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Film-Cooling Performance and Heat Transfer over an Inclined Film-Cooled Surface

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Experiments have been performed to study and obtain the adiabatic wall film-cooling effectiveness and the heat transfer over a film-cooled surface that is made inclined at various angles with respect to a uniform flow. The vertical temperature distribution was measured to infer the flow structure and the rate of mixing of the film jet with the freestream. The inclination angle of the film-cooled surface ranges from −20 to 20 deg with an increment of 5 deg, the blowing parameter from 0.5 to 2.0 (0.5, 0.7, 1.0, 1.5, 2.0). It is found that the mixing of the film jet with the freestream is significantly affected by the inclination angle of the film-cooled surface. The divergence of the film-cooled surface can make the flow unstable and cause flow separation, whereas the convergence has both stabilization and impingement effects on the film jet. This has a very complicated effect on both the film-cooling performance and the heat transfer under the film. More detailed discussion on the flow and the heat transfer is presented. Correlations for both the film-cooling effectiveness and the heat transfer under the film are provided.

Nomenclature

c_p	=	specific heat of air
h	=	heat transfer coefficient $= q/(T_w - T_{\mathrm{aw}})$, $\mathrm{W/m^2K}$
k	=	thermal conductivity, W/mK
M	=	blowing ratio $= \rho_c U_c / \rho_m U_m$
Nu_x	=	Nusselt number $= hx/k$
Pr	=	Prandtl number $= \rho \nu c_p / k$
Q	=	dimensionless heat flux, as defined in nondimensional analysis
q	=	heat flux, $\mathrm{W/m^2}$
Re_s	=	Reynolds number $= U_m y_c / \nu$
Re_x	=	Reynolds number $= U_m x / \nu$
St	=	Stanton number $= h / \rho_m c_{pm} U_m$
T	=	temperature, °C
Tu	=	turbulence intensity of the mainstream
t	=	slot lip thickness, mm
U_c	=	average velocity of the film jet, m/s
U_m	=	average velocity of the mainstream, m/s
x	=	axial distance from film slot, mm
y_c	=	slot height, mm
η	=	film-cooling effectiveness, $(T_m - T_{\mathrm{aw}})/(T_m - T_c)$
θ	=	convergent or divergent angle, degree
ν	=	kinematic viscosity, $\mathrm{m^2/s}$
ρ	=	density, $\mathrm{kg/m^3}$

Subscripts

aw	=	adiabatic wall
c	=	film jet at slot exit
m	=	mainstream
o	=	uniform flow
s	=	slot
w	=	wall

I. Introduction

FILM-cooling techniques have been widely used in cooling the high-temperature systems such as turbine blades, combustors, afterburners, and nozzles. Cooling air is usually injected from arrays of small holes on a turbine blade or a step slot on the liner of a combustor, which is usually at very high temperature, and forms an insulation layer to protect surfaces from being overheated. In a practical design of film-cooling in the combustor or afterburner, an ideal situation is usually assumed and considered, i.e., the mainstream is uniform and parallel to the film-cooled surface. The heat transfer under the film adopts correlations for a uniform flow over a heated flat plate. That means that the film-cooled heat transfer data are available only for a uniform flow over a heated flat plate. Extensive review of both analytical and experimental studies on film-cooling is available [1–3]. However, some more careful studies have moved into more realistic flow conditions, such as the presence of pressure gradient [4–7], acceleration [8,9], or swirling flow [10,11], in the mainstream, the appearance of turbulence intensity in either the mainstream [12–16] or the film jet [17], the surface roughness [18], and the surface curvature effect [19,20] of the wall. However, in a real situation at the entrance of the combustor, the main flow may expand, decelerate, or separate from the film-cooled surface, and may have a significant effect on both the film-cooling performance and the heat transfer under the film. It appears that the situations described previously still could not be used in this real application.

Work relevant to the current condition is the study of occurrence of pressure gradient or acceleration in the mainstream on the film-cooling effectiveness and the heat transfer [4–9]. However, in most of the previous experiments, the pressure gradients are made by contoured roofs rather than by the film-cooled surface. Therefore, the

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results could not be used for the current condition. The experiments performed by Carlson and Talmor [8] were the only design in which the pressure gradients are made by the film-cooled surface. However, the results are only for the film-cooled surface that is convergent at very large angles ($\theta = 30, 45$, and 60 deg). Work at a divergent angle or at a mild or small convergent angle is still lacking. In addition, most of the work measures only the film-cooling effectiveness but leaves out the heat transfer coefficient under the film. Up to the present, however, the effect of the complex flow mentioned previously on the film-cooling effectiveness and heat transfer is still unknown. Data for the film-cooling performance and the heat transfer under this complicated flow are lacking.

Therefore, experiments were performed to study and obtain the film-cooling effectiveness and the heat transfer over a film-cooled surface that was made inclined at various angles to the mainstream. When the film-cooled surface becomes convergent with respect to the mainstream, it may impinge and destroy the film jet structure. When the film-cooled surface becomes divergent, the mainstream becomes unstable and may separate from the film jet flow and enhance the mixing of film jet with the mainstream. The effect of this complicated flow on both the film-cooling performance and the heat transfer under the film will be studied. Instead of film cooling, the method of film heating is used to obtain the adiabatic wall temperature. That is, the film-cooling flow is heated before entering into the inclined surface that is well insulated. The heat transfer coefficient can be obtained when the wall surface is heated with a desired heat flux. The radial temperature distributions at several locations will be measured and used to infer the film jet structure and the mixing process of film jet with the mainstream. More detailed discussion on the effects of these complicated flows on the film-cooling performance and the heat transfer is presented.

