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TWO-DIMENSIONAL COLD-AIR CASCADE STUDY OF A FILM-COOLED TURBINE STATOR BLADE

IV - COMPARISON OF EXPERIMENTAL AND ANALYTICAL AERODYNAMIC RESULTS FOR BLADE WITH 12 ROWS OF 0.076-CENTIMETER- (0.030-in.-) DIAMETER HOLES HAVING STREAMWISE EJECTION ANGLES

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SUMMARY

Previously published, experimentally determined changes in aerodynamic efficiency due to coolant ejection from a film-cooled turbine stator blade having 12 spanwise rows of coolant holes are compared with efficiency changes predicted by two published analytical methods. One of the methods was used as published; the other was modified. The experimental results were from tests in a two-dimensional cascade with both the coolant flow and the primary flow at near-ambient temperatures. The tests included coolant discharge from each of the single rows and from various combinations of rows, including full-film discharge.

For single-row and multirow coolant discharge from blade surface regions with static pressures greater than the blade-exit pressure, both analytical methods predict the efficiency changes quite well. For instance, when the total pressures of the coolant and primary flow are equal, the range of differences between the experimentally measured and the predicted efficiency changes is less than 1 percent for all cases and less than 1/4 percent for most cases.

For coolant discharge from blade surface regions where static pressures are lower than the blade-exit pressure, the efficiency changes predicted by both analytical methods are less than the experimental results. However, the modified method does give the better prediction. An example is full-film cooling in which 3 of the 12 rows of holes are discharging coolant to blade surface regions where the static pressures are lower than the blade-exit pressure. For this case, when the total pressures of the coolant and primary flows are equal, the efficiency changes predicted by the modified method are about 2 percent less than the experimental results, while those predicted by the other method are about 6 percent less than the experimental results.

For all cases of coolant discharge, the experimentally determined trends of efficiency change are predicted well by both analytical methods.

INTRODUCTION

Different means of ejecting cooling air from the turbine blade surface are known to have significantly different effects on turbine aerodynamic efficiency. Extensive research programs (e.g., refs. 1 to 17) are in progress at the Lewis Research Center and elsewhere to investigate both analytically and experimentally the effects of various coolant ejection schemes. As examples, references 1 and 2 present two simple analyses for predicting the effect of different coolant schemes on turbine aerodynamic efficiency. References 3 to 6 report the results of an experimental investigation of the effect of stator blade transpiration cooling on turbine performance, and references 7 to 11 report the results of an experimental investigation of the effect of stator blade trailing-edge ejection on turbine performance. Reference 12, which summarizes the results of the stator blade transpiration and trailing-edge ejection investigations, shows trailing-edge ejection to be more efficient aerodynamically than transpiration ejection.

Currently, a program is in progress at Lewis to study the factors affecting the aerodynamic performance of full-film cooled stator blades. Two analytical methods used to predict the effect of coolant on performance are described in references 1 and 2. Experimental results from three full-film cooled stator-blade tests are being compared with results predicted by these two analyses in order to improve their usefulness by applying empirical corrections. The first blade tested (refs. 13 and 14) had 12 spanwise rows of coolant holes with nominal ejection angles in a streamwise direction. The second blade tested (ref. 15) had the same configuration of coolant holes, but the coolant-hole diameter was half of what was used on the first blade tested. The third blade studied (ref. 16) had 45 spanwise rows of holes and was tested with seven different coolant-hole angle orientations, including streamwise, spanwise, and compound-angle orientations.

This report compares the experimental efficiency results of the first-tested stator blade (refs. 13 and 14) with results predicted by the analytical methods of references 1 and 2. The comparison includes changes in efficiency due to coolant ejection from each of the individual rows and various combinations of coolant rows, including full-film cooling.

The experimental results were obtained from tests conducted in a two-dimensional cascade with the temperatures of the primary and coolant air near the ambient temperature. The tested nominal ideal primary-air critical velocity ratios at the blade-row exit were 0.5, 0.65, and 0.8. Tests were conducted with ejection from each of the separate coolant rows and from various combinations of coolant rows, including full-film cooling.

In computing the predicted results by the analytical procedures of references 1 and 2, the analysis of reference 1 was used as published, while the analysis of reference 2 was modified by an empirical correction to better fit the data. Brief descriptions of the two published analyses and the modification to the one of reference 2 are presented.

The comparative experimental and predicted results of the subject investigation are reported in terms of fractional change in primary efficiency relative to the efficiency of the noncooled solid blade. Primary efficiency is defined as the ratio of the actual kinetic energy of the total flow (primary plus coolant) relative to the ideal kinetic energy of the primary flow only. Predicted results are compared with experimental results for all cases of single- and multiple-row coolant ejection reported in references 13 and 14. Results are reported only for an ideal primary-air critical velocity ratio of 0.65 since references 13 and 14 show that primary efficiency is only slightly affected by critical velocity ratios in the range of 0.5 to 0.8.

SYMBOLS

D	coolant-hole diameter, cm; in.
L	coolant-hole length, cm; in.
L_{pr}	pressure-surface length from leading edge to trailing edge (fig. 1(c)), cm; in.
L_s	suction-surface length from leading edge to trailing edge (fig. 1(c)), cm; in.
p	absolute pressure, static, N/cm^2; psia
p'	absolute pressure, total, N/cm^2; psia
V	absolute velocity, m/sec; ft/sec
w	mass flow rate, kg/sec; lbm/sec
x	local position along blade surface from leading edge (fig. 1(c)), cm; in.
y	total coolant fraction, w_c/w_p
y_i	coolant fractions of individual coolant rows discharging to diffusion region
β	angle between coolant-hole axis and local blade surface tangent in plane parallel to end-wall surface, deg
η_o	blade-row efficiency with no coolant flow
η_p	primary efficiency (ratio of actual kinetic energy of total flow to ideal kinetic energy of primary flow only)

η_{th} thermodynamic efficiency (ratio of actual kinetic energy of total flow to ideal kinetic energy of total flow)

Subscripts:

c coolant flow

cr conditions at Mach 1

cp compression

id ideal quantity corresponding to isentropic process

m blade exit station where flow conditions of coolant and primary flows are assumed to be uniform (fully mixed flow)

p primary flow

s blade surface

t total flow (primary plus coolant)

0 station at blade-row inlet

Superscript:

' total-state conditions

DESCRIPTION OF TEST BLADE

The test blade of references 13 and 14 is shown in figure 1. The blade is hollow, untwisted, and of constant cross section. The blade profile corresponds to the mean-section profile of the stator blade of reference 18, in which the blade is described in detail. Significant dimensions of the blade are as follows: span, 10.16 centimeters (4.0 in.); chord, 5.74 centimeters (2.26 in.); pitch, 4.14 centimeters (1.63 in.).

Figure 1(c) shows the profile of the subject blade with the numbering system for the coolant-hole rows. Table I lists the locations of the coolant-hole rows and the orientations and geometries of the holes. (The symbols used in table I are illustrated in fig. 1(c), and all symbols are defined in SYMBOLS section.) The axes of all coolant holes are parallel to the planes of the blade end walls. The diameter and pitch of the coolant holes in all rows are 0.076 centimeter (0.030 in.) and 0.114 centimeter (0.045 in.), respectively.

