HEAT TRANSFER AND EFFECTIVENESS ON FILM COOLED TURBINE BLADE TIP MODELS

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ABSTRACT

In unshrouded axial turbine stages, a small but generally unavoidable clearance between the blade tips and the stationary outer seal allows a clearance gap leakage flow to be driven across the blade tip by the pressure-to-suction side pressure difference. In modern high temperature machines, the turbine blade tips are often a region prone to early failure because of the presence of hot gases in the gap and the resultant added convection heating that must be counteracted by active blade cooling. The blade tip region, particularly near the trailing edge, is often very difficult to cool adequately with blade internal coolant flow, and film cooling injection directly onto the blade tip region can be used in an attempt to directly reduce the heat transfer rates from the hot clearance flow to the blade tip. An experimental program has been designed and conducted to model and measure the effects of film coolant injection on convection heat transfer to turbine blade tips. The modeling approach follows earlier work that found the leakage flow to be mainly a pressure-driven flow related strongly to the airfoil pressure loading distribution and only weakly, if at all, to the relative motion between blade tip and shroud. In the present work the clearance gap and blade tip region is thus modeled in stationary form with primary flow supplied to a narrow channel simulating the clearance gap above a plane blade tip. Secondary film flow is supplied to the tip surface through a line array of discrete normal injection holes near the upstream or pressure side. Both heat transfer and effectiveness are determined locally over the test surface downstream of injection through the use of thin liquid crystal coatings and a computer vision system over an extensive test matrix of clearance heights, clearance flow Reynolds numbers and film flowrates. The results of the study indicate that film injection near the pressure-side corner on plane turbine blade tips can provide significant protection from convection heat transfer to the tip from the hot clearance gap leakage flow.

NOMENCLATURE

A heat transfer area
\( c_p \) specific heat
\( b \) injection hole width
\( D_h \) channel hydraulic diameter
\( G \) mass velocity, = \( \rho V \)
\( G_m \) main flow mass velocity
\( G_f \) film flow mass velocity
\( H \) clearance gap height
\( h \) local convection coefficient in \( q = h (T_r - T_w) \)
\( h_{av} \) spanwise-averaged convection coefficient
\( k \) test surface thermal conductivity
\( k_f \) fluid thermal conductivity
\( m \) mass flow rate
\( M \) blowing parameter = \( G_f / G_m \)
\( N_{Du} \) Spanwise-averaged Nusselt number = \( h_{av} D_h / k_f \)
\( Pr \) Prandtl number = \( \mu c_p / k_f \)
\( q \) heat flux
\( R_{en} \) Reynolds number = \( G_m D_h / \mu \)
\( t \) time
\( T_i \) initial temperature
\( T_f \) film temperature
\( T_r \) reference temperature
INTRODUCTION

Convection heat transfer phenomena continues to play an important role in the development of improved gas turbine engines, for both aerospace and terrestrial applications. Improvements in overall engine performance almost always involve increases in hot gas temperature, and necessitate the development of better cooling schemes and more efficient use of the cooling air that is used to control temperatures in the engine components. To achieve acceptable durability, both temperature levels and temperature gradients must be controlled in the components exposed to the hot gas stream.

The sought-after temperature and temperature gradient control must be achieved with a minimum use of coolant, since most modern engines, regardless of application, use costly compressed air diverted from the engine compressor stages. Minimization of cooling air use is one of the primary motivations for the development of improved convection heat transfer knowledge and predictive ability.

