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AIR COOLING OF DISK OF A SOLID INTEGRALLY CAST TURBINE ROTOR FOR AN AUTOMOTIVE GAS TURBINE

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16. Abstract

A thermal analysis is made of surface cooling of a solid, integrally cast turbine rotor disk for an automotive gas turbine engine. Air purge and impingement cooling schemes are considered and compared with an uncooled reference case. Substantial reductions in blade temperature are predicted with each of the cooling schemes studied. It is further shown that air cooling can result in a substantial gain in the stress-rupture life of the blade. Alternatively, increases in the turbine inlet temperature are possible.

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AIR COOLING OF DISK OF A SOLID INTEGRALLY CAST TURBINE ROTOR FOR AN AUTOMOTIVE GAS TURBINE

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SUMMARY

The surface cooling of a solid, integrally cast turbine rotor disk for an automotive gas turbine engine was thermally analyzed. Air purge and impingement cooling schemes were considered and are compared with an uncooled reference case. These cooling schemes consist of using compressor discharge air either to maintain a positive purge in the cavities between rotating and stationary parts or to impingement cool the turbine disk.

Either method results in a substantial reduction in blade temperatures at the blade critical spanwise location. This, in turn, results in a substantial increase in stress-rupture life or, alternatively, will permit the turbine to operate at increased gas temperatures. Temperature reductions at the blade critical location are predicted to be between 40 and 70 K. Redesigning the turbine to augment the surface cooling area results in a predicted temperature reduction of about 100 K at the blade critical location.

INTRODUCTION

The maximum inlet gas temperature for uncooled turbines is currently limited, by available materials, to about 1280 K. Air cooling, however, provides a means for either increasing the turbine inlet temperature or reducing the blade wall temperatures. Increasing the turbine inlet temperature can increase the engine power output. Lowering the wall temperatures can increase component life or permit the use of less costly materials.

The purpose of the study was to evaluate analytically the potential benefits that can be provided by air cooling the surface of a solid, integrally cast turbine rotor disk from an automotive gas turbine engine. The compressor drive turbine of a two-shaft gas turbine engine was chosen for the study. Radial temperature distributions in the solid disk and blades were calculated for an uncooled turbine and for two different cooling schemes. In the first cooling scheme, compressor discharge air was used to purge the cavities between the rotating and stationary parts. In the second scheme, the discharge air was used to impingement cool the disk. Temperature distributions calculated for the air-cooled case were compared with those of the uncooled case to indicate the reductions in blade and disk temperatures attainable. In addition, the temperature reductions obtained were used to predict an increase in blade stress-rupture life.

A computer model of the turbine was used to determine the steady-state, radial temperature distribution at maximum design rotor speed. The study was made for a turbine inlet temperature and pressure of 1280 K and 3 atmospheres, a turbine speed of about 44 000 rpm, and a compressor discharge temperature of about 500 K. The results are presented as a comparison of radial temperature distributions for an air-cooled case and an uncooled reference case.

ANALYSIS

The turbine rotor analyzed is shown in figure 1. The turbine is a solid, integral casting of blades, rim, and disk. Noted in the figure is a pocket region just inside the rim radius. The purpose of this region is to reduce the mass of the turbine and to reduce heat conducted to the disk. Other pertinent information concerning the turbine is as follows:

Tip diameter, cm	13.7
Hub diameter, cm	10.9
Chord, cm	1.1
Number of blades	53

Additional operating information is contained in reference 1.

The analytical procedure was first to predict the radial temperature distribution in the uncooled state. Then, temperature distributions were predicted for two different cooling schemes. The first scheme was an air purge of the cavities between the stationary and rotating parts. The second scheme consisted of impingement cooling of various parts of the disk face.

Only radial and axial conduction heat flow were considered in the model of the turbine. The convection heat transfer and fluid flow were solved separately from the conduction solution and were input as boundary conditions. Circumferential temperatures were assumed to be uniform and, therefore, this heat flux was negligible. A one-blade segment of the turbine, shown schematically in figure 2(a), was analyzed. In addition to these assumptions, the model considered only radial heat flow in the blade.

