STUDY OF THE HEAT TRANSFER DURING FILM COOLING

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An analysis is made of the results obtained in an experimental study of heat transfer in both the main and the transition stage of film cooling a plate.

Film cooling is lately used more often in various branches of industry as a protection of apparatus components exposed to high temperatures.

It is to be noted that numerous studies, e.g., [1-3] have been made to determine the efficiency of film cooling when the cooling medium is injected through an orifice parallel to the protected surface. At the same time, only a few studies [4-6] really have been concerned with the heat transfer during film cooling. The data shown in these references indicate that it may be possible to calculate the heat-transfer coefficient during film cooling at a far distance from the point of coolant injection ($x/s > 70$), using the relations based on an analysis of the heat transfer during the flow of a turbulent stream over a plate. In this case the heat-transfer coefficient is calculated from the difference between the actual wall temperature and the equilibrium temperature, the latter being equal to the wall temperature under adiabatic conditions. We will show that for parts of the surface far removed from the point of injection one may use the data in [7-10], which pertain to the heat transfer at surfaces located behind an interposed heat-exchanger section or behind a porous partition through which the coolant is passed. It is not correct to use the results of these studies for both the initial and the transition stage of the film cooling process, because the flow patterns in these regions are very different then. During the transition stage of film cooling there occurs a gradual modification of the velocity profile from one which is characteristic of a jet along the wall and thus peculiar to the initial stage to one which corresponds to a regular turbulent boundary layer in the main stage. With an interposed heat exchanger, the velocity profile across the entire protected surface is that of a regular turbulent boundary layer. An injection of coolant through a porous partition at low velocities also produces some distortion of the regular velocity profile. Large quantities of air injected through the porous partition squeeze out the boundary layer completely, which is also very much unlike the flow patterns in both the initial and the transition stage of film cooling [10]. It is to be noted that in this case, too, a calculation of the heat-transfer coefficients from the relations of flow over a flat plate is hardly reasonable if applied to surface areas immediately behind the porous partition.

The purpose of the study presented here was to obtain generalizing relations for the heat transfer in both the main and the transition stage under zero-gradient conditions in both the main and the injected stream.

These problems were studied on a test stand consisting of an aerodynamic open channel with an air conditioning system (a receiver, a honeycomb moderator, and a mixer), the active segment equipped with measuring instruments, and electric heaters with a semiautomatic system for stabilizing the temperatures of both the main and the injected stream. The active segment was rectangular in cross section, 150×120 mm^2. The preheated air was injected through a special orifice panel made of asbestos cement. The orifice height was 3 mm. The orifice distributor cap was made of Textolite 1 mm thick and tapered to 0.1 mm at the end. In front of the orifice panel, at a 0.05 m distance, was placed a suction chamber to take in the boundary layer. Behind the orifice panel was placed a 0.61 m long measuring membrane made of asbestos cement. The tests were performed with hot air injected into a cold main stream, which is

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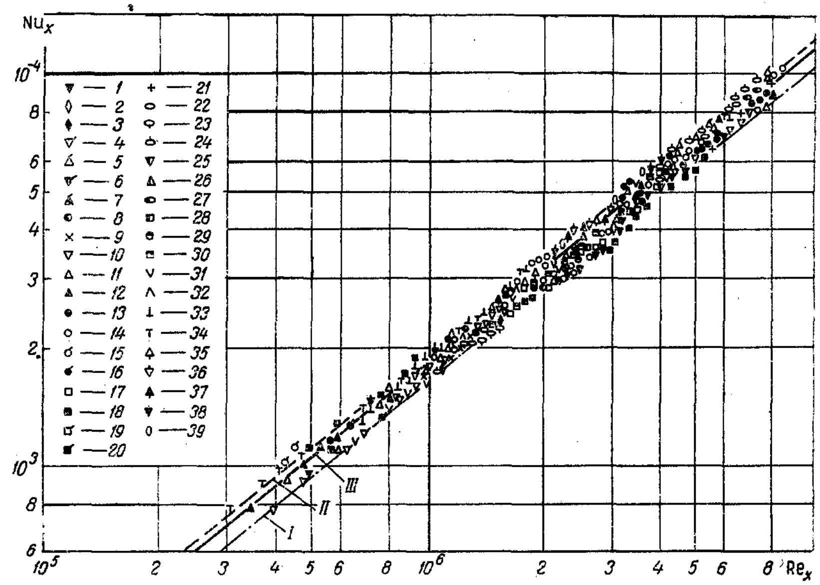

Fig. 1. Heat transfer in the main stage: I) according to (2); II) according to Eq. (3); III) according to Eq. (4). Legend is given in Table 1.

permissible because of the film heating and the film cooling curves being identical when the temperature difference between both streams is small.

