REVIEWS

# SOME RESULTS OF THE INVESTIGATION OF TRANSPIRATION COOLING OF GAS TURBINE BLADES

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UDC 629.7.048.7

The most widely used method for cooling gas turbine blades is convection air cooling. However, it is practically no longer possible to use this method after the inlet gas temperature $T_g^*$ has reached 1500-1600 deg K. One could say that this method of cooling has reached a crisis stage [1]. A certain delay of the cooling crisis can be achieved, for example, by adding a film cooling of some blade areas to convection cooling, and by increasing the heat transfer in internal air channels by the use of some type of air flow agitators. Besides increasing the heat transfer coefficient on the cooling agent side ($\alpha_a$) it is advisable to reduce it on the gas side ($\alpha_g$) and to increase the contact area between cooling air and the surface being cooled.

Transpiration cooling meets both the last-mentioned conditions. Air is here pressed through a tight network of microchannels (pores) onto the external surface of the blade's curved porous skin thereby producing a thermally protective film on this surface. Transpiration cooling has been investigated in the USA since 1940 and was initially associated only with the liquid-propellant rocket engine. However, as early as in 1946 the advantages of its use in turbojet engines were stressed in [2].

Systematic investigations of transpiration cooling of gas turbine blades started in about 1950. The first porous blades were of hollow design and had a single cooling channel. The cooling air distribution over blade profile was not controlled and the result was that the condition of the blade surface was far from isothermal.

In 1952 Pametrada tested a 3500 hp gas turbine engine (GTE) with $T_g^* = 968$ deg K with nozzle blades made of a porous sheet material [3]. In 1953 the same firm tested a marine GTE with porous blades (Fig. 1) working at $T_g^* = 1473$ deg K. The blades were made of a 1.25 mm thick shell secured to a solid metal rod with longitudinal grooves for controlling the air flow to the shell.

In 1954 information was published on the research carried out at the NACA [4]. On blades similar to those of Pametrada only the temperature distribution along the load rod was measured. A sharp increase in temperature was observed along the rod starting from 3/8 of the blade height up to its tip; this led to the conclusion that the air fed through the blade root leaves it mainly through the base of the profile shell. This was apparently the cause of the failure of the porous shell at the periphery in the region of the outlet edge.

In 1954 data were published in [5] on the testing of nozzle boxes with transpiration-cooling which also had a porous blade shell mounted on the load rod; and the results obtained in measuring blade surface temperature in the central (height) section were published for the first time.

In the sixties a considerable increase in $T_g^*$ and the development of cooling methods noticeably increased the interest in transpiration cooling of turbine blades. The biggest research program was carried out by Curtiss-Wright.

The first investigations were concerned with full-size turbines with $T_g^* = 1480$ deg K (for short periods temperature of $T_g^* = 1810$-$1920$ deg K) were obtained; investigations were also carried out into the production of blades with transpiration cooling. Most of the re-

N. E. Bauman Moscow Polytechnic Institute. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 28, No. 3, pp. 533-544, March, 1975. Original article submitted December 27, 1973.

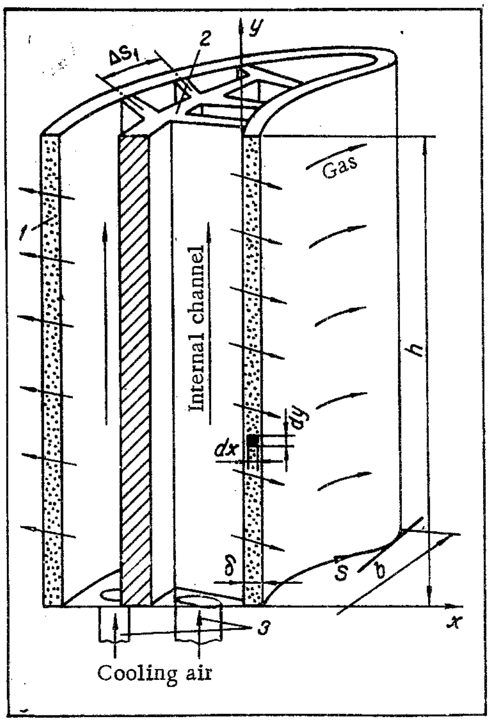


Fig. 1

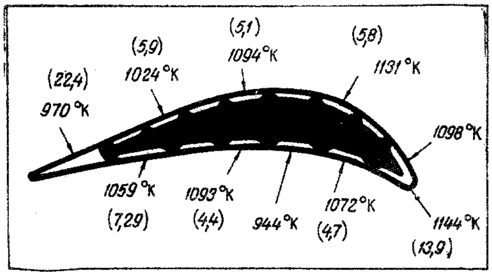


Fig. 2

Fig. 1. General view of a Pametrada blade with porous shell: 1) shell; 2) rod; 3) cooling air channels. a) Internal channel, b) cooling air.

