Report No. 101

DESIGN STUDY FOR THE MECHANICAL
ATTACHMENT OF BRITTLE MATERIAL
BLADES TO GAS TURBINE ROTORS

Contract Nonr 484(01)
Task Order NR 094-179

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Prepared by: _[signature]_

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## PREFACE

This Document is the final report of the research activities conducted in compliance with the Office of Naval Research Contract Nonr-484(01), Task Order NR 094-179.

This task was entitled "Design Studies for the Application of Brittle Materials to Gas Turbines". The scope of the task was clarified in subsequent correspondence and discussions which emphasized that the design studies (and test work as required) should seek a solution to the problems associated with the attachment of non-ductile material turbine blades to a ductile rotor disk. The studies were based upon dimensional and operating data for a small auxiliary gas turbine as approved by the Office of Naval Research.

We wish to acknowledge the cooperation and assistance of the many industrial organizations and government agencies which have been so helpful in the conduct of this research.

# DESIGN STUDY FOR THE MECHANICAL ATTACHMENT

## OF BRITTLE MATERIAL BLADES TO GAS TURBINE ROTORS

### SUMMARY

A design study was conducted resulting in the design of a mechanical attachment for a brittle turbine blade to a ductile rotor disk and an analytical determination of the maximum effective gas temperature and service life of the attachment in a specific application. Included are brief analyses of the required assembly clearances to provide for dissimilar thermal expansion rates, the effect of rotor rim cooling on the attachment problem, and a detailed description of the methods used for the readers' reference. Tests were conducted to determine a suitable ductile material insert and to qualitatively compare designs. Analysis of the final design when applied to a small auxiliary gas turbine indicates that, with an uncooled rotor, at an effective gas temperature of $1820°F$. the attachment life is 100 hours and blade life is 1000 hours. With a cooled rotor, at a temperature of $1920°$ F. the attachment life exceeds 1000 hours while the blade life is 100 hours.. Figures are based upon calculated temperature distributions and upon available stress-rupture data for several cermet materials since no high temperature life tests were conducted. It is concluded that the attachment design is satisfactory for the application and would permit increased turbine operating temperatures insofar as the blade and attachment strengths are concerned. Considerations which may limit the use of brittle blades are noted as are the assumptions for and limitations of the analyses presented.

### INTRODUCTION

In the search for improved performance, the gas turbine designer looks for solutions to a great number of problems. While they may disagree as to the relative importance of the various problems and in the best approach to their solution, all will agree that substantial increases in power without sacrificing efficiency can be obtained by increasing turbine operating temperatures. Present designs are limited by the ability of the available highly-alloyed materials to withstand higher temperatures. This is generally true of all the hot parts in the engine such as combustion chamber liners, nozzle box and vanes, turbine rotor and blades, and the exhaust system.

It is expected that solutions to the oxidation and thermal shock problems involved will be found in metallurgical advances, development of temperature resistant coatings, and improved cooling methods, etc. Research and development toward the solution of these problems is progressing in each of these fields and shows considerable promise. The problems of oxidation and thermal shock resistance are common to all of the above components and are the major obstacles in the path to higher temperatures for all of the components except the turbine rotor and blades which are subjected to high stress as well as high temperature.

Rotor disks are subjected to varying biaxial stresses due to centrifugal loads and temperature distribution. Elastic deformation and, in some cases, plastic flow of the disk material results in reduced maximum stresses and more uniform stress distribution. Thus, ductility is essential to the design of gas turbine rotor disks and to the efficient use of disk materials. Therefore, as far as the turbine rotor is concerned, the increase of operating temperatures depends largely upon the development of cooling methods to limit the disk material temperature and/or the development of new ductile materials capable of withstanding higher temperatures.

Turbine blade cooling can reduce blade material temperature substantially below the gas temperature. Considerable progress has been made in developing cooling methods and blade designs to accomplish this and thus to increase gas temperatures. In addition, new materials have been developed which exhibit satisfactory oxidation and thermal shock resistance for use in turbine blades at higher temperatures. However, the ductile materials lack sufficient stress-rupture strength for even small temperature increases, and the materials which promise relatively large temperature changes are primarily non-metallic and present difficult design problems in their attachment to metallic rotor disks. The attachment problems are due to the dissimilar physical characteristics of the two materials being joined, such as coefficient of thermal expansion, and to the fact that the non-metallic materials are extremely brittle.

The design studies reported herein are directed at the problem of achieving a satisfactory mechanical attachment between a brittle turbine blade and a ductile rotor disk, and an analytical determination of the maximum effective gas temperature and service life of this attachment design in a specific application. Included in the study are brief analyses of the required assembly clearances to provide for dissimilar thermal expansion rates, the effect of rotor rim cooling on the attachment problem, and a detailed description

of the methods used for the readers' reference. While it
is recognized that brittle materials lack impact resistance
and that this shortcoming could seriously reduce the pre-
dicted service life of brittle turbine blades, this problem
is considered to be associated with the selection of material
rather than the design of an attachment. Therefore, no
consideration is given to impact resistance in this report.

It should be noted here that the results of most of
the above research and development will make a valuable
contribution to the conservation of strategic materials. The
scarce alloying elements are used primarily for improved
properties at elevated temperatures. The development of
temperature-resistant coatings, new cooling methods, and
cermet materials will undoubtedly permit the use of less
strategic alloys at present temperatures in many applica-
tions where temperature increases are either not desirable
or not feasible until satisfactory solutions to the remain-
ing problems are found.

## THE PROBLEM

The primary load involved in attaching a blade to a
turbine disk is tension caused by the centrifugal force of
the rotating blade itself. The bending load resulting from
the gas forces is small relative to the centrifugal loads.
It can be counteracted by offsetting the blade centroidal
axis from the radial line through the neutral axis of the
attachment by an amount which equalizes the resulting
moment due to centrifugal load and the moment due to gas
load. Vibration is a serious problem if the natural fre-
quency of the blade is less than six to eight times the
rotational speed of the rotor.

The design of a mechanical attachment for such a
blade involves one or more notches and a like number of
lands or load-bearing surfaces. The load is transferred
from the blade base to the rotor segment on the disk
periphery either by mating the shape of the contacting
surfaces in the two components or through a pin which
mates with notches in each of the bodies. In either
case the area available to resist the tension load is
reduced adjacent to the load-bearing lands. The maximum
tensile stresses occur at such points. In order to keep
bearing-land area to a minimum, blade attachments are run
with high bearing stresses. This demands that the bearing
contact surfaces mate nearly perfectly under load in order
to insure a uniform load distribution and prevent local
failure.

For ductile material blades these problems have been

solved by using various modification of the "fir-tree" or "pine-tree" attachment. Due to the ability of the material to deform and distribute load, a relatively small load is taken on eachone of the multiple load-bearing lands. The lands and, therefore, the notches can be small, and relatively large cross-sectional areas are available to resist the tension load. Reasonable production tolerances can be allowed in fabricating the attachment because of the material's ability to deform and achieve a perfect fit under load.

Blades of a brittle material, on the other hand, will not deform under load and aresubject to high stress concentrations in tension due to notches or changes in section and in bearing due to lack of fit or surface conditions which result in line-contact or point-contact rather than in a uniform load distribution over the bearing area. It is apparent that a satisfactory design for the attachment of brittle material blades to a gas turbine disk must minimize the concentration of stress in tension and incorporate some means of insuring uniform bearing load distribution. These are the two major problems to be solved.

In addition to the above requirements, a satisfactory attachment design must not allow the brittle material blade base to be subjected to large bearing or crushing loads which can result fromthe combined effects of rotor plastic flow and the different rates of thermal expansion of the two materials. Also, cognizance must be taken of the fact that the the thermal conductivity of the cermet materials considered is greater than that of current blade alloys and will permit more heat flow into the rotor rim. A satisfactory design for use with current rotor disk materials must include provisions for limiting the temperature distribution in the disk if higher operating temperatures are to be obtained. Vibration is not considered to be a serious problem with brittle material blades since the natural frequency of such a bladeis in the order of twice the frequency of itsmetal alloy counterpart.

## PROCEDURE

The procedure followed in conducting the design study was primarily an analytic one. Experimental work in the form of strength testing was used as a design tool only in those instances where problems were intangible and not amenable to analytic solution. The procedure is described in the approximate order in which the study work was conducted.

A non-dimensional design chart was prepared showing the effects of notch-radius and notch-depth upon the stress

concentration factor to be applied to the tensile stress in the neck of a notched member in tension. This chart makes it possible to select the proportions of a blade base such that the stress concentration factor will be minimum.

Comparative strength tests were conducted to determine a satisfactory means of obtaining a reasonably uniform load distribution on a load-bearing surface. Brittle material specimens were loaded in a centrifugal test apparatus in order to simulate actual load conditions.

Several attachment designs were analyzed and compared. The designs were consistant in that the rotor segment strength was held constant as was the space envelope containing the blade base and rotor segment. Comparative strength tests were conducted to establish the merit of any design having features which were expected to be improvements the extent of which were not determinable by analysis.

The design considered to be superior was selected and subjected to a complete analysis to determine the maximum temperature vs. life expectancy when used in a small auxiliary gas turbine rotor. Temperature distributions were calculated at several effective gas temperatures for both an uncooled rotor and one with rim-cooling fins. The available stress-rupture data for several cermet compositions and one currently-used disk alloy were used in determining the optimum attachment configuration and the maximum temperature capability for several service life expectancies.

A rotor disk profile was designed to incorporate the necessary changes due to reduced blade loading and increased temperature gradient. Several suggested designs for methods of retaining the blades in the disk were prepared, and a brief analysis was made to determine the assembly clearances required to insure that the blades would not be over-constrained at any time during the operating cycle.

## Stress Concentration

The non-dimensional design chart of Figure 1 shows the effect of notch-radius and notch-depth upon the concentration of tensile stress in the blade base. The dimensional nomenclature is shown in Figure 2. The family of curves are obtained from the data of references 1 and 2. The loading condition of the references is pure tension through the notched area while a turbine blade attachment is loaded in the notch with accompanying large bearing loads. The determination of the effect of notch-loading involves a test program which is beyond the scope of this study. Consequently, the bearing limit curve of Figure 1 is taken from reference 3 which reports the results of such a test program.

Notch proportions above the limit curve result in bearing failures while those below result in tension failures.

The limit curve applies strictly to methyl methacrylate plastic specimens loaded in a universal testing machine equipped with a polariscope to preclude eccentric or variable loading. The curve is assumed to be a maximum for brittle material turbine blade attachments for the following reasons:

(1) The plastic material shows somewhat more ductility than ceramics or cermets.

(2) Bearing loads will not be uniform in turbine blade attachments due to the mass distribution of the cambered airfoil.

(3) Load eccentricity between bearing lands is more possible with turbine blades due to fabrication tolerances.

(4) Centrifugal loading results in bearing loads on the lands which are greater than the tension load in the neck of the blade base due to the mass below the neck.

The curve as shown was used for design purposes since little else is known about the characteristics of notch loading and such factors as the ratio of bearing stress to tensile stress of different materials and the inherent variations of experimental data might favor the use of somewhat higher bearing load limitations. However, consideration is given to items (2), (3), and (4) above in accounting for unexpected bearing failures during test, and in improving attachment designs which otherwise appear to warrant such development.

It is apparent from Figure 1 that the optimum proportions of a blade base for maximum load factor with adequate bearing strength are approximately as follows:

$$r/D_b = 0.4 \text{ to } 0.5$$
$$d_b/D_b = 0.65 \text{ to } 0.70$$
$$r/h = 2.3 \text{ to } 2.7$$

These proportions result in a maximum load factor of approximately 0.462 which indicates that the blade neck will carry 46 percent of the tensile load that the base would carry without notches.

It should be noted that an attachment configuration with the maximum load factor is not necessarily the optimum for a particular type of design since the optimum design requirement is minimum blade neck stress at the total centrifugal load of both the blade and base while maintaining adequate strength in the rotor rim segment. This is shown by the equation for blade neck stress.

$$\sigma_{t,\,b} = \frac{P_b}{(d_b/D_bK_f)\ D_bt} \tag{1}$$

Where $P_b$ equals centrifugal load due to the blade and the portion of the base above the neck. The base thickness (t) is considered a constant throughout the study since it is determined by the root chord of the blade. Since $P_b$ and $D_b$ are affected by several factors including the parameters which determine the load factor $(d_b/D_bK_f)$, it is evident that a minimum stress may occur with other than a maximum load factor. The section of this report entitled "Preliminary Attachment Design Studies" deals more fully with this subject.

## Preliminary Attachment Design Studies

In order to make a preliminary comparison of several different types of attachment designs, it is necessary to establish a standard set of operating conditions. The conditions assumed are for the small auxiliary gas turbine at its design speed of 36,000 RPM. These are 55 blades per wheel on a diameter at the root platform of 5.11 inches. The centrifugal load at the blade root platform is 980 pounds based upon a cermet density of 0.21 pounds per cubic inch.

For each type of attachment the critical rotor rim stress is held constant at approximately 36,000 p.s.i. which is equivalent to the 100 hour stress-rupture strength of Timken 16-25-6 alloy at 1270° F. This value is arbitrarily chosen as representative of a reasonable maximum value for this material. Each attachment configuration is varied until a minimum blade neck stress including stress concentration factor is obtained. Comparisons are then made on the basis of blade neck stress. The configuration having the lowest stress is considered to be the best design.

The attachment types investigated are shown in Figure 3. Several other types were considered but not subjected to preliminary analysis because of obvious lack of advantage in comparison with those reported here. Conventional and modified fir-tree bases were rejected because of the difficulty in obtaining generous radii and the extreme

precision of fabrication required for uniform loading
between bearing lands. Multiple round-pin type attach-
ments were rejected for the same reasons. All types which
require a split turbine disk were rejected because of the
variations in temperature, stress, and deformation between
the two disks during a normal operating cycle.

Blade **neck** stress is calculated using Equation (1).
The load, $P_b$, is computed for each configuration at the
design conditions using the following equation.

$$P_b = 980 + 36,780 \ \rho_c \ V_c \ \bar{r} \quad (\text{lb}) \tag{2}$$

where:

$\rho_c$ = density of cermet (lb/cu.in.)(0.21 for study)

$V_c$ = volume of cermet base above neck (cu. in.)

$\bar{r}$ = mean radius to centroid of $V_c$ (in.)

The load factor, $(d_b/D_b K_f)$, is taken from Figure 1. The
maximum width, $D_b$, is determined by, (1) the requirement
for 55 blades or, (2) the optimum blade base proportions
to give minimum stress. The thickness, t, is held con-
stant at 0.663 inch throughout the study.

## Dovetail Attachment

The dovetail type of attachment is a simple mechanical
attachment which can be easily fabricated, incorporates
large radii, and involves no pins or auxiliary attachments
other than a means of retaining the blade in the disk. It
can be designed to attain the maximum load factor shown on
Figure 1 ($d_b/D_b K_f = 0.462$). In this optimum configuration
the stress in the blade neck is approximately 21,700 p.s.i.
with a rotor segment stress of 36,000 p.s.i. The shear
stresses across the lands of both the blade base and the
rotor segment have adequate margins of safety. Sufficient
space is available to provide for variations of the attach-
ment loads, stresses, and design details without compromising
the essential features of the design or seriously affecting
the size and weight of the rotor disk.

This attachment design when used in conjunction with
a thin, ductile metal insert is judged to be superior to
the other types investigated. It is subjected to a thorough
analysis later in this report in an attempt to arrive at an
optimum attachment for a given application in terms of
operating temperature and service life.

## Pin-in-Longitudinal Shear Attachment

Round pin type attachments are attractive because they are easy to fabricate and assemble in mass production and because the attaching pins may be made of a soft, ductile material to distribute the bearing loads evenly across the lands of the cermet blade base. It is possible to obtain a configuration with a maximum load factor. However, the accompanying total width, $D_b$, is very small and the corresponding rupture strength is relatively low. The maximum strength configuration occurs with a notch radius to notch depth ratio of approximately unity $(r/h = 1.0)$. The minimum blade neck stress with a rotor segment stress of 36,000 p.s.i. is approximately 26,000 p.s.i. The attaching pins are only slightly larger than 1/32-inch in diameter and must carry a shear stress of over 27,000 p.s.i. It is likely that a ductile material insert would be required to distribute the bearing loads since any material capable of this shear stress at elevated temperatures would be quite hard. Shear strength is assumed to be 70% of the tensile stress-rupture strength for all materials since actual shear strength data are not available.

This attachment is not as strong as the dovetail, requires more parts since both pins and ductile inserts are needed for load distribution, and is very sensitive to changes in design. That is, the optimum proportions do not remain the same as the ratio of blade neck width to rotor segment width are varied in order to obtain the optimum attachment for a given application. It is concluded that the basic weakness of this type is that both the blade base and rotor segment must be reduced in strength in order to provide space for the attaching pins. Two other attachment types are the result of attempting to avoid this weakness in the design of a pin-type attachment.

