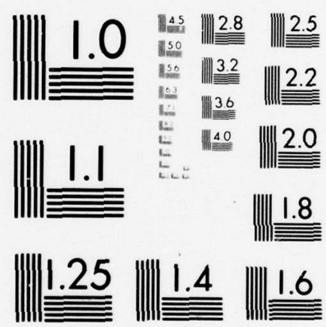
AD-A065 974 FOREIGN TECHNOLOGY DIV WRIGHT-PATTERSON AFB OHIO F/G 21/5
CALCULATION OF THE HYDRAULIC FRICTION OF THE COOLING CHANNELS O--ETC(U)
OCT 77 L N ODIVANOV

UNCLASSIFIED FTD-ID(RS)T-1827-77 NL

1 OF 1
AD
A065974

END
DATE
FILMED
5-79
DDC



MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

FTD-ID(RS)T-1827-77

1

AD-A065974

FOREIGN TECHNOLOGY DIVISION

AIR FORCE SYSTEMS COMMAND

CALCULATION OF THE HYDRAULIC FRICTION OF THE COOLING CHANNELS OF GAS-TURBINES BLADE

by

L. N. Odivanov

FOREIGN TECHNOLOGY DIVISION

Approved for public release; distribution unlimited.

11 09 155

FTD-ID(RS)T-1827-77

MACHINE TRANSLATION

FTD-ID(RS)T-1827-77 27 Oct 1977

MICROFICHE NR: FTD-77-C-001352

CALCULATION OF THE HYDRAULIC FRICTION OF THE COOLING CHANNELS OF GAS-TURBINES BLADE

By: L. N. Odivanov

English pages: 27

Source: Trudy Kazanskogo Aviatsionnogo Institute, No. 128, 1971, pp. 3-13.

Country of origin: USSR
Translated by: Robert D. Hill
Requester: FTD/PDRS

Approved for public release; distribution unlimited.

In the interest of economy and timeliness, the original graphics have been merged with the computer output and editing has been limited to that necessary for comprehension. No further processing is anticipated.

U. S. BOARD ON GEOGRAPHIC NAMES TRANSLITERATION SYSTEM

Block	Italic	Transliteration	Block	Italic	Transliteration
А а	*А а*	A, a	Р р	*Р р*	R, r
Б б	*Б б*	B, b	С с	*С с*	S, s
В в	*В в*	V, v	Т т	*Т т*	T, t
Г г	*Г г*	G, g	У у	*У у*	U, u
Д д	*Д д*	D, d	Ф ф	*Ф ф*	F, f
Е е	*Е е*	Ye, ye; E, e*	Х х	*Х х*	Kh, kh
Ж ж	*Ж ж*	Zh, zh	Ц ц	*Ц ц*	Ts, ts
З з	*З з*	Z, z	Ч ч	*Ч ч*	Ch, ch
И и	*И и*	I, i	Ш ш	*Ш ш*	Sh, sh
Й й	*Й й*	Y, y	Щ щ	*Щ щ*	Shch, shch
К к	*К к*	K, k	Ъ ъ	*Ъ ъ*	"
Л л	*Л л*	L, l	Ы ы	*Ы ы*	Y, y
М м	*М м*	M, m	Ь ь	*Ь ь*	'
Н н	*Н н*	N, n	Э э	*Э э*	E, e
О о	*О о*	O, o	Ю ю	*Ю ю*	Yu, yu
П п	*П п*	P, p	Я я	*Я я*	Ya, ya

*<u>ye</u> initially, after vowels, and after ъ, ь; <u>e</u> elsewhere.
When written as ё in Russian, transliterate as yё or ё.

RUSSIAN AND ENGLISH TRIGONOMETRIC FUNCTIONS

Russian	English	Russian	English	Russian	English
sin	sin	sh	sinh	arc sh	$\sinh^{-1}$
cos	cos	ch	cosh	arc ch	$\cosh^{-1}$
tg	tan	th	tanh	arc th	$\tanh^{-1}$
ctg	cot	cth	coth	arc cth	$\coth^{-1}$
sec	sec	sch	sech	arc sch	sech^{-1}
cosec	csc	csch	csch	arc csch	csch^{-1}

Russian	English
rot	curl
lg	log

SUBJECT CODE 5147D

Page 3.

