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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

I - Experimental Results

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16. Abstract

Five different trailing-edge slot configurations were investigated in a two-dimensional cascade of turbine stator blades. The trailing-edge slots were incorporated into blades with round trailing edges. The five blade configurations investigated included blades with two different trailing-edge thicknesses and four different slot widths. The results of the investigation showed that there was, in general, a significant increase in primary-air efficiency due to the coolant flow, the increase varying with slot configuration. For the five configurations tested, the average percent change in primary-air efficiency per percent coolant flow varied almost linearly from zero to about 1.4 percent over a range of coolant- to primary-air exit-velocity ratios between 0 and 1.2. However, for different configurations there was considerable deviation from the average values in the lower range of exit velocity ratios.

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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

I - EXPERIMENTAL RESULTS

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SUMMARY

The investigation was conducted in a two-dimensional cascade. Separate tests were conducted for each of the different slot configurations. The investigation was conducted at nominal ideal exit primary-air critical velocity ratios of 0.5, 0.65, and 0.8 and over a range of coolant flow rates corresponding to coolant- to primary-air exit-velocity ratios from zero to about 1.2. The principal results are reported in terms of primary-air efficiency, which relates the actual kinetic energy output of the total flow to the ideal energy output of the primary flow only.

For the five configurations tested, the average percent change in primary-air efficiency per percent coolant flow varied approximately linearly from zero to 1.4 percent over the range of coolant- to primary-air exit-velocity ratios from 0 to 1.2. However, there was rather poor agreement with these average values between bladings with different slot widths in the lower range of coolant- to primary-air exit-velocity ratios.

For all the bladings, except for the two with the smallest slot width at low coolant flow rates, the primary-air efficiency increased significantly with increased coolant flow rate. The percent change in primary-air efficiency of the five bladings varied differently with coolant flow rate. At coolant flow rates up to about 2 percent, the change in primary-air efficiency was larger for the bladings with larger slot widths than for those with smaller slot widths. At 2 percent coolant flow rate, the change in primary-air efficiency for all bladings was about the same. Above 2 percent coolant flow rate, an increase in slot width resulted in a smaller increase in efficiency as the coolant flow rate increased.

For the tested range of primary-air critical velocity ratios, there was essentially no effect of primary-air critical velocity ratio on percent change in primary-air efficiency.

INTRODUCTION

Concerning the performance of cooled turbines, several analytical and experimental studies (e.g., refs. 1 and 2) have shown that different means of ejecting compressor bleed coolant air from the turbine blade surface have 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 on other aspects of turbine performance.

Several means of coolant ejection have been investigated previously. For instance, reference 3 reports the results of an experimental investigation of the effect on stator blade performance of coolant ejection from four rows of coolant holes located in or near the diffusion region on the suction surface, with the axis of the holes parallel to the end walls, at an angle of 35° to the blade surface. References 4 to 6 report the results of experimental and analytical investigations of the influence of turbine stator blade trailing-edge coolant ejection on turbine stator and stage performance. And references 7 to 10 report the results of experimental and analytical investigations of the effect of stator blade transpiration discharge on turbine stator and stage performance. The results of the investigations of references 4 to 10 are summarized in reference 11.

The following are the main conclusions of the investigations of references 3 to 10: Coolant flow discharged from rows of coolant holes on the suction surface, in or near the diffusion region, with the axis of the holes parallel to the end walls, at an angle of 35° to the blade surface would decrease the turbine work output at low coolant ejection velocities and increase the turbine work output at high coolant velocities. Coolant flow ejected from a particular trailing-edge slot parallel to the main stream significantly increased the turbine work output. Coolant flow ejected over the complete blade surface at an angle normal to the blade surface contributed little or nothing to the turbine work output.

As mentioned, references 4 to 6 report on investigations of the influence on turbine stator and stage performance of coolant ejection from stator blading having a given trailing-edge slot configuration. The investigation reported herein is an extension of the work of these references and was conducted to determine the effect of five different trailing-edge slot configurations on stator blade performance.

The testing for this investigation was conducted in a two-dimensional cascade. The temperatures of the primary air and the coolant air were nearly the same, and atmospheric air was used as the primary flow fluid. The tests were conducted at ideal primary-air exit critical velocity ratios of 0.5, 0.65, and 0.80. The range of coolant-air to primary-air mass flow percentages investigated varied for different slot geometries. This variation resulted because the maximum coolant- to primary-air exit-velocity ratio was limited to about 1.2 for all slot geometries, while the slot width for some geometries was larger than for others.

The principal results are reported in terms of primary-air efficiency, either as a function of coolant to primary air mass flow percentages or as a function of coolant- to primary-air exit-velocity ratios. In addition, experimentally determined values of coolant slot discharge coefficients, which are of engineering interest, are presented in appendix A.

APPARATUS, INSTRUMENTATION, AND PROCEDURE

Blading

The five trailing-edge configurations are shown in figure 1. As indicated, the blades are hollow and of constant cross section.

Cross-sectional sketches showing the geometries and significant dimensions of the five different trailing-edge slot configurations are presented in figure 2. Two of the five test blade configurations (fig. 2(a)) had trailing-edge thicknesses of 0.178 centimeter (0.070 in.), with coolant slot widths of 0.051 centimeter (0.020 in.) and 0.102 centimeter (0.040 in.). (These are shown on the left side of fig. 1.) The other three configurations (fig. 2(b)) had trailing-edge thicknesses of 0.330 centimeter (0.130 in.), with coolant slot widths of 0.051 centimeter (0.020 in.), 0.127 centimeter (0.050 in.), and 0.203 centimeter (0.080 in.). (These are shown on the right side of fig. 1.) 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 1.968 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 12. Detailed dimensions and geometry of the blading may be found in that reference. Some significant dimensions of the bladings are as follows: 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 13.

Cascade

The blading was tested in the simple two-dimensional cascade shown in figure 3. There are 12 blades in the cascade; however, only three blades near the middle are cooled. Other details of the cascade are described in reference 14.

Referring to figure 3, primary (atmospheric) air enters the cascade inlet, shown on the right side of the photo, and coolant air enters the inside of the three middle blades of the cascade through the coolant manifold and associated piping. The survey probe actuator operates a slide in which a multipurpose survey probe is mounted downstream of the blading. The coolant flow and primary flow passing through the blading is discharged from the cascade through exhaust piping attached to the circular base of the cascade.

Instrumentation

A calibrated multipurpose survey probe of the type shown in figure 4 was used to determine the angle, static pressure, and loss in total pressure downstream of the blading. (A detailed description of this type probe is given in ref. 14.) Coolant total pressure p_c' inside the blade was measured with a total pressure probe having the sensing element of the probe located 2.54 centimeters (1.0 in.) from the blade end walls on the coolant manifold side. (Symbols are defined in appendix B.) The circular sensing end of the probe faced the coolant flow entering the blading so that the total pressure inside the blade was measured as accurately as was practically possible.

Coolant flow was measured with the use of various size calibrated sharp-edged orifice plates located in a 5.08 centimeter (2.0 in.) orifice run. The orifice run, including instrumentation and orifice plates, all conformed to ASME specifications.

All pressure data were measured with calibrated strain-gauge transducers.

Test and Calculation Procedures

Only three test blades were used in the investigation of each blade configuration. These were installed in the middle of the 12-blade cascade. Data were then taken for only the center blade of the three test blades, so that the measured data simulated data for a blade in a completely cooled blade row having adjacent blades of the same design and with the same flow conditions. Also, to eliminate the effect of end-wall conditions on the measurements, data were taken at the mean section of the blading only.

To operate the test facility, primary (atmospheric) air is caused to flow through the cascade by use of the laboratory altitude exhaust system, which is piped to the cascade outlet. Desired primary-air pressure ratios across the blade row are maintained by regulation of an exhaust control valve. Coolant air flow is provided by the laboratory combustion air system. After installing the proper size orifice plate for a desired range of coolant flow rates, desired coolant flow rates are obtained by first setting the desired upstream orifice pressure by means of an upstream pressure regulator and then setting the desired pressure ratio across the orifice plate by regulating a throttling valve downstream of the orifice plate.

