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REMARKS ON COOLING THE TURBINE BLADES OF AIRCRAFT ENGINES, (U)
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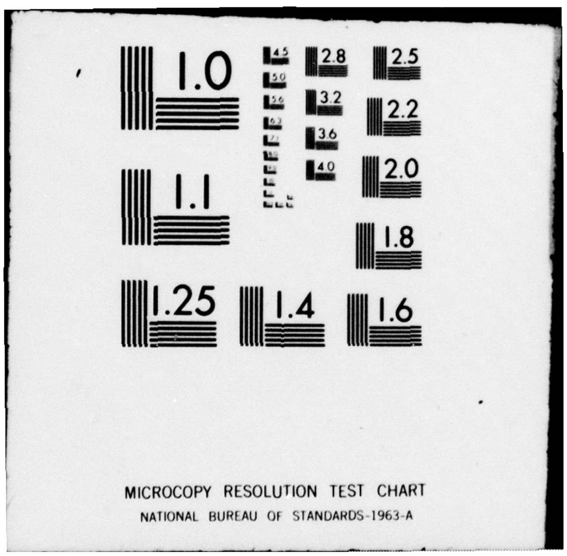
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REMARKS ON COOLING THE TURBINE BLADES OF AIRCRAFT ENGINES

by

J. Otys, S. Szczecinski

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FTD-ID(RS)T-1924-77

EDITED TRANSLATION

FTD-ID(RS)T-1924-77 1 February 1978

MICROFICHE NR: FTD-78-C-000148

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English pages: 11

Source: Technika Lotnicza i Astronautyczna, Vol. 27, Mar. 1972, pp. 5-8.

Country of origin: Poland
Translated by: LINGUISTIC SYSTEMS, INC.
F33657-76-D-0389
M. B. Biskupski
Requester: AEDC/XRFI
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FTD-ID(RS)T-1924-77 Date 1 Feb 1978

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Remarks on Cooling the Turbine Blades
of Aircraft Engines

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In this article are presented the structural and stress-endurance effects of cooling the guide and rotor blades of turbine aircraft engines, as well as a method for approximate calculation for blade cooling.

As is known, the increase in the temperature of the gasses before the turbine allows the decrease in the overall dimensions of the turbine engine, and, in the case of the turbine engine shaft -- also of unitary fuel use. The decrease in the geometric dimensions allows the simultaneous decrease in construction weight.

As a result, there exists the continual effort to increase the temperature of the gasses to the highest possible degree permissible with regard to the strength of the materials used in the several parts of turbine assembly. Many years have already passed since essential changes were noted in the permissable amounts of stress (in high temperature conditions) for the use of creep-resisting construction materials. Hence, also, a temperature reduction is often used before the turbine in engines used in civilian transport aviation (in comparison to the same engines used in military aircraft), which has the goal, first of all, of increasing the longevity of the turbine assembly -- as a result there is a decrease in the creep rate of materials used in the blades and bearing plates of the turbine rotors. In drawing 1 is presented the relationship between the strength of creep-resisting materials and the temperature as well as the relationship of the creep rate to the function time of the load at different temperatures.

From the cited data it can be seen that the only means to increase the longevity of a turbine assembly is to decrease the temperature of its parts. With simultaneously ever-increasing demands relative to engines' geometric dimensions, their weight, and other parameters, the necessity of turbine cooling arises (as has been noted in Aircraft and Astronautic Technology, 1971, No. 4).

The degree to which a stream of air can cool a turbine stator was tested long ago (e.g., the Jumo 004 engine), this degree of cooling of the rotor blades became possible only after the attainment of the relevant technological methods for turbine blade testing which allowed the blades' cooling by an air stream flowing along the length through appropriate canals in the blades. Direct cooling of the blades restricts the influx of heat Q_t to the bearing plates, which reduces the internal temperature stress, thus allowing the decrease in their weight.