## Pin-in-Sectional Shear Attachment

This attachment design permits the blade base width to be increased by transferring the load from the blades to the rotor disk through leading-and trailing-edge flanges rather than through rotor segments. This configuration allows the blade base and attaching pins to occupy the entire perimeter of the rotor disk. Assembly of the blades to the wheel should be simple and the pins can be retained by peening the ends as is currently done with alloy blades.

Investigation of this design discloses that the major problem is to provide sufficient shear strength in the pins.

In order to realize the inherent advantages of this type of attachment, the pin area must be kept reasonably small. The high allowable shear stress required is furnished only by alloy materials with high cobalt and/or nickel content. Round pins are desirable to permit fabrication of the pin retaining holes in the wheel rim by drilling and reaming rather than by broaching. However, round pins of sufficient area reduce the blade base neck width and thus the load factor below practical values. A high load factor near the maximum can be obtained by using non-circular pins. A design of this type is shown in Figure 3. The blade neck stress is approximately 21,000 p.s.i. and the pin shear stress is approximately 39,000 p.s.i. It is apparent that this attachment is much larger and deeper than the other types and incorporates a large odd-shaped pin with a small edge distance to resist bearing loads at the rim of the rotor disk.

While the calculated stress conditions appear to be superior, several considerations make the realization of these conditions quite doubtful, and further, if realized, whether the relatively small stress advantage over the dovetail type would exceed the disadvantages. These considerations are listed below.

(1) The wheel rim flanges extend up to the top of the blade platform and are subjected to the full inlet gas temperature. The resulting wheel rim temperatures will be higher.

(2) The edge distance in bearing is small, and any increase will increase the operating loads and reduce the load-resisting areas. Combined with par.(1) above, this design might prove inadequate to realize the stress conditions shown.

(3) The larger blade load and the load from the flanges at the wheel rim result in rotor disk loads which are greater than for the other types. Therefore, the rotor will be larger and heavier.

(4) The retaining pins are operating at very high shear stresses even when the best of the super-alloys at reasonably moderate temperatures are considered. It is difficult to modify the design to reduce these stresses without nullifying the blade base neck stress advantage.

(5)  One of the pins contains approximately
     the same amount of critical material as
     an entire conventional blade for this
     turbine made of the same alloy.

(6)  The wheel rim attachment holes are very
     difficult to fabricate, as are the pins
     themselves to a lesser extent.

### Pin-in-Compression Attachment

This type of attachment allows the blade base and
the rotor segment to interlock with each other such that
the pin size does not limit the blade neck width as in
the pin-in-longitudinal shear type. With round pins it is
not as easy to fabricate as the dovetail or pin-in-
longitudinal shear types, but with non-circular pins it is
perhaps the simplest of all types. Since the pin is in
compression it can be made of relatively soft material and
require no additional ductile material insert to dis-
tribute the bearing loads.

Using round pins, somewhat lower stresses are
developed in the blade neck than with the pin-in-longitu-
dinal shear attachment. The stress level is still in-
ferior to the dovetail configuration. However, with non-
circular pins, as shown in Figure 3, the maximum load
factor can be developed and the stress level is com-
parable to that of the dovetail type. It is slightly in-
ferior because the loads are greater due to the greater
depth of attachment required to furnish adequate shear
area in the outer portion of the rotor segment since the
pin occupies space which is used to resist shear load in
the dovetail design. This configuration is judged to be
a close second to the dovetail since strength is about
equal and ease of fabrication and assembly is about the
same except for the non-circular pins used with the inter-
locking design.

In summary of the preliminary attachment design
studies, the evaluations of the various types of attachment
designs fall in the following order of superiority.

(1)  Dovetail
(2)  Pin-in-Compression (non-circular pin)
(3)  Pin-in-Sectional Shear (non-circular pin)
(4)  Pin-in-Longitudinal Shear
(5)  Pin-in-Compression (round pin)
(6)  Pin-in-Sectional Shear (round pin)

As a result of this evaluation, the Dovetail attachment is subjected to a detailed design analysis for various operating conditions and using several representative cermet blade materials in a later section of this report.

## Test Apparatus and Procedure

Tests are necessary to determine an effective means of preventing attachment failure due to local concentrations of bearing stress on the blade base bearing lands and to evaluate design modifications whose effects cannot be determined satisfactorily by analysis. These objectives can be achieved by a test program based upon the comparison of one design or technique with another in order to select the better of the two, just as well as, by a more elaborate program which would yield absolute values for each possibility. Absolute magnitudes are analytically determined in this investigation, and checking their validity requires full-scale engine testing. It is required, however, that all aspects of the test program be consistent to insure valid comparisons, be capable of simulating the significant material properties and loading conditions, and be economical. Consequently, a centrifugal test apparatus of simple design is used in order to simulate actual turbine loading conditions. The tests are conducted at room temperature since the cermets under consideration are most brittle at low temperatures. The specimens are made of an inexpensive but very brittle ceramic material. Great care is exercised to insure consistency between groups of specimens both in their fabrication and in their testing.

## Test Equipment

The test installation is shown in Figure 4 and consists of a spin chamber with a vacuum-sealed removable top enclosed in a concrete and steel protective barricade, an air-turbine drive system, a vacuum system, a mechanical hoisting system, and appropriate instrumentation. The spin chamber consists of a 24 inch diameter steel cylindrical shell 3/8 inch thick permanently sealed at the bottom with a one inch steel plate, and covered at the top with a 1-1/2 inch steel plate which seals by dead weight and external pressure against a neoprene seal mounted in a ring welded to the top of the shell. The top includes two small plexiglas observation windows and

mounting provisions for the drive system and electronic tachometer components.  Inside the spin chamber 48 wooden blocks of special shape retained by steel rings at top and bottom absorb the impact of fragments from the ruptured specimens.

The air turbine drive system consists of an air turbine and bearing housing adapted from a Thompson Products Turbine Pump Assembly, an extension shaft and bearing housing which mounts to the spin chamber top, an emergency air brake system which mounts to the top of the turbine, and a muffler through which the turbine exhaust passes.  The air is supplied by the plant compressed air system through a small reservoir to a pair of control valves, one each for turbine control and brake operation.  The Thompson turbine is a radial flow type with 26 nozzles, and is designed for a much greater power output than required for this application.  The plant air system is supplied by a 5HP compressor and it was found after some experimentation that with all but two of the turbine nozzles plugged, the turbine output was adequate and yet the plant air supply was not exhausted.  The muffler is necessary to reduce the noise level of the exhaust and the turbine while in operation.

The vacuum system consists of a Kinney VSD-778 vacuum pump with oil separator and solenoid valve, a vacuum control valve, and air-line filter to protect the valve seats and pump from damage by ceramic dust, a vacuum release valve, and a vacuum gage.  The vacuum pump has a rated capacity of 27 cubic feet per minute of free air and will pull a 10 micron Hg. vacuum in blank tests.  However, the working vacuum in this installation is 29.5 inches of Hg. due primarily to air and water vapor drawn from the wooden blocks in the spin chamber.

The mechanical hoist system consists of a specially-designed three-legged sling which can be attached by hooks to either the spin chamber or its top and is connected at the upper end to a simple manila rope block and tackle arrangement and then to a heavy wooden frame.

The instrumentation consists of an air pressure gage, a vacuum gage, and a Hewlitt-Packard Model 505(a) Electronic Tachometer.  The air pressure gage and vacuum gage are used only to determine the initial conditions for starting a test run, while the tachometer furnishes the test data to determine the relative strength of test specimens.

## Test Specimens and Holders

The specimens subjected to test are shown in Figure 5. The Calibration and Standard specimens are used to check the physical properties of the material, the reproducibility of the test data and equipment, and the effect of test blade configuration on the apparent strength of the specimens. Data from tests of these specimens are also used to evaluate the relative merits of various types of ductile material inserts. The Dovetail and Skewed Dovetail specimens are used to determine the effects of the modifications incorporated in the skewed base configuration. This base is rotated 15 degrees with respect to the turbine axis and also incorporates flat attachment bearing lands which are tangent to the notch radii and subtend an included angle of 60 degrees. The effects of these changes are not readily determinable by analysis. The reasons for believing these modifications to be improvements are explained along with the methods used to determine the extent of the changes in the section of this report entitled "Modification of Dovetail Design".

Specimens are fabricated of feldspathic porcelain. Fabrication begins with a brass pattern of the specimen made sufficiently oversize to allow for shrinkage. Patterns are made in two pieces so that each blade pattern can be attached to each of the **four** base patterns. This precludes the necessity of duplicating the complex blade pattern for eachbase design.

Plaster of Paris molds are made from the patterns, and a low-pressure casting technique is used to cast the blended feldspathic porcelain raw materials. All specimens are fabricated from a single batch of blended materials to insure consistent composition. The casting is dried in the mold for approximately 24 hours and air-dried for at least 48 hours after removal from the mold. The specimens are then fired at Cone 12 for 24 hours. Extreme care is exercised to insure that the firing conditions for each batch of specimens are essentially identical. This procedure results in very consistent dimensions from specimen to specimen with maximum differences amounting to plus or minus .005 inch.

After firing, the specimens are inspected by X-ray and by penetrant dye for internal and external defects. Rejections average about 10% for all batches except the early ones during which the casting technique was being developed. After inspection, the specimen base is ground to the final contour to an accuracy of plus or minus .001 inch. The surface finish is very smooth. A silicon

carbide grinding wheel, grade G or H, 80 or 120 grit, is used. The work is flooded with coolant during the grinding operation. Grinding fixtures are surface ground blocks with a hole to receive the blade portion of the specimen and three locating points for the root platform surface. The specimen is inserted in the hole and lightly clamped with the base portion extending out of the fixture and the hole is filled with molten sulphur which solidifies to retain the specimen and then is melted to release the specimen after grinding. The final operation is to grind the tip of the blade to finish length using the test holder as a grinding fixture. Final inspection includes weighing each specimen; all agree within plus or minus 4% of the mean weight within a group of comparable specimens.

Specimen holders are made using standard machine shop techniques. The attachment contour is formed using a special end mill ground to shape using the same wheel used to grind the specimen base. This wheel is dressed after each complete operation using a ten-times scale template and a diamond-point dressing tool. The holders used in the initial tests are made of 24ST aluminum alloy rolled plate and SAE 1018 cold rolled steel to determine the effect of holder ductility and hardness. Cold rolled steel holders are used in all subsequent tests to more nearly approximate turbine materials. Counterweights are made of the same material as the holders and are dynamic duplicates of the specimen-holder combinations.

## Test Procedure

In all except Run 1, one specimen and its corresponding counterweight are installed on the wheel as shown in Figure 4(c). The specimen is held loosely in the holder by a set-screw to prevent its falling out during handling and initial acceleration of the test rig. In most of the runs a ductile material insert is placed between the mating surfaces of the specimen and holder. The assembled wheel is then checked for static balance. The entire rotating assembly less specimen, holder, and counterweight had previously been balanced statically and dynamically at all speeds up to 25,000 RPM.

The absolute pressure in the spin chamber is reduced to less than one inch of mercury, and the specimen is rotated up to the soaking speed and held at that speed for approximately one minute to insure that the specimen does not move in the holder and that the mating surfaces

are contacting each other properly. Rotation is stopped
and a visual inspection is made through the inspection
windows to check the above conditions. The speed is
then increased as uniformly as possible until failure
occurs. The loading rates are very low in terms of
psi/second. The speed of the test rig is controlled
manually, but the rate of speed increase is coordinated
with vocal check points from a stop-watch in order to
keep the loading rate as consistent as possible. The
failure speed is recorded and the specimen is removed
and examined to determine the location and type of failure.

## Test Results

The test data is given in Table 1. Runs 1 through
7 represent the familiarization phase to get acquainted
with the equipment and the ceramic material. These speci-
mens are the only ones furnished by the first supplier
who cancelled further participation in this program due
to a change in the nature of his business. The first
tests indicate that holder ductility has a pronounced
effect upon the apparent strength of the specimens. The
specimens tested in 24ST aluminum alloy holders average
47% greater strength at fracture than those tested in
cold rolled steel holders. The location of the first
fracture was in doubt during the first five runs because
of the large number of fractures evident after each test.
The nature of this fragmentation of the specimens is
shown in Figure 6. Runs 6 and 7 include a special 24ST
retainer assembly to furnish additional support to the
base but not the blade and to retain the base fragments.
The specimen after fracture is shown in Figure 7. There
are three small fragments at the mating surface just
above the arc-shaped neck fracture which indicate that
the initial failure is due to bearing stress at that point.
The 25% increase in fracture load corroborates this con-
clusion. The fracture of the blade at the root is pro-
bably explained by the phenomenon of double fracture in
tensile testing of brittle materials. An analysis by the
method given in reference 4 indicates that a secondary
fracture at the blade root is probable following a primary
fracture at the base neck.

## Selection of satisfactory Ductile Insert.

Runs 8 through 28 are intended to determine a
satisfactory ductile insert and to determine the effect
of specimen blade shape on the apparent strength of the
specimens. Ductile materials considered for use as inserts
are .005 and .010 inch solid copper sheet, .010 inch brass

wire screen,.005 in. electrolytic copper plating, and
.005 inch nickel-plated, perforated copper screen.  There
is little difference between the two gages of solid copper
sheet.  The average rupture speed for all standard and cali-
bration specimens tested with these inserts is 10,170 RPM.
The average for the brass screen is 10,950 RPM.  The copper
plating does not adhere to the ceramic material and was
dropped from consideration after a single test.  The
average rupture speed for all calibration and standard
specimens tested with perforated screen inserts is 11,000
RPM.  The perforated screen has a negligible strength
advantage over the woven brass screen but it has two other
advantages which result in its recommendation.   It is
easier to form to the shape of the attachment contour and
its resistance to high temperature and to the oxidizing
atmosphere of gas turbine operation should be decidedly
superior.

The calibration specimens with circular blades
demonstrate only 8% greater strength than the standard
specimens with airfoil shaped blades.  However, the air-
foil-shaped blades are not as difficult to fabricate as
had at first been anticipated and thus there is no ad-
vantage to be gained by using the simpler, less represen-
tative circular blade.  The remainder of the tests are
conducted using airfoil blade specimens and perforated
screen for inserts with one exception.

## Dovetail vs. Skewed Dovetail Specimens

The objective in modifying the dovetail base to
the skewed configuration with flat bearing lands is to
reduce local bearing load concentrations by distributing
the bearing loads more uniformly with a skewed base and
by achieving better surface contact conditions with flat
bearing lands.  The Lewis Laboratory of NACA has found
that skewing the base increases the operating life of
cermet blades greatly in their full-scale engine tests.
Pratt and Whitney Aircraft have recommended flat attach-
ment surfaces as a result of their investigations in this
field.

Comparative test results taken from Table 1 indicate
that, for those specimens which failed in bearing, the
skewed base withstood 28% greater load.  However, very few
of either type failed in bearing.  Tension and bearing
fractures in the base neck, and tension fracture in the
blade root fillet are shown in Figures 8, 9, and 10. The
majority of the dovetail specimens failed in tension across
the base neck, which is the anticipated type and location
of fracture, while the majority of the skewed dovetail

specimens failed in the blade root fillet at an average
load of only 88% of the majority of the dovetail type
and only 90% of the load sustained by those dovetail
specimens that also failed in the blade root fillet. The
explanation for these results is not readily apparent,
and a thorough inspection is required of several of each
type of specimen to determine the validity of a direct
comparison of the test results.

Inspection of the skewed dovetail specimens reveals
that the centroidal axis of the blade does not lie in
the plane of the center-line between the bearing lands
but is at an angle with it. This causes bending loads in
the blade and blade root fillet under centrifugal load.
The blade is canted so as to increase the tension load
on the concave side of the blade root fillet which means
that the leading and trailing edge tension loads are in-
creased. The leading edge overhangs the root platform
in the skewed base design, and the platform is quite thin
and unsupported in this region. In addition, the root
platform of the skewed dovetail specimen is 15% thinner
than that of the dovetail specimen as shown by inspection.
It appears reasonable to assume that, in combination,
these factors could readily result in root fillet failure
of the skewed dovetail specimens. It is presumed that
the misalignment of the blade and bearing lands occurred
while grinding the bearing lands due to not locating the
specimens in the fixture properly. It is further pre-
sumed that if the skewed dovetail attachment were re-
designed to incorporate a slightly thicker root platform
and were fabricated to eliminate bending loads, it
would demonstrate a similar superiority in overall
strength with respect to the dovetail attachment as it
currently does in regard to bearing strength.