To conduct survey tests, the desired primary-air set-point critical velocity ratio and coolant flow rate were established for the blading by regulation of the primary-flow and coolant-flow control valves. After the desired primary and coolant flow conditions were set, a mean-section survey was made with the multipurpose probe across one blade pitch of the middle test blade to determine the downstream flow condition of the test blading. During the survey, all data, including probe survey data, coolant flow data etc., were digitized and recorded on magnetic tape. Also during testing, pertinent survey data were monitored on x-y recorders, and all data were monitored by teletype feed-back from the laboratory data processing center.

The survey investigations of coolant ejection from each of the different trailing-edge slot configurations were conducted at three nominal primary-air ideal exit critical velocity ratios $(V/V_{cr})_{p,i,3}$ of 0.5, 0.65, and 0.8. The range of coolant flow rates investigated varied from zero to different maximum percentages for different configurations, depending on the slot size and the fact that the coolant- to primary-air exit-velocity ratio was limited to about 1.2.

The general procedure for computing the test results was as follows: Coolant flow rates were computed by the method specified in the ASME code for sharp-edged orifices. Local values of mass flow, momentum, flow angle, static pressure, and kinetic energy at each data point included in the survey were then computed. The local values of primary mass flow, momentum, etc., were next integrated over one blade pitch to obtain total values of the same quantities at the measuring station. Then, with the assumption of the conservation of tangential momentum, total values of flow, axial momentum, and tangential momentum at the measuring station were equated to the same quantities at the hypothetical after-mixed downstream station. These equations were then solved simultaneously to obtain the after-mixed flow conditions. With the after-mixed flow conditions known, the primary-air efficiency, as well as other results of interest, could be computed at fully mixed flow conditions. At the after-mixed station, the equation used for computing primary-air efficiency was

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

Accuracy of Results

Concerning the accuracy of the results, statistical evidence obtained from many tests by several investigators using this same facility indicate the maximum probable error in primary-air efficiency is about ±0.25 percent for uncooled blading. The exact reasons for the error are not known. However, the investigators have speculated that the error could result from the following reasons: measurement inaccuracy; actual changes in the efficiency of the same blading due to fluctuations in flow or fluctuations in location of the boundary layer transition point from laminar to turbulent flow on the blade surfaces; or temporary collection of foreign material on either the blade or survey-probe surfaces during testing.

RESULTS AND DISCUSSION

A cold-air experimental investigation was conducted in a two-dimensional cascade to determine the effect on turbine stator blade aerodynamic performance of coolant ejection from five different trailing-edge slot configurations. Separate tests were conducted for bladings with each of the different slot configurations. The testing was done only at the mean section of the blading.

The investigation was conducted at nominal ideal primary-air exit critical velocity ratios $\left(V/V_{cr}\right)_{p,i,3}$ of 0.5, 0.65, and 0.8. (Symbols are defined in appendix B.) The range of coolant-air to primary-air mass flow percentages investigated varied for different slot geometries. This variation occurred because the maximum coolant- to primary-air exit-velocity ratio $\left(V_c/V_p\right)_3$ was limited to about 1.2 for all slot geometries, while the slot width for some geometries was different than for others. (It should be noted that the velocity of the coolant $V_{c,3}$ is based on the static pressure at the after-mixed station. This was done since the local static pressure directly at the exit of the trailing edge would be difficult, if not impossible, to determine, particularly with trailing-edge coolant ejection (see ref. 15).)

The principal results are reported in terms of after-mixed primary-air efficiency $\eta_{p,3}$ which relates the actual kinetic energy output of the total flow to the ideal output of the primary flow at the hypothetical downstream location where flow conditions are uniform (see eq. (1)).

The results are presented in three sections. The first section presents the experimentally determined values of primary-air efficiency as a function of coolant flow rate. The second section presents the percent change in primary-air efficiency of each of the bladings with different slot configurations, relative to the respective blading with the slots filled, as a function of coolant flow rate. Also in this section, the changes in primary-air efficiency resulting from different slot configurations are compared as functions of coolant flow rate, and the reasons for the differences are discussed. In the last section, the changes in efficiency of the different blade configurations are correlated with coolant- to primary-air exit-velocity ratio. The average changes in efficiency obtained from this correlation are then used to compute the change in efficiency of each of the blade configurations as a function of coolant rate. The computed results are then compared with the experimental results to determine if using the average correlation of the change in efficiency as a function of coolant- to primary-air exit velocity satisfactorily predicts the experimental change in efficiency with coolant flow rate of all five configurations.

In addition to the main results presented in the text, experimentally determined values of trailing-edge slot discharge coefficients, which are of engineering interest, are presented in appendix A as functions of ideal slot Reynolds number, with the effect of primary-air critical velocity ratio also indicated.

Experimental Efficiencies

The experimentally determined values of stator blade primary-air efficiency for the five tested trailing-edge slot configurations are presented in figure 5 as functions of coolant flow rate. The results include data for the three primary-air critical velocity ratios considered.

The results indicate that for all the test blade configurations, the effect of primary-air critical velocity ratio had little effect on the change in primary-air efficiency. Further, the results show for all the blading except the two with the smallest slot width at between zero and about 1 percent coolant flow rate, that the primary-air efficiency increased significantly with increasing coolant flow rate. For the bladings with the smallest slot width, between zero and about 1.0 percent coolant flow (figs. 5(a) and (c)), the primary-air efficiency first decreased slightly, then increased.

In addition, the increase in primary-air efficiency with coolant flow rate is indicated to be generally greater for the blading with smaller trailing-edge slots than for the blading with the larger trailing-edge slots. For instance, in figure 5(c), between zero and 3.5 percent coolant flow rate, the change in efficiency for the blading with 0.051-centimeter (0.020-in.) slots is about $4\frac{1}{2}$ points; whereas, in figure 5(d), for the blading with 0.127-centimeter (0.050-in.) slots, the change in efficiency between the same coolant flow rates is about $2\frac{1}{2}$ points. In the next section, the effects of primary-air critical velocity ratio and trailing-edge slot width are presented in a manner that more clearly shows these effects. Also, the reason for, and exceptions to, these general effects are discussed in greater detail.

Effect of Slot Geometry and Primary-Air Critical Velocity Ratio

on Variations in Primary-Air Efficiency

In this section, changes in primary-air efficiency resulting from primary-air critical velocity ratio and the different trailing-edge slot configurations are reported and compared.

The changes in primary-air efficiency are presented as percent variations in primary-air efficiency of the slotted blade relative to the efficiency of the corresponding unslotted blade as a function of coolant flow rate. In equation form, the percent variation in primary-air efficiency is equal to the following:

$$\left(\frac{\Delta\eta}{\eta_0}\right)_{p,3} (100) = \left(\frac{\eta_p - \eta_0}{\eta_0}\right)_3 (100) \tag{2}$$

The percent change in primary-air efficiency for the five bladings with different slot configurations are presented in figure 6. The results show that the percent change in primary-air efficiency for blading with a given slot configuration is little, if at all, affected by the primary-air critical velocity ratio, since the small differences shown due to primary-air critical velocity ratio are within test accuracy. For blading with a given slot geometry, the percent change in primary-air efficiency would logically be expected to be dependent on coolant- to primary-air exit-velocity ratio and coolant flow rate only (see ref. 2). As shown in figure 7, for a given slot configuration, the variation of coolant- to primary-air exit-velocity ratio as a function of coolant flow rate is the same for the three primary-air critical velocity ratios tested. Therefore, the percent change in primary-air efficiency as a function of coolant flow rate (fig. 6) for blading with a given slot geometry would not be expected to be influenced by the primary-air critical velocity ratio.

A comparison of percent change in primary-air efficiency as a function of coolant flow rate for the five blade configurations tested is presented in figure 8. The curves shown for the different configurations are the average curves presented in figure 6.