DESCRIPTION OF EXPERIMENTAL TESTS

In the experimental investigations of references 13 and 14, separate tests were first made with coolant ejection from each of the 12 single rows of coolant holes. Next, the effects of multirow coolant ejection from the suction and pressure surfaces were individually determined. In the multirow tests, the combinations of rows considered were the two rows nearest the leading edge, the three rows nearest the leading edge, etc., until all six rows on the separate surfaces were included. Finally, tests were conducted to determine the effect on stator-blade performance of full-film cooling from all 12 coolant-hole rows. In addition, a multirow test was made with coolant discharge from the four rows of holes nearest the leading edge, two of the rows being on the pressure surface and two on the suction surface. All the multiple row configurations that were tested are listed in table II.

DISCUSSION AND DEFINITION OF PRIMARY EFFICIENCY

There are a number of efficiency expressions commonly used to describe the performance of high-temperature turbines requiring coolant. For cold aerodynamic tests without internal blade inserts, actual hot-engine heat-transfer and pressure-drop processes are not duplicated, and the selection of an efficiency definition becomes arbitrary. The major parameter studied in the subject aerodynamic investigation was the effect of ejected coolant on the output kinetic energy of the combined flow (primary plus coolant). This effect is well shown by the efficiency expression termed primary efficiency. Therefore, the results of this investigation were computed and reported in terms of primary efficiency. Primary efficiency relates the actual energy of the total flow to the ideal energy of only the primary flow and is expressed as

$$\eta_p = \frac{w_t V_{t,m}^2}{w_p V_{p,id,m}^2} \tag{1}$$

Thermodynamic efficiency is the same as primary efficiency except that the ideal energy of the coolant flow is included in the denominator.

$$\eta_{th} = \frac{w_t V_{t,m}^2}{w_p V_{p,id,m}^2 + w_c V_{c,id,m}^2} \tag{2}$$

A significant state condition for comparing primary efficiency results is when the ideal specific energies of the coolant and primary flows are equal. For cold-air tests having equal inlet temperatures, this condition occurs when the inlet total pressures of the primary and coolant flows are equal ($p'_c = p'_{p,0}$). For this condition, the change in primary efficiency relative to the efficiency of the uncooled blade is caused by the coolant and represents that part of the output kinetic energy contributed by, or charged to, the coolant. In effect, it represents the efficiency of the coolant flow and is a useful measure for evaluating various coolant schemes for the same or different blades.

PREDICTION METHODS

The predicted results of this report are based on two published analyses. In this report, the analysis of reference 1 is referred to as "method 1," and the analysis of reference 2 as modified herein is referred to as "method 2."

For convenience to the reader and to indicate differences between the methods, somewhat abbreviated descriptions of the two published analyses and of the modification to the method of reference 2 follow. For detailed descriptions of the published analyses, the reader should refer to the references.

Method 1

In the TOTLOS method of reference 1, called "method 1" in this report, the model depicted in figure 2 is used to determine the effect of coolant ejection from single rows of coolant holes on blade-row output.

The following procedure is applied for coolant ejection from each blade surface. At the first upstream ejection location, the coolant flow and the mixing-layer flow, which is a fractional part of the entering primary flow (see fig. 2), are mixed one-dimensionally at constant static pressure corresponding to the blade surface pressure at the ejection location. In applying the mixing equation, it is assumed that the mixing-layer flow expands isentropically from blade row inlet (station 0 in fig. 2) to the ejection location and that only the component of coolant flow momentum in the same direction as the mixing layer contributes to the total momentum of the combined flow of the coolant and the mixing layer.

For successive points of ejection farther downstream, the contents of the mixing layer, which includes the mixed flow of the mixing layer and coolant flow from all upstream locations, is expanded isentropically from each ejection point to successive downstream locations. At each successive ejection location, the same one-dimensional, constant-static-pressure mixing process is applied as at the first upstream location.

When this procedure has been completed for the last ejection location on the individual blade surfaces, the total contents of each of the mixing layers is expanded isentropically from the last ejection location to the average static pressure at the blade exit (station m in fig. 2).

After expansion of the contents of the two mixing layers to the blade exit, the velocity of the total flow (coolant plus primary) is determined by constant-pressure, one-dimensional mixing of the contents of the two mixing layers and the unaffected isentropic primary flow at the blade exit plane.

When the velocity of the total flow at the blade exit, $V_{t,m}$, is known, the primary efficiency - without viscous, trailing-edge, and wake mixing losses - can be computed. (Primary efficiency is defined as the kinetic-energy output of the total flow relative to the ideal kinetic energy of the primary flow only.) Thus,

$$\eta_p = \frac{(1 + y)V_{t,m}^2}{V_{p,id,m}^2} \tag{3}$$

Having determined the primary efficiency without viscous, trailing-edge, and wake mixing losses, the fractional change in primary efficiency relative to the efficiency of the noncooled solid blade can be obtained from the expression

$$\frac{\Delta\eta_p}{\eta_o} = \frac{\eta_p}{\eta_o} - 1 \tag{4}$$

where $\eta_o = 1$, since to have consistent efficiencies for computing equation (4), the same viscous, trailing-edge, and wake mixing losses that were excluded from the primary efficiency of the cooled blade (eq. (3)) must be excluded from the efficiency of the noncooled solid blade.

Analysis of Reference 2

In the analysis of reference 2, the model depicted in figure 3 is used to determine the effect of coolant ejection from single rows and multiple rows of coolant holes on blade-row output. There are two differences between this analysis and that of reference 1 (or method 1) in the treatment of coolant discharge. First, the analysis of reference 1 assumes that the coolant and the primary flow mix at the location of coolant discharge; whereas the analysis of reference 2 assumes that the coolant and the primary flow mix at the blade-row exit. Second, in the analysis of reference 1, the same procedure is used for the expansion and the diffusion regions of the blade surface; whereas in the analysis of reference 2, different procedures are used for ejection from the expansion and diffusion regions. As shown in figure 4, the expansion and diffusion regions of the blade surface are defined as those regions where the blade-surface static pressure is higher or lower, respectively, than the blade-exit static pressure.

To determine the kinetic energy output of the blade row with coolant discharge, the velocities of the coolant and primary flows at the blade exit must be known.

Coolant exit velocity for expansion region. - To determine the coolant exit velocity in the expansion region, the total pressure of the coolant at the ejection location must be obtained. In determining the coolant total pressure, the effective dynamic pressure (dynamic head) of the coolant at the ejection location is first computed by assuming that only the component of coolant flow velocity parallel to the mainstream flow contributes to the total pressure of the coolant.

When the effective dynamic pressure of the coolant is known, the total pressure of the coolant at the ejection location is determined by adding the effective dynamic pressure to the blade-surface static pressure at the ejection location. By using the total pressure of the coolant at the ejection location and the blade-exit static pressure, the coolant velocity at the exit of the blade is computed with assumed isentropic expansion.

Coolant exit velocity for diffusion region. - With ejection from the diffusion region, the static pressure of the coolant at the location of discharge is less than that at the blade exit. To determine the coolant exit velocity in this region, the analysis of reference 2 assumes that the coolant velocity at the blade exit is equal to the component of coolant velocity parallel to the mainstream flow at the ejection location. With these assumptions, the energy required for compressing the coolant must, of course, be accounted for in the computation of the blade-row output after the coolant and the primary flows are mixed at the blade exit.