Before the experiments, it is necessary to perform a nondimensional analysis to obtain the relevant nondimensional parameters that affect either the film-cooling effectiveness or the heat transfer process under the film. These nondimensional parameters can be used to perform both experiments and analysis. The measured film-cooling effectiveness and the heat transfer data can then be correlated in terms of these dimensionless parameters.

II. Nondimensional Analysis

In the literature [1–3], especially for Re, either Re_s or Re_x has been used as the correlation parameter. Sometimes, the Re_c, which is defined based on the exit velocity of the jet, is also used in the correlation for the Nusselt number. In addition, the slot lip thickness may also be used for characteristic length used in Re. The analysis presented is to clarify which of the Res defined is the most appropriate one. Consider the case that the flow is laminar, as shown in Fig. 1, and all the thermal and transport properties are constant. By introducing the nondimensional parameters

$$x' = \frac{x}{y_c}, \qquad y' = \frac{y}{y_c}, \qquad t' = \frac{t}{y_c}, \qquad y'_h = \frac{y}{y_c}, \qquad u' = \frac{u}{U_m}$$

$$v' = \frac{v}{U_m}, \qquad \theta' = \frac{T - T_m}{T_c - T_m}, \qquad Q = \frac{\dot{q}}{\rho c_p U_m (T_c - T_m)}$$

the two-dimensional steady boundary-layer equations become as follows:

continuity equation:

$$\frac{\partial u'}{\partial x'} + \frac{\partial v'}{\partial y'} = 0 \tag{1}$$

momentum equation:

$$u' \frac{\partial u'}{\partial x'} + v' \frac{\partial v'}{\partial y'} = \frac{1}{Re_s} \frac{\partial^2 u'}{\partial y'^2} \tag{2}$$

energy equation:

$$u' \frac{\partial \theta'}{\partial x'} + v' \frac{\partial \theta'}{\partial y'} = \frac{1}{PrRe_s} \frac{\partial^2 \theta'}{\partial y'^2} \tag{3}$$

and boundary conditions

$$x' = 0, \qquad 0 < y' < 1, \qquad u' = U_c/U_m, \qquad v' = 0$$

$$\theta' = 1, \qquad x' = 0, \qquad 1 < y' < 1 + t', \qquad u' = v' = 0$$

$$x' = 0, \qquad 1 + t' < y' < y'_h, \qquad u' = \cos\theta, \qquad v' = \sin\theta$$

$$\theta' = 0 \qquad x' > 0, \qquad y' = 0, \qquad u' = v' = 0$$

$$Q = \text{constant}$$

Thus,

$$\eta_{\mathrm{aw}} = \frac{T_m - T_{\mathrm{aw}}}{T_m - T_c} = f(Re_s, Pr, U_c/U_m x, t, \theta, y_c) \tag{4}$$

$$St = \frac{Nu_x}{PrRe_x} = \frac{hx}{k} PrRe_x = \frac{qx}{(T_w - T_{\mathrm{aw}})k} PrRe_x$$
$$= \frac{Q(T_c - T_m)y_c}{(T_w - T_{\mathrm{aw}})} PrRe_s x = f\left(Re_s, Pr, \frac{U_c}{U_m}, x, t, \theta, y_c\right) \tag{5}$$

In the experiment, Re_s, y_c, and the slot lip thickness are constant. For small temperature differences, Pr is constant and U_c/U_m may be replaced by blowing parameter M, because the pressure gradient is related to the acceleration and can be absorbed into the term θ. Therefore, Eqs. (4) and (5) can be written as

$$\eta = f(M, x, \theta) \tag{6}$$

and

$$Nu = f(Re_s, M, x, \theta) \quad \text{if } Re_s \text{ is not constant} \tag{7}$$

III. Experimental Apparatus and Procedures

The test section at the outlet of contraction is rectangular, as shown in Fig. 2a, and has inside dimensions of 200 mm in height, 302 mm in width, and 664 mm in length. Mainstream air is supplied from and discharged into the environment by an open type of wind tunnel. The turbulence intensity at the entrance of the test section is 1%, which was measured with a TSI hot-wire anemometer. The injection air is supplied by a different air system that can be heated at desired temperatures. Therefore, instead of the film-cooled method, the film-heated method is adopted to measure both the film-cooling effectiveness and the heat transfer coefficient. The flow rate of the injection air was measured with a rotary volumetric flow meter. Thus, the exit velocity of the film jet can be obtained. The turbulence intensity of the injection air measured at the exit is 4.5%.

The original thought for the film-heated wall is a 300 × 294 mm aluminum plate, which has a thickness of 5 mm and is heated by an electric heater attached on the back. Then, the heated wall is insulated on the back with a 20-mm-thick piece of balsa wood and a 10-mm-thick piece of Plexiglas. To reduce heat conduction toward downstream in the aluminum, however, the plate is cut parallel into 17 strips and a 2-mm-wide strip of balsa wood is inserted between the neighboring strips, as shown in Fig. 2b. The first seven strips in the upstream are all 10 mm wide and are followed by seven strips of 20 mm wide and three strips of 30 mm wide in the downstream. Therefore, each of the aluminum strips is attached on the back with a strip heater having the same size as the aluminum strip. All the plates are bolted together and the surface is well polished. Both sidewalls of the heated plate are all insulated with 30-mm-thick balsa wood.

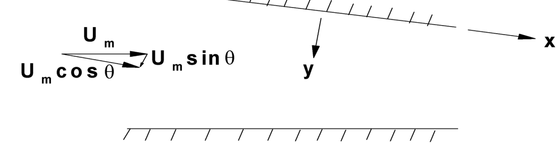

Fig. 1 Schematic diagram for the physical model.