Fig. 2. Temperature at some points of the external shell and rod surfaces of a blade in the mean radius cross section and the flow rates of cooling air (in brackets in %) when testing the J-65 engine.

search program of Curtiss-Wright consisted of testing a full-size modified J-65 turbojet engine with porous blades of a single-stage turbine. In testing the turbine at $T_g^* = 1644$ deg K the following data were obtained on the temperature field of the shell and nozzle rod and on the distribution and rate of cooling air flow along the channels in the central section of the blade (Fig. 2) [6]. The temperature field was obtained for a cooling air flow rate of $g_{coo} = 0.0385$ and an average pressure difference across a 0.762 mm thick porous shell of 0.21 $kg/cm^2$ (20,600 $N/m^2$).

Figure 3 also shown the distribution of average shell and rod temperatures over blade height. An important conclusion was that the manufacture of porous blades does not present special problems and their cost is only slightly higher than the cost of blades with convection cooling [6]. During the next research stage the temperature $T_g^*$ was increased to 1783 deg K [7].

The temperature distribution over the blade surface within this temperature range, with $g_{coo} = 0.045$, is shown in Fig. 4; Fig. 5 shows the variation of temperature over blade height for some sections of the load rod (the tests were carried out on a modified J-65-W-5 engine). It is interesting to note that the time needed to reach maximum temperature when testing the J-65-W-5 engine was shorter than in the case of a standard J-65 engine with convection cooling. In a number of cases a coating of aluminum suspension was used for changing the porosity of individual blade sections (in particular, for reducing the porosity at blade root) and this produced a more favorable temperature distribution on the blade surface.

The manufacture and finishing of experimental engines with transpiration cooling of blades required shell materials with specified porosity and needed new shell and blade production methods. The first important problem in developing such blades was the selection of a material for the wire needed for the wire cloth of which the shell is made. The main requirement which shell material must satisfy is the resistance to oxidation characterized by such measureable parameters as the change of specimen weight during the test, the change in thickness of the specimen and of the oxide surface layer under various temperatures and

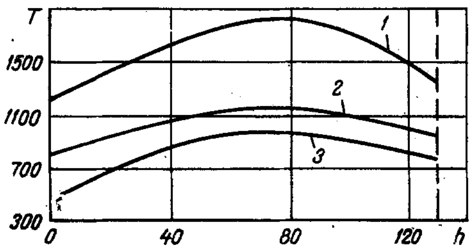


Fig. 3. Distribution of mean shell and rod temperatures over the blade height under the conditions described in Fig. 2: 1) gas temperature in front of the rotor; 2) inlet edge temperature; 3) rod temperature.

test durations. Investigations showed that among alloys suitable for the manufacture of wire the following produced best results: chromium—nickel alloys TD (Ni base; 2% $ThO_2$; 20% Cr), DH = 242 (Ni base; 20% Cr; 1% Cb), Hastalloy X (Ni base; 22% Cr; 18.5% Fe; 9% Mo; 1.5% Co), and GE - 1541 (Fe base; 15% Cr; 4% Al; 1% V) [8].

Porous sheet specimens made of GE - 1541 alloy retained permeability after heating for 600 h at T = 1256 deg K. The work carried out by Curtiss-Wright resulted in the selection of alloy Nichrome—V—Cb for blade shells. This alloy was sufficiently plastic and heat resistant up to temperatures of 1200 - 1250 deg K.

The diameter of wire used was in most cases 0.08-0.13 mm.

The blade rod, made by precision casting, accurate stamping, or machining, represents the carrying element of the structure. For securing the shell and the distribution of the cooling air over the blade contour special ribs were provided on the rod surface. Rods were made of cast alloys Heins HS 31, Raneg 41, Inconel 713, Inconel 700, SM 322; workable alloys Raney 41, IN - 100, Nimonic 115.

The porous shell can be secured to the load rod by soldering with a high-temperature solder, by resistance welding, by electron-beam welding, or by diffusion. The two last-mentioned methods produce a joint of highest quality both with regard to strain and to quality of the blade's working surface [9].

Generally the process of producing blades with transpiration cooling does not involve any fundamental problems apart from oxidation.

Transpiration cooling of blades was investigated not only on actual engines; a number of researches [10, 11, 14, 16] were concerned with the laws governing the heat transfer and the aerodynamic characteristics of blades when this type of cooling is used. Consider some results of these investigations.

