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**Summary of Research and
Development Effort on Air and
Water Cooling of Gas Turbine Blades**

A. P. Fraas

**OPERATED BY
UNION CARBIDE CORPORATION
FOR THE UNITED STATES
DEPARTMENT OF ENERGY**

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AND WATER COOLING OF GAS TURBINE BLADES

A. P. Fraas

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CONTENTS

	<u>Page</u>
ACKNOWLEDGMENT	v
ABSTRACT	1
INTRODUCTION	2
BACKGROUND	2
Summary of Program Costs and Running Time	6
RECOMMENDATIONS	7
PRINCIPAL PROBLEMS AND PARAMETERS	8
Methods for Air Cooling Turbine Blades	9
Principal arrangements for air cooling	9
Gains and losses associated with air cooling	11
Local heat transfer coefficients	12
Local temperature distribution	15
Stress distribution	16
Methods of Water Cooling Blades	16
Water flow supply arrangements	19
Limiting heat flux	19
Temperature distribution in a water-cooled blade	24
Thermal stresses	24
Heat losses	34
Erosion	35
Deposits in the water-cooling passages	35
Reliability	36
CURRENT STATUS OF DEVELOPMENT	37
REFERENCES	37
APPENDIX I	41

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SUMMARY OF RESEARCH AND DEVELOPMENT EFFORT ON AIR AND WATER COOLING OF GAS TURBINE BLADES

A. P. Fraas

ABSTRACT

A review of the research and development work on air- and water-cooled gas turbines from the 1904 Lemale-Armengaud water-cooled gas turbine, the 1948 to 1952 NACA work, and the present program at the General Electric Co. indicates that the potential of air cooling has been largely exploited in reaching temperatures of 1100°C ($\sim 2000^{\circ}\text{F}$) in utility service and that further increases in turbine inlet temperature may be obtained with water cooling. However, the local heat flux in the first-stage turbine rotor with water cooling is very high ($\sim 440 \text{ W/cm}^2$, or $\sim 1.4 \times 10^6 \text{ Btu/hr-ft}^2$), yielding high-temperature gradients and severe thermal stresses. Analyses and tests indicate that by employing a blade with an outer cladding of an $\sim 1\text{-mm}$ -thick oxidation-resistant high-nickel alloy, a sublayer of a high-thermal-conductivity, high-strength, copper alloy containing closely spaced cooling passages $\sim 2 \text{ mm}$ in ID to minimize thermal gradients, and a central high-strength alloy structural spar, it appears possible to operate a water-cooled gas turbine with an inlet gas temperature of 1370°C (2500°F). The cooling-water passages must be lined with an iron-chrome-nickel alloy to avoid serious trouble with corrosion of the copper, and these passages must be bent 90° to extend in a neatly spaced array through the platform at the base of the blade. The complex geometry of the blade design presents truly formidable fabrication problems. Further, the water flow rate to each of many thousands of coolant passages must be metered and held to within rather close limits because the heat flux is so high that a local flow interruption of only a few seconds would lead to a serious failure.

Heat losses to the cooling water will run $\sim 10\%$ of the heat from the fuel. By recovering this waste heat for feedwater heating in a compound cycle, these heat losses will give a degradation in the power plant output of $\sim 5\%$ relative to what might be obtained if no cooling were required. However, the associated power loss is less than half that to be expected with an elegant air cooling system.

INTRODUCTION

This is one of a series of topical reports reviewing the research and development (R&D) effort on advanced energy conversion systems.¹⁻¹¹ The first portion of each report briefly summarizes the background by giving a chronological history of the research and development on the system up to 1976 with the cumulative costs and system operating times. The next section points up the vital problem areas and recommends experiments deemed most likely to resolve these problems. This is followed by a more detailed discussion of the principal problem areas, performance parameters, and figures of merit characteristic of the system together with the current status of the development work with particular reference to progress in improving the principal performance parameters.

This report represents a special case in that it is a corollary to the reports on open-cycle gas turbines³ and combined gas turbine-steam cycles.⁴ It is concerned primarily with the special problems of water cooling the turbine blades, but inasmuch as these problems are closely related to those of air cooling of the blades, the principal elements of the heat transfer, stress analysis, and blade fabrication problems for both approaches are presented.

This work was carried out at the request of the Office of Program Planning and Analysis of the Fossil Energy Program of the Department of Energy (DOE) with funds provided for a general appraisal of advanced fossil energy systems.

BACKGROUND

As a consequence of the growing shortages of petroleum and natural gas, the combined gas turbine-steam cycle utilizing gaseous or liquid fuel derived from coal has been judged to be one of the most promising of the advanced energy conversion concepts.^{12, 13} This judgment was based on the premise that the gas turbine can be operated with an inlet temperature of over 1200°C with 1370°C (2500°F) or more commonly assumed for base load conditions. This will require the use of either an advanced air or water-cooling system for the upper-stage stator and rotor blades or ceramic blades. This report is concerned with the special problems

of air and water cooling of turbine blades; a companion report treats the problems of ceramic blades.⁶ Air cooling and water cooling of turbine blades have many points in common, particularly from the standpoint of the local heat transfer coefficients between the high-temperature combustion gases and the blade surface and the fabrication problems involved in the mass production of blades with complex internal cooling passages.

An immense background of experience with air-cooled blades has been accumulated in the course of the development of turbojet engines for aircraft, and much of this is applicable to gas turbines for stationary power plants. Water cooling has not been applicable to aircraft engines because the weight of water required would be prohibitive; however, water cooling appears attractive for stationary power plants.

A number of efforts involving water cooling of high-temperature gas turbine blades have been carried out in the past. In fact, one of the very earliest gas turbines to operate successfully, the Lemale and Armengaud 400-hp turbine in 1904, employed water cooling.¹⁴ Professor K. Brammert at the Braunschweig Technischen Hochschule included both air- and water-cooled turbines in the gas turbine work that he initiated there in 1938, and this work stimulated the initiation of British work on water-cooled gas turbines beginning in 1945 and continuing off and on to the present.¹⁵ The NACA built and tested two aluminum and one stainless steel water-cooled gas turbines in the later 1940's.¹⁶⁻¹⁸ The NACA work was covered by some 15 reports. In both the 1904 and NACA tests, the heat losses to the cooling water seriously degraded the turbine output. Fortunately, these heat losses become a smaller fraction of the system output as the size of the turbine is increased because the surface-volume ratio becomes more favorable. In the latter 1950's, the Solar Aircraft Co., under the Bureau of Ships of the U.S. Navy, designed and built a small water-cooled gas turbine with a closed cooling system. The turbine was operated a total of 219 hr but was troubled with thermal stress cracking of the blades.¹⁹ Subsequently, a Siemens project involved operation of a 1200-kW unit in Germany in the early 1960's with a turbine inlet temperature of 1000°C.²⁰ A closed water-cooling circuit was used with the water circulated out to the blades and then back to

the center of the rotor. This arrangement made it necessary to use thick-walled passages to contain the high pressures generated by centrifugal force and gave trouble with deposits of solids centrifuged out of the water near the blade tips. Work was discontinued after ~1000 hr of testing because of difficulties with plugging of coolant passages, instabilities in the cooling water flow, leaks, and most particularly severe thermal stresses in the buckets.