In general, a heat balance on an element of the turbine (fig. 2(b)) results in equation (1).

$$h_{21}A_{21}(T_2 - T_1) + k_{31}A_{31}\left(\frac{T_3 - T_1}{\Delta x_{31}}\right) + k_{41}A_{41}\left(\frac{T_4 - T_1}{\Delta x_{41}}\right) + k_{51}A_{51}\left(\frac{T_5 - T_1}{\Delta x_{51}}\right) = 0 \qquad (1)$$

where $k_{31} = k_{41} = k_{51}$. The boundary conditions required for the analysis are gas and coolant temperatures and the associated heat transfer coefficients (i. e., T_2 and h_{21} of eq. (1)). Average Nusselt number correlations on different segments of a turbine are discussed in reference 2. These correlations are used herein as a basis for determining the various heat transfer coefficients required. The analysis is discussed in four parts: the turbine blade, the blade platform, the disk, and the allowable blade metal temperatures.

Turbine Blade

The relative gas temperature to which the blade is exposed can be determined from the inlet total temperature to the stator and the inlet velocity diagram for the blade. The relative total gas temperature determined by equation (2) was assumed to apply to the entire blade surface (ref. 3):

$$T'_{g,\,rel} = T'_g - \left(\frac{V^2 - W^2}{2C_p}\right) \qquad (2)$$

The heat transfer coefficient was also assumed to be a constant over the entire blade surface. An approximate value of the heat transfer coefficient was obtained from figure 2 of reference 2. Gas property data were based on the relative total gas temperature.

Blade Platform

The gas temperature to which the blade platform is exposed was assumed to be the same as that of the blade. Although in reality there would be a thermal boundary-layer profile with a lower gas temperature at the wall, it was assumed that there is sufficient turbulent mixing in the gas stream to maintain a constant temperature across the duct.

The heat transfer coefficient on the blade platform was determined by equation (3) (ref. 2):

$$(\overline{Nu})_p = 0.021(Re)_p^{0.8}(Pr)^{0.6} \tag{3}$$

$$(Re)_p = \left(\frac{2LS_H}{L + S_H}\right)\frac{\overline{G}}{\mu} \tag{4}$$

Turbine Disk

Uncooled reference case. - The original turbine design did not incorporate forced-air cooling over the disk surface. Consequently, the rotation of the disk between stationary parts of the engine would generate a pumping action that would draw hot gases into the cavities from the hot gas stream. There would be higher ambient air temperatures within these cavities than would exist if the cavities were purged with compressor bleed air. The airflow rate for each cavity due to the rotational pumping action was determined by equation (5) (ref. 2):

$$\frac{Q}{\pi r_o \mu (Re)_o} = 0.0702(Re)_o^{-0.202} \tag{5}$$

This is also the amount of compressor bleed air required to prevent hot gas from entering each cavity. A nominal value of the air temperature in the cavity was selected such that it represents an average between the static gas temperature and the compressor discharge air temperature. This temperature was used to evaluate the cooling air properties.

The average heat transfer coefficients on the disk face were obtained from an equation in reference 2 that considers a disk rotating between stationary parts. The equation was developed for turbulent airflow over the disk face.

$$(\overline{Nu})_o = 0.0346(Re)_o^{0.8}K_v^{-0.1}\left(\frac{r_o}{r_c}\right)^{-0.3}\left(\frac{z}{r_o}\right)^{0.06}(Pr)^{0.6} \tag{6}$$

where

$$(Re)_o = \frac{\omega r_o^2 \rho}{\mu} \tag{7}$$

$$(Re)_z = \frac{w}{2\pi z\mu} \tag{8}$$

$$K_v = \frac{(Re)_o}{(Re)_z}\left(\frac{r_c}{r_o}\right)^2 \tag{9}$$

Forced convection cooling. - Two simple cooling schemes are considered. One consists of an air purge of the cavities between the rotating and stationary parts with compressor discharge air (case 1). This scheme would also restrict the influx of hot gas into the cavities and result in a boundary temperature similar to the compressor discharge air temperature. The heat transfer coefficients would be the same as those of the uncooled reference case. The other scheme consists of impingement cooling of the disk. This cooling scheme was considered for four different regions of the disk: (1) the upstream and downstream faces of the disk in a local area just below the pocket region (case 2), (2) the upstream and downstream edges of the rim (case 3), (3) the entire upstream face of the disk below the pocket region (case 4), and (4) the upstream and downstream faces of the disk below the pocket region (case 5). An average heat transfer coefficient for these impingement cooling cases was determined from figures 2 and 4 of reference 4. Here, again, the boundary temperature in the cavities was similar to the compressor discharge temperature. The cooling airflow rates were equated to the rate of flow due to pumping determined by equation (5).