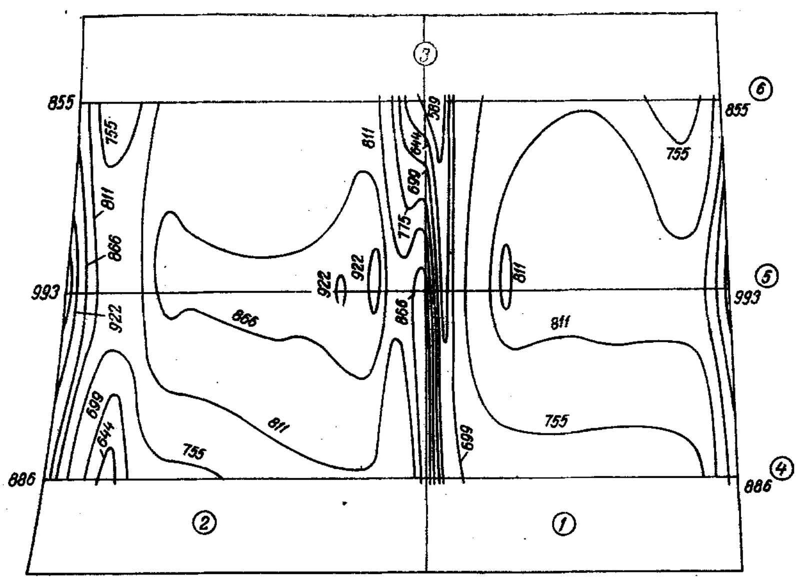


Fig. 4. Temperature field on blade surface: 1) blade front; 2) blade back; 3) inlet edge; 4) root section; 5) central section; 6) peripheral section.

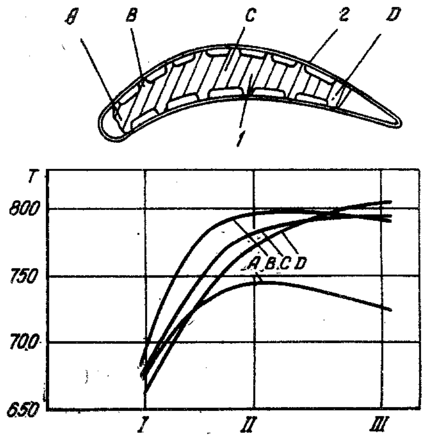


Fig. 5. Variation of temperature of some sections of the rod over blade height: I) in the root section; II) in the central section; III) in the peripheral section; 1) rod; 2) porous shell.

## 1. Investigation of Heat Transfer Processes in Blades with Transpiration Cooling

The investigation of special features of heat transfer in nozzle boxes and blades with transpiration cooling (Fig. 1) involves the investigation of three independent processes; heat transfer in internal channels for cooling air which, in effect, is a blind channel with one porous wall (Shell 1); heat transfer in a porous shell; and heat transfer through the external shell wall. Here one can solve either the direction problem, i.e., the determination of porous shell temperature for a given flow rate distribution over the blade contour, or the reverse problem, i.e., the distribution over the blade contour of a flow rate of air needed to maintain a specified blade surface temperature.

In earlier investigations the efficiency of transpiration cooling was associated only with the reduction of thermal flux from gas to blade by creating a thermally insulating air film on the porous surface. Thus, only one of the heat transfer processes was considered: on the external side of the porous shell. The effect of shell cooling by the flow of a cooling agent through its pores was considered to be very weak, while the convection heat transfer in the internal channels was completely neglected. It will be shown below that such simplifications of the processes under consideration are unfounded.

Heat Transfer on the External Surface of the Porous Shell. The determination of heat transfer from hot gas to blade surface is an indispensible stage in designing any cooling system. However, in the widely used convection cooling systems the theoretical calculations do not always agree with the experimental data because of unsatisfactory semiempirical theories of a nonisothermal boundary layer on a curved surface. It is therefore difficult to assume that the development of these theories for calculating the boundary layer on a permeable surface can produce a satisfactory agreement between $\alpha_g$ values and the experimental data.

