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COLD-AIR STUDY OF THE EFFECT ON TURBINE STATOR BLADE AERODYNAMIC PERFORMANCE OF COOLANT EJECTION FROM VARIOUS TRAILING-EDGE SLOT GEOMETRIES

II - Comparison of Experimental and Analytical Results

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16. Abstract
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SUMMARY

An investigation was made to compare experimentally determined efficiencies of turbine stator blades having trailing-edge coolant ejection with efficiencies predicted from two previously published approximate analytical methods. The experimental results, also previously published, were obtained from two-dimensional data with the temperatures of the coolant and primary flows both being nearly ambient. Five stator blade configurations having two different trailing-edge thicknesses and four different trailing-edge slot widths were included in the comparison. The data covered coolant- to primary-air exit-velocity ratios from 0 to 1.2.

An average of the absolute values of differences between experimental and predicted efficiencies for all five blade configurations showed that one of the methods gave average efficiency differences which were about 1.3 percent different than experimental efficiencies, while the other method gave average efficiency differences that were about 0.7 percent different than experimental efficiencies. However, maximum differences of as much as 4 percent occurred between predicted and experimental efficiencies for the two bladings with larger slot widths. Differences of this magnitude are certainly larger than desired.

In general, the agreement between experimental and analytical results was considerably better for the two blade configurations having the smallest trailing-edge slot width. One of the analytical methods gave results which agreed better with experimental results for the two bladings with the smallest slot width. The other analytical method agreed better with experimental results for the three bladings with larger slot widths.

The comparison between experimental and analytical results strongly indicate that the ratio of trailing-edge slot width to trailing-edge thickness influences the measured efficiencies to a greater extent than is accounted for by either of the analytical models. Therefore, an empirical prediction method was derived from the experimental results; however, the method may not be applicable to other stator blading.

INTRODUCTION

Several analytical and experimental studies on the performance of cooled turbines (e.g., refs. 1 to 12) have shown that different means of ejecting compressor bleed coolant air from the turbine blade surface cause significantly different effects on turbine efficiency.

Since high turbine efficiency is important in most engine designs, an extensive research program is in progress at the Lewis Research Center to investigate both experimentally and analytically the effect of different means of coolant ejection on turbine efficiency as well as other aspects of turbine performance.

In previous experimental investigations, several means of coolant ejection have been investigated (see refs. 3 to 11). In references 4 to 6, the results of a three-dimensional experimental and analytical investigation of the influence on turbine stator blade and stage performance of stator blade trailing-edge coolant ejection from a particular slot geometry are reported. The main conclusion of the investigation of references 4 to 6 was that coolant flow ejected from a particular trailing-edge slot parallel to the main stream significantly increased the turbine work output.

The experimental part of the investigation of references 4 and 5, to determine the effect on stator performance of coolant ejection from a particular trailing-edge slot geometry, has now been extended to include the effect on stator blade performance of coolant ejection from five different trailing-edge slot configurations. The results of the extended experimental investigation, which was conducted in a two-dimensional cascade, are reported in reference 12. The general finding of the investigation of reference 12 for the five test blade configurations was that the average percent change in kinetic-energy output per percent coolant flow varied approximately linearly from 0 to 1.4 percent over a range of coolant- to primary-air exit velocity ratios from 0 to 1.2. However, there was considerable variation from these average values between the different blade configurations, particularly in the lower range of exit velocity ratios.

This report presents a comparison between the experimentally determined efficiencies with trailing-edge coolant ejection reported in reference 12 and efficiencies computed from two approximate analytical prediction methods. The analytical methods used are those of reference 1, as published, and reference 2, with minor modifications. In addition, an empirical prediction method based on the experimental results of reference 12 is developed and presented.

The comparison of results are reported principally in terms of percent change in stator blade primary-air efficiency relative to the noncooled blading as a function of coolant flow rate. The stator blade primary-air efficiency is defined as the ratio of the actual kinetic energy output of the total flow to the ideal kinetic energy output of the primary flow only.

DESCRIPTION OF TESTED BLADING

A photograph of the different test blading having five trailing-edge configurations is presented in figure 1. As indicated, the blading is hollow and of constant cross section.

A cross-sectional sketch showing the geometry and significant dimensions of the five different trailing-edge slot configurations is presented in figure 2. Two of the five test blade configurations had a trailing-edge thickness of 0.178 centimeter (0.070 in.) with coolant slot widths of 0.051 and 0.102 centimeters (0.020 and 0.040 in.). (These are shown on the left side of fig. 1.) The other three configurations had a trailing-edge thickness of 0.330 centimeter (0.130 in.) with coolant slot widths of 0.051, 0.127, and 0.203 centimeters (0.020, 0.050, and 0.080 in.). (These are shown on the right side of fig. 1.) As indicated, the slots for all the blading were machined through round trailing edges. As shown in figure 1, all the coolant slots had structural support webs spaced at spanwise intervals. The slot length between webs was 1.969 centimeters (0.775 in.) in the test area near the mean section. The spanwise web widths were 0.127 centimeter (0.050 in.), and the lengths of the slots and the webs, in the direction of coolant flow, were the same.

Except for the incorporation of trailing-edge slots, the blading with the thinner trailing edges corresponds to the mean section of the stator blading of reference 13. Detailed dimensions and geometry of the blading may be found in that reference. Some significant dimensions of the blading are: span, 10.16 centimeters (4.0 in.); chord, 5.74 centimeters (2.26 in.); pitch, 4.14 centimeters (1.63 in.). The blading with the thicker trailing edges was modified so as to have the same flow path (except at the leading and trailing edges) as the blading with the thinner trailing edges. Details of the method of modification are given in reference 14.

ACCURACY OF EXPERIMENTAL RESULTS

The maximum probable error in experimentally determined values of primary-air efficiency obtained from several hundred tests of blading without coolant flow is about ±0.25 percent.

With coolant flow, the maximum probable error in determining the actual kinetic energy output of the total flow at the particular spanwise location being tested is also about ±0.25 percent. However, as discussed in reference 12, there was some question concerning even distribution of coolant flow both among the test blades and also spanwise through the coolant slots. Uneven distribution of coolant flow would result in inconsistencies in experimentally determined values of primary-air efficiency since the total kinetic energy was determined at the mean section of the blading only, whereas the ideal kinetic energy output of the primary flow is computed by assuming even distribution of the coolant flow.

As a result of the possible maldistribution of coolant flow, the author has some reservations concerning the absolute accuracy of the experimental efficiencies; however, the trends of efficiency with coolant flow rate and the relative efficiencies of the bladings with different slot configurations agree, with one exception, with what would logically be expected (see ref. 12).

ANALYTICAL METHODS

This section of the report is presented in considerable detail to provide convenient background material for understanding the differences in results between the two analytical methods used herein and the experimental results presented later in the report. It is presented in three parts. In the first part, the analytical method of reference 1 is presented and described. In the second part, the analytical method of reference 2, as modified to provide for recently determined differences in trailing-edge drag coefficients resulting from different trailing-edge geometry (ref. 14), is presented and described. And in the last part, differences in theories of the two methods are discussed.

