addition the effective stresses in the first-stage wheel leave little safety margin. An additional problem is internal leakage. In the curvic coupling-tiebolt design the first-stage wheel will have to be designed around the tendency of the head of the tiebolt to take a permanent set due to the limited strength of the TZM molybdenum alloy at the first stage temperature level. The curvic coupling permits vapor to flow through it. Vapor internal to the rotor will flow to lower temperature areas and will condense inside the turbine rotor. This condensation inside the rotor is considered a cause for rotor distortion in steam turbine design and as a rule is not permitted. The curvic coupling-tiebolt design as well as the bolted design would be preferable for gas-turbine applications, where condensation is not a problem. In two-phase flow, however, the welded rotor eliminates the condensation problem because it is leak tight. The weldable materials

also offer the advantage of better ductility with concomittant advantages in forging. Concentric welding of turbine stages without undue distortion is considered feasible by using electron-beam welding. In contrast to the bolted and curvic coupling-tiebolt designs which use TZM molybdenum alloy, the welded designs using ASTAR 811C had strain rates of 0.6 and 0.3 percent in stages one and two, respectively, rather than the preferred value for minimum tip clearance of 0.1 percent. For the 24,000 rpm case the welded rotor was overstressed. It appears that the welded rotors require more developmental effort than the molybdenum alloy bolted and curvic coupling-tiebolt designs. It seems premature at this moment to rule out one or the other. It is suggested that development continue on the basis of interchangeable rotors and that a selection be made only after hardware and test experience have been established.

Erosion and Condensate Removal

Although not yet conclusive, test results from the three-stage potassium test turbine (Contract NAS3-10606) indicate that rotor-blade erosion may be expected on (TZM) molybdenum alloy blades at tip speeds lower than the 900 ft/sec. used on the KTA. If these test results are confirmed in subsequent tests it will be necessary to resort to condensate removal devices to keep the condensate level below the acceptable level throughout the turbine. The vortex separator has been developed to the point where its performance is highly predictable, test results indicating effectiveness values greater than 80 percent. There is little penalty associated with the installation of four vortex separators in parallel between a high- and low-pressure turbine, as was

done in the ten-stage split-spool design. This turbine also had the highest efficiency of those proposed.

Condensate removal from the stator blades of selected stages is another possible means of controlling condensate level. If used alone an effectiveness of 20 to 25 percent would have to be achieved. However, if this type of device can be made effective it could be applied in addition to the vortex separator thus further improving the life-expectancy of the turbine.

Synthetic Oil Lubricant

The KTA evaluation showed that employment of a synthetic oil as a lubricant and coolant at a temperature of 400°F is highly beneficial to alternator design, permitting a 24,000 rpm alternator design instead of being limited to 19,200 rpm with 800°F liquid potassium coolant. Overall, it results in a weight reduction of 400 lbs. out of a total KTA weight of 1800 lbs. This weight saving on the turboalternator will, however, be more than offset by the need for a separate oil system and large radiator areas for maintaining the 400°F coolant temperature level. A decision in regard to using oil really should be made from a systems point of view. As far as the turboalternator is concerned, it has been found necessary to vent the synthetic oil and/or potassium vapor to space from the oil-to-potassium seals and the alternator cavity. The potassium-lubricated system does not require venting to space, but is hermetically sealed. The venting to space over three to five years time may have deleterious effects on the space vehicle, such as causing the fogging up of windows or deposition to other surfaces. In regard to the oil-to-potassium seal design, the safe sealing between oil and

potassium requires an inert buffer gas. The buffer gas must either be discarded or reclaimed, but each approach results in a severe weight penalty for the space power system. The seal without buffer gas depends on low vacuum and remains a risk item. Accordingly, the use of synthetic oil in a potassium-turbine-driven KTA is not recommended. It is further recommended that the synthetic-oil designs not be considered for Phase II.

Rotor Response

The liquid metal lubricated bearing provides an important damping function in addition to its load carrying capability. Rotor-bearing response analyses indicated that the proper placement of rotor vibrational nodal points becomes as important in suppressing the response to third critical speeds as the stiffening of the shafts. This makes symmetric, straddle-mounted arrangements less attractive, especially when shaft rigidity is reduced by the effects of high temperature on the elastic modulus or the effects of low section modulus.