One of the critical areas of gas turbine engines, in terms of durability and use of cooling air, is the blade tip region. In axial turbine stages under almost all operating conditions, a clearance gap exists between the blade tips and the outer stationary seal. Even with sophisticated clearance control methods and hardware this gap is never eliminated at all operating conditions (Hennecke, 1984). Thus in normal operation the pressure difference between the convex and concave sides of the blades drives a leakage flow across the tip which has detrimental effects on both aerodynamic performance and heat transfer. Near the pressure side of the gap, hot mainstream flow is turned into the gap (Fig. 1) with high acceleration levels and thin boundary layers (Mayle and Metzger, 1982; Metzger and Rued, 1989). Because of this boundary layer thinning and together with strong secondary flows within the hot gas blade path, the flow entering the gap is often primarily composed of fluid at or near the maximum temperature of the hot gases, particularly in the downstream tip region near the trailing edge. The resultant thermal loading at the blade tip can be very significant and very detrimental to tip durability, especially since the blade tip region near the trailing edge can be difficult to cool adequately with blade internal cooling flows.

As a result of their effects on turbine efficiency and performance, blade tip leakage flows have been the subject of fairly intense investigation for more than a decade (Allen and Kofskey, 1955; Booth, et al, 1982; Bindon, 1986; Moore and Tilton, 1988); but only in more recent years has attention focused on the heat transfer aspects of these leakage flows (Mayle and Metzger, 1982; Metzger and Rued, 1989; Rued and Metzger, 1989; Moore, et al, 1989). One of the results of these more recent studies has been demonstration that convection heat transfer on the blade tip itself is virtually independent of the relative velocity between the tip and the stationary outer ring seal. Despite the fact that the clearance gap is normally very small (the order of one percent of blade height), this independence has been established for both plane and grooved blade tip configurations both experimentally and numerically (Chyu, et al, 1986; Chyu, et al, 1987) for relative velocities greater than those expected in practice. The essence of the situation is sketched in Fig. 2, where the effects of the relative velocity are seen to be confined to a thin layer next to the shroud, with the velocity profile near the tip virtually unchanged from the shape it would have without relative seal motion. This independence has been previously used to enable the experimental study of tip heat transfer with stationary test sections (Metzger and Bunker, 1989), and the present study is a
continuation of those efforts with film coolant injection onto simulated plane blade tips. The plane tip configuration is representative of a large class of turbine blade designs practice, particularly for larger engine sizes where the size of the clearance gap relative to blade span can be kept smaller. The modeled situation is shown generally in the sketch of Fig. 1, where film coolant injection is provided at the tip from within the blade along the pressure-side corner to protect the tip from the deleterious effects of the hot leakage flow.

In the present experiments, the simulated blade tip surface downstream of a single line of film cooling injection sites is constructed of acrylic plastic and coated with a thin liquid crystal layer capable of giving a visual color indication of the local test surface temperature distribution. A transient test procedure is used, with heated primary and secondary flows applied to the test surface to cause the coating to display colors that are locally viewed and processed with a workstation-based computer vision system. The test method allows determination of detailed local convection coefficient and film cooling effectiveness distributions, and these distributions have been acquired over an extensive test range of clearance heights, clearance flow Reynolds numbers and film flowrates.

**EXPERIMENTAL APPARATUS AND PROCEDURES**

**Experimental Apparatus** A schematic arrangement of the test setup is shown in Fig. 3. The test section consists of a plenum chamber with calming section feeding a narrow channel with the bottom, or test, surface of this channel representing a plane turbine blade tip surface. Heated flow can be provided to both the primary and secondary lines through diverter ball valves which allow sudden application of the flow(s) to the test section. The test surface is liquid crystal coated, and the display from this coating is viewed and processed by the vision system components shown in the figure. The test section is constructed entirely of acrylic plastic: transparent on the top of the clearance gap channel to allow viewing of the surface from outside the channel, and black on the test surface itself to provide an optimum visual background for the liquid crystal coating display.

The main flow circuit consists of filtered and dried laboratory compressed air feeding an inline electric air heater with autotransformer controlled power input, a three-way ball-type flow diverter valve, and a flow balancing valve (not shown). The diverter valve is used to bypass the heated air from the test section until the heated air temperature reaches a desired value. The diverted air passes through the balancing valve which is adjusted to equalize the flow resistance between the test section and bypass circuit so that the flow rate is unchanged when the flow is suddenly routed to the test section.