Data on the heat transfer during film cooling were obtained by the electrical calorimeter method, which essentially reduces to a comparison between the quantity of heat generated by conversion of electrical energy to thermal energy on the heater plates and the quantity of heat transmitted convectively to the stream under study plus the quantity of heat leaked conductively through the insulation. As heaters for this study were used three strips of stainless steel 1.8 cm wide and 0.1 mm thick, two of which served for compensation and for reducing the lateral heat leakage to zero. In order to prevent temperature distortions in the heaters due to the heat conducted away to the heat sinks, the heater plates were made longer than the test plate. The heater plates were bonded to the test plate with a special adhesive. The heaters were energized from the 220 V ac line through a voltage stabilizer and a stepdown power transformer for supplying 40-50 A at 12 V. For a smooth regulation of the heater power, a voltage regulator was connected in series with the primary winding of the transformer. The temperature of the heater plates was recorded by means of Chromel–Copel thermocouples and electrodes 0.2 mm in diameter welded with a capacitor welder to the heater plates on the inactive side behind the bonded area. Operating on alternating current was proper, because the thermocouple emfs were measured by the potentiometer with a model R-330 instrument. The circuit for measuring the power supplied to the heaters was made up of an instrument-type current transformer, a class 0.2 ammeter, and a class 0.5 voltmeter. In order to account for possible nonuniformity of the current density, a special circuit was set up along the heaters consisting of switches, a voltmeter, and Copel thermoelectrodes with which the voltage drop across heater segments between neighboring thermocouples could be measured.

The test procedure for determining local values of heat-transfer coefficients was as follows. After the beginning of steady state on the test plate under certain operating conditions had been noted on the basis of thermocouple readings, the plate temperature was measured at 40 points. The heaters were turned off for this part of the experiment and thus almost adiabatic conditions were created at the test plate surface (the heat dissipation from the surface of the entire test plate through the insulation was not more than 3% of the excess heat content in the injected stream). The results of these measurements were used for determining the equilibrium temperature and for the subsequent calculation of α. Then, without changing the apparatus operating mode, the heaters were turned on and, after a new steady state had been reached, the temperatures of the test plate were recorded again. In addition, the temperature was also measured at check points underneath the plate, and thus the downward heat leakage from the heaters through the insulation could be calculated. Along with the other apparatus operating parameters were also measured both the velocity and the temperature profile in the exit section of the orifice and above it. The formula for calculating the heat-transfer coefficients on the basis of this test procedure was