CALCULATION OF THE HYDRAULIC FRICTION OF THE COOLING CHANNELS OF GAS-TURBINES BLADE

L. N. Odivanov.

Is given the derivation of the calculation formulas for determining the hydraulic friction of the cooling channels of blades. In formulas is considered the effect of the preheating of air, losses from friction and centrifugal backwater ~~to~~ on the capacity of cooling channels.

For blades with radial (longitudinal) cooling channels are given the calculation formulas, suitable for a programming ~~by~~ on small ETsVM [- digital computer]. For blades with transverse channels is given an example of the calculation on the basis of the selection of the loss factors in handbook. Calculation data satisfactorily coincided with

the experimental data of other authors.

In gas turbine construction there was received ~~the~~ wide acceptance of blades air-cooled. The diagrams of some of them are given in Fig. 1, 2.

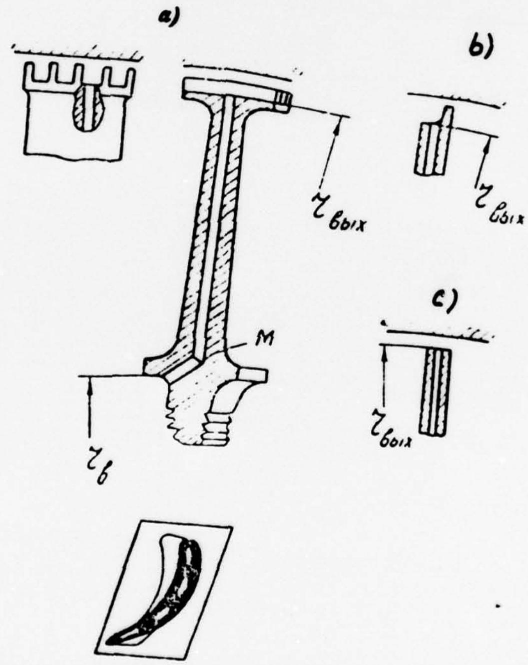


Fig. 1. ~~Schematics~~ Diagrams of the cooled blades with the longitudinal channels.

Page 4.

~~During~~ In the ~~layout~~ design of cooling channels it is necessary correctly to estimate hydraulic losses. In the existing methods of calculation [1], etc. there is not examined the combined effect of some special ~~feature/~~peculiarities of air flow in the cooling channels of the blades: the effect of the preheating of air from the heat, ~~applied~~ fed through walls from gas; the heat, isolated from friction; the effect of centrifugal forces, etc. This leads to certain arbitrariness in the calculations, frequently ~~it~~ impedes the account of the specific conditions of the flow of coolant and the generalization of the loss factors.

In work [2] is experimentally investigated the effect of the factors indicated for a rotor with the blades, which have ~~within~~ inside deflector (Fig. 2). The clearance between the external wall and the deflector was maintain/withstood because of point punch-outs on deflector.

In works [3, 4] is investigated the ~~construction~~ design of blade with the transverse motion of air coolant, with its ejection in the area of trailing edge to the side of trough. Experimental data [4, 5], etc. confirm the possibility of applying reference data [6] with some corrections.

In the present work is proposed the method of the calculation of hydraulic friction and determination of the flow rate of cooling air for blades with the longitudinal and cross flow of coolant with the assigned/~~prescribed~~ geometry of channels and parameters of air coolant.

Below is set forth the derivation of the calculation formulas on the basis of the equations of energy and continuity.

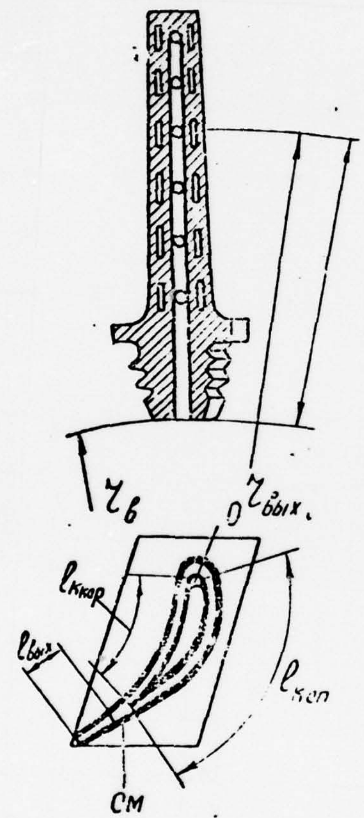


Fig. 2.
(Caption on next page.)

