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Curvature Effects on a Turbine Blade Cooling Film

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Analysis of the equations governing the equilibrium conditions of a cool gas film injected along a curved wall in a hot gas stream shows that Archimedes type forces may either increase or decrease the cooling efficiency of the film, according to the value of the mass flow rate ratio, the temperature ratio of the flows and the sign of the curvature of the wall. Experiments have been performed to verify these predictions: in the case of a concave wall and relatively high mass flow rate ratios, a gain in performance has been obtained, if compared to the constant pressure flat plate; in the case of a convex wall, the cooling efficiency of the film is less than that of a film on a plate with the same pressure gradient, although it is still higher, for the same mass flow ratio than the efficiency of a film on a constant pressure plate.

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INTRODUCTION

Whenever turbine blade cooling by means of a gas film is considered, one always wonders if and how much the blade surface curvature interferes with the efficiency of the process.

The centrifugal forces due to the local curvature of the streamlines are balanced by pressure gradients that build up between suction and pressure sides of a blade channel. But if a gas jet is injected into this equilibrium flow, with a momentum that is not that of the main flow, Archimedes force type forces are induced in the fluid.

If these forces have the effect of detaching the cooling film from the blade walls, a natural convection type motion may take place in the blade channel. This results in an increase of the turbulence level of the flow and decreases the cooling efficiency of the film, due to a rapid mixing of this film with the main flow.

On the other hand, if these Archimedes type forces tend to maintain a long time the film along the wall, the cooling efficiency may be increased, and it will become easier to cool a curved wall than a straight one.

In order to distinguish between the favorable and the unfavorable effects of curvature on the cooling efficiency of a gas film, comparative experiments have to be made, in which the major parameters are maintained at same values.

This paper presents the results obtained in such tests. The main experiments were performed on a curved channel that is a big scale reproduction of a high deflection, high Mach number turbine blade channel. Theoretical calculation of the flow field in this curved channel has shown that the velocity remains almost constant on the pressure side. Therefore, comparison of film cooling efficiency on the pressure side of the turbine blade has to take as reference the constant pressure flow along a flat plate.

On the contrary, along the suction side of the curved channel, the flow is accelerated on the first half of the blade wall and a deceleration takes place on the second half. A similar, though slightly different pressure field, was built up along a flat plate by means of a contoured opposite wall that induces pressure gradients; however, the comparison of the cooling performances of the film along a flat plate (no curvature effects) and a curved wall is more difficult.

Due to the complexity of the aerothermodynamical processes of film cooling (1),[1] a simplified theoretical analysis has been made, in order to define the fundamental parameters of similarity, and also the conditions necessary for comparison of the efficiency of a cooling film along a flat plate and along a curved wall. The first results of such a comparison are given here.

A SIMPLIFIED ANALYSIS OF CURVATURE EFFECTS ON A COOLING FILM

Conservation Equations

In the usual case of the cooling of a gas turbine blade, the mass flow of the film is always very small compared to the main mass flow. This main flow is, therefore, only slightly disturbed; thus, the pressure field along the blade and the pressure gradients can be taken as those of the undisturbed flow. Cooling air injection is made in the vicinity of the blade leading edge at a point where the blade wall boundary layer is still thin and the pressure distribution that will be taken into account is that of an irrotational flow. Since all the experiments described in the following were performed on a non-rotating channel, the theoretical approach will also neglect the effect of the forces due to rotation.

In the local reference frames centered at the center of curvature of a streamline, the x axis being tangent to this streamline, the momentum equations take the following forms:

[1] Underlined numbers in parentheses designate References at end of paper.

NOMENCLATURE

C_o = oxygen mass concentration in main flow (inlet conditions)
C_W = oxygen mass concentration near the wall
c_p = total pressure coefficient
e = characteristic length for heat conduction
g = gravity
h = height of main flow
h_c = height of film
l = length to be protected
m = mass flow rate ratio
p = pressure
P_f = Archimedes force
$(P_f)_o$ = Archimedes force at film injection section
P_{NC} = Archimedes force for natural convection
P_i = total pressure
Pr = Prandtl number of the film
r = radius, radius of curvature
Re = Reynolds number
s = width of the slot
T = temperature of main flow
T' = hot wall temperature (natural convection)
T_a = adiabatic wall temperature
T_c = film temperature
T_W = wall temperature
$\bar{T}$ = mean temperature of main flow
$\bar{T}_c$ = film mean temperature
T_o = main flow inlet temperature
T_{co} = film inlet temperature
V = velocity of main flow
V_c = film velocity
x = abscissa
y = ordinate

Greek letters
α = mass flow ratio (film/main flow)
Δr = deflection of a streamline
$\Delta\rho$ = density difference (natural convection)
ΔT = temperature difference (natural convection)
η = film cooling efficiency
η_c = concentration measured efficiency
ξ = geometrical reduced abscissa
ζ = aerodynamical reduced abscissa
θ = temperature ratio
λ = thermal conductivity
μ = eddy viscosity
ϖ = Archimedes force ratio to dynamic force
ρ = main flow density
ρ_c = film density

$$\left.\begin{aligned} \rho V \frac{\partial V}{\partial x} &= -\frac{\partial p}{\partial x} \\ \rho \frac{V^2}{r} &= -\frac{\partial p}{\partial z} \end{aligned}\right\} \qquad (1)$$

where V is the velocity, p is the pressure, and ρ is the density of the undisturbed flow (Fig. 1).

If the aerothermodynamic parameters of the cooling film in its injection section, velocity V_c, density ρ_c, are different from those of the main flow, the local equilibrium conditions become (2)

$$\left.\begin{aligned} \rho_c V_c \frac{\partial V_c}{\partial x} - \rho V \frac{\partial V}{\partial x} &\simeq \frac{1}{2}\frac{\partial}{\partial z}\left[\mu_c \frac{\partial V_c}{\partial z} - \mu_c \frac{V_c}{r}\right] \\ \rho_c V_c \frac{\partial V_{z,c}}{\partial x} &\simeq \frac{\rho_c V_c^2}{r} - \frac{\rho V^2}{r} \end{aligned}\right\} \qquad (2)$$

where μ_c is the eddy viscosity of the film.

If there is no flow separation near the blade wall, the streamlines and curvature radii are those of the undisturbed flow, and these latter are those of the blade wall itself.

The second equation (2) shows that there exists a buoyancy effect, normal to the streamlines:

$$P_f = \frac{\rho_c V_c^2}{r} - \frac{\rho V^2}{r} = \frac{\rho V^2}{r}\left(\frac{\rho_c V_c^2}{\rho V^2} - 1\right) \qquad (3)$$

and the direction of this thrust depends on the sign of

$$\frac{\rho_c V_c^2}{\rho V^2} - 1 \qquad (4)$$

For positive values of this Archimedes type force, the film is pushed toward a concave wall and is detached from a convex wall. For negative values of equation (4), the effects are inversed.