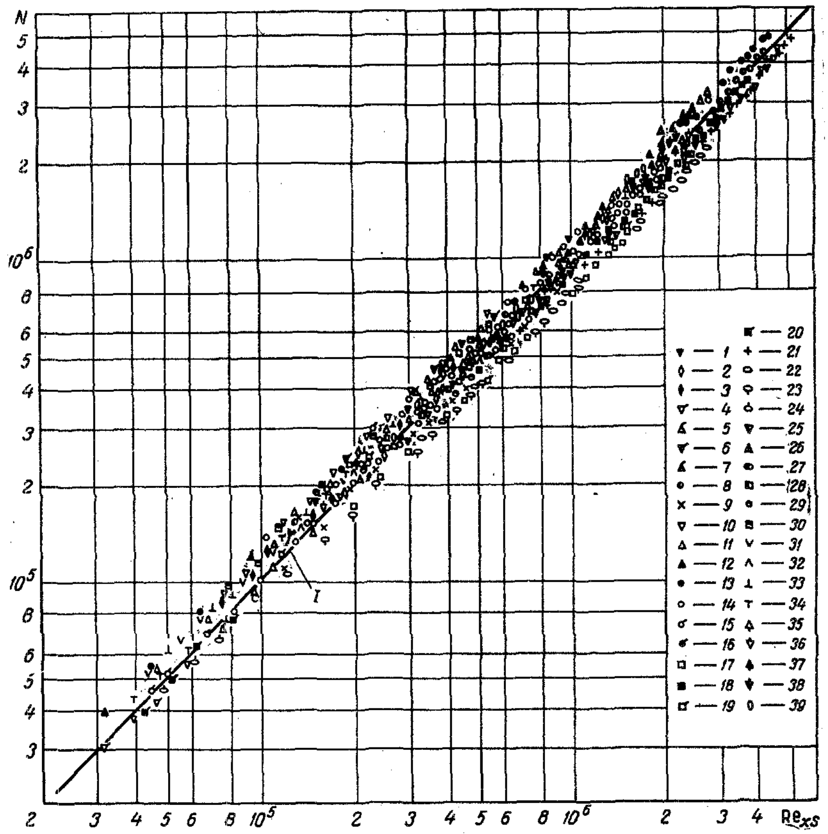

Fig. 2. Heat transfer in the transition stage: I) according to Eq. (7). (Legend is given in Table 1.)

$$\alpha = \frac{1}{T_n - T_{aw}} \left(\frac{IU}{F} - \lambda_i \frac{T_n - T_i}{h_i} \right). \tag{1}$$

The maximum relative error here is 12-15%.

The validity of this procedure for determining the local heat-transfer coefficients was confirmed by a series of control tests from which data were obtained on the heat transfer at a flat plate with a turbulent boundary layer and without coolant injection. A comparison of these test results with values calculated by the well-known formula [11]

$$\mathrm{Nu}_x = 0.0296\,\mathrm{Re}_x^{0.8}\mathrm{Pr}^{0.43}, \tag{2}$$

has shown that the maximum difference between them is 12%, the test values being generally somewhat higher than the calculated ones.

A long series of tests was then performed to determine local heat-transfer coefficients in both the main and the transition stage of film cooling, with the basic parameters varied as follows: Re_s from 5000 to 45,000, m from 0.2 to 1.06, and θ from 1.0 to 1.2.

The data for the main stage have been evaluated in critical terms, namely in terms of the local Nu_s number as a function of the Re_x number. The wall temperature under adiabatic conditions served as the reference temperature here. The test results have been plotted in this form in Fig. 1, where the dashed -dotted curve represents relation (2) and the dashed curve represents the theoretical relation according to [11] with Pr = 1:

$$\mathrm{Nu}_x = 0.0296\,\mathrm{Re}_x^{0.8}. \tag{3}$$

The solid curve generalizes the experimental data with a 10% dispersion; its analytical equation is

$$\mathrm{Nu}_x = 0.028\,\mathrm{Re}_x^{0.8}. \tag{4}$$

Relation (4) applies within the following variation limits for the basic parameters characterizing the main stage of film cooling: Re_s from 5000 to 45,000, m from 0.2 to 1.06, θ from 1.0 to 1.2, and Re_x from