The coating used is a commercially available microencapsulated chiral nematic thermochromic liquid crystal (TLC), applied to the test surface using an airbrush. This TLC displays colors in response to temperature changes as a result of lattice reorientation of the crystal. When sprayed as a thin layer, the TLC is essentially clear (showing the black background) and displays color with increasing temperature in sequence of red, green, blue, and back to clear. The nominal temperatures for red, green, and blue displays of the TLC formulation used are nominally 38.4 °C, 39.8 °C, and 43.5 °C, respectively. It is expected that this coating (the order of
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10^-3 cm thick) will have a response time of only a few milliseconds as shown by Ireland and Jones (1987). This time is negligible in comparison with the length of the thermal transients used in the present study.

To minimize experimental uncertainties, the temperatures of the supplied flows are chosen so that the color threshold is not reached until sufficient time has elapsed after the start of flow (usually 15 seconds or more) to insure that the elapsed time can be determined accurately. Also, the flow temperature is chosen so that the elapsed time and corresponding penetration of the temperature pulse into the surface are small enough (usually less than 60 seconds) to insure that the test surface can be treated as semi-infinite as discussed above. Experimental uncertainties have been assessed by the methods of Kline and McClintock (1953) and are estimated to be ±8 percent for convection coefficients, h, and ±10 percent for film cooling effectiveness, η.

The present computer-vision system employs the following three major components: (i) SUN SPARC II workstation with ethernet mainframe connection, (ii) RasterOps RGB color frame grabber, and (iii) Pulnix RGB CCD color video camera. Additional auxiliary components include a video cassette recorder, micro-computer and color printer. The use of the system to evaluate film cooling performance is described in the following section.

A cross-section and plan view of the test surface is shown in Fig. 4. The test surface is 4 in. (10.16 cm) wide in the spanwise direction and 3.125 in. (7.94 cm) long in the streamwise direction across the simulated blade tip from pressure side to suction side. Secondary film flow is supplied from the heater to a manifold which in turn distributes the film flow equally to a single line of five elongated injection holes, 0.125 in. (0.318 cm) wide (b) in the streamwise direction and 0.375 in. (0.953 cm) long in the spanwise direction. The five holes are equally spaced in the spanwise direction on a pitch of 0.5625 in. (1.429 cm), and the upstream edge of the hole array is located at a distance, x₀, of 0.1 in. (0.254 cm) from the pressure side corner. The channel height, H, (clearance gap) is set in the present tests to give values of H/b of both 1.5 and 2.5. For each value of H/b, primary channel flow Reynolds numbers were set at values of 15 x 10^3, 30 x 10^3, and 45 x 10^3, and secondary film flow rates were set to give values of M equal to 0. 0.1, 0.3, 0.5, and 0.9.

Figure 4 Test Surface Details

Measurement Theory and Procedures: The measurement of the test surface convection characteristics and evaluation of the performance of the film coolant injection follows the methods of Vedula and Metzger (1991) with local heat transfer rate expressed as:

\[ q = h (T_r - T_w) \]  \hspace{1cm} (1)

where \( T_w \) is local test surface temperature and \( T_r \) is the reference, or convection driving, temperature that renders \( h \) independent of the temperatures. For small temperature differences and constant fluid properties, the appropriate \( T_r \) reduces the convection coefficient \( h \) to a function of the aerodynamic character of the flowfield alone, and allows local surface convection behavior to be condensed into a constant of proportionality (h) for a given flowfield.