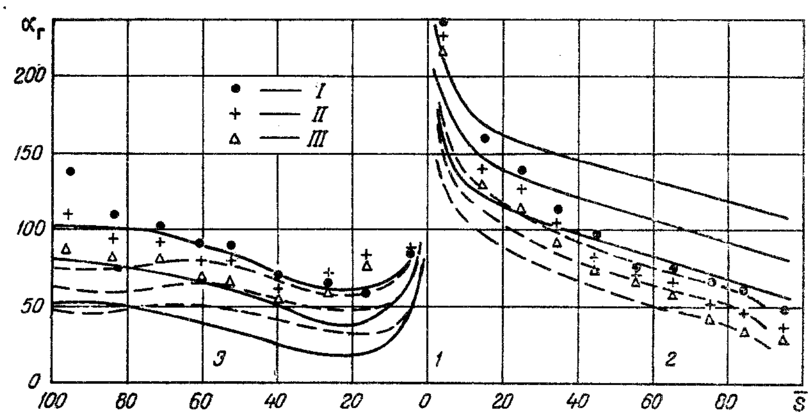


Fig. 6. Variation of the coefficient of heat transfer $\alpha_g$ from gas to blade profile with a uniform distribution of injected air over the profile at $M_2 = 0.77$ and three different rates of injection; I) m = 2920 kg/mm$^2$ h; II) 3900 kg/mm$^2$ h; III) 4880 kg/mm$^2$ h; 1) inlet edge; 2) blade back; 3) blade front; dotted curves — according to [13]; solid curves — according to [12]. $\bar{S}$, %.

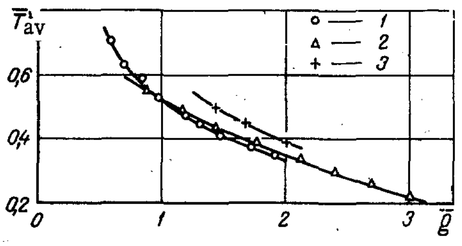


Fig. 7. Average surface temperature $T_{av} = (T_w - T_{a\ in})/(T_g - T_{a\ in})$ as a function of the amount of injected air $g_{coo}$, %: 1) isothermal blade at low (0.45%) main flow turbulence; 2) constant (over the contour) distribution of injected air; 3) isothermal blade at high (3%) level of the main flow turbulence.

However, papers [10, 11] showed a method for carrying out satisfactorily accurate calculations. Figure 6 shows the experimentally obtained $\alpha_g$ values for three different air injection rates and compares them with corresponding values calculated from the widely known theories [12] and [13] (the data are obtained for a constant cooling air flow rate). It will be noted that heat transfer in the inlet edge zone is exceptionally high. The processing of the results of several experimental investigations made it possible to calculate the air consumption for maintaining the blade surface temperature at the required level. Figure 7 shows, as an example, these characteristics for three different cases: isothermal blade surface at low (0.45%) and high (3%) main flow turbulence values as well as for a uniform distribution (over the contour) of the air flow rate m (with a turbulence of 0.45%).

In order to simplify the calculations some workers use simpler hypotheses assuming an independence (conservatism) of the law of heat transfer from the longitudinal pressure gradient on the blade surface. For example, in [14] use is made of the experimental $Nu_0$, $St_0$ numbers obtained for a plane plate without injection; these data are used for calculating Nu values for a permeable profile from the equation

$$Nu = \frac{A}{1-A}\,\frac{Nu_0}{St_0}\cdot\frac{\rho_w v_w}{\rho_0 v_0},$$

where $\rho_w$, $\rho_0$ are the local density of the injected air and the density of gas in the flow core respectively; $v_w$, $v_0$ are the local injected air velocity and the velocity in the flow core; $A = 1 + 1/3St_0 \cdot \rho_w v_w/\rho_0 v_0$-is an empirical relation.

New porous materials are characterized by increased surface coarseness. As a result the heat transfer conditions on blade surfaces differ considerably from those prevailing on aerodynamically smooth surfaces: in the case of pores with a diameter of 50-100 μ the coarseness has a marked effect on the location of the point of transmission from the laminar boundary layer to the turbulent boundary layer, and this finds its expression in increased Nu numbers over the profile shown in Fig. 8 [14] (Nu numbers were determined by a theoretical method using the Scholz theory). However the authors would like to draw attention to a considerable discrepancy between calculated and experimental Nu values; the best agreement was obtained for an assumed insignificant effect of coarseness, which suggests that the effect of coarseness, which is in principle important within the assumed theory, is considerably exaggerated.

Heat Transfer in a Porous Shell. The calculation of an unsteady three-dimensional temperature field of a porous shell (function T = T (x, y, s) Fig. 1) is at present a very difficult problem. For this reason most investigators only use the solution of a one-dimensional problem for determining the function T = T(x) which characterizes the variation of temperature of a porous material over shell thickness according to Grootenhuis [15]. Here the most difficult part is the calculation of the volumetric coefficient of heat transfer $\alpha_v$ in the pores of the material, because its calculated values differ considerably from experimental data.

In most cases $\alpha_v$ is determined by a purely experimental method. Figure 9 shows typical measurement results [11] compared with values calculated after Grootenhuis.