The rotor-bearing response analyses indicated that the configuration with 9 stages overhung was dynamically acceptable and resulted in lower amplitudes than straddle-mounted rotors. These analyses also indicated that the heavy alternator rotor supported on two bearings posed a considerable dynamic problem, requiring a number of iterations to find the proper combination of bearing diameter and stub-shaft length to arrive at a satisfactory dynamic behavior. Further, it was found that the alternator dynamic behavior could be improved by the addition of an overhung turbine on one side.

11.3 KTA ARRANGEMENTS

The basic concept of the study was a two-shaft, four-bearing arrangement, with elastic coupling as a connecting member between the turbine and alternator. This concept has been fully confirmed by the study, as being feasible and practical. The coupling adds little or nothing to the dynamic behavior of turbine and alternator and permits separate development and testing of the two components until it becomes desirable to combine them.

In excess of the limited objectives of the study, a two-bearing KTA design was investigated. By overhanging the turbine from one side of the alternator it was found dynamically to be highly attractive, improving considerably the dynamic behavior of the alternator rotor. Mechanically, however, this arrangement resulted in bimetallic joining problems on rotor as well as stator, which make it less attractive. This arrangement also is limited to the single-spool turbine.

KTA Installation

Although outside the scope of the subject contract, installation of the KTA is an important consideration. In the space power system there will be 4-inch lithium pipes at 2200^oF and 5-inch and 12-inch potassium vapor pipes at 2100^oF and 1380^oF, respectively. The major components of the power plant may have to be mounted rigidly on a frame running at about the radiator temperature, less than 1380^oF, to withstand inertial forces up to 20 G's. Provisions will then have to be made for flexible joints in the power plant piping, including the large pipes mentioned above, so that the differential thermal expansion can be accommodated. In the case of the KTA the attachment

to the frame and the connecting pipes will have to be flexible enough to provide vibrational isolation. Stops will have to be provided on the KTA mounts so that at time periods when the power plant is subjected to high inertial forces the deflection of the KTA mounts will be limited.

11.4 KTA EFFICIENCIES AND WEIGHTS

The efficiencies and weights accomplished in the Phase I study are listed on Table 11.4-1. The KTA efficiencies range from $\eta = 0.656$ to 0.692, the weights from 1438 lbs. to 1815 lbs, corresponding to specific weights of $W_s = 3.6$ to 4.54 lbs/kw_e. To appraise the relative importance of weight versus efficiency the effects of efficiency improvements of the turboalternator on overall systems weight have to be determined. A 300 kw_e nuclear space power system with burnup limited reactor and shadow shielding for manned applications may have a specific weight in the order of 50 to 70 lbs/kw_e. It is commonly assumed that every kw loss saved in the turboalternator would result in a net saving of 50 to 70 lbs. This assumption is not entirely correct due to the fact that most component weights do not increase linearly with power. Also, many components such as structure, electrical conversion equipment, electrical coolant system and others are not affected by KTA efficiency changes, as long as the electric power output is kept constant.

TABLE 11.4-1

KTA COMPARISON

<u>Coolant & Lubricant</u>	Potassium 800°F 9-Stage 19,200 rpm <u>Overhung</u>	Potassium 800°F 6 & 4 Stages 19,200 rpm <u>Counterflow</u>	Synthetic Oil 400°F 8 Stage 24,000 rpm <u>Split Flow</u>	Synthetic Oil 400°F 5 & 3 Stages 24,000 rpm <u>Counterflow</u>	Potassium 800°F 9 Stage 2 Bearing 19,200 rpm <u>Overhung</u>
Turbine Efficiency, Including Bearing, Seal and Ducts	0.720	0.747	0.709	0.720	0.735
Alternator Efficiency	0.927	0.927	0.925	0.925	0.927
KTA Efficiency η	0.667	0.692	0.656	0.666	0.681
$\eta - \eta_1 = \Delta\eta$	-	.025	- .011	- 0.001	.014
Turbine Weights, lbs.	400	542	568	540	291
Alternator Weight, lbs.	1273	1273	898	898	1294
KTA Weight, lbs.	1673	1815	1466	1438	1585
Spec. Weight W_s , lbs/ kw_e , based on 400 kw_e alternator output (1)	4.18	4.54	3.66	3.60	3.96
$E = \frac{W_t(1) + (\eta - \eta_1) 12000}{W_t}$		<u>1973</u>	<u>1542</u>	<u>1661</u>	<u>1841</u>
		1815	1466	1672	1585
=	1.00	1.087	1.053	0.995	1.162

Full establishment of the exact tie-in of KTA efficiency and systems weight requires a detailed systems design study and the establishment of weight vs. power dependence on all components - a study, which by far exceeds the bounds of this contract. Some of this, however, has been done, permitting the assumption that every kw loss saved in the KTA results in at least a saving of half the system weight per kw, or 25 to 35 lbs/kw_e.