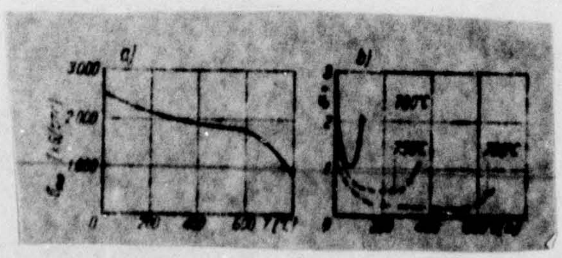
We present below a general method for the evaluation of temperature decrease of the walls of the turbine blades, at a given gas temperature and a given cooling intensity, as well as the construction strength resulting from the cooling method employed.

According to published information, the cooling of turbine blades (given the use of materials of comparable quality to those used in non-cooled turbines) allows the increase of pre-turbine temperature to 1100-1250°C, assuring thus a several thousand-hours endurance for turbine components.

Approximate Calculation of Blade Cooling

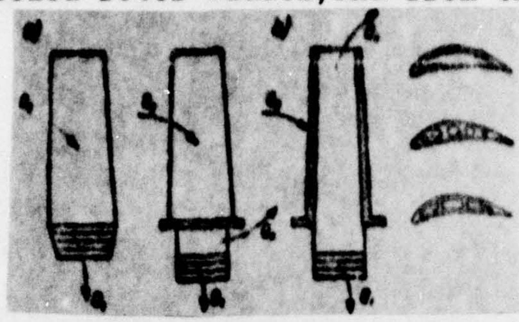
The presentation of calculations for the cooling of turbine blades has the purpose of determining the median wall temperature for both guide and rotor blades at a given temperature for the gasses flowing around it and at a determined expenditure of cooling air or, in other words -- the designation of the essential amount of cooling air needed to attain the desired wall temperature.

Simplifying this problem requires us to disregard the effect of heat radiation and abstraction, due to conduction, to the turbine cylinder (in the example of stator blade) -- or to the bearing plate Q_t (in the example of the rotor blades). Disregarding the effects of conduction is permissible, especially in regard to rotor blades, due to the way in which air cooling is supplied (to the interior of the blade) from the disc side in the direction of radiation.



1. Durability qualities of creep-resisting materials: a - durability in relation to temperature, b- rate of creep under constant stress at various temperatures.

In drawing 2 is given a schemata of the inflow and outflow of heat from un-cooled rotor blades, and from those cooled by air current.



2. Schemata of heat inflow and outflow in turbine blades: a - un-cooled blades, b - cooled blades.

After achieving this result, the heat balance has the form:

$$Q_2 = Q_p \qquad (1$$

where Q_2 = the flow of heat onto the blades from the gasses

Q_p = the flow of heat taken away from the blades by a flow of cooled air

The middle stage of conducting heat through the walls of the blades (permeation) can be disregarded due to the negligible heat resistence of the heavy ____________+ turbine walls.

In developed form the heat balance assumes the following form:

$$\int_{(A_g)} a_g (t_g - t_w)\, dA_g = \int_{(A_p)} a_p (t_w - t_p)\, dA_p \qquad (1a)$$

where +a_g, a_p = coefficients of heat absorption, relating gasses to the blade walls, and the blade walls to the cooling air

+A_g, A_p = blade surface bathed correspondingly by gasses and air

---

+Illegible

+ = stagnation temperature of the gasses before the blade lattice in ^{o}C.

+ = wall temperature in ^{o}C.

+ = stagnation temperature of cooling air.

At the same time, the heat flow Q_p^{+} can be expressed:

$$Q_p = G'_p c_p \Delta t^*_p \qquad (2)$$

where $G'_p = \frac{3600 \cdot G_p}{z}$ = expenditure of cooling air in one hour by one blade

G_p = expenditure of cooling air in one second

z = number of blades

$^{+}Q_p$ = median real heat of the cooling air

$\Delta t^*_p = t^*_{p2} - t^*_{p1}$ = temperature stagnation increase of air in the turbine

In solving the equation (1a) with equation (2) we gain the following result:

$$a_g A_g (t^*_{g1} - t_w) = G'_p c_p (t^*_{g1} - t^*_{p1})(1 - e^{-k_1}) \qquad (3)$$

where t^*_{g1} - stagnation temperature of the gasses at the entry section of the blade lattice -- along the blade vane