Mixed conditions at blade exit. - To determine the kinetic energy output of the total flow at the blade exit after mixing, the velocities of both the coolant and the primary flows at the blade exit are required. The procedures for obtaining the coolant velocities at the blade exit were described in the immediately preceding sections of the report. The velocity of the primary flow at the blade exit is determined by assuming isentropic expansion. With the coolant flow rates and the coolant and primary air velocities at the blade exit known, the velocity of the mixed flow at the blade exit is found by assuming one-dimensional, constant-pressure mixing of the coolant and primary flows. The net kinetic energy output of the total flow at the blade exit is then equal to the kinetic energy output of the mixed flow (as determined from the mixed velocity) less the isentropic compression energy required to increase the pressure of any coolant flow discharged in the diffusion region. Dividing the net kinetic energy output of the total flow at the blade exit by the ideal energy of only the primary flow yields the primary efficiency. Thus,

$$\eta_p = \frac{(1 + y)V_{t,m}^2 - \sum y_i V_{cp,id}^2}{V_{p,id,m}^2} \tag{5}$$

where the minus term in the numerator is the required compression energy.

When the primary efficiency is known, the fractional change in primary efficiency can be computed from equation (4) in the same manner as for method 1 (analysis of ref. 1).

Although, as mentioned, the analyses of references 1 and 2 for computing the effect of coolant discharge on blade-row output are somewhat different, the net effects of coolant discharge on blade-row output computed from both analyses are very similar.

Method 2

Method 2 was used to compute one of the two sets of predicted results presented later in this report and is the same as the analysis of reference 2, except that the compression-energy term of equation (5) was deleted. The compression term was deleted because comparisons of experimental and analytical results for the subject blade and other blades show that deletion of this term results in appreciably better agreement between the experimental and the predicted results. Considerable effort has been spent to find the cause for the better correlation when this term is deleted, but no physical reason has been found yet.

RESULTS AND DISCUSSION

A comparison is presented between experimental and predicted changes in efficiency due to coolant ejection from a film-cooled turbine stator blade having 12 spanwise rows of coolant holes. The comparison includes changes in efficiency due to coolant ejection from each of the 12 rows and from various combinations of rows including, full-film cooling.

The experimental results, previously published in references 13 and 14, were obtained from tests conducted in a two-dimensional cascade at a nominal ideal critical velocity ratio of 0.65 with the coolant and primary-air temperatures essentially equal to the ambient temperature. The predicted results were computed from the published analytical procedures of references 1 and 2. The two procedures herein called method 1 and method 2, respectively, are described under PREDICTION METHODS.

Method 1 was the analytical procedure of reference 1 as published. Method 2 was a modified form of the analytical procedure of reference 2. Comparative experimental and predicted results are given in terms of percent change in primary efficiency relative to the efficiency of the solid, noncooled blade. Primary efficiency is defined as the ratio of the actual kinetic energy of the total flow (primary plus coolant) relative to the ideal energy of the primary flow only. In comparing the results at various coolant and primary flow conditions, the condition of equal total pressures of the coolant and primary flows is considered the most significant, since it is representative of the condition in the first turbine stage of a gas turbine engine.

The comparative results are presented in three parts. The first part concerns single-row and multiple-row coolant ejection from the expansion region of the blade suction and pressure surfaces; the second part, single-row ejection from the diffusion region of the blade suction surface; and the third part, multiple-row ejection from the suction surface and also from all 12 rows (full-film cooling).

Comparison of Results for Ejection From the Expansion Region of Blade Surface

Single-row results. - The comparative experimental and predicted changes in primary efficiency relative to the efficiency of a noncooled blade ($\Delta\eta_p/\eta_p$) as a function of coolant fraction y for the nine test cases of single-row ejection from the expansion region of the blade surface (i.e., rows 1 to 9, figs. 1(c) and 4) are shown in figure 5. The two analytical methods give the same results for ejection from all rows except rows 1 and 7. For these two rows, there is a small difference of about 1/2 percent in the upper range of coolant fractions between the results predicted by the two methods. This difference is attributed to the fact that rows 1 and 7 have ejection angles of 90°, whereas the other rows have nominal ejection angles of 35°. Small differences between results predicted by the two methods for cases of coolant ejection from the expansion region are maximum when the coolant is ejected normal to the blade surface.

The agreement between experimental results and the results predicted by both analytical methods is considered very good for all cases of single-row ejection from the expansion region except from rows 6 and 9 (figs. 5(f) and 5(i)). Except for rows 6 and 9, the maximum difference between experimental and predicted results over the range of tested coolant fractions is less than 0.5 percent with both methods. For ejection from rows 6 and 9 (figs. 5(f) and 5(i)), the maximum differences between predicted and experimental results are somewhat larger, being 0.6 percent less than experimental for row 6, and 0.9 percent less than experimental for row 9. These differences, which are the same with both prediction methods, occur at the condition when the total pressures of the primary and coolant flows are equal ($p'_{p,0} = p'_c$). The reason for the larger differences for ejection from rows 6 and 9 than for the other seven rows is perhaps related to the fact that the local static pressures at these two rows are close to the blade-exit static pressure (see fig. 4).

Multirow results. - Figure 6 presents the comparative experimental and predicted efficiency changes for the five test cases of multirow coolant ejection from the expansion region of the blade pressure surface. For ejection from 2 to 5 rows on the pressure surface (figs. 6(a) to 6(d)), the results predicted by both methods agree very well with experimental results for the range of coolant fractions considered. When $p'_{p,0}$ equals p'_c, the experimental results and predicted results from both methods agree very closely. The maximum differences occur in the upper range of coolant fractions and are about 1/4 percent for method 1 and about 3/4 percent for method 2.

As shown in figure 6, the agreement between experimental and predicted results for multirow coolant ejection from 2 to 5 rows on the pressure surface is quite good. The plots of figure 5 also showed that the agreement between experimental and predicted results is quite good for ejection from the associated single rows 1 to 5. This indicates that the single-row and multirow results are related. Such a relationship is confirmed in references 13 and 14, where it is shown that experimental multirow efficiency changes closely approximate the efficiency changes obtained by properly adding the associated experimental single-row efficiency changes. Further, multirow efficiency changes of both prediction methods are obtained by adding predicted single-row results in the same or similar manner to that used in references 13 and 14 for adding experimental single-row results. Consequently, in this report, if single-row experimental and predicted results agree, the multirow results with ejection from corresponding rows will also agree. Conversely, if the single-row results do not agree, the multirow results will also not agree by an amount roughly equal to the algebraic difference between the associated single-row results.

For ejection from all six rows on the pressure surface (fig. 6(e)), the agreement between predicted and experimental results is poorer than for the other cases of multirow ejection from the pressure surface. When $p'_{p,0}$ equals p'_c, the efficiency changes predicted by both analytical methods are about 3/4 percent less than experimental, and in the upper range of coolant fractions, this difference between predicted and experimental results gradually increases to roughly 3 percent less than experimental for both methods.

As mentioned, the plots of figure 5 showed good agreement between experimental and predicted results for ejection from all single rows on the pressure surface except row 6. For row 6, when $p'_{p,0}$ equals p'_c, the efficiency change predicted by both methods was 0.6 percent less than the experimental result. This compares with the predicted change of about 3/4 percent less than the experimental result for multirow ejection from all six rows. Again, this shows that differences in predicted and experimental single-row results are reflected in the differences in predicted and experimental multirow results.