Allowable Blade Metal Temperatures

In order to determine the allowable blade radial metal temperatures, the radial distribution of centrifugal stresses in the blade was first calculated by using equation (10):

$$S = \frac{\rho\omega^2}{2}\left(r_t^2 - r_x^2\right) \tag{10}$$

The blade stress distribution calculated by equation (10) and the stress-rupture data for a given life of the blade alloy can be used to determine an allowable blade radial metal temperature distribution. The point of tangency of the allowable blade temperature distribution and the actual temperature distribution is the critical spanwise location on the airfoil.

RESULTS AND DISCUSSION

The calculated, steady-state disk and blade temperature distributions are shown in figures 3 to 8. The calculated uncooled (reference case) temperature distribution is included in each figure for comparison purposes. The boundary conditions for each case analyzed are summarized in table I.

The disk and blade radial temperature distributions shown in figure 3(a) suggest that purging of the cavities between the stationary and rotating parts with compressor discharge air (at 500 K) reduces the blade temperature at the blade critical location by about 70 K. The critical location for the reference case is indicated by a small arrow. The quantity of coolant required for the purge is determined by equation (5) to be approximately 1.5 percent of the hot gas flow. This represents the amount of flow due to the pumping action of rotation.

The calculated blade temperature distributions for the reference case and the air purge case are shown in figure 3(b). This figure also shows the allowable blade temperatures for 3000 and 30 000 hours of blade life based on a turbine speed of 44 000 rpm and stress-rupture data for material 713C (ref. 5). The critical location for the reference case occurs at an x/L of about 0.2 with a life expectancy of 3000 hours. For the air purge case the critical location is moved outward to an x/L of about 0.5 and the life expectancy is increased to 30 000 hours. Alternatively, for the air purge case, if the 3000-hour stress-rupture life criterion is used for the maximum allowable blade temperature, the turbine inlet total temperature could be increased by about 50 K.

Figure 4 shows the result of impingement cooling on the upstream and downstream faces of the disk in a small area just below the pocket region (case 2). The temperature reduction near the blade critical location is about 60 K.

The results of impingement cooling on both the upstream and downstream faces of the rim (or hub platform) are shown in figure 5 (case 3). The temperature reduction near the blade critical location is predicted to be about 70 K. Doubling the coolant flow with this scheme had little effect on conduction cooling of the blade and only reduced the temperature at the critical location by about 6 K.

The results of impingement cooling the entire upstream face of the disk (case 4) are shown in figure 6. The temperature reduction at the blade critical location is predicted to be about 40 K.

Applying impingement cooling to the entire upstream and downstream faces of the disk (case 5) results in the temperature distribution shown in figure 7. The temperature reduction near the blade critical location is predicted to be about 50 K.

The mechanical details of incorporating these various cooling schemes into the turbine have not been considered. The air purge case would probably be the least costly because it only requires ducting from the compressor exit to the wheel hub region. This cooling scheme also appears to be as effective as the other schemes, which are more complex.

The cooling schemes discussed so far can be applied to an existing turbine. In order to remove heat more efficiently from the turbine, a redesign of the rim region was considered. This redesign consisted of increasing the heat transfer surface area and locating this surface at an optimum distance from the cooling air impingement nozzle. Figure 8 compares temperature distributions for a possible redesign and for the uncooled reference case. A sketch of the redesign is also shown in the figure. The blade temperature at the critical location is 100 K lower than that of the uncooled turbine.

SUMMARY OF RESULTS

Surface cooling of a solid, integrally cast turbine rotor disk was analyzed. Two cooling schemes were considered: air purge of the cavities between stationary and rotating parts, and impingement cooling of the disk face. The blade temperatures for the schemes were compared with those for an uncooled turbine, with the following results:

1. The disk temperature can be reduced significantly by each of the two cooling schemes investigated. The least complicated of the cooling schemes, air purge of the cavities between stationary and rotating parts, was as effective as impingement cooling of the disk.

2. The blade temperatures were reduced by 40 to 70 K at the critical location by conduction heat transfer to the air-cooled disk. This reduction in blade temperature can increase the blade life expectancy by an order of magnitude.

3. Redesigning the disk rim region by augmenting the surface area with fins may reduce the blade wall temperature at the critical location by 100 K from that of the uncooled turbine.

4. The turbine inlet gas temperature can be increased by 50 K by allowing the blade temperature at the critical location of the air-cooled turbine to remain the same as that of the uncooled turbine.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, October 19, 1976,
505-04.