In two-temperature convection situations, with a single convecting fluid at temperature \( T_m \), the reference temperature is simply \( T_m \), and only \( h \) must be determined in the experiments. In film cooling with two flows present, the reference temperature is at some generally unknown level that depends on the supply temperatures of the two interacting streams and the degree of mixing that has occurred between them before they arrive at the various locations on the surface. For these situations, both \( h \) and \( T_r \) must be considered unknowns to be determined by experiment. Note that in general if the surface is locally adiabatic, \( T_r = T_w \) (adiabatic wall
temperature). Thus traditionally $T_r$ distributions over the surface have been obtained on adiabatic surfaces, but separate testing on a non-adiabatic surface is required to determine the $h$ distribution. With the present test method, both $T_r$ and $h$ are determined with use of the same surface.

In the present experiments, the test surface is suddenly exposed to the flow(s) and the transient response of the test surface as indicated by the TLC color display is observed. The wall material including the test surface is initially at a uniform temperature at all depths, and the initial response near the surface is governed by a semi-infinite formulation of the transient heat conduction where the temperature at the surface is given by the classical solution:

$$
\frac{T_w - T_1}{T_r - T_1} = 1 - \exp \left[ \frac{h^2a(t - t_1)}{k^2} \right] \text{erfc} \left[ \frac{h\sqrt{a(t - t_1)}}{k} \right] \tag{2}
$$

The semi-infinite description is appropriate as long as the transient temperature penetration does not exceed the thickness of the wall material being used, and thus the penetration time becomes the criteria for deciding test wall thickness and subsequent allowable transient test duration. The solution is applied locally at all points on the surface in accordance with the findings of Metzger and Larson (1986) and Vedula, et al (1988) which showed that lateral conduction effects within the test surface are very small, even with strong variations of $h$ over the surface.

For situations where $T_r$ is known, for example tests in the present program conducted without secondary flow present, $h$ can be determined from Eq.(2) by measuring the time $t$, required for the surface temperature to reach a prescribed value as indicated by the coating color display. The method is extended to film cooling and other three-temperature situations (Metzger and Vedula, 1991) by noting that both $h$ and $T_r$ can be obtained as the simultaneous solution of two equations of the form of Eq. 2 obtained either from a single transient test with two surface temperature indications at different times during the transient, or from two separate related transient tests. For example, if during the transient a liquid crystal surface coating indicates one surface temperature $T_{wg}$ at time $t_g$ corresponding to the green display and another $T_{wb}$ corresponding to the blue display at $t_b$, then $h$ and $T_r$ are determined from the simultaneous solution of:

$$
\frac{T_{wg} - T_1}{T_r - T_1} = 1 - \exp \left[ \frac{h^2a(t_g - t_1)}{k^2} \right] \text{erfc} \left[ \frac{h\sqrt{a(t_g - t_1)}}{k} \right] \tag{3}
$$

$$
\frac{T_{wb} - T_1}{T_r - T_1} = 1 - \exp \left[ \frac{h^2a(t_b - t_1)}{k^2} \right] \text{erfc} \left[ \frac{h\sqrt{a(t_b - t_1)}}{k} \right] \tag{4}
$$

In the experiments, for both two- and three-temperature situations, an additional complication is introduced, since true step changes in the applied fluid temperatures are usually not possible and the reference temperatures are thus functions of time. This complication is accounted for by modifying the equations through use of superposition and Duhamel's theorem (Metzger and Larson, 1986). The actual gradual change is obtained by using a series of steps. The solution is represented as:

$$
T - T_1 = \sum_{i=1}^{N} U(t - t_i) \Delta T_r \tag{5}
$$

where

$$
U(t - t_i) = 1 - \exp \left[ \frac{h^2a(t - t_i)}{k^2} \right] \text{erfc} \left[ \frac{h\sqrt{a(t - t_i)}}{k} \right] \tag{6}
$$

Here, $T_r$ is time-varying and unknown but related to the time variation in $T_m$ and $T_f$ and to the film cooling effectiveness $\eta$ such that:

$$
\Delta T_r = (1 - \eta)\Delta T_m + \eta\Delta T_f \tag{7}
$$

So:

$$
T - T_1 = \sum_{i=1}^{N} U(t - t_i)(1 - \eta)\Delta T_m + \eta\Delta T_f \tag{8}
$$

The two simultaneous equations are solved in the form of Eq. 8 to obtain the two unknowns, $\eta$ and $h$.

