DESIGN OF GAS-TURBINE BLADES FOR AUTOMOBILE APPLICATIONS

by

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Summary

The combustion chamber of an automobile gas-turbine engine can be designed to produce a gas temperature distribution at the inlet of the turbine increasing from blade root to blade tip. It is shown in the paper, by means of comparative calculations, that by using such distributions of temperatures blade life can be sustantially increased, or else, unexpensive materials can be used.

Such gas temperature distributions produce non-isentropic flow conditions. It is developed in the paper a method for the aerodynamic design of blades within a non-isentropic flow and it is also shown that if the blades are designed by taking an average gas temperature, as it is usually made, important errors are introduced in the resulting shape of the blade, which reduces the efficiency of the turbine.

1.—INTRODUCTION.

The turbine is one of the most important components of a gas-turbine engine regarding to life and cost. Blades and disc bear large stresses at high temperature, to such extent, that superalloys or high-alloy steels must be used. Therefore, life and cost of such turbine are always critical.

In automobile applications both power/weight ratio of the engine and fuel consumption are very important parameters, which implies that high working gas temperatures have to be selected. This makes more difficult all design problems of the turbine, especially considering that in the high competitive market of automobile vehicles, life and cost of the engines are extremely important.

As a result, a careful design of the turbine in order to increase its life, or else, in order to make possible the use of unexpensive materials is of fundamental importance.

Blade temperature at the root, where stresses usually reach their maximum value, can be reduced by imparting to the gas flow a radial temperature distribution increasing from blade root to blade tip. This can be achieved by means of a proper design of the gas-turbine combustion chamber.

Such forms of radial temperature distributions have been utilized in turbojet engines and in industrial gas turbines. In this case the flow is not isentropic, which introduces an essential difficulty in the aerodynamic design of the blade. Such difficulty has been customarily avoided by designing the blades taking an average value for the gas temperature. However, it will be shown that the blades cannot be properly designed disregarding the actual radial distribution of the gas temperature, because in such a way very important errors are introduced in the gas velocities and, then, in the blade shape.

Furthermore, there is little or no information available of the effect of such radial gas temperatures distributions on the laws of variation of temperatures within the blades, and therefore, on the influence of them on life and materials of blades and turbine discs.

A method for the aerodynamic design of blades within a non-isentropic flow is developed in the present paper, as well as a method to calculate blade temperature for any gas temperature distribution. These calculations together with stresses considerations give the optimum shape and the optimum radial law of variation of blade thickness.

By comparing a blade designed to suit a radial gas temperature distribution increasing from blade root to blade tip, to another blade designed for constant gas temperature (equal average gas temperature for both cases), it will be shown that blade life can be subtantially increased if the same materials are used, or else, that a considerably cheaper material can be selected if lifes are taken equal in both cases.

2.—RADIAL VARIATION OF MAGNITUDES IN A NON-ISENTROPIC FLOW WITH AXIAL SYMMETRY.

a) Fundamental assumptions

The assumptions on which the model of the process will be based are those usually admitted in the aerodynamic of turbomachinery, but considering the non-isentropic character of the fluid. They are as follows:

1°) Ideal fluid, except friction losses which could be considered by taking a politropic exponent k instead of the isentropic exponent γ.
2°) Axial symmetry and stationary conditions.*
3°) Radial or quasi-radial blades.
4°) Non isentropic flow. The first assumption implies that the flow is isentropic along each stream line, but the entropy constant is different for every stream line.
5°) Radial deviations of the stream lines will be disregarded.

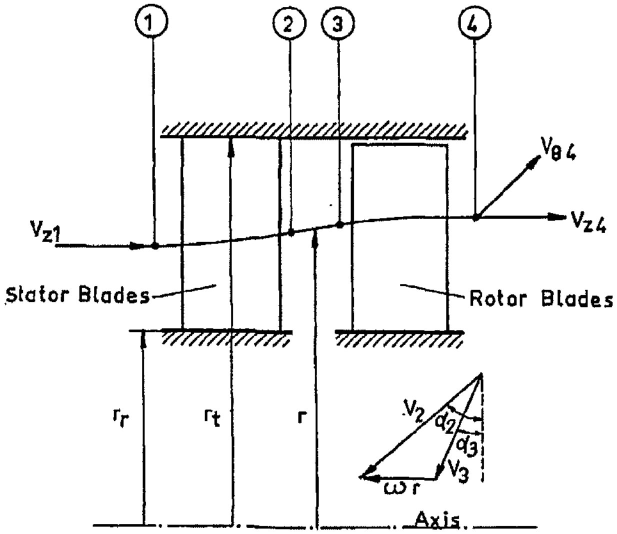

Fig. 1. Notation

*—These two conditions imply that the formal hypothesis of assuming an infinite number of blades infinitely thin is admitted.