Near the injection section, the initial value of the Archimedes force is

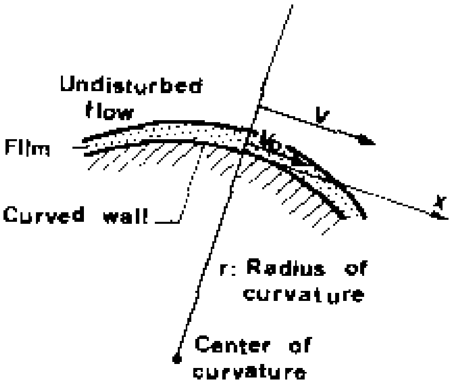

Fig. 1 Schematic representation of a film along a curved wall

$$\left(P_f\right)_0 = \rho \frac{V^2}{2}\left(m^2 \frac{\rho}{\rho_c} - 1\right) \tag{5}$$

where

$$m = \frac{\rho_c V_c}{\rho V} \tag{6}$$

ratio of the mass flow rates in the film and in the main flow appears to be one of the fundamental parameters of film cooling. For a perfect gas, $(P_f)_0$ can also be written

$$\left(P_f\right)_0 = \rho \frac{V^2}{2} \frac{m^2 T_c - T}{T} \tag{7}$$

where T and T_c are the temperatures of the main flow and of the film.

Comparison of the Film Cooling Process and the Natural Convection Process

It is interesting to compare the Archimedes force that appears near a film-cooled wall with the natural convection phenomenon (3). Near a horizontal heated wall, there exists in the gas a thrust

$$P_{nc} = g\,\Delta\rho \tag{8}$$

where g is the gravity and $\Delta\rho$, the difference of density between the undisturbed air and the plate heated air. Since in a constant pressure medium

$$\Delta\rho \cong \rho \frac{\Delta T}{T} \tag{9}$$

where

$$\Delta T = T' - T$$

is the difference between the temperature of the plate-heated gases and the ambient temperature, this natural convection force can be written

$$P_{nc} = \rho g \frac{T' - T}{T} \tag{10}$$

Table 1 gives the correspondence between the various terms of the Archimedes force in the film cooling process along a curved wall and in natural convection phenomena; Table 2 gives some further parameters for the comparison of the film cooling process and the natural convection process.

It appears in Table 2 that the Archimedes force in the film cooling process is some 2.10^5 times larger than the one in the natural convection process. But natural convection effects extend over lengths of several meters and sometimes tens, hundreds of meters, when film cooling acts only on lengths of a few centimeters. (The chord of a turbine blade is maybe 4 cm.) The residence time of the cooling film will be

$$\tau = \frac{\ell}{V} = \frac{0.04}{325} = 1.3 \cdot 10^{-4}\ \text{sec} \tag{11}$$

and during this time, the radial displacement of a cooling film particle will be

$$\Delta r = \frac{P_f}{\rho}\frac{\tau^2}{2} = 15\ \text{mm} \tag{12}$$

It can be seen on the expressions (7) of the Archimedes force and (11) of the residence time that this displacement does not depend on the actual value of the velocity, but on the parameter, m, and on blade geometry only, the relative displacement of the particle

$$\frac{\Delta r}{r} = \frac{P_f}{\rho}\frac{\tau^2}{2r} = \left(\frac{\ell}{r}\right)^2 \frac{m^2 T_c - T}{T} \tag{13}$$

is function of the ratio of the blade chord to the radius of curvature.

Similarity Parameters of the Film Cooling Process

Since a complete theoretical analysis of the film cooling process is not available, the following elementary description is proposed. Only flat plates are considered, since similarity parameters only are researched (Fig. 2).

Heat transfer between the main flow and the film and between the film and the wall is

Table 1 Comparison between Film Cooling and Natural Convection

	Film cooling	Natural convection
Force	$\frac{\rho V^2}{r} \frac{m^2 T_c - T}{T}$	$\rho g \frac{T' - T}{T}$
Gravity	$\frac{V}{r}$	g
Temperature difference	$m^2 T_c - T$	$T' - T$

Table 2 Typical Values of Aerothermodynamical Parameters in Film Cooling and Natural Convection

Cooling air	Velocity of the main flow	$V = 325$ m/sec
	Radius of curvature of the blades	$r = 0.03$ m
	Mass flow rate ratio	$m = 2$
	Centrifugal acceleration	$V^2/r = 3\ 10^6$ m/sec^2
	Archimedes force	$P_f = 2.5\ 10^6$ n/cu m
Natural convection	Temperature difference	$T' - T = 72$ K
	Gravity	$g = 9.81$ m/sec^2
	Archimedes force	$P_{NC} = 2.5$ N/cu m
Ratio of the Archimedes forces		$P_f/P_{NC} = 10^6$

governed by equations of the type

$$\left.\begin{aligned} \rho V c_p \frac{\partial T}{\partial x} &= \lambda \frac{\partial^2 T}{\partial y^2} \\ \rho_c V_c c_p \frac{\partial T_c}{\partial x} &= \lambda \frac{\partial^2 T_c}{\partial y^2} \end{aligned}\right\} \qquad (14)$$

where λ is the eddy heat-transfer coefficient is assumed to take the same value in the main flow, in the vicinity of the film, and in the film itself. If both equations (14) are integrated from the blade surface, where

$$\frac{\partial T_c}{\partial y} = 0$$

toward infinity where again

$$\frac{\partial T}{\partial y} = 0$$

one obtains

$$\left.\begin{aligned} \rho \bar{V} c_p h \frac{\partial \bar{T}}{\partial x} &= -\lambda \left(\frac{\partial T}{\partial y}\right)_f \\ \rho_c V_c c_p h_c \frac{\partial \bar{T}_c}{\partial x} &= \lambda \left(\frac{\partial T}{\partial y}\right)_f \end{aligned}\right\} \qquad (15)$$

where h and h_c are the respective heights of the main flow and of the film, and $\bar{T}$ and $\bar{T}_c$ are the