TABLE 1. Basic Parameters of the Studied Modes

Legend numbers	m	$Re_s \cdot 10^{-4}$	θ	$Re_x \cdot 10^{-5}$	$Re_{xs} \cdot 10^{-5}$
1	0,881	2,36	1,008	22,6—42,8	2,68—13,8
2	0,768	2,30	1,01	19,3—43,0	1,83—13,9
3	0,430	2,32	1,018	10,3—43,4	0,75—4,16
4	0,192	2,28	1,018	3,89—42,8	0,31—0,61
5	0,266	4,00	1,005	10,7—74,9	0,76—2,20
6	0,530	3,98	1,003	25,5—74,0	1,50—9,40
7	0,760	3,98	1,000	33,3—74,3	3,15—23,8
8	0,917	3,98	1,000	46,9—74,3	5,00—41,0
9	0,892	1,15	1,005	9,65—21,5	1,07—8,08
10	0,725	1,17	1,004	8,12—21,8	0,76—5,46
11	0,600	1,16	1,005	5,78—21,8	0,47—2,93
12	0,386	1,16	1,005	3,49—21,6	0,32—1,20
13	0,862	4,47	1,002	50,5—83,2	5,36—41,8
14	0,512	4,47	1,004	28,7—83,2	1,66—10,3
15	0,944	0,52	1,038	4,11—9,70	0,44—3,50
16	0,435	1,42	1,028	5,29—26,8	0,45—2,30
17	1,067	1,38	1,008	14,2—25,7	1,91—14,2
18	1,053	2,81	0,988	34,5—52,3	5,00—36,2
19	0,572	2,81	0,988	18,0—52,3	1,15—7,55
20	0,204	2,81	1,007	4,76—52,3	0,42—0,811
21	0,968	4,20	1,000	53,3—78,0	7,08—51,2
22	1,040	2,29	1,003	27,0—42,8	3,30—27,0
23	0,988	1,19	1,003	11,6—22,1	1,62—10,9
24	0,2100	3,95	1,095	10,3—72,0	0,48—1,49
25	0,881	3,95	1,124	47,6—69,7	4,53—45,8
26	0,714	4,01	1,125	35,8—68,0	3,43—27,4
27	0,492	4,01	1,115	24,7—71,5	1,29—9,70
28	0,904	2,30	1,149	24,8—39,2	2,68—23,9
29	0,740	2,30	1,119	18,6—39,2	1,68—14,2
30	0,518	2,30	1,112	13,8—40,0	0,78—5,31
31	0,289	2,30	1,103	6,53—40,4	0,44—1,78
32	0,920	1,11	1,110	10,4—19,3	1,34—9,70
33	0,740	1,10	1,089	6,72—19,4	0,49—3,55
34	0,210	1,12	1,093	3,10—19,8	0,20—0,77
35	0,837	2,86	1,177	26,0—46,8	3,00—23,5
36	0,825	2,35	1,170	22,4—38,6	2,42—20,0
37	0,816	1,65	1,165	15,2—27,9	1,67—13,2
38	0,834	2,63	1,105	26,2—45,3	2,74—22,4
39	0,845	2,03	1,105	19,6—35,3	2,15—16,9

310,000 to 8,320,000. A comparison between the curves in Fig. 1 points clearly to a close agreement between the results of this study and the data in [4-10], where it has been suggested that the heat transfer in the main stage be calculated by the formulas for a plate with a turbulent boundary layer.

The data for the transition stage have been evaluated in terms of the local Nu_x number as a function of the $Re_{x,S}$ number, this relation having been calculated from the injected stream velocity at the reference temperature (the wall temperature under adiabatic conditions), and their analysis has shown that the rate of heat transfer still depends on the injection factor m. The trend of the $Nu_x = f(Re_{x,S})$ curves plotted in logarithmic coordinates indicates that Nu_x must be the following function of the dimensionless parameters m and $Re_{x,S}$:

$$Nu_x = c\,Re_{xS}^n, \text{ where } c = f(m);\ n = f(m). \tag{5}$$

A further evaluation yielded an analytical form for c = f(m) and n = f(m). The resulting expressions can be written as follows:

$$c = 0.25\,m^{6.15};\ n = 0.66m^{-0.55}, \tag{6}$$

so that, finally, the formula for calculating the rate of heat transfer in the transition stage of film cooling becomes

$$Nu_x = 0.25m^{6.15} Re_{x,S}^{0.66m^{-0.55}}. \tag{7}$$

In Fig. 2 are shown test data which have been obtained for the transition stage, with the basic parameters varied as follows: Re_S from 5000 to 45,000, m from 0.2 to 1.06, θ from 1.0 to 1.2, and $Re_{x,S}$ from 20,000 to 5,000,000. It is evident here that formula (7) obtained from these tests and shown in Fig. 2 in logarithmic coordinates

$$\left(\frac{Nu_x}{0.25m^{6.15}}\right)^{\frac{1}{0.66m^{-0.555}}} = N;\ Re_{x,S},$$