Fig. 2. ~~Schematic~~ Diagram of ~~the~~ a cooled blade with transverse channels.

Page 5.

1. Equation of energy.

The equation of energy for air flow in cooling channels (in relative motion) is ~~record/~~written in the form:

$$c_p(T_в - T_{вых}) + Q + \frac{u^2_{вых} - u^2_в}{2} - \frac{w^2_{вых} - w^2_в}{2} = 0. \qquad (1)$$

Here Q - the heat, applied to air from the walls of the heated channel; ~~x~~ w is ~~a~~ relative speed; u - peripheral speed; T - the temperature of air in channels; c_p is the specific heat of air.

~~Indices~~ Subscripts "в" and "вых" designate the parameters of air coolant at the duct inlet and at ~~output/yield~~ outlet from it respectively (Figs. 1, 2).

According to the equation of the 1st law of the thermodynamics

$$dL_r + dQ = c_p dT - \frac{dp}{\gamma}, \qquad (2)$$

where L_r is the work lost in friction; p and γ - the pressure and air density.

In the process of the expansion of air its preheating through walls from gas, from friction and from centrifugal compression let us consider ~~through~~ in terms of the single average polytropic index ~~electromesons~~ ~~of the polytrope~~ n, i.e., let us accept $p/\rho^n = \text{const}$. Then integration of (2) will give

$$Q = c_p(T_{\text{вых}} - T_{\text{в}}) + \frac{n}{n-1} R T_{\text{в}}\left[1 - \left(\frac{p_{\text{вых}}}{p_{\text{в}}}\right)^{\frac{n-1}{n}}\right] - \int_{\text{в}}^{\text{вых}} dL_r. \qquad (3)$$

It is known that

$$\int_{\text{в}}^{\text{вых}} dL_r = L_r = \frac{w^2_{\text{вых.т}} - w^2_{\text{вых}}}{2}, \qquad (4)$$

where $w_{\text{вых.т}}$ is ~~pitch~~ the theoretical speed without taking into account ~~of~~ hydraulic losses.

Taking into account expressions (3) and (4) equation (1) takes the form

$$w^2_{\text{вых.т}} = 2\frac{n}{n-1} R T_{\text{в}}\left[1 - \left(\frac{p_{\text{вых}}}{p_{\text{в}}}\right)^{\frac{n-1}{n}}\right] + u^2_{\text{вых}} - u^2_{\text{в}} + w^2_{\text{в}}. \qquad (5)$$

Polytropic Index ~~electromesons~~ it is possible to determine from the known relation~~ship/ratio~~

$$\frac{n-1}{n}=\frac{\lg\frac{T_{max}}{T_{н}}}{\lg\frac{p_{max}}{p_{н}}}, \qquad (6)$$

and outlet temperature from channel from the equation

$$T_{max}=T_{н}+\Delta T^{*}-\frac{w^{2}_{max}-w^{2}_{н}}{2\frac{k}{k-1}R}. \qquad (7)$$

Page 6.

For determining w_{max} from value $w_{max.н}$ it is necessary to find the factor of hydraulic loss.

2. Factors of hydraulic loss.

For blades of the type a, b, c (Fig. 1) ~~Of~~ duct losses ~~they~~ are composed of the following forms: input, exit and loss from friction in channel. It is convenient to refer ~~relate~~ losses to real outlet velocity from channel, then $L_r=\xi_k\frac{w^{2}_{max}}{2}$ either according to expression (4) we will obtain

$$w^2_{\text{вых}} = \frac{w^2_{\text{вых.п}}}{1+\xi_k}$$

or

$$w^2_{\text{вых}} = \frac{2\frac{n}{n-1}RT_{\text{в}}\left[1-\left(\frac{p_{\text{вых}}}{p_{\text{в}}}\right)^{\frac{n-1}{n}}\right] + u^2_{\text{вых}} - u^2_{\text{в}} + w^2_{\text{в}}}{1+\xi_k}. \qquad (8)$$