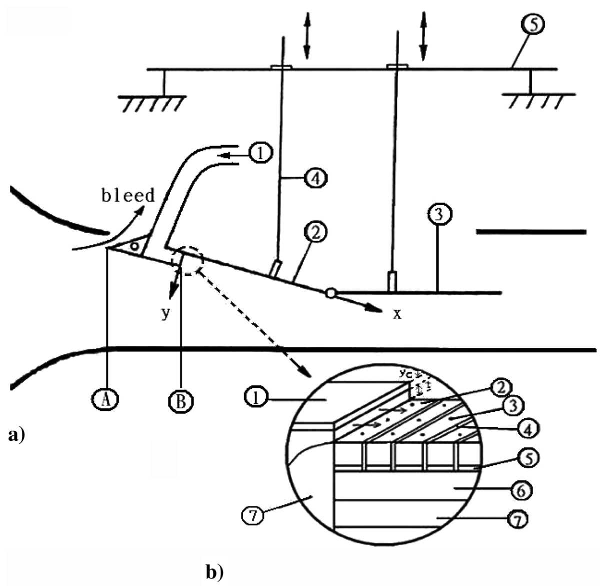

Fig. 2 Schematic diagram of the test section: a) 1) film jet, 2) film-cooled surface, 3) guide plate, 4) rotatable threaded rods, and 5) fixed frame; b) 180 deg rotation and enlarged view of the portion shown in Fig. 2a: 1) cover plate for the film flow, 2) aluminum strip, 3) thermocouple, 4) 2-mm-thick balsa wood strip, 5) electric heater, 6) balsa wood, and 7) Plexiglas plate.

Each aluminum plate is embedded with three sets of thermocouples. This is done by drilling three small holes on the back. Each of the thermocouples is then inserted into the small hole from the back, through the 20-mm-thick Plexiglas plate, the 20 mm thick balsa wood, and the strip heater. One of the thermocouples is in the central region of the aluminum strip. The other two are located at 50 mm distance from the two ends and are used to check for heat loss. The readings from these three thermocouples in each strip are used to evaluate the average temperature of the strip. The junctions of thermocouples are kept very close to the top surface of the aluminum strip. There are also additional thermocouples inserted in the mainstream and at the exit of the film jet for measurement of temperatures. When the heaters are turned on, the maximum heat loss is estimated to be 5.4%. A movable thermocouple probe is also made, which is mounted on an automatic traversing mechanism, to measure the vertical temperature distributions at various locations and the temperature at the film slot exit. All the thermocouples are calibrated in a constant temperature bath and have a measurement accuracy of $\pm 0.1°C$.

For measurements of film-cooling effectiveness, all the heaters on the film-heated surface were turned off. With good thermal insulation outside the wall, the adiabatic wall temperatures can be measured. With temperature measurements in the mainstream and at the exit of the film slot, the film-cooling effectiveness can be obtained, as described in Eq. (4) or defined in the nomenclature, by substituting all the temperatures into this equation. During the heat transfer measurements, the heaters on the film-cooled surface were turned on. With measurements of all the temperatures on the heated wall and the adiabatic wall temperature obtained previously, the heat transfer coefficient can be obtained from Eq. (5) or the one defined in the nomenclature. The injection air is heated $15 \sim 25°C$ higher than the ambient. The error analysis is performed by following the method reported in Kline and McClintock [21]. The maximum uncertainty of the measured film-cooling effectiveness is $\pm 9\%$ and that of the heat transfer coefficient is 7.4%.

The boundary layer of the mainstream at the outlet of the contraction is bled into the atmosphere, as shown in Fig. 2a, so that a more uniform air flow at the upper region of the entrance of the test section can be obtained. The film-heated surface is hinged at the upstream on an aluminum rod and can be adjusted at a desired inclination with the vertical threaded rods. A guide plate is attached to the end of the film-heated surface and the angle of the plate can be adjusted to ensure that the downstream flow does not move upstream and affect the mixing process between the film jet and the mainstream. Therefore, when the channel is convergent, the guide plate is made horizontal. When the channel is divergent, both the guide plate and the film-heated surface are maintained at the same flat plane. The average velocity along the plane B, as shown in Fig. 2a, is obtained by dividing the total mass flow rate in the wind tunnel by the cross section area of plane B, and is maintained at 20 m/s during the experiments.

IV. Results and Discussion

For the case when the film-heated surface is horizontal and $M = 1.0$, the film-cooling effectiveness is measured and compared with the results of others, as shown in Fig. 3. The agreement with the results of Ko and Liu [22] and Lefebvre [2] is found to be very good. In general, the present data are higher than others [23,24], except in the downstream where it is slightly lower than the correlation of Lefebvre [2]. This can be attributed to the smaller turbulence intensity in both the mainstream and the film jet in the current experiments. It has been found [12–16] that the turbulent intensity in either the mainstream or the film jet can significantly increase the rate of mixing of film jet with the mainstream and cause a lower film-cooling effectiveness. In the experiments of Ko and Liu [22], the turbulence intensity in the mainstream is 2% and that in the film jet is 10%, and is much greater than in the current flow conditions.

Figure 4 shows the comparison of the Stanton numbers between the present data with the published correlations for the case of uniform flow and $M = 1.0$. In the upstream region, where the film jet still preserves its jet flow structure, the present data have a good agreement with the result of Seban and Back [25], but has a large deviation with the results of others. The discrepancy is attributed to the difference of the film jet flow structure, i.e., the turbulence intensity and velocity profile of the film jet, in the different experiments. However, in the downstream when $x/y_c > 35$, the present data approach the results of others and have a very good agreement with the correlation of Jakob [26]. This is the situation when the film jet has mixed well with the mainstream and becomes somewhat like a turbulent boundary layer. In fact, the discrepancy in this stage is primarily attributed to the difference in the freestream or the film jet turbulence intensity among different experiments. The higher turbulence intensity in both the mainstream and the film jet in the case of Ko and Liu [22] causes higher Stanton numbers than the present results, as shown in Fig. 4.