The following considerations are based on an average saving of 30 lbs per kw_e saving in losses. The power output of the KTA may be 400 kw_e. A one percent efficiency improvement corresponds to 4 kw_e or 4 x 30 = 120 lbs. systems weight. A five percent improvement in KTA efficiency would permit a 5 x 120 = 600 lbs weight increase of the KTA, without affecting overall systems weight.

The relationship of weight savings to be made by KTA lightweight design and by efficiency improvements is expressed by an effectiveness number E, which is used in evaluating and comparing KTA designs. The 66 percent efficient KTA of Weight (1) has the same effectiveness as the 73 percent efficient KTA of Weight [(1) + 7 x 120] lbs =

$$\text{Weight } [(1) + 840] \text{ lbs.}$$

$$\text{Effectiveness } E = W_s(1) + \Delta\eta \times 120;$$

where W_s = specific weight in lbs/kw_e.

Design (1) and Weight (1) are used as a comparison standard; it is the KTA specified as single spool, 9 stage, 19,200 rpm, 800° F coolant, potassium bearings. It's weight is 1673 lbs., its specific weight $W_s = 4.18 \text{ lbs/kw}_e$ based on 400 kw_e KTA output. The results indicate effectiveness improvals to be made by reducing parasitic losses, by increasing the number of turbine stages and by using higher rotational speeds and lower alternator coolant temperatures.

The Phase I result is the outcome of a limited number of design iterations. It should be considered indicative rather than final. Any design shown can be further improved and optimized. The basis obtained permits a forecast of efficiency/weight combinations foreseeable for Phase II. The parameters involved are number of turbine stages, number of bearings, alternator coolant temperature and speed. The most attractive solutions, combined with high turbine efficiencies can be obtained by using the two bearing approach and further increasing the number of turbine stages. Rotor response encourages this approach. Significant weight reductions can be obtained by using 600⁰F alternator coolant temperature, 24,000 rpm and titanium casing, Table 11.4-2. The best foreseeable split spool designs with interspool condensate extraction are indicated on Table 11.4-3.

The split spool design remains a four bearing approach. A two bearing split spool is not recommended for reasons of rotor response and differential thermal expansion.

11.5 RECOMMENDATIONS

The critical review of weights and efficiencies indicates that the designs generated so far do not represent optimum solutions. Considerable weight reductions can be obtained by using higher rotational speeds and by applying light-weight design methods and materials to reducing the weight of the alternator casing. The alternator weight could be further reduced by sizing it for 450 kw_e, instead of 550 kw. The key to higher speeds on the alternator is a lowering of coolant temperatures; 700⁰F coolant makes the 24,000 rpm feasible, 600⁰F would permit a lightweight titanium casing. 1900⁰F turbine inlet would limit

TABLE 11.4-2

FORECAST - TWO BEARING KTA'S

19,200 rpm, 600^oF Potassium Coolant & Lubricant
Titanium Alternator Casing

Ten Stage Overhung Turbine	$\eta = 0.83$	;	$W_t = 300$	lbs.
Two Bearings	$= 0.95$	;	$= 60$	lbs.
450 kw _e Alternator	$= 0.927$	;	$= 966$	lbs.
KTA	$\eta = 0.73$	;	$W_t = 1326$	lbs.
	$\Delta\eta = .063$	;	$W_s = 2.94$	

$$E = \frac{1673 + 756}{1326} = \frac{2429}{1326} = 1.835$$

24,000 rpm, 600^oF Potassium Coolant & Lubricant
Titanium Alternator Casing

Nine-Stage Overhung Turbine	$\eta = 0.82$	;	$W_t = 275$	lbs.
Two Bearings	$= 0.95$	;	$= 50$	lbs.
450 kw _e Alternator	$= 0.925$	;	$= 785$	lbs.
KTA	$\eta = 0.72$	;	$W_t = 1110$	lbs.
	$\Delta\eta = .053$	;	$W_s = 2.47$	

$$E = \frac{1673 + 636}{1110} = \frac{2309}{1110} = 2.080$$

TABLE 11.4-3

FORECAST - SPLIT SPOOL KTA'S

19,200 rpm, 600^OF Potassium Coolant and Lubricant
Titanium Alternator Casing

6 + 4 Stage Turbine $\eta = 0.82$; $W_t = 500$ lbs.