Following the precedent of reference 12, which reports the experimental results for trailing-edge coolant discharge, the results of this report are presented in terms of primary-air efficiency which is defined as the actual kinetic energy output of the total flow relative to the ideal energy of the primary flow at the hypothetical downstream station where flow conditions are uniform. Thus,

$$\eta_{p,3} = \left(\frac{m_p + m_c}{m_p}\right)\left(\frac{V}{V_{p,i}}\right)_3^2 = (1 + y)\left(\frac{V}{V_{p,i}}\right)_3^2 \qquad (1)$$

where y is the ratio of the mass flow of the coolant to the mass flow of the primary. (Symbols are defined in appendix A.)

Description of Analytical Method of Reference 1

The method of reference 1 for predicting primary-air efficiency with trailing-edge coolant ejection is based directly on that of reference 15 with some modification to include a coolant jet in the blade trailing edge. The effects on primary-air efficiency of trailing-edge blockage and coolant energy are included in the method. Viscous losses are included; however, they are considered to be constant so any effects of coolant ejection on viscous losses are not accounted for. The two-dimensional model used in developing the method is shown in figure 3. (In this report, the height of the trailing-edge coolant jet $s\ell$ at station 1 was assumed equal to the physical trailing-edge slot width.) Referring to the model, the first step in developing the method is to compute the uniform downstream fluid conditions at station 3 from the conditions existing at the trailing-edge station 1. To do this, the mass flow, tangential momentum, and axial momentum at the aftermixed station 3 are equated to the same quantities at the trailing-edge station 1. Thus,

$$(\cos\,\alpha_3)(\rho V)_3 = (\cos\,\alpha_1)(\rho V)_{p,i,1}\left[(1 - \delta_{te} - \delta^*) + (\delta_{s\ell})(\rho V)_{c,1}\right] \tag{2}$$

$$(\cos\,\alpha_3)(\sin\,\alpha_3)(\rho V^2)_3 = (\cos\,\alpha_1)(\sin\,\alpha_1)\left[(\rho V^2)_{p,i,1}(1 - \delta_{te} - \delta^* - \theta^*) + (\delta_{s\ell})(\rho V^2)_{c,1}\right] \tag{3}$$

$$(\cos^2\alpha_3)(\rho V^2)_3 + p_3 = \left(\cos^2\alpha_1\right)\left[(1 - \delta_{te} - \delta^* - \theta^*)(\rho V^2)_{p,i,1} + (\delta_{s\ell})(\rho V^2)_{c,1}\right] + p_1 \tag{4}$$

By using the equations of energy and state, the simultaneous system of equations may be solved directly to obtain any one of the unknown variables α_3, ρ_3, p_3, or V_3 at the mixed station. After one of the variables is determined, the other variables and then the primary-air efficiency may be determined from standard relations.

Description of Analytical Method of Reference 2

The prediction method of reference 2 (slightly modified as described in appendix B) considers the change in primary-air efficiency due to trailing-edge coolant ejection to result from two effects. One is that the coolant flow contributes to the kinetic energy of the total flow leaving the blade row. The other is that the discharge of coolant flow from the trailing edge results in a trailing-edge loss that is smaller than that of a comparable uncooled blade. The blade surface friction loss (viscous losses) are assumed to be unaffected by trailing-edge coolant ejection which implies that the efficiency of the primary flow is unaffected by the coolant flow.

Considering first the effect of the coolant flow kinetic energy on the change in output of the uncooled blading, the following assumptions are made: (1) the coolant flow mixes with the primary flow without loss due to exchange of momentum between the two streams, (2) the mixing occurs at constant static pressure equal to that of the trailing-edge plane, and (3) the aftermixed static pressure is equal to that of the trailing-edge plane. With these assumptions, the effect of the coolant flow energy on the change in primary-air efficiency relative to the uncooled blade row is equal to the ratio of the kinetic energy of the coolant flow to the kinetic energy of the primary flow at the trailing-edge plane. Thus,

$$\left(\frac{\Delta\eta_{p,c,e}}{\eta_o}\right)_3 = (y)\left(\frac{V_c}{V_p}\right)_1^2 \tag{5}$$

Next, considering the effect of trailing-edge coolant discharge on the blade trailing-edge loss, reference 2 assumes that the flow of coolant from the trailing-edge slot reduces the momentum deficit occurring in the trailing-edge region and thus reduces the trailing-edge loss that occurs in the absence of coolant flow. To determine the effect of coolant flow on trailing-edge loss then requires that the trailing-edge loss with and without coolant flow be determined.

The maximum trailing-edge loss that can result occurs without coolant flow. This loss results from the trailing-edge geometry of the blading without the slot. In reference 2, this loss is computed using a modification of an equation for base drag developed in reference 16. Thus,

$$\left(\frac{\Delta\eta_{te}}{\eta_o}\right)_3 = 0.340\left[\frac{t}{(\bar{e})(th)}\right]^{1/3}\left(\frac{t}{th}\right)c_{D,o} \tag{6}$$

The minimum trailing-edge loss that can result for the primary flow is assumed to be dependent upon the geometry and thickness of the two halves of the trailing-edge slot walls adjacent to the primary flow. Modifying equation (6) to provide for one half the slot wall thickness and the geometry of the halves of the slot wall adjacent to the primary flow gives the following equation for the minimum trailing-edge loss of the primary flow for a given blading:

$$\left(\frac{\Delta\eta_{te,p,min}}{\eta_o}\right)_3 = 0.340\left[\frac{w}{(\bar{e})(th)}\right]^{1/3}\left(\frac{w}{th}\right)c_{D,p} \tag{7}$$

where the flow coefficient $c_{D,p}$ is a function of the trailing-edge geometry (see appendix B).

Reference 2 further assumes that the fraction of the maximum recoverable trailing-edge loss attributed to the primary flow that is actually recovered is proportional to $(V_c/V_p)_1$. This assumption is used to obtain the following equation for the reduction in primary-air trailing-edge loss with coolant flow by subtracting the primary-air trailing-edge loss with coolant flow from the trailing-edge loss without coolant flow:

$$\left(\frac{\Delta\eta_{te,p}}{\eta_o}\right)_3 = \left(\frac{\Delta\eta_{te}}{\eta_o}\right)_3 - \left[\left(\frac{\Delta\eta_{te}}{\eta_o}\right)_3 - \left(\frac{\Delta\eta_{te,p,min}}{\eta_o}\right)_3\right]\left(\frac{V_c}{V_p}\right)_1 \tag{8}$$

Having described the manner of reference 2 for treating the trailing-edge loss ascribed to the primary flow, we now consider the manner of treating the trailing-edge loss of the coolant flow. Reference 2 assumes that the trailing-edge loss ascribed to the coolant flow is dependent upon the geometry and thickness of the halves of the slot wall adjacent to the coolant flow. Modifying equation (6) to provide for one half the thickness and the geometry of the halves of the slot walls adjacent to the coolant flow gives the following trailing-edge loss attributed to the coolant flow:

$$\left(\frac{\Delta\eta_{te}}{\eta}\right)_{c,3} = 0.340\left[\frac{w}{(\bar{e})(th)}\right]^{1/3}\left(\frac{w}{sl}\right)c_{D,c} \tag{9}$$