A. Divergent Channel

For film-cooling in a divergent channel, the mainstream flow becomes slower in comparison with the film jet and unstable as flow

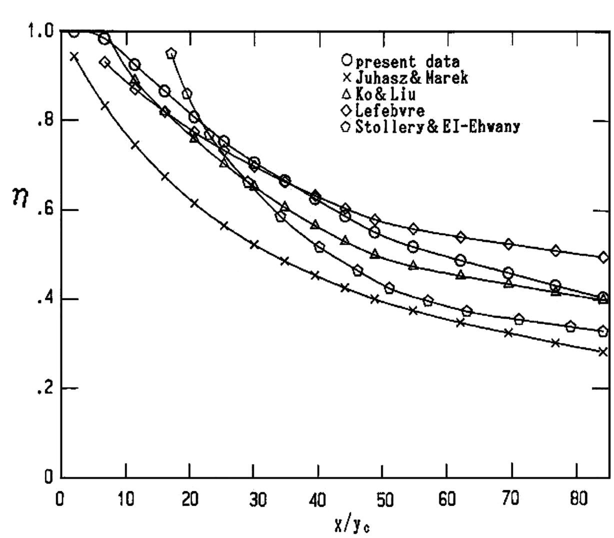

Fig. 3 Comparison of the present data with the published results.

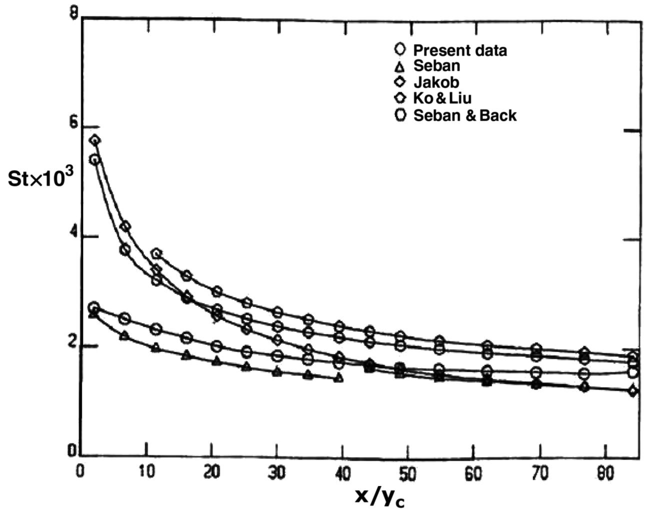

Fig. 4 Comparison of heat transfer results between the present data and the published results for uniform flow with M = 1.0.

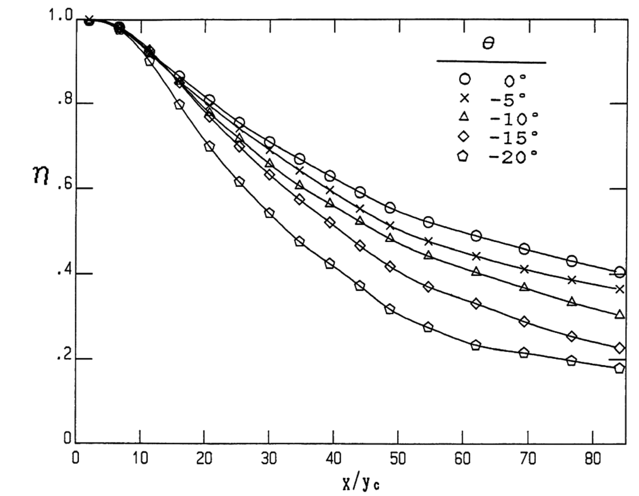

Fig. 6 Effect of the divergent angle on the film-cooling effectiveness for $M = 1.0$.

moves downstream, and turbulence can be readily generated. Generation of a similar kind of unstable flow was also found in a small angle of divergent channel [27–29]. In addition, for a large angle of divergence, the mainstream may separate from the film-cooled surface, and separation of flow can have a profound influence on the mixing rate of the film jet with the mainstream and the film-cooling performance. It is expected that as the angle of inclination is small, flow separation can occur in the downstream. As the inclination angle θ increases, the separation of flow may gradually extend upstream and the recirculation of flow becomes intense. Therefore, a more rapid rate of mixing of the film jet with the mainstream and a lower film-cooling effectiveness can be expected inside the region where separation of flow occurs. The film jet structure and the rate of mixing of the film jet with the mainstream can be inferred from the temperature distribution measurements normal to the film-heated surface at several locations and at different angles of inclination, as shown in Fig. 5. A sharp temperature gradient near the film-heated surface at $x/y_c = 20.7$ suggests that the film jet structure is well preserved at this stage. As the inclination angle of the film-heated surface increases, the film jet diffuses more rapidly from the wall. The increase in the diffusion rate of the temperature is attributed to both generation of turbulence in the highly unstable, decelerated flow and the upstream expansion of the recirculation zone accompanied with an increase in the recirculation intensity inside. The more rapid diffusion of temperature suggests the more rapid rate of mixing and the lower film-cooling effectiveness. At a large angle of inclination, the temperature gradient near the wall in the downstream is very small, which suggests that the film jet structure is severely destroyed.