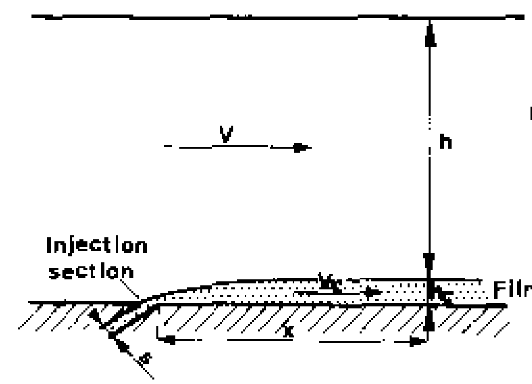

Fig. 2 Schematic representation of a film along a flat plate

corresponding main temperatures, the latter one being very close to the wall temperature. Since temperature gradient $(\partial T/\partial y)_f$ at the film boundary can be assumed to be proportional to the temperature difference

$$\left(\frac{\partial \overline{T}}{\partial y}\right)_f = \frac{\overline{T} - \overline{T}_c}{e} \tag{16}$$

where e is some characteristic length, then one obtains

$$\overline{T}_c = \frac{T_o + \alpha T_{c_o} - (T_o - T_{c_o})\, \exp\left(-\frac{(1+\alpha)\lambda}{\rho_c u_c e c_p}\frac{x}{h_c}\right)}{1+\alpha} \tag{17}$$

where α is the ratio of film mass flow to the main mass flow. Then for a given experimental setup, for which the main mass flow is constant, the following similarity parameters can be used:

1 The mass flow rate ratio already defined

$$m = \frac{\rho_c V_c}{\rho V}$$

2 The temperature ratio

$$\frac{T_{c_o}}{T_o}$$

3 The product of main flow Reynolds number by Prandtl number

$$R_e P_r = \frac{\rho V e}{\mu} \times \frac{\mu c_p}{\lambda} = \frac{\rho V c_p e}{\lambda}$$

4 The distance from injection section divided by the film height and the mass flow rate ratio

$$\zeta = \frac{x}{m h_c}$$

and since the film mass flow is always a small fraction of the main flow, $\alpha \cong 0$, the following equation gives an approximate expression of the cooled wall temperature as function of the non-dimensional parameters defined in the foregoing.

$$\overline{T}_c \cong T_o - (T_o - T_{c_o})\, \exp\left(-\frac{\zeta}{R_e P_r}\right) \tag{18}$$

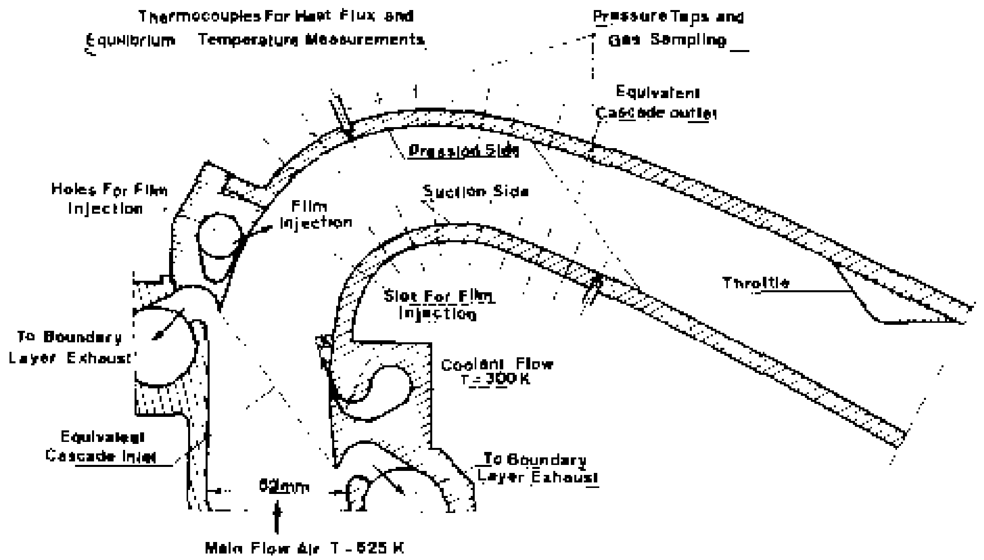

Fig. 3 Curved channel for film cooling experiments

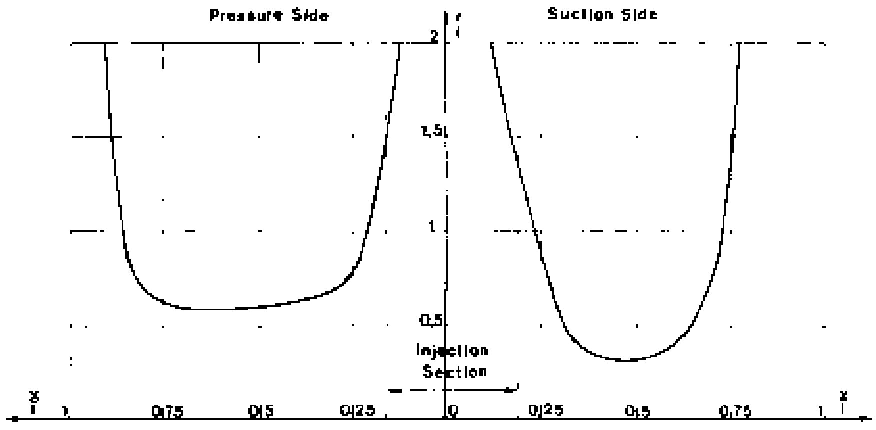

Fig. 4 Nondimensional radius of curvature of a typical turbine blade (4)

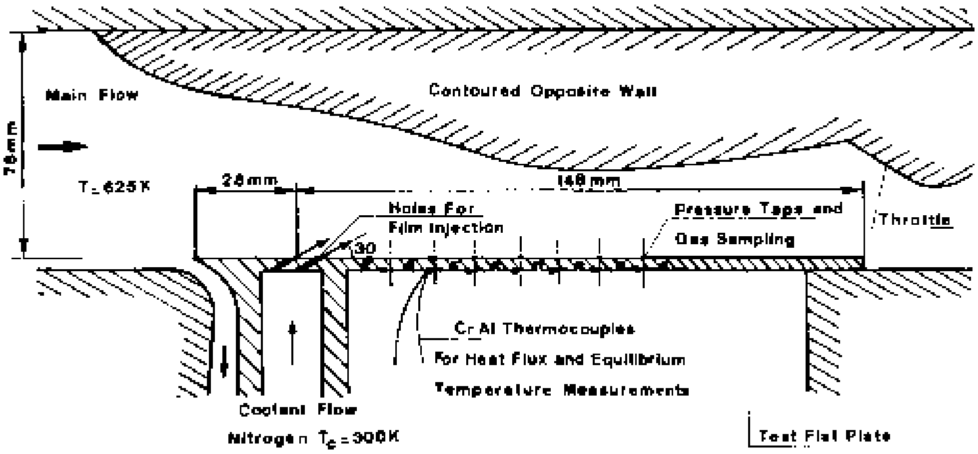

Fig. 5 Reference flat plate for film cooling experiments

Efficiency of the Film Cooling Process

Usually, the efficiency of the film cooling process is defined by

$$\eta = \frac{T_a - T_w}{T_a - T_{co}} \tag{19}$$

where:

T_a = adiabatic temperature of the wall without film cooling
T_w = actual wall temperature ($T_w \cong \bar{T}_c$)
T_{co} = temperature of the film in inlet conditions.

