which cannot be verified with the present test facility. However, this generalization seems justified by previous film cooling studies.

Depending on the mainstream temperature, the available coolant temperature, and the desired surface temperature, together with the total thermal environment of the part to be cooled, practically any value of θ^* could conceivably be of interest. However, in gas turbine applications, the film-air temperature depends primarily on the compressor pressure ratio, and the range of interest of θ^* will probably be:

$$1 \leq \theta^* \leq 5$$

For example, with a main gas temperature, t_m of 2000 deg F, a film temperature, t_f, of 1000 deg F, and a surface temperature of 1600 deg F

$$\theta^* = \frac{1000 - 2000}{1600 - 2000} = 2.5$$

Application. The surface thermal boundary condition for the present results is a uniform surface temperature. This situation is approached in many actual film cooling designs for the relatively short downstream region of interest. Longitudinal conduction in the cooled part and the desire to reduce thermal stresses are design factors which tend to even the surface temperature distribution.

The three final correlating equations, (10), (11), and (12) facilitate comparisons of the overall reduction in heat transfer rates provided by different combinations of the design parameters. When a tentative design is found to be feasible on this basis, reference to more detailed local heat transfer rates for the particular design chosen would be desirable. Although not a part of the present study, local heat transfer coefficients downstream of the injection slots can be provided approximately by repeated transient tests at the same flow conditions, but with surface blocks of several different lengths. The difference in average heat transfer rates to two different size blocks reflects the heat transfer rate in the local region comprising the additional length of the longer block.

Conclusions

The principal conclusions of the study are summarized as follows:

1 An experimental procedure has been described which facilitates comparisons of various film cooling schemes in regions where adiabatic wall temperatures alone are not sufficient for calculating heat transfer rates. Interpretation of the transient results obtained with this procedure has been verified by comparison with steady-state tests.

2 Secondary injection through flush, angled slots has been studied. The effects of the injected film thickness, angle, temperature, and flow rate can be represented by

$$\Phi = 1.09 - 7.9\theta^* G^{*0.6}(s/l)^{0.5} \quad \text{for} \quad \beta = 20 \text{ deg} \qquad (10)$$

$$= 1.18 - 7.5\theta^* G^{*0.6}(s/l)^{0.5} \quad \text{for} \quad \beta = 40 \text{ deg} \qquad (11)$$

$$= 1.24 - 6.8\theta^* G^{*0.6}(s/l)^{0.5} \quad \text{for} \quad \beta = 60 \text{ deg} \qquad (12)$$

in the range

$$0.25 \leq G^* \leq 1.0$$

$$35 \leq l/s \leq 70$$

References

1 Scesa, S., "Effect of Local Normal Injection on Flat Plate Heat Transfer," PhD thesis, University of California, Berkeley, Calif., 1954.

2 Hartnett, J. P., Birkebak, R. C., and Eckert, E. R. G., "Velocity Distributions, Temperature Distributions, Effectiveness and Heat Transfer for Air Injected Through a Tangential Slot Into a Turbulent Boundary Layer," *Journal of Heat Transfer*, Trans. ASME, Series C, Vol. 83, No. 3, Aug. 1961, pp. 293–306.

3 Stollery, J. L., and El-Ehwany, A. A. M., "A Note on the Use of a Boundary-Layer Model for Correlating Film-Cooling Data," *International Journal of Heat and Mass Transfer*, Vol. 8, 1965, pp. 55–65.

4 Wieghardt, K., "Hot-Air Discharge for De-icing," A. A. F. Translation No. F-TS-919-RE, 1946.

5 Chin, J. H., et al., "Adiabatic Wall Temperature Downstream of a Single, Tangential Slot," ASME Paper No. 58—A-107.

6 Papell, S. S., and Trout, A. M., "Experimental Investigation of Air Film Cooling Applied to an Adiabatic Wall by Means of an Axially Discharging Slot," NASA TN D-9, 1959.

7 Seban, R. A., "Heat Transfer and Effectiveness for a Turbulent Boundary Layer With Tangential Fluid Injection," *Journal of Heat Transfer*, Trans. ASME, Series C, Vol. 82, No. 4, Nov. 1960, pp. 303–312.

8 Samuel, A. E., and Joubert, P. N., "Film Cooling of an Adiabatic Flat Plate in Zero Pressure Gradient in the Presence of a Hot Mainstream and Cold Tangential Secondary Injection," *Journal of Heat Transfer*, Trans. ASME, Series C, Vol. 87, No. 3, Aug. 1965, pp. 409–418.

9 Papell, S. S., "Effect on Gaseous Film Cooling of Coolant Injection Through Angled Slots and Normal Holes," NASA TN D-299, 1960.

10 Swank, L. R., "Film Cooling Effects With Angled Slots Using a Transient Technique," MS thesis Arizona State University, Tempe, Ariz., 1967.