A different approach employing a once-through water circuit discharging a steam-water mixture from the blade tips was initiated by General Electric in the early 1960's. The first experiments were carried out with a 24.7-cm-diam (9.7-in.) simple turbine wheel with turbine inlet temperatures ranging from 1540 to 1925°C (2800 to 3500°F) and, as in the Lemale-Armengaud and NACA turbine tests, the weight flow rate of water was only a little less than that of the fuel.^{14, 17, 21} Results of the initial tests in 1964 were encouraging. There were no difficulties with water flow instabilities or passage plugging as a consequence of deposits in the short period of the tests. This work was followed a few years later by the design, construction, and operation of a second turbine in which the design took advantage of the lessons learned from the first unit. This unit was operated for some 20 hr with 50 starts and stops in a series of tests in which the maximum gas temperature into the turbine was as high as 1925°C (3500°F). A local heat flux in the turbine rotor of 440 W/cm² (1.4×10^6 Btu/hr-ft²) was measured. A third turbine was then built and is currently under test. About 100 hr of operation had been obtained on these three turbines as of February 1978. There has been no effort to run an endurance test — the first objective was to determine whether the new approach eliminated the problems encountered in the Siemens work. Subsequently, the effort has been directed mainly toward assuring adequate cooling with acceptable thermal stresses but has also included studies of corrosion in the cooling-water passages and erosion of the casing and stator by liquid droplets thrown off the rotor.²²⁻²⁷ The effort has also included the preparation of conceptual designs for full-scale systems. Figure 1 shows a typical flowsheet and Table 1 gives the principal parameters for a reference design system.

Table 1. Water-cooled gas turbine combined cycle design parameters given in Ref. 22

	Flow (lb/sec)	Pressure (psia)	Temp. (°F)
A. Air			
1. Compressor inlet	700	14.62	59
2. Bleed air			
8th stage	3.05	46.81	346.9
13th stage	13.91	96.84	523.5
Compressor discharge			
a. Turbine cooling	3.28	200.36	696.8
b. Bearing seal lkg.	2.71	200.36	696.8
3. Combustor inlet	677.05	200.36	696.8
B. Fuel			
4. Fuel heater inlet	29.48		125
5. Combustor inlet	29.48		250
C. NO_x Suppression steam			
6. Combustor inlet	8.12	300.36	417.5
D. Combustion products			
7. Combustor discharge	714.65	192.34	2877.1
8. Turbine exhaust	744.38	15.42	1384.6
9. HRSG bypass	7.44	15.42	1384.6
10. HRSG inlet	736.94	15.42	1384.6
11. HRSG outlet	736.94	14.88	300.0
12. Stack	744.38	14.70	311.7
E. Nozzle coolant			
13. Turbine inlet	98.83	1500	225
14. Turbine discharge	98.83	1250	512.4
15. Flash tank inlet	60.63	1250	512.4
16. Fuel heat exchanger inlet	38.20	1250	512.4
17. Fuel heat exchanger outlet	38.20	1250	463.9
18. Flash tank return	52.51	300	417.3
19. Deaerator inlet (nozzle return)	38.20	21.7	395.0
20. Deaerator inlet (flash tank return)	52.51	21.7	395.0
F. Bucket coolant			
21. Turbine inlet			
1st stage	19.00	250	100
2nd stage	11.27	250	100
3rd stage	4.86	250	100
22. Coolant lost to gaspath			
1st stage	5.70	103.84	337.3
2nd stage	2.82	49.14	291.8
3rd stage	0.97	21.70	253.5
23. Recovered coolant			
1st stage	13.30	103.84	337.3
2nd stage	8.45	49.14	291.8
3rd stage	3.89	21.70	253.5
24. Feedwater heater discharge	25.64	14.70	130.5
25. Makeup water	19.39	14.70	59.0
26. Polisher inlet	35.13	14.70	99.7
27. Steam turbine condenser inlet	9.90	14.70	99.7
G. Steam cycle			
28. Feedwater pump inlet	177.97	21.7	222.3
29. Economizer inlet	177.97	1485	225.0
30. Boiler blowdown	1.78	1415	588.5
31. Evaporator inlet	176.19	1415	588.5
32. Superheater inlet	176.19	1415	588.5
33. Steam turbine inlet	176.19	1250	950.0
34. Condenser inlet	176.19	1.23	108.7
35. Feedwater heater inlet	186.09	24.11	108.8
36. Deaerator inlet	186.09	21.7	134.4
Gas turbine output (MW)	176.24		
Steam turbine output (MW)	82.16		
Net plant output (MW)	258.40		
Gas turbine efficiency (%)	0.3244 (HHV)		
Combined cycle thermal efficiency (%)	0.4757 (HHV)		

ORNL-DWG 78-8060

Fig. 1. Flowsheet for a water-cooled gas turbine fueled by a coal gasifier and coupled to a conventional steam cycle (Ref. 26).

Since May 1974, the General Electric program on a water-cooled gas turbine has been supported by EPRI and subsequently by ERDA-FE (now DOE). The funding to date amounts to approximately \$8,000,000. The prime effort has entailed an analytical and experimental attack on the fundamental problems backed up by a continuing series of design studies on full-scale water-cooled gas turbines.²²⁻²⁷ The latter has included the fabrication of typical blades to investigate both fabrication processes and the effectiveness of thermal bonds together with construction of a series of simulated turbine wheels with elaborately instrumented electrically heated blades.²²⁻²⁷ Shakedown tests of the latest unit, which will operate at full-scale turbine conditions, started in February 1978.

Summary of Program Costs and Running Time

The totals for both the experimental program cost and the running time on air-cooled gas turbines are at once very large and indeterminate. The R&D work on water-cooled gas turbines in the U.S. is much more limited, so that reasonably good estimates of the cost and running time could be made to give the curves in Fig. 2. Only the NACA work around 1950, the Solar Aircraft work, and the more recent G.E. work are included in Fig. 2.

ORNL-DWG 78-8061R

Fig. 2. Summary of R&D program costs and running time on water-cooled gas turbines in the U.S. (costs are given in then-year dollars with no corrections for inflation).

RECOMMENDATIONS

The most crucial feasibility question of the water-cooled, high-temperature gas turbine is the practicality of providing water-cooling passages in such a way as to avoid severe thermal stresses in the blades, particularly in the geometrically complex region at the blade root. These questions are so intimately tied to water flow in a strong centrifugal field and the high heat fluxes associated with high gas temperatures that there seems to be no choice but to investigate them experimentally.