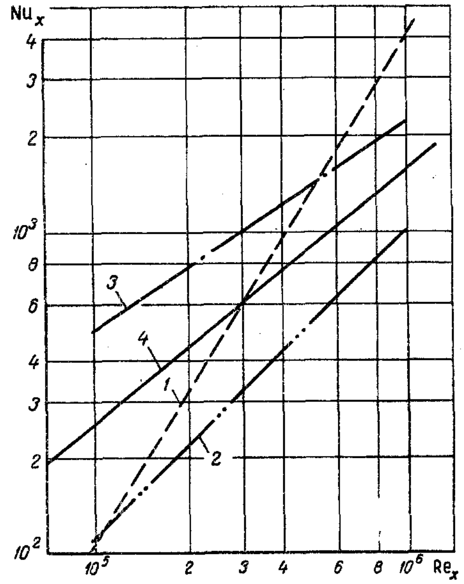

Fig. 3. Comparison of data on heat transfer in the transition stage: 1) according to relation (9) obtained in this study for m = 0.2; 2) 0.5; 3) and 1.0; 4) according to the data in [4-10].

generalizes the test data with a maximum dispersion of 15-18%. In Fig. 3 we compare relation (7), which has been obtained in this study, with those suggested in [4-10] for calculating the heat-transfer coefficients. For this purpose, with the aid of the relation

$$\mathrm{Re}_{x,S} = m\,\mathrm{Re}_x \tag{8}$$

expression (7) was transformed into

$$\mathrm{Nu}_x = 0.25\, m^{6.15+0.66m^{-0.55}} (\mathrm{Re}_{x,S})^{0.66m^{-0.55}}. \tag{9}$$

It is clearly evident here that in the transition stage of film cooling the heat-transfer coefficients may, depending on the injection factor m, be larger or smaller than calculated for a flat plate.

We note that the boundaries between the main stage and the transition stage should be defined by the values of the parameter $A = \Delta^{1.5}\mathrm{Re}_S^{-0.25} m^{-1.25} \theta^{-1.25} x/S$, for example, in [3] – these values defining the boundaries of both stages as far as the efficiency of film cooling is concerned, with $1 \le A \le 11$ for the transition stage and $A > 11$ for the main stage.

We note, in conclusion, that relations (4) and (7) obtained as a result of this study, as well as the results in [3], make it possible to calculate the values of heat-transfer coefficients in both the main and the transition stage, with the basic parameters varying as follows:

$$\mathrm{Re}_S = (0.5\text{—}4.5)\cdot 10^4;\; m = 0.2\text{—}1.06;\; \theta = 1.0 \text{ -- } 1.2.$$

NOTATION

u	is the velocity, m/sec;
ρ	is the density, kg/m^3;
T	is the temperature, °K;
S	is the orifice height, m;
x	is the distance from the point of injection, m;
μ	is the dynamic viscosity, N·sec/m^2;
$m = \rho_S u_S/\rho_0 u_0$	is the injection factor;
$\theta = T_S/T_0$	is the temperature parameters;
$\mathrm{Re}_S = \rho_0 u_0 S/\mu_0$	is the Reynolds number based on mainstream parameters and orifice height;
$\mathrm{Re}_x = \rho_{aw} u_0 x/\mu_{aw}$	is the Reynolds number based on mainstream velocity, space coordinate x, and parameters defined in terms of the adiabatic wall temperature;
$\mathrm{Re}_{x,S} = \rho_{aw} u_S x/\mu_{aw}$	is the Reynolds number based on the velocity of the injected stream, the space coordinate x, and the parameters defined by the adiabatic wall temperature;
α	is the heat-transfer coefficient, W/m^2·deg;
λ	is the thermal conductivity, W/m·deg;

$Nu_x = \alpha x/\lambda_{aw}$ is the Nusselt number based on space coordinate x and parameters defined in terms of the temperature of the adiabatic wall;
Pr is the Prandtl number;
Δ is the parameter characterizing the effect of the initial boundary layer [3];
F is the area of heat-transfer surface, m^2;
h_i is the insulation thickness, m;
I is the heater current, A;
V is the voltage drop across heater segment, V.

Subscripts

0 refers to main stream;
S refers to injected stream;
aw refers to adiabatic wall and to parameters defined in terms of the adiabatic wall temperature T_{aw};
i refers to insulation;
n refers to the wall when heaters are turned on.

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