Heat Transfer in Internal Channels. The main purpose of internal channels in blades under consideration is the supply of air to the porous shell regions and its distribution according to the necessity of maintaining the required blade surface temperature. In their original form, as it was mentioned above, the porous blades were hollow; the distribution

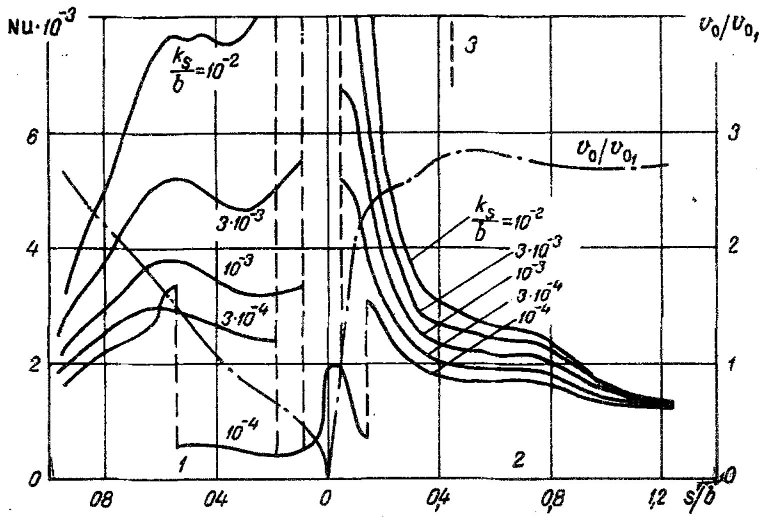


Fig. 8. Distribution of relative velocities $v_0/v_{01}$ ($v_{01}$ is the flow velocity at channel inlet) and Nu numbers over the contour of blades of different coarseness: 1) profile front; 2) profile back; 3) transition from laminar to turbulent flow, $Re_1 = 5.1\cdot10^5$; $k_s/b$ is the relative coarseness.

of injected air over the blade contour was not controlled or controlled only by the use of shells with variable thickness and porosity. The construction of blades by the last-mentioned method appears very difficult.

In the case of a single internal channel (in hollow blades) at relatively low cooling air flow rates the rate of internal convection heat transfer was very low, which suggests that this process has, in principle, little effect on the total characteristics of the cooling system. However, a detailed examination of this process [11] revealed a considerable effect of heat transfer in the internal channels on the temperature conditions of a porous blade.

Figure 10 shows the variation of temperatures of the cooling agent and of the porous wall along one of the cooling channels the blade for two blade designs.

The process of heat transfer in internal channels has not been sufficiently investigated theoretically. This can be attributed to a very complex physical nature of the flow in a relatively low blind rectangular channel with an increased coarseness of the porous wall; in this case the cooling agent leaves the channel through the porous wall with the result that the flow rate in the channel changes between channel inlet and blade periphery from $g_{c00}$ to 0. Using for process analysis the characteristic equation for turbulent flow in a tube with nonpermeable walls

$$\mathrm{Nu_c} = c\mathrm{Re_c^n} = 0.02\mathrm{Re_c^{0.8}}$$

(where $Nu_c$, $Re_c$ are the corresponding Nu and Re numbers calculated for the internal channel), it is possible to describe the flow rate variation along the channel length due to the penetration of the cooling agent through the porous wall.

The loss of air through penetration is apparently the reason for a slow development of the boundary layer, which has a considerable effect on the thermal interaction between cooling agent and shell.

Thus the flow under consideration differs considerably from the normal flow in tubes, which is bound to change the characteristic equation.

For an approximate evaluation the following relation can be used [11]