Figure 7 shows the comparative experimental and predicted results for multirow ejection from two and three rows in the expansion region of the forward part of the blade suction surface. The results for both cases show good agreement between the trends of experimental and predicted changes and fair agreement between the levels of experimental and predicted changes. Also, the agreement between experimental and predicted changes is slightly better for method 2 than for method 1.

Figure 7 shows that the agreement between experimental and predicted results is better for two-row coolant ejection than for three-row ejection. When $p'_{p,0}$ equals p'_c, with two-row ejection the predicted efficiency changes are about 1/10 percent less than the experimental results for method 2 and about 1/4 percent less than experimental for method 1. With three-row ejection at the same condition, the predicted efficiency changes are about 1/2 percent less than the experimental results for method 2 and about 3/4 percent less than experimental for method 1. These differences in the predictions of the two analytical methods are again a reflection of the differences in the single-row ejection results shown in figure 5.

Figure 8 shows a comparison of predicted and experimental efficiency changes for four-row ejection from the expansion region of the blade surface. (The four rows are the two nearest the leading edge on the suction surface and the two nearest the leading edge on the pressure surface.) The agreement between the experimental results and the results predicted by both methods is good. In the lower range of coolant fractions, agreement is better with method 2, while in the upper range, agreement is better with method 1.

In summary, the effect of coolant ejection from the expansion region of the blade surface on blade-row output is predicted very well by both analytical methods. Neither method is clearly better than the other.

Comparison of Single-Row Results for Ejection From the Diffusion Region of Blade Surface

The experimental and predicted efficiency results for the three test cases of single-row ejection from the diffusion region of the blade surface are compared in figure 9. The experimental and predicted trends agree well for both methods; however, the level of experimental results is predicted considerably better by method 2 than by method 1. For these three cases, when $p'_{p,0}$ equals p'_c, the efficiency changes predicted by method 2 are between 1/10 and 1/2 percent less than the experimental results;

while the changes predicted by method 1 range between 1 and $1\frac{1}{4}$ percent less than the experimental results. The better agreement with method 2 than with method 1 is due to the different manners in which the two methods treat the effect of coolant ejection in the diffusion region. Method 2 ignores the compression energy required to pump the coolant flow to the blade-exit static pressure, whereas method 1 includes it.

Comparison of Multirow Results for Ejection From the Suction Surface and Also From All 12 Rows (Full-Film Cooling)

Figure 10 compares the experimental and predicted efficiency changes for the three test cases of multirow ejection from the suction surface. Results are shown for four-, five-, and six-row ejection from the suction surface, with three of the rows in the expansion region and either one, two, or three rows in the diffusion region (see figs. 1(c) and 4).

Figure 10 shows that both analytical methods predict the trends of experimental efficiency changes very well. However, as expected from single-row differences, the level of experimental efficiency changes is predicted considerably better by method 2 than by method 1. For instance, when $p'_{p,0}$ equals p'_c, for four-, five-, and six-row ejection from the suction surface, the efficiency changes predicted by method 2 are about 1/4 to 1 percent less than the experimental results, whereas the changes predicted by method 1 are about 2 to $4\frac{1}{2}$ percent less than the experimental results. The maximum differences for both prediction methods occurred with six-row ejection.

Figure 11 compares the results for ejection from all 12 rows on both the suction and pressure surfaces (full-film cooling). When $p'_{p,0}$ equals p'_c, the results predicted by method 2 are about 2 percent less than the experimental results, while those predicted by method 1 are about 6 percent less than the experimental results. Again, the principal reason for the better agreement with method 2 than with method 1 is that method 2 ignores the compression energy required by the coolant ejected in the diffusion region of the blade surface.

CONCLUDING REMARKS

This comparison of predicted and experimental results for one film-cooled blade geometry indicates that the two prediction methods used herein may be usable as guides in the aerodynamic design of cooled blades. However, before too much credence is placed in the accuracy of either method, both should be tested further by comparisons with other experimental results for a variety of cooled-blade configurations at various temperature levels. If the methods were proven, they could serve as valuable design tools for greatly reducing the amount of experimental work required to optimize cooled-blade designs.

SUMMARY OF RESULTS

A comparison was made between experimental and predicted changes in efficiency due to coolant ejection from a film-cooled turbine stator blade having 12 spanwise rows of coolant holes. The comparison included changes in efficiency due to coolant ejection from each of the 12 rows and from various combinations of rows, including full-film cooling.

The previously published experimental results were obtained from tests conducted in a two-dimensional cascade at a nominal, ideal, primary-air critical velocity ratio of 0.65 with the coolant and primary-air temperatures essentially equal to the ambient temperature.

The predicted results were computed by two published analytical procedures, herein called method 1 and method 2. Method 1 was used as published. Method 2 was modified from the published procedure for ejection from the diffusion region of the blade surface. The expansion and diffusion regions of the blade surface are defined as the parts of the blade surface where the local static pressure is higher or lower, respectively, than the blade-row exit static pressure.

Experimental and predicted results are compared in terms of the percent change in primary efficiency relative to the efficiency of the uncooled (solid) blade. Primary efficiency is defined as the ratio of the actual kinetic energy of the total flow (primary plus coolant) relative to the ideal kinetic energy of the primary flow only. The comparisons are summarized for the condition of equal total pressures of the coolant and primary flows. The following are the results of the comparisons:

1. For multirow ejection with simultaneous coolant discharge from both the expansion and diffusion regions of the blade surface, both analytical methods predicted too small an increase in efficiency compared to experimental results. However, method 2 predicted the efficiency changes better than did method 1. For example, with full-film cooling, which had 3 of the 12 rows of coolant holes in the diffusion region, the efficiency increases predicted by method 2 were about 2 percent less than the experimental results, whereas the predictions of method 1 were about 6 percent less than the experimental results.

2. The major reason for the better agreement with method 2 than with method 1 for full-film cooling is that method 2 better predicts experimental results for ejection from the diffusion region of the blade surface. That is, for single-row ejection from

the diffusion region of the blade surface, the results predicted by method 2 ranged from 1/10 to 1/2 percent less than the experimental results, while those predicted by method 1 ranged from 1 to $1\frac{1}{4}$ percent less than experimental results. This difference between the predictions of the two methods for single-row ejection from the diffusion region is reflected in all cases of multirow ejection that include coolant discharge in the diffusion region. An additional example, besides full-film cooling, is the case of six-row ejection from the suction surface. For this case, which also has three rows of coolant holes in the diffusion region, the results predicted by method 2 were about 1 percent less than the experimental results, while the predictions of method 1 were about $4\frac{1}{2}$ percent less than the experimental results.

3. For all cases of single- and multirow ejection in the expansion region of the blade surface, both methods predicted the experimental results quite well. That is, differences between experimental and predicted results for all cases were less than 1 percent, and for most cases they were less than 1/4 percent.

4. The experimental trends of efficiency change as a function of coolant fraction were well predicted by both analytical methods.