The total energy losses in channel, referred to the unit of mass, will be:

$$\xi_k \frac{w^2_{\text{вых}}}{2} = \Sigma \int_l \frac{w^2 \zeta}{2D_{\text{г}}} dl + \Sigma \frac{\xi_{\text{м}} w^2_{\text{м}}}{2}.$$

Hence

$$\xi_k = \frac{1}{w^2_{\text{вых}}} \Sigma \int_l \frac{w^2 \zeta}{D_{\text{г}}} dl + \Sigma \frac{\xi_{\text{м}} w^2_{\text{м}}}{w^2_{\text{вых}}}, \qquad (9)$$

where $\xi_{\text{м}}$ is factor of local losses; ζ - the drag coefficient of the sections of channel; $D_{\text{г}}$ - the hydraulic diameter of the sections of channel; l is length of channel.

If we approximately accept a linear change in the losses from the wall friction on the section of channel, then the first term of expression (9) **will take the form:**

$$\xi = \frac{\zeta_{в} l}{2D_{г.в}}\left(\frac{w_{в}}{w_{вых}}\right)^{2} + \frac{\zeta_{вых} l}{2D_{г.вых}}.$$

Speed change on section from output~~/yield~~ to the inflection point of channel is negligibl~~y little~~e, i.e., we can ~~be accepted~~ assume that $w_{м} = w_{в}$.

Page 7.

Taking this into account the formula for determining the coefficient of the losses of the channels, depicted on Fig. 1, will take the **form:**

$$\xi_{k} = \frac{\zeta_{вых} l}{2D_{г.вых}} + \left(\frac{w_{в}}{w_{вых}}\right)^{2}\left(\xi_{м} + \frac{\zeta_{в} l}{2D_{г.в}}\right). \qquad (10)$$

The velocity ratio is determined from the equation of the continuity

$$\frac{w_{в}}{w_{вых}} = \frac{p_{вых} F_{вых} T_{в}}{p_{в} F_{в} T_{вых}}. \qquad (11)$$

From equations (6), (7), (8), (10), (11) we ~~is~~ **determine~~d~~** $w_{вых}$ **and** $T_{вых}$. In this case pressure at output~~/yield~~ **from holes** $p_{вых}$ **can be** accepted approximately [1] that which is being **chang~~ing~~ed** according to linear law from the gas pressure on periphery $p_{1н}$ **at** ~~entrance~~ inlet (into lattice) to $p_{2н}$ - at output~~/yield~~.

FOOTNOTE [1]. More precise according to data [7] p_{max} a little increases ~~flow/rises~~ with a decrease in the clearance δ and with an increase in the dynamic head of air coolant. ENDFOOTNOTE.

For the blades, depicted on Fig. 2, are characteristic still other forms of ~~the~~ losses whose coefficients are enumerated below:

the coefficient, which evaluates loss from air distribution in central channel ξ_p;

coefficients, the evaluating ~~entry~~ loss at inlet into hole 0 (Fig. 2), losses ~~by~~ from shock of flow into wall (internal surface of the spout of blade) and from the rotation of flow in the direction of transverse channels $\xi_{в}$ and $\xi_{у}$;

the coefficient, which considers losses from the mixing of flows upon their ~~meeting~~ encounter and from sudden expansion ~~on output/yield~~ at outlet from the channels of back and trough $\xi_{см}$.

Are examined below recommendations ~~to to~~ on the selection of the loss factors.

On the periphery of blades (Fig. 1b, c) the air flow at output/~~yield~~ from hole depends on the effect of the flow of gas; the mechanism of interaction is very complex; therefore the coefficient of outlet ~~leaving~~ losses $\xi_{вых}$ can be determined experimentally. Tentative calculations show that $\xi_{вых}=0.3\div1$ (the upper limit corresponds to small radial gap δ). For the blades, shown in Fig. 2, $\xi_{вых}$ it is possible to determine by Borda-Carnot's formula: $\xi_{вых}=\left(1-\frac{F_{вых}}{F_{п}}\right)^2$.