It is appropriate here to mention that a distinct advantage of the testing scheme described is that all wetted surfaces are thermally active. This is in contrast to many other testing techniques where only the test surface itself is thermally active and adjacent surfaces that may interact through convection with the test surface have unrealistic thermal boundary conditions. Moreover, the thermal boundary conditions on the test and adjacent surfaces are close to spatially isothermal, which is the wall surface condition usually desired in gas turbine engine component design.

RESULTS AND DISCUSSION

The immediate results of the test procedures described in the preceding section are two color-coded maps displayed on the workstation screen describing the local distributions of both
convection coefficients and film cooling effectiveness over the entire test surface downstream of injection for the various parameter combinations included in the test program (Kim and Metzger, 1992). For the purposes of clarity and comparison, various linear variations and averages can be derived from these maps, and a sampling of those sufficient to describe the character of the film cooling performance are presented in this section.

**Typical Local Convection Coefficient and Effectiveness Variations**

Figure 5 shows variations in convection coefficients along the spanwise direction over a nominal three-pitch distance centered at the spanwise center of the hole array (s=0) for a typical set of results acquired with \( Re_m = 30,000 \), \( M = 0.3 \), and \( H/b = 1.5 \). Data and results were acquired for this case and for all others over the entire test surface span but this central spanwise zone is used for presentation purposes to minimize array and test channel end effects. The distributions of Fig. 5 are presented for four streamwise locations downstream of injection presented in terms of number of hole widths, \( x/b \).

It is evident from this Fig. 5 that the overall variation in \( h \) across the central test span is small at the injection location and remains so at all stations downstream. The local variation is also quite small: no more than ±10-15 percent. This spanwise variation is typical of the intermediate blowing rates examined in this study for both \( H/b = 1.5 \) and 2.5. The local spanwise variations for \( M = 0.1 \) are very small, the order of ±5 percent, while the variations for \( M = 0.9 \) range upwards of 30-40 percent at the streamwise locations closest to the holes. As a general observation, however, it can be said that the spanwise variations in convection coefficient are small, particularly when compared with the corresponding variations in effectiveness. The small spanwise variation is probably attributable to the fact that the coefficients are elevated and dominated by the channel entrance effect, and are only elevated significantly by injection at the highest values of film flow rate.

In contrast to the situation for convection coefficients, Fig. 6 shows the corresponding variation in film cooling effectiveness for the same conditions and locations of Fig. 5. At the streamwise location closest to injection, and aligned with the center of the injection holes, effectiveness values of nearly 1.0 are recorded. Even exactly half-way between the holes only one hole width downstream of injection, the minimum local effectiveness is above 0.5. This lack of any totally uncooled region is of course an objective of the film cooling configuration design and is undoubtedly the result of the close spanwise spacing of the injection holes. Again, the local distributions presented in Fig. 6 are typical.

In general, the spanwise variations tend to be highest at the highest blowing rates, with higher spanwise-averaged effectiveness values close to injection, monotonically decreasing in the downstream direction. In all cases, the close spanwise spacing of the holes prevents any zero-effectiveness regions between the holes, with spanwise variations in local \( \eta \) at \( x/b = 1.0 \) varying from ±10-50 percent and decreasing with increasing \( x/b \).
Another way to visualize the spanwise variation in local effectiveness is with linear cuts through the surface effectiveness map aligned in the downstream direction and positioned at various positions on the span. Figure 7 is such a presentation, again for a typical, but different, set of conditions: Rem = 45,000, M = 0.5, and H/b = 1.5. Here three cuts at three positions on the span are presented: one aligned with the center of the holes, one aligned at the edge of the holes, and the third aligned halfway between the holes. The values shown are averages of results from similar positions within the center-span region defined in the discussion of Figs. 5 and 6. The behavior shown is again typical, with spanwise differences in effectiveness for this set of conditions varying somewhat with downstream distance, but in the range of ±20 percent.