$$\mathrm{Nu_{\bar{c}}} = 0.045\mathrm{Re_{\bar{c}}^{0.8}}.$$

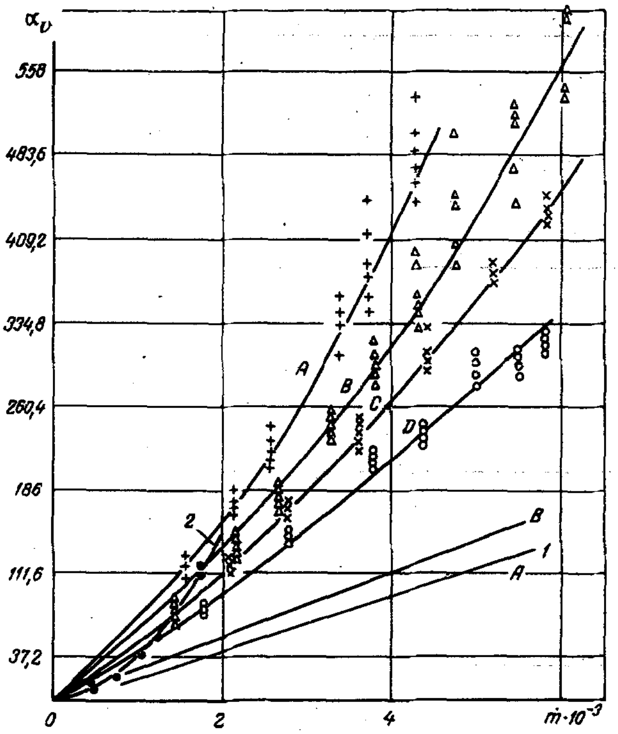


Fig. 9

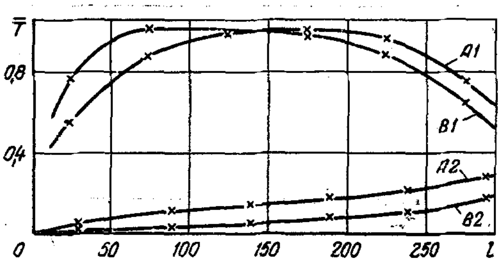


Fig. 10

Fig. 9. Results obtained in measuring volumetric coefficients of heat transfer $\alpha_v$ in different porous materials and their comparison with values calculated in [15]: 1) calculated data [15]; 2) porous material Poroloy (with stainless steel as the base); A, B, C, D) are porous bronzes Porosint of four different types; $\alpha_v$ is the volumetric coefficient of heat transfer, W/m$^3$ °C$\cdot10^{-4}$.

Fig. 10. Variation of the nondimensional temperature $\bar{T}$ of a porous wall (curves A1, B1) and of the cooling agent (A2, B2) along a rectangular channel with one porous wall, $\bar{T} = (T-T_{a\ in}/T_{wmax}-T_{a\ in}$; A — for a sharp variation of the cooling channel area at the entry; B — for smooth entry; $l$ — distance from channel inlet mm.

## 2. Investigation of Aerodynamical Characteristics of Porous Nozzle Boxes

A transverse injection of a cooling agent has a strong effect on the structure of the dynamic boundary layer on the blade surface: the layer thickness increases, the turbulent flow zone widens, the velocity profile of the boundary layer changes and becomes, as under the action of a positive pressure gradient, less "filled," and an earlier breaking away of the layer from the surface becomes likely. The effect of increased coarseness of porous blade surfaces becomes substantial. Because of all these factors there is a danger of a sudden deterioration of aerodynamic properties of nozzle boxes of turbines with porous cooling of blades. Definite conclusions on this problem can be drawn from the results of investigations carried out by Bayley and Wood [16]. The investigations were carried out mainly in order to obtain comparative characteristics of stationary plane aerodynamic turbine bladings with nonpermeable surface and porous surface for various rates of air injection. It should be noted that in these experiments the law of distribution of injected air over the profile contour did not meet the requirements resulting from the need to maintain the necessary blade surface temperature, for example, constant; during the first test series the ratio of local injection velocity $v_w$ to local velocity at the outer boundary of the boundary layer $v_o$ was maintained constant; $v_w/v_o$ = const (the results are given in Fig. 11a); during the second test series the mass flow rate of injected air per unit area a (Fig. 11b) was the constant quantity.

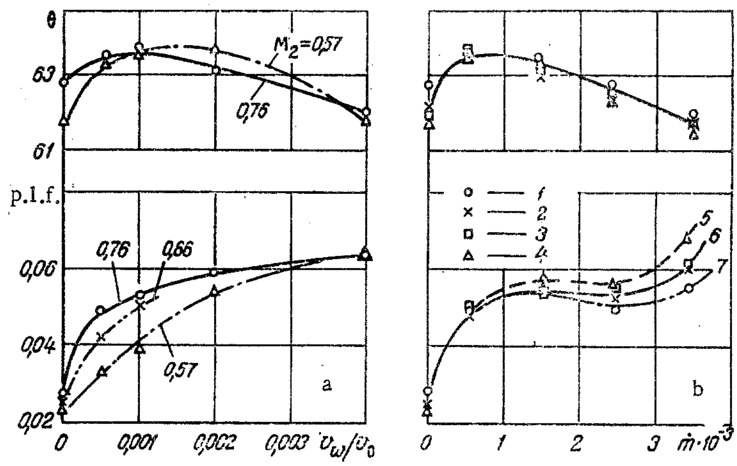


Fig. 11. Effect of injection on the turbine blading characteristics: a) effect of the rate of injection on the blading characteristics at constant $M_2$ values at a constant value of $v_w/v_0$ along the profile; b) effect of cooling agent flow rate at constant $M_2$ values and a constant cooling agent flow rate m along the profile (for b), 1) - $M_2$ = 0.76; 2) 0.70; 3) 0.66; 4) 0.57; 5) 0.57; 6) 0.66; 7) 0.76; θ is the average angle of flow outlet; p.l.f. = profile loss factor).