As indicated by the term $(\Delta\eta_{te}/\eta)_{c,3}$, equation (9) relates the trailing-edge loss of the coolant flow to the kinetic energy output of the coolant flow. To relate the trailing-edge loss of the coolant flow to the kinetic energy of the primary flow without coolant then requires that the relative energies of the coolant flow and uncooled primary flow be considered. Thus,

$$\left(\frac{\Delta\eta_{te,c}}{\eta_o}\right)_3 = (y)\left(\frac{V_c}{V_p}\right)_1^2\left(\frac{\Delta\eta_{te}}{\eta}\right)_{c,3} \tag{10}$$

If the trailing-edge loss with and without coolant flow is known, the fractional improvement in blade row efficiency resulting from reduced trailing-edge loss with coolant may be computed from the following expression:

$$\left(\frac{\Delta\eta_{p,te}}{\eta_o}\right)_3 = \left(\frac{\Delta\eta_{te}}{\eta_o}\right)_3 - \left[\left(\frac{\Delta\eta_{te,p}}{\eta_o}\right) + \left(\frac{\Delta\eta_{te,c}}{\eta_o}\right)_3\right] \tag{11}$$

The total fractional change in blade row efficiency relative to the uncooled blade row is then the sum of the fractional change in efficiency due to the coolant flow energy and the fractional change in efficiency due to reduction in trailing-edge loss. Thus equations (5) and (11) result in

$$\left(\frac{\Delta\eta_p}{\eta_o}\right)_3 = \left(\frac{\Delta\eta_{p,c,e}}{\eta_o}\right)_3 + \left(\frac{\Delta\eta_{p,te}}{\eta_o}\right)_3 \tag{12}$$

Comparison of Theories of the Two Analytical Methods

The major differences between the methods of references 1 and 2 are discussed and compared in this section.

Concerning the differences in treatment of the energy of the coolant flow, the method of reference 1 assumes turbulent mixing of the coolant and primary flows; whereas, the method of reference 2 assumes that the energy of the coolant flow, which is introduced in the regime of low primary-air momentum at the trailing edge, is utilized without mixing loss.

The different effects on primary-air efficiency that result from the different assumptions of the two methods concerning utilization of the energy of the coolant flow are compared in figure 4. The comparison is presented in terms of percent change in primary-air efficiency relative to the uncooled blading per percent coolant flow as a function of the coolant- to primary-air exit-velocity ratios. Figure 4 shows, for all coolant- to primary-air velocity ratios from 0 to 1.2, except 1.0, that the method of reference 1 results in less output due to the energy of the coolant than that of reference 2. (At zero coolant- to primary-air velocity ratio there is, of course, no coolant flow and no effect on the output.) At a coolant- to primary-air velocity ratio of 1.0, the figure shows the method of reference 1 results in the same output due to the coolant energy as reference 2. As a matter of interest, it is also noted that the results in figure 4 obtained from the method of reference 1 are essentially equivalent to those that would be obtained by computing the mixed velocity V_3 from the simple relation for turbulent mixing at constant pressure; that is,

$$V_3 = \frac{(V_{p,1}) + (y)(V_{c,1})}{1 + y} \tag{13}$$

The assumption of reference 2, that the coolant energy is utilized without mixing loss, may seem illogical from a physical viewpoint. However, the comparison between experimental results and analytical results for the two methods, to be shown in the next section of this report indicates that the assumption of utilization of the coolant energy without mixing gives generally better agreement with experimental results for three of the five test blade configurations than the assumption of turbulent mixing used in reference 1.

Concerning the differences in treatment of reduction in trailing-edge loss due to coolant ejection of the two analytical methods, the method of reference 1 for computing the trailing-edge loss without coolant flow assumes that the trailing-edge loss results from a sudden enlargement in flow area due to the thickness of the trailing edge. With coolant flow, the method assumes that the trailing-edge loss results from the sudden enlargement due to the two wall thicknesses that result when the slot is incorporated in the trailing edge. However, the method does not consider differences in trailing-edge loss due to the geometry (i.e., round, blunt, tapered, etc.) of the trailing edge either without the slot or the geometry of the two trailing edges resulting from the intersection of the blade surfaces with the slot surrfaces when the slot is incorporated in the trailing edge. In addition, it does not consider the effect of boundary layer thickness on trailing-edge loss that is shown to occur in the analysis of reference 16.

Reference 2 treats reduction in trailing-edge loss due to coolant ejection somewhat differently than reference 1. As previously discussed, the trailing-edge loss both with and without coolant flow is computed using a modification of the analytical method of reference 16. The method of reference 2 considers the effect on trailing-edge loss of factors not considered in reference 1. The factors are boundary layer thickness of the coolant and primary flows and the blade trailing-edge geometry as well as thickness, either with or without the incorporation of the slot.

An example, for one of the tested trailing-edge slot configurations, of the effect on primary-air efficiency that result from the different assumptions of the two analytical methods regarding reduction in trailing-edge loss with coolant flow is shown in figure 5. (The results shown for method 2 are based on experimental results of ref. 14. See appendix B.) The figure is presented in terms of percent change in primary-air efficiency per percent coolant flow as a fraction of coolant- to primary-air exit-velocity ratio. The example, for this configuration, indicates the general trend of differences in primary-air efficiency due to trailing-edge reduction with coolant flow for the two analytical methods for all five tested configurations. That is, the method of reference 2 results in a larger reduction in trailing-edge loss relative to the noncooled blading than the method of reference 1.

COMPARISON BETWEEN EXPERIMENTAL AND ANALYTICAL RESULTS

The results of a two-dimensional cold-air experimental investigation of the effect on stator blade primary-air efficiency of coolant ejection from five different trailing-edge slot configurations (ref. 12) are compared with results obtained from two approximate analytical prediction methods. The analytical methods used are those of references 1 and 2. The analytical methods and the geometry of the different trailing-edge slot configurations are discussed in detail in preceding sections of this report.

Results are reported primarily in terms of efficiency differences as a function of coolant fraction. The curves shown are the average of data taken at all three test ideal critical velocity ratios $(V/V_{cr})_{p,i,3}$ of 0.5, 0.65, and 0.8. The averages were used because the effect of primary-air velocity level on the experimental (ref. 12) and analytical results was small, comparable with measurement accuracy.

Efficiency differences are reported in terms of percent change in primary-air efficiency relative to the efficiency of the comparable uncooled blade. In equation form, the percent change in primary-air efficiency is equal to the following:

$$\left(\frac{\Delta\eta_p}{\eta_o}\right)_3 (100) = \left(\frac{\eta_p - \eta_o}{\eta_o}\right)_3 (100) \tag{14}$$

where $\eta_{p,3}$ the primary-air efficiency, is equal to the ratio of the actual kinetic energy output of the total flow relative to the ideal kinetic energy output of the primary flow only at the fully mixed station. Thus, (see eq. (1))

$$\eta_{p,3} = (1 + y)\left(\frac{V}{V_{p,i}}\right)_3^2$$

The comparison between experimental and predicted changes in primary-air efficiency (obtained using the methods of refs. 1 and 2) as a function of coolant flow rate for the stator bladings having five different trailing-edge slot configurations are shown in figure 6. The figure is in five parts with one part for each of the five bladings having different slotted trailing-edge geometries.