Page 8.

In works [3, 4] the total losses $\xi_{п}+\xi_{у}$ are determined experimentally; their value is more at the smaller relative clearances $\bar{h} = h/d$ between the leading edge of the deflector and the internal surface of blade. According to experiments [3] $\xi_{п}+\xi_{у}=4\div5$ with $\bar{h} \approx 1$, while according to the data [4] 0.7-1.7 when ~~with~~ $\bar{h} \approx 2$.

The satisfactory agreement of these data is obtained, if they ~~them~~ are selected on graphs for circular exhaust shafts ~~mines~~ with hood, upon consideration of additional entry loss into hole (account of the evenness of entrance) [1, 6]. The coefficients of losses $\xi_{п}$ at the inlet ~~entrance~~ into cooling channels, ξ_{p}, $\xi_{см}$ can be found from handbook [6].

The values of drag coefficients $\zeta_{к}$ of the transverse channels, obtained in experiments [4], satisfactorily coincide with values calculated [6]. The drag coefficients of radial channels $\zeta_{п}$ and ζ_{max} are determined from formulas for ducts. One must take into account that the friction of the rotating radial channels is more than ~~being~~ nonrotating ~~unrotated~~ ones. According to experiments [5] excess ~~composes~~ consists of 15o/o.

These coefficients of losses and recommendation regarding their selection require further refinement (especially for rotor blades) with the accumulation of experimental data.

3. Determination of the preheating of coolant.

$$c_p dT^* = dQ + udu. \qquad (12)$$

The equation of energy in the complete parameters, the differential form and in relative motion has the form:

~~A~~ The quantity of heat, applied to coolant (with flow rate per second ~~of~~ G) through walls, on the elementary section of the length cf channel dz is equal to:

$$dQ_0 = GdQ - k_0(T_r^* - T^*)\,d\bar{z},$$

where

$$k_0 = \frac{l}{\dfrac{1}{\alpha_\text{в} \Pi_\text{в}} + \dfrac{\delta}{\lambda_\text{м} \Pi_{cp}} + \dfrac{1}{\alpha_\text{г} \Pi_\text{г}}};$$

$$\bar{z} = \frac{z}{l};$$

z - coordinate in the direction of a radius; δ is wall thickness; α - heat-transfer coefficient; $\lambda_\text{м}$ - the thermal conductivity of wall; Π - the perimeters of the cross section/~~cut~~ of wall.

By converting equation (12), we will obtain

$$dT^* = \left[k_0 \frac{(T_r^* - T^*)}{G} + (\omega l)^2 (\bar{r} + \bar{z}) \right] \frac{d\bar{z}}{c_p}.$$

Page 9.

At the average/~~mean~~ value of quantities k_0, T_r^* the definite integral of this equation in ~~range~~ interval from 0 to $\bar{z}$ takes the form:

$$\Delta T^* = \left[T_r^* - T_\text{в}^* + \frac{\omega^2 l^2}{c_p k_1} \left(\bar{r}_\text{в} - \frac{1}{k_1} \right) \right] \left(1 - e^{-k_1 \bar{z}} \right) + \frac{\omega^2 l^2 \bar{z}}{c_p k_1}, \quad (13)^{*)}$$

where $\bar{r}_\text{в} = \frac{r_\text{в}}{l}$, $r_\text{в}$ - radius of the inlet ~~entrance~~ of hole; $T_\text{в}^*$ - the stagnation temperature of coolant at the ~~entrance~~ inlet in relative motion:

$$k_1 = \frac{k_0}{Gc_p}.$$

FOOTNOTE 1. In book [8] is given the formula for determining the preheating of coolant in nozzle blades (ω = 0). ENDFOOTNOTE.