In both cases the loss increased by no more than 2-3.5 times. Injection had little effect on the flow outlet angle θ. However, these tests failed to provide information on how much the losses increase, since in both test series, as it has already been mentioned above, the law of injection was not linked with the law of surface temperature distribution. The increase of loss in the case of low-intensity injection was attributed by the authors to the boundary layer on the blade profile back becoming turbulent; another characteristic feature is the repeated increase of losses at high injection intensity values (Fig. 11b) due to the breaking away of the boundary layer, especially at low external flow velocities (lower $M_2$ values).

The integral characteristics of Fig. 12 are of some practical interest, they provide a means for determining the effect which injection has on profile losses and the outlet angle for a known ratio of cooling air flow rate to gas flow rate in the channel (the injection law corresponds to a constant (over the profile) m, i.e., to the flow rate of cooling air). The results of calculating the effect of increased coarseness of porous blades on the total increase in profile loss, calculated as the ratio of the total head loss across the blading $\Delta p^* = p_1 - p_2^*$ to the velocity head of the flow at blading inlet $q_1 = \rho_1 (v_{01}^2/2$ (Fig. 13, [14]), show that the coarseness along, even in the absence of injection, substantially increases the profile loss. Interesting from the point of view of a possible use of transpiration cooling in connection with increased level of profile loss during injection, are the results of theoretical assessment given in [16].

A comparison was carried out of the performance characteristics of an aircraft gas turbine engine performance with convection and transpiration cooling systems obtained for the following engine parameters: Increase of pressure in compressor 15; initial air parameters at GTE inlet (under static condition): pressure, 1.033 kg/cm² (1.013 · $10^5$ N/m²); temperature, 288 deg K; adiabatic efficiency of the compressor, 0.9; No. of turbine stages, 2; blade surface temperature; 1200 deg K.

Air consumption with convection cooling of the first stage was assessed at 4% of working medium consumption at a maximum gas temperature of 1450 deg K; profile loss was 0.022; secondary loss 0.03; overflow loss through radial clearance 0.025; all this together produced an adiabatic efficiency of the turbine $\eta_{ad\ t}$ of 0.92. The specific thrust $R_{sp}$ of GTE was here 819 N sec/kg at specific fuel consumption $c_R$ = 0.099 kg/N h. In the case of transpiration cooling the cooling agent consumption in the same GTE was 1.75%, profile loss was assessed at 0.062.

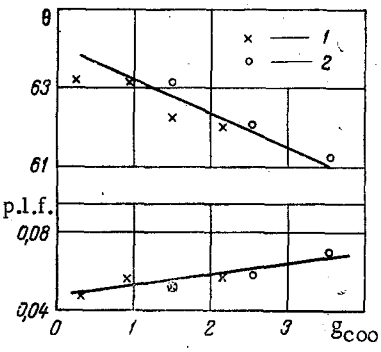


Fig. 12

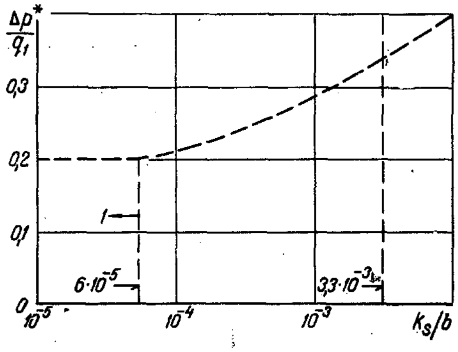


Fig. 13

Fig. 12. Variation of characteristics (profile loss and outlet angles) as functions of specific injection air flow consumption: 1) $M_2 = 0.76$; 2) 0.57. a) profile loss factor; b) $g_{coo}$.

Fig. 13. Calculated effect of the coarseness of porous blades on the total increase of profile losses: 1) hydraulically smooth surfaces.

Assuming that the secondary loss and the clearance loss are the same for both cooling methods, for transpiration cooling $\eta_{ad\ t} = 0.89$; $R_{sp} = 832$ N sec/kg, and $c_R = 0.098$ kg/N h.

Thus at moderate gas temperature the characteristics of GTE with both cooling systems being compared are very close to each other.

However, a similar comparison for $T_g^* = 1800$ deg K leads to the conclusion suggesting that the transpiration cooling is more advantageous. With cooling agent consumption of 7.2%, $\eta_{ad\ t} = 0.89$; even taking into account the additional work spent on pumping the cooling agent, the specific thrust increased to 1010 N sec/kg at $c_R = 0.114$ kg/N h. Similar calculations were carried out in a number of other investigations; in particular, in [14] a comparison of the efficiencies of convection and transpiration cooling was carried out for a two-stage TJE with a pressure increase of 24.