Under the aforementioned assumptions the general equations of the process are as follows (Fig. 1):

Momentum (radial direction)

$$\frac{1}{\rho_i}\frac{\partial p_i}{\partial r} = \frac{V^2_{\theta i}}{r} \tag{1}$$

Energy

$$\frac{\gamma}{\gamma - 1}\frac{p_i}{\rho_i} + \frac{1}{2}(V^2_{zi} + V^2_{\theta i} - 2\omega r V_{\theta i}) = = \frac{\gamma}{\gamma - 1}\frac{p_{i+1}}{\rho_{i+1}} + \frac{1}{2}(V^2_{zi+1} + V^2_{\theta i+1} - 2\omega r V_{\theta i+1}) \tag{2}$$

in which

$$\tau = \omega r(V_{\theta i} - V_{\theta i+1}) \tag{3}$$

is the adiabatic head or specific power of the turbine.

Continuity

$$\rho_i \cdot V_{zi} = \rho_{i+1} V_{zi+1} \tag{4}$$

and:

$$\dot{m} = \int_{r_r}^{r_t} \rho_i V_{zi} r dr \tag{5}$$

Isentropic motion along a stream line

$$\frac{p_i}{\rho_i^{\gamma}} = \frac{p_{i+1}}{\rho^{\gamma}_{i+1}} \tag{6}$$

Assuming that $\tau = 0$ (stator) or $\tau =$ constant (rotor), from the above equations it is obtained:

$$\frac{1}{\gamma - 1}\frac{p_i}{\rho_i}\frac{\partial}{\partial r}\ln\left(\frac{p_i}{\rho^{\gamma}_i}\right) + \frac{V^2_{\theta i}}{r} + V_{zi}\frac{\partial V_{zi}}{\partial r} + V_{\theta i}\frac{\partial V_{\theta i}}{\partial r} = c_p \frac{\partial T_i - 1}{\partial r} \tag{7}$$

For the isentropic case, this equation reduces to the well known equation:

$$\frac{V^2_{\theta i}}{r} + V_{zi}\frac{\partial V_{zi}}{\partial r} + V_{\theta i}\frac{\partial V_{\theta i}}{\partial r} = 0 \tag{8}$$

Equation (7) gives the radial variation of the velocities, through equations (2) and (6), once the torsional law of variation $V_{zi} = f(V_{\theta i})$ or $V_{\theta i} = f(r)$ is given.

Usually, equation (7) is applied to the stator, and the rotor is designed from the resulting conditions at station 3 and from the value of τ.

For the usual case of taking a constant efflux angle α_2 at the stator, that is:

$$\tan \alpha_2 = \frac{V_{\theta 2}}{V_{z2}} = \text{constant}, \tag{9}$$

equation (7) reduces to:

$$\frac{dV^2_{z2}}{V^2_{z2}} = \frac{dT_{1i}}{T_{1i}} - 2\sin^2\alpha_2 \frac{dr}{r} \tag{10}$$

which can readily be integrated.

3.—NUMERICAL APPLICATION

A practical application has been performed for the compressor turbine of an automobile gas turbine of 150 HP approximately. Blade temperatures are more important in that turbine than in the power turbine, in which gas temperatures are lower.

A preliminary design of such turbine has been made, from which the following pertinent data are taken:

Inlet pressure: p_{1t} = 3,5 kg/cm² (constant)

Turbine speed: $\begin{cases} n & = 36{,}400 \text{ r. p. m.} \\ \omega & = 3{,}800 \text{ rad/sec.} \end{cases}$

Stator politropic efficiency: $\eta_k = 0.93$

Mass flow: $\dot{m}$ = 3.2 kg/sec.