From equation (18), the film cooling efficiency takes the following approximate value

$$\eta \cong \exp\left(-\frac{\zeta}{Re_c\, \mathcal{R}}\right) \tag{20}$$

in which the role of the various nondimensional parameters defined in the foregoing appears clearly.

EXPERIMENTAL RESEARCH

Test Facilities and Equipment

The experimental study of the effect of blade surface curvature on the cooling efficiency of thin films has been performed: (a) on a curved channel representing a turbine blade channel, and

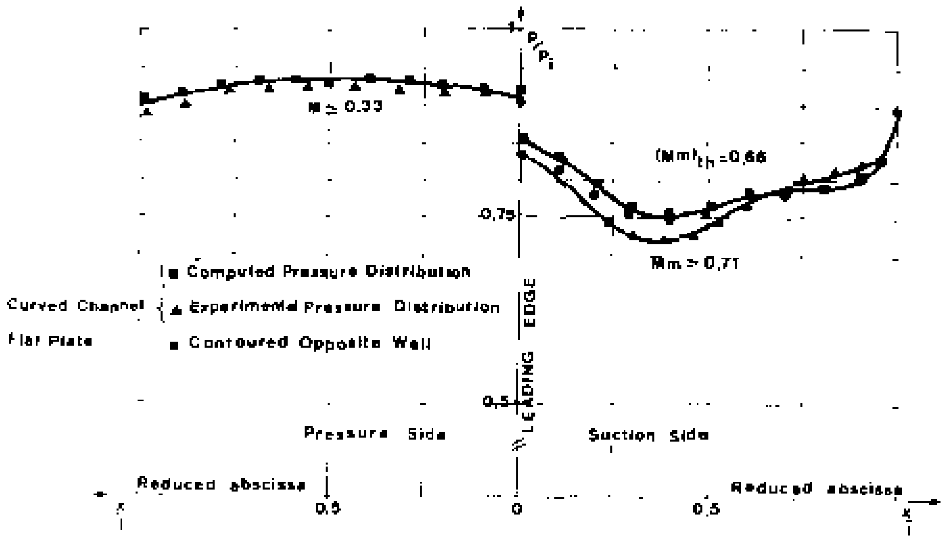

Fig. 6 Suction side and pressure side pressure distributions on the walls of a curved channel and on a flat plate with contoured opposite wall (M_o = 0.40)

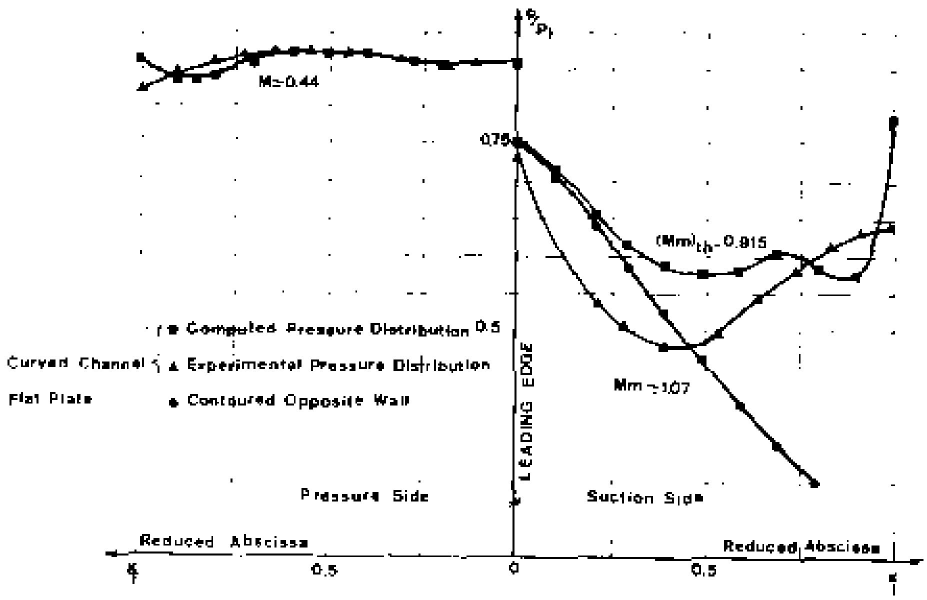

Fig. 7 Suction side and pressure side pressure distributions on the walls of a curved channel and on a flat plate with contoured opposite wall (M_o = 0.53)

(b) on a reference flat plate along which similar pressure gradients were imposed.

The curved channel (Fig. 3) represents, at a larger scale, the actual blade channel corresponding to an analytically defined high deflection, high Mach number turbine blading (4).

The evolution of the nondimensional radius of curvature r/ℓ, where ℓ is the length measured along the wall between the film-injection section and the trailing edge is plotted in Fig. 4 as a function of the nondimensional abscissa, $\xi = x/\ell$, where the origin of x is also at the injection section.

A leading edge has been conserved both for

Table 3 Aerothermodynamic Parameters of a Typical Film-Cooled Turbine Blade Flow and Laboratory Conditions: Curved Channel and Flat Plate

	Turbine		Laboratory tests: Curved Channel		Laboratory tests: Flat Plate	
Flows	Main	Film	Main	Film	Main	Film
Gas	Burned gases	Air	Air	Nitrogen	Air	Nitrogen
Temperature (deg K)	1500	700	625	300	625	300
Pressure (atm)	13	20	2	2.3	2	2.3
Flow rate (Kg/m^2-sec)	1300	800 to 2600	240	240 to 480	240	240 to 480
Mass flow rate rate ratio	0.6 to 2		1 to 2		1 to 2	
Chord Reynolds number	10^6		10^6		10^6	
Film Reynolds number	1 to 3 10^4		1 to 3 10^4		1 to 3 10^4	
Turbulence level	8%		5%		5%	

the suction and pressure side of the curved channel and the wall boundary layer is sucked out through slots upstream of these leading edges in order to establish along the test section conditions similar to those on an actual blade. The nondimensional distance between leading edge and film injection section is: (a) $\xi_i = x_i/\ell = 0.15$ for the pressure side and (b) $\xi_i = x_i/\ell = 0.22$ for the suction side.

The reference flat plate (Fig. 5) is mounted in a parallel side wall channel. The wall opposite to the test plate is either straight, and gives then a constant pressure flow or contoured in order to induce along the flat plate a given pressure distribution.

Figs. 6 and 7 compare the pressure distributions obtained on the curved channel pressure and suction sides to the pressure distributions realized on the flat plate in the corresponding conditions and also to the pressure distributions calculated by means of a direct numerical method.

At the relatively low inlet Mach number, M_0 = 0.40 (Fig. 6), computed and measured values are close together; the maximum measured suction side Mach number is M_m = 0.71 for a predicted maximum Mach number $(M_m)_{th}$ = 0.66 and the contoured opposite wall gives a correct pressure distribution along the flat plate; on the pressure side, a nearly uniform Mach number M = 0.33 is measured and computed.