$e \approx 2.72$ - basic natural logarithm

$k_1 = \frac{a_g \cdot A_g}{G'_p \cdot c_p}$ - exponent

Designating $K = (G_p C_p)/(a_g A_g)$ allows us to gain from equation (3) the requirement for median wall temperature, <u>viz.</u>:

$$t_{w} = \frac{t^*_{g1} + k_1 t^*_{p1} (1 - e^{-k_1})}{1 + k_1 (1 - e^{-k_1})} \qquad (4)$$

Equation (3) permits the determination, by indicating the wall temperature, of the essential expenditure of cooling air for the blade, which is performed by the method of successive approximations. Then, for air cooled blades, we take as a first approximation the value of coefficient K_1 in the limits $K_1 = 0.2 - 0.4.^+$

The median values of the coefficients for heat absorption along the blade profile is expressed by equations from determined criteria which were developed on an experimental basis. For example, the value of coefficient $a_y{}^+$ is determined on the basis of the equation:

$$Nu = A\,Re^{[illegible]} \, [illegible] \qquad (5)$$

in which equation (Nuszelt's Number)

A = 0.1315 + 0.16 - the coefficient comprehending the effect of the shape of the blade profile. Lesser values relate to profiles with increased radius of attack edge

$^+Re = \frac{Cdg}{v} = \frac{C?dz}{?}$ - Reynold's number

$^+? = \frac{Tw}{T?}$ - The relation of median absolute blade wall temperatures and gasses

$^+t? = \frac{t}{b}$ - relative density of the lattice, equating the proportionate lattice scale (t) to its breadth (b).

After considering the preceeding expressions, equation (5) can be transformed to:

$$[illegible] \qquad (6)$$

where: $C = \frac{C_1 + C_2}{2}$ - Median gas speed in the lattice equating the median arithmetical speed before the lattice a+, and beyond the lattice C_2.

+ - median arithmetical values of the coefficients conveying heat, density, and the dynamic gas viscosity coefficient, which is determined in relationship to gas temperature in the entry and exit section of the lattice. In calculating the approximations for the values noted above the tables for the physical properties of air may be used.

$T_2 = \frac{T_1 + T_2}{2}$ - median gas temperature in the entry and exit sections of the lattice.

Tsc - median wall temperature which value is accepted for the first approximations, and in case of a difference with calculations from formula (4), succeeding approximation are performed.

+ = + - equivalent diameter, equating the relative blade profile perimeter bathed by gasses to x.

Formula (6) allows the value of coefficient (+) to be determined, for a stationary blade lattice, e.g., a stator. As the experiments of many authors have shown, the coefficient for conducting heat a'g for rotor glades increases on the average 20-30% in comparison with the same value for a stationary blade lattice:

$$a'_g = (1.2 + 1.3)ag$$

It is necessary to bear in mind here that in utilizing formula (6) for a lattice of rotor blades, absolute speeds C_1 and C_2 should be replaced by relative speeds w_1 and w_2.

If there exists the necessity to determine the value of the coefficient for heat conducting ap, then in this regard we may utilize the formula:

$$Nu = 0.0246\,Re^{0.79}\,\Theta^{-0.5}$$

where: $Nu = \frac{\alpha_p d_p}{\lambda}$ – Nusselt's number

$Re = \frac{c \rho d_p}{\eta}$ – Reynold's number

$\Theta = \frac{T_w}{T_p}$ – the relationship of the median absolute blade wall temperature to the median absolute tempa-rature of the air in the blades, which can be calculated on the basis of the air temperature increase in the blade

$$\Delta t_p^* = (t_{sc} - t_{p1}^*)(1 - e^{-k_1})$$

c = speed of the cooling air influx at the entrance to the blade interior

$^+$ = median arithmetical values for the coefficient of heat conduction, density, and the dynamic coefficient of air viscosity, determined by the temperature at the blade intake and exit.

$d_p = + \frac{4 \cdot A_{cw}}{U}$ – equivalent diameter, relatively equating the quadruple surface of the lateral section of the cooling canal in the blade to the perimeter of this canal bathed by air.

The utilization of internal cooling for turbine rotor blades in order to decrease the wall temperature to the range 200-300°C requires the most frequent expenditure of cooling air, which would account for 1.5 - 2.5% of all the air expended by the engine.