Additional points of interest resulting from this
simulated test program are the consistency of the data
from successive tests of identical specimens in view of
the use of inexpensive ceramic material and of a rela-
tively simple test rig, and the close relationship of the
attachment strength to the blade root strength of the
dovetail design. One-quarter of these specimens failed
in the blade root at an average load only 1% less than those
that failed in the base neck.

## Modification of Dovetail Design

The proportions of the dovetail base attachment are
based upon data from Figure 1 for the maximum load factor.

Figure 1 resulted from tensile tests of rectangular
plastic specimens in a universal testing machine. A
polariscope was used to insure uniform loading in the
entire notch and bearing land region. These ideal con-
ditions result in the bearing limit curve showing maxi-
mum load factors which are difficult to realize in an
attachment which is loaded non-uniformly by an airfoil-
shaped blade.

An analysis of the dovetail base is made to det-
ermine what the load distribution on each bearing land
is and what modifications can be made to improve it. The
load distribution analysis is an approximate one since
it is based upon the summation of the loads developed
in radial columns without consideration of the pro-
portion of load that is transferred to adjacent columns.
This kind of approximation results in load distributions
which are worse than would actually occur.

The root platform is divided into a grid of
squares such that there are 20 squares in the chordwise
direction and 11 squares perpendicular thereto. This
grid is extended along the length of the blade such
that the corners lie in radial lines. The squares thus
become rectangles of more unequal proportions from blade
root to tip. Large scale drawings are made of several
cross-sections of the blade and oriented in true posi-
tion on the grid for the radius at which the cross-
section is located. These drawings are superimposed
in adjacent pairs and the blade volume and centrifugal
load within each column is calculated. This process
is repeated for each pair of sections from tip to root
and the individual load values are summed up to give
the total load at the root for each column. The total
of all the column loads is checked against the pre-
viously computed total blade load. The moment of each
column load about the neutral axis of the attachment
bearing lands is calculated and the bearing land
reaction loads are obtained for each of the 20 rows
of columns in the chordwise direction.

The results of this analysis indicate that rotat-
ing the base and bearing lands about the centroidal
axis of the blade should improve the load distribution.
Consequently, the bearing land load distributions are
calculated for several angles of base rotation from
zero to twenty degrees. The load distribution for
fifteen degrees is considered to yield the best overall
picture and is used for the skewed dovetail base
configuration. The calculated load distributions for

no rotation and for 15 degrees rotation are shown in
Figure 11. The peak load intensity on the convex
side can be reduced by moving the blade on the base,
but this is not recommended since it would introduce
bending loads into the neck of the base.

The unit bearing stress on the lands is a
function of the area in contact and also depends upon
the absence of local surface conditions which cause
concentrations. The former is determined by the pro-
portions of the base and notch but the latter is de-
termined primarily by the accuracy with which the con-
tacting surfaces are fabricated. The blade base
contour would undoubtedly be ground and the rotor
segment slots would be broached in production. The
accuracy and particularly the consistency during pro-
duction runs with each of these fabricating techniques
should be better for flat surfaces rather than curved.
It is believed, therefore, that the incorporation of
flat bearing lands in the design of the dovetail base
in conjunction with rotating the base should make the
realization of the maximum load factor shown in Figure
1 quite probable. Consequently, flat bearing lands
subtending an angle of 60 degrees are incorporated
in the design of the skewed dovetail base.

Tests conducted to establish the superiority
of the skewed dovetail base with flat bearing lands
over the plain dovetail base are not conclusive for the
reasons given in the section of this report entitled
"Dovetail vs. Skewed Dovetail Specimens".

## Temperature Distribution

Radial temperature distributions for several gas
temperatures and two rotor cooling conditions are
computed for the rotor disk and blades of the small
auxiliary gas turbine which is used as a reference
throughout this study. These data are for use in making
an analysis of the temperature-vs-service life to be
expected from the dovetail base attachment. The calcu-
lations are based on a dovetail attachment with a blade
neck width-to-rotor segment neck width ($d_b/d_r$) ratio of
approximately unity. The method of calculating the
temperature distributions are adapted from references
5, 6, and 7. Heat transfer coefficients are obtained
using the method of reference 8 and gas property and
gas flow data supplied by the turbine manufacturer.

Temperature distribution curves for the turbine wheel without special cooling fins are shown in Figure 12 for effective gas temperatures of 1300, 1500, and 1800° F. Temperature distribution curves for the wheel with cooling fins on the rim are shown in Figure 13 for effective gas temperatures of 1300, 1500, 1800, and 2200° F. Effective gas temperatures are used since they are involved in the calculations. Inlet gas temperatures are approximately 5% or 65° to 110° F. higher than the effective gas temperatures shown. Rim cooling fins of adequate strength are calculated to increase the heat transfer coefficient from the disk to the cooling air by a factor of 4 or from $q_o = 30$ to $q_o = 120$. The cooling air temperature is assumed to be stabilized at 200° F. in all cases. The heat transfer coefficient from the hot gas to the turbine blade is computed to be constant at a value of $q_1 = 247$ BTU per hour per square foot per degree Fahrenheit. Other simplifying assumptions made to facilitate calculations are:

(1) Calculations are made on the basis of dividing the wheel into five sections i.e. blade, base above neck, base below neck, rotor segment, and rotor disk.

(2) Constant mean values of the heat flow area, perimeter, thermal conductivity, and heat transfer coefficients are used for each section.

(3) No temperature gradients other than radial are considered.

(4) Radiation effects are neglected.

(5) Cooling on both sides of the turbine rim and disk is assumed.

(6) The blade tip is insulated and the gas temperature is constant along the blade span.

In order to determine more accurately the temperatures in the critical region of the attachment, a two-dimensional distribution is calculated using the relaxation method of reference 9. A typical distribution is shown by the isothermal lines of Figure 14 for an effective gas temperature of 1600° F. without rim cooling. The temperatures used in the detail analysis of the dovetail base attachment are obtained in this manner.

## Cermet Material Data

Titanium carbide in conjunction with metallic binders is the most advanced cermet material for which data adequate for design purposes is available. This basic material has been combined with many different combinations of binders by two principal techniques, i.e., pressing and sintering as dry mixed powders or infiltrating the TiC matrix with molten metal during final sintering. Both methods have produced very satisfactory materials from the high temperature strength standpoint. In general, the high nickel content materials show the best overall combination of properties. They combine high temperature strength with satisfactory oxidation resistance and thermal shock resistance. However, they significantly lack impact strength which is a universal shortcoming of all the brittle cermet materials for which data are available.

Of the several remaining types of cermet materials, the borides appear to show the greatest promise. Zirconium boride compositions show very good high temperature strength and thermal shock resistance but they start to decompose in an oxidizing atmosphere after 250 hours. Sufficient data are not available to evaluate the chromium boride compositions. However, preliminary information indicates that high temperature strength equal to other cermets is possible with satisfactory oxidation and thermal shock resistance and, most hopeful of all, considerably more ductility. More ductility indicates the possibility of greater resistance to impact, the lack of which is the one major reason why the present cermet materials probably will not find wide use in gas turbine blade applications.

The compositions of several representative cermet materials are given in Table 2 to the extent that they are available. Physical property data as obtained from the manufacturers are furnished in Table 3 for several materials. Stress-rupture curves representing the best available data for the most promising materials are given in Figure 15. It is readily apparent that the availability of data is limited and that such information as is available is sketchy and incomplete. It is also noted that comparison of material properties is difficult because of the many different methods of testing for a given characteristic which are employed by the manufacturers of these materials. Standardization of test procedures would facilitate comparison and selection of these materials.

### Detail Analysis of Dovetail Attachment

The dovetail attachment is subjected to a more
detailed analysis to determine its maximum theoretical
temperature vs. service life performance under the
design load conditions for the small auxiliary gas
turbine. This analysis is carried out for each of the
three most promising cermet materials for which data
were available and also for each of the two turbine
cooling conditions.

Most turbine blade stress-rupture failures occur
in one of three places; in the blade itself, in the
blade base neck, or in the rotor segment neck. In
this analysis it is assumed that the blade dimen-
sions and proportions are established by turbine per-
formance requirements and are held constant during the
study. To determine the location and time of failure
of the blade at any given gas temperature, it is
necessary to make a plot relating the operating stress
in the blade and the allowable stress for the material
at the local temperature corresponding to the effective
gas temperature. Such a plot is shown in Figure 16 for
K 162-B material at an effective gas temperature of
$1300^{\circ}$ F. with rim-cooling ($q_0 = 120$). The operating
stress curve is calculated at a speed of 36,000 RPM and
using the density of the material under consideration.
The allowable stress curves for 10,100, and 1000 hours
are taken from stress rupture data for the cermet
material at the local temperature obtained from the
temperature distribution curves. For the conditions
shown in Figure 16, the blade will fail at a radius of
approximately 0.252 feet after about 750 hours of opera-
tion at $1800^{\circ}$ F. Cross-plots are made to determine
the time and location of failure with greater accuracy
and the resulting data are plotted as the time-vs-
temperature curve for blade failure of Figure 13 for
$q_0 = 120$. This procedure is repeated for each material
and for both rotor cooling conditions and result in the
six blade failure curves of Figures 13, 19, and 20.

The maximum strength of the attachment will occur
when it is proportioned such that the blade base neck and
the rotor segment neck are equal strength. The optimum
ratio between the blade base neck and the rotor seg-
ment neck varies with the operating temperature,
time-to-failure, and the cermet material used because
of the difference in temperature gradients and the varia-
tion in slope of the stress - rupture curves for the
different materials. The optimum ratio is determined
for each set of conditions (material and cooling) as shown
in Figure 17 for K162-B material and rim cooling. It
indicates that for a 10 hour life an attachment with a

$D_b/d_r$ ratio of 1.76 will have equal strength in the blade base and rotor rim and will withstand a maximum effective gas temperature of $2200^\circ$ F. At greater $D_b/d_r$ ratios the rotor rim will fail and at lesser ratios the blade base will fail. The same data are furnished for 100 hour and 500 hour life and the optimum points are used to plot the temperature-vs-life curves for attachment failure (in this case with cooling fins) shown on Figure 18. Similar plots are prepared for each of the materials and for both cooling conditions and result in the attachment failure curves of Figures 18, 19, and 20 for optimum $D_b/d_r$ ratio. Additional curves are plotted for $D_b/d_r$ ratios near the optimum to indicate the sensitivity of attachment strength to small changes in $D_b/d_r$ ratio.

The curves illustrated by Figure 17 are prepared using the following relationships.

For equal strength:

$$\left(\frac{D_b}{d_r}\right)_{Opt.} = \frac{1}{(d_b/D_b K_f)}\left(\frac{S_r}{S_b}\right)\left(\frac{P_b}{P_r}\right) \qquad (3)$$

For rim failure:

$$\left(\frac{D_b}{d_r}\right)_{Max.} = t\left(\frac{2\pi R_r}{55}\right)\left(\frac{S_r}{P_r}\right) - 1 \qquad (4)$$

For blade base failure:

$$\left(\frac{D_b}{d_r}\right)_{Min.} = \frac{1}{t\left(\frac{2\pi R_r}{55}\right)\left(\frac{d_b}{D_b K_f}\right)\left(\frac{S_b}{P_b}\right) - 1} \qquad (5)$$

Where:

$D_b$ = blade base width at bottom of notch

$d_b$ = blade base neck width

$d_r$ = rotor segment neckwidth

$t$ = thickness of blade base and rotor rim

$R_r$ = radius from rotor axis to rotor segment neck

$\dfrac{d_b}{D_b K_f}$ = blade base load factor (.462 for optimum dovetail)

$S_b$ = stress-rupture strength of cermet material in blade neck

$S_r$ = stress-rupture strength of Timken 16-25-6 material in rotor segment neck.

$P_b$ = design operating load in blade base neck

$P_r$ = design operating load in rotor segment neck

These relationships are derived from the stress equations at each critical point and the geometric relationship that $D_b + d_r = 2\pi R_r/55$ where 55 blades are used in the turbine wheel. Data from the temperature distribution calculations in conjunction with stress-rupture data from the three cermet materials and for the disk alloy, Timken 16-25-6, are used to obtain solutions to Equations (3), (4), and (5). The solutions are plotted on curves similar to Figure 17, and the optimum solutions are in turn plotted to form the temperature-vs-life curves of Figures 18, 19, and 20.

Figure 18 shows the theoretical maximum temperature-vs-life performance of the dovetail attachment and K162-B material when used in a small auxiliary gas turbine. The maximum effective gas temperature for 100 hour service life is limited by the attachment without cooling to 1820° F. With rim cooling it is limited by the blade itself to 1860° F. A small amount of rim cooling will apparently increase the attachment temperature limits to provide some margin over the blade limit.

Figure 19 shows the temperature-vs-life performance of K161-B material under identical circumstances. The maximum temperature for 100 hours is limited by the attachment without cooling to 1820° F. With cooling fins it is limited by the blade to 1920° F. It is apparent that rim cooling fins permit a substantial increase in temperature with this material.

Figure 20 shows the temperature-vs-life performance of TC-66-1 material under the same conditions. The maximum temperature for 100 hours without cooling is limited by the attachment to 1820° F. With cooling fins it is limited by the blade to 1915° F.

It is readily apparent that there is little difference as regards strength between the cermet materials included in this analysis. This conclusion is emphasized when the limited nature of the stress-rupture data is considered. Several extrapolations of basic data were required to complete the analysis and since the accuracy of extrapolated physical property data is questionable, this factor alone might well change the comparison of these materials. However, it is also apparent that the dovetail attachment when applied to a small gas turbine using brittle blade material will permit the use of higher gas temperatures without cooling. With rim cooling it appears probable that the attachment

strength can be made to exceed the blade strength with
an additional increase in operating temperature. These
increases are more pronounced for long service-life
operation due to the relatively flat stress-rupture
characteristics of the cermet materials.

## Additional Considerations

In addition to the basic attachment problem
associated with the application of brittle materials as
gas turbine blades, there are several other problems
or effects to be considered. The rotor disk is sub-
jected to smaller centrifugal loads due to the low den-
sity of the cermet but receives more heat from the
blades due to increased thermal conductivity. As the
rotor rim expands and contracts with temperature changes
the blade bases do not change at the same rate due to
different coefficients of thermal expansion. The
blades must be installed with sufficient clearance to
insure that they do not fail as a result of overcon-
straint. Alloy blades are commonly retained in the
rotor rim by ball-peening the ends of the blade bases
or the rotor segments. This cannot be done with
brittle material blades because they will not peen and
they are extremely sensitive to the stress concentra-
tions resulting from peening the rotor segments.

A rotor disk profile is designed for the small
auxiliary gas turbine with cermet blades and subjected
to stress analysis with design loading at 1600° F. with-
out rim cooling, and at 1800° F. and 2200° F. with cool-
ing fins. The methods used are taken from reference 10.
The profile and the stress conditions are shown in Figure
21. The disk is thinner and lighter than its counterpart
using alloy blades at an effective gas temperature of
approximately 1400° F. It is noted that plastic flow
of the rotor rim material occurs only with the 2200° F.
temperature.

A brief analysis of the thermal expansion problem
indicates that the blade bases will tighten in their
slots due to differential expansion, however, a minimum
clearance of .002 inch is sufficient to prevent over-
constraint in the case of the small auxiliary gas tur-
bine. The use of a ductile insert of approximately .005
inch thickness on each side of the blade base should
help to eliminate overconstraint and crushing. The
progressive crushing effect resulting from plastic flow
of the rotor rim after cycling the engine does not appear
to be a serious problem in this case due to the absence of
plastic flow at practical operating temperatures.

A satisfactory method of retaining the brittle material blades must be simple but must not cause any additional stresses in the blade base. Three suggested methods of retaining the blades are shown in Figure 22. The rivet type is simple and inexpensive but care is required to prevent cracking the bases while driving the rivets. The self-locking pin type is simpler yet, and while it increases the bearing land loads this should have a negligible effect. The snap-ring type is suggested to provide a quick method of removing and installing a complete set of blades. This may have advantages because of the fact that all blades may be destroyed if one blade fails due to the lack of impact resistance of the brittle material. Many other methods of retaining the blades could be suggested but these are representative of the possibilities and appear to be satisfactory for most applications.

## CONCLUSIONS AND RECOMMENDATIONS

1. Brittle material blades can be satisfactorily attached to ductile rotor disks. The problems of stress concentration, distribution of attachment loads, differential thermal expansion, and retension can be solved by careful design of the attachment configuration, incorporation of a ductile insert, attention to fits and tolerances, and selection of proper retaining methods.

2. Substantial temperature increases are possible in turbines with cermet blades due to their excellent high temperature strength and their low density. Some cooling of the turbine is required to realize the full potential, but the simplest means of cooling, i.e., fins on the rotor rim, is adequate. Under such conditions cermet blades can be operated for 100 hours or more at inlet gas temperatures approximately 2000° F.