The comparisons in figure 6 generally show the following. An average of the absolute values of differences between experimental and predicted efficiencies for all five blade configurations showed that the method of reference 1 gave average efficiency differences which were about 1.3 percent different than experimental efficiencies while the method of reference 2 gave average efficiency differences that were about 0.7 percent different than experimental efficiencies. However, in some instances, maximum differences of as much as 4 percent occurred between predicted and experimental efficiencies for the two bladings with larger slot widths. Differences of this magnitude are certainly larger than desired. However, considering the fact that there is some possibility of error in experimental results due to uneven distribution of coolant flow (see ref. 12) and also the fact that both of the analytical methods are based on approximate theory, the agreement might be considered reasonable. (The largest difference shown is for the configuration in fig. 6(b)). As mentioned in ref. 12, experimental results for this configuration are somewhat suspect since they are inconsistent with the results for the other configurations.) The analytical results do agree better with experimental results for the two bladings with the smallest slot width (fig. 6(a) and (c)) than for the three bladings with larger slot widths, the maximum difference between experimental and predicted efficiencies for the bladings with the smallest slot width being about

0.5 percent. Also, the method of reference 2 gives generally better agreement with experimental results than that of reference 1 for three of the tested trailing-edge configurations (fig. 6(b), (d), and (e)), and the method of reference 1 gives better agreement than that of reference 2 for two of the configurations (figs. 6(a) and (c)).

Concerning the comparison between experimental and predicted results from the method of reference 1, in the lower range of coolant rates for all the configurations tested, the method predicts too small a slope of change in efficiency; whereas, in the upper range of coolant flows the method generally predicts about the correct slope of change in efficiency for the blade configurations represented in figures 6(a), (c), and (d) and too large a change of slope in efficiency for the blade configurations represented in figures 6(b) and (e).

Concerning the predicted comparisons for the method of reference 2, in the lower range of coolant rates, the method predicts too large a slope of change in efficiency for the two bladings with the smallest slot width (figs. 6(a) and (c)) and too small a slope of change in efficiency for the bladings with the larger slot widths (figs. 6(b), (d), and (e)). In the upper range of coolant rates, the method predicts slightly too small a change of slope in efficiency for the two bladings with the smallest slot width and too large a change of slope in efficiency for the three bladings with larger slot widths.

Regarding specific comparisons between experimental and analytical results for the two methods, for the two bladings with the smallest slot width (figs. 6(a) and (c)), there is fair agreement between test and predicted values of percentage change in primary-air efficiency obtained using both methods, the method of reference 1 resulting in slightly better agreement, particularly at low coolant rates, than the method of reference 2. For the three bladings with larger trailing-edge slot widths (figs. 6(b), (d), and (e)), both of the analytical methods predict too small a change in slope of efficiency in the lower range of coolant flows and too large a change in slope of efficiency in the upper range of coolant flows. However, over the full range of coolant ratios considered, the method of reference 2 agrees better with experimental results than the method of reference 1. In fact, for the blading represented in figure 6(d), the difference between experimental results and results obtained using the method of reference 2 is within 1 percent which is considered to be reasonably good agreement.

It will be noted that for all test blade configurations the method of reference 1 predicts a smaller percent change in primary-air efficiency with coolant flow rate than the method of reference 2. The reasons for this occurring were indicated in the preceding section of this report, ANALYTICAL METHODS, under the subheading Comparison of Theories of the Two Analytical Methods. As discussed in that section, the principal reason for the differences between the two methods is that the method of reference 1 assumes turbulent mixing of the coolant, whereas the method of reference 2 assumes that the coolant flow sustains no mixing loss. As shown in figure 4, for given flow conditions, these assumptions result in the computed energy output due to the coolant flow being greater for the method of reference 2 than for the method of reference 1 (except at coolant- to primary-air velocity ratios of 1.0). In addition, as previously discussed and indicated in figure 5, the method of reference 2 results in a little larger increase in output due to trailing-edge loss recovery than the method of reference 1.

ANALYSIS OF RESULTS AND EMPIRICAL METHOD

As indicated in the previous section, neither of the two published analytical methods (refs. 1 and 2) used herein are considered by the author to satisfactorily predict the experimental results for all of the five tested blade configurations even though it is realized that the analytical methods are approximate and there is some reservation concerning the absolute accuracy of the experimental data. The experimental results were therefore further analyzed in an effort to improve the methods.

As discussed previously under ANALYTICAL METHODS, the analytical methods of references 1 and 2 assume that the major effect on blade row output of discharging coolant from a trailing-edge slot results from the energy of the coolant. The method of reference 1 assumes turbulent mixing of the coolant and primary flows and the method of reference 2 assumes that the coolant energy is utilized without mixing loss. However, as shown in figure 4, for the same coolant- to primary-air exit-velocity ratio, both methods assume that the percent change in output per percent coolant flow resulting from the coolant flow energy is independent of the slotted trailing-edge geometry. In the immediately preceding section of this report, it was shown that the results obtained from both analytical methods agree better with experimental results for blading with some trailing-edge geometries than for blading with other trailing-edge geometries. It is thus indicated that the percent change in efficiency per percent coolant flow resulting from the coolant flow energy depends, at least in part, upon the slotted trailing-edge geometry.

To show more clearly that the experimental results of reference 12 indicate that, for the same coolant- to primary-air exit-velocity ratio, the percent change in output per percent coolant flow resulting from the coolant flow energy is a function of the slotted trailing-edge geometry, the results of reference 12 were further analyzed and rearranged as explained and presented in the following. The percent change in primary-air efficiency relative to the uncooled blading resulting from reduction in trailing-edge loss with coolant flow $(\Delta\eta_{p,te}/\eta_o)_3$ was first computed for each of the five test bladings using the previously described analytical method of reference 2 for trailing-edge loss reduction as modified in appendix B. The computed percent change in primary-air efficiency due to reduction in trailing-edge loss was then subtracted from the experimentally determined values of total change in primary-air efficiency. In this manner, the change in primary-air efficiency resulting from the energy of the coolant was obtained. Thus

$$\left(\frac{\Delta\eta_{p,c,e}}{\eta_o}\right)_3 = \left(\frac{\Delta\eta_p}{\eta_o}\right)_3 - \left(\frac{\Delta\eta_{p,te}}{\eta_o}\right)_3 \tag{15}$$

Next, the percent change in primary-air efficiency per percent coolant flow $(\Delta\eta_{p,c,e}/\eta_o)_3(1/y)$ resulting from the effect of the coolant flow was determined as a function of coolant- to primary-air exit-velocity ratio for each of the five blade configurations.