Erosion of the casing and downstream stator by water droplets emerging from the water-cooled rotor blade tips presents formidable problems. Experimental work is under way on several ways of dealing with this set of problems, and this work should continue. Note that water collection must

be accomplished in a fashion to avoid the serious churning losses that will act as a drag on the rotor.

The fundamental investigation currently under way at General Electric should provide a firm basis for determining whether a water-cooled gas turbine can be made attractive. The analytical work being done is excellent, the experiments are well conceived, and the test results to date are promising. However, these tests should be sufficiently comprehensive to permit interpretation of the results to include allowance for deviations from ideality in a full-scale turbine in which approximately 20,000 small cooling passages operating in parallel must each have very nearly its proper water flow rate or a failure will occur.

PRINCIPAL PROBLEMS AND PARAMETERS

A variety of methods for cooling gas turbine blades have been employed. The first and obvious step taken in the 1930's was to take advantage of thermal conduction in the turbine rotor buckets and remove heat from them by directing a cooling air stream over the rotor disk to which they were attached. This had the advantage that it reduced the blade temperature in the vicinity of the blade root, where the stresses from centrifugal force are the highest. This approach was used in the development of exhaust turbosuperchargers in the early 1940's. It was also obvious at that time that more effective cooling could be accomplished by employing hollow turbine buckets supplied with cooling air from the rotor hub. This approach was used in the early German turbojet engines and in some of the U.S. exhaust turbosuperchargers for aircraft engines.

Other approaches that have been considered include the fabrication of hollow turbine buckets partially filled with an alkali metal such as potassium and sealed in much the same way as the stems of sodium- or potassium-cooled exhaust valves for internal combustion engines. With this arrangement, heat would be removed from the blade tip by boiling of the alkali metal and transferred in vapor form to the blade root where the vapor would condense. This tends to give a uniform temperature over the length of the blade, which in turn means a higher root temperature for a given load on the air stream employed to cool the rotor, and hence is a less desirable approach than other simpler systems.

The bulk of the effort on cooling of turbine blades has been concerned with bleeding air through internal passages in the blade and exhausting it through ports into the hot combustion gas stream passing over the blade. In addition, there has been a substantial amount of work on water cooling. These two sets of work are summarized in the following sections.

Methods for Air Cooling Turbine Blades

The tremendous background on turbine blade cooling that has accumulated in the course of developing air-cooled blades for turbojet engines provides a splendid background for water cooling. In fact, a familiarity with the work on air cooling is vital to an understanding of the problems of water cooling the blades of gas turbines for utility service, and hence it seems best to begin with a review of the work on air-cooled blades. Note, for example, that the combustion gases enter the turbine with a nonuniform temperature distribution, the temperature in the middle of the flow annulus commonly running several hundred degrees higher than that near the inner and outer boundaries (i.e., over 100°C above the mean).

Principal arrangements for air cooling

The principal methods that have been used for cooling gas turbine rotor and stator blades are indicated in Fig. 3, which was taken from a summary of NASA work on blade cooling.²⁸ The simplest, most obvious approach is indicated in Fig. 3a, where the cooling air flows radially outward through passages inside the blade. Somewhat higher heat transfer coefficients can be obtained with a given cooling air flow but a higher pressure drop by making use of the impingement cooling arrangement of Fig. 3b, where air is supplied to the blade through a central perforated duct and small air jets from the duct impinge on the outer shell of the blade. This arrangement has the advantage that the coolant flow distribution can be varied readily to suit the differing requirements of local blade regions; the nose of the blade, for example, is subject to a higher heat transfer coefficient than any other portion. The third approach to blade cooling is indicated in Fig. 3c, which employs slits in the blade

ORNL-DWG 78-8062

Fig. 3. Methods of air cooling of turbine blades (Ref. 28).

that inject cooling air into the boundary layer passing over the outer surface of the blade and thus reduce the heat transfer rate from the combustion gas to the blade surface. This approach can be made more effective by using a large number of small holes rather than a few slits in the blade; such an arrangement is indicated in Fig. 3d. This approach, if carried to the limit, makes use of a porous blade shell — an arrangement called transpiration cooling (Fig. 3e). Because of the variations in local heat transfer coefficient with position around the perimeter of the blade, the blade cooling air can be used more efficiently if some

portions of the blade are cooled by convection of air streams flowing radially outward, while other portions such as the nose may be cooled by impingement, some regions can be film cooled by exhausting some of the air through slits in the blade surface, and the thin trailing edge region of the blade can be cooled by a slit as indicated in Fig. 3f. Note that ejection of the cooling air through the trailing edge of a rotor blade might be expected to give a Hero turbine type of jet reaction that would contribute to the efficiency of the turbine. In practice, it has been found that at best this arrangement avoids aerodynamic losses sustained as a consequence of disturbance of the main combustion gas stream by the ejected cooling air. Figure 3g shows a blade in which the main portion is cooled by impingement and only the trailing edge is cooled by chordwise convection. Figure 3h shows a transpiration-cooled blade that might be formed by winding layers of fine wire steeply pitched in alternating directions on a central strut which constitutes the main structural element of the blade and the assembly sintered to give a strong, rigid body.²⁹ Other variations of these basic cooling methods include elegant passages to conduct the air radially outward, return it radially inward, and then pass it radially outward again.

Extensive tests of many different geometries have been carried out by NASA and by the gas turbine manufacturers. This work has had to be closely coordinated with the development of fabrication techniques for making the complex geometries required of the difficult-to-work iron-chrome-nickel-cobalt alloys required to obtain good strength and oxidation resistance.

Gains and losses associated with air cooling

As the amount of air bled from the compressor discharge and used for blade cooling is increased, the turbine inlet temperature can be increased, thus increasing both the output from a given size engine and the thermal efficiency. However, both of these factors are offset by the pumping power losses involved because the air bled from the compressor does not yield work from the turbine stage into which it is injected or from turbine stages upstream of the stage in which it is injected. Consequently, increasing the amount of cooling airflow brings one to the

point of diminishing returns. Figure 4 shows this effect for three typical methods of blade cooling (i.e., convection, impingement, and transpiration cooling).²⁸ The overall effect on plant performance is indicated in Fig 5, which shows the effects on both thermal efficiency and specific power output as a function of the turbine inlet temperature obtainable from elegantly designed cooling provisions for several different percentages of cooling air bled from the compressor discharge.³⁰

Local heat transfer coefficients

The local heat transfer coefficient varies with position around the perimeter of the airfoil. Figure 6 shows the local heat transfer coefficient as a function of distance from the leading edge as calculated by an elegant analysis.³¹ Also shown are local surface temperatures for an uncooled blade with a turbine inlet total temperature of 1850°F. An

Fig. 4. Potentials of cooling methods. Outside metal temperature, 1255°K (1800°F); coolant temperature, 811°K (1000°F); wall thickness, 0.127 cm (0.05 in.); gas pressure, 20 atm; convection efficiency, 0.7; film cooling efficiency, 0.6; transpiration cooling efficiency, 0.8 (Ref. 28).