3. A dovetail type attachment is superior in strength to others considered because of its low stress concentration factor due to large notch radius. It is simple to fabricate and assemble. A modification of the dovetail attachment which includes rotating the base 15 degrees toward the convex side of the blade and which incorporates flat bearing lands is theoretically superior to the plain dovetail. The tests conducted to establish the relative merits of these modifications were in conclusive. However, it is recommended that the skewed dovetail attachment with flat bearing lands be given serious consideration as a superior design.

4. Nickel-plated, perforated copper screen is a thoroughly satisfactory ductile insert for use between the blade base and rotor segment. Woven brass screen and thin, copper sheet show good load distribution characteristics in tests but are not as consistently superior as are the perforated screen inserts.

5. Considerable savings of critical materials can be achieved by using cermet blades, since the cermets are low in strategic material content, and the highly alloyed rotor disks can be smaller and lighter.

6. Currently available data on cermets indicate that their impact resistance is very poor. Until further metalurgical research results in sizeable improvements in this property, it is unlikely that these materials will find wide use in gas turbine blade applications. Such research is recommended in view of the tremendous potential of these materials.

7. It is recommended that further tests be conducted to determine the validity of the analyses contained in this report with respect to maximum operating temperatures and operating service life. However, since full scale engine tests are required, it is recommended that such tests only be undertaken if lack of impact resistance is judged to be unimportant in the application under consideration, or if a cermet material having adequate impact resistance is available.

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   RM E50C28, July, 1950.

9. Emmons, H. W.
   The Numerical Solution of Partial Differential
   Equations, Quarterly, Applied Mathmatics, October, 1944.

10. Manson, S. S.
    Direct Method of Design and Stress Analysis of
    Rotating Disks with Temperature Gradient, NACA
    Report No. 952, 1950.

## TABLE 1 - TEST DATA

| RUN NO. | SPEC.(1) NO. | DUCTILE INSERT | SOAK RPM. | FAILURE RPM. | FRACTURE TYPE | FRACTURE LOCATION | REMARKS (CRS holder unless noted |
|---|---|---|---|---|---|---|---|
| 1 | C3 | NONE | 2500 | 11,500 | Bear. | Neck | 24ST holder |
| 2 | C7 | NONE | 5000 | 11,100 | Bear. | Neck | 24ST holder |
| 3 | C4 | None | 5000 | 11,600 | Bear. | Neck | 24ST holder |
| 4 | C11 | None | 5000 | 9,600 | Bear. | Neck | CRS holder |
| 5 | C10 | None | 5000 | 9,200 | Bear. | Neck | CRS holder |
| 6 & 7 | C5 | None | 5000 | 12,700 | Bear. | Neck | 24ST holder & retainer on root platform |
| 8 | C17 | 005 Cu Sheet | 6000 | 11,000 | Bear. | Neck | CRS holder |
| 9 | C18 | 005 Cu Sheet | 6000 | 8,800 | Bear. | Neck | X-Ray reject * |
| 10 | C19 | None | 6000 | 8,250 | Bear. | Neck | |
| 11 | C20 | 010 Cu Sheet | 6000 | 9,300 | Tens. | Root fil. | Fracture not explained * |
| 12 | C21 | 010 Cu Sheet | 6000 | 8,250 | Bear. | Neck | X-Ray reject * |
| 13 | C22 | 010 Cu Sheet | 6000 | 10,200 | Bear. | Neck | |
| 14 | S2 | 010 Cu Sheet | 6000 | 10,000 | Bear. | Neck | 1st airfoil blade |
| 15 | S3 | 010 Cu Sheet | 6000 | 8,250 | Bear. | Neck | X-Ray reject* |
| 16 | S4 | 010 Wov.Screen | 6000 | 11,200 | Bear. | Neck | |
| 17 | S5 | 010 Cu Sheet | 6000 | 10,200 | Bear. | Neck | Large chip off blade tip |
| 18 | S6 | 005 Cu Plating | 6000 | 9,750 | Bear. | Neck | Plating not well bonded * |
| 19 | C28 | 010 Wov.Screen | 6000 | 11,500 | Bear. | Neck | |
| 20 | C29 | 010 Wov.Screen | 6000 | 9,700 | Undet. | Above Neck | Base chipped-unusual fracture* |
| 21 | C23 | 010 Wov.Screen | 6000 | 10,950 | Bear. | Neck | |
| 22 | S10 | 010 Wov.Screen | 6000 | 11,000 | Bear. | Neck | |
| 23 | S9 | 010 Wov.Screen | 6000 | 10,600 | Bear. | Neck | |
| 24 | S11 | 010 Wov.Screen | 6000 | 10,600 | Bear. | Neck | Small chip at blade tip |
| 25 | S13 | 005 Cu Sheet | 6000 | 9,450 | Bear. | Neck | Chipped blade tip |
| 26 | S12 | 005 Cu Sheet | 6000 | 8,200 | Bear. | Neck | Low strength not explained * |
| 27 | S7 | 005 Perf.Screen | 6000 | 11,000 | Bear. | Neck | Chip at base of notch |
| 28 | S8 | 005 Perf.Screen | 6000 | 9,600 | Bear. | Neck | Gas hole in neck * |
| 29 | D4 | 005 Perf.Screen | 6000 | 15,200 | Tens | Neck | 1st Dovetail-stopped twice for inspection |
| 30 | D2 | 005 Perf.Screen | 6000 | 15,500 | Tens | Neck | Small gas hole in neck |

## TABLE 1 (Cont) - Test Data

| RUN NO. | SPEC(1) NO. | DUCTILE INSERT | SOAK RPM. | FAILURE RPM. | FRACTURE TYPE | FRACTURE LOCATION | REMARKS (CRS holder unless noted) |
|---|---|---|---|---|---|---|---|
| 31 | D14 | 005 Perf.Screen | 6000 | 15,500 | Tens | Root Fil. | Small holes in neck |
| 32 | D3 | 005 Perf.Screen | 7500 | 12,300 | Bear. | Neck | Blade tip not ground off |
| 33 | D15 | 005 Perf.Screen | 7500 | 9,200 | Undet. | Blade | Failed at 2/3 blade span * |
| 34 | D7 | 005 Perf.Screen | 6000 | 14,700 | Tens. | Neck | Fract.slightly spalled |
| 35 | D12 | 005 Perf.Screen | 6000 | 15,400 | Tens. | Root.Fil | Blade tip not ground off |
| 36 | D10 | 005 Perf.Screen | 6000 | 14,300 | Tens. | Neck | |
| 37 | D8 | 005 Perg.Screen | 6000 | 17,100 | Tens. | Neck | Strongest specimen |
| 38 | D1 | 005 Cu Sheet | 6000 | Stopped for severe vibration-test rig repaired | | | |
| 38A | D1 | 005 Cu Sheet | 6000 | 16,400 | Tens. | Neck | Tract. spalled |
| 39 | SD9 | 005 Perf.Screen | 6000 | 14,400 | Tens. | Root Fil | 1st Skewed Dovetail |
| 40 | SD5 | " " " | 6000 | 13,500 | Bear | Neck | Fracture spalled |
| 41 | SD4 | " " " | " | 14,200 | Tens. | Root Fil, | )Platform too thin- |
| 42 | SD8 | " " " | " | 14,400 | " | " | )Blade axis not |
| 43 | SD10 | " " " | " | 15,300 | " | " | ) radial through |
| 44 | SD18 | " " " | " | 14,500 | " | " | ) base center |
| 45 | SD6 | " " " | " | 14,300 | Bear | Neck | Fracture spalled |

\* These runs are not representative and arenot included in the analysis of these data.

(1)Specimen numbers include the type of specimen i.e.  C for Calibration
                                                       S for Standard
                                                       D for Dovetail
                                                      SD for Skewed Dovetail

## TABLE 2

### COMPOSITION OF CERMET MATERIALS

% COMPOSITION BY WEIGHT

| SOURCE OF DATA | MATERIAL | TiC | Cb-TaTiC | Ni | Mo | Al | Cr | Co | Fe |
|---|---|---|---|---|---|---|---|---|---|
| Thompson Products | TC-86-I | 50% | | 39.5%* | | | 7.5%* | | 3.75%* |
| Kennametal | K 152-B | 64% | 6% | 30% | | | | | |
| | K 161-3 | 62% | 8% | 30% | | | | | |
| | K 162-B | 64% | 5% | 25% | 5% | | | | |
| | K 163-B | 57% | 8% | 30% | 5% | | | | |
| | K 153-B | 54% | 6% | 40% | | | | | |
| Amer. Electro Metal Corp. | Borolite I | Primarily Zirconium Boride | | | | | | | |
| | Borolite III | Primarily Chromium Boride | | | | | | | |
| Pratt & Whitney | JR | 42% | 3% | 33.8% | 12.7% | 3.5% | | | |
| | JR-5 | 54% | 3% | 27% | 10.2% | 2.8% | | | |
| | JR-6 | 46% | | 46% | 2% | 2% | | 4% | |

\* Constituents of Inconel

## TABLE 3

### PHYSICAL PROPERTIES OF CERMET MATERIALS

| | TC-?-I | K 152-B | K 152-? | Porolite |
|---|---|---|---|---|
| Source of Data | Thompson | Kennametal | Kennametal | ?? |
| Modulus of Elasticity (PSI) | | $47x10^6$ | $69x10^6$ | $3.?x10^6$ |
| Thermal Expansion $(in/in/{}^{\circ}F x10^{-6}$ to 1200 ${}^{\circ}F)$ | | $5.3x10^{-6}$ | $.3x1?^{-6}$ | $5.7x1?^{-6}$ $(1?00 ?)$ |
| Thermal Conductivity $(Btu/hr)(sq.ft)({}^{\circ}F$ per ft.$)$ | | 13.? | 17.? | 1?.0 |
| * Oxidation Resistance (wt. Change after 100 hrs.) $(mg/cm^2)$ | 7 $(2,000{}^{\circ}F)$ | 11 $(?,?00{}^{\circ}F)$ | 7 $(1,?00{}^{\circ}F)$ | ? $(?,?00{}^{\circ}F)$ |
| * Thermal Shock | Good | Good | Good | Good |
| Charpy Impact Strength 3/16" at 1000${}^{\circ}F$ (in. lb.) | 6.3,4.9 (None are alloys) | 1.?,3.? considered | 3.?,2.? good, compared to | |
| Density (lb./in$^3$) | .217 | .207 | .?20 | .1? |
| Stress Rupture 1800 ${}^{\circ}F$ 10 hr. (PSI) | 51500 | 26000 | 41000 | |
| 100 hr | 3?000 | 19000 | 31500 | |
| 500 hr | 3?500 | | ?4000 | |
| 1800 ${}^{\circ}F$ 10 hr (PSI) | ?2500 | 13000 | 2?5?0 | ?5,?00 |
| 100 hr | 15000 | ?500 | 13000 | 1?,500 |
| 500 hr | 11500 | | ?00 | |
| 1000 hr | | | | 1?,000 |

* Test conditions vary over a wide range. Numerical results can be misleading.

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Figure 1. - Theoretical Design Chart relating stress concentration load factor with notch parameters and showing bearing stress limitation.
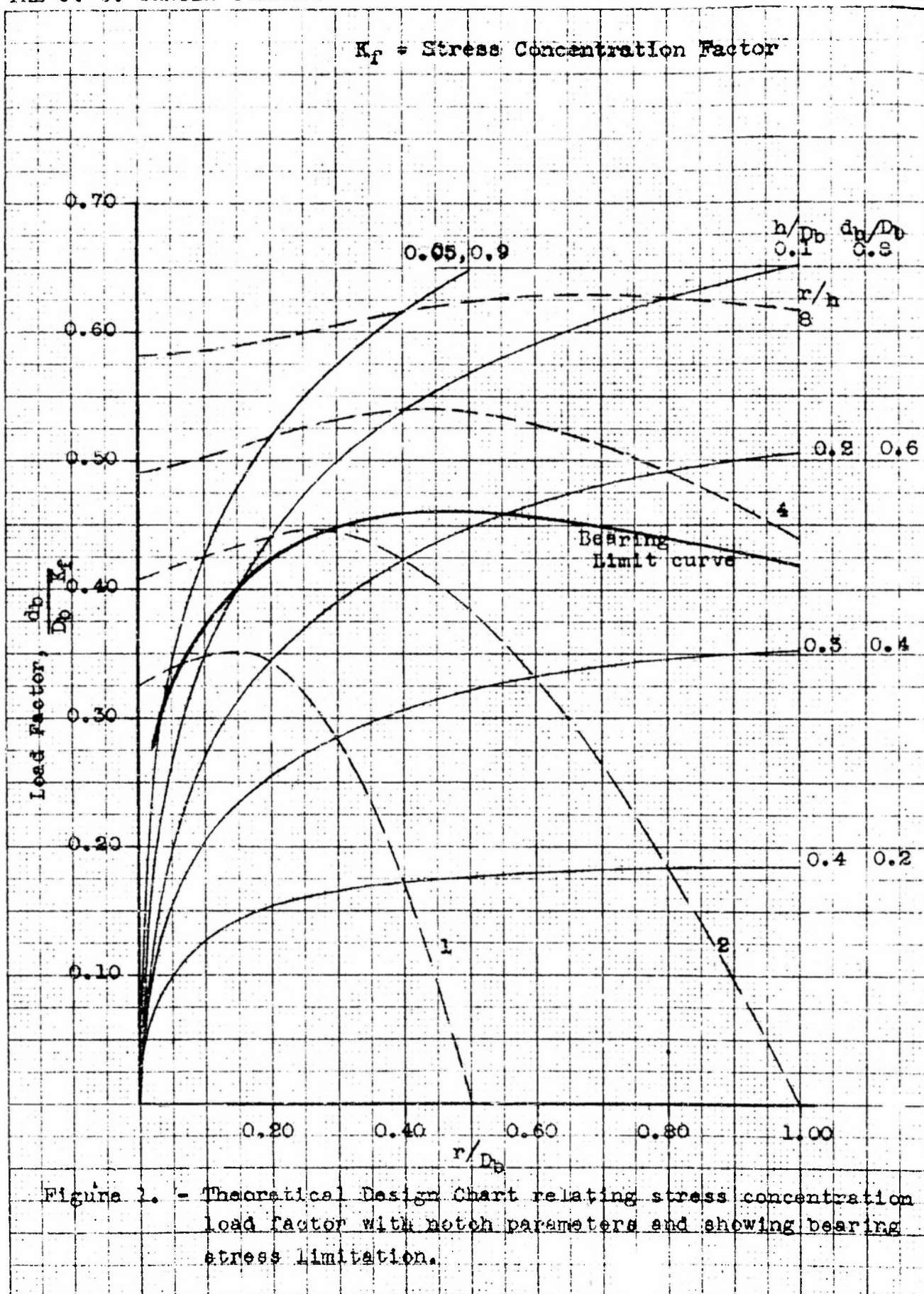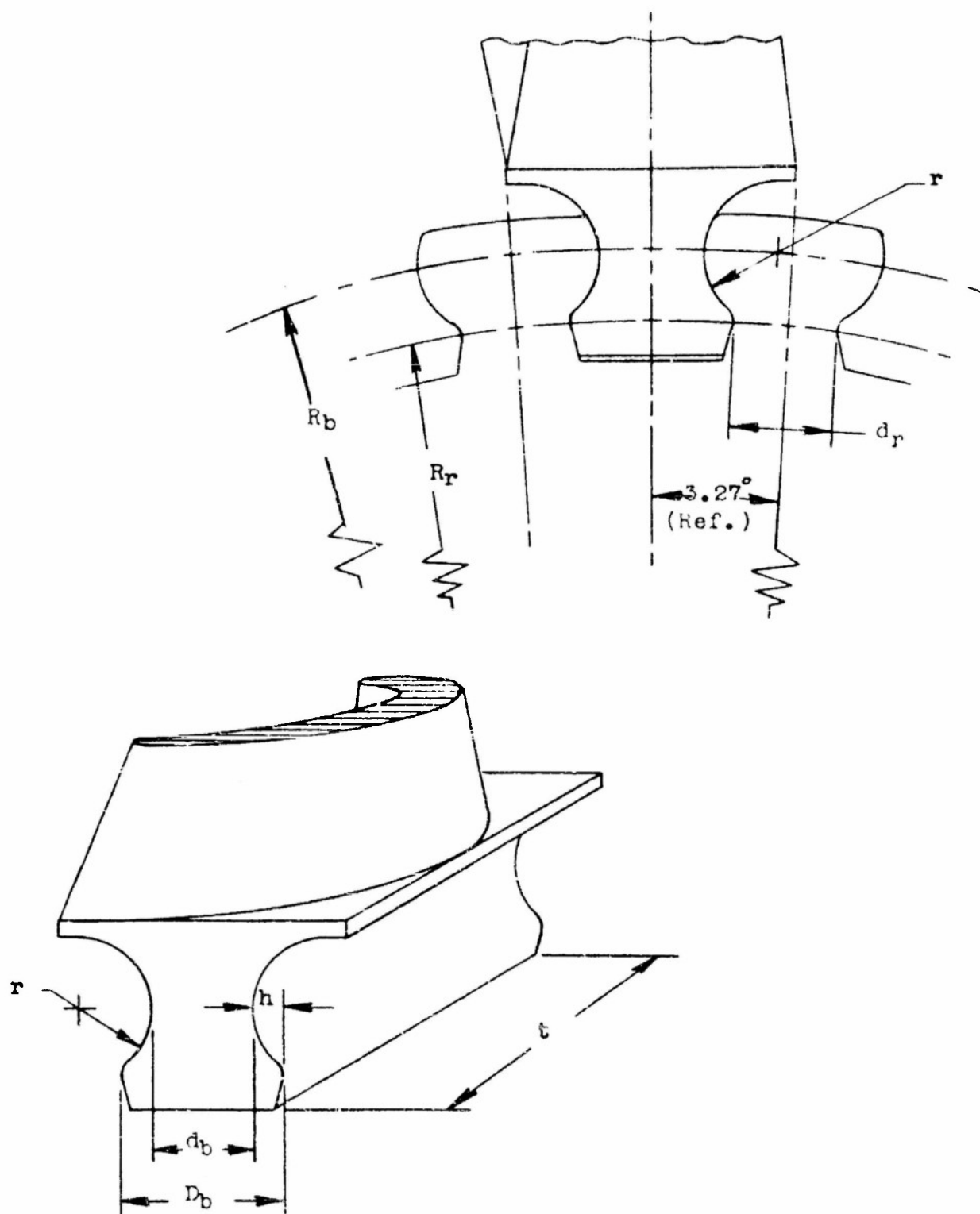
Figure 2.- Notation of blade base and rotor rim segment dimensions.