The change in primary-air efficiency resulting from trailing-edge coolant ejection has been shown (refs. 2 and 5) to result from both the kinetic energy of the coolant flow relative to the kinetic energy of the primary flow at blade exit and the reduction in trailing-edge loss that occurs with trailing-edge coolant discharge.

Figure 7 showed that for a given coolant flow rate, the coolant- to primary-air exit-velocity ratios increase almost inversely with the size of the slot width (as would be expected, since to obtain constant mass flow with constant exit density, the velocity of the flow must increase inversely as the flow area). For a given coolant flow rate, the increase in output resulting from the coolant flow kinetic energy would then be larger for the blading with smaller slot widths than for the blading with larger slot widths.

From zero to about 2 percent coolant flow rate, the results in figure 8 show, in general, an increased gain in efficiency with increased slot widths. Considering the discussion of the preceding paragraphs, the increased gain in efficiency of the blading with increased slot widths between zero and 2 percent coolant flow rate would apparently have to result from the following: At the lower coolant flow rates, the increase in kinetic-energy output of the blading with smaller slot widths must be less than the increase in output due to reduction in trailing-edge loss of the blading with larger slot widths. At 2 percent coolant flow rate, the percent increase in efficiency of all the blading is about the same. Above 2 percent coolant flow rate, the results generally show an increase in percent change in primary-air efficiency with coolant flow rate as the slot size of the bladings decreased. This trend apparently results from the fact that, at larger coolant flow rates, the increase in kinetic energy output of the blading with smaller slot widths is larger than the increase in output due to reduction in trailing-edge loss of the blading with larger slot widths.

In the preceding discussion it was stated that above 2 percent coolant flow rate, the results generally show an increase in percent change in primary-air efficiency with coolant flow rate as the slot size of the blading is decreased. There is an exception, however. This exception occurs for the blading with the 0.102-centimeter (0.040-in.) slot width and the blading with the 0.127-centimeter (0.050-in.) slot width. Above about 4 percent coolant flow rate, the rate of percent change in efficiency for the blading with the smaller slot width is less than for the blading with the larger slot width. Finally, at 8 percent coolant flow rate, the blading with the smaller slot width has about $1\frac{1}{4}$ percent less change in efficiency than the blading with the larger slot width. This is not as expected, since at 8 percent coolant flow rate, the blading with the smaller trailing-edge slot would be expected to have a substantially larger increase in kinetic energy output (about 2.5 percent) than the blading with the larger slot width, as can be deduced from figure 7.

Reasons for the discrepancy in results noted in the last paragraph can only be speculated upon. There is some evidence that there may have been uneven distribution of coolant flow both among the three test blades and also spanwise through the coolant ejection slots. This would result in test errors and subsequent inconsistencies in primary-air efficiency calculations, since all data relating to the total flow were obtained only for the middle test blade near the mean section of the blading; whereas, all coolant flow data were based on even distribution of the coolant flow both spanwise and among the three test blades. The inconsistency in primary-air efficiency calculations that would result are indicated by equation (1).

If unequal distribution of coolant flow to the three test blades did occur, it may have been caused by an inadequate-size coolant manifold, inadequate flow area from the manifold to the blading, or by the effects of the structural webs or other design features inside the hollow blading (see figs. 1 and 3).

Although it is not known whether or not the speculative maldistribution of flow actually occurred, future research tests should be conducted with larger coolant manifolds, with the coolant flow inlet area from manifold to blading made as large as possible.

Correlation of Primary-Air Efficiency With Coolant- to Primary-Air Exit-Velocity Ratio

As discussed in the previous section of the report, part of the additional output of the coolant flow results from its kinetic energy. Regardless of the trailing-edge slot configuration, it might be expected that the additional kinetic energy output per percent coolant flow for a given ratio of coolant- to primary-air velocity would be the same.

In addition to the extra output due to the kinetic energy of the coolant, there is also evidence that some additional output occurs due to the coolant flow reducing the trailing-edge loss (see refs. 1, 2, and 5). It seems reasonable to assume that for a given trailing-edge configuration, the reduction in trailing-edge loss might also be a function of coolant- to primary-air velocity ratio. This assumption seems reasonable, since, with zero coolant flow, the trailing-edge loss might be expected to be maximum and then decrease as the coolant flow (and, consequently, the momentum) is increased in the stagnation area at the blade trailing edge.

Based on the speculation of the preceding paragraphs, the percent change in primary-air efficiency per percent coolant flow is presented in figure 9(a) as a function of coolant- to primary-air exit-velocity ratio. The figure shows that the correlation between the two bladings with the small slot width and between the three bladings with the larger slot widths is reasonably good. However, the correlation between the blades with small slot width and the blades with larger slot widths is rather poor, particularly in the lower range of exit velocity ratios.

The results in figure 9(a) show that for coolant- to primary-air exit-velocity ratios from zero to about 1.0, the change in primary-air efficiency per percent coolant flow is less for the two blade sets with the small slot width than for the three with the larger slot widths. The general trend of increasing improvement in efficiency per percent coolant with increasing slot width for the same trailing-edge thickness was not unexpected, since the reduction in trailing-edge loss would logically be expected to increase with increasing slot width (ref. 2).

The correlation of percent change in efficiency per percent coolant flow as a function of coolant- to primary-air exit velocity, although somewhat unsatisfactory, was the best method found by the authors for correlating the results. Therefore, as a matter of interest, the results shown in figure 9(a) were used to compute the arithmetic average percent change in efficiency per percent coolant flow as a function of coolant- to primary-air exit-velocity ratio for the five different slot configurations. The results, presented in figure 9(b), show that the average percent change in efficiency per percent coolant flow varies approximately linearly from 0 to 1.4 percent in the range of coolant- to primary-air velocity ratios between zero and 1.2.

The data of figure 9(b) were then used, together with the data of figure 7, to obtain an estimated percent change in primary-air efficiency as a function of coolant flow rate for the five test blade configurations. The estimated percent change was obtained as follows: At a given ratio of coolant- to primary-flow exit velocity, the average percent change in efficiency per percent coolant flow was read from figure 9(b). For the particular blade configuration in question, the value of coolant flow rate corresponding to the coolant- to primary-air exit-velocity ratio being considered was then read from figure 7. The percent change in primary-air efficiency was then computed for the particular blade configuration and coolant flow rate in question. Thus, as is obvious,

$$\left(\frac{\Delta\eta}{\eta_0}\right)_{p,3}(100) = \frac{\Delta\eta}{\eta_0}\left(\frac{100}{y}\right)(y) \tag{3}$$

The computed change in primary-air efficiency as a function of coolant rate was then compared with the experimental change in primary-air efficiency for the five test blade configurations. The results are presented in figure 10. In all cases, except for the blading with the 0.102-centimeter (0.040-in.) slot width in the upper range of coolant rates, the agreement between the computed results and the experimental results is within about 1 percent.

It is not known whether the computed change in primary-air efficiency obtained from the correlation of percent change in primary-air efficiency per percent coolant flow for the five tested trailing-edge configurations would apply to blading with other trailing-edge configurations. However, if better information were not available for a particular configuration, the computation method would provide some means, based on experimental data and some logic, of predicting the change in primary-air efficiency due to trailing-edge ejection. Using the correlation of percent change in efficiency per percent coolant flow as a function of coolant- to primary-air exit-velocity ratio to predict the change in primary-air efficiency would, of course, require that the coolant- to primary-air exit-velocity ratio and the percent coolant flow of the blading in question be somehow determined.

Also, 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 should be applicable at other temperature ratios because it is dependent on coolant- to primary-air exit-velocity ratio and coolant flow rate, variables which would be determined independent of the method, regardless of coolant- to primary-air temperature ratio.

SUMMARY OF RESULTS

A cold-air experimental investigation was conducted in a two-dimensional cascade to determine the effect on turbine stator blade aerodynamic performance of coolant ejection from five different trailing-edge slot geometries. Separate tests were made for each of the different slot geometries.

The test blades have 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 trailing-edge thicknesses 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 trailing-edge thicknesses 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.).