Lewis Research Center,
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TABLE I. - COOLANT-HOLE SPECIFICATIONS

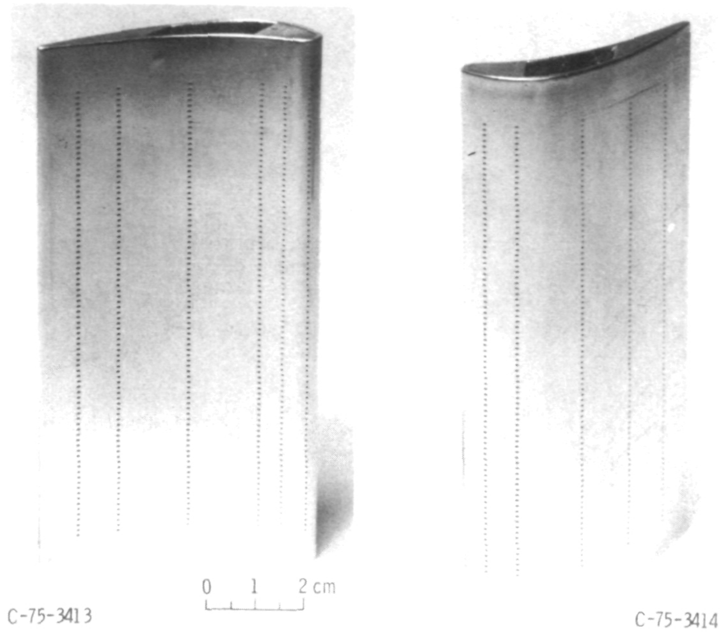
[See fig. 1(c) for geometry.]

Coolant-hole row	Distance from leading edge to hole row as fraction of pressure- or suction-surface length, x/L_{pr} or x/L_s	Angle between coolant-hole axis and tangent to local blade surface in plane parallel to blade end walls, β, deg	Coolant-hole length-to-diameter ratio, L/D
1	0.035	90	2.2
2	.12	34	3.7
3	.20	33	3.3
4	.45	35	↓
5	.70	33	↓
6	.85	34	↓
7	.035	90	2.2
8	.105	36	3.7
9	.20	39	4.5
10	.40	38	4.0
11	.60	38	3.8
12	.80	35	3.8

TABLE II. - COOLANT-HOLE MULTIROW CONFIGURATIONS TESTED

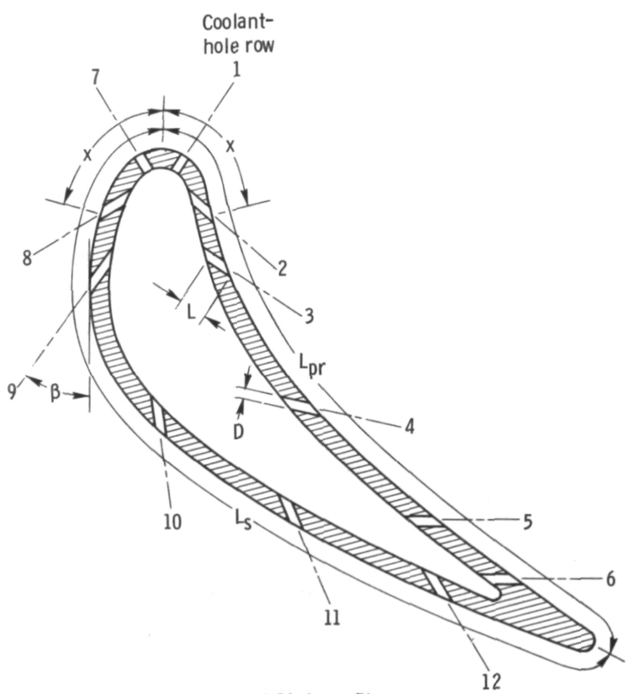
[See fig. 1(c) and table I for geometry and specifications.]

Configuration	Coolant-hole rows included	Region of blade surface
1	1 and 2	Pressure surface
2	1 to 3	↓
3	1 to 4	↓
4	1 to 5	↓
5	1 to 6	↓
6	7 and 8	Suction surface
7	7 to 9	↓
8	7 to 10	↓
9	7 to 11	↓
10	7 to 12	↓
11	1, 2, 7, and 8	Both pressure and suction surfaces
12	1 to 12	Both pressure and suction surfaces



(a) Pressure surface.

(b) Suction surface.



(c) Blade profile.

Figure 1. - Film-cooled stator blade.

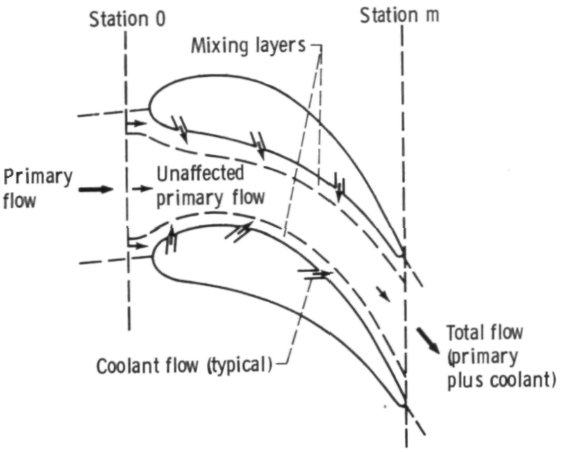


Figure 2. - TOTLOS model of reference 1 for determining output of blade row with surface coolant ejection.

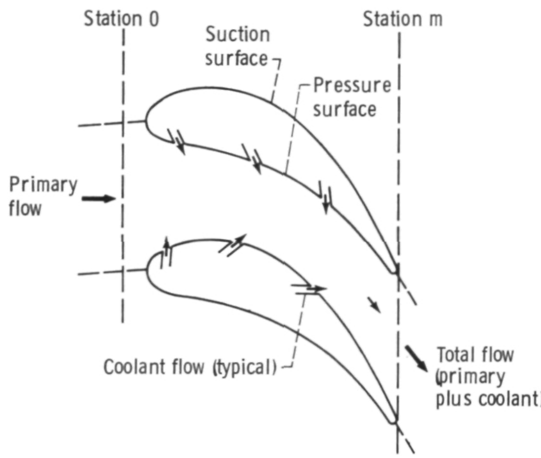


Figure 3. - Model of reference 2 for determining output of blade row with surface coolant ejection.

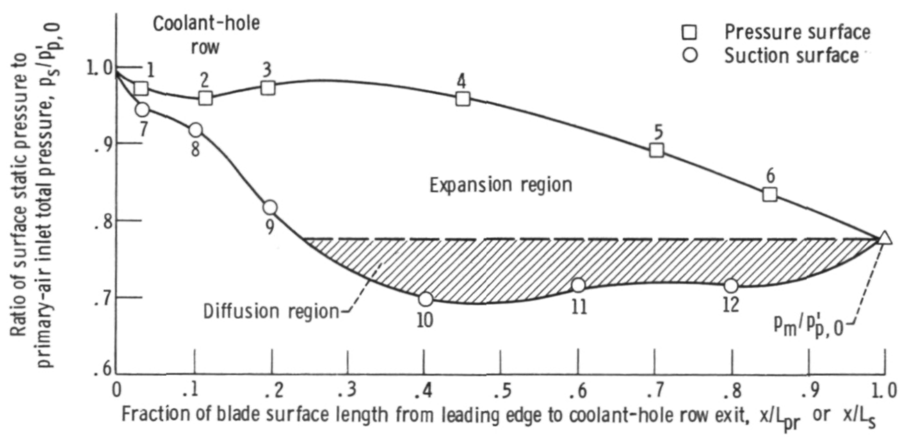


Figure 4. - Comparison of coolant-hole-row local static pressures on suction and pressure surfaces of blades for primary-air critical velocity ratio $(V/V_{cr})_{p,id,m}$ of 0.65.