ORNL-DWG 78-8064

Fig. 5. Analytical estimates, for an idealized cycle, of the effects of the amount of cooling air bled from the compressor discharge on the thermal efficiency as a function of turbine inlet temperature and on the specific power output as a function of the optimum pressure ratio. The dashed lines indicate the amount of cooling air required to maintain an acceptable blade temperature (Ref. 30).

indication of the validity of the analysis technique is given in Fig. 7, which shows a curve for the calculated surface temperature along the pitch line of a cooled blade together with some experimentally measured points.

The local heat transfer coefficient around the perimeter of a turbine blade varies somewhat with the airfoil shape. Figure 8 shows another curve for the local heat transfer coefficient as a function of position around the perimeter of the blade for a different airfoil section.³² Note the similarity between Figs. 6 and 8 in that the highest heat transfer coefficient is at the leading edge, but note also that in Fig. 8 the heat transfer coefficient is also high at the 45% chord point on the suction surface of the blade and is substantial on the pressure side near the trailing edge. As implicit in Fig. 6, secondary flows

Fig. 6. Typical gas-side heat-transfer coefficients and adiabatic wall temperatures (Ref. 31).

Fig. 7. Comparison of analysis of blade temperature distribution with engine test data (Ref. 31).

ORNL-DWG 78-8067

Fig. 8. Typical curve for the local heat transfer coefficient at the blade surface (Ref. 32).

lead to differences in the chordwise position of the transition from laminar to turbulent flow on the suction side of the blade; in this instance, the transition is delayed substantially near the hub while it occurs quite early near the tip. If slots are provided in the blade for film cooling as in Fig. 3f, aerodynamic conditions in the boundary layer are obviously affected and can substantially reduce the nominal local heat transfer coefficient.

Local temperature distribution

It is interesting to examine some local temperature distributions both as calculated and as measured in experiments. Figure 9 shows an interesting case for which both calculated and experimentally determined temperatures are available for a transpiration-cooled turbine rotor blade. Note that in spite of the difficulty in estimating the complex effects of cooling airflow through a porous blade surface into the boundary layer of the high-velocity air stream over the airfoil, it apparently was possible in this case to calculate the local temperature within $\pm 200^\circ\text{F}$ of the values found in experiments.

Fig. 9. Calculated and measured airfoil temperatures in a transpiration-cooled turbine rotor blade (courtesy Curtiss-Wright Corp.).

Stress distribution

Variations in the local temperature through the blade section lead to thermal stresses as a consequence of differential thermal expansion. These thermal stresses are superimposed on the stresses induced by centrifugal force coupled with the bending stresses on the blades stemming from the aerodynamic load. Figure 10 shows both the temperature and the stress distributions at the midsection for a typical air-cooled blade. Note that in this instance the hottest element of the blade occurs on the suction side about 30% of the chord while the coolest section of the blade is the trailing edge. These temperatures and stresses were calculated for the local heat transfer coefficient distribution shown in Fig. 8; hence it is not surprising that the peak temperature in the blade is as indicated.

Methods of Water Cooling Blades

At first glance, one might consider quite a variety of water-cooled blades in a turbine rotor. Perhaps the first logical possibility would

ORNL-DWG 78-8068

Fig. 10. Calculated distributions of temperature and stresses at the midsection of a cooled rotor blade under steady-state operating conditions with a gas temperature of 986°C and a cooling air inlet temperature of 380°C (Ref. 32).

be to circulate water radially outward through one set of passages of the blade and then return it to the rotor through another set. This approach was employed in NACA tests of two aluminum turbine wheels and one stainless steel wheel in the 1948 to 1952 period. The wheel diameters were about 30 cm (1 ft). The bulk of this work was carried out with rotor tip speeds of 214 to 305 m/s (700 to 1000 fps) with turbine inlet temperatures of 540 to 1095°C (1000 to 2000°F). Both natural thermal convection and forced convection flow of the water through the rotor were investigated, and a wealth of heat transfer data were obtained.¹⁷⁻¹⁹ Siemens, in Germany, subsequently built a full-scale 1200-kW gas turbine employing this method of blade cooling.²⁰ The testing included 700 hr at temperatures above 1000°C (1830°F). However, the blade tips operate in a centrifugal field of 10,000 to 20,000 g , leading to a radial pressure differential between the rotor and the blade tip of hundreds of atmospheres; this gives severe pressure stresses in the blades which are superimposed on the stresses from centrifugal and aerodynamic loads as well as severe thermal stresses. To avoid the pressure stresses, the water can be admitted to the blade root and allowed to flow radially

outward to a discharge port near the blade tip with boiling occurring along the length of the passage.²¹ The water flow rate must be limited to avoid an excessive water consumption and quenching of the hot gas stream. Ideally, one would like to have a thin film of water flowing on the wall while steam would flow through the central portion of the cooling passage. This approach is complicated by the fact that the water film not only is subject to centrifugal forces but also to Coriolis forces associated with the tangential acceleration of the water as it flows radially outward along the blade.²⁴ Consider the cross section through such a passage as shown in Fig. 11. As a consequence of the Coriolis force, the liquid film is swept to the trailing side of the circular water passage to give the crescent shape shown in Fig. 11. The lower radial velocity in the boundary layer reduces the Coriolis forces in that region and leads to the secondary circulation shown with the small arrows in Fig. 11. Evaporation takes place from the surface of the liquid film rather than by the formation of tiny bubbles at the heated surface as in conventional nucleate boiling.

Fig. 11. Schematic view of the fluid mechanics in the cooling passage for a water-cooled turbine rotor blade (Ref. 24).

The concentration of the liquid film along one side of the cooling passage has been found to be so pronounced in test work that ordinarily only about 20% of the perimeter of a radial passage having a bore of 1.1 mm is effective in transmitting heat from the hot metal to the liquid film. Slanting the passage so that it is swept back against the direction of rotation has been found helpful by opposing centrifugal force to the Coriolis force.²⁴ However, canting the passages in this way presents problems in cooling the leading edge of the turbine blade near the tip, thus complicating the blade geometry and hence the fabrication. It is also possible to texture the internal surface of the passage in such a way as to spread the crescent-shaped meniscus to cover as much as 45% of the wall perimeter.²⁵

To provide a good water flow distribution around the perimeter of the airfoil with the limited water flow rate available, it appears best to make use of an array of small-diameter passages arranged around the perimeter of the airfoil as shown in Fig. 12. The flow to each passage must be individually metered to restrict the water flow to the proper level.

Water flow supply arrangements

Getting water into the blade root presents a problem. A promising configuration entails the use of a groove near the rim of the rotor into which a water jet can be directed as in Fig. 13 so that it enters the rotor at a pressure substantially equal to that of the local hot gas (Refs. 25, 26, 33). Small passages in the blade root distribute the water to the radial passages as indicated in Fig. 14. Water feed from the bore of the rotor is also being considered.