APPENDIX - SYMBOLS

A	heat transfer area
C_p	specific heat at constant pressure
$\overline{G}$	mass velocity of coolant
h	heat transfer coefficient
K_v	parameter in eq. (6)
k	thermal conductivity
L	blade length
$\overline{Nu}$	average Nusselt number
Pr	Prandtl number
Q	volumetric flow rate
Re	Reynolds number
r	disk radius
S_H	blade pitch at hub
T	temperature
V	absolute velocity
W	relative velocity
w	mass flow rate
x	distance from blade platform
Δx	distance between nodes
z	disk-shroud clearance
μ	viscosity
ρ	density of air or metal
ω	angular velocity

Subscripts:

c	coolant entry radius
g	gas
o	platform radius
p	platform

rel	relative
t	blade tip radius
x	position along blade span
z	disk-shroud clearance
1, 2, 3, 4, 5	typical nodal points (see fig. 2(b))
21, 31, 41, 51	node face designation (see fig. 2(b))

Superscript:

'	total

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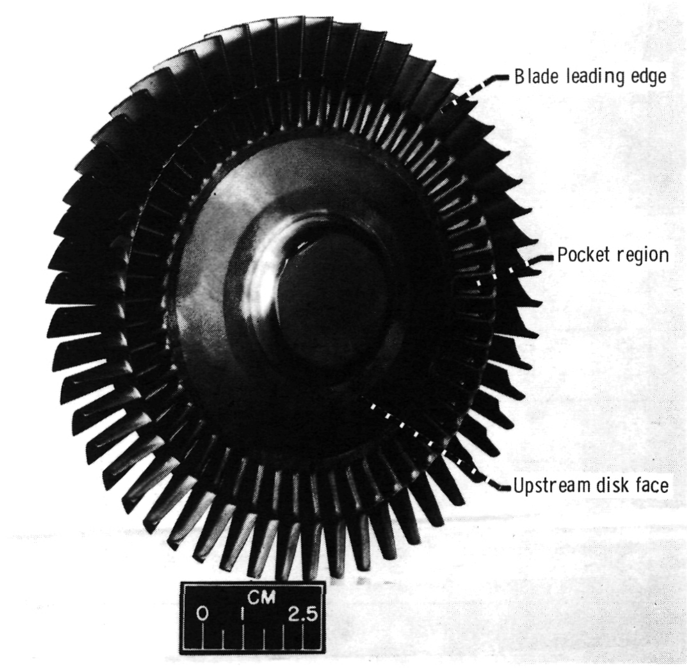
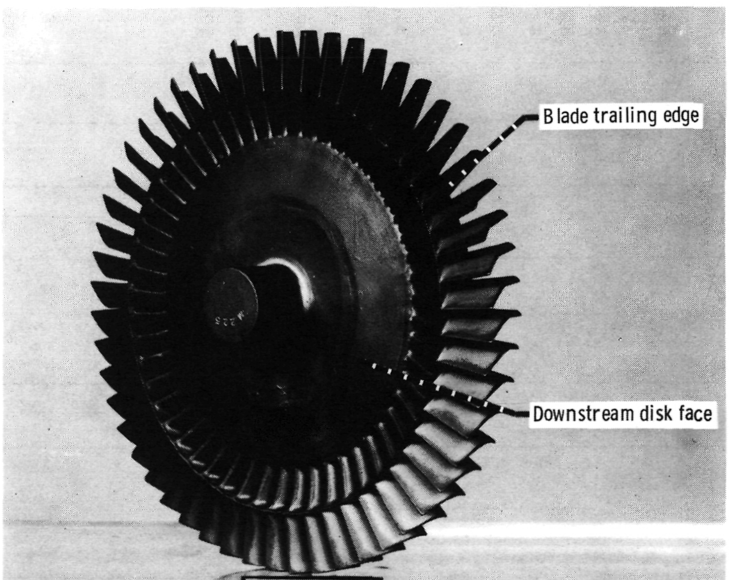
TABLE I. - BOUNDARY CONDITIONS

