Stress Analysis Study in Cooled Radial Inflow Turbine

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Abstract

THIS paper presents a study of the stresses associated with different cooling patterns in a radial inflow turbine rotor. The finite-element method is used in the stress calculations, taking into consideration centrifugal, thermal, and aerodynamic loading. The projected blade life, as determined from the steady-state stress and temperature distributions, is compared for the different cooling arrangements.

Contents

The different cooling arrangements under consideration are shown schematically in Fig. 1. They include external rotor disk cooling by radially outward coolant flow on the rotor backside and two internal cooling arrangments of the rotor blades referred to as single path and double path. The results are also reported for a case involving a combination of internal and external cooling. The details of the calculated steady-state stress distribution for each separate loading are reported in Ref. 1.

The model of the radial inflow turbine rotor used in the three-dimensional thermal and stress analyses consists of a wedge section including one blade and the corresponding hub section extending between the two midchannels on both sides. A detailed description of the method of thermal analysis, as well as the temperature distribution in the rotor, can be found in Ref. 2. The same node placement used in the thermal analysis was also used in the stress analysis of the turbine rotor using NASTRAN. This involved 256 nodal points in the rotor section in the case of solid blade and 300 and 350 nodal points in the cases of internal cooling by single path and double path, respectively.

The octahedral shear stress, τ_o, is used for comparing the different cooling arrangements based on Von Mises theory, which is defined as

$$\tau_o = \tfrac{1}{3}\sqrt{(\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_3-\sigma_1)^2}$$

where σ_1, σ_2, σ_3 are the principal stresses.

The results of the analysis are presented for a twelve-bladed radial inflow turbine rotor. While the data which are relevant to the thermal and stress analysis can be found in Ref. 2, a brief summary is also given here. The mass flow rate in the rotor is 4.9 lb/s at 67,000 rpm. The stagnation inlet temperature and pressure are 2685°R and 257.5 psia, respectively. The rotor material is IN100 and the rotor tip diameter is equal to 3.948 in. The coolant inlet temperature was 850°F and the coolant mass flow rate was 3% of the 4.9 lb/s main mass flow rate in all the cases reported here.

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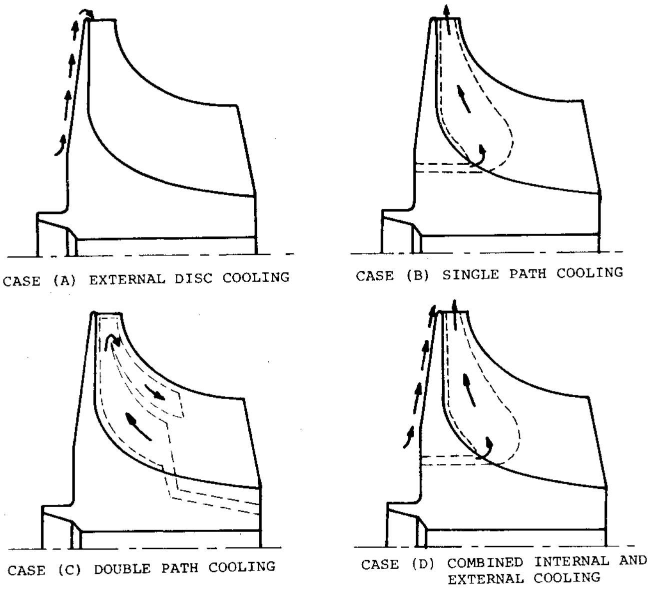

Fig. 1 Rotor blade cooling configurations.

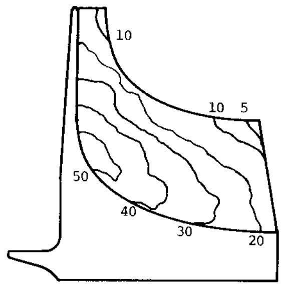

Fig. 2 Stress distribution for case A.

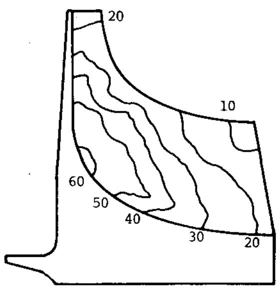

Fig. 3 Stress distribution for case B.