In figure 7, the correlation between the percent change in output per percent coolant flow as functions of the ratio of coolant- to primary-air exit-velocity ratio $(V_c/V_p)_1$ and the ratio of trailing-edge slot width to trailing-edge thickness sl/t is presented. (Since the reader may wish to relate the coolant- to primary-air exit-velocity ratio to the coolant fraction, the relation between these two variables for the five test configurations is shown in fig. 8. Fig. 8 is taken from Part I of ref. 12.) The results in figure 7 strongly indicate, particularly at low coolant- to primary-air exit-velocity ratios, that the geometry of the slotted trailing edge is a major parameter affecting the utilization of the coolant flow and that the ratio of the slot width to the trailing-edge thickness appears to be the major trailing-edge geometric parameter affecting the utilization of coolant energy.

The results in figure 7 show that, at lower values of coolant- to primary-air exit-velocity ratios, the percent change in output per percent coolant flow resulting from the coolant flow energy, in general, decreases with decreasing values of slot width to trailing-edge thickness ratios. As the coolant- to primary-air velocity ratio is increased from low values up to 1.0, the effect of slot to trailing-edge width ratio on the percent change in output per percent coolant flow is shown, in general, to decrease. As the coolant- to primary-air velocity ratio reaches 1.0, the results indicate little effect of trailing-edge geometry on the percent change in output per percent coolant flow. Finally, as the coolant- to primary-air velocity ratio is increased above 1.0, an increase in percent change in output per percent coolant flow with decreasing slot width ratio is indicated.

As shown in figure 9, the results in figure 7 indicate that at lower values of slot to trailing-edge width ratios, the experimental method of reference 1, which assumes turbulent mixing of the coolant flow, better represents the net effect of the coolant flow energy on the output of the total flow; while at higher values of slot to trailing-edge width ratios the method of reference 2, which assumes no mixing loss of the coolant flow, better represents the net effect of the coolant flow energy on the output of the total flow. (The results in fig. 9 were taken from figs. 4 and 7.)

It was hoped that an easy means, based on simplified theory, could be found which would allow the present methods to be corrected for the effect of slotted trailing-edge geometry on blade row output. However, no such means, amenable to simple mathematical expressions, were found by the author.

Although no simple means could be found for correcting either of the two methods, the correlation shown in figure 7 may be used in an empirical procedure to predict the change in primary-air efficiency of other stator blading having different trailing-edge geometry than the test bladings. However, it is pointed out that the correlation for this stator blade may not apply to other blade configurations. Nevertheless, if better information were not available for a particular blading, it would provide some means, based on experimental data and some logic, of predicting the change in primary-air efficiency due to trailing-edge coolant ejection.

It is also noted that if the method is applicable at the tested coolant- to primary-air temperature ratio of 1.0, it should also be applicable at other temperature ratios. The method would be applicable because it is dependent upon coolant- to primary-air exit-velocity ratio and coolant rate only and independent of the temperature ratio.

To use the correlation to predict the primary-air efficiencies, for other blading would require the following. The coolant flow rate y and the coolant- to primary-air exit-velocity ratio $(V_c/V_p)_1$ would have to be determined for the blading in question. Knowing these values and the ratio of coolant slot to trailing-edge width, the change in primary-air efficiency per percent coolant flow relative to the uncooled blading $(\Delta\eta_{p,c,e}/\eta_o)_3(1/y)$ resulting from the coolant flow energy could be read from figure 7. The change in efficiency due to reduction in trailing-edge loss relative to the uncooled blading $(\Delta\eta_{p,te}/\eta_o)_3$ would next be computed from equation (11). If $(\Delta\eta_{p,c,e}/\eta_o)_3(1/y)$ and $(\Delta\eta_{p,te}/\eta_o)_3$ are known, the total change in primary-air efficiency could be computed for the coolant fraction in question. Thus,

$$\left(\frac{\Delta\eta_p}{\eta_o}\right)_3 = \left(\frac{\Delta\eta_{p,c,e}}{\eta_o}\right)_3 \frac{1}{y} (y) + \left(\frac{\Delta\eta_{p,te}}{\eta_o}\right)_3 \qquad (16)$$

CONCLUDING REMARKS

A study was made to compare experimentally determined efficiencies of turbine stator blades having trailing-edge coolant ejection with efficiencies predicted from two previously published approximate analytical methods. The experimental results were obtained from two-dimensional data with the temperatures of the primary and coolant flows both being nearly ambient. Five stator blade configurations having two different trailing-edge thicknesses and four different slot widths were included in the comparison.

The test blading has a span of 10.16 centimeters (4.0 in.) and a chord width of 5.74 centimeters (2.26 in.). Two of the five test trailing-edge slot configurations had a trailing-edge thickness of 0.178 centimeter (0.070 in.) with slot widths of 0.051 centimeter (0.020 in.) and 0.102 centimeter (0.040 in.), and the other three had a trailing-edge thickness of 0.330 centimeter (0.130 in.) with slot widths of 0.051 centimeter (0.020 in.), 0.127 centimeter (0.050 in.), and 0.203 centimeter (0.080 in.).

An average of the absolute values of differences between experimental and predicted efficiencies for all five blade configurations showed that one of the methods gave average efficiency differences which were about 1.3 percent different than experimental efficiencies while the other method gave average efficiency differences that were about 0.7 percent different than experimental efficiencies. However, in some instances, maximum differences of as much as 4 percent occurred between predicted and experimental efficiencies.

The agreement between experimental and analytical results was generally considerably better for the two blade configurations having the smallest slot width than for the other three configurations with larger slot widths. One of the analytical methods gave results which agreed better with experimental results for the two bladings with the smallest slot width, particularly at low coolant rates. The results of the other analytical method agreed better with experimental results for the three bladings with larger slot widths.

The comparison between experimental and analytical results strongly indicate that the ratio of trailing-edge slot width to trailing-edge thickness influences the measured efficiencies to a greater extent than is accounted for by either of the analytical models. Therefore, an empirical method was derived from the experimental results. However, it may not be applicable to other stator blading.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, November 15, 1974,
505-04.

APPENDIX A

SYMBOLS

c_D trailing-edge drag coefficient

$\bar{e}$ fractional loss in energy due to blade surface friction loss

m mass flow, kg/sec; lbm/sec

p absolute pressure, N/m^2; lbf/ft^2

s blade pitch, m; ft

$s\ell$ height of trailing-edge coolant jet at station 1 (assumed equal to slot width)

t trailing-edge thickness, diameter of circular arc forming blade trailing edge, m; ft

th blade throat width, m; ft

V absolute velocity, m/sec; ft/sec

w thickness of trailing-edge slot wall at blade exit, m; ft

y ratio of coolant to primary-air mass flow

α fluid flow angle measured from axial direction, deg

δ total boundary layer displacement thickness at blade trailing-edge based on average free stream conditions, m; ft

δ^* $\delta/s \cos \alpha_1$

$\delta_{s\ell}$ $s\ell/s \cos \alpha_1$

δ_{te} $t/s \cos \alpha_1$

η_o efficiency of uncooled blade, ratio of actual to ideal kinetic energy at aftermixed station