Specific power: τ = 17800 m.

Dimensions: $\begin{cases} r_r & = 8.5 \text{ cm.} \\ \bar{r} & = 10 \text{ cm.} \\ r_t & = 11.5 \text{ cm.} \end{cases}$

Stator efflux angle: α_2 = 65°

(choking conditions at mean radius $\bar{r}$)

For the inlet temperature three cases have been considered:

1°. A constant temperature T_{1t} = 1070°K (isentropic case).

2°. A temperature distribution increasing from blade root until a point near the blade tip, where temperature reaches a maximum, decreasing from it till the blade tip.

This is the type of temperature distributions which occurs in practice, although in the ideal case, gas temperature should increase up to the blade tip.

Maximum and minimum temperatures have been taken equal to ±20% of the average temperature, which is also 1070°K.

3°. A temperature distribution equal to that of 2° case, but with maximum deviations equal to ± 10% of the average values.

Fig. 2 shows temperature profiles at the inlet and outlet stations 1 and 2 of the stator.

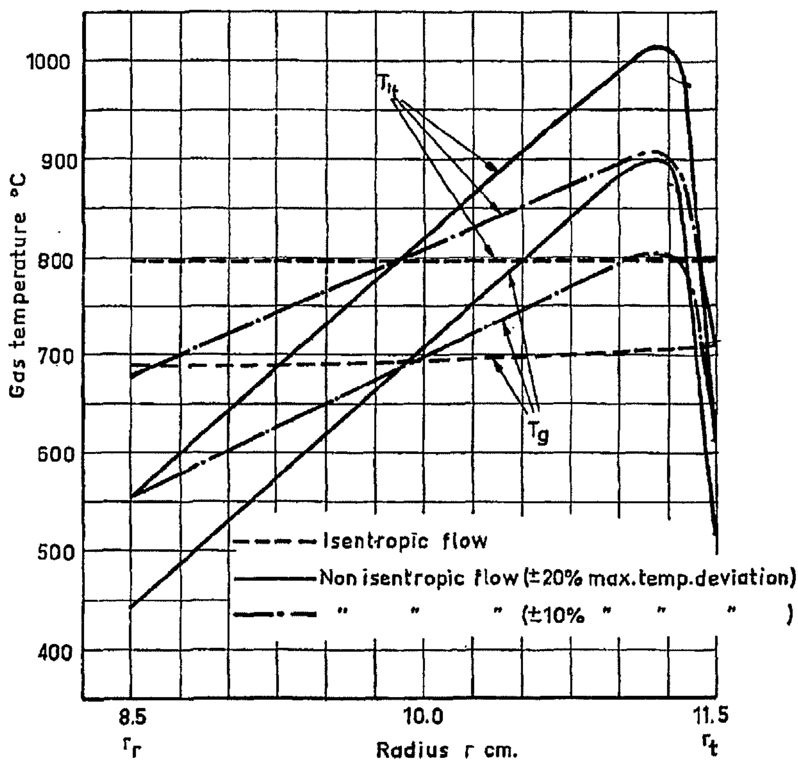

Fig. 2. Gas temperature profiles

4. AERODYNAMIC RESULTS

The selected temperature profiles $T_{1t}(r)$ at the stator inlet are shown in Fig. 2. Temperature T_g of the gas in the boundary layer surrounding the rotor blades is also shown. Temperature T_g has been calculated by adding to the static temperature T_s ninety per cent of the dynamic temperature resulting from the relative velocity V_3. This temperature T_g will be used to calculate the temperature within the blades.