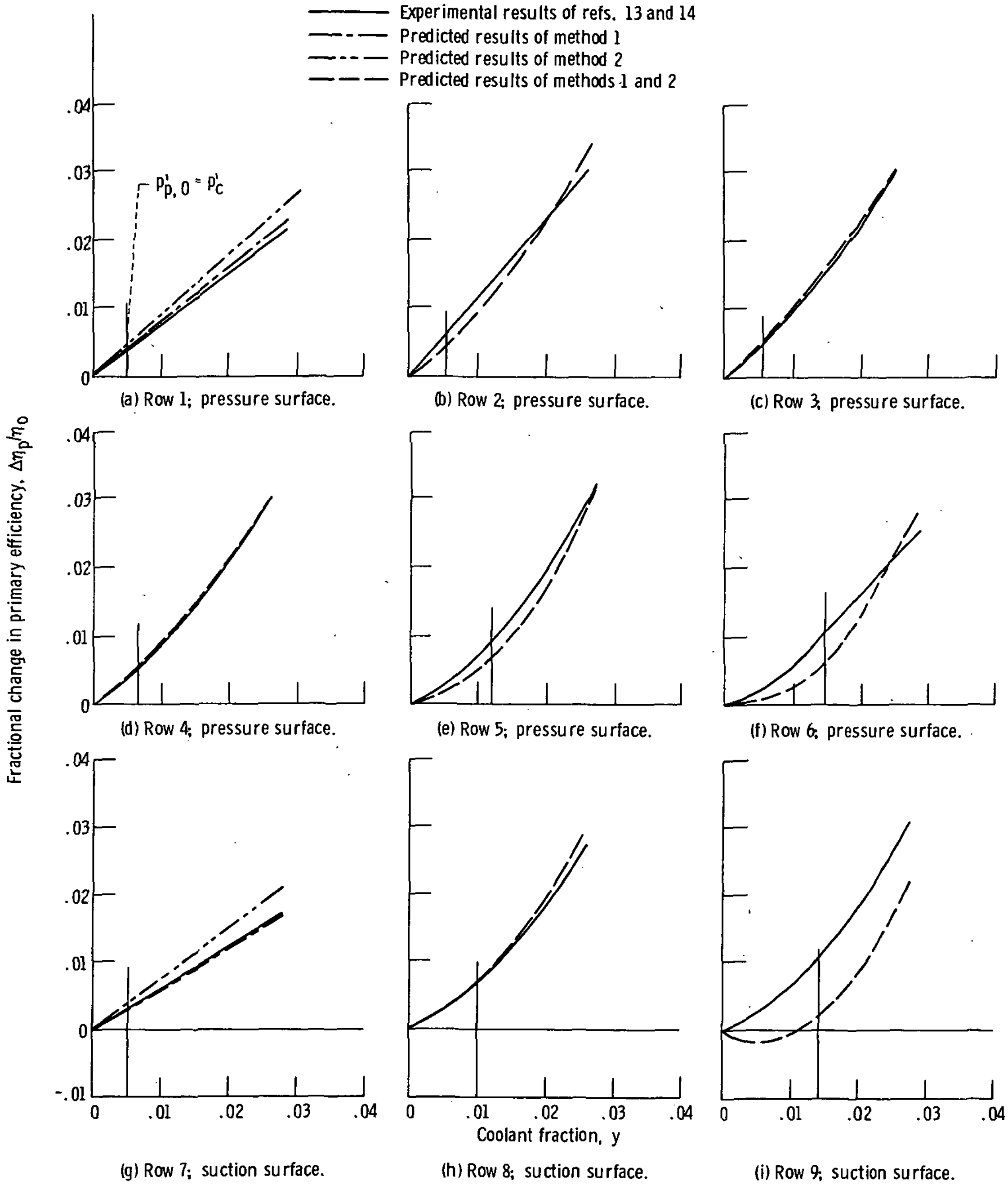
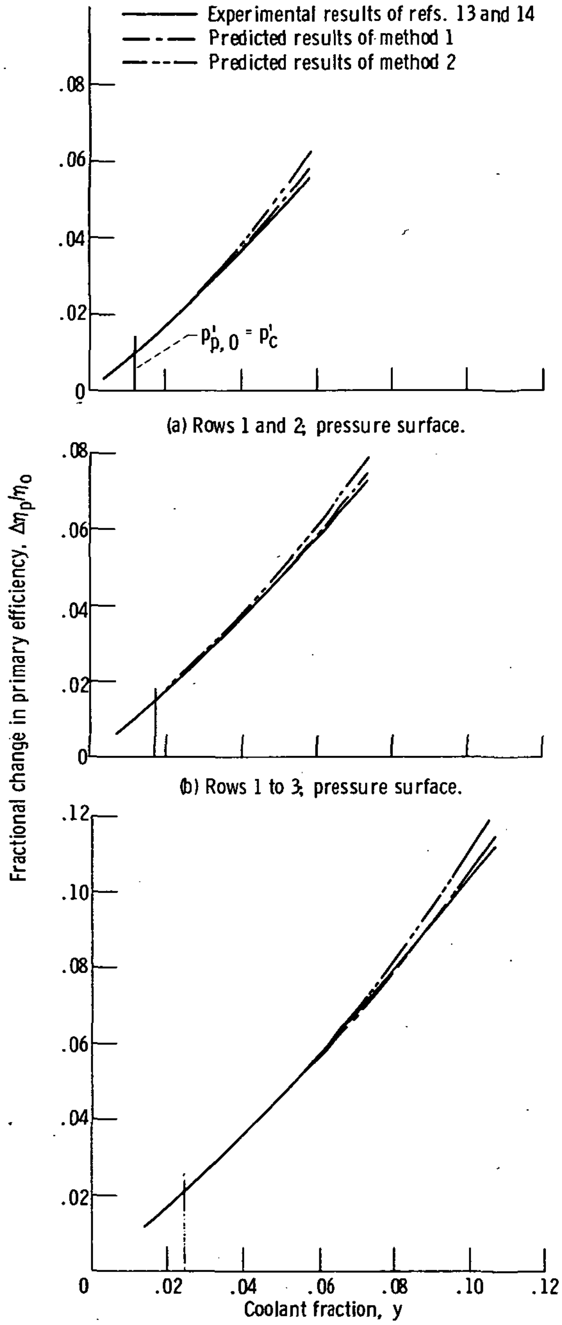
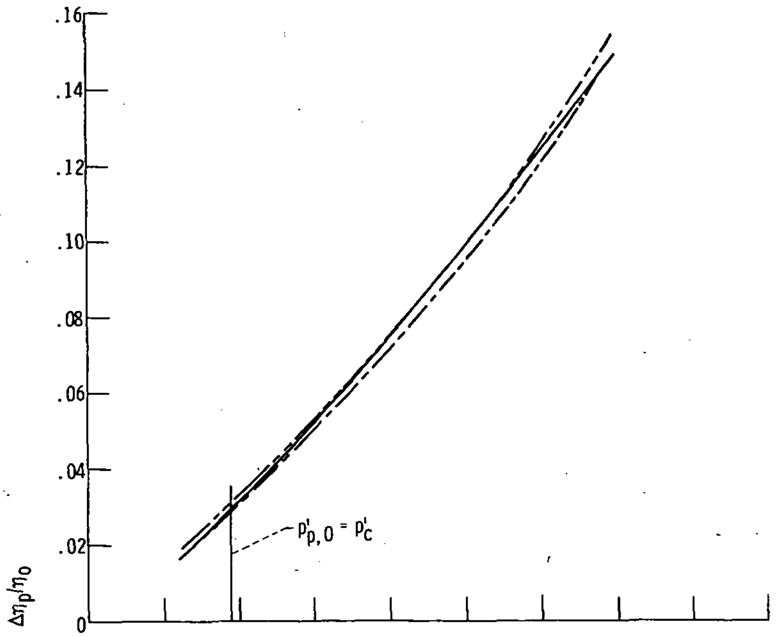


Figure 5. - Comparison of results for single-row ejection from the expansion region of the blade surface.