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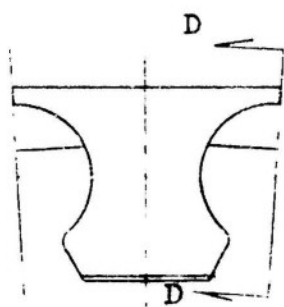
Dovetail
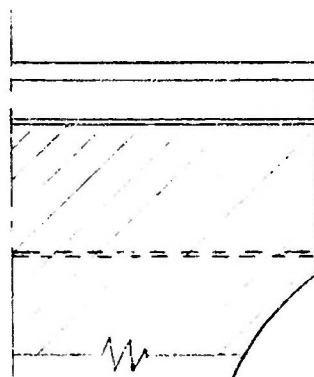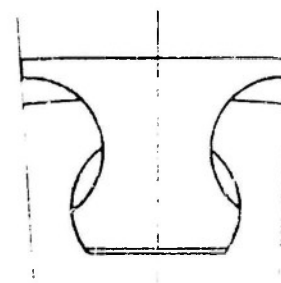
1/2 Section D-D

Pin-in-Compression
or Interlock
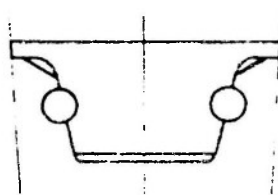
Pin-in-Longitudinal Shear

**FIGURE 3** - Attachment types investigated in preliminary analysis
(approximately to same scale)
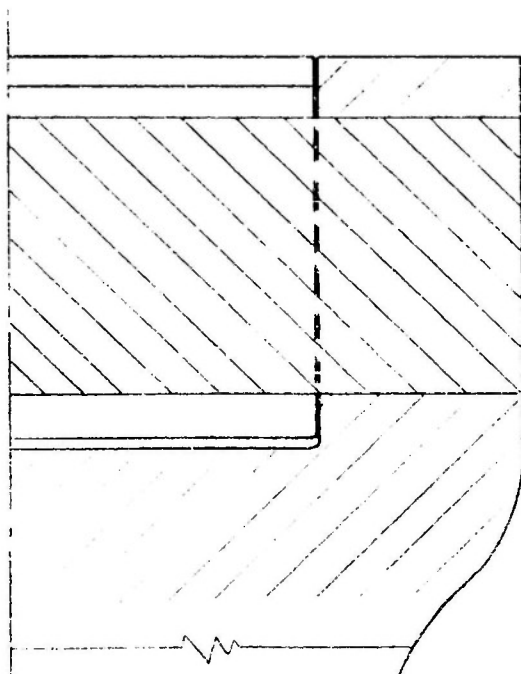
1/2 Section P-P
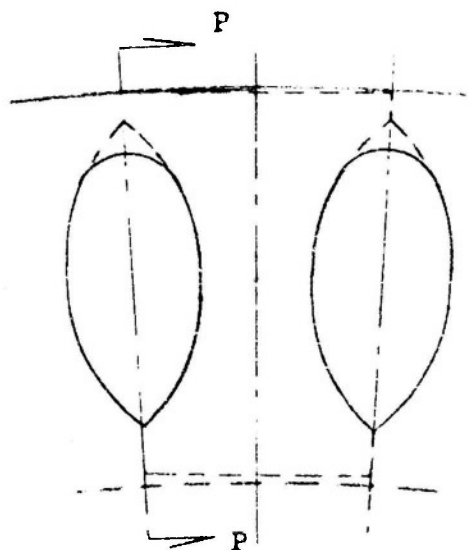
Pin-in-Sectional Shear
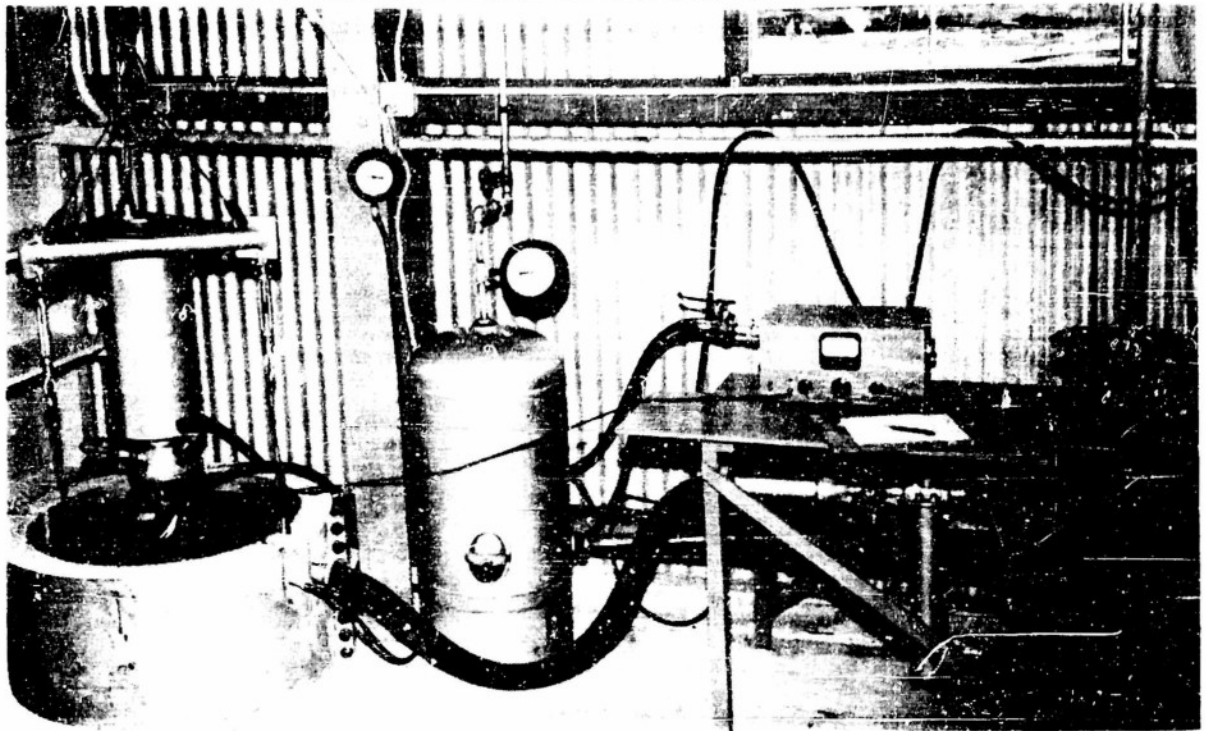
Figure 4(a) - Test Installation, showing l. to r.: protective
barricade, test chamber, hoist equipment, vacuum gage,
air tank and gage, control valves, techometer, vacuum
valves, filter, and pump.
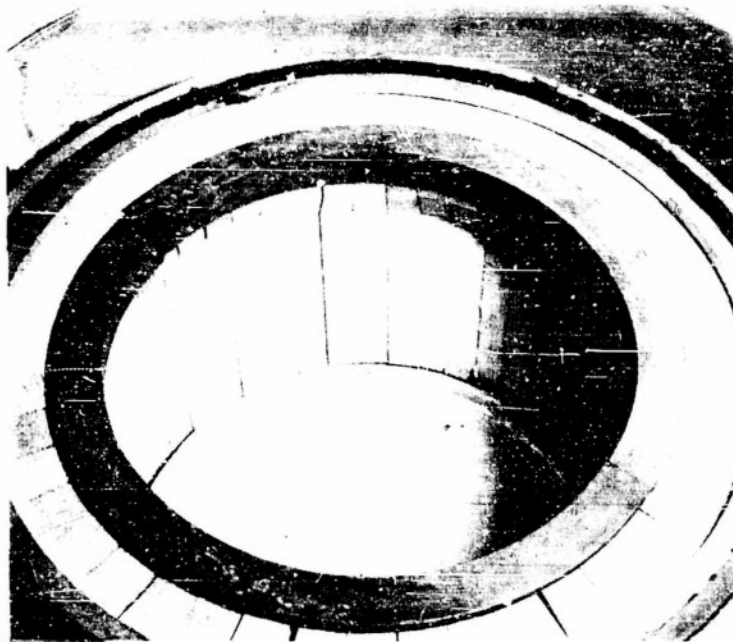


Figure 4(b) - Test Chamber Interior, showing: vacuum seal, wooden
safety blocks, block retaining rings, and portion
of barricade.
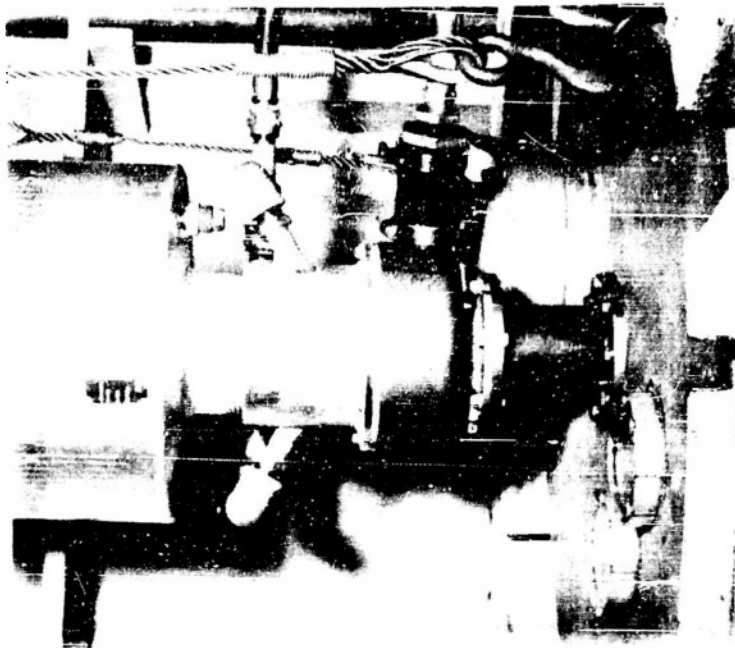
Figure 4(d) - Test Chamber Top showing: inspection windows, drive assembly, air brake, muffler, tachometer head and power supply.
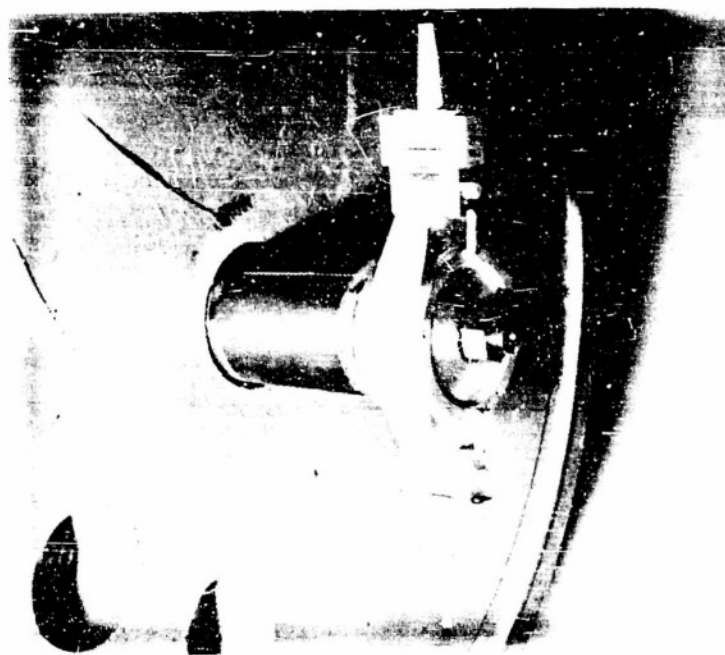


Figure 4(c) - Test Wheel Assembly, showing l. to r.: counterweight, wheel, specimen and holder, and method of assembly.
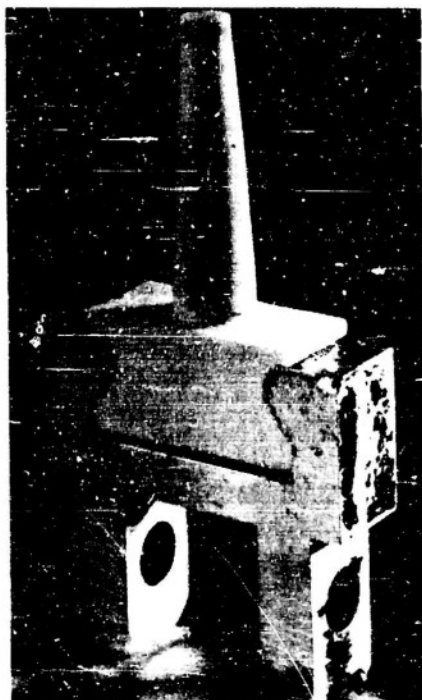
Calibration



Standard
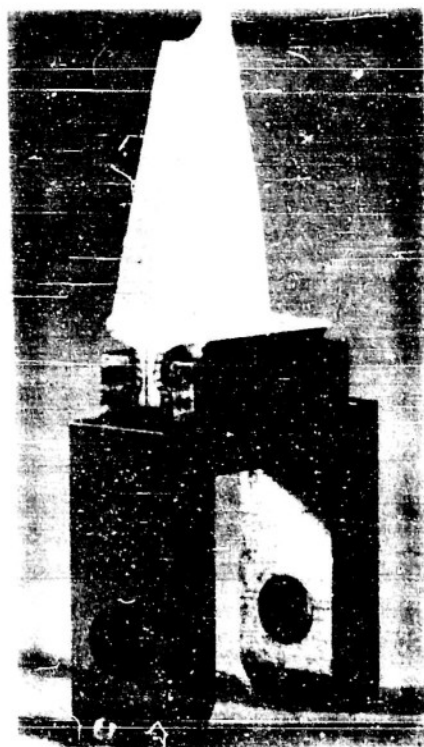


Dovetail with perf.screen
insert (rejected spec.)