At the higher Mach number M_0 = 0.53 (Fig. 7), corresponding to more realistic turbine data, transonic operation appears on the suction side, though the computed pressure distribution on suction side shows only a high subsonic maximum Mach number, $(M_m)_{th}$ = 0.915, experimental pressure measurements show a local supersonic region with a maximum Mach number, M_m = 1.07, terminated by a shock wave. In this case, the pressure distribution obtained on the flat plate with the contoured opposite wall is continuously decreasing with a supersonic outlet; no experiments were

Table 4 Expected Effect of Mass Flow Rate Ratio and Blade Curvature on a Film Along a Curved Wall

Mass flow rate ratio	$\bar{m}$	Pressure Side	Suction Side
1	-0.5	Film separation from the wall	Film pushed toward the wall
1.5	0	no Archimedes force	
2	2	Film pushed toward the wall	Film separation from the wall

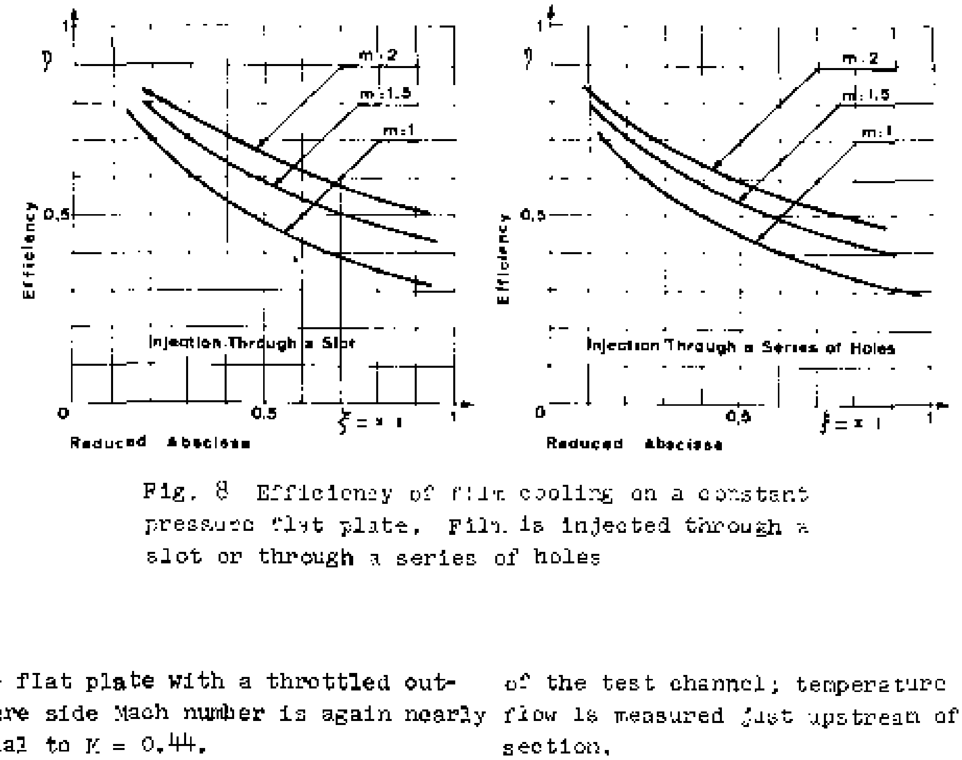

Fig. 8 Efficiency of film cooling on a constant pressure flat plate. Film is injected through a slot or through a series of holes

performed on the flat plate with a throttled outlet. The pressure side Mach number is again nearly constant and equal to $M = 0.44$.

Main Flow and Cooling Gas. The main flow is a heat exchanger heated air at $T_0 = 623$ K and at a pressure of approximately 2 atm; the cooling gas is nitrogen at room temperature and a slightly higher pressure. Injection of the cooling gas is made either to a slot or through series of holes.

The slot has a δ - 1 mm width, and injection angle of the film is 30 deg; the injection holes are distributed on three equidistant rows along an alternative pattern; each hole has a 1 mm dia, and the overall flow passage is equivalent to a 1.13-mm slot.

The main flow, the bypassed boundary-layer flow, and the coolant flow are separately measured by means of sonic throat flow meters. Temperature of the main flow is measured upstream of the test channel; temperature of the coolant flow is measured just upstream of the injection section.

Wall Temperature and Pressure Measurements, Gas Sampling. Series of flux meters are distributed along the channel walls and also along the flat plate; they give the local heat-transfer coefficient and the equilibrium wall temperature, that in the case of no-film injection is the adiabatic wall temperature.

Pressure taps give the wall pressure distribution, and they can also be used for boundary-layer gas sampling. These are 0.4-mm-dia pressure taps, and for gas sampling, the boundary layer is exhausted by means of a vacuum pump and an oxygen analyser gives directly the oxygen content of the gas layer near the wall.

Film cooling efficiency is defined either by means of temperature measurements, equation