4. Determination of the ~~consumption~~ flow rate of coolant.

Outlet velocity from hole is ~~located~~ found ~~through~~ from converted expression (8)

$$w^2_{\text{вых}} = \frac{2\frac{n}{n-1}R(T_{\text{в}} - T_{\text{вых}}) + u^2_{\text{вых}} - u^2_{\text{в}} + w^2_{\text{в}}\left(1 - \zeta_{\text{м}} - \frac{\zeta_{\text{в}}}{2D_{\text{г.в}}}\right)}{1 + \frac{\zeta_{\text{вых}}}{2D_{\text{г.вых}}}}. \quad (14)$$

The drag coefficient of channel with technical roughness with Re numbers > 2300 it is possible to determine from the general-purpose formula of Colbrook and White [6] taking into account an increase by 15o/o in the data [5] ~~during~~ with the rotation

$$\zeta = \frac{1{,}15}{\left[-2\lg\left(\frac{2{,}51}{\mathrm{Re}\sqrt{\zeta}} + \frac{\Delta}{3{,}7D_{\text{г}}}\right)\right]^2}, \quad (15)$$

where Δ is roughness.

Moreover Re it is convenient to calculate according to the formula

$$\mathrm{Re} = \frac{4G}{\Pi\,\eta}, \qquad (16)$$

where η is dynamic viscosity of coolant.

For determining $w_{\text{в}}$ they are assigned at first approximately by the ~~consumption of~~ flow rate coolant and then they find appropriate $p_{\text{в}}$, $T_{\text{в}}$ and $w_{\text{в}}$ through the following equations:

$$p_{\text{в}} = p_{\text{в}}^{*}\left[1 - \frac{w_{\text{в}}^{2}(1+\xi_{\text{в}})}{2\frac{k}{k-1}RT_{\text{в}}^{*}}\right]^{\frac{k}{k-1}}; \qquad (17)$$

$$T_{\text{в}} = T_{\text{в}}^{*} - \frac{w_{\text{в}}^{2}}{\frac{2k}{k-1}R}; \qquad (18)$$

$$w_{\text{в}} = \frac{GRT_{\text{в}}}{p_{\text{в}}F_{\text{в}}}. \qquad (19)$$

Page 10.

The calculation in channel is performed according to sections i. ~~On~~ at output/~~yield~~ from which from equations (6), (7), (13), (14) and the continuity

$$p_{\text{вых}\,i} = \left(\frac{GRT}{wF}\right)_{\text{вых}\,i} \qquad (20)$$

by the method successive approximations ~~they~~ there are determined $p_{\text{вых}\,i}$, $w_{\text{вых}\,i}$, $T_{\text{вых}\,i}$.

At output/~~yield~~ from the last/~~latter~~ section, ~~it~~ there is calculated

$$p_{\text{вых}\,p}=p_{\text{вых}\,i}\left(1-\frac{\xi_{\text{вых}}\,w^{2}_{\text{вых}\,i}}{2\frac{k}{k-1}RT^{*}_{\text{вых}\,i}}\right)^{\frac{k}{k-1}}.$$

The entire calculation is fulfilled also by the method of ~~approach/~~approximations G before ~~agreement~~ of $p_{\text{вых}\,p}$ with $p_{\text{вых}}$ assigned~~/prescribed~~

The calculation is programmed for computation on ~~by~~ ETsVM "Nairi" ~~"press"~~.

The calculations show that the correctly approximate equality

$$\frac{n}{n-1}T_{\text{в}}\left[1-\left(\frac{p_{\text{вых}}}{p_{\text{в}}}\right)^{\frac{n-1}{n}}\right]\approx$$

$$\approx\frac{k}{k-1}(T_{\text{в}}+0{,}5\Delta T^{*})\left[1-\left(\frac{p_{\text{вых}}}{p_{\text{в}}}\right)\right]^{\frac{k-1}{k}},\qquad(21)$$

which is used below.

For the blades, depicted on Fig. 2, due to the presence of many different forms of friction in coolant passage the calculation ~~to~~

conveniently to conduct ~~consecutively~~ sequentially totaling hydraulic losses by sections along channel.

If we write equation (5) in stagnation parameters and ~~to~~ consider relation~~ship/ratio~~ (21), then we will obtain expression for determining the theoretical total pressure p_n^* on the appropriate radius of the blade

$$p_n^* = p_в^* \left[1 + \frac{u^2 - u_в^2}{2\frac{k}{k-1} R\,(T_в + 0{,}5\Delta T^*)}\right]^{\frac{k}{k-1}}, \qquad (22)$$

where ΔT* - the preheating of air on section from the entrance into radial channel before its distribution into transverse channels.