Figure 6 indicates that the film-cooling effectiveness decreases significantly with the increase of the inclination angle. At a large angle of inclination, i.e., $\theta = -20$ deg, the film-cooling effectiveness decreases sharply even at a very early stage and maintains a very low value thereafter. A similar kind of physical process was also observed in the previous temperature distribution measurements. In the downstream, the film jet is weakened and the film jet structure has been almost completely destroyed. This can be inferred from both the temperature distribution (Fig. 5) and the film-cooling effectiveness measurements (Fig. 6).

The effect of the blowing parameter on the film-cooling effectiveness is also studied, as shown in Fig. 7. The increase in the blowing parameter can make the film jet structure stronger and increase the film-cooling effectiveness. However, the blowing parameter does not alter the fact that the increase of the angle of inclination can both enhance turbulence generation in the decelerated flow and extend the recirculation zone upstream. This can increase the rate of mixing of the film jet with the mainstream, and decrease the film-cooling performance. However, at a higher blowing parameter, the reduction in the film-cooling effectiveness with the inclination angle becomes greater. Therefore, it can be inferred that at

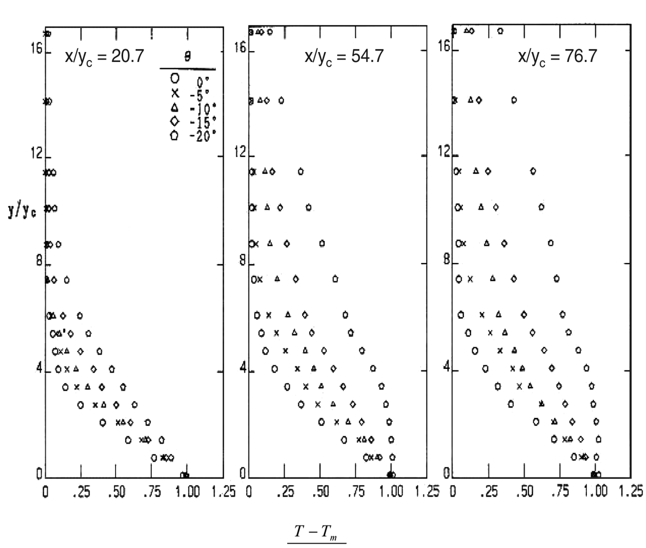

Fig. 5 Dimensionless temperature distributions normal to the film-heated surface at different divergent angles and axial locations for $M = 1.0$.

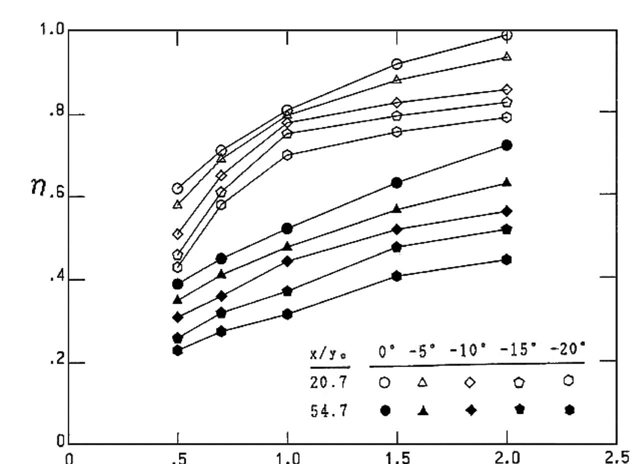

Fig. 7 Effect of the divergent angle on the film-cooling effectiveness at different blowing parameters.

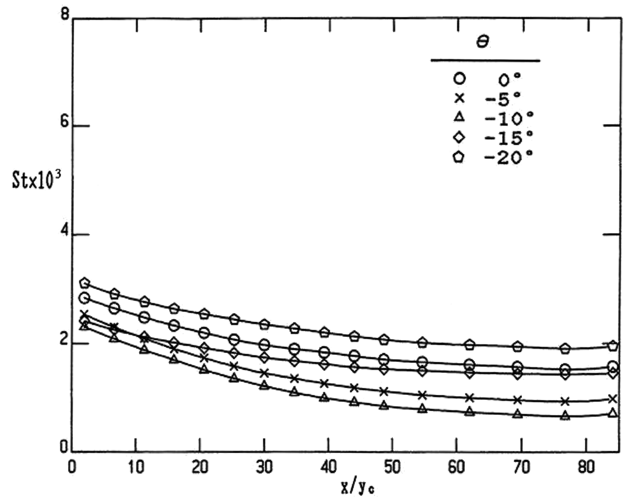

Fig. 8 Effect of the divergent angle on the Stanton number for $M = 1.0$.

a large angle of inclination, the film jet is significantly weakened and the film jet structure is almost completely destroyed even when the blowing parameter is very large.

The effect of inclination angle on the Stanton number is more complicated and not in proportion, as shown in Fig. 8. The Stanton number decreases with an increase of the inclination angle θ when θ is less than 10 deg, and increases with θ when it is greater than 10 deg. The peculiar trend of Stanton number which decreases with θ could not be explained clearly with the current data available because this variation does not follow the trend of η, which decreases with θ when θ is less than 10 deg. In this case, the rapid mixing of the film jet with the mainstream that can cause a decrease of η is expected to enhance the heat transfer under the film. However, the opposite results were obtained. One possibility is that the turbulence intensity in the film jet may decrease with increasing θ when θ is small and increase significantly with θ due to strong mixing of the film jet with the mainstream when θ is large. The reduction in the turbulence intensity in the film jet may be attributed to the mixing of the mainstream that has a much lower turbulence intensity, which has an effect of suppression on the film jet turbulence. However, this argument needs further verification with the measurements of the turbulence intensity in the film jet.