The investigation was conducted at ideal primary-air exit critical velocity ratios of 0.5, 0.65, and 0.8 and over a range of coolant rates corresponding to coolant- to primary-air exit-velocity ratios from 0 to about 1.2.

The principal results of the investigation are presented in terms of primary-air efficiency as a function of coolant flow rate and also as a function of coolant- to primary-air exit-velocity ratios. The primary-air efficiency relates the actual kinetic energy of the combined flow to the ideal kinetic energy of the primary flow only. Changes in primary-air efficiency reported in the results refer to changes in efficiency of the slotted blading relative to the corresponding unslotted blading. The principal results follow:

1. For the five blade configurations tested, the average percent change in primary-air efficiency per percent coolant flow varied approximately linearly from 0 to 1.4 percent over the range of coolant- to primary-air exit-velocity ratios from zero to 1.2. However, there was considerable deviation from these average values between the two bladings with the small slot width and the three bladings with the larger slot widths in the lower range of exit-velocity ratios.

2. For the five blade configurations tested, except for the two blades with the small slot width at low values of coolant flow rate, the primary-air efficiency increased with increased coolant flow rate. For the two bladings with the small slot width at coolant flow rates between zero and about 1.0 percent, the primary-air efficiency first decreased slightly and then increased.

3. For the five different configurations tested, the percent change in primary-air efficiency varied differently at different coolant flow rates. At coolant flow rates up to 2 percent, the percent change in primary-air efficiency generally increased with in-

creased slot width. At 2 percent coolant flow, the percent change in primary-air efficiency was about the same for all configurations. Above 2 percent coolant flow, except for one blade configuration, an increase in slot width resulted in a smaller increase in efficiency as the coolant flow rate increased.

4. For the range of primary-air critical velocity ratios included in the investigation, there was essentially no effect of primary-air critical velocity ratio on percent change in primary-air efficiency.

5. The best correlation of the results was obtained by plotting percent change in efficiency per percent coolant flow as a function of coolant- to primary-air exit-velocity ratio. The change in primary-air efficiency as a function of coolant flow rate computed from the average percent change in efficiency per percent coolant flow as a function of exit velocity ratio agreed within 1 percent or less with the test results for all the configurations except one. This exception for this configuration occurred only in the upper range of coolant flow rates.

CONCLUDING REMARKS

The computation method presented for predicting the change in primary-air efficiency of the test blade configurations may or may not apply to blading with different trailing-edge slot geometries. However, if data were not available for blading with different trailing-edge slot geometries, the computation procedure would provide some means, based on experimental results and some logic, of predicting the change in efficiency due to coolant ejection for that geometry. Also, 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 should be applicable at other temperature ratios because it is dependent only on coolant- to primary-air exit-velocity ratio and coolant flow rate, variables which would be determined independently of the method, regardless of the coolant- to primary-air temperature ratio.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, November 14, 1973,
501-24.

APPENDIX A

TRAILING-EDGE SLOT DISCHARGE COEFFICIENTS

This appendix presents the trailing-edge slot discharge coefficients as functions of ideal slot Reynolds number and primary-air critical velocity ratio for the five blade configurations tested. The equations and assumptions used in obtaining the results are discussed.

The discharge coefficient is, of course, the ratio of the actual to the ideal flow through the trailing-edge slot. Thus,

$$C_{D,ts} = \frac{m_c}{(\rho V)_{c,i,3}(sl)(L)} \tag{A1}$$

As the subscript 3 in equation (A1) indicates, the ideal exit flow conditions were based on after-mixed conditions. The reason for this is that the local static pressure directly at the exit of the trailing edge would be difficult, if not impossible, to predict, particularly with trailing-edge coolant ejection (see ref. 15). The upstream flow conditions used for the determination of ideal density and velocity were obtained from the measured total pressure p'_c inside the blade and the coolant total temperature T'_c. The actual coolant flow was determined from data and calculations as specified by the ASME code for flat-plate orifices.

The ideal slot Reynolds number was computed from the following equation:

$$Re_{i,ts} = \frac{4R(\rho V)_{i,3}}{\mu} \tag{A2}$$

where

$$R = \frac{(sl)(l)}{2(sl + l)}$$

Figure 11 presents the trailing-edge slot discharge coefficients as functions of ideal slot Reynolds number for primary-air ideal critical velocity ratios of 0.5, 0.65, and 0.8 for the five blade configurations.

The discharge coefficients for the different trailing-edge slot widths varied with Reynolds number and primary-air ideal critical velocity ratio.

The maximum coefficients, at the maximum Reynolds number considered, varied from about 0.7 to 0.85 for the different slot widths. For all slot configurations, the coefficients decreased quite rapidly with decreasing Reynolds number.

At the higher Reynolds numbers, the coefficients were little affected by primary-air critical velocity ratio. However, as the Reynolds number decreased, the slot coefficients were increasingly influenced by primary-air ideal critical velocity ratio. For all slot configurations, at lower given values of Reynolds number, the coefficients decreased with increasing primary-air critical velocity ratio.

APPENDIX B

SYMBOLS

C_D discharge coefficient (ratio of actual to ideal mass flow)

L total spanwise length of trailing-edge slot, excluding webs, m; ft

l length of trailing-edge slot between adjacent structural webs, m; ft

m mass flow rate per blade, kg/sec; lbm/sec

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

R profile radius (ratio of cross-sectional area to wetted perimeter of flow area), m; ft

Re Reynolds number

sl trailing-edge slot width, m; ft

T temperature, K; $^{\circ}R$

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

V absolute velocity, m/sec; ft/sec

y ratio of coolant- to primary-air mass flow

η efficiency

η_p primary-air efficiency (ratio of kinetic energy of total flow to ideal kinetic energy of primary flow only)

μ viscosity, $(N)(sec)/m^2$; lbm/(sec)(ft)

ρ density, kg/m^3; lbm/ft^3

Subscripts:

c coolant flow

cr conditions at Mach 1

i ideal quantity corresponding to isentropic conditions

p primary flow

ts trailing-edge slot

0 condition for blading having no provisions for coolant flow

3 hypothetical station downstream of blading where flow conditions are considered uniform

Superscript:

' total state

REFERENCES

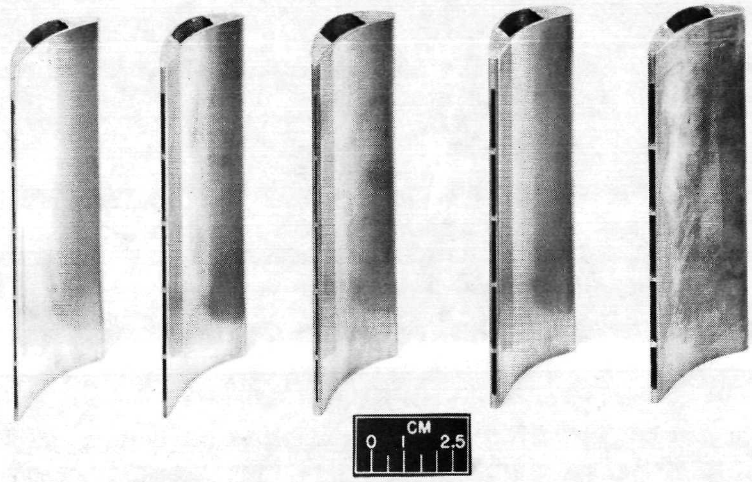
1. Hartsel, J. E.: Prediction of Effects of Mass-Transfer Cooling on the Blade-Row Efficiency of Turbine Airfoils. Paper 72-11, AIAA, Jan. 1972.
2. Prust, H. W., Jr.: An Analytical Study of the Effect of Coolant Flow Variables on the Kinetic Energy Output of a Cooled Turbine Blade Row. Paper 72-12, AIAA, Jan. 1972.
3. Brown, Douglas B.; and Helon, Ronald M.: Cold-Air Aerodynamic Study in a Two-Dimensional Cascade of a Turbine Stator Blade With Suction-Surface Film Cooling. NASA TM X-2685, 1973.
4. Whitney, Warren J.; Szanca, Edward M.; and Behning, Frank P.: Cold-Air Investigation of a Turbine With Stator-Blade Trailing-Edge Coolant Ejection. I - Overall Stator Performance. NASA TM X-1901, 1969.
5. Prust, Herman W., Jr.; Behning, Frank P.; and Bider, Bernard: Cold-Air Investigation of a Turbine With Stator-Blade Trailing-Edge Coolant Ejection. II - Detailed Stator Performance. NASA TM X-1963, 1970.
6. Szanca, Edward M.; Schum, Harold J.; and Prust, Herman W., Jr.: Cold-Air Investigation of a Turbine With Stator-Blade Trailing-Edge Coolant Ejection. III - Overall Stage Performance. NASA TM X-1974, 1970.
7. Prust, Herman W., Jr.; Schum, Harold J.; and Szanca, Edward M.: Cold-Air Investigation of a Turbine With Transpiration-Cooled Stator Blades. I - Performance of Stator With Discrete Hole Blading. NASA TM X-2094, 1970.
8. Szanca, Edward M.; Schum, Harold J.; and Behning, Frank P.: Cold-Air Investigation of a Turbine With Transpiration-Cooled Stator Blades. II - Stage Performance With Discrete Hole Stator Blades. NASA TM X-2133, 1970.
9. Behning, Frank P.; Prust, Herman W., Jr.; and Moffitt, Thomas P.: Cold-Air Investigation of a Turbine With Transpiration-Cooled Stator Blades. III - Performance of Stator With Wire-Mesh Shell Blading. NASA TM X-2166, 1971.
10. Behning, Frank P.; Schum, Harold J.; and Szanca, Edward M.: Cold-Air Investigation of a Turbine With Transpiration-Cooled Stator Blades. IV - Stage Performance With Wire-Mesh Shell Stator Blading. NASA TM X-2176, 1971.
11. Moffitt, Thomas P.; Prust, Herman W., Jr.; Szanca, Edward M.; and Schum, Harold J.: Summary of Cold-Air Tests of a Single-Stage Turbine With Various Cooling Techniques. NASA TM X-52968, 1971.

12. Whitney, Warren J.; Szanca, Edward M.; Moffitt, Thomas P.; and Monroe, Daniel E.: Cold-Air Investigation of a Turbine for High-Temperature-Engine Application. I - Turbine Design and Overall Stator Performance. NASA TN D-3751, 1967.

13. Prust, Herman W., Jr.; and Helon, Ronald M.: Effect of Trailing-Edge Geometry and Thickness on the Performance of Certain Turbine Stator Blading. NASA TN D-6637, 1972.

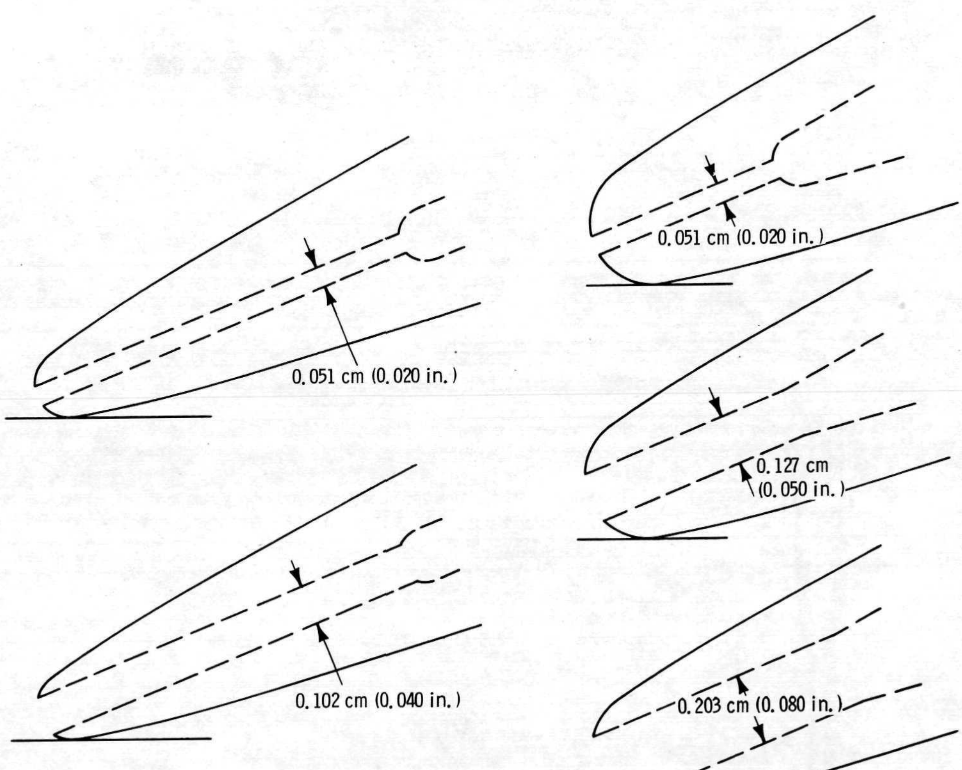
14. Stabe, Roy G.: Design and Two-Dimensional Cascade Test of Turbine Stator Blade With Ratio of Axial Chord to Spacing of 0.5. NASA TM X-1991, 1970.

15. Prust, Herman W., Jr.; and Helon, Ronald M.: Flow Conditions Around the Exit and Downstream of Certain Stator Blading With Various Trailing-Edge Thicknesses and Geometries. NASA TM X-2659, 1972.



C-73-2142

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.

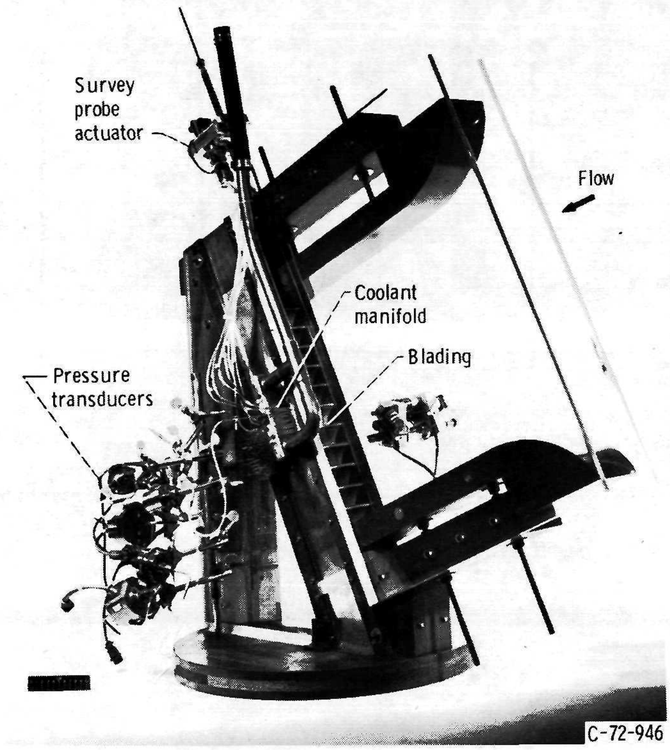


Figure 3. - Stator blade cascade.