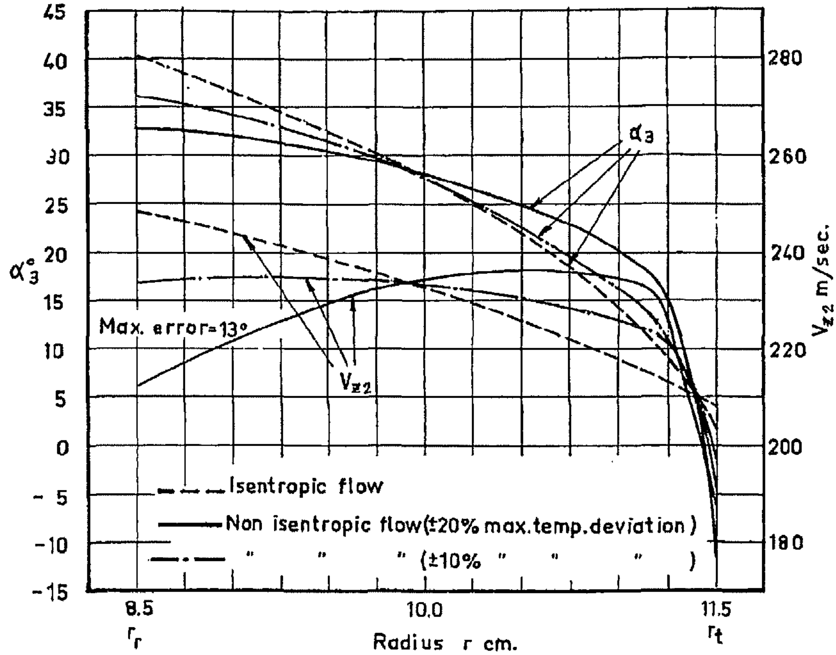

Fig. 3. Velocities and angles profiles

Fig. 3 shows the axial velocity profiles and the resulting gas angles α_3 at the inlet of the rotor blades.

It can be seen that velocities and angles differ considerably from those calculated by taking a constant temperature. This shows that with gas temperature distributions such as those selected a blade cannot be properly designed by taking an average value of the temperature. In this case the blade shape would be incorrect, resulting high positive or negative values of the angle of incidence $(\alpha_3 - \beta_3)$, with differences up to 10–15°. These incorrect angles of incidence can exert an important effect on the efficiency of the turbine.

5. BLADE TEMPERATURES

The geometric characteristics of the blade are shown in Fig. 4. Blade thickness has been selected from stresses considerations.

Once the size and shape of the blade have been determined and the gas temperature T_g is known, blade temperature is calculated by assuming one-dimensional and stationary heat flux conditions within the blade, by means of the equation:

$$(T_g - T_b)\Omega\alpha = -\lambda_b \frac{d}{dr}\left(\sigma \frac{dT_b}{dr}\right) = = -\lambda_b \sigma \frac{d^2T_b}{dr^2} - \lambda_b \frac{dT_b}{dr}\frac{d\sigma}{dr}, \qquad (11)$$

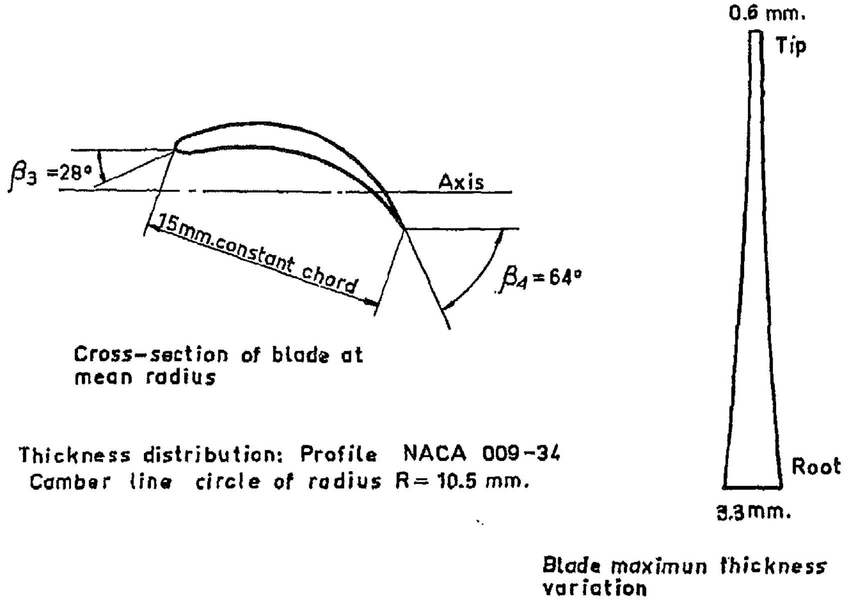

Fig. 4. Geometric characteristics of blade

which expresses the heat balance in a blade element of length dr, area σ and perimeter Ω. In that formula λ_b is the blade thermal conductivity, α the heat transfer coefficient and T_b is the blade temperature.