Limiting heat flux

Extensive experience with heat transfer under boiling conditions has shown that the two principal parameters are the local heat flux and the temperature difference between the hot metal surface and the saturation temperature of the boiling liquid for the pressure at which it is operating.³⁴ As the heat flux is increased, the temperature difference between the metal surface and the saturation temperature of the boiling liquid increases relatively little until a critical heat flux

Fig. 12. Stage 1 bucket coolant tube spacing at pitch line for high and low water film coverage (Ref. 25).

is reached; after this further increases in the metal surface temperature lead to a drastic reduction in the heat flux (see Fig. 15). The peak heat flux is commonly alluded to as the "critical heat flux" or "burnout." For conventional stationary boiler tubes, the burnout condition is

ORNL-DWG 78-8071

Fig. 13. Conceptual design for a water-cooled gas turbine (Ref. 26).

commonly reached when the vapor quality is 30 to 70% with heat fluxes of the order of 30 to 100 W/cm² (100,000 to 300,000 Btu/hr-ft²).

The situation in a water-cooled turbine blade is clearly substantially different from that in a stationary boiler tube; hence, experiments have been run in the G.E. program to investigate the situation.^{24, 25} Figure 16 shows a typical set of data for the heat flux to a 1.1-mm-diam, 12.5-cm-long passage in an electrically heated copper block mounted on a rotor that gave a centrifugal field of 8000 *g*. Analyses showed that the critical, or burnout, heat flux (i.e., the heat flux at which the metal temperature began to increase rapidly) nominally was about what one would expect in static boiler tubes. However, a detailed analysis carried out to determine the fraction of the passage perimeter wetted by the water film indicated that only about 20% of the perimeter was wetted; hence, the local heat flux under the wetted film was about five times what one would expect in static boiler tubes. This probably stems from the high radial velocity and the large Coriolis forces acting to keep the film tightly

Fig. 14. First-stage bucket platform cooling and water distribution (Ref. 25).

Fig. 15. Heat flux to boiling water as a function of the difference between the metal temperature and the saturated temperature of the water.

Fig. 16. Experimental data for the metal temperature in the wall of a 0.043-in.-ID G.E. Status-design cooling passage with a reduced 0.0025-gpm flow in the rotor of an electrically heated simulated blade in a test rig (Ref. 24).

against the heated surface. Subsequent tests indicate that with approximately 2-mm-diam passages, special treatment of the internal surface of the passage, and canting the passage to reduce the Coriolis forces, it appears possible to increase the fraction of the wetted perimeter to ~45% and the total heat removal capacity correspondingly.²⁵⁻²⁷

Temperature distribution in a water-cooled blade

At first, one might expect to be able to employ the array of small water-cooling passages of Fig. 12 in otherwise solid stainless steel or nickel alloy blades. A brief look at the heat transfer, temperature distribution, and thermal stress situation, however, quickly shows that in the first stage the heat flux is so high that this approach would give completely unacceptable thermal stresses, although it can be used in the third stage and possibly in the second stage where the heat fluxes are lower. This stems from the fact that with hot gas temperatures of 1370 to 1650°C (2500 to 3000°F) and cooling water saturation temperatures of 170 to 235°C (340 to 455°F) corresponding to gas pressures of 8 to 30 atm and with blade surface temperatures of 540 to 650°C (1000 to 1200°F), the gas film temperature drop will be around 850°C (1500°F). From Figs. 6 and 8, one can see that for conventional 1000°C gas turbines the local heat transfer coefficient will run as much as 2800 W/m²-°C (500 Btu/hr-ft²-°F), which gives a local heat flux of about 240 W/cm² (750,000 Btu/hr-ft²). At the higher pressure ratios required to take advantage of turbine inlet temperatures in the 1370 to 1650°C (2500 to 3000°F) range, the heat transfer coefficient would be still higher. As an example, in the small (24.7-cm-diam) water-cooled turbine run by G.E. at 16 atm, a heat flux of 440 W/cm² (1.4 x 10⁶ Btu/hr-ft²) was measured experimentally.²¹ Such high heat fluxes will lead to much steeper temperature gradients in the metal blade between the heated surface and the cooling passage than in air-cooled blades.

Thermal stresses

In the extensive studies of water-cooled gas turbines carried out by NACA around 1950, the severe thermal stresses associated with the high heat fluxes and steep thermal gradients caused serious concern even with

solid aluminum alloy blades. The thermal conductivity for these alloys is nearly ten times that of the iron-chrome-nickel alloys ordinarily used in gas turbines. For example, in a homogeneous iron-chrome-nickel alloy blade for a heat flux of 330 W/cm^2 (10^6 Btu/hr-ft^2), the temperature drop across a 2.5-mm thickness would be $\sim 400^\circ\text{C}$ (720°F). Inspection of Fig. 12 indicates that the bulk of the metal would be at a temperature perhaps 50°C above that of the saturation temperature of the water [i.e., around 230°C (450°F)]; hence, a great deal of elastic and plastic deformation would take place in the surface skin, particularly in the hot streak along the leading edge and probably an additional hot streak on the suction surface at 30 or 40% of the chord. Changes in power level for the plant would lead to changes in the gas inlet temperature and hence in the blade temperature, so that these power changes, coupled with startups and shutdowns, would subject the hot streak regions to severe thermal strain cycling.

A classic curve showing the relation between the temperature range in thermal strain cycles and the numbers of cycles to failure is presented in Fig. 17, which was obtained experimentally for stainless steel for a metal temperature of 593°C (1100°F).³⁵ Note that for the 400°C

Fig. 17. Effects of strain range on the number of cycles to failure for type 347 stainless steel cycled at 1100°F with uniaxial constraint.

temperature differential cited above, Fig. 17 indicates that, in a homogeneous stainless steel blade, failure would occur in about 300 cycles if the stresses were uniaxial and no other stresses were present. In a turbine rotor blade, the thermal stresses will be biaxial and will be superimposed on large stresses from the centrifugal load on the blade. Thus failure should be expected in a much smaller number of cycles — probably between 10 and 100. Note that this rough analysis is for an ideal simple slab geometry, whereas in an actual blade (see Fig. 12) the heat flow path from the surface to the portion of the passage cooled by the liquid film would be longer than 2.5 mm; in the critical region close to the leading edge, the metal cross-sectional area effective in heat conduction from the hot surface to the reduced area of the cooling liquid meniscus would roughly double the heat flux between adjacent cooling passages. This would lead to temperature differences greater than used in this example. Further, the cooling passages constitute stress concentrations.