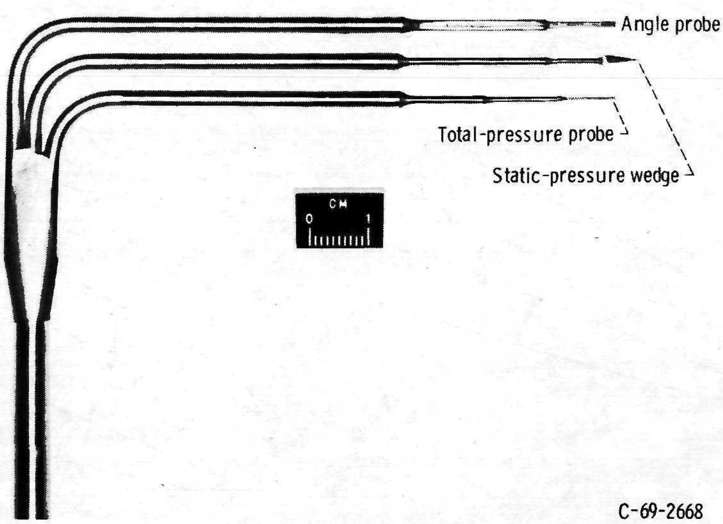
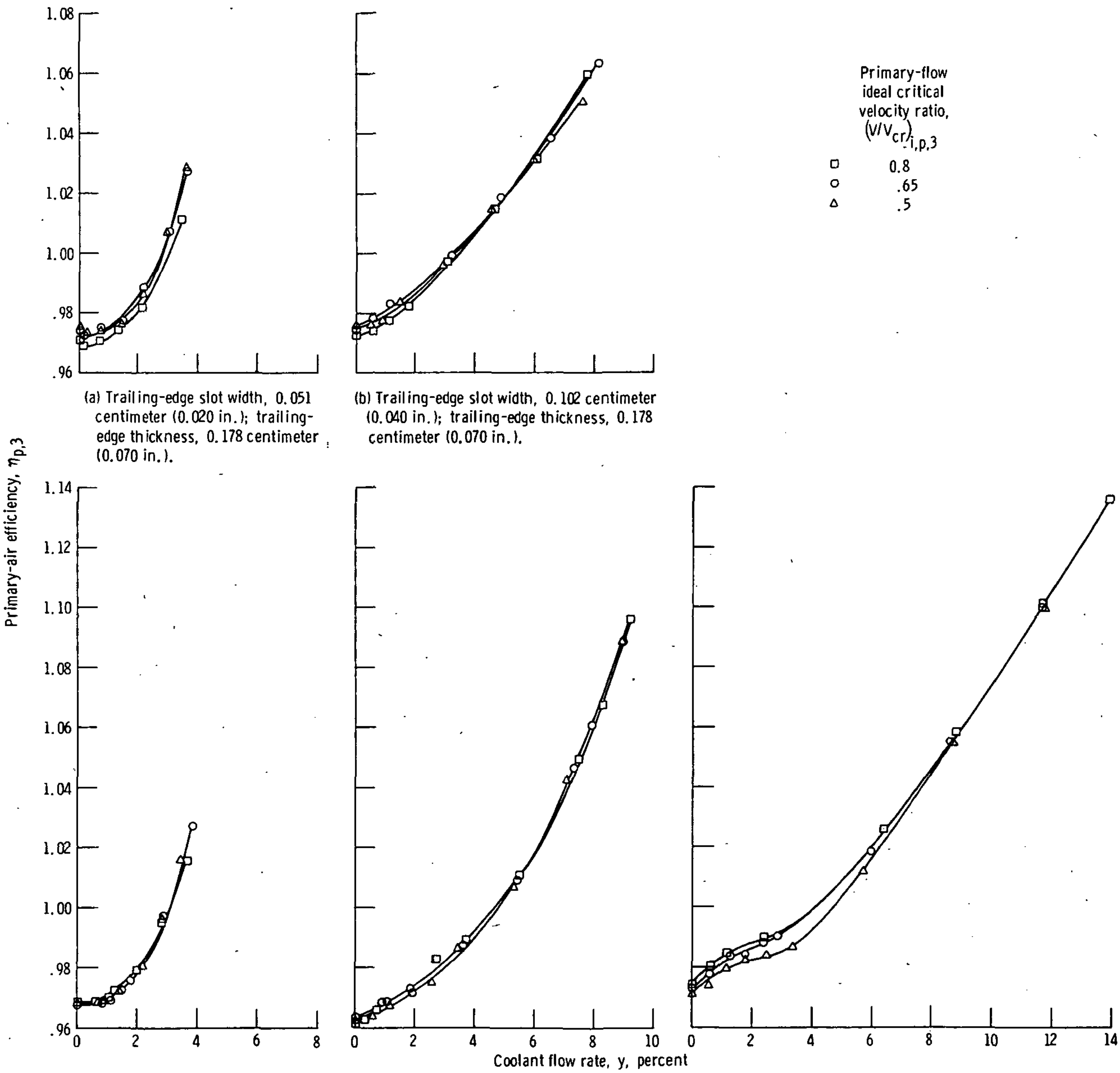


Figure 4. - Combination exit survey probe.



(a) Trailing-edge slot width, 0.051 centimeter (0.020 in.); trailing-edge thickness, 0.178 centimeter (0.070 in.).

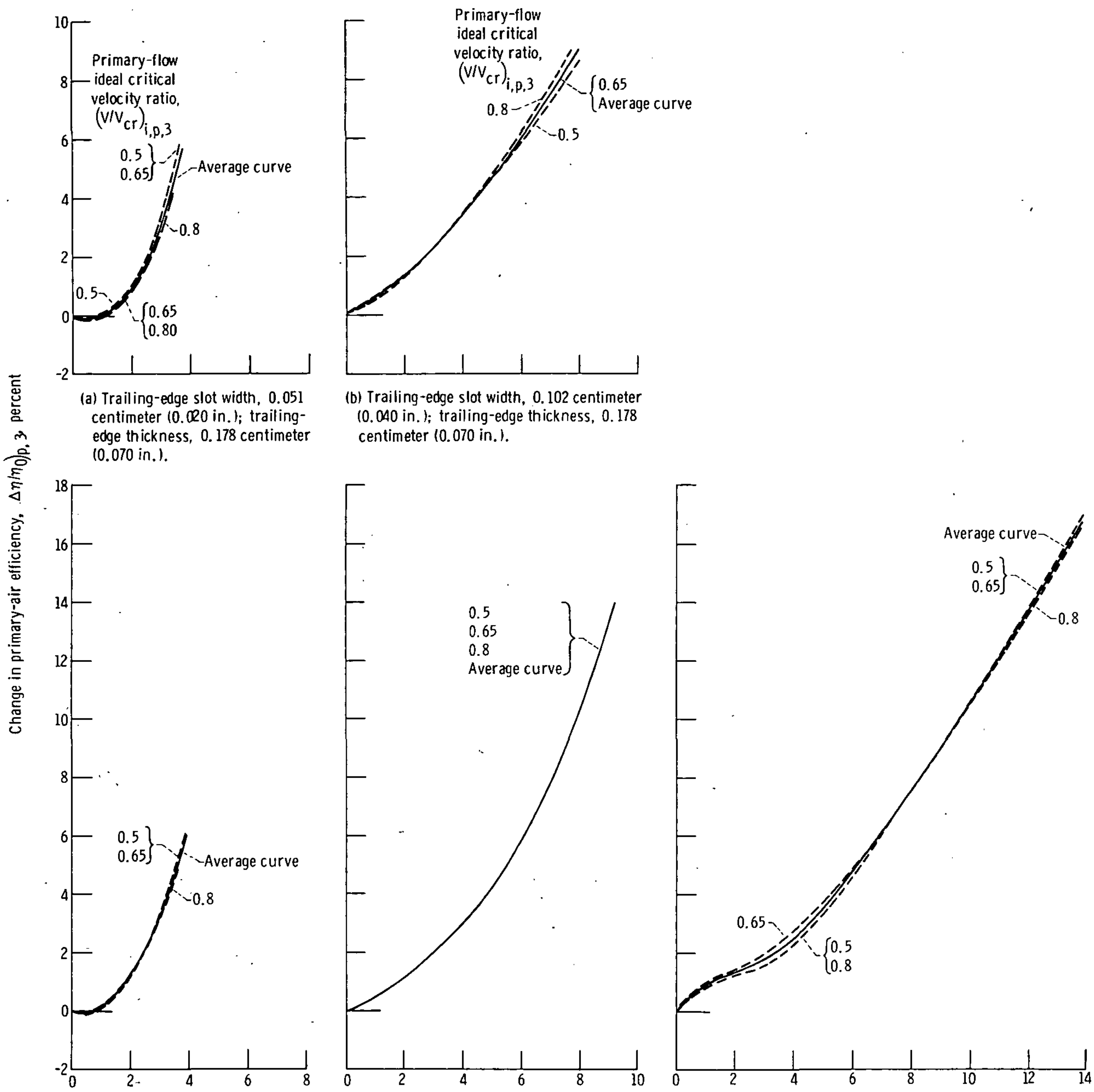
(b) Trailing-edge slot width, 0.102 centimeter (0.040 in.); trailing-edge thickness, 0.178 centimeter (0.070 in.).