Boundary node	Uncooled reference case		Air purge		Impingement cooling								Redesigned disk case	
			Case											
			1		[a]2		[b]3		[c]4		5			
	T, K	h, $W/(m^2)(K)$	T, K	h, $W/(m^2)(K)$	T, K	h, $W/(m^2)(K)$	T, K	h, $W/(m^2)(K)$	T, K	h, $W/(m^2)(K)$	T, K	h, $W/(m^2)(K)$	T, K	h, $W/(m^2)(K)$
1	420	---	420	---	420	---	420	---	420	---	420	---	420	---
32	920	654	530	654	530	230	530	230	920	230	530	230	530	230
33	920		530		530		530		920		530	230	530	
34	930		545		530		530		920		530	230	530	
35	940		555		545		545		930		505	2500	545	
36	955		565		555		555		940				555	
37	970		580		565		565		955				565	
38	980		590		505	2500	580		970				580	
39	920		530		530	230	530		505			230	530	
40	930		545		530		530					230	530	
41	940		555		545		545			2500		2500	545	
42	955		565		555		555						555	
43	970		580		565		565						565	
44	980		590		505	2500	580						580	
45	1030	245		245	590	230	590		590	230	590	230	590	
46									590					
47									590					
48									980					
49														
50														
51	1090						505	2000					505	3300
52	1090						505	2000	590				505	3300
53	1030						590	230	590				590	230
54	1200	1060	955	1060	1090	1060	1090	1060	1090	1060	1090	1060	1090	1060
55	1200	2700	1200	2700	1200	2700	1200	2700	1200	2700	1200	2700	1200	2700

[a]Cooling of upstream and downstream faces of disk just below pocket region.
[b]Cooling of rim upstream and downstream edges.
[c]Cooling of entire upstream face of disk below pocket region.
[d]Cooling of upstream and downstream edges of disk below pocket region.