Conventional gas turbine blade materials are stronger than stainless steel, which helps the situation, but they are less ductile and hence may be more subject to failure from strain cycling. The situation is quite complex. Many factors influence the ability of a material to withstand thermal strain cycling. These include its ductility, the temperature at which it is operated, and the range of plastic strain per cycle. Although comprehensive test data are not available for most of the materials and conditions of interest here, a recent paper by Coffin³⁶ reviews an extensive set of experimental data on the low-cycle fatigue of materials at high temperature and shows that good correlation can be obtained through the use of the empirical equation presented in Appendix I. In the appendix, this has been applied following the analysis of Ref. 37 using basic thermal stress equations to derive the curves of Fig. 18 using the physical property data of Table 2. Note that copper at 20°C gives about the same resistance to thermal strain cycling as stainless steel at 600°C and that Haynes 188, even at 600°C, is more resistant than either copper or stainless steel. Experimental data for IN 718, an exceptionally strong high-temperature alloy tentatively chosen by G.E. for their water-cooled turbine blade spar, were found in Ref. 38 and are

Table 2. Properties of materials used for estimating low-cycle-fatigue life

Material property	Type 316 SS 600°C	Haynes 188 500°C	IN 718 550°C	A-286 430°C	Commercial oxygen-free Cu 150°C	Glidcop AL-20 425°C	AMAX-MZC 425°C
c_p , cal/g-°C	0.105	0.12	0.104 ^a		0.092		0.094
ρ , g/cm ³	8.0 ^a	8.95 ^a	8.2 ^a		8.96 ^a		8.8 ^a
k , cal/sec-cm-°C	0.041	0.0475	0.049	0.045	0.923	0.81	0.74
$\alpha \times 10^6$ /°C	17	14.8	14.2	16.7	16	19.6	20
$E \times 10^{-6}$							
psi	21	27.6	25.5	25	17 ^a	16 ^a	20 ^a
kg/cm ²	1.48	1.94	1.79	1.76	1.2	1.12	1.4
σ_y							
psi	10,000	43,800	80,000	60,000	5,000	26,000	55,000
kg/cm ²	700	3,100	5,600	4,200	350	1,760	3,800
$\Delta\epsilon_{1/4}$	0.4	0.7	0.20	0.20	0.4	0.07	0.09
μ	0.30	0.30	0.28		0.36	0.36	0.36

^aValue at room temperature; elevated-temperature data not available.

Fig. 18. Equivalent temperature range as a function of the number of cycles to failure as calculated (see App. I) for the principal alloys of interest for water-cooled turbine blades.

plotted in Fig. 18. (This alloy has the best resistance to strain cycling at high temperatures that the writer has ever seen.) The principal characteristics desired in an alloy for resistance to thermal strain cycling are a high thermal conductivity to minimize the temperature differential for a given heat flux, a low coefficient of thermal expansion and a low modulus of elasticity to minimize the thermal stress for a given temperature differential, and a high elastic limit and a high ductility at the desired operating temperature to withstand the thermal strain cycling.

In using Fig. 18, it should be remembered that in strain cycling laboratory test work, alloy specimens are held at a constant temperature and are subjected to mechanical rather than thermal strain cycling. The strain ordinate in the figure is replaced by the temperature range that would induce a thermal strain equivalent to the mechanical strain required to produce failure in the mechanical strain cycling tests. Thus Fig. 18 is a bit artificial, especially for large strains and few cycles, because the properties of the alloys vary with temperature. At first thought, one might employ a second scale for the mechanical strain per cycle, but this cannot be done because the conversion from thermal to mechanical

strain involves the modulus of elasticity and the coefficient of thermal expansion, both of which vary from one alloy to another. Thus in trying to make a direct comparison of different alloys from the thermal strain cycling standpoint, the scales of Fig. 18 are by far the most convenient. One simply must keep in mind that they are an approximation made with the simplifying assumption that they are for a set of ideal materials having the temperature-invariant properties of the materials specified at the temperatures specified. For the region of interest here the peak temperature in the thermal cycle is that for which the curves were drawn.

If one compares the equivalent temperature range indicated by the experimental data of Fig. 17 with the value for stainless steel given by Fig. 18 at the 4000-cycle level (the G.E. design objective), one finds that the experimental value is roughly double the value given by Fig. 18. This stems from the fact that test data were obtained with uniaxial strains in relatively slender test specimens, whereas the synthesized curves were for biaxial thermal stress conditions in a plate geometry. This geometric difference stems from the effects of Poisson's ratio, μ , which are accounted for in Eq. (6) of Appendix I by the term $(1 - \mu)/(1 + \mu)$, which makes biaxial stresses nearly twice as severe as uniaxial conditions. Note that the thermal stresses in a turbine blade in the course of a temperature transient are biaxial, and the surface stresses are tensile for an air blast quench.

The temperature differential between the outer surface and the coolant passages can be reduced by employing copper in the blades. Since copper could not withstand the centrifugal stresses, the oxidizing conditions on the hot gas side or the corrosive attack of high-pressure, high-temperature steam, it will be necessary to clad the outer surface of the copper with a high-strength oxidation-resistant material and make the central structural spar and the cooling passage liners of a high-strength iron-chrome-nickel alloy that has a higher coefficient of thermal expansion.^{24,25} The current G.E. choices for these materials are, respectively, IN 718 (an exceptionally high-strength Cr-Ni alloy containing Mo, Mn, V, Ti, and Si) for the central spar, A-286 for lining the cooling passages, and IN 617, an exceptionally high-strength alloy of ~54% nickel for the cladding. These

alloys will have to be thermally bonded to the copper, and the bonds will have to withstand severe thermal strain cycling.

Some interesting insights into the effects of design parameters on the temperature of both the outer skin and the copper core are given by Fig. 19, which was prepared by G.E. The base case is represented by the outer skin temperature of 538°C (1000°F) with an assumed 100% coverage of the capillary tube walls by the liquid film. Then, using a computer

Fig. 19. Analytical estimates of the effect of the fraction of the coolant passage surface covered by the type of cooling water on bucket temperature for several sets of conditions (Ref. 24).

analysis, G.E. calculated the skin and copper temperatures for progressively lower fractional coverage of the capillary cooling passages by the liquid film assuming a heat transfer coefficient between the copper and the water of $15.3 \text{ W/cm}^2\text{-}^\circ\text{C}$ ($27,000 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$). Note that this was done for a saturation temperature in the cooling water of 205°C (400°F). Three additional cases were then considered. For the first, the upper temperature limit in the copper imposed by creep stress considerations was taken as 432°C (800°F), and this in turn imposed a requirement that at least about 50% of the coolant passage must be wetted by the water film. This is more than double the coverage found experimentally in the tests of simple radial passages, a point for which case is plotted in the upper left; the low coverage of $\sim 20\%$ would have yielded a skin temperature of nearly 815°C (1500°F) with a nominal heat transfer coefficient of $4.5 \text{ W/cm}^2\text{-}^\circ\text{C}$ ($8000 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$). The third condition considered was for a larger diameter tube [i.e., 1.75 mm (0.070 in.)] instead of 1 mm (0.040 in.). This still gave too high a temperature in the copper. By reducing the hot gas temperature from the reference design case of 1538°C (2800°F) to 1427°C (2600°F), it was found possible to bring the copper temperature within the creep stress limited temperature of 427°C (800°F). More recent designs envision a cooling passage ID of 2.3 mm (0.090 in.). Note that much of the copper would be close to the saturation temperature of 204°C (400°F) for any of the cases considered.