η_p primary-air efficiency, ratio of kinetic energy of total flow to ideal energy of primary flow only at aftermixed station

$\frac{\Delta\eta_p}{\eta_o}$ $\frac{\eta_p - \eta_o}{\eta_o}$

θ total boundary layer momentum thickness at blade trailing edge based on average free stream conditions, m; ft

θ^* $\theta/s \cos \alpha_1$

ρ density, kg/m^3; lbm/ft^3

Subscripts:

c	coolant flow
cr	conditions at Mach 1
e	kinetic energy
i	ideal conditions corresponding to isentropic conditions
o	without coolant flow
p	primary flow
te	trailing-edge
min	minimum
1	station at trailing-edge plane
3	hypothetical downstream station where flow conditions are uniform

MODIFICATIONS TO THE ANALYTICAL METHOD

In appendix B of reference 2, equations (B17), (B18), and (B20) for computing the trailing-edge loss with and without coolant flow do not provide for different drag coefficients resulting from changes in trailing-edge geometry with and without the trailing-edge slot. To distinguish between these different drag coefficients, equation (B17) for computing the trailing-edge loss without the trailing-edge slot should have the drag coefficient term c_D renamed $c_{D,o}$; equation (B18) for computing the trailing-edge loss of the primary flow with the slot should have the drag coefficient c_D renamed $c_{D,p}$; and equation (B20) for computing the trailing-edge loss of the coolant flow with the slot should have the drag coefficient c_D renamed $c_{D,c}$.

In consideration of the foregoing and the results of reference 14 in which the drag coefficients for various slot geometries were determined, the following drag coefficients for the tested trailing-edge geometries (see fig. 2) were used for this study.

Trailing-edge configuration	Trailing-edge thickness, t		Height of trailing-edge coolant jet at station 1, s_l		Trailing-edge drag coefficients		
	cm	in.	cm	in.	$c_{D,o}$	$c_{D,p}$	$c_{D,c}$
1	0.178	0.070	0.051	0.020	0.130	0.070	0.200
2	.178	.070	.102	.040			.070
3	.330	.130	.051	.020			.200
4	.330	.130	.127	.050			.070
5	.330	.130	.203	.080			.070

The previous values were selected from reference 14 on the following basis. The base drag coefficient without the trailing-edge slot $c_{D,o}$ is that of a round trailing-edge and equal to 0.130. With the trailing-edge slots, the geometry of the halves of the trailing-edge on the primary flow side of all the blading resembles that of a tapered trailing-edge having a drag coefficient of 0.070. With the trailing-edge slots, the geometries of the halves of the trailing-edge on the coolant flow side of the blading are somewhat different for different bladings. For configurations 1 and 3, coolant side drag coefficients $c_{D,c}$ of 0.200 for a square trailing-edge geometry were used, and for configurations 2, 4, and 5, coolant side drag coefficients of 0.070 for a tapered trailing-edge geometry were used.

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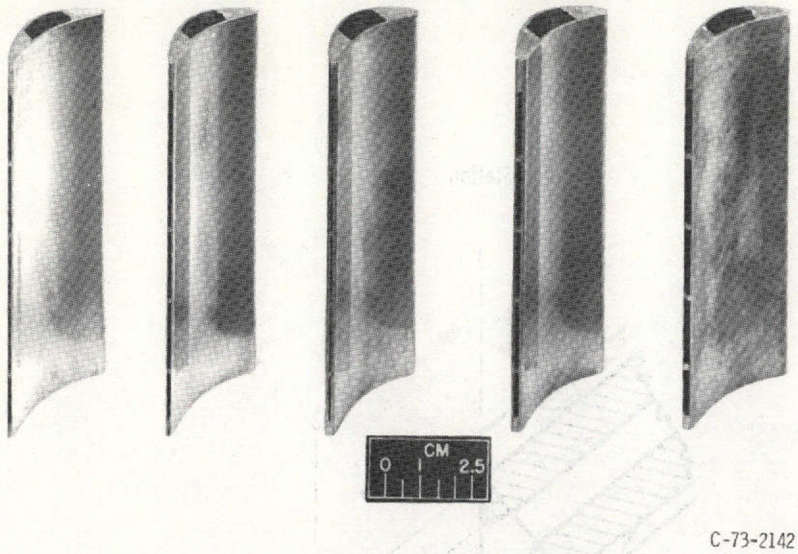
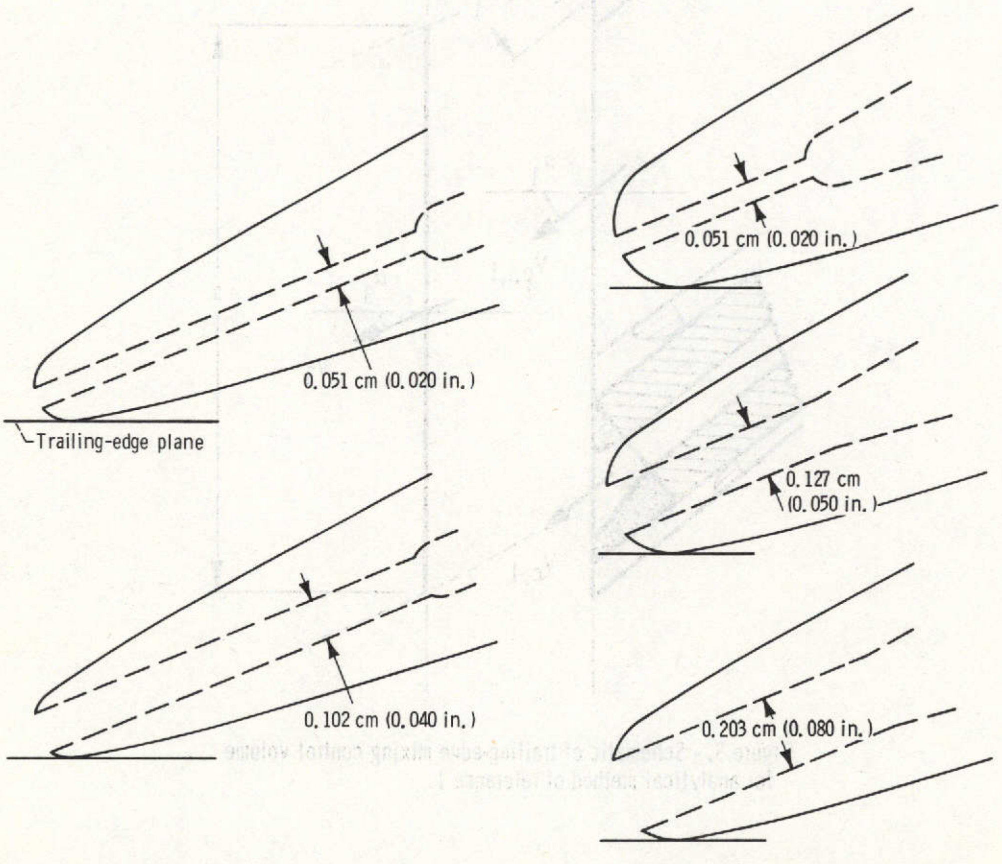


Figure 1. - Five test blade configurations.