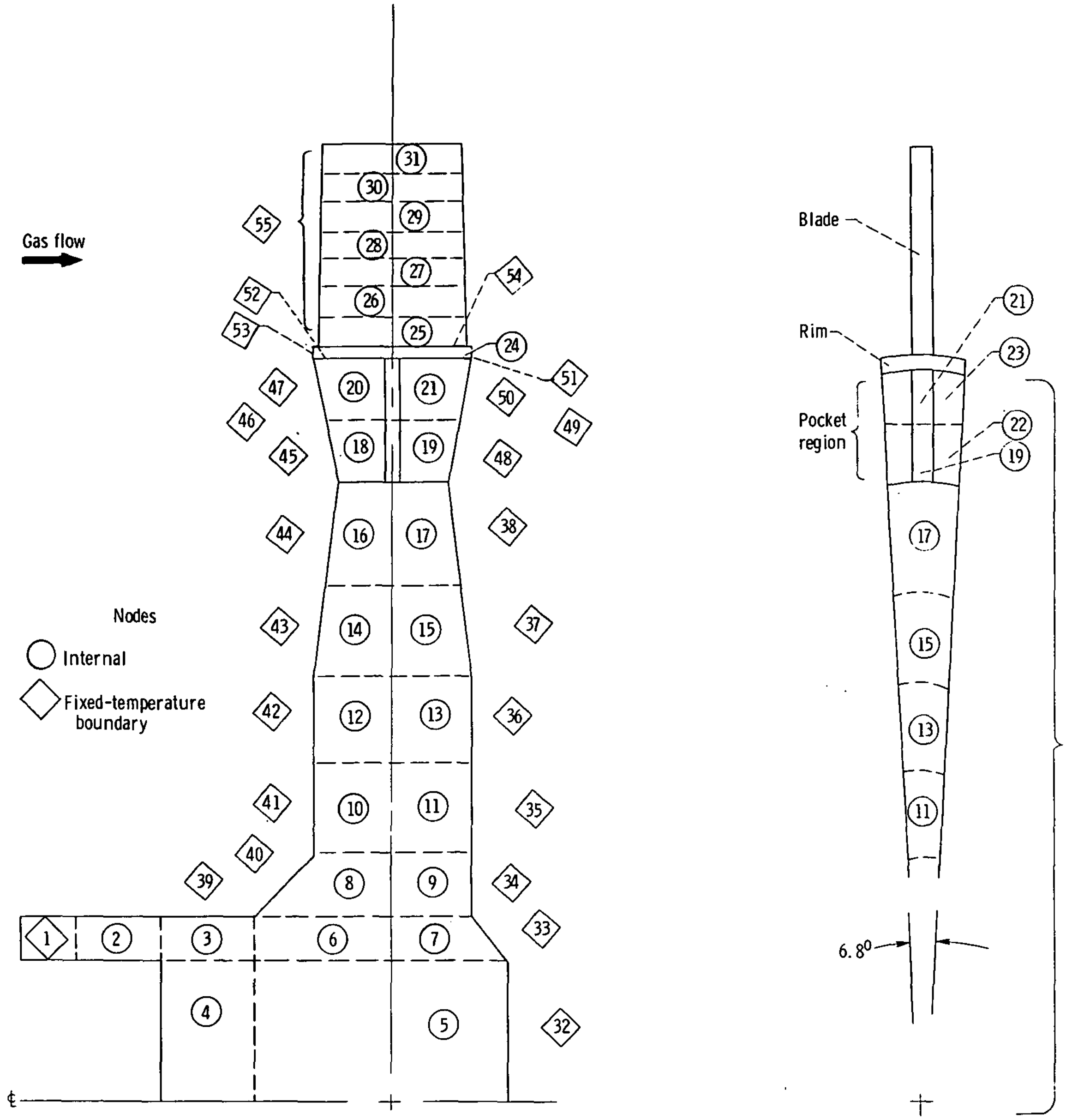


(a) View showing upstream face.

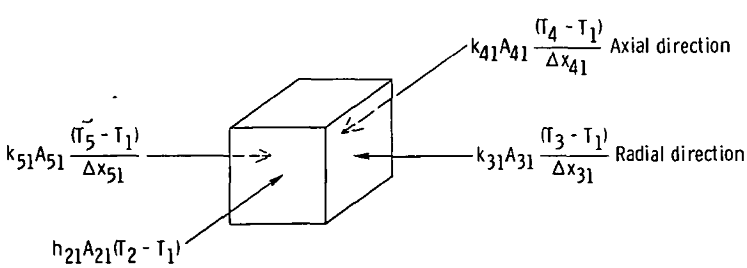


(b) View showing downstream face.

Figure 1. - First stage of automotive gas turbine rotor.

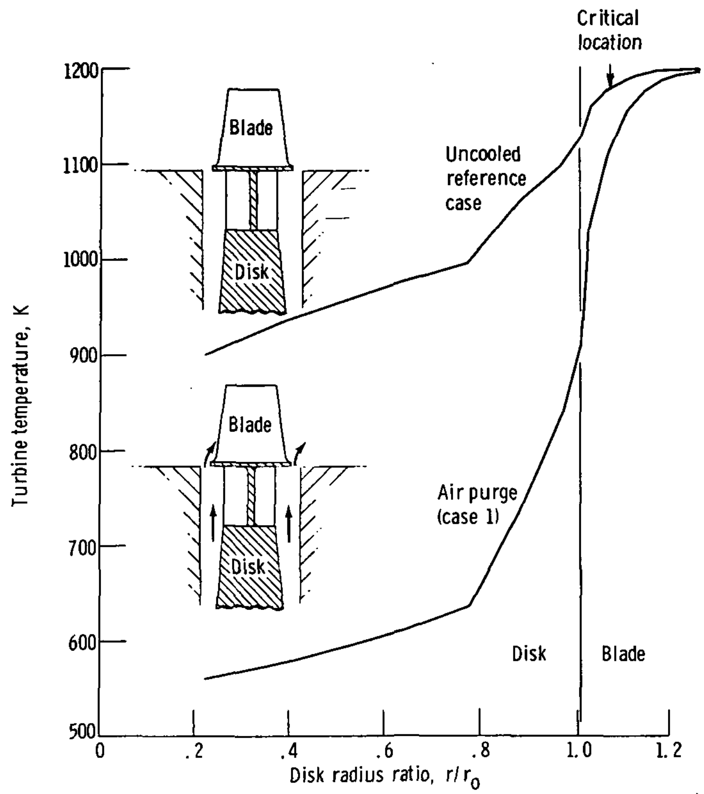


(a) Nodal representation.



(b) Typical element showing heat balance.

Figure 2. - Model of turbine rotor segment.



(a) Uncooled reference case and air purge case.

(b) Calculated blade temperature and allowable blade temperature distributions.

Figure 3. - Comparison of calculated temperature distributions.