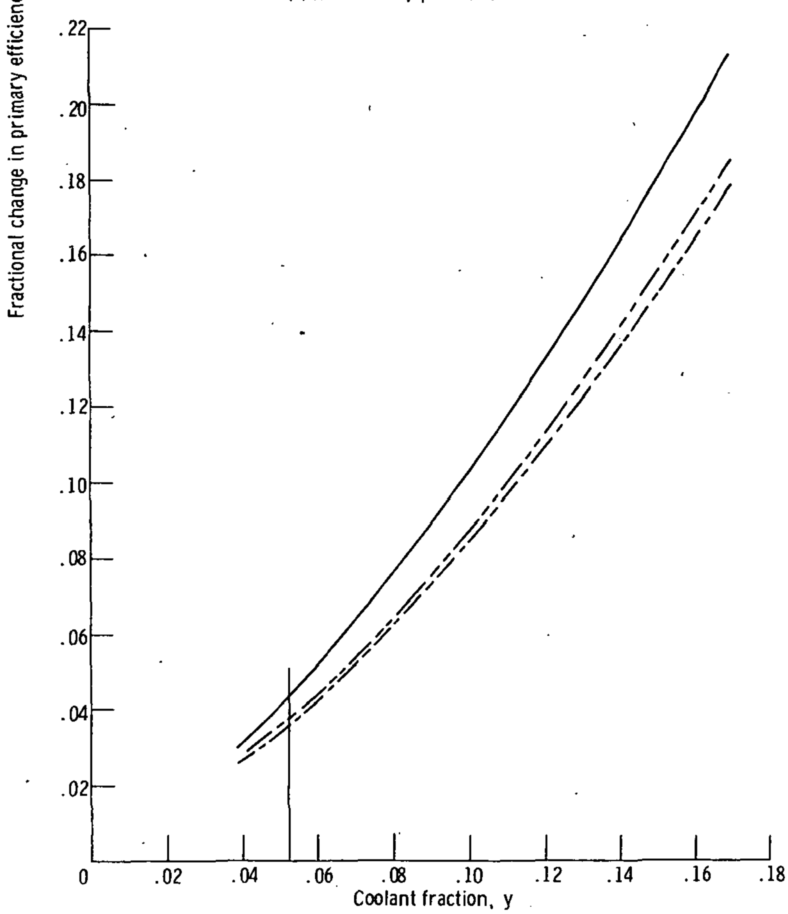


(c) Rows 1 to 4; pressure surface.

Figure 6. - Comparison of results for multirow ejection from the expansion region of the blade pressure surface.

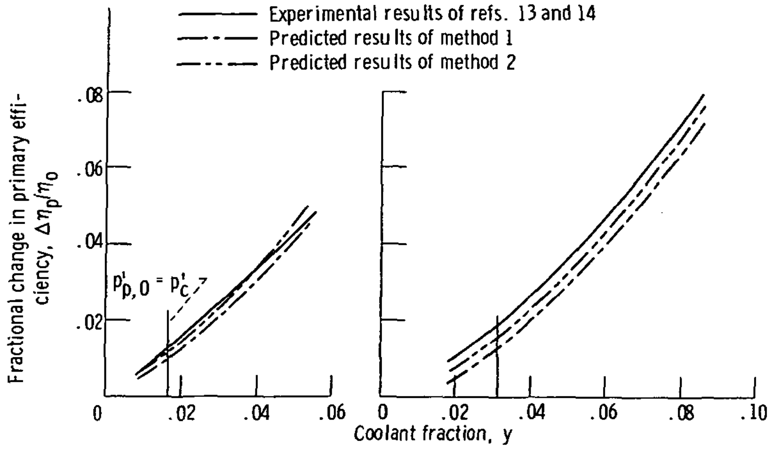


(d) Rows 1 to 5; pressure surface.



(e) Rows 1 to 6; pressure surface.

Figure 6. - Co[illegible]ed.



(a) Rows 7 and 8; suction surface.

(b) Rows 7 to 9; suction surface.

Figure 7. - Comparison of results for multirow ejection from the expansion region of the blade suction surface.

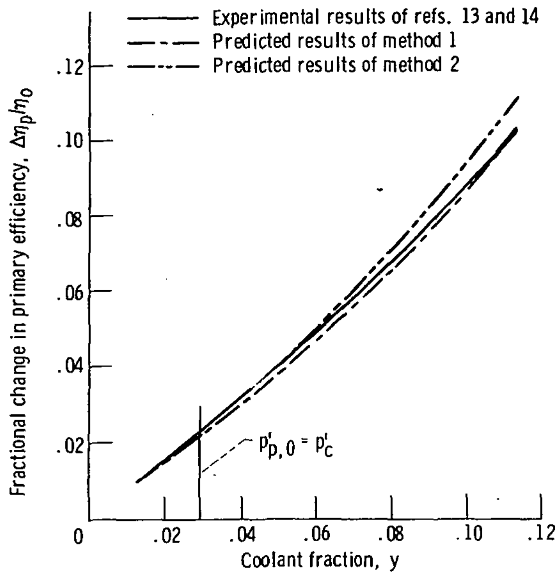
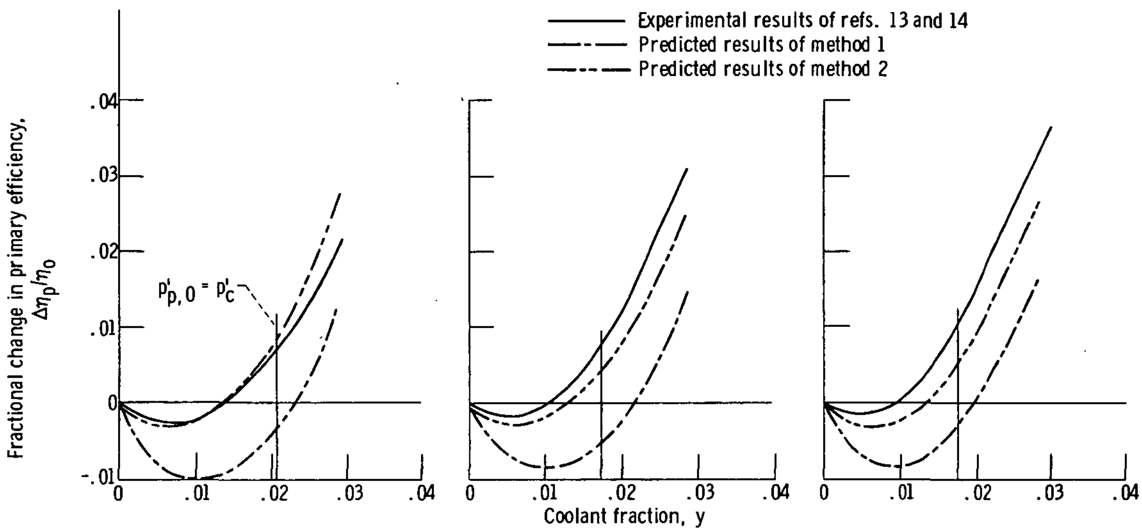


Figure 8. - Comparison of results for multirow ejection from the forward portions of both the suction and pressure surfaces of the blade (rows 1, 2, 7, and 8).