(c) Trailing-edge slot width, 0.051 centimeter (0.020 in.); trailing-thickness, 0.330 centimeter (0.130 in.).

(d) Trailing-edge slot width, 0.127 centimeter (0.050 in.); trailing-edge thickness, 0.330 centimeter (0.130 in.).

(e) Trailing-edge slot width, 0.203 centimeter (0.080 in.); trailing-edge thickness, 0.330 centimeter (0.130 in.).

Figure 5. - Primary-air efficiency as function of coolant flow rate and primary-flow ideal critical velocity ratio for bladings with different trailing-edge slot geometries.



(a) Trailing-edge slot width, 0.051 centimeter (0.020 in.); trailing-edge thickness, 0.178 centimeter (0.070 in.).

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(e) Trailing-edge slot width, 0.203 centimeter (0.080 in.); trailing-edge thickness, 0.330 centimeter (0.130 in.).

Figure 6. - Percent change in primary-air efficiency relative to uncooled blading as function of coolant flow rate and primary-flow ideal critical velocity ratios for bladings with different trailing-edge geometries.

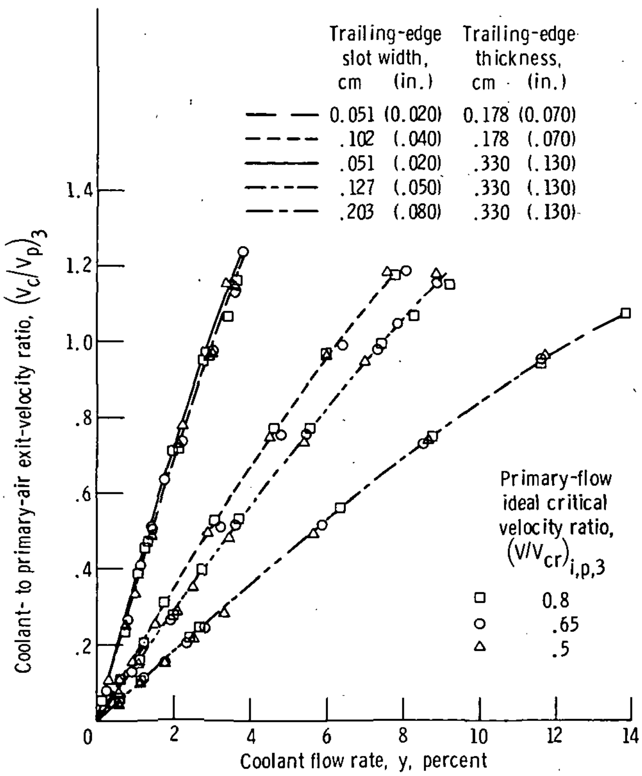


Figure 7. - Variation of coolant- to primary-air exit-velocity ratio as a function of coolant flow rate for different trailing-edge slot geometries and primary-flow critical velocity ratios.

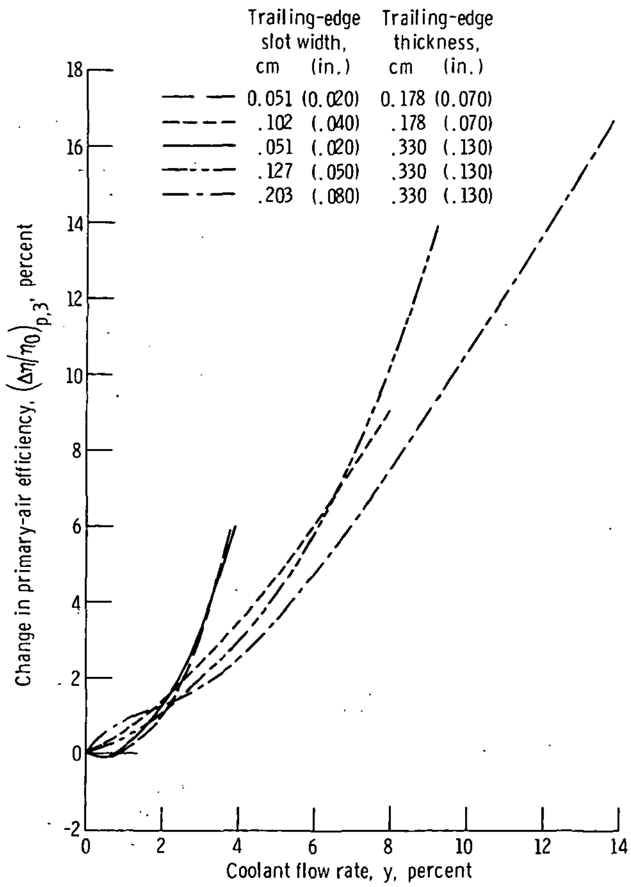
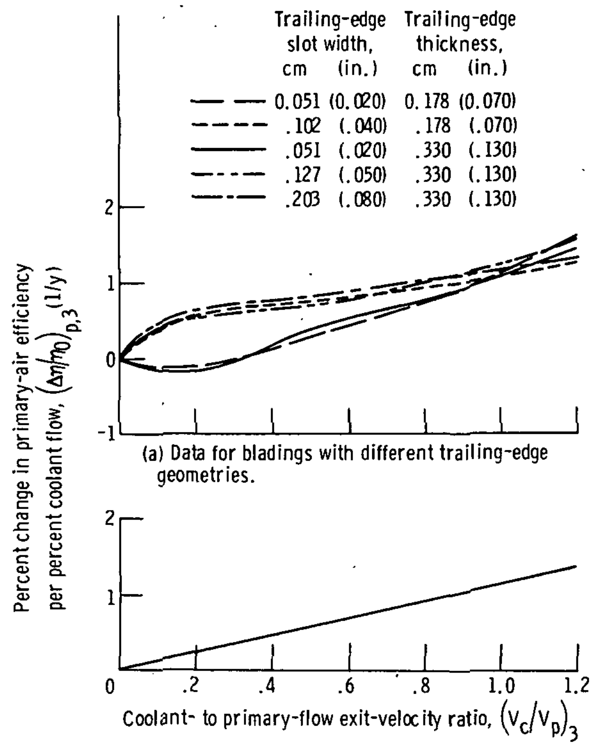
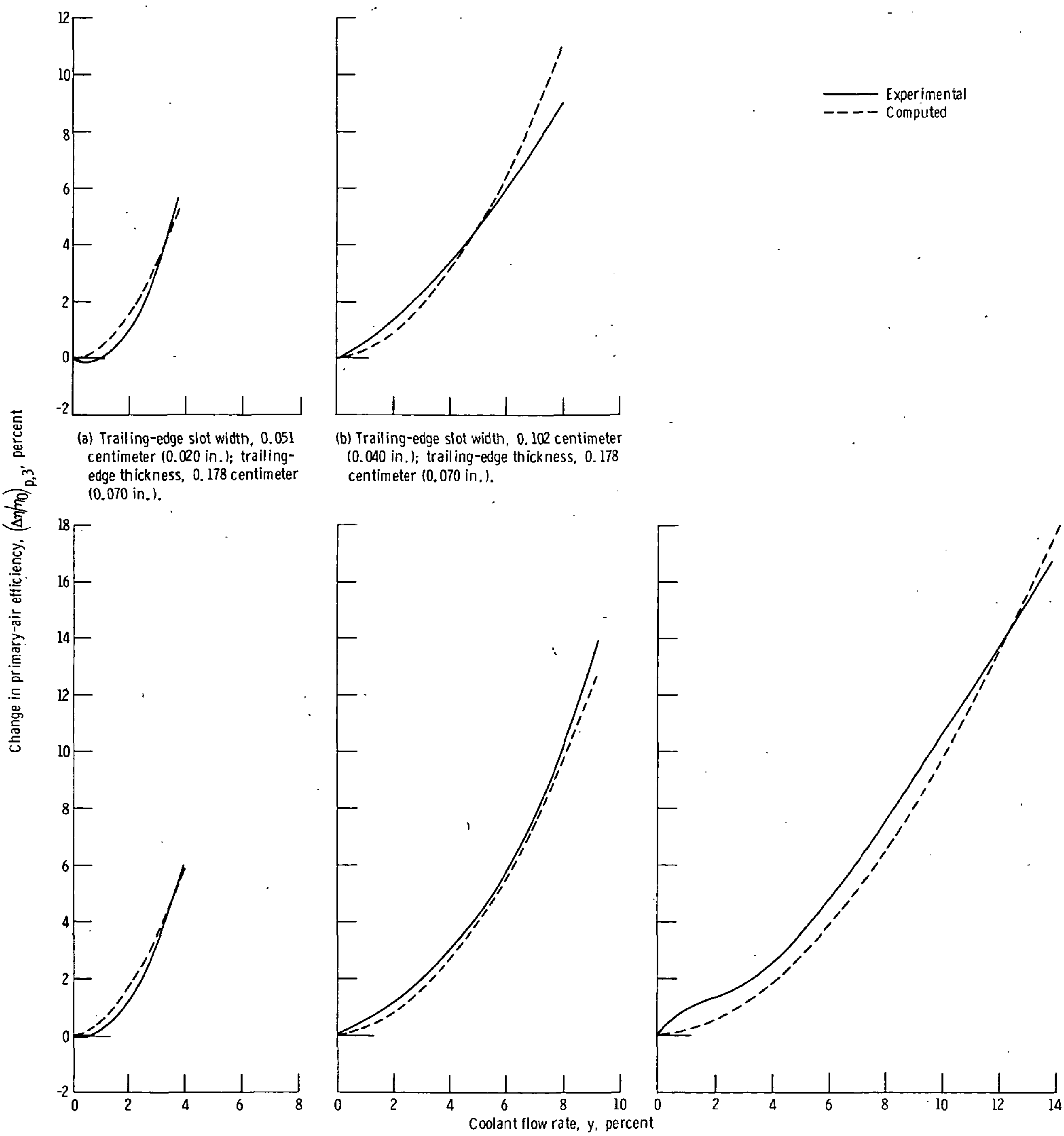