The spanwise variations in effectiveness are important in practice since they play a major role in establishing the magnitude of local metal temperature gradients on the cooled surface, and in turn the size of the thermal stresses. In general, the present set of results show that the spanwise differences in effectiveness are very small for M = 0.1 and 0.3, but increase with increasing values of M to a maximum at M = 0.9. Also for all other parameters held constant, the spanwise differences are greater for H/b = 1.5 than for H/b = 2.5. Figures 8-10 illustrate and quantify these trends.
narrow channel decreases the size of the separated zone to the extent that it is contained within the narrow region upstream of \( x = 0 \). This may also place reattachment within the injection holes, which are present on the test surface even though zero film flow is injected. Also, it should be noted that acceleration levels associated with the contracting of the flow into the narrow clearance gap are high (Mayle and Metzger, 1982), and are higher as Reynolds number increases at constant gap spacing and higher for the narrow gap than for the wide gap. The degree of suppression of the \( \text{Nu}_D \) distributions below expected fully established values appears to increase as acceleration increases, suggesting that relaminarization may be a factor, but this speculation will need to be examined in future work. Also left for future work is the issue of unsteadiness and embedded vorticity within the ingested flow, as can exist in practice. For the purposes of the present work, Figs. 11 and 12 represent the no-injection \( \text{Nu} \) behavior that will be used as baselines to compare with cases with injection present.

Spanwise- Averaged Distributions

**Baseline Results, \( M = 0 \)** Figure 11 shows results conducted in the absence of film injection for the larger clearance gap, \( H/b = 2.5 \), for all three values of Reynolds numbers used throughout the study. The convection coefficient, presented here in Nusselt number form, display a classical sharp-edge channel entrance distribution with low values immediately downstream of the entrance (pressure side) corner attributable to flow separation. The values then rise abruptly in the streamwise direction, peaking at values of \( x/b \) that monotonically increase from 2.5 to 3.5 as Reynolds numbers increase, as expected and attributable to reattachment on the test surface. As flow develops streamwise, Nusselt numbers decrease, and are generally in agreement with expected fully established values near the downstream end of the channel. Fully established Colburn equation predictions are shown as solid symbols near the right edge of the figure, and show that the predictions systematically over-predict the measured downstream values as Reynolds number increases, although the overall agreement is good and establishes confidence in the experimental procedures used.

Figure 12 shows corresponding zero film injection results for the smaller clearance gap configuration, \( H/b = 1.5 \), and these values differ significantly both in character and in magnitude from the wider gap results. The values are generally lower than those in the previous figure, and the peak \( \text{Nu} \) behavior evident in that figure is absent, suggesting that the greater confinement associated with the

Effect of Injection Rate on \( \text{Nu}_D \) Figures 13-15 and 16-18 display the measured effects of sequentially increasing coolant injection rates on the spanwise-averaged Nusselt numbers, with the \( M = 0 \) baseline values repeated on each figure for reference. The important effect for all parameter combinations is the general tendency for convection coefficients to increase with increasing injection rate over virtually all of the protected surface. The increases for the maximum values of \( M \) used in the study are in the range of 30-50\%, and this is a significant factor for use of these results in assessing the probable effect of film cooling on blade tip thermal
loading in practice. This injection induced elevation in convection coefficients means that if the film injection temperature is not low enough, relative to the clearance leakage flow temperature, injection can actually increase the thermal loading on the tip, opposite to the desired effect.