(a) Row 10; suction surface. (b) Row 11; suction surface. (c) Row 12; suction surface.

Figure 9. - Comparison of results for single-row ejection from the diffusion region of the blade suction surface.

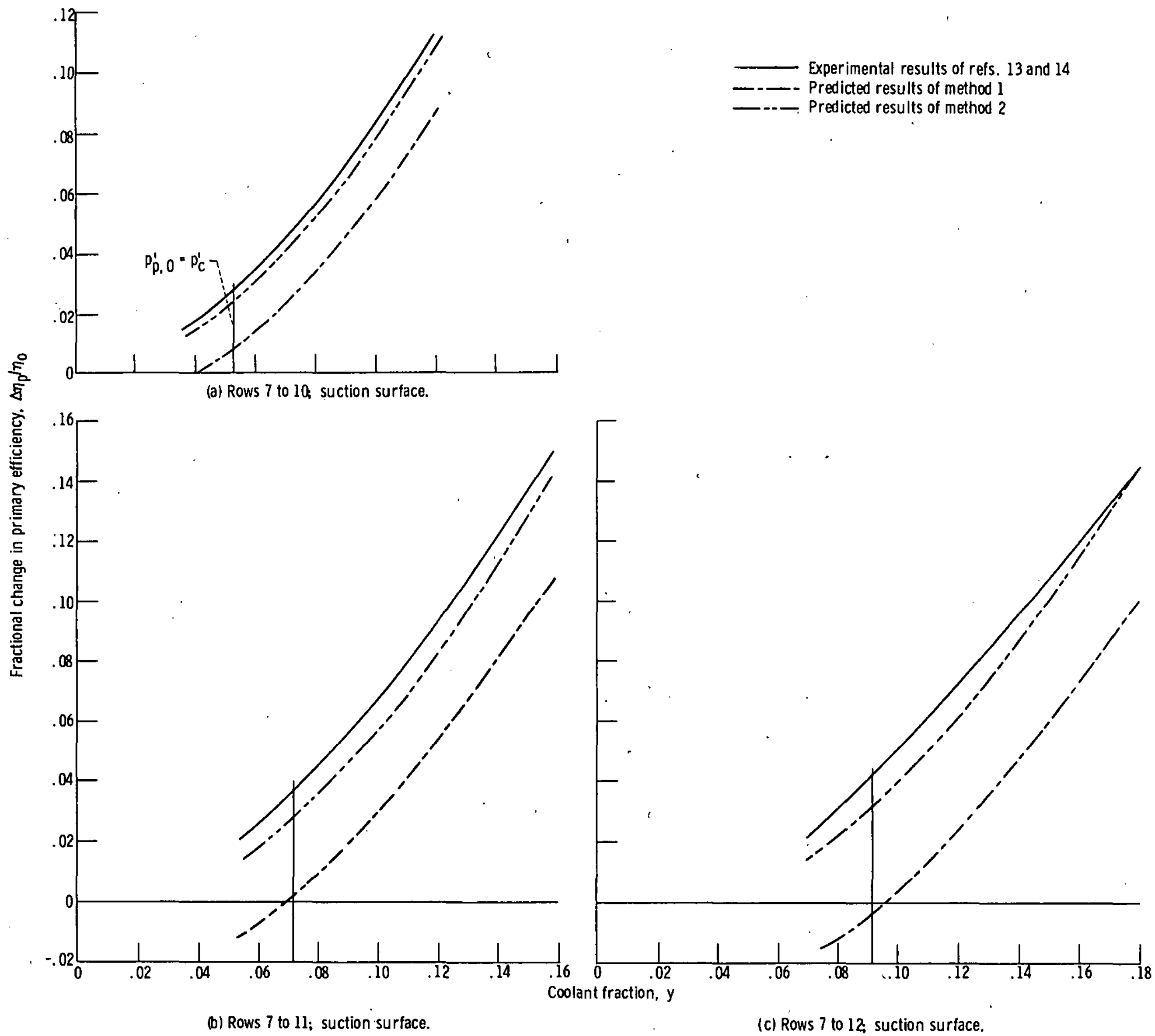


Figure 10. - Comparison of results for multirow ejection with simultaneous discharge from the expansion and diffusion regions of the blade surface.

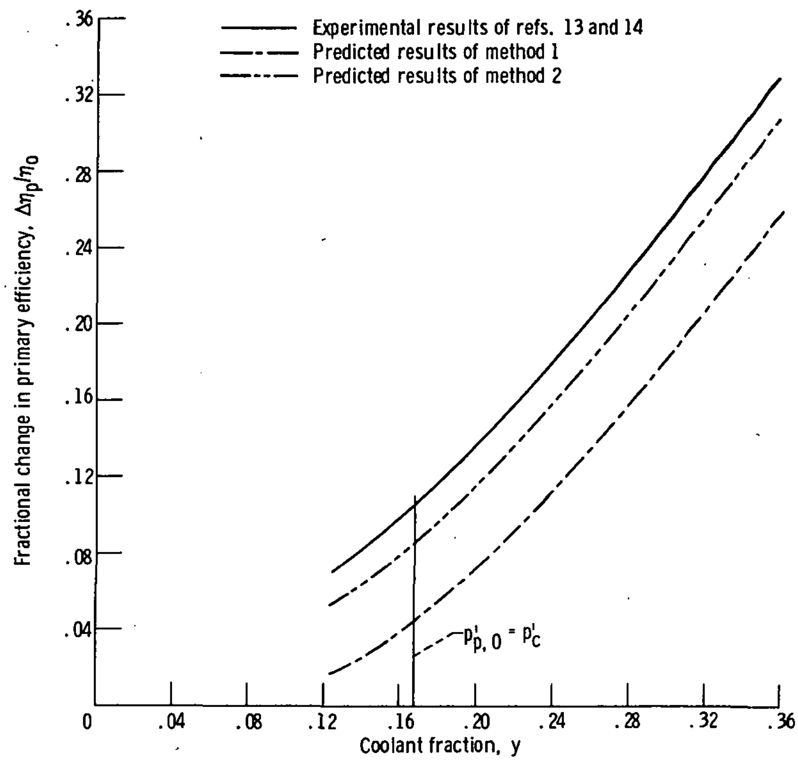


Figure 11. - Comparison of results for ejection from all 12 coolant rows (full-film cooling).

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16. Abstract

Previously published experimental aerodynamic efficiency results for a film-cooled turbine stator blade are compared with analytical results computed from two published analytical methods. One method was used as published; the other was modified for certain cases of coolant discharge from the blade suction surface. For coolant ejection from blade surface regions where the surface static pressures are higher than the blade exit pressure, both methods predict the experimental results quite well. However, for ejection from regions with surface static pressures lower than the blade exit pressure, both methods predict too small a change in efficiency, but the modified method gives the better prediction.

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