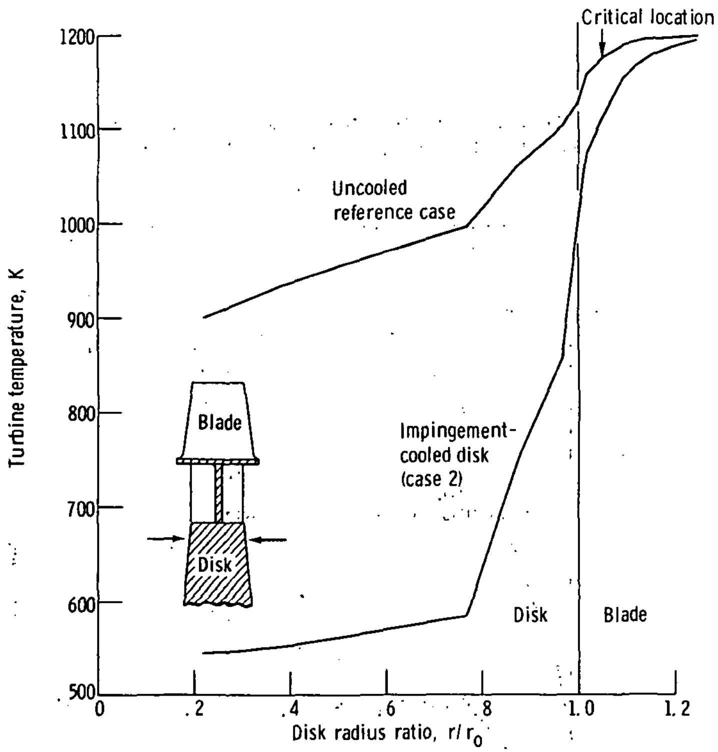


Figure 4. - Comparison of calculated temperature distributions for impingement-cooled disk (case 2) and uncooled reference case.

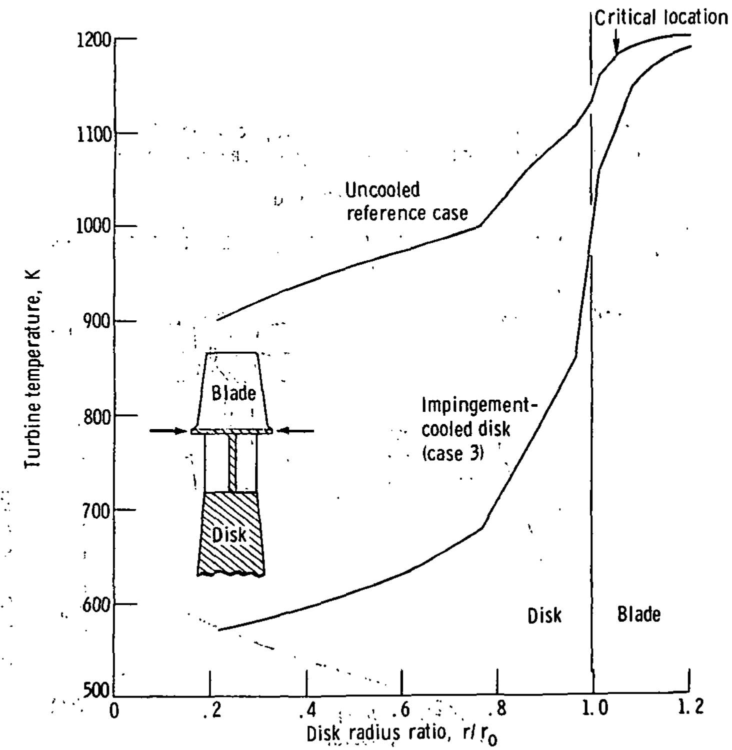


Figure 5. - Comparison of calculated temperature distributions for impingement-cooled disk (case 3) and uncooled reference case.

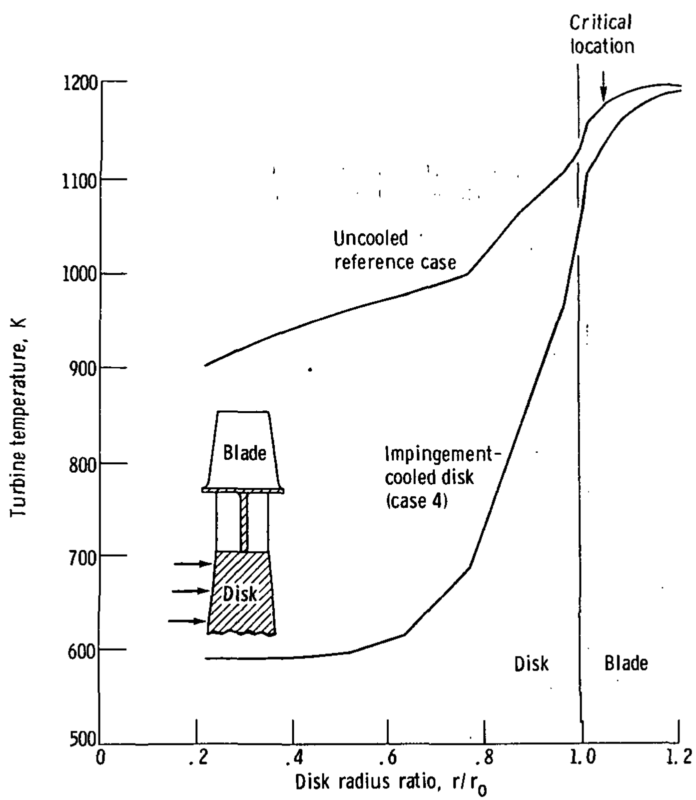


Figure 6. - Comparison of calculated temperature distributions for impingement-cooled disk (case 4) and uncooled reference case.