Skewed Dovetail (perf.
screen insert)

Figure 5 - Test specimens
and holders

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Figure 6 - Specimen after test
showing reassembled
major fragments and
location of fractures.
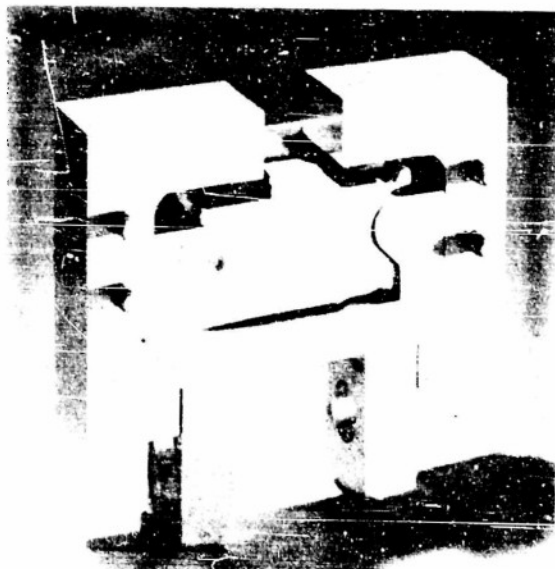


Figure 7 - Specimen base retained
after Run 7 showing
initial failure in bearing
just below neck of base
(on left).
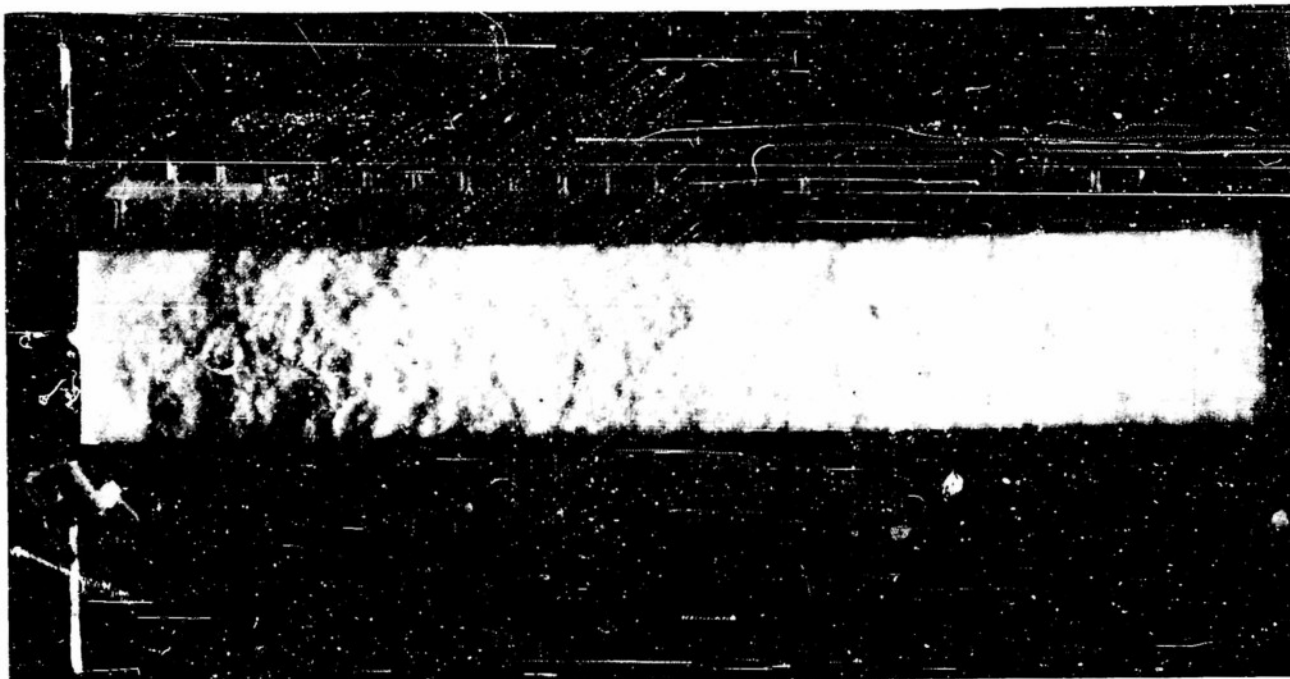
Plan View
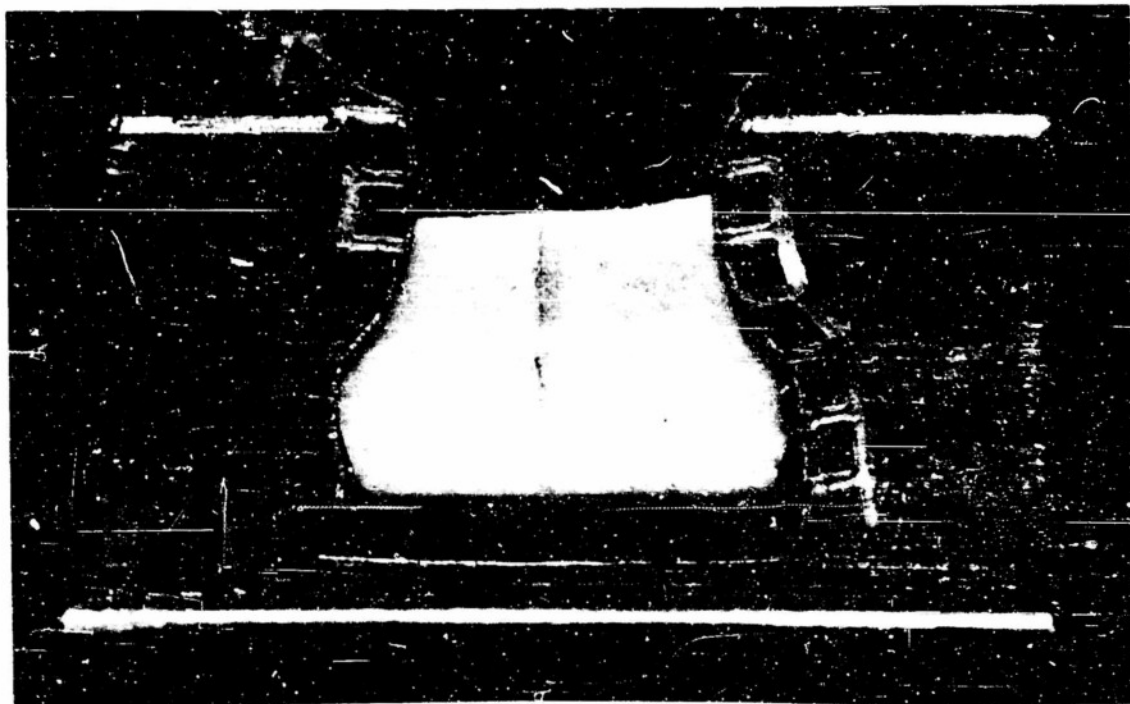


End View

Figure 8 - Typical tension fracture in neck of base, dovetail specimen (approx 9 x size)
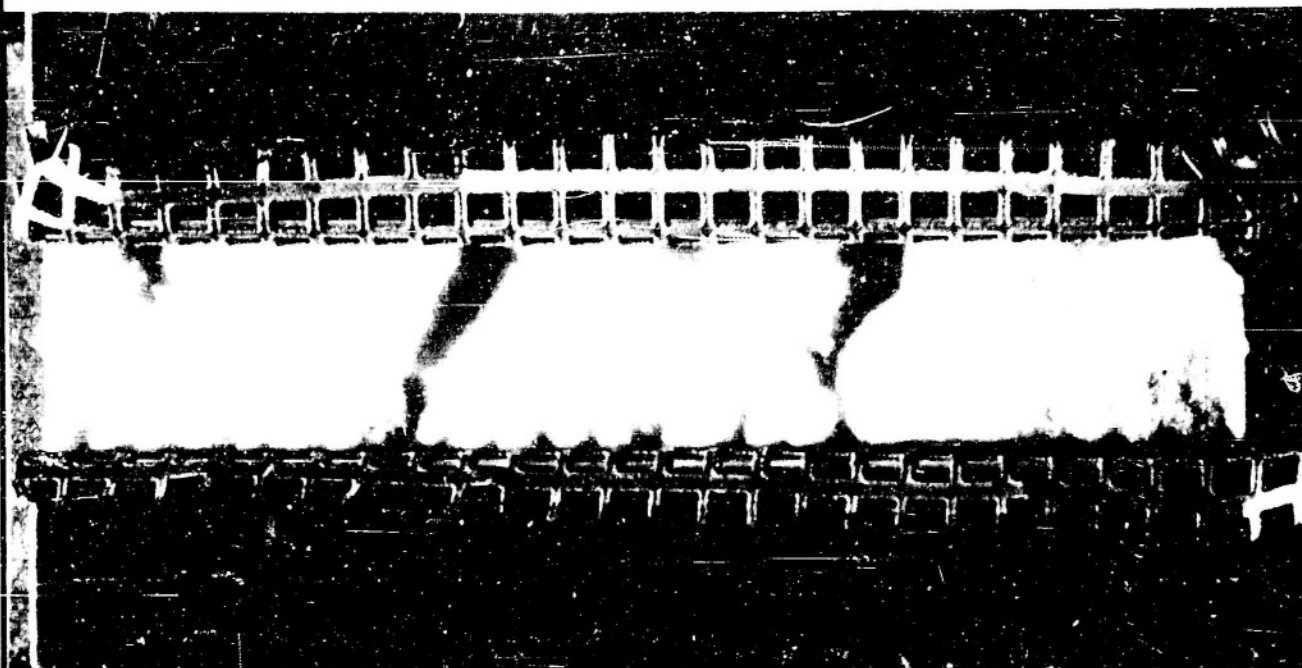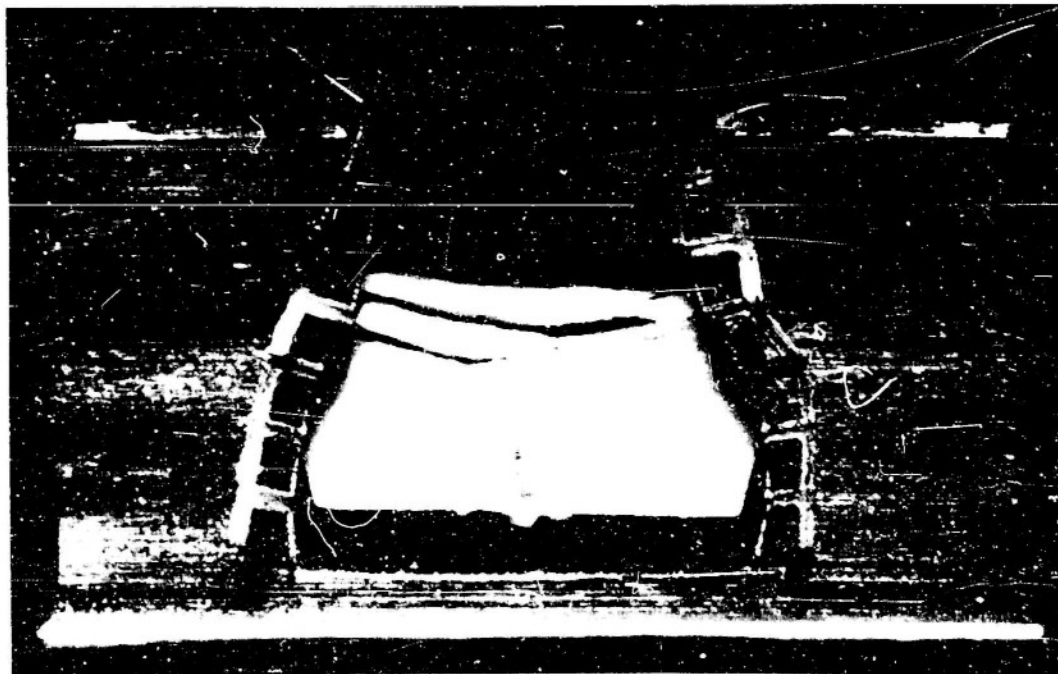
Plan View



End View

Figure 9 - Typical bearing fracture in neck of base, dovetail
specimen (approx. 9 x size).
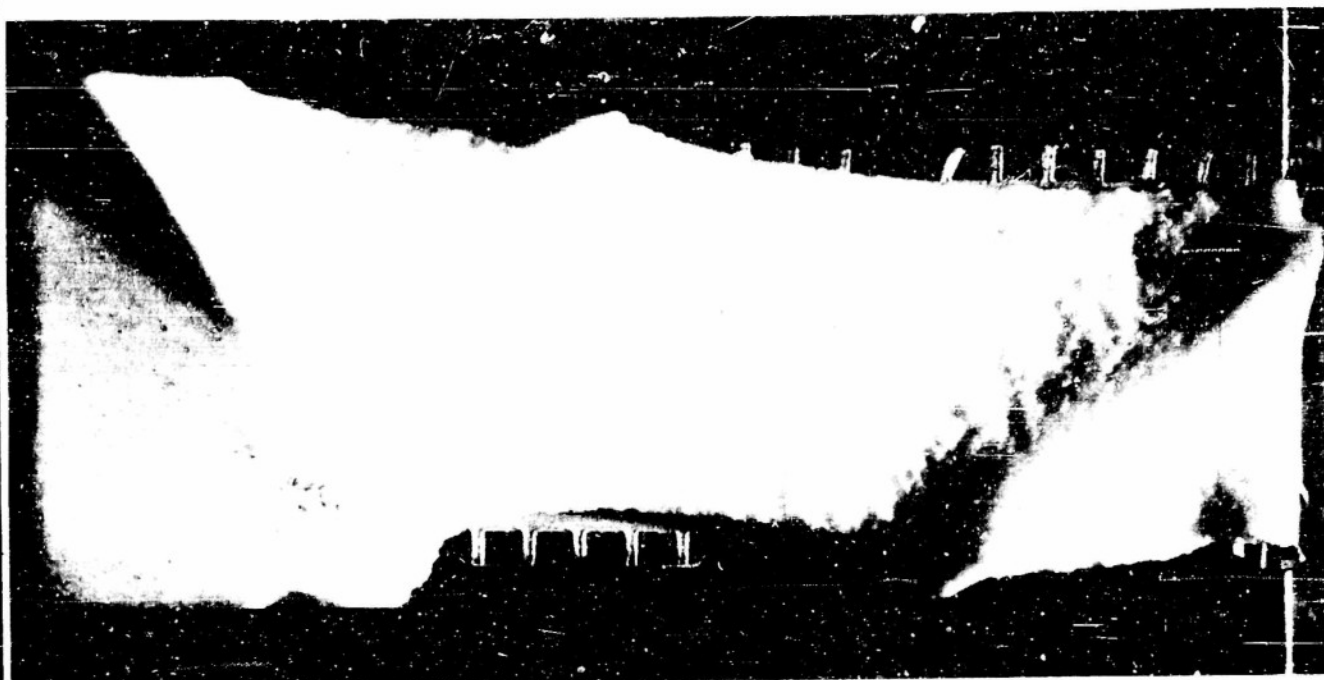
Plan View
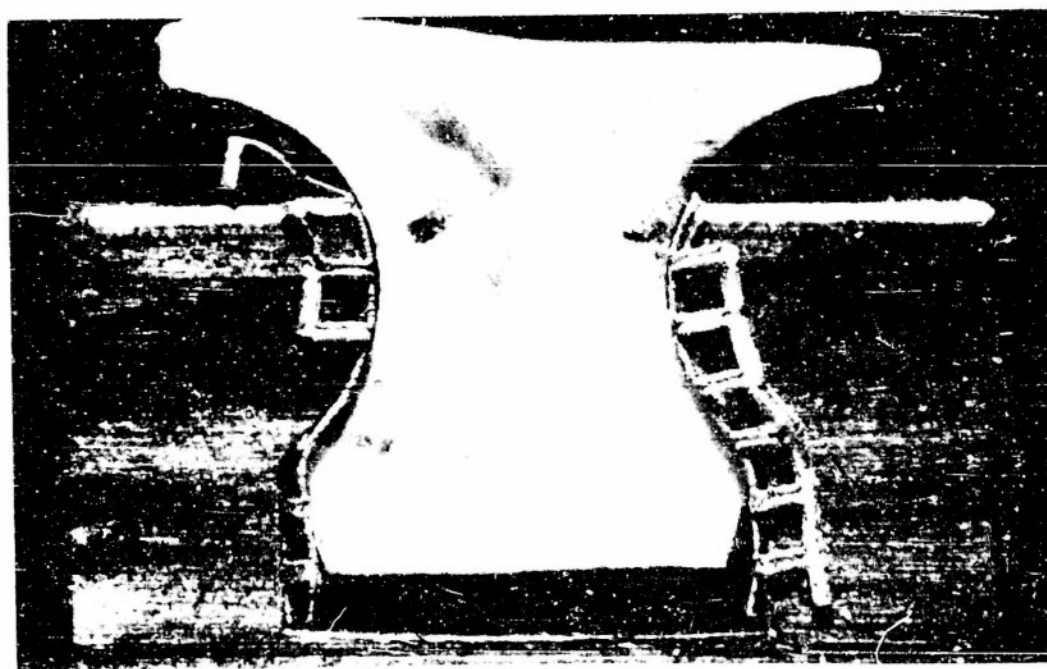