One of the boundary conditions is established at the blade tip, where a finite area σ_t exists:

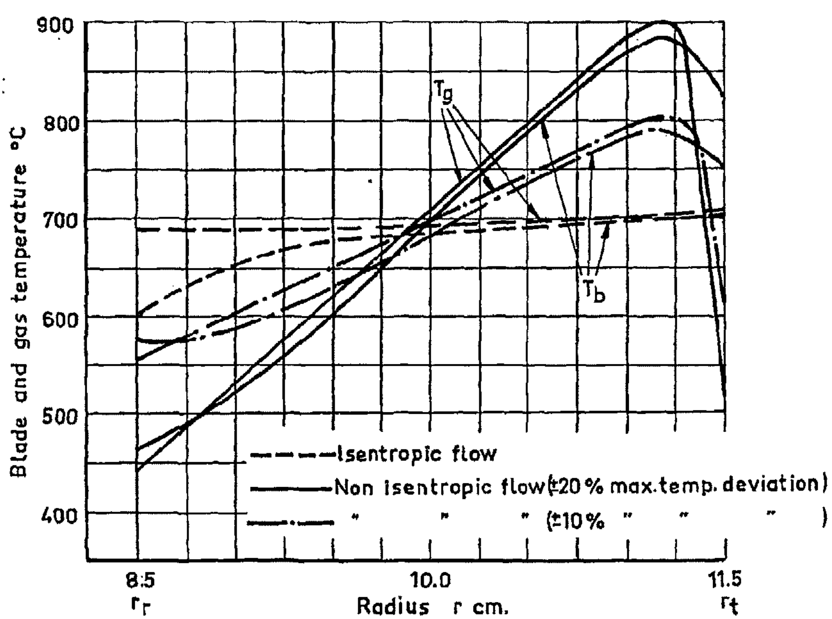

Fig. 5. Gas and blades temperature profiles

$$(T_g - T_b)_t \sigma_t \alpha = \lambda_b \sigma_t \left(\frac{dT_b}{dr}\right)_t \tag{12}$$

In order to compare results between the isentropic and non-isentropic flow cases, the heat taken away from the turbine disc has to be considered. The main cooling mechanism of a ventilated or air-cooled turbine disc is forced convection. Therefore, the boundary condition at the root of the blade can approximately be expressed as follows:

$$\lambda_b \left(\frac{dT_b}{dr}\right)_r = B(T_{br} - T_a) \tag{13}$$

in which T_{br} is the blade temperature at the root, equal to the disc temperature at the rim; T_a is the temperature of the air surrounding the disc and B is a heat transfer constant which is determined by taking a value for T_{br} equal to 600°C for the isentropic case.

Equation (11) with boundary conditions (12) and (13) has been integrated by means of a relaxation method, taking the following values for the parameters:

α = 0,058 cal/cm² sec.