Any attempt to appraise the thermal stress problem in the blade geometry of Figs. 12 and 14 requires specific physical property data for the materials to be used. Table 2 gives data for some of the leading candidates. Note that stainless steel and even Haynes 188 have lower yield points than the IN 718 and A-286 chosen by G.E. and that commercial oxygen-free copper has a far lower yield point than either of two new copper alloys recently placed on the market [i.e., Glidcop (a dispersion of Al_2O_3 in pure copper) and AMAX-MZC ($\text{Cu} + 0.06\% \text{ Mg}$, $0.15\% \text{ Zr}$, and $0.40\% \text{ Cr}$)].

Elegant analyses of the temperature and thermal stress distribution in the rotor and stator blades have been under way at G.E. The results of one typical estimate of the temperature distribution in a blade element are shown in Fig. 20. Note that the central spar is massive relative

Fig. 20. Model for stage-1 bucket thermal stress (temperatures in °F) (Ref. 25):

to the rest of the blade, so that differential expansion and hence local thermal stresses depend on the difference between the local temperature and the mean temperature of the central spar. For example, note that the 475°F in an element of the A-286 coolant passage liner gives a temperature differential of $\sim 240^\circ\text{F}$ between this local region and the mean temperature of the spar. The resulting thermal stress would be $\sim 58,000$ psi, which is below the yield point (see Table 2) and hence acceptable. The peak temperature in the cladding is $\sim 255^\circ\text{F}$ above the mean temperature of the spar, but this is partially compensated by the lower coefficient of thermal expansion. The net effect is to give a thermal stress in the cladding of $\sim 30,000$ psi, well within the elastic limit. However, these rough estimates of the thermal stresses were made for simple uniaxial conditions with no allowances for three-dimensional effects, the stresses from centrifugal

force and aerodynamic loads, and stress concentration effects associated with changes in section in the vicinity of the blade root. For example, stretching the central spar under the centrifugal load will act to alleviate the axial thermal stress in the cladding, but Poisson's ratio effects will act to increase the tangential stress in the cladding. The effects are exceedingly complex, but a satisfactory solution to the stress problem appears possible in view of the very clever and sophisticated design analyses carried out at G.E.²²⁻²⁷

In reviewing the above, a number of key requirements for acceptable thermal stresses in the critical element, the first-stage rotor, have become evident as follows [these are pointed out in the G.E. reports (e.g., on p. 61 of Ref. 23)].

1. A layer of high-conductivity copper must be used beneath a cladding layer that is less than 1 mm (0.040 in.) thick if skin temperatures below 542°C (1000°F) are to be maintained with open-circuit water cooling.
2. The cladding, the copper layer, and the central spar must all have unusually high yield points and exceptional resistance to thermal strain cycling.
3. The cladding material must have a lower coefficient of thermal expansion than the material of the central spar if thermal stresses and thermal strain cycling effects are to be kept low enough to give the design life goal of ~4000 starts.
4. The bores of the cooling passages in the copper must be lined with an iron-chrome-nickel alloy to avoid serious corrosion of the copper, and the inner surface of the coolant passage must be treated to assure that the liquid meniscus covers close to 50% of the surface during operation.
5. The base of the blade and blade platform must have cooling provisions similar to those for the airfoil section in spite of the geometric complexity of the region (see Fig. 14). The cost of fabrication must be kept to reasonable levels.
6. The cladding, copper, cooling passage liners, and central spar must be metallurgically bonded to give an essentially perfect thermal bond that will not develop interface cracks when subjected to severe

thermal strain cycling. This must be accomplished with the complex geometry of Fig. 14.

Heat losses

In the 1904 Lemale-Armengaud and 1950 NACA tests of water-cooled turbines, the heat lost from the hot gas to the cooled walls substantially reduced the power output. This type of loss would be reduced in a large turbine because of the more favorable surface-volume ratio, but it is not trivial. In the NACA tests, the heat loss per stage amounted to $\sim 5\%$ of the energy input in the fuel with a temperature difference of 950°C (1700°F) between the hot gas and the metal surfaces. An indication of the losses recently estimated by G.E. for a large turbine is given by Table 5.2-1, p. 5-5, of Ref. 22. The total heat loss to the cooling water for a three-stage turbine was estimated as 155 kJ/kg (66.8 Btu/lb) of gas, which is equivalent to 9.3% of the heat input from combustion. Almost one-third of this was estimated to occur in the first-stage stator, and over 20% would be in the first-stage rotor. The writer estimated that the total heat loss to the cooling water would reduce the gas turbine output by $\sim 6\%$ and the steam turbine output by $\sim 7\%$. This effect would be partly offset by the increase in mass flow through the gas turbine represented by the steam escaping from the rotor blade tips, which would increase the gas turbine output by $\sim 0.5\%$. The heat flowing to the cooling water for the stator and casing, coupled with the heat in the liquid water thrown off the rotor and collected by slots in the casing, could be used for feedwater heating; this would represent an increase in output by $\sim 2\%$. Thus the net overall effect would be a loss of a little over 5% relative to the ideal uncooled case for the same combustor outlet temperature. Note that this assumes no water cooling of the combustion chamber and includes no allowance for water churning losses associated with the liquid water thrown off the rotor or the pumping power requirement for the cooling water system. Note too, that if the liquid water thrown off the turbine rotor is not collected, its vaporization will act to quench the hot gas stream and further reduce its temperature.

It is interesting to compare these losses with those estimated for air-cooled turbines in Fig. 5. The ideal thermal efficiency indicated

by Fig. 5 for 1370°C (2500°F) is ~50% for zero cooling air flow, whereas the dashed curve that indicates the minimum cooling air flow probably required gives a thermal efficiency of ~45%, or a 10% drop in power plant output. In comparing this with the losses estimated above for a water-cooled turbine, it appears that water cooling will give half the losses of an air-cooled turbine.

It has been suggested that the heat losses might be reduced by applying a ceramic coating to the stator and rotor blades.²⁵ The temperature drop through this coating would reduce the temperature difference between the hot gas and the blade for a given peak metal temperature and thus reduce the heat loss. The problem will be to find a ceramic coating that will not crack and spall off under the severe thermal cycling conditions that it must withstand.⁶ It should be noted that recent tests of such coatings at United Technology Corporation have shown that to be effective they must be polished. In one test with a coating having a surface roughness of 600 $\mu\text{in.}$, the increased heat transfer coefficient essentially offset the thermal insulation effect, but polishing to a surface roughness of 125 $\mu\text{in.}$ did not completely avoid this effect.