The increase of the blowing parameter is equivalent to the increase in the film jet velocity. This can significantly increase the heat transfer under the film, as shown in Fig. 9, and the increase appears very linear. However, the peculiar trend of heat transfer variation with the inclination angle does not change even when the blowing parameter increases or decreases significantly. The minimum heat transfer occurs at $\theta = -10$ deg for M varying from 0.5 to 2.0. The previous argument can still be used for M varying from 0.5 to 2.0.

B. Convergent Channel

In a convergent channel, the mainstream can be accelerated. It has been found [30–33] that flow acceleration due to convergence of the channel can suppress the turbulence and promote retransition to a laminar flow. Therefore, it is expected that the accelerated flow can relaminarize the turbulent flow in the mixing zone and reduce the rate of mixing of the film jet with the mainstream. This can cause an increase in the film-cooling effectiveness. However, in the current configuration, a decrease in the film-cooling effectiveness is obtained, as shown in Fig. 10, as the angle of convergence increases up to 10 deg. The decrease in the film-cooling effectiveness is attributed to another possible physical process that occurs in the convergent flow, i.e., the flow in the mainstream may impinge the film-cooled surface and destroy the film jet structure. This action can cause a decrease in the film-cooling effectiveness. One can expect that the impingement of flow occurs only in the upstream region where uniform flow enters into the convergent channel. Therefore, a lower film-cooling effectiveness in the upstream can be expected, as shown in Fig. 10. It appears that whether a lower or a higher film-cooling effectiveness can be obtained depends on whether the effect of the relaminarization or the impingement of the flow is dominant.

Figure 11 shows the nondimensional temperature distributions at several locations normal to the film-cooled surface. It appears that the nondimensional temperature distribution does not change very much with the convergent angle of the film-cooled surface despite a small deviation of the temperature distribution among different cases that still exists. This implies that the turbulent mixing of the film jet with the mainstream has been suppressed in some degree by the acceleration of the mainstream. The film-cooling effectiveness will not be changed too much by the convergent angle of the film-cooled surface. The film-cooling effectiveness η in the convergent channel is, in general, greater than that in the divergent channel. One can obtain the same conclusion by comparing the η in Fig. 6 and the η in Fig. 10. However, the η shown in Fig. 10 decreases with θ for θ less than 10 deg and increases with θ for θ greater than 10 deg. Similar correspondence has been observed in the temperature distribution measurements, as shown in Fig. 11, although the temperature variation with the convergent angle is not large. That means that, due to strong mixing of the film jet with the mainstream, the film-cooling effectiveness decreases which corresponds to increases of the temperature in the core as θ increases from 0 to 10 deg. Notice that for θ less than 10 deg the η value in the downstream region does not change too much with the convergent angle but approaches the case of $\theta = 0$ deg. It appears that in the near slot region, the effect of the flow impingement plays a more important role than the effect of

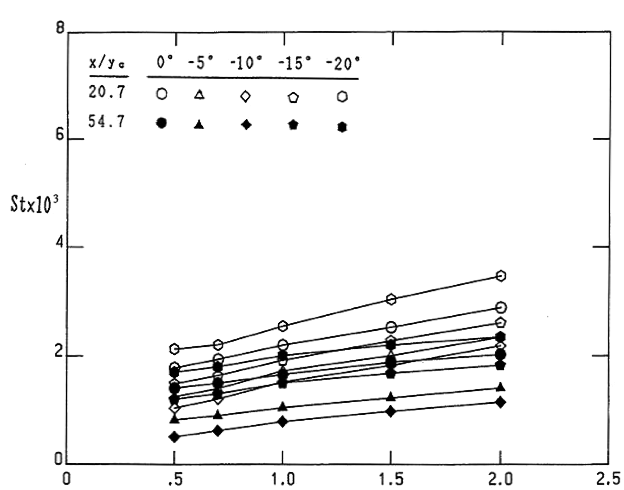

Fig. 9 Effect of the divergent angle on the Stanton number at different blowing parameters.

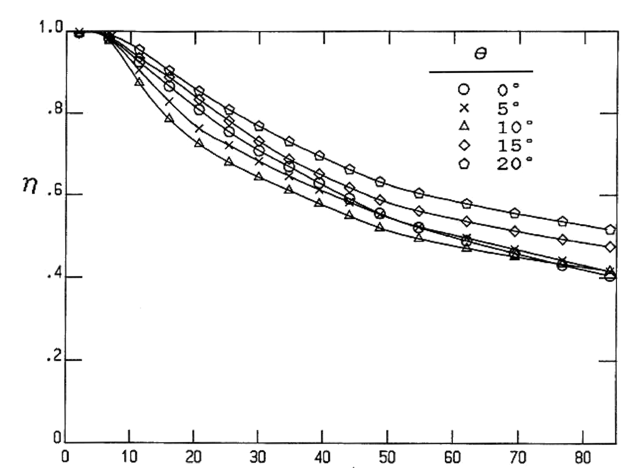

Fig. 10 Effect of the convergent angle on the film-cooling effectiveness for $M = 1.0$.