The overall octahedral shear stress distribution on the rotor blade suction surface are shown for all the cases of rotor cooling in Fig. 2-5. The values labeled on the various contours give the magnitudes of the stresses in kilopounds per square inch. The results of the analysis are summarized in Table 1 for all the cooling arrangements investigated, as well as for the uncooled case. Additional figures showing the stress distribution on the blade pressure side and the stresses due to the separate thermodynamic, centrifugal, and aerodynamic loadings were reported in Ref. 1. It was generally observed that, at the high rotor speed of 67,000 rpm, the centrifugal

Table 1 Comparative data for the different cooling arrangements

Case	Type of loading	Maximum stress in the blade, ksi	Blade temperature at the maximum stress, °F	Yield stress corresponding to the temperature at the position of maximum stress, ksi	Maximum temperature in the blade material °F	Minimum temperature in the blade material, °F
Solid uncooled blade	Thermal	3.93	1757.80	65.00		
	Centrifugal	51.67	1535.25	110.15	1779.66	1521.7
	Overall	52.02	1545.50	110.15		
External disk cooling, 3%	Thermal	13.95	1699.40	73.00		
	Centrifugal	51.67	1573.80	110.15	1748.91	1275.11
	Overall	53.19	1455.00	121.00		
Single path internal cooling, 3%	Thermal	17.19	1526.3	110.15		
	Centrifugal	53.98	1398.0	127.00	1697.30	1237.00
	Overall	61.04	1329.70	127.00		
Double path internal cooling, 3%	Thermal	32.65	1389.35	127.00		
	Centrifugal	53.15	1341.25	127.00	1689.98	1174.47
	Overall	65.02	1265.92	128.00		
Combined internal and external cooling, 1.5% + 1.5%	Thermal	10.22	1491.42	121.00		
	Centrifugal	53.30	1310.28	127.00	1693.44	1269.47
	Overall	58.26	1284.27	128.00		

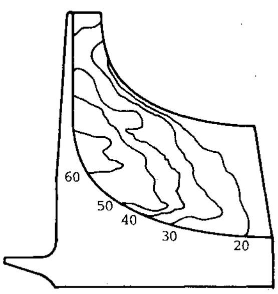

Fig. 4 Stress distribution for case C.

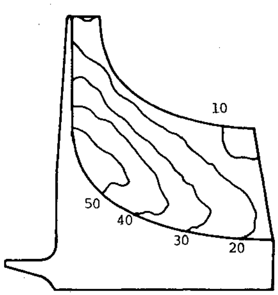

Fig. 5 Stress distribution for case D.

loading has the dominant effect on the blade stress distribution. The presence of the internal cooling passage did not result in a significant change in the blade stresses in general, nor in the magnitude and location of the highest blade stresses which are due to the centrifugal loading in particular. The main difference between the stress distributions for the different cooling cases was therefore mainly due to the difference in the resulting thermal stresses. The highest magnitude of thermal stresses in the cooled rotor blades was 32.7 ksi near the blade leading edge in case C, of the double path internally cooled blade. On the other hand, the lowest thermal stresses in any cooled rotor resulted in case D of the combined internal cooling of 1.5% in a single path plus a 1.5% external disk cooling by radially outward flow on the rotor backside. The thermal stress did not exceed 10.2 ksi in this combined cooling arrangement, in which the thermal stresses in the critically stressed blade region were also the lowest at about 3 ksi. These low thermal stresses for case D were also associated with the largest reduction in the blade temperatures.

A simplified study was carried out for the purpose of obtaining comparative projected blade life data in the cases under consideration. For this purpose, the steady-state stress and temperature distributions were used in conjunction with the blade material stress rupture data to determine a blade life potential. The stress rupture life potential (SRLP) thus computed under steady-state conditions will naturally be high and is therefore used merely for the purpose of comparison. The results of calculations suggest a 72.6 timefold improvement in the blade life potential with the combined cooling as compared to the uncooled case. On the other hand, the other three cooling cases, A, B, and C, resulted in 7.2, 8.2, and 42.0 timefold improvements in the blade life over the uncooled case, respectively.

It can be concluded from the data presented that from all cases considered, the combined external and internal cooling is superior and results in the most desirable blade temperature fields. While in the combined internal-external cooling case studied here the 3% coolant flow was equally split; further performance improvement might be realized with some other division ratio.

Acknowledgments

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[2] Hamed, A., Baskharone, E., and Tabakoff, W., "A Numerical Study of the Temperature Field in a Cooled Radial Turbine Rotor," NASA CR 137951, March 1977; also National Technical Information Service No. N77-32152.