The effect of injection rate on Nusselt number near the pressure side corner is more complex than the overall simple effect discussed above, but does appear reasonably consistent across the set of results for each channel height. In Figs. 13-15 for $H/b = 2.5$, the lower injection rates apparently allow the distribution of initial low NuD values followed by a peak to be retained at elevated values over $M = 0$, but the elevation in this zone near the pressure corner is not always monotonic with $M$. For the higher injection rates, $M = 0.5$ and 0.9, this pattern is broken and the heat transfer coefficients in the separated flow zone are elevated. In Figs 16-18 for $H/b = 1.5$, the higher $M$ values, particularly for the highest velocity flows of Fig. 18, appear to allow the local coefficient values to emerge from their suppressed levels right at injection, but they return to their suppressed levels almost immediately. In this regard it should be noted that at $Re_m = 45,000$ at $M = 0.5$ in Fig. 18, NuD is still below even the fully developed Colburn values over almost all the test surface.
Fig. 15 Effect of $M$ on Spanwise-Averaged Nusselt Numbers, $H/b = 2.5$, $Re_m = 45,000$

Fig. 16 Effect of $M$ on Spanwise-Averaged Nusselt Numbers, $H/b = 1.5$, $Re_m = 15,000$

Fig. 17 Effect of $M$ on Spanwise-Averaged Nusselt Numbers, $H/b = 1.5$, $Re_m = 30,000$

Fig. 18 Effect of $M$ on Spanwise-Averaged Nusselt Numbers, $H/b = 1.5$, $Re_m = 45,000$
Effect of Injection Rate on Effectiveness  Figures 19-21 and 22-24 display the measured effects of sequentially increasing coolant injection rates on spanwise-averaged effectiveness values for \( H/b = 2.5 \) and 1.5, respectively. These display essentially expected trends, but establish both the maximum values at injection and the decay rates for this particular family of configurations. Spanwise-average effectiveness increases essentially monotonically with increasing blowing rate up to \( M = 0.5 \). At all cases where the higher rate \( M = 0.9 \) was included in the test sequence, this rate resulted in slightly lower effectiveness near the injection site, but higher effectiveness downstream. This behavior is probably explained in terms of more separation of coolant jet from the surface at small \( x/b \), together with enhanced entrainment of the primary stream toward the surface at the edges of each film jet. The film subsequently reattaches to the surface, and the greater amount of injected coolant provides better downstream coverage. An anomalous set of results is noted in Fig. 22 for \( M = 0.1 \), where effectiveness is unexpectedly low upstream, and drops off to essentially zero almost immediately. This could again be the result of some interaction between the injection site and initial entrance separation and reattachment, and provides a further reason for future more detailed investigation of the behavior right at the pressure side corner.

Effect of Reynolds Number on Effectiveness  Finally, Figure 25 shows a set of results displaying the relative independence of the spanwise effectiveness values from changes in Reynolds number. This is behavior typical of the present results set, although close examination of Figs 19-24 will show that the spread of the results with \( Re_D \) is sometimes greater than shown in Fig. 25.
Fig. 22 Effect of M on Spanwise-Averaged Effectiveness, $H/b = 1.5, Re_m = 15,000$

Fig. 23 Effect of M on Spanwise-Averaged Effectiveness, $H/b = 1.5, Re_m = 30,000$

Fig. 24 Effect of M on Spanwise-Averaged Effectiveness, $H/b = 1.5, Re_m = 45,000$

Fig. 25 Typical Effect of $Re_m$ on Spanwise-Averaged Effectiveness
CLOSURE

The results of the study indicate that film injection near the pressure-side corner on plane turbine blade tips can provide significant protection from convection heat transfer to the tip from the hot clearance gap leakage flow. The results, for a closely spaced single row of elongated injection sites, indicates that injection provides nearly continuous spanwise film coverage, and for a high enough blowing rate, effectiveness should remain significantly above zero over the entire distance to the suction-side corner. Injection generally increases the convection coefficients in a straightforward manner over most of the covered surface. The results for the smallest clearance spacing investigated indicate some unexplained trends and complexities that warrant further study, but the overall results are in reasonable accord with expectations, and should aid designers in predicting the film cooling performance on plane turbine blade tips.

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