DISCUSSION

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The preceding paper presents welcome information on heat transfer occurring in film cooling. The experimental results have been evaluated to obtain dimensionless correlations for the heat transfer coefficients connected with this cooling method. The purpose of the present discussion is to suggest an alternative evaluation procedure which has been established by previous experimental investigations. An adiabatic wall temperature t_{ad} is

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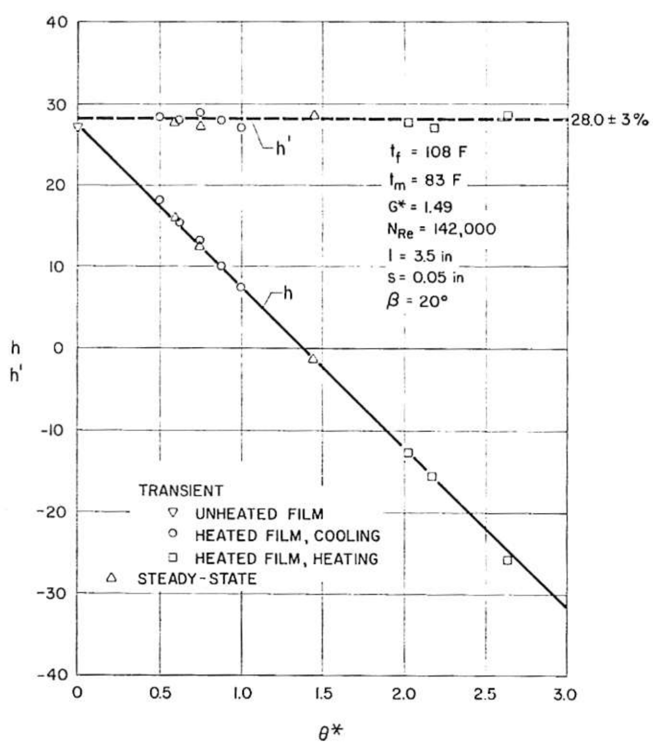

Fig. 12

introduced, defined as that temperature which the film-cooled wall assumes when the specific heat flux $\dot{q}$ in equation (7) assumes the value zero. This adiabatic wall temperature is expressed in a nondimensional way by an effectiveness parameter

$$\eta = \frac{t_{\mathrm{ad}} - t_m}{t_f - t_m} = \frac{1}{\theta_{\mathrm{ad}}^*} \tag{13}$$

identical to the reciprocal of the temperature parameter θ^* in the preceding paper for the condition $t_w = t_{\mathrm{ad}}$. A heat transfer coefficient is now introduced by the equation

$$\dot{q} = h'A(t_w - t_{\mathrm{ad}}) \tag{14}$$

This method has the advantage that for locations on the film-cooled wall which are not too close to the slot, this heat transfer coefficient h' agrees with good approximation with the heat transfer coefficient which is caused by the main flow alone when the flow of the film coolant is zero.

This method has been tested on the results of the preceding investigation presented in Fig. 3. The parameter θ_{ad}^* is found in this figure at the intersection of the inclined line with the abscissa ($h = 0$). It has the value $\theta_{\mathrm{ad}}^* = 1.387$. The corresponding effectiveness is $\eta = 0.722$. The heat transfer coefficient h' is found from the equation

$$\frac{h'}{h} = \frac{\theta_{\mathrm{ad}}^*}{\theta_{\mathrm{ad}}^* - *} \tag{15}$$

The values obtained for this parameter are plotted in Fig. 12 as the symbols which are well approximated by the dashed line within ±3 percent. The original data of Fig. 3 of the paper have also been repeated. It may be observed that the heat transfer coefficient h' is with good accuracy independent of whether cooling or heating occurs in the film cooling arrangement. Actually, the fact that h' is independent of θ^* follows immediately from the turbulent boundary-layer equations of a constant property fluid. The heat transfer coefficient without fluid injection is, according to Fig. 5, about 15 percent smaller than the value of h' in Fig. 12. This suggests that there is a good possibility that the proposed evaluation procedure will correlate the experimental data of the paper better than the evaluation, the results of which are presented in Figs. 9 through 11.

Authors' Closure

The authors are grateful to Professor Eckert for his comments and his interest in this paper. As pointed out in the Introduction, the adiabatic wall temperature is a particularly convenient quantity in film cooling at locations far downstream from the slot where h' can be determined from the behavior of the main flow alone. As can be observed from the results of the paper, this simplification is not consistently accurate for use in determining the average heat transfer rate on surfaces over at least the first seventy slot widths downstream of the injection site. For these important regions near the injection location, data with heat transfer must be obtained.

The choice of a method of presentation and use of this data is largely a matter of convenience. As Professor Eckert points out, there is a direct correspondence between the θ^* presentation, where $\dot{q}$ is based on $(t_w - t_m)$, and an effectiveness presentation with $\dot{q}$ based on $(t_w - t_{\mathrm{ad}})$. The linearity of the $h - \theta^*$ results, as well as the lack of dependence of h' on θ^*, follows from the governing equations, and the verification of this in the present results provides a check on the experimental work. The $\Phi - \theta^*$ presentation used in this paper in effect provides both h' and t_{ad} on the same graph. As Professor Eckert suggests, the h' results can be correlated separately, but such a correlation must be used with a correlation of the t_{ad} results.