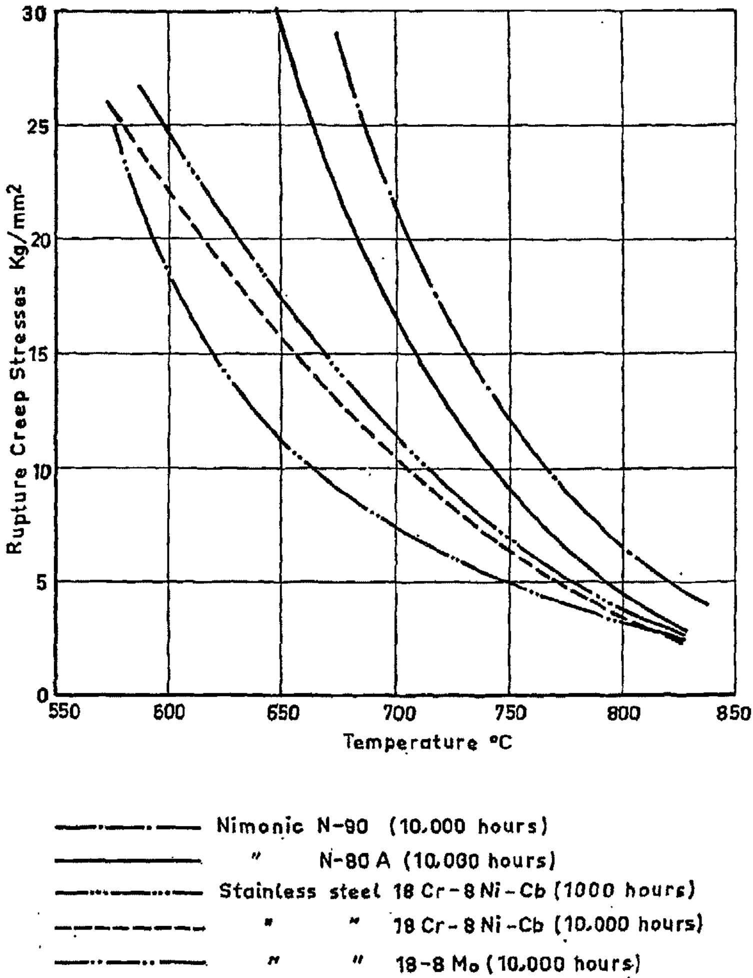

Fig. 6. Creep stresses for nimonic and stainless steel

λ_b = 0,05 cal/cm °C sec.
T_a = 150 °C

Results are shown in Fig. 5, where it can be seen that blade temperature is considerably smaller in the lower part of the blade for the two non-isentropic cases.

6. BLADE LIFE AND BLADE MATERIALS

Two kinds of possible materials have been considered for the blades: superalloys Nimonic N–80A and N–90 and stainless-steels 18 Cr–8Ni Cb and 18–8 Mo. Rupture stresses of these materials are shown in Fig. 6 as functions of the temperature*. A normal life of 10,000 hours has been considered.

From the curves of stresses and temperatures within the blades and from the rupture data of Fig. 6, it is possible to determine the maximum (rupture) stresses in each section of the blade. If for a certain material such curves are located under the curves of the actual stresses in the blade, such material cannot be utilized, and conversely.

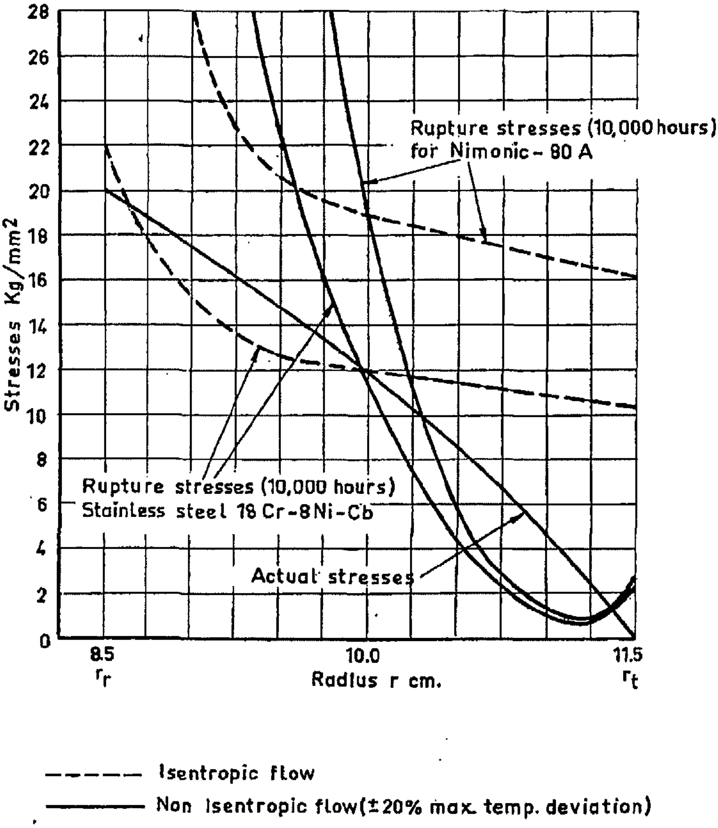

Fig. 7. Blade actual stresses and maximum working stresses

Fig. 7 shows the results obtained for the isentropic case (T_{1t} = constant) and those corresponding to a gas temperature with ±20% maximum deviations from the mean value.