End View

Figure 10 - Typical tension fracture in blade root fillet - dovetail specimen (approx. 9 x size).
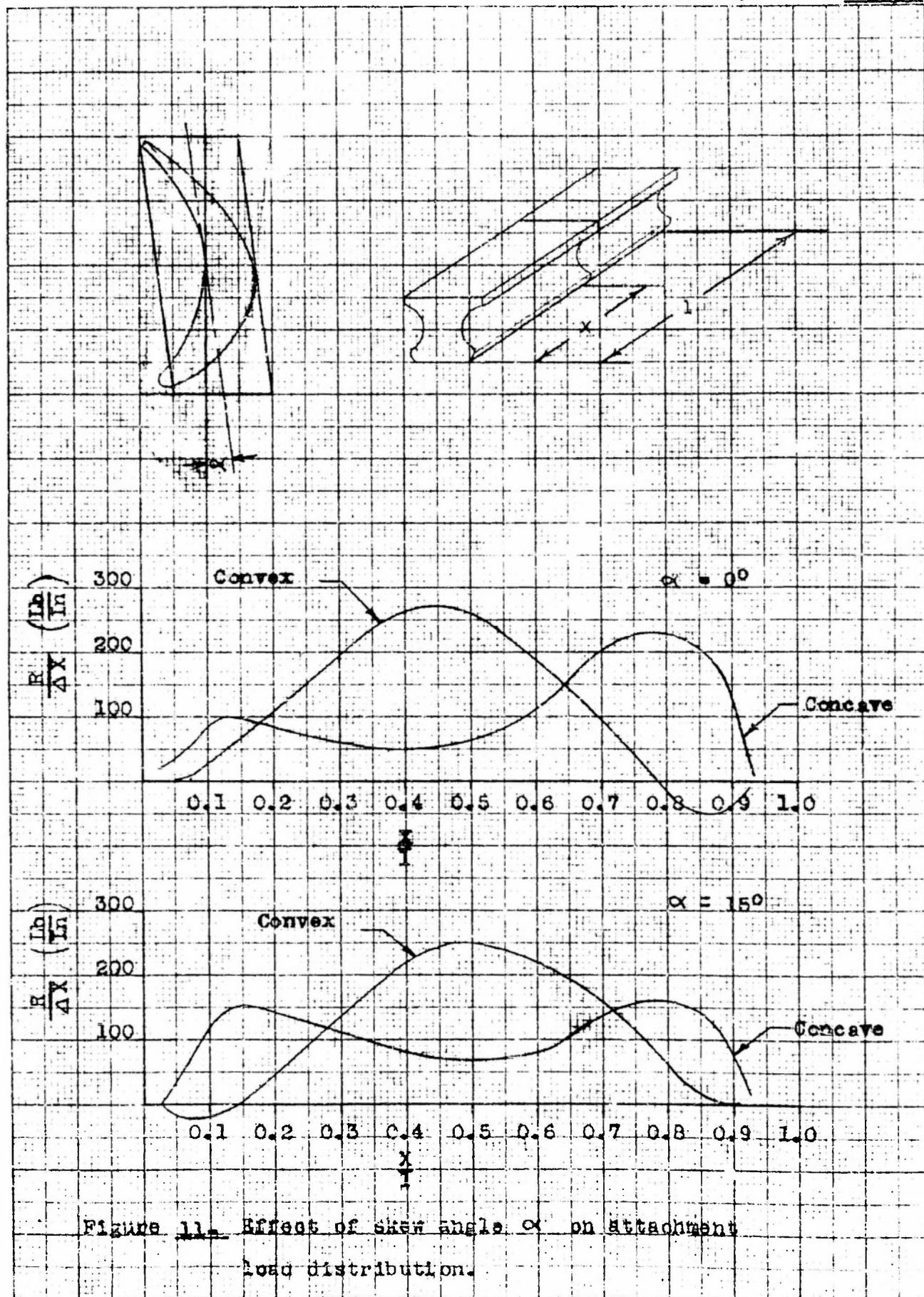
Figure 11. Effect of skew angle α on attachment load distribution.
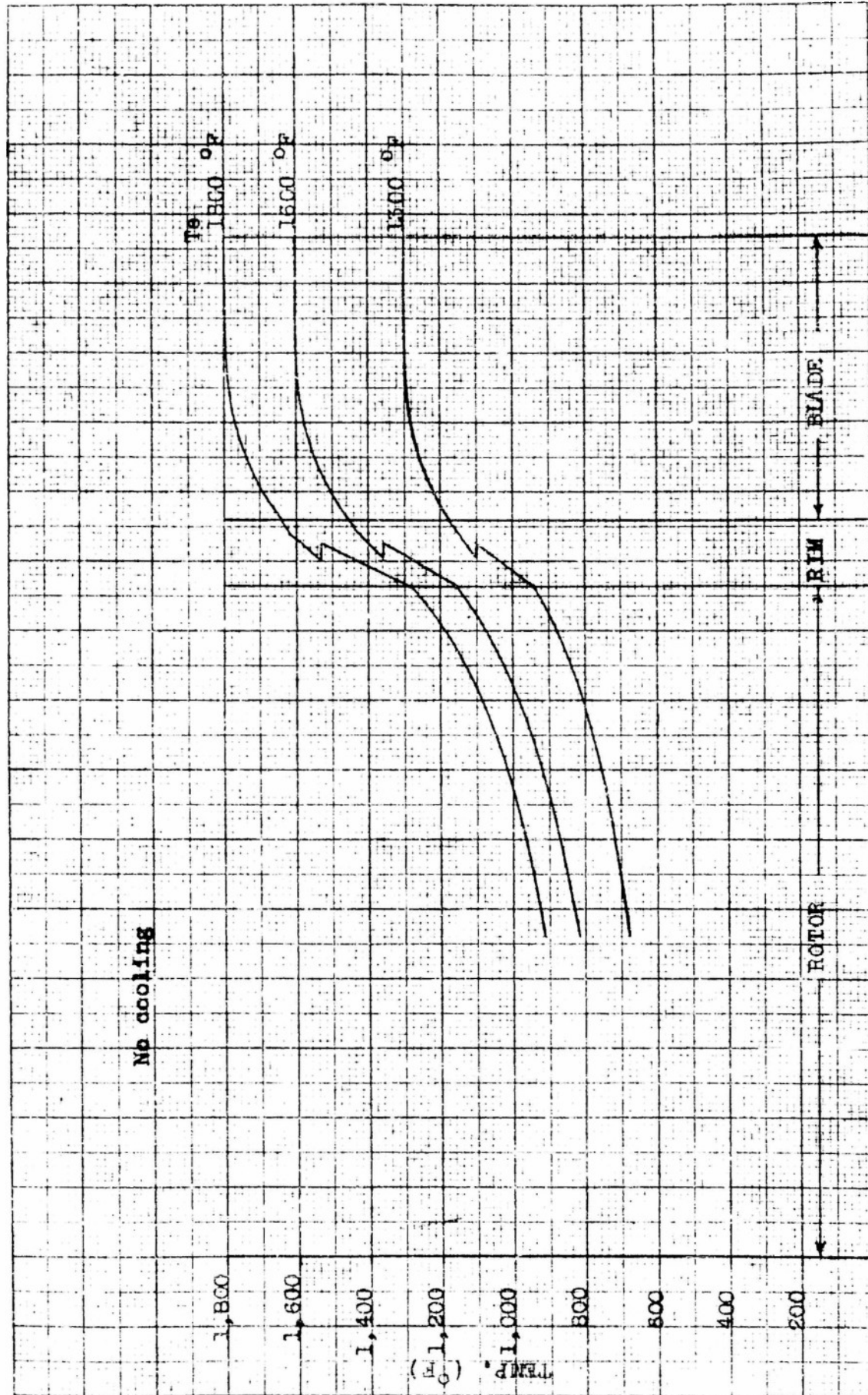
Figure 13: Effect of gas temperature on calculated temp. distribution for a small turbine with dovetail blade attachment ($D_b/dr = 2.7$) and $q_o = 30$, $q_i = 24$, $Ta = 247$, $Ta = 200\,°F$
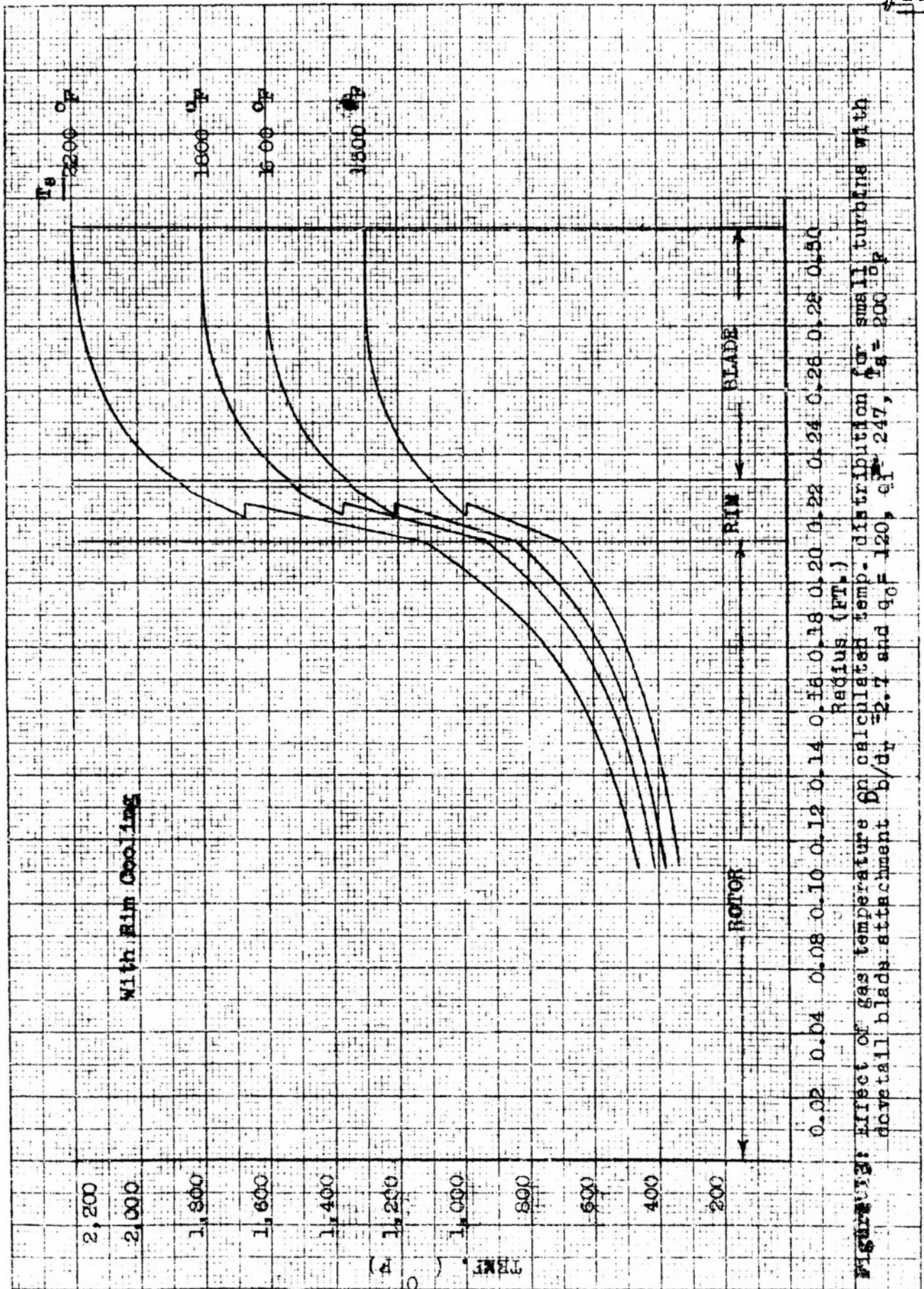
Figure 13: Effect of gas temperature on calculated temp. distribution for small turbine with dovetail blade attachment $b/d_r = 2.7$ and $q_0 = 120$, $q_1 = 247$, $T_s = 200 \,°F$

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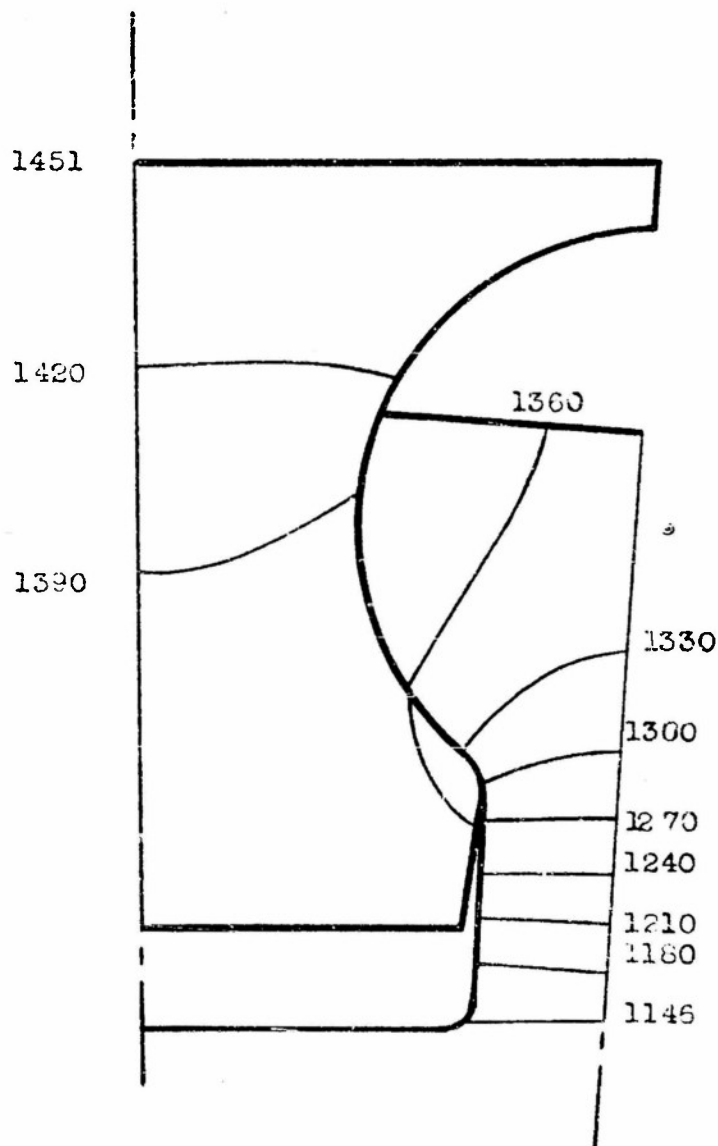
**Figure 14:** Temperature distribution through dovetail type attachment. Te = 1,600 °F , $Q_o$ = 30, $D_b/d_r$ = 2.7
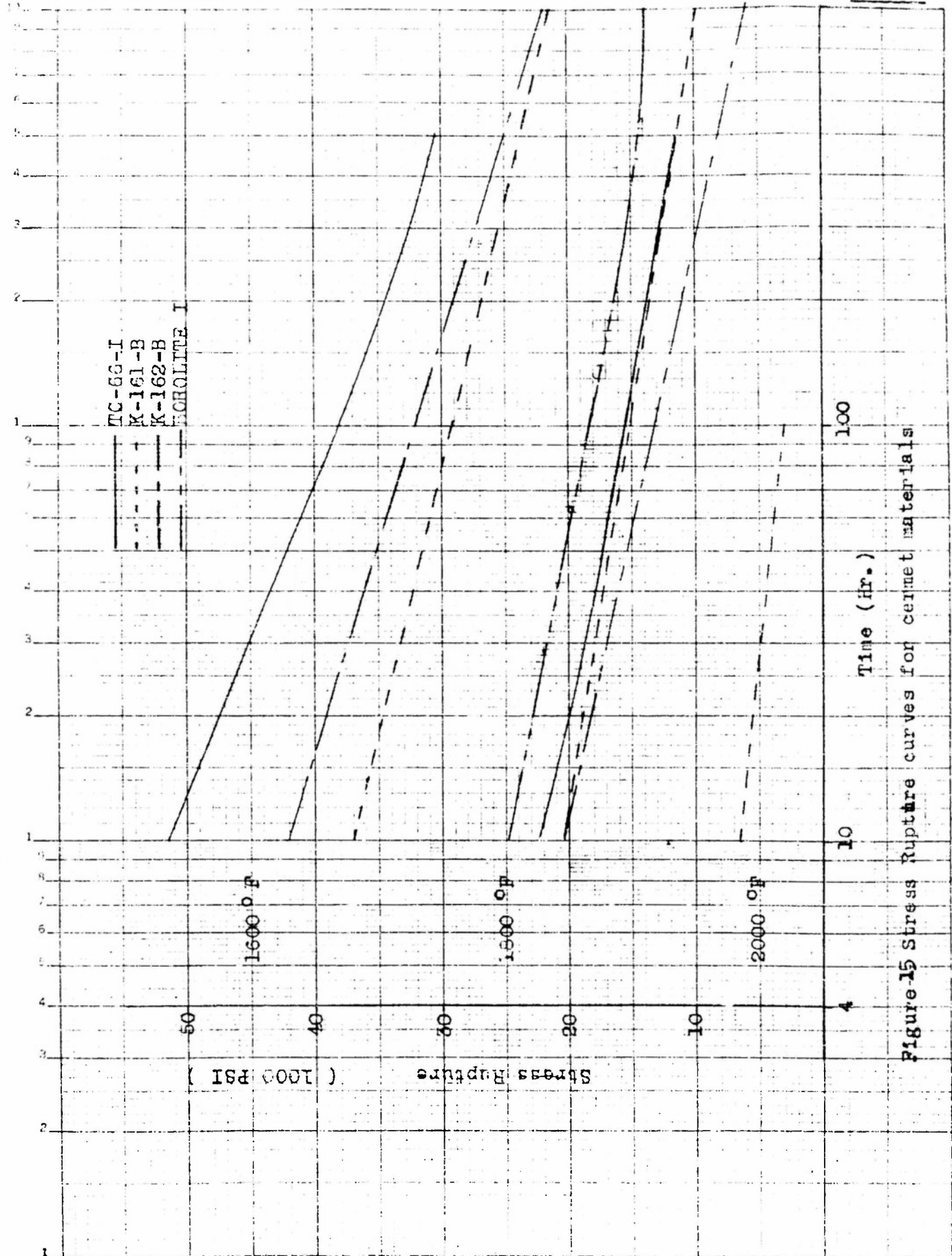
Figure 15 Stress Rupture curves for cermet materials