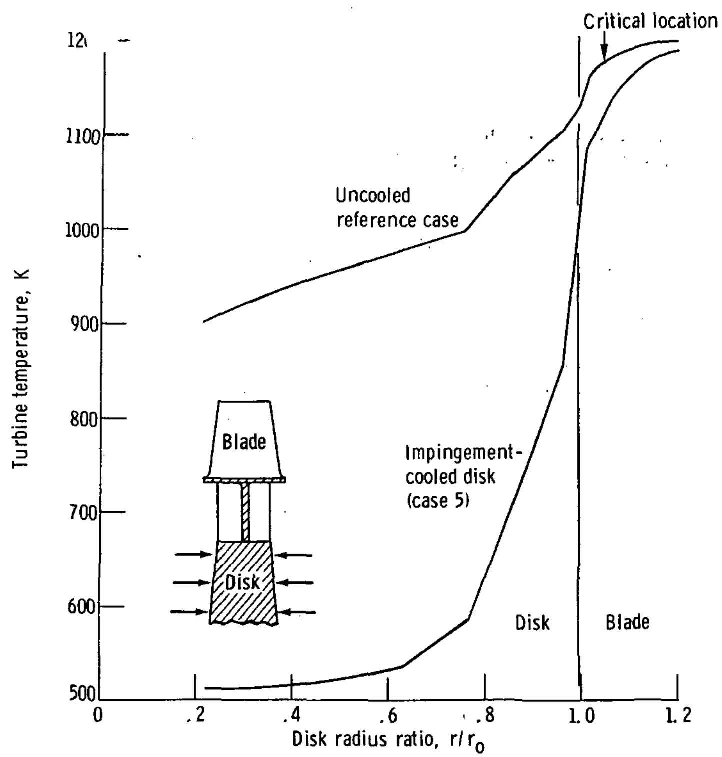


Figure 7. - Comparison of calculated temperature distributions for impingement-cooled disk (case 5) and uncooled reference case.

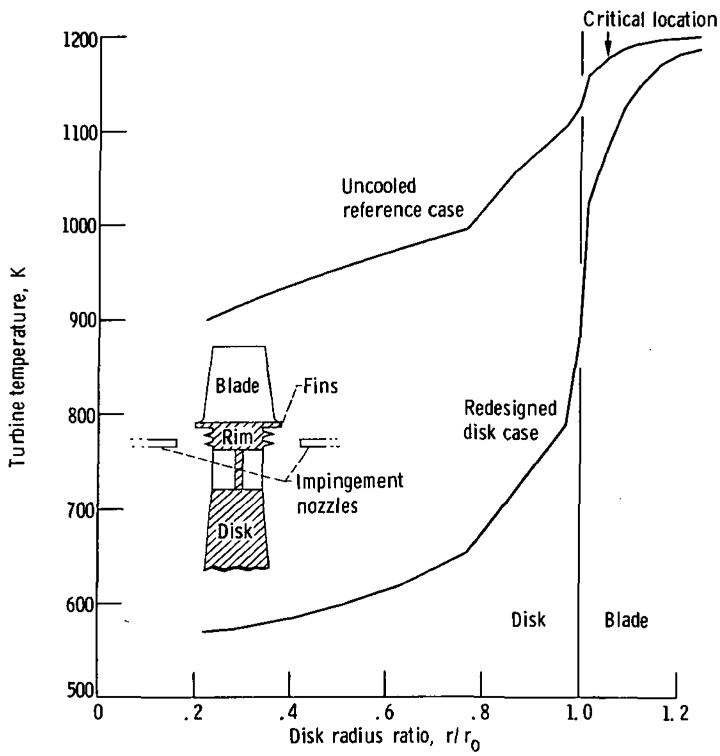


Figure 8. - Comparison of calculated temperature distributions for impingement-cooled, redesigned disk and uncooled reference case.

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