It can be seen that stainless steel cannot be utilized in the case of isentropic flow; only

*—In gas-turbine blades in which temperature changes considerably along the blade, rupture and not strain is the variable which, mainly, determines the life of the blade.

superalloys would give a life to the blades longer than 10,000 hours.

For the non-isentropic case, stainless steel would be sufficient at the root of the blade, but at the tip the blade is too hot, and even Nimonic 80A could not be used. This shows, that temperatures deviations of $\pm 20\%$ are too high in this case.

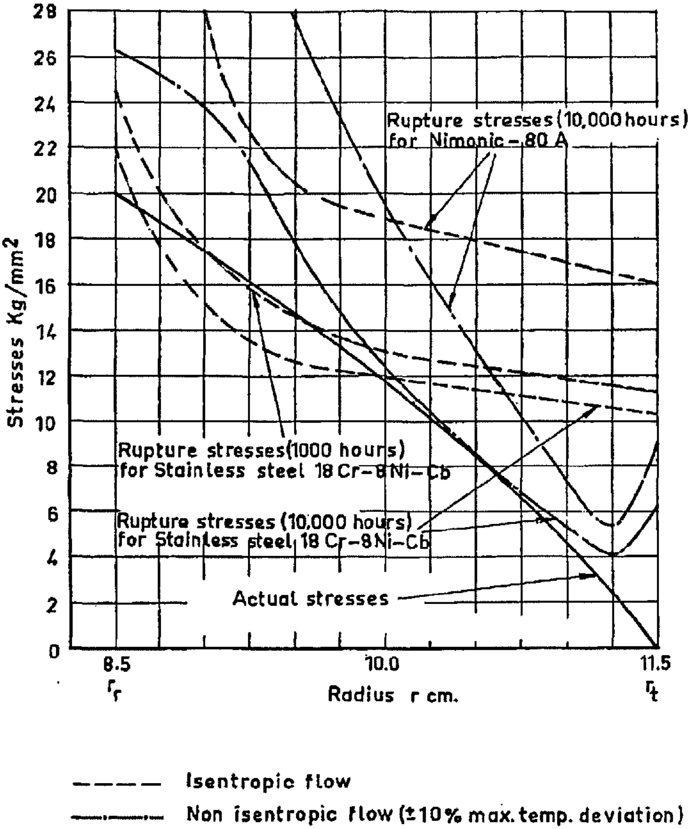

Fig. 8. Blade actual stresses and maximum working stresses

Finally, Fig. 8 shows again the results for the isentropic case, and for a non-isentropic flow with a temperature distribution with $\pm 10\%$ maximum deviations.

Such distribution of temperature is approximately the best for the case studied. The superalloys needed for the isentropic case can be substituted, for the same 10,000 hours life, by stainless steel.

On the other hand, if stainless steel would be used with isentropic flow, the life of the blade would be only 1,000 hours approximately, as shown in the same figure.

Considering that in small turbines, such as those used for automobile application, disc and blades are usually integral, and then of the same material, and that the price of stainless steel 18–8 is of the order of one fifth of that superalloy, it can be seen that an important reduction in price of the engine can be achieved. On the other hand, a very important gain in turbine life could be obtained if the same materials were utilized in both cases.

CONCLUSIONS

1°. By means of a proper selection of radials distributions of gas temperature increasing from blade root to blade tip, the life of an automobile turbine could be subtantially increased in many cases, or else, less expensive materials could be utilized instead of superalloys.

2°. For such non-isentropic flows the aerodynamic design of blades has to be made considering the actual gas temperatures distributions, and not by taking an average value of the temperature. Otherwise, the efficiency of the turbine would be reduced.

NOTATION

Symbols are described in the text. Therefore, only the most significant are listed.

c —blade chord
p_t —gas pressure
r —radius
T_t, T_b, T_a—temperatures of the gas, blade and of the cooling air of the turbine disc.
V_z, V_θ —axial and tangential gas velocities
α —heat transfer coefficient
α_t, β_t —angles
γ —isentropic exponent
λ —thermal conductivity
ω —angular speed
ρ_t —gas density
σ —blade area.
Ω —blade perimeter

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