Page 11.

The temperature of stagnation was determined from converted expression (7)

$$T^* = T_в^* + \Delta T^*. \qquad (23)$$

The actual the total pressure at the ~~output/yield on of~~ outlet from each section will be

$$p_{вых}^* = p_в^* - \Delta p^*, \qquad (24)$$

where $\Delta p^* = \xi \frac{\gamma w^2}{2}$.

In handbook [6] the coefficients of losses ξ are given on the basis of experiments with small pressure differentials under isothermal conditions. In long channels with ~~the large~~ great preheating of coolant proceeds a considerable change in the losses and a~~n~~ ~~incidence/~~drop in the total pressure from heat supply. Taking this into account total pitot loss (in the channels of constant cross section) approximately [1] will be

$$\Delta p^* \approx \left(\frac{\xi_{\text{в}}}{2}-1\right)\frac{\gamma_{\text{в}}\, w_{\text{в}}^2}{2}+\left(\frac{\xi_{\text{вых}}}{2}+1\right)\frac{\gamma_{\text{вых}}\, w_{\text{вых}}^2}{2}$$

(for channels $\xi=\zeta\frac{l}{D_{\text{г}}}$).

FOOTNOTE [1]. The more precise value of the total pressure at output~~/yield~~ from section with the changing cross section can be found from equations (14) and (20). With the taken ~~consumption of~~ flow rate G is determined at first $p_{\text{вых}}$ and then $p^*_{\text{вых}}$. ENDFOOTNOTE.

For computation $\frac{\gamma w^2}{2}$ it is convenient to ~~apply~~ use gas-dynamic functions. With the flow rate assigned~~/prescribed~~ in the first approximation, ~~consumption~~ is determined the function $q(\lambda)=\frac{G\sqrt{T^*}}{mFp^*}$ and from tables - λ.

Then ~~is located~~ we find $\frac{\gamma w^2}{2} = \frac{G}{2F} \lambda \sqrt{\frac{2k}{k+1} RT^*}$ and at the selected value ξ is determined Δp^*.

The calculation is fulfilled by the method of successive approximations, changing G before agreement in exit section $\lambda_{\text{вых}}$ from $\pi_2 = \frac{p_{\text{вых}}}{p^*_{\text{вых}}}$ ($p^*_{\text{вых}}$ is obtained by consecutive subtraction of Δp^* sections) and from the expression

$$y(\lambda_{\text{вых}}) = \frac{G\sqrt{T^*_{\text{вых}}}}{mF_{\text{вых}}\, p_{\text{вых}}}.$$

Page 12.

In this case pressure at output/~~yield~~ from hole can be taken as equal to the pressure of gas p_2 at output/~~yield~~ from lattice [1].

FOOTNOTE [1]. Work [2] shows the negligible effect of ejecting effect on trailing edge (from the flow of gas) on the ~~consumption~~ flow rate of air coolant. ENDFOOTNOTE.

The proposed calculation method makes it possible to determine coolant distribution according to span of the blade (Fig. 2).

Example of the calculation.

For ~~testing~~ checking the method of calculation and selection of the loss factors was determined the flow rate of the air coolant through the channels of blades [2]. The loss factors were selected from handbook [6]:

$\xi_0 = 0{,}53 \div 0{,}56$ - **for the holes of the** rim of disk with transition **to deflector;**

$\xi_{\text{д}} = 1{,}58 \div 1{,}66$ - **for a deflector** (taking into account the **separation of flow** on holes on leading edge);

$\xi_{\text{вх}} + \xi_{y} = 1{,}1;$

$\xi_{\text{щ}} = 4{,}76 \div 6{,}42$ - **for the** slots between deflector and wall of blade (was designed by formulas for lamellar-tubular radiators);

$\xi_{\text{см}} = 1{,}14;$

$\xi_{l\,\text{вых}} = 0{,}141 \div 0{,}202;$

$\xi_{\text{вых}} = 0{,}77.$

In this case some geometric data, which are not given in work [2], are ~~undertaked~~ taken approximately from the figure of blade.