(a) Trailing-edge thickness, 0.178 centimeter (0.070 in.).

(b) Trailing edge thickness, 0.330 centimeter (0.130 in.).

Figure 2. - Cross sections of blade trailing-edge slot geometries. Trailing-edge thickness is equal to diameter of circular arc at blade trailing edge.

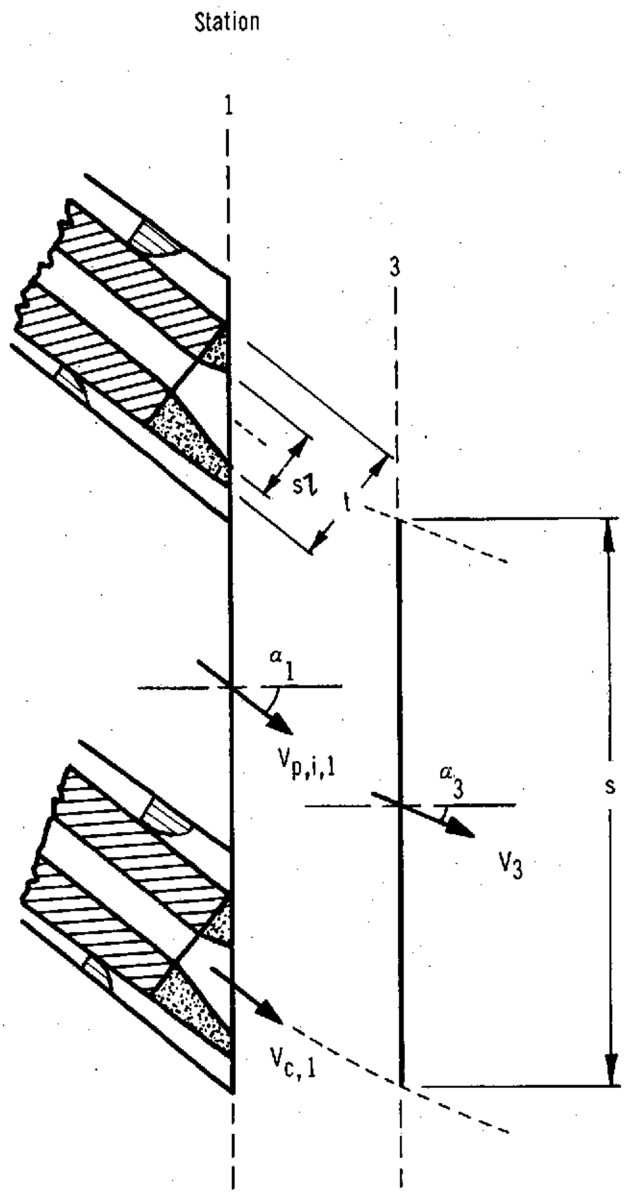


Figure 3. - Schematic of trailing-edge mixing control volume for analytical method of reference 1.

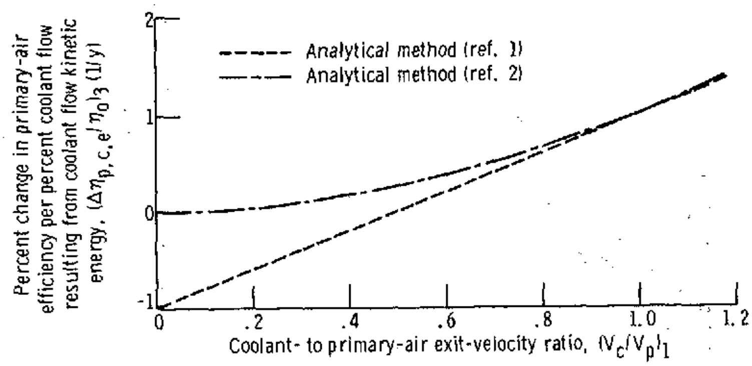


Figure 4. - Predicted effect of coolant flow energy on primary-air efficiency as function of coolant- to primary-air exit-velocity ratio for two analytical methods.

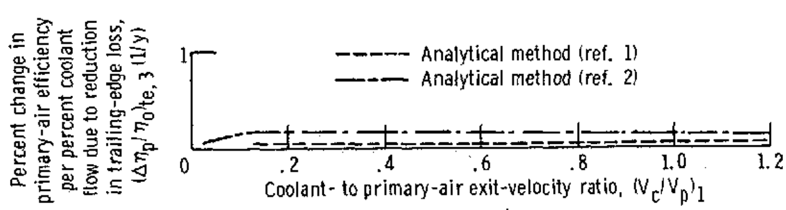


Figure 5. - Example of predicted gain in primary-air efficiency resulting from reduction in trailing-edge loss as function of coolant- to primary-air exit-velocity ratio for two analytical methods. Blade trailing-edge slot width, 0.203 centimeter (0.080 in.); trailing-edge thickness, 0.330 centimeter (0.130 in.).