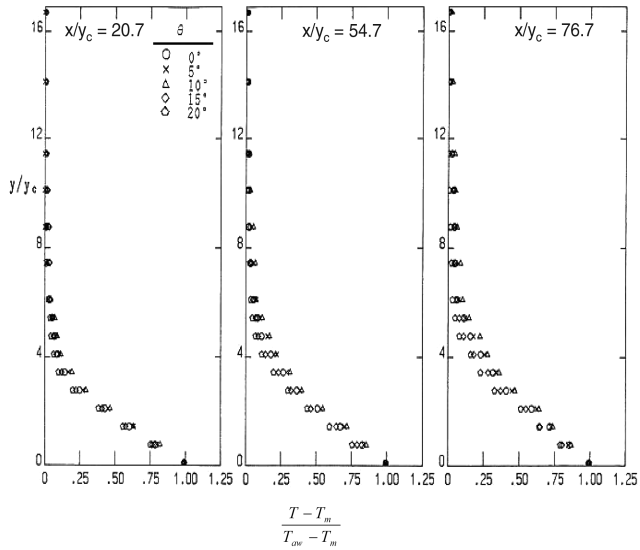

Fig. 11 Dimensionless temperature distributions normal to the film-heated surface at different convergent angles and axial locations for $M = 1.0$.

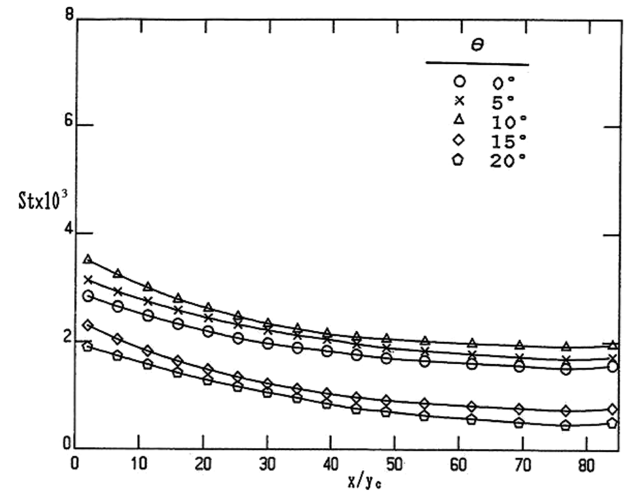

Fig. 13 Effect of the convergent angle on the Stanton number for $M = 1.0$.

relaminarization, whereas, in the downstream, both effects become less significant. For θ greater than 10 deg, the increase of η with θ is attributed to the relaminarization effect of the accelerated flow which suppresses the rate of turbulent mixing of the film jet with the mainstream.

The effect of blowing parameter on the film-cooling performance at various convergent angles is shown in Fig. 12 for two specific locations. The film-cooling effectiveness increases significantly with the blowing parameter unless it reaches unity. For the case of $M > 1.0$, the variation of the η with the θ is very similar to the case of $M = 1.0$. For $M < 1.0$, however, the film-cooling effectiveness in the upstream at $x/y_c = 20.7$ decreases significantly with an increase in θ for θ less or even greater than 10 deg. It appears that for $M < 1.0$, the film jet structure is very weak, and the impingement effect of the mainstream can significantly destroy the film jet structure in the upstream. For $M = 0.5$, the film jet structure is so weak that the film-cooling effectiveness decreases significantly by the impingement of the accelerated flow, and approaches approximately the same low value for θ from 5 to 20 deg, as shown in Fig. 12. In the downstream at $x/y_c = 54.7$, where the impingement effect of the accelerated flow becomes not as significant as in the upstream region, however, the variation of η with θ is similar to the case of $M = 1.0$.

The Stanton numbers measured along the film-cooled surface at different convergent angles are presented in Fig. 13. The variation of the Stanton number with the θ indicates an opposite trend to the variation of the η with the θ. This is expected because the rapid mixing of the film jet with the mainstream can cause a decrease in η but a increase in the Stanton number. On the other hand, the lower rate of mixing of the film jet with the mainstream can cause a increase in η but a decrease in the Stanton number. The Stanton numbers increases with θ for $\theta \leq 10$ deg and decreases with θ for $\theta > 10$ deg. It appears that for $\theta \leq 10$ deg, the impinging effect of the mainstream, which tends to destroy the film jet structure, can enhance the heat transfer under the film. The impinging effect is more significant in the upstream region, so that the enhancement of the heat transfer in the upstream is slightly larger than that in the downstream. For $\theta > 10$ deg, when the relaminarization effect becomes dominant, the relaminarization effect, which tends to reduce the mixing rate of the film jet with the mainstream, can decrease the heat transfer under the film.

The increase of the blowing parameter is equivalent again to the increase in the film jet velocity. This can also increase the heat transfer under the film in the convergent channel, as shown in Fig. 14, and the increase appears very linear. Therefore, the Stanton number variation with the convergent angle does not change very much when the blowing parameter increases or decreases. The Stanton number variation with the convergent angle at a higher blowing parameter is more significant than at a lower blowing parameter. This is attributed to the fact that the heat transfer under the film at higher film jet velocity is more sensitive to the impinging or relaminarization effect of the mainstream. At a lower blowing parameter, the film jet

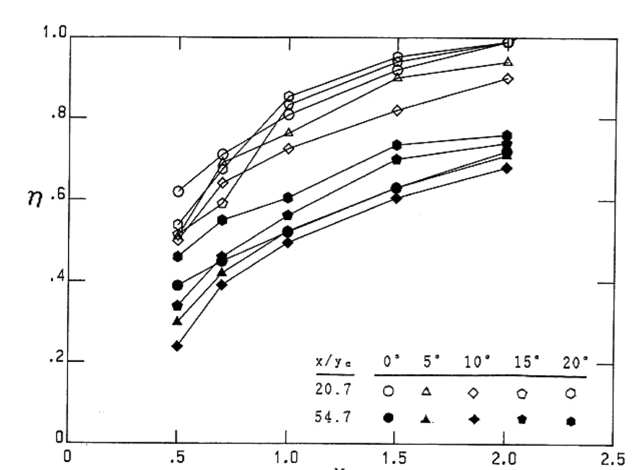

Fig. 12 Effect of the convergent angle on the film-cooling effectiveness at different blowing parameters.