At a maximum gas temperature of 1600 deg K the specific thrust and fuel consumption become the same for both cooling systems, the cooling air consumption for transpiration cooling is 0.53 of its consumption in the convection cooling system, the efficiency of the high-pressure turbine (HPT) decreased by 3%. For future turbojet engines with $T_g^* = 1860$ deg K the cooling air consumption is equal to its consumption in the convection cooling system of the initial engine with $T_g^* = 1600$ deg K, the specific thrust increases by 34%, fuel consumption decreases by 1% with the efficiency of the HPT decreasing by 3.6%.

## NOTATION

$c_R$, specific fuel consumption, kg/N·h; M, Mach number; Nu, Nusselt number; Re, Reynolds number; $R_{sp}$, specific thrust, N·sec/kg; St, Stanton number; T, temperature, deg K; b, profile chord length, m; $g_{coo}$, relative cooling air consumption (ratio of cooling air flow rate to gas flow rate through turbine), %; h, blade height, m; $k_s$, average height of irregularities of coarse blade surface, m; $\Delta p^* = p_1^* - p_2^*$, change in total pressure of flow in the blading, N/m$^2$; $q = \rho v^2_0/2$, velocity head, N/m$^2$; s, coordinate along the periphery of the external blade shell surface, starting from the critical point at blade front, m; $\bar{s} = s/L$, nondimensional coordinate, %, where L is the length of the periphery along the back or front of the blade, m; v, velocity, m/s–c or m/h; x, coordinate along the normal to the inner shell surface, m; y, coordinate along the blade length, m; α, heat transfer coefficient, W/m$^2$·°C; $\alpha_v$, volumetric coefficient of heat transfer, W/m$^2$ deg; δ, blade shell thickness, m; $\eta_{ad}$, adiabatic efficiency; ρ, density, kg/m$^3$; $m = \rho_w v_w$, flow rate of the injected cooling agent, kg/m$^2$ h. Indices: a, air (cooling agent); in, inlet of blade cooling channel; g, gas; c, channel; w, wall; t, turbine; *, parameters of retarded flow; 0, parameter values for flow core; max, maximum value of a parameter; 1, blading inlet; 2, blading outlet.

## LITERATURE CITED


1. V. I. Lokai, "Increasing gas temperature in front of turbine — main direction of development of modern GTE," Izv. Vuz, Aviats. Tekh. 4 (1972).
2. P. Grootenhuis and N. Moor, British Patent No. 619634 (1946).
3. "Pametrada Gas Turbine Experience (Research on Research on Sweat Cooling and a Liquid Alloy Heat Exchanger)," Oil Engine and Gas Turbine, 20, 231 (1952).
4. P. Donough and A. Diaguillo, Exploratory Engine Test of Transpiration-Cooled Turbine Rotor Blade with Wire Cloth Shell, NACA, E 53 K 27 (1954).
5. S. Andrews, H. Ogden and J. Marshall, Some Experiments on an Effusion - Cooled Turbine Nozzle Blade, NGTE, 132 (1954).
6. S. Lombardo, N. Lauziere, and D. Kump, Design and Fabrication Aspects of Transpiration Air Cooled Blades for 2500 deg F Turbine Operation, SAE Paper No. 820A (1964).
7. S. Moskowitz and S. Lombardo, 2750 deg F Engine Test of a Transpiration Air-Cooled Turbine, ASME Publication, Paper No. 10-WA/GT-1 (1971).
8. F. W. Cole, J. B. Padden, and A. R. Spencer, Oxidation Resistance Materials for Transpiration Cooled Gas Turbine Blades, Part I, NASA CR-930 (1970).
9. S. Lombardó, and S. Moskowitz, Experience with Transpiration Cooled Bayley AGARD-CP-73-71 (1971).
10. F. J. Banley and A. B. Turner, The Transpiration-Cooled Gas Turbine, ASME Paper No. 10-GT-56 (1970).
11. F. J. Bayley and A. B. Turner, Transpiration-Cooled Turbines, AGARD-CP-73-71, High-Temperature Gas Turbines (1971).
12. S. S. Kutateladze and A. I. Leont'ev, Turbulent Boundary Layer of Compressible Gas [in Russian], Novosibirsk (1962).
13. S. V. Patankar and D. B. Spalding, Heat and Mass Transfer in Boundary Layers, Morgan-Grampian, London (1967).
14. H. Prechter, A. Schönbeck, and N. Scholz, Effusion Cooling of Turbine Blades, AGARD-CP-73-71 (1971).
15. P. Grootenhuis, "The mechanism and application of effusion cooling," Roy. Aeronaut. Soc., 63, 578 (1959).
16. F. J. Bayley and G. R. Wood, "Aerodynamic performance of porous gas turbine blades," J. Roy. Aeronaut. Soc., 73, 705, Sept. (1969).