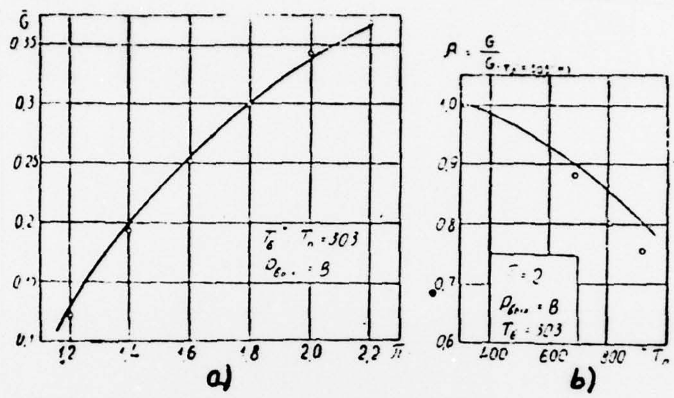


Fig. 3. The dependence of the ~~given~~ normalized air flow rate on the ~~relation~~ ratio of pressures and temperature of the blades: —— - the calculation; O- - according to experiments [3].

Page 13.

The preheating of air ΔT* from the sections of channel was determined from the equations of heat exchange and energy (13).

The results of the calculation ~~in~~ as a function of $\bar{G} = \frac{G\sqrt{T_n^*}}{p_{max}F_{max}}$ on $\pi = \frac{p_0^*}{p_{max}}$ and ~~from~~ on the temperature of blade $T_л$ were given in Fig. 3, where was visible the satisfactory convergence of calculated and experimental data [2].

REFERENCES

1. I. T. Shvets, Ye. I. Dyban. Air cooling of the rotors of gas turbines. Publishing house of Kiev university, 1959.

2. S. Z. Kopelev, S. V. Gurov. Hydraulic friction of the system of air cooling the rotor of gas turbine. (Republic interdepartmental collection). Heat exchange in power plants. Kiev, "Scientific Thought", 1967.

3. S. Z. Kopolev, S. V. Gurov. Study of the hydraulic friction of the cooled blade. "thermal-power engineering", 1966, No.10.

4. V. G. Tyryshkin, M. I. Grunfest, V. A. Mikhaylova. Experimental study of gas-turbine blade with internal air cooling. "power equipment construction", 1967, No. 3.

5. V. V. Mal'tsev. Investigation of radially ventilation Rotors of turbogenerators. The "herald of electrical industry", 1960, No. 8.

6. I. Ye. Idel'chik. Handbook on hydraulic friction. M., Gosenergoizdat, 1960.

7. V. M. Kapinos, A. F. Slitenko. Experimental study of the distribution of pressure on the end/~~face~~ of ~~the~~ a cooled blade. Power machine-building. (Republic interdepartmental scientific collection), iss. 5. Publishing house of Kharkov university, 1967.

8. G. S. Zhiritskiy, V. I. Lokay, M. K. Maksutova, V. A. Strunkin. Gas turbines of aircraft engines. M., Oborongiz, 1963.

Submitted on 28 November 1968.

DISTRIBUTION LIST

DISTRIBUTION DIRECT TO RECIPIENT

ORGANIZATION		MICROFICHE
A205	DMATC	1
A210	DMAAC	2
B344	DIA/RDS-3C	8
C043	USAMIIA	1
C509	BALLISTIC RES LABS	1
C510	AIR MOBILITY R&D LAB/FIO	1
C513	PICATINNY ARSENAL	1
C535	AVIATION SYS COMD	1
[redacted]	[redacted]	[redacted]
C591	FSTC	5
C619	MIA REDSTONE	1
D008	NISC	1
H300	USAICE (USAREUR)	1
P005	ERDA	1
P055	CIA/CRS/ADD/SD	1
NAVORDSTA (50L)		1
NASA/KSI		1
AFIT/LD		1

ORGANIZATION		MICROFICHE
E053	AF/INAKA	1
E017	AF/RDXTR-W	1
E404	AEDC	1
E408	AFWL	1
E410	ADTC	1
E413	ESD	2
	FTD	
	CCN	1
	ETID	3
	NIA/PHS	1
	NICD	5