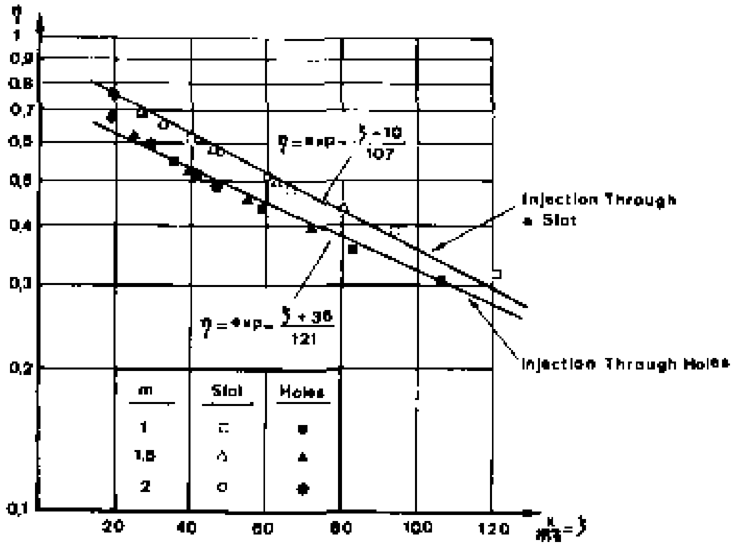

Fig. 9 Correlation of film cooling efficiency on a constant pressure plate

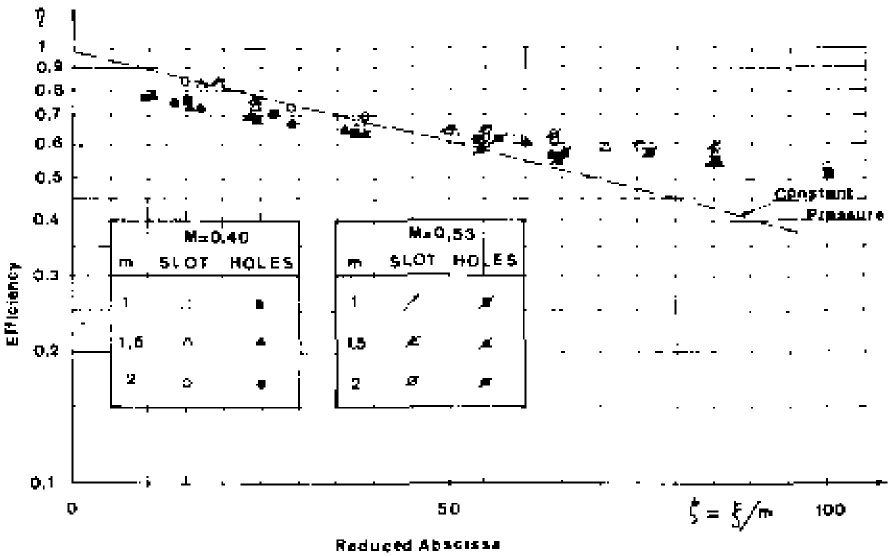

Fig. 10 Efficiency of film cooling on a flat plate with a positive and negative pressure gradient (M_o = 0.40) or a purely negative pressure gradient (M_o = 0.53)

(19), or from the oxygen concentration near the wall.

In the first case, corrections have to be made to take into account the longitudinal heat conduction; this phenomenon is largely reduced by using heat flux meters instead of wall thermocouples. In most cases, this correction is negligible.

The determination of film cooling efficiency based on concentration measurements uses the Reynolds analogy, assuming, as it is the case in the low temperature range of the experiments, that the Lewis number equals 1. Then film cooling efficiency is defined by

$$\eta_c = \frac{C_o - C_w}{C_o} \qquad (21)$$

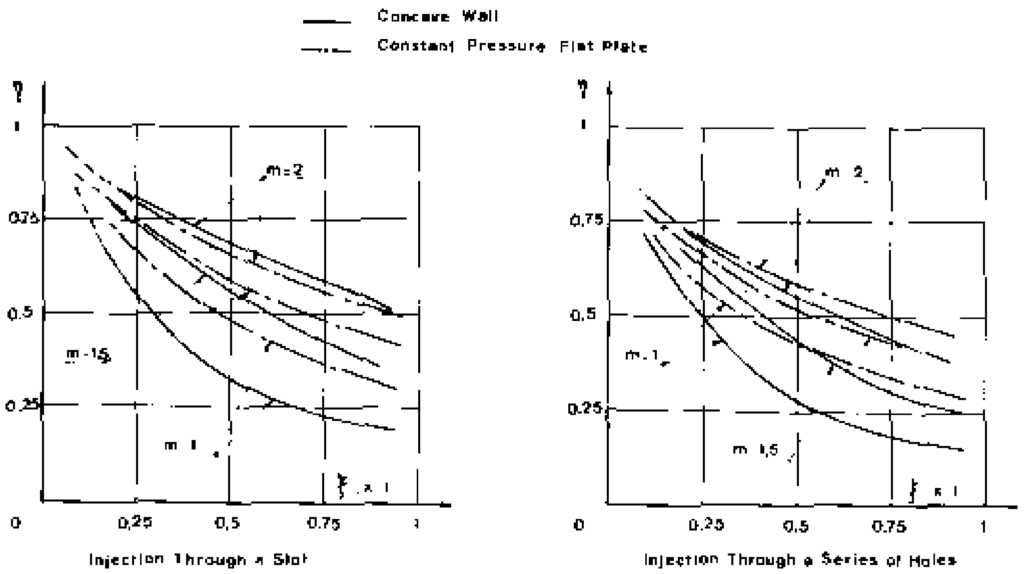

Fig. 11 Comparison of film cooling efficiency on a flat plate at constant pressure and on a concave wall

where $C_0 = 0.23$ is the oxygen mass concentration in air, and C_w, the oxygen concentration of the film near the wall, is zero at the inlet injection section. Precautions have to be taken to suck just the amount of film necessary for concentration measurements, and it was often verified that both techniques give the same value of film cooling efficiency.

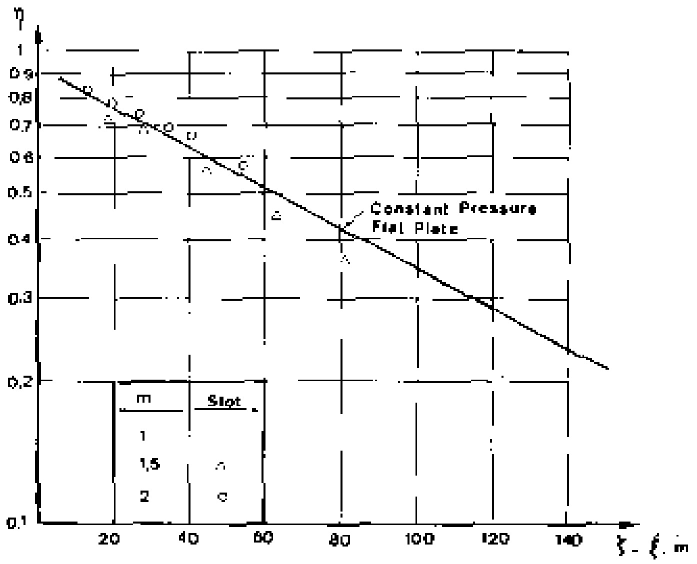

Fig. 12 Correlation of film cooling efficiency on a constant pressure curved surface (injection through a slot)

Aerothermodynamic Parameters of the Experiments

The main aerothermodynamic parameters defining the test conditions are given in Table 3, both for the curved channel and the reference flat plate. In order to show the representativity of these laboratory tests, typical cooled turbine blade conditions are also given in Table 3.

The expected effect of mass flow rate ratio and local curvature, characterized by the nondimensional parameter, r/ℓ, is summarized by the ratio of Archimedes pressure to dynamic pressure

$$\varpi = \frac{P_{A_o}}{\frac{1}{2}\rho V^2}\left(\frac{\ell}{r}\right) = \frac{m^2 T_c - T}{T}$$

the typical values of which are plotted in Table 4.

Remarks: (a) Effect of inlet boundary-layer sucking was tested in auxiliary experiments; the sucked boundary-layer mass flow was slightly modified (± 10 percent) without any substantial change in the efficiency of the cooling process; and (b) the inlet turbulence level was decreased to 1.5 percent without change in efficiency.

