EFFICIENCY OF FILM COOLING IN A CURVILINEAR CHANNEL BETWEEN GUIDE BLADES

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Results of an experimental study are analyzed which concern the efficiency of film cooling at the end wall of a curvilinear channel between guide blades in a turbine.

Film cooling has recently found more engineering applications in areas where structural components of equipment must be protected against the effects of high temperatures and chemically aggressive media.

Many experimental and theoretical studies have been made [1-4] concerning the efficiency of film cooling under ideal conditions, i.e., with uniform parameter distributions in both the main stream and the injected stream at the entrance section while there are no external forces acting. The basic performance indicator which could be evaluated under such conditions was the efficiency of film cooling defined as

$$\eta = \frac{T_0 - T_{a.w.}}{T_0 - T_s}. \tag{1}$$

In all these studies the efficiency of film cooling was measured under so-called ideal conditions, and most authors singled out three zones: an initial zone, a transition zone, and a main zone, with the efficiency being a different function of m, Θ, Re_s, and x/s in each. Only a few studies [5-10] have dealt with the effect of a longitudinal pressure gradient on the cooling efficiency, as such a gradient may cause both streams in actual turbines to mix during film cooling.

The authors here have attempted to obtain test data on the efficiency of film cooling at the end wall of a curvilinear channel between guide blades, i.e., when all factors which characterize the conditions in actual turbo-machinery prevail almost simultaneously.

The tests were performed on an experimental turbine and the cooling efficiency was measured at the end wall of the passage between two blades of the first stage. In one channel we measured the temperature of the medium in the boundary layer and in another channel we measured the surface temperature at the blade hubs. The thermocouples were located as shown schematically in Fig. 1. We also measured the temperature of the metal at points inside the blade hubs, in order to check the heat dissipation by conductive heat transfer into the stator. The measurements were made by means of Chromel–Alumel thermocouples with wires 0.2 mm in diameter. The thermal emf was recorded by model ÉPP-09 (class 0.5) potentiometers, yielding a final measurement accuracy within 3°C and a determination of the cooling efficiency at the end of the main zone within an accuracy of approximately 8%.

The velocity of the injected stream was determined from readings of a total-pressure Pitor tube, with the inlet orifice in the throat section of the passage and with the pressure pickup at the channel wall also located in the exit section of the passage. The mainstream velocity was measured in terms of mean values, on the basis of the flow rate, and referred to the swept area of the turbine at a section corresponding to the said passage section.

The tests were performed with either forward or reverse heat flow, with specially electrically preheated air injection into the cold mainstream. Such an injection was effected through a tangential orifice

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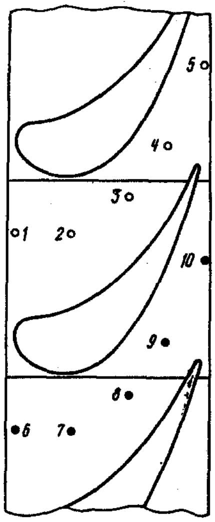

Fig. 1. Schematic diagram indicating the locations of thermocouples in the test channels: 1-5) temperature measurements in the boundary layer of the medium; 6-10) temperature measurements at the surface to be protected.

[5, 10] of a special design. The orifice height was 1.6 mm for a cooled channel with a maximum median length of 61 mm.

The basic parameters in these tests were varied over the following ranges: $Re_s = (1.2\text{–}2.6) \cdot 10^3$, $m = 0.33\text{–}1.0$, and $\Theta = 0.65\text{–}1.3$.

The array of guide blades and the end walls of the passages set up for these tests were operated as follows: $T_0 = 300\text{–}770°K$, $T_s = 420\text{–}520°K$, $P_0/P_1 \approx 1.1$, with the exit angle 19°40', and with the stream accelerated through the array from 30-50 to 170-190 m/sec. For this reason, the results from [10] were not used for evaluating our test data.

A preliminary evaluation of our test data was made in η; $A_2 = m^{-1.25}\, Re_s^{-0.25}(x/s)$ coordinates. In order to calculate η, we replaced $T_{a.w}$ in (1) by the temperature of the medium which had been measured