Figure 8. - Comparison of percent change in primary-air efficiency relative to uncooled blading as function of coolant flow rate for bladings with different trailing-edge geometries.



(b) Average data for bladings with different trailing-edge geometries.

Figure 9. - Percent change in primary-air efficiency per percent coolant flow as function of coolant- to primary-flow exit-velocity ratio for bladings with different trailing-edge geometries.



(a) Trailing-edge slot width, 0.051 centimeter (0.020 in.); trailing-edge thickness, 0.178 centimeter (0.070 in.).

(b) Trailing-edge slot width, 0.102 centimeter (0.040 in.); trailing-edge thickness, 0.178 centimeter (0.070 in.).

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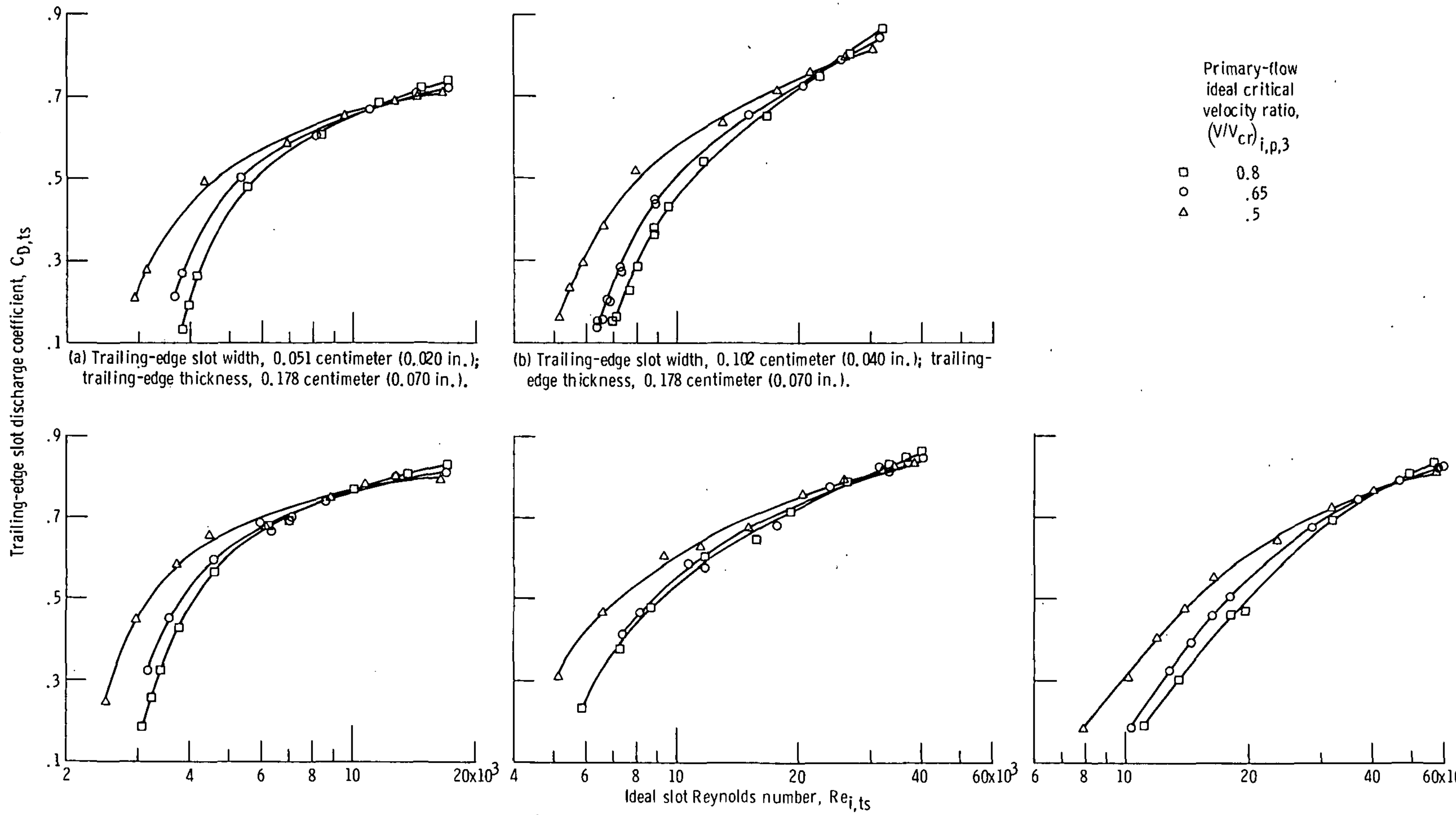
(e) Trailing-edge slot width, 0.203 centimeter (0.080 in.); trailing-edge thickness, 0.330 centimeter (0.130 in.).

Figure 10. - Comparison of experimental and computed variations in primary-air efficiency as functions of coolant flow rate.



(a) Trailing-edge slot width, 0.051 centimeter (0.020 in.); trailing-edge thickness, 0.178 centimeter (0.070 in.).

(b) Trailing-edge slot width, 0.102 centimeter (0.040 in.); trailing-edge thickness, 0.178 centimeter (0.070 in.).

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(e) Trailing-edge slot width, 0.203 centimeter (0.080 in.); trailing-edge thickness, 0.330 centimeter (0.130 in.).

Figure 11. - Trailing-edge slot discharge coefficient as function of ideal slot Reynolds number and primary-air critical velocity ratio for different slot geometries.

NASA-Langley, 1974 E-7743

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