Test Results

Basic Results on a Flat Plate. In order to separate the effects of the various parameters on film cooling efficiency, experiments were made on the flat plate with straight opposite wall (constant pressure) and with a contoured wall (negative and positive pressure gradients).

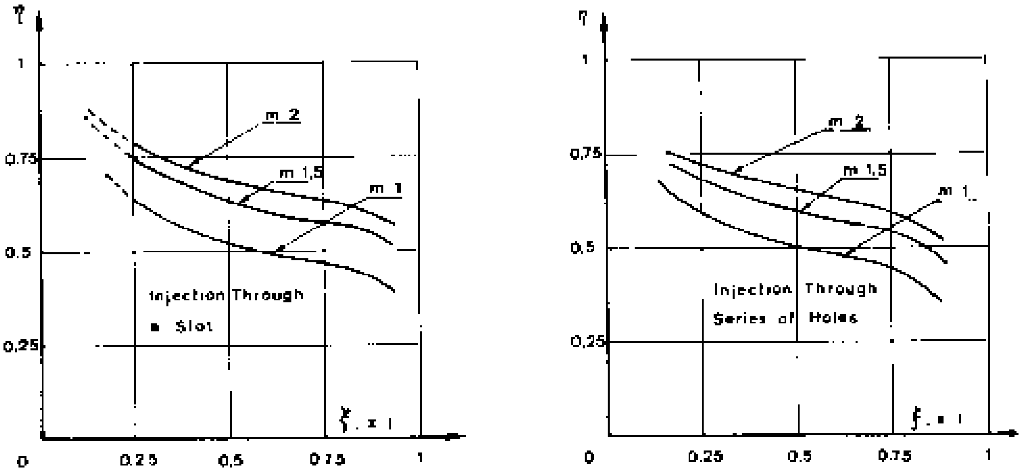

Fig. 13 Film cooling efficiency on a convex wall ($M_o = 0.55$)

Fig. 8 gives the variation of film cooling efficiency as function of geometrically reduced abscissa

$$\xi = \frac{x}{\ell}$$

in the case of the constant pressure operation: ℓ is the length of the plate downstream of the film-injection section, practically ℓ is the length to be protected. Injection was made through a slot and through series of holes, with, in both cases, the mass flow rate ratios

$$m = \frac{\rho_c V_c}{\rho V} = 1 - 1.5 - 2$$

As expected, efficiency is an increasing function of mass flow rate ratio and a decreasing function of abscissa.

In order to check formulas of the type (20), a plot of efficiency versus the aerothermodynamically reduced parameter

$$\zeta = \frac{x}{m s}$$

was made on a semi-logarithmic diagram (Fig. 9): In the case of a gas film-cooled constant pressure plate, the following correlations were found:

$$\left.\begin{array}{ll} \textbf{injection through a slot} & \eta = \exp - \dfrac{\zeta + 10}{107} \\ \textbf{injection through series of holes} & \eta = \exp - \dfrac{\zeta + 36}{121} \end{array}\right| \quad (22)$$

Comparison of the experimental correlations (22) with the predicted efficiency, equation (20), shows that the film acts as if it would be injected at some upstream abscissa. The length necessary for the reorganization of an obliquely injected film is shorter for a slot than for a series of holes. It has, of course, to be understood that the values shown on correlation (22) are specific of the experiments described in the foregoing, but they give a correct idea of the general trend.

The effect of a negative pressure gradient on the first half of a flat plate is shown in Fig. 10. Due to this favorable pressure gradient, the thermal protection is maintained further downstream, and injection through the holes gives

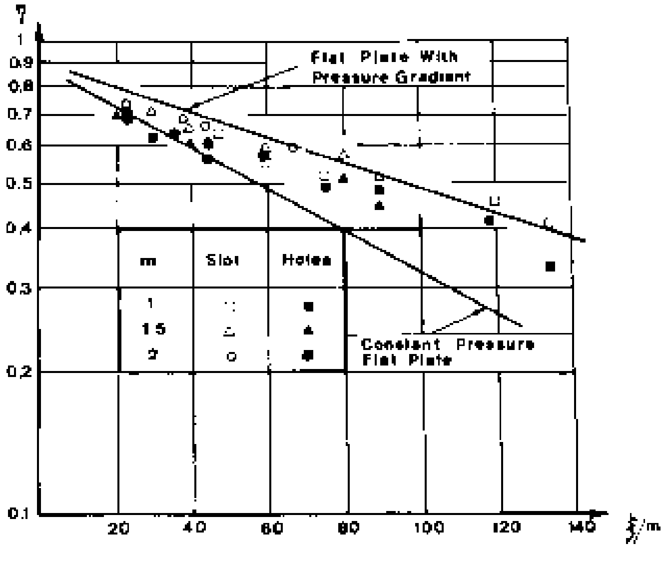

Fig. 14 Comparison of film cooling efficiency on a convex wall and on a flat plate

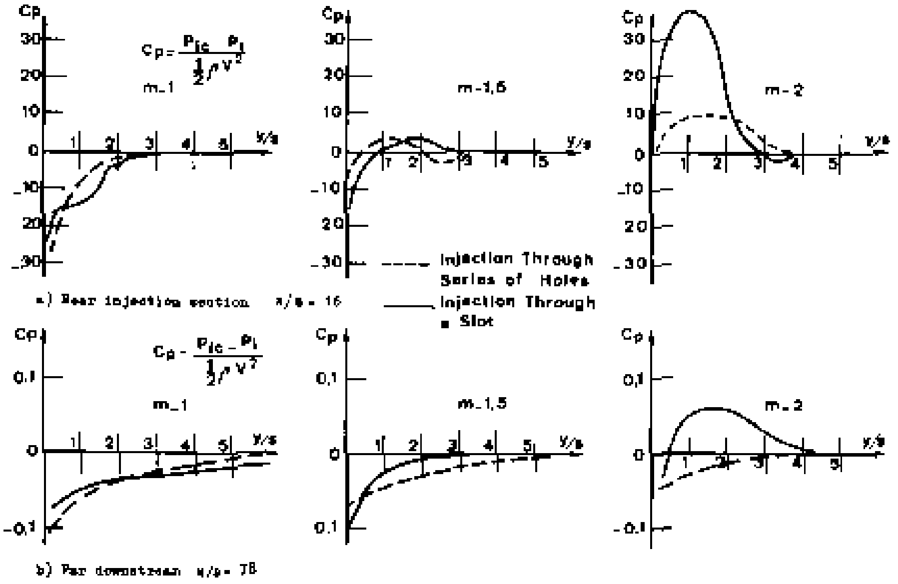

Fig. 15 Decay of total pressure difference between film and main flow

practically the same efficiency than injection through a slot. Fig. 10 shows also the negligible effect of Mach number on cooling efficiency (the experiments were performed at a constant value of the main flow mass flow rate), though, as has been shown (Fig. 6), there exists for the lower Mach number, M = 0.40, a positive pressure gradient which is obtained on the rear part of the plate, when at M = 0.55 (Fig. 7), the pressure gradient is negative all along the plate. One concludes, therefore, that the favorable effect of negative pressure gradient on the film overcompensates the effect of the adverse pressure gradient due to flow deceleration.