Erosion

The mixture of water and steam leaving the blade will have a quality no higher than 70% if satisfactory cooling is to be assured.²²⁻²⁵ The droplets of water emerging from the tips of the blades will be thrown off at high velocity and will tend to impinge on either the turbine casing or the downstream stator blades; in either case erosion will be a major problem. Methods for coping with this at the low-temperature end of steam turbines have been developed, but they have entailed bleeding off substantial quantities of steam from the main stream flowing through the turbine stage and using it for feedwater heating; this would represent a major loss if it were done near the inlet of a gas turbine. The erosion problems in a water-cooled gas turbine will also be rendered more severe by the high-temperature environment.

Deposits in the water-cooling passages

Trace amounts of solids dissolved in the feedwater may lead to deposits in the small-diameter cooling passages in the blade and blockage

of the channels. By keeping the quality of the steam leaving the blade below 70%, it seems likely that this kind of difficulty can be avoided. However, the small cooling passages in the turbine blades will give a situation far more sensitive to trace amounts of deposits than in any steam boiler built to date. This will be particularly important in the case of the cooling passages in the stator and casing because these entail closed-circuit flow with subcooled boiling. Any nonuniformity in solids deposition from one passage to another can imbalance the flow distribution; this could lead to boiling in some channels, flow instabilities, and a rapidly progressing failure condition.

Reliability

In attempting to appraise the reliability of the water-cooled turbine system design that is emerging from the G.E. analyses and tests, five points appear to be particularly important:

1. The cooling passages are small in diameter (between 1 and 2.6 mm) and hence may be subject to plugging.
2. To keep heat losses, etc., to acceptable levels, the water flow rate through the rotor blades must be kept to a fairly low level so that the vapor exit quality will be $\sim 70\%$.
3. In the first stage, in particular, the heat flux is close to that for burnout.
4. The water flow rate to each of many thousands of passages must be metered and kept within close limits to avoid excessive heat losses on the one hand and burnout on the other.
5. If the cooling-water flow to a region of a blade is interrupted, the local surface temperature will rise at a rate of $\sim 280^\circ\text{C}/\text{sec}$ ($\sim 500^\circ\text{F}/\text{sec}$) in high flux regions and failure will occur in a few seconds.

Thus the crux of the reliability problem is the probability that the water flow rate to any one of thousands of small cooling passages will drop more than $\sim 30\%$ below the design value for even a few seconds. That probability must be kept exceedingly low.

CURRENT STATUS OF DEVELOPMENT

The most significant reference design prepared to date for a water-cooled gas turbine is from a G.E. study carried out for EPRI.²² This assumed an oil-fired combustor giving a gas turbine inlet temperature of 1580°C (2877°F) combined with an 85-bar, 510°C (1250 psi, 950°F) steam system as indicated in the flowsheet of Fig. 1. The principal design parameters are presented in Table 1. The overall thermal efficiency was estimated to be 47.5%.

The information presented in this report emphasizes that a water-cooled gas turbine of the sort proposed presents extremely formidable problems. As indicated in the previous section, these are being attacked in a comprehensive program that includes elegant analyses, extensive rig tests to investigate the crucial fluid flow and heat transfer problems, and the fabrication of specimens of the complex composite blades required. The latter work is currently directed toward fabricating elements of the blade, assembling them, and consolidating and bonding the blade assembly by a hot isostatic pressing operation.

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Appendix I

EMPIRICAL CORRELATION OF TEST DATA ON LOW-CYCLE FATIGUE

In a recent paper on the design problems of fusion reactors³⁷ Thompson evolved a convenient design chart for the first wall of the blanket where thermal strain cycling considerations are paramount. That approach has been modified to suit the problems of a water-cooled gas turbine. The point of departure is a recent paper by Coffin³⁶ which reviews the experimental evidence concerning the fatigue of materials at high temperature and correlates the data with the equations

$$\Delta \epsilon_p \nu^{\beta(k-1)} = C N_f^{-\beta} \quad (1)$$

and

$$\Delta \epsilon_e \nu^{-k'} = \frac{A}{E} N_f^{-\beta'} \quad (2)$$

where $\Delta \epsilon_p$ is the range of plastic strain (the sum of the negative and positive amounts by which the strain goes into the plastic region during a cycle) and $\Delta \epsilon_e$ is the range of the elastic strain (positive and negative). N_f is the number of cycles to failure under the conditions imposed and E is the modulus of elasticity. A , C , k , k' , β , and β' are experimentally determined quantities for each material. ν is the frequency with which the stress application is repeated.

The total range of the strain, $\Delta \epsilon$, is the sum of the plastic and elastic ranges or

$$\Delta \epsilon = \Delta \epsilon_p + \Delta \epsilon_e .$$

Upon substitution of the expressions from Eqs. (1) and (2),

$$\Delta \epsilon = C \frac{N_f^{-\beta}}{\nu^{\beta(k-1)}} + \frac{A}{E} \frac{N_f^{-\beta'}}{\nu^{-k'}} \quad (3)$$

When a thin layer of the surface of a material is heated by an amount $T_s - T_0$, the resulting thermal strain is found by the methods of the theory of elasticity to be

$$\Delta\epsilon = \frac{1 + \mu}{1 - \mu} \alpha (T_s - T_0) . \quad (4)$$

The thermal stress which accompanies this strain in an ideal elastic material is

$$\sigma = \frac{E\alpha(T_s - T_0)}{1 - \mu} . \quad (5)$$

The number of cycles of applied surface energy input required to cause failure as indicated by the above assumptions and the physical properties of Table 1 are shown in Fig. 18 in the main body of the report.

In order to apply these equations to the prediction of the capacity of a given material to withstand a pulse of energy into its surface, it is necessary to assign values, preferably from experiments, to the constants β , β' , C , A , k , and k' . For most materials of interest here, the values of these constants are not well known. For our present purposes the following assumptions are made:

1. $\beta = 0.5$ and $\beta' = 0$. For many materials that have been tested, values of β are in the region $0.45 < \beta < 0.6$ and β' is small so that for a first approximation it can be neglected.
2. $k = 1$ and $k' = 0$. Under most conditions of failure in thermal fatigue, the frequency effect appears to be small. The assumptions above drop the frequency effect altogether.
3. The constant A is taken equal to the yield strength σ_y .
4. The constant C is related to the ductility of the material, which of course depends upon the temperature and the rapidity of the deformation. Its value here is determined from Eq. (1), with the estimated true strain, $\Delta\epsilon_p = \Delta\epsilon_{1/4}$, corresponding to that of a conventional tensile pull test which is one-fourth that of a complete strain cycle ($N_f = 1/4$), so that $C = \Delta\epsilon_{1/4}/2$. This yields the relation

$$T_s - T_0 = \left[\frac{(1 - \mu) \Delta \epsilon_{1/4}}{2(1 + \mu) N_f^5 \alpha} + \frac{(1 - \mu) \sigma_y}{(1 + \mu) \alpha E} \right] . \quad (6)$$

The values used for the properties of materials in evaluating Eq. (6) when preparing Fig. 18 in the main body of the report are shown in Table 2.

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