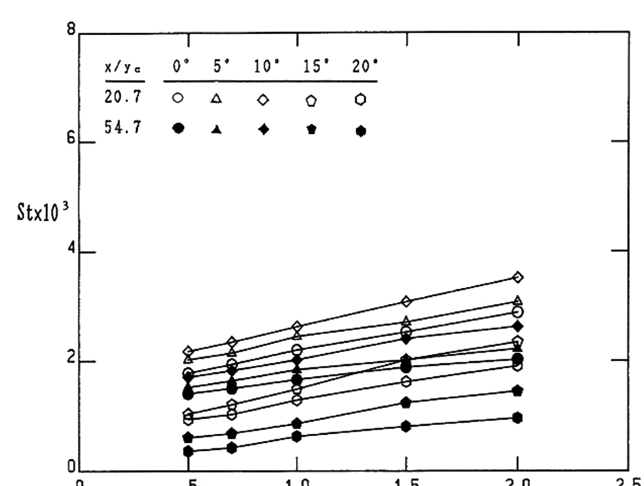

Fig. 14 Effect of the convergent angle on the Stanton number at different blowing parameters.

Table 1 Values of constant and standard deviations of the correlations

	A_1	Derivation	A_2	Derivation	A_3	Derivation
0 deg	3.16	7.9%	9.95	8.2%	0.00745	9.6%
−5 deg	3.43	8.1%	17.87	7.9%	0.00706	7.9%
−10 deg	3.41	8.5%	15.78	8.9%	0.00728	8.8%
−15 deg	4.23	7.6%	19.66	8.1%	0.00881	8.5%
−20 deg	4.61	8.3%	28.95	8.5%	0.0102	10.2%
5 deg	3.08	8.8%	15.81	8.2%	0.00879	8.8%
10 deg	3.11	7.9%	11.86	8.2%	0.0104	9.8%
15 deg	3.89	8.3%	13.84	8.4%	0.00822	9.6%
20 deg	4.23	8.4%	24.82	9.1%	0.00737	9.8%

structure has been destructed for a certain level, so that the heat transfer under the film becomes less sensitive to the impinging or relaminarization effect of the mainstream. An attempt was made to correlate both the film-cooling effectiveness and the heat transfer Nusselt number in terms of the nondimensional parameters obtained in the previous section. However, both the film-cooling effectiveness and the Stanton number variation with the inclination angle are not proportional. Therefore, correlation can only be made at each inclination angle. For a better correlation, the film-cooling effectiveness was correlated in different regions as follows:

For $1 \le M \le 2$, $-20 \le \theta \le 20$ deg, $6 \le x/y_c \le 50$,

$$\eta = A_1(\theta) M^{0.262} (x/y_c)^{-0.515} \tag{8}$$

For $1 \le M \le 2$, $-20 \le \theta \le 20$ deg, $50 \le x/y_c \le 85$,

$$\eta = A_2(\theta) M^{0.506} (x/y_c)^{-0.894} \tag{9}$$

In the preceding correlations, $A_1(\theta)$, $A_2(\theta)$ are constant. Their values and the standard deviations of the correlations at each inclination angle are listed in Table 1.

For the Nusselt number correlations, the Re_s is selected in the correlation. It is found that in the correlation, the power of Re_s and x/y_c are very close. Therefore, these two parameters are combined into a single one, i.e., Re_x, and y_c is absorbed in the constant of the correlation. The correlations for the Nu_x are written as follows:

For $1 \le M \le 2$, $-20 \le \theta \le 20$ deg, $0 \le x/y_c \le 85$,

$$Nu_x = A_3(\theta) M^{0.311} Re_x^{0.858} \tag{10}$$

In the preceding correlation, $A_3(\theta)$ is constant. The value of $A_3(\theta)$ at each inclination angle and the standard deviation of the correlation are also listed in Table 1.

V. Conclusions

The experiments were designed to study the film-cooling effectiveness and the heat transfer Stanton number under the film when the film-cooled surface is made inclined with respect to the mainstream. When the film-cooled surface is made horizontal, the present data of the film-cooling effectiveness are close to the correlation of Ko and Liu [22] and results of Lefebvre [2]. At the same time, the heat transfer coefficient is very close to the results of Seban and Black [25] and Jakob [26].

When the film-cooled surface is divergent, significant reduction in the film-cooling performance is obtained, especially in the downstream region. This is attributed to the separation of flow and generation of turbulence that can severely destroy the film jet structure. This can also be inferred from the rapid diffusion of temperature as θ increases. Therefore, film-cooling effectiveness decreases significantly with increasing the inclination angle. However, the Stanton number measurements do not indicate a trend of similar correspondence. Further measurements on the film jet flow structure are necessary to give a more clear explanation.

When the film-cooled surface is convergent at angle equal or less than 10 deg, the impingement effect of the mainstream is attributed to the major mechanism causing the decrease in the film-cooling effectiveness and the increase in the heat transfer under the film. The impingement effect can severely destroy the film jet structure, which can decrease the film-cooling performance and enhance the heat transfer under the film. However, when the convergent angle is greater than 10 deg, the relaminarization of the mainstream becomes significant, which can decrease the mixing rate of the film jet with the mainstream and increase the film-cooling performance. This action can significantly reduce the heat transfer under the film. The blowing parameter can increase the film jet structure, which leads to an increase in the film-cooling performance, and enhance the heat transfer due to the increase in the film jet velocity. Correlations for both the film-cooling effectiveness and the heat transfer under the film at each inclination angle are very successful.

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