Film Cooling Efficiency on Pressure Side. The constant pressure distribution measured on the channel pressure side should give results similar to those obtained on the flat plate with straight opposite wall (constant pressure flow).

Fig. 11 compares the film cooling efficiency as function of reduced abscissa, $\xi = x/\ell$, on the flat plate and on the pressure side of the curved channel:

1 The film injected through a slot gives at mass flow rate ratio, m = 2, an efficiency higher than the one obtained on a flat plate; at m = 1.5, the results are equivalent on a flat plate and on a constant pressure curved surface near the leading edge; at m = 1, the film cooling efficiency on a flat plate is higher.

2 In the experiments with injection through a series of holes, no gain in efficiency was observed due probably to the downstream mixing of the discrete jets that gives supplementary pressure losses. It is thought however, that at higher values of the mass flow rate ratio, m, the efficiency of the film on the constant pressure curved surface might be higher than on the flat plate, even when the film is injected through a series of holes, but such an operation may not be realistic for actual turbine blade pressure side protection.

Correlating the experimental results on the semi-logarithmic plot shows the marked advantage of film blade cooling on a constant pressure curved surface compared to that on a flat plate (Fig. 12).

Film Cooling Efficiency on Suction Side. The favorable effect due to the negative pressure gradient on the first half of the curved channel suction side improves the efficiency of the film (Fig. 13). However, the expected change due to buoyancy that would give higher efficiencies at low mass flow rates, according to Table 4, was not obtained experimentally: The film cooling efficiency on the suction side of the curved channel is higher than on a flat plate in a constant pressure flow, and increases steadily with the mass flow rate ratio, m (Fig. 14).

Pressure and Temperature Profiles Near the Wall. In order to analyze more in detail the cooling process using a gas film, total pressure and total temperature measurements were made near the

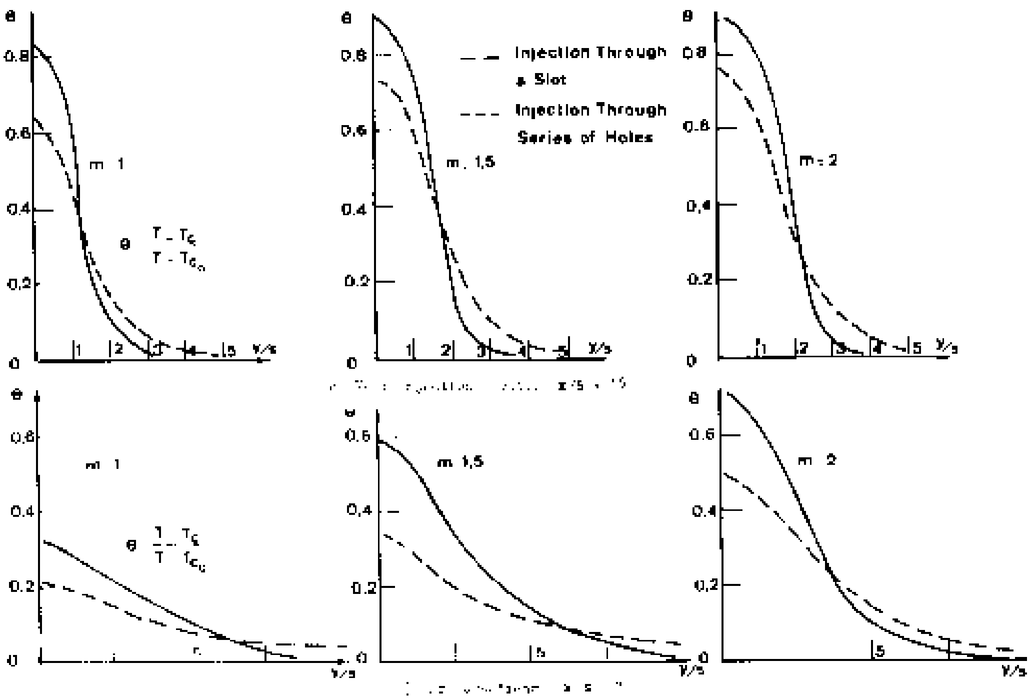

Fig. 16 Decay of the temperature profile and mixing of film and main flow

wall, in the vicinity of the injection section (reduced abscissa, $x/s = 16$) and further downstream, ($x/s = 18$).

Fig. 15 shows the nondimensional parameter

$$C_p = \frac{P_{1c} - P_1}{\frac{1}{2}\rho V^2}$$

where:

P_1 = main flow total pressure,
P_{1c} = film total pressure
$1/2\rho V^2$ = dynamic pressure of main flow, function of reduced ordinate

$$\frac{y}{s}$$

where y is measured normal to the wall. The difference in pressure loss when slot or holes are used appears clearly on the diagram and explains the corresponding loss in efficiency. Since the effect of injection pressure losses decreases further downstream (Fig. 15), the difference of efficiency between film injected through a slot and film injected through series of holes decreases also.

It is interesting to notice that the afore-defined parameter, C_p, is proportional to the Archimedes force.

Fig. 16 gives the temperature profile near the wall

$$\theta = \frac{T - T_c}{T - T_{c_0}}$$

(On the wall itself, parameter θ is equivalent to film cooling efficiency.) The change in rate of decay of the temperature profile with the value of m shows the fundamental role played by this parameter on film cooling efficiency.

CONCLUDING REMARKS

Comparison of film cooling efficiency on a flat plate, with or without pressure gradients, and on a curved surface (pressure or suction side of a blade channel) shows:

1 Cooling efficiency of a slot-injected film on nearly constant pressure side of the channel is equivalent at moderate mass flow rate ratio m = 1.5 to the efficiency on a constant pressure flat plate; it becomes better than on a flat plate at higher mass flow ratios (m = 2) and less good at lower values of m (m = 1). Efficiency of a film injected through a series of holes is slightly less good.

2 Mach number variation does not affect the

efficiency of the film-cooling process on the pressure side, at least in the moderate subsonic range ($M_D = 0.40$).

3 At the same mass flow rate ratio, the efficiency of a film injected through a series of holes is less than the efficiency of the film injected through a slot (experiments performed at constant main flow).

4 On the suction side of the channel, the efficiency of the film is always better than on a constant pressure flat plate. This efficiency is not diminished by a positive pressure gradient near the trailing edge.

As a conclusion, it can be stated that moderate amounts of cool gas can give a correct temperature protection of curved walls such as turbine blade pressure or suction sides, if well-designed film injection systems are used.

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