Cooling of the blades is connected with decreasing the overall temperature of the gasses flowing through the blade lattice. This decrease can be determined from this formula:

$$\Delta t_g^* = t_{g1}^* - t_{g2}^* = (t_{g1}^* - t_{w})(1 - e^{-k_3})$$

where:

$k_3 = \frac{\alpha_g A_g}{G_g' G_{pg}}$ - exponent

$G_g' = \frac{3600\,G_g}{z}$ - expenditure of gasses for one blade in the course of an hour

G_g - gas expenditure by the engine in one second

z - number of blades

C_{pg} - average characteristic heat of gasses

Practically, the value of (+) attains only a few degrees, and may be disregarded in approximate calculations of blade cooling.

Structural and Durability Effects on Blade Cooling

As has already been mentioned, prevailing technicalogical methods which enable the flow of cooling air by means of internal canals in turbine blades have allowed for considerable temperature diminution for the materials of these blades. Simultaneously the weight of the rotor blades is reduced decreasing the rim burden of the bearing plates, which permits the reduction of the plates' weight while retaining its strength.

The cross sections of the blade canals can also be selected so that the general character of the cross-sectional changes does not undergo change in relation to the "full" (not cooled) vanes. As is well known, the tensile stress on a turbine blade, on an arbitrary radius, is defined by the formula:

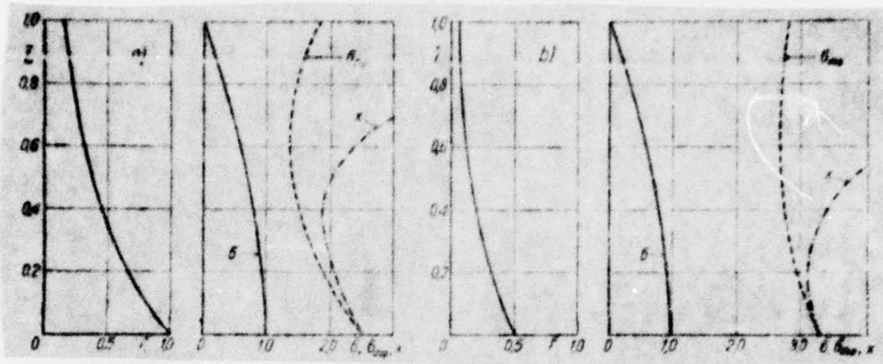
$$\sigma(R) = \gamma \omega^2 \frac{1}{F(R)} \int_R^{R_2} f(r)\, r\, dr$$

where: r - weight density of the material

w - angular velocity of the rotor

F(R), f(r) - consecutive blade lateral section fields

R, r, R_2 - consecutive and external blade radius



Drawing 3. Characteristic patterns for lateral sections F, stresses (+) and safety factors along the turbine rotor blade: a = uncooled blades, b = cooled blades.

In drawing 3 are presented the characteristic course of changes, in lateral section, in turbine rotor blades, the distribution of stress along the blade and the course of permissible stress for material for both un-cooled and cooled blades -- under conditions of the same gas temperature. Retaining the constancy of the relationship f(r)n + P(r)cht+ along the radius, the stress on the blade remains without change -- which, in the case of cooling, increases the amount of permissible stress, simultaneously increasing the value of the safety factor for the blade. The safety factor at each blade section assumes the form:

$$x(R) = \frac{\sigma_{dop}[T(R)]}{\sigma(R)}$$

The problem may be stated in the reverse -- the evaluation of how much the gasses's temperature can be increased, by using rotor blade cooling, and still retain the safety factor at this same level. While retaining the same value for the safety factor, the gasses's temperature can be raised 200-300°C (in the case of cool blades) depending upon the intensity of cooling, the creep speed of the material used and the predicted durability of the turbine.

Cooling of turbine stator blades has been employed for many years quite broadly, however cooling of rotor blades has come into use relatively recently -- and then primarily in jet engines, i.e., in engines with blades of relatively large geometric dimensions. Due to blade cooling, both stator and rotor, of two first stage turbines, the possibility has been attained of enduring considerable temperatures (before the turbine assembly), for example in the Rolls-Royce RB211 engine -- 1145°C, in the General Electric TF39 engine -- 1258°C; perhaps these will not comprise the limit of technical possibilities.

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