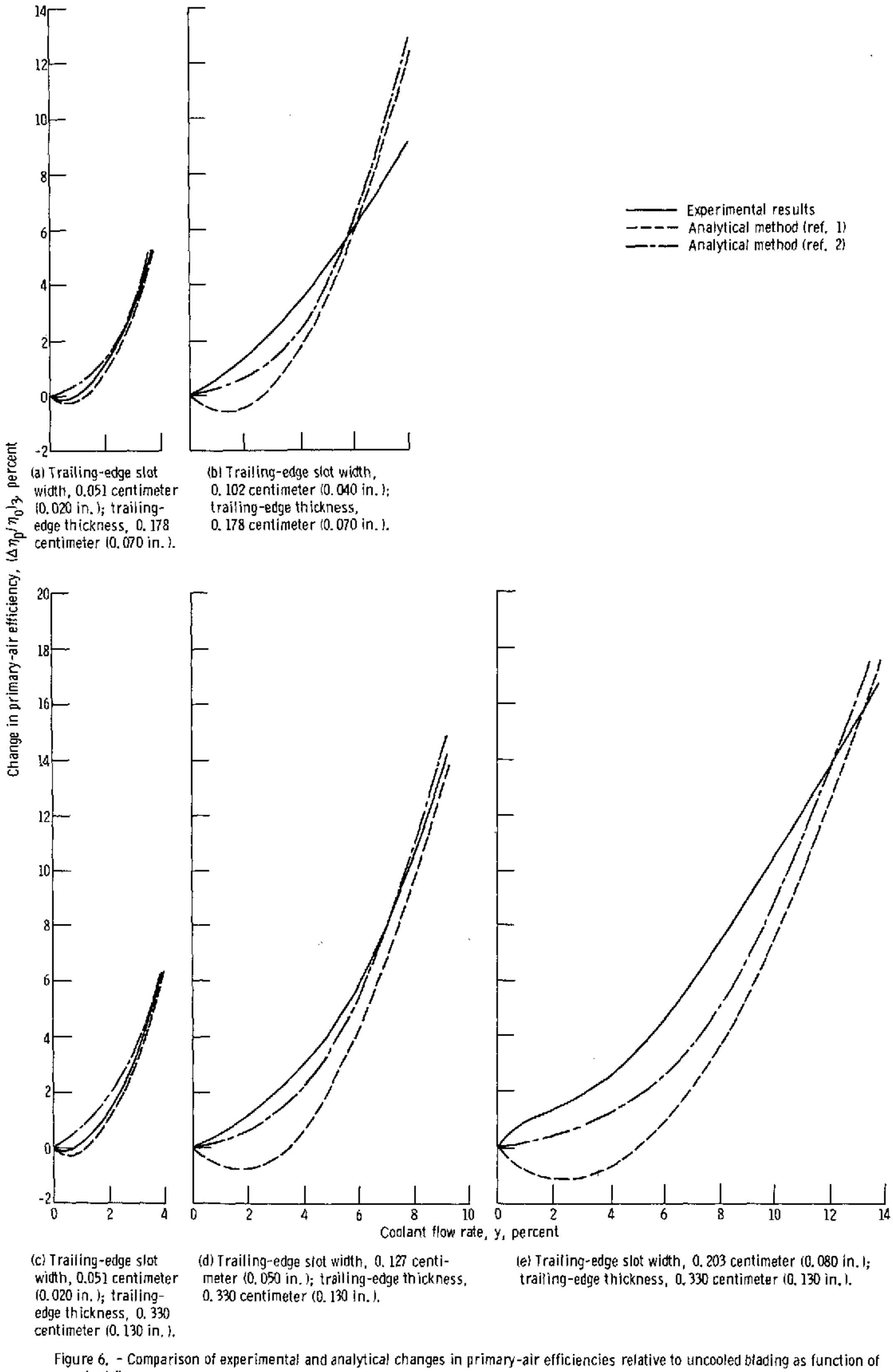


Figure 6. - Comparison of experimental and analytical changes in primary-air efficiencies relative to uncooled blading as function of coolant flow.

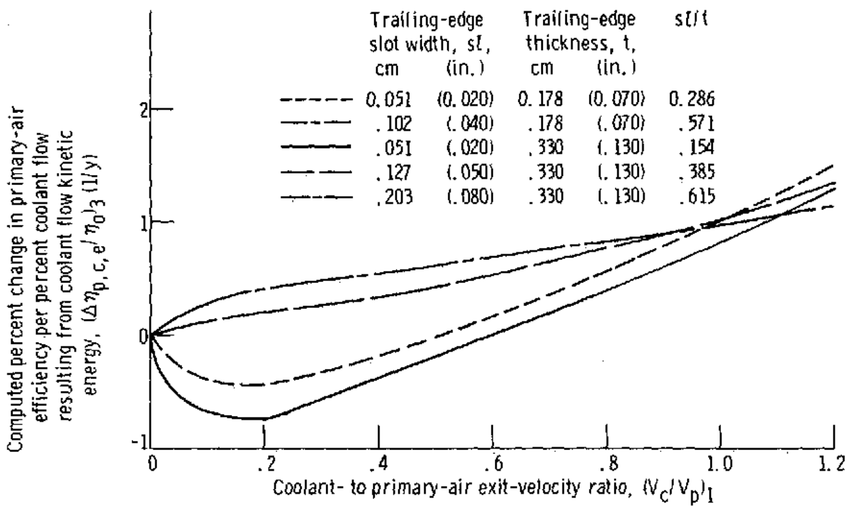


Figure 7. - Effect of coolant flow energy on primary-air efficiency as function of coolant- to primary-air exit-velocity ratio and trailing-edge geometry computed from experimental results.

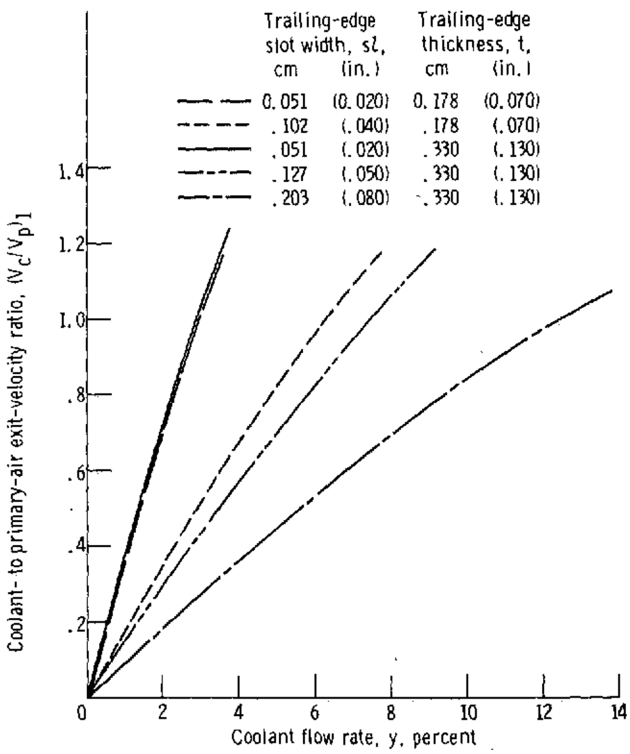


Figure 8. - Variation of coolant- to primary-air exit-velocity ratio as function of coolant flow rate for different trailing-edge slot geometries and primary flow critical velocity ratios.

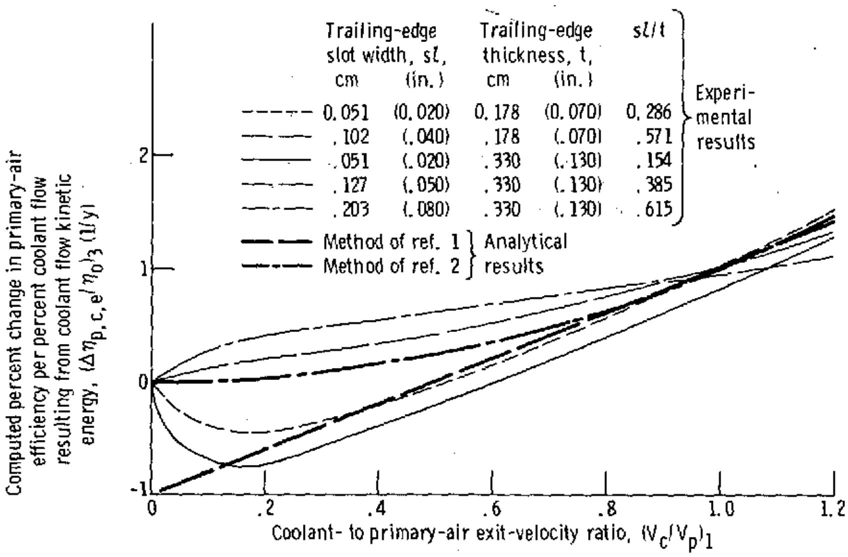


Figure 9. - Effect of coolant flow energy on primary-air efficiency as function of coolant- to primary-air exit-velocity ratio and trailing-edge geometry computed from experimental results and analytical methods.

NASA-Langley, 1975 E-8121