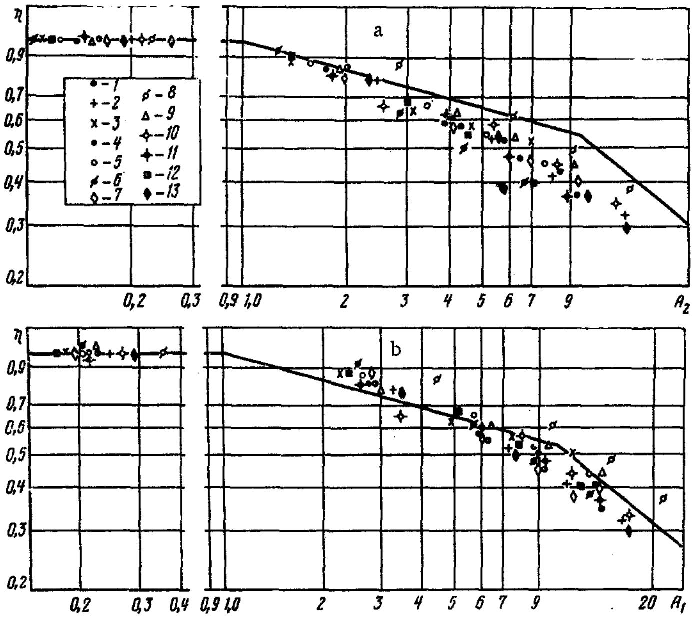

Fig. 2. Efficiency η as a function of (s) the parameter A_2 and (b) the parameter A_1: 1) $m = 0.79$, $Re_s = 1.2 \cdot 10^3$, $\Theta = 0.76$; 2) respectively 0.61, $1.2 \cdot 10^3$, 0.81; 3) 0.95, $1.1 \cdot 10^3$, 0.7; 4) 0.88, $0.94 \cdot 10^3$, 0.73; 5) 0.92, $0.94 \cdot 10^3$, 0.68; 6) 1.0, $1.4 \cdot 10^3$, 0.81; 7) 0.74, $2.6 \cdot 10^3$, 1.3; 8) 0.37, $1.4 \cdot 10^3$, 0.71; 9) 0.73, $1.24 \cdot 10^3$, 0.75; 10) 0.62, $1.0 \cdot 10^3$, 0.79; 11) 0.79, $1.0 \cdot 10^3$, 0.74; 12) 0.99, $1.0 \cdot 10^3$, 0.67; 13) 0.61, $2.6 \cdot 10^3$, 1.26.

in the boundary layer. The results of such an evaluation are shown in Fig. 2a. The solid line represents the efficiency of film cooling under ideal conditions [5]. The maximum departure of test points from this curve at the end of the initial zone ($A_2 \approx 10$) is here 17% (absolute).

As has been mentioned earlier, the general relation for the cooling efficiency is $\eta = f(Re_S, m, \Theta, x/s)$, with the temperature parameter Θ characterizing the effect of the temperature difference between streams. The effect of Θ on the cooling efficiency was studied in [1] over a narrow range of this parameter, while in [5] Θ was calculated according to the recommendations made in [11]. In our study here we attempted to generalize the test data in the form $\eta = f(A_1)$ with $A_1 = m^{-1.25} Re_S^{-0.25} \Theta^{-1.25} (x/s)$, i.e., following the suggestions made in [5, 11], although the range of Θ variation in our tests was much wider than in those other studies.

The results of such an evaluation (Fig. 2b) show a satisfactory correspondence between test data and the efficiency curve based on ideal conditions. For the end of the transition zone and for the main zone, for instance, our values of η lie only somewhat below (5-8%) that curve. It is to be noted that we replaced Θ in A_1 by Θ^{-1} for the evaluation of test data pertaining to reverse heat flow (film heating), because increasing the parameter which characterizes the rate of heat transfer from injected stream to mainstream could not possibly raise the cooling efficiency. The more the temperatures of both streams differ, obviously the stronger will the streams affect one another and the faster will the temperature change in the boundary layer.

The satisfactory agreement between test data on the efficiency of film cooling and the curve obtained under ideal conditions (Fig. 2b) allows us to use the latter for estimating the cooling efficiency at the end wall of passages in turbomachinery. We must point out the necessity, however, of careful and separate further studies concerning the effect of Θ and the combined effect of longitudinal and transverse pressure gradients on the cooling efficiency over a wide range of all those parameters.

NOTATION

T	is the temperature, °K;
$m = \rho_S u_S / \rho_0 u_0$	is the ratio of injection mass flow rate to mainstream mass flow rate;
ρ	is the density, kg/m^3;
u	is the stream velocity, m/sec;
s	is the passage height, m;
x	is the distance from the injection point, m;
$Re_S = u_0 s \rho_0 / \mu_0$	is the Reynolds number referred to the mainstream parameters and to the passage height;
μ	is the dynamic viscosity, $N \cdot sec/m^2$;
$\Theta = T_S / T_0$	is the temperature parameter.

Subscripts

0	denotes the mainstream;
s	denotes the injected stream;
a.w	denotes the wall under adiabatic conditions.

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