Figure 24 Comparison between calculated centrifugal stress and allowable stress as computed from stress – rupture data for K-162-B and calculated temperature distribution for a small gas turbine. $T_e = 1800°F$, $Q° = 120$

Figure 17 Effect of $D_b/d_r$ ratio upon allowable effective gas temperatures for 10, 100, and 500 Hr. service. Blade material K-162-B, Rotor, material 16-25-6. For a small gas turbine with cooling fins. ($Q_o = 120$)

Figure 18 Effect of effective gas temperature upon the operating life of a small gas turbine with and without cooling fins. Blade material is K 162 B
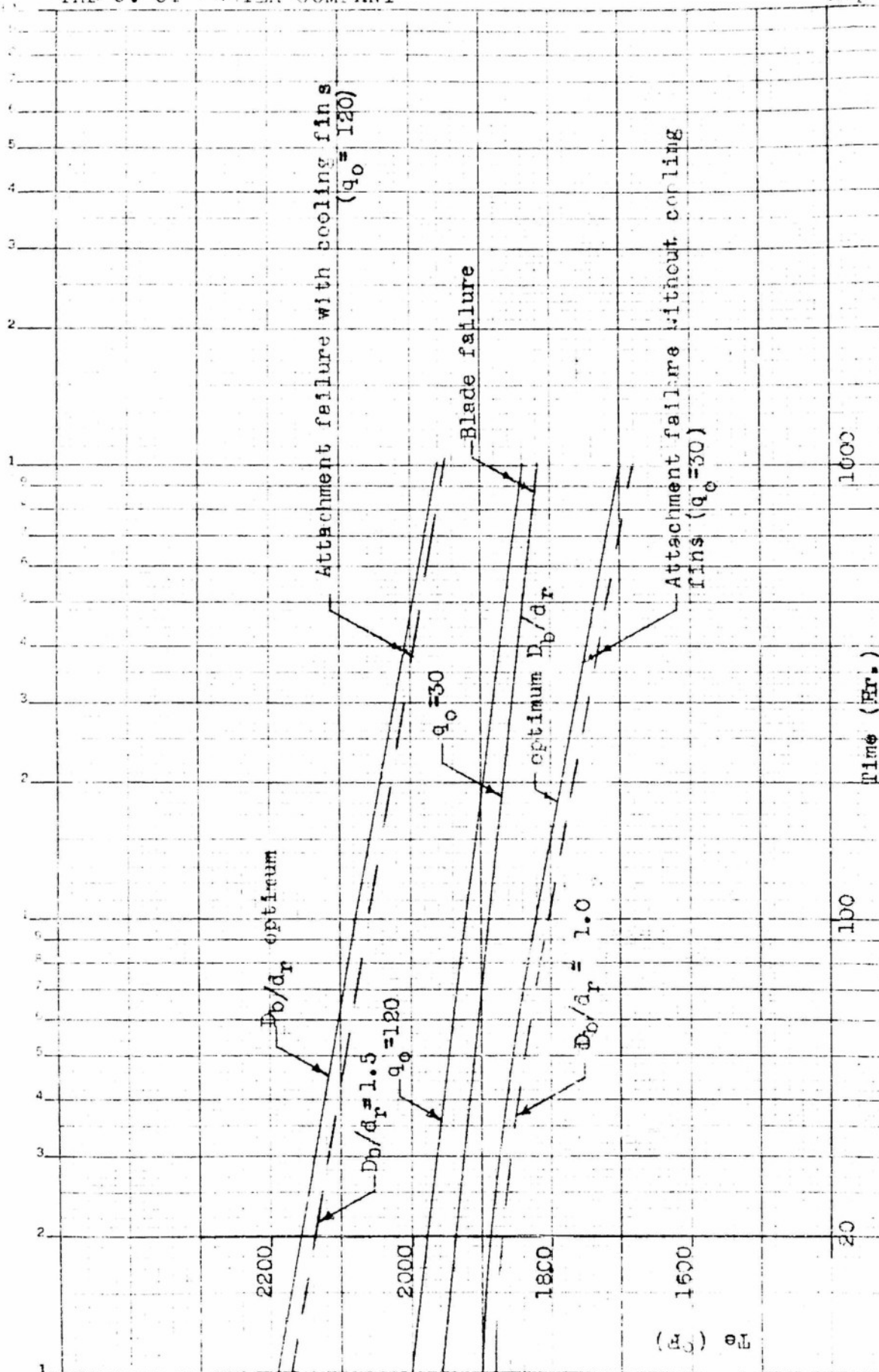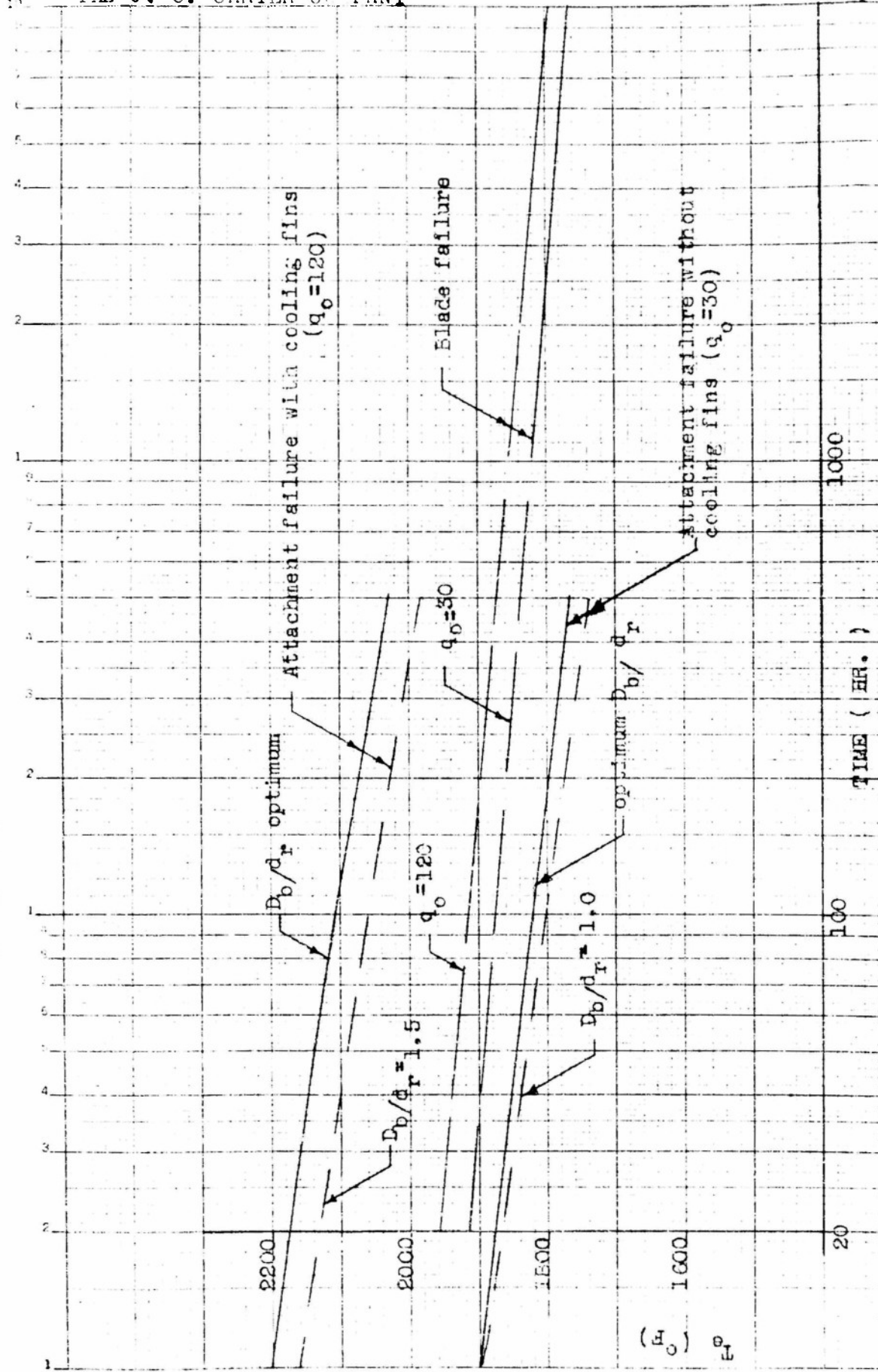
Figure 19  Effect of effective gas temperature upon the operating life of a small gas turbine with and without cooling fins. Blade Material is K 161-B

Figure 20 Effect of effective gas temperature upon the operating life of a small gas turbine with and without cooling fins. Blade material is TC-65-1.
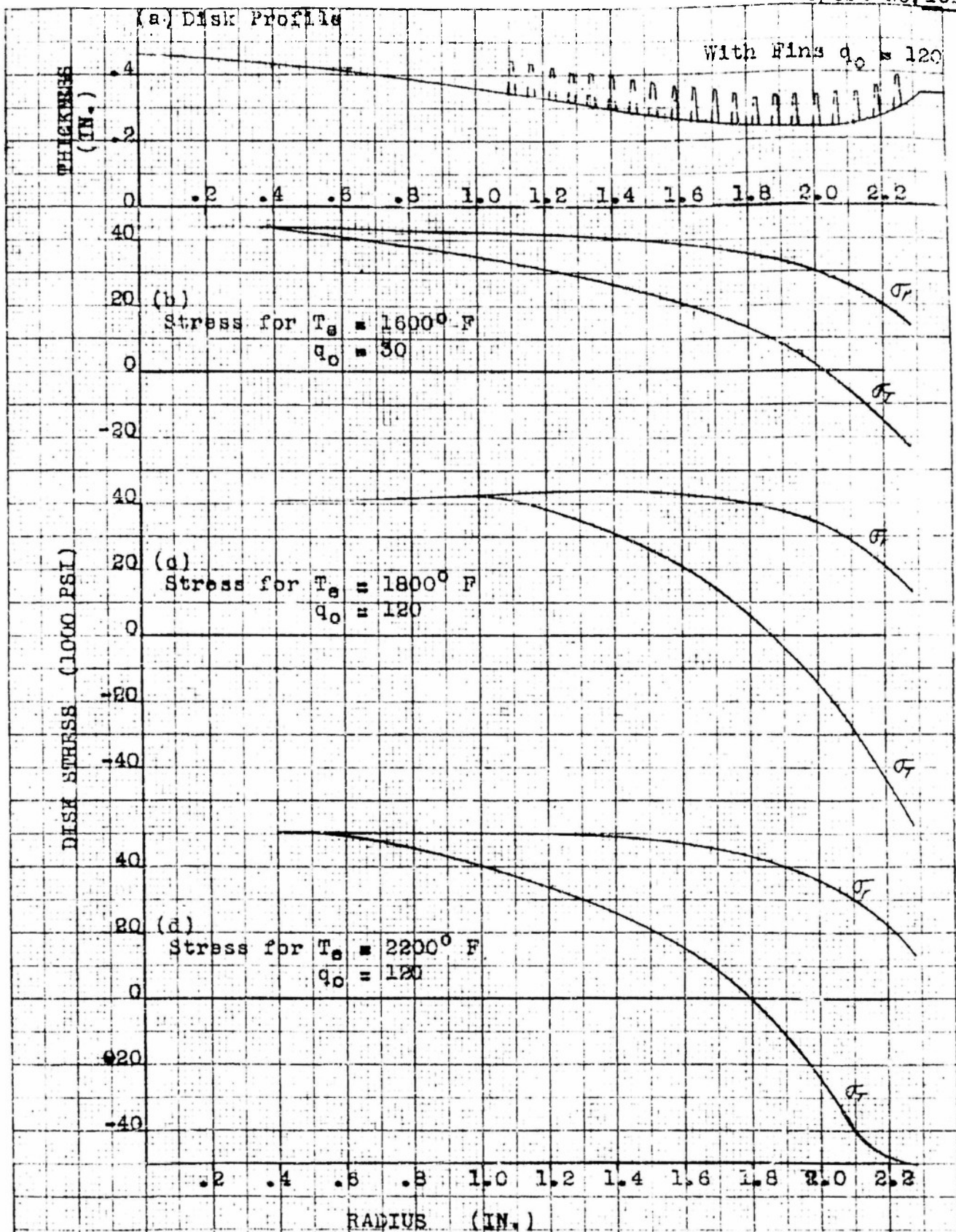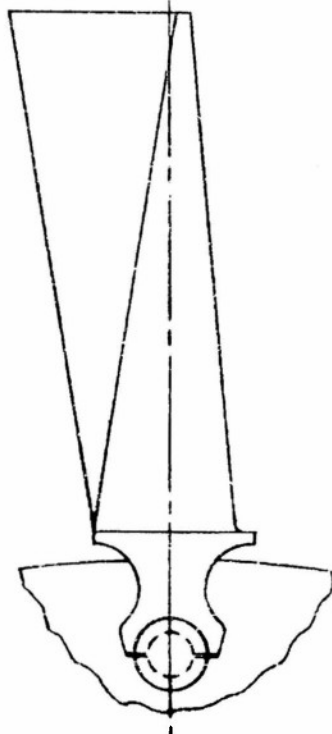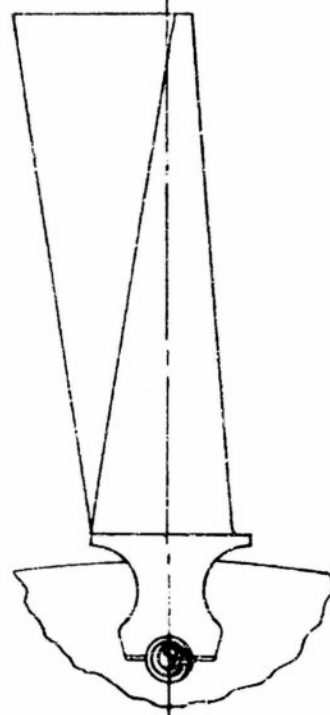
Figure 21 — Effect of gas temperature on rotor stress for a small gas turbine equipped with cermet blades at 35,000 RPM, $\sigma_{rms} = 15,000$ PSI.

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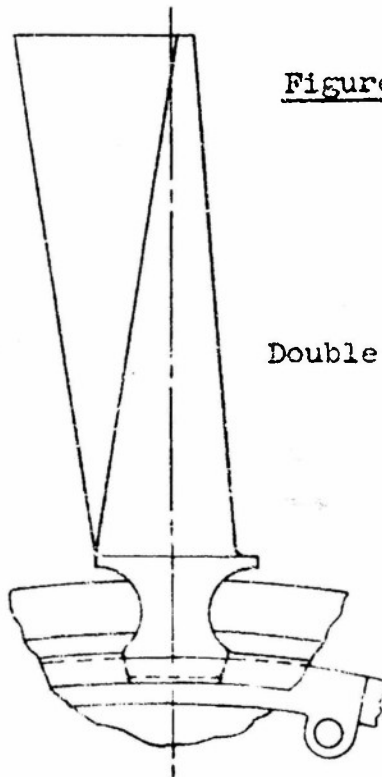      CALIFORNIA

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Rivet Retainer

Self-locking Pin Retainer

Figure 22 - Suggested blade retaining methods.

Double Snap-Ring Retainer