4 Bearings $= 0.92$; $= - -$

450 kw_e Alternator $= 0.927$; $= 1039$ lbs.

KTA $\eta = 0.70$; $W_t = 1539$ lbs.

$\Delta\eta = .033$; $W_s = 3.42$ lbs/kw_e

$$E = \frac{1673 + 396}{1539} = \frac{2069}{1539} = 1.345$$

24,000 rpm, 600^OF Potassium Coolant and Lubricant
Titanium Alternator Casing

6 + 3 Stage Turbine $\eta = 0.82$; $W_t = 480$ lbs.

4 Bearings $= 0.92$; $= - -$

450 kw_e Alternator $= 0.925$; $= 833$ lbs.

KTA $\eta = 0.697$; $W_t = 1313$ lbs.

$\Delta\eta = 0.030$; $W_s = 2.92$ lbs/kw_e

$$E = \frac{1673 + 360}{1313} = \frac{2033}{1313} = 1.545$$

the creep rate of the first two stages to 0.1 percent in 3 years and would make the 24,000 rpm feasible. The higher speed, in general, yields higher turbine efficiency. Additional efficiency gains can be made by further increasing the number of stages, up to a total of 11 or 12. Bearing and seal losses, however, tend to increase with rpm and have to be checked carefully.

In selecting the optimum configuration for further study a decision becomes hardest.

Considering the fact that turbine erosion seems to be the greatest obstacle of the wet vapor turbine in obtaining 3 to 5 years operating capability, it is suggested to select the turbine type most suitable for condensate removal; the ten stage split spool turbine, in counter-flow arrangement, with centrifugal separators between spools. By adding condensate removal from stators, the moisture level in the turbine can be further reduced, without affecting rotor or casing design. This concept permits the addition of stages to obtain still higher efficiencies, without jeopardizing rotor dynamics.

In case a single spool turbine is desired because of less complexity, then General Electric recommends the overhung turbine with 9 to 10 stages for its outstanding dynamic behavior, its simplicity of bearing and seal design and its freedom in inlet and exhaust ducting. By extensive use of interstage condensate removal it may be possible to adequately control the moisture level in the turbine.

The cradled, in-line turbine, which was a concept of the proposal phase, is not recommended by General Electric as a result of this study. The reasons for eliminating the bearing at turbine inlet are

listed on Table 11.5-1, subdivided in effects on Performance, Reliability and Manufacture.

Depending on how long a developing cycle is foreseen it appears practical to pursue the 2 shaft, 4 bearing KTA with a split spool turbine at 19,200 rpm. It can be expected that in due time the methods of condensate removal from the turbine, actual creep rates and life capability will be established. It then will be appropriate to take a second look, reconsider turbine inlet and coolant temperatures and take advantage of the capabilities of the single shaft, 2 bearing approach with one-sided overhung single spool turbine at 24,000 instead of 19,200 rpm. The promise is a gain in efficiency/weight effectiveness of 55 percent over the 19,200 rpm KTA with split spool turbine. This corresponds to a systems weight saving of $0.55 \times 1673 = 920$ lbs. or 2.3 lbs/kw_e , based on an actual KTA power output of 400 kw_e .

TABLE 11.5-1

Reasons for eliminating forward bearing at turbine inlet side:

Performance

1. High Vapor Seal Losses
 - Stepped down seal 9.05 kw
 - Step Seal + Vapor 17.20 kw
 Exchange
2. Difficulty of sufficiently insulating bearing and seal area to prevent large heat input by condensation.

Reliability

3. High thermal stresses as a result of steep temperature gradients.
 - 25,000 psi 1500^oF/inch
4. Unsymmetric scroll inlet creates casing distortion.
5. Unwanted extension of high temperature rotor structure.
6. 2000^oF turbine end affects bearing/rotor response by low modulus of elasticity.
7. 2000^oF vapor seals prone to creep and seizure.
8. Tiebolt creates leakage problem between high pressure high temperature vapor (~100 psia, 2000^oF) and bearing cavity (~30 psia, 800 - 1100^oF).

Manufacture

9. Difficulty of forging first stage wheel and stub shaft integral.

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