NASA TECHNICAL MEMORANDUM

NASA TM X-3442

NASA TM X-3442

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

III - Effect of Hole Size on Single-Row and Multirow Ejection

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • OCTOBER 1976

1. Report No.	2. Government Accession No.	3. Recipient's Catalog No.
NASA TM X-3442		

4. Title and Subtitle	5. Report Date
TWO-DIMENSIONAL COLD-AIR CASCADE STUDY OF A FILM-COOLED TURBINE STATOR BLADE. III - EFFECT OF HOLE SIZE ON SINGLE-ROW AND MULTIROW EJECTION	October 1976
	6. Performing Organization Code
7. Author(s)	8. Performing Organization Report No.
Herman W. Prust, Jr. and Thomas P. Moffitt	E-8720
	10. Work Unit No.
9. Performing Organization Name and Address	505-04
Lewis Research Center National Aeronautics and Space Administration Cleveland, Ohio 44135	11. Contract or Grant No.
	13. Type of Report and Period Covered
12. Sponsoring Agency Name and Address	Technical Memorandum
National Aeronautics and Space Administration Washington, D.C. 20546	14. Sponsoring Agency Code

15. Supplementary Notes

16. Abstract

The effect of coolant discharge on the aerodynamic performance of a film-cooled turbine stator blade was determined. The blade had the same number, location, and injection angle of coolant holes, but the coolant hole diameters were one half that of a previously investigated blade. Otherwise the blades were the same. The study included tests with discharge from individual coolant rows and multiple coolant rows, including full-film discharge. The results of the blade with smaller holes are reported and compared with the blades with larger holes.

17. Key Words (Suggested by Author(s))		18. Distribution Statement
Turbine	Film-cooling	Unclassified - unlimited
Stator blade	Cascades	STAR Category 02
Aerodynamics		

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified	33	$4.00

* For sale by the National Technical Information Service, Springfield, Virginia 22161

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

III - EFFECT OF HOLE SIZE ON SINGLE-ROW AND MULTIROW EJECTION

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SUMMARY

An experimental investigation was conducted to determine the effect of coolant discharge on the aerodynamic performance of a full film-cooled turbine stator blade. The blade had the same number, location, and injection angle of coolant holes, but the coolant hole diameters were half that of a previously investigated stator blade. The blade had 12 spanwise coolant rows, 6 on the suction surface, 6 on the pressure surface. Tests were made with discharge from each of the coolant rows as well as various combinations of coolant rows, including full film discharge.

The investigation was made to determine the performance of a stator blade with 0.038-centimeter (0.015-in.) diameter coolant holes and to compare its performance with the performance of a stator blade with larger coolant holes (0.076 cm (0.030 in.) diam).

Comparative full film cooling results show that the blade with the larger holes utilized the ideal specific energy of the coolant flow more efficiently than the blade with the smaller coolant holes. For instance, when the ideal specific energies of the coolant and primary flows are equal for both blades, the percent increase in primary efficiency per percent coolant flow is about 0.65 for the larger hole blade compared with about 0.45 for the smaller hole blade. This was shown to be due to better utilization of the coolant flow ideal energy by the larger hole blade discharging to surface static pressures lower than blade exit static pressure.

As with the larger hole blade the experimental multiple coolant row efficiency results for the smaller hole blade agreed quite well with multiple coolant row results predicted by adding experimental single-row efficiencies. Apparently coolant ejected from upstream rows of holes did not significantly affect the output of coolant ejected from downstream rows.

INTRODUCTION

An extensive research program is in progress at Lewis to investigate experimentally and analytically (e.g., refs. 1 and 2) the effect of coolant ejection on turbine efficiency. There are different means of ejecting coolant air from the turbine blade surface, and these have significantly different effects on turbine efficiency. Some of the coolant schemes previously tested were stator blade trailing-edge ejection (refs. 3 to 7) and stator blade transpiration cooling (refs. 8 to 11). The results of the investigations of references 3 to 5 and 9 to 11 are summarized in reference 12. Currently a program is in progress to study the factors affecting the aerodynamic performance of full-film-cooled stator blades.

The results of an experimental investigation of the influence of coolant ejection on the performance of a full-film-cooled stator blade were reported in references 13 and 14. Tests were conducted with coolant ejected from each of the coolant rows separately, from combinations of coolant rows, and from all coolant rows together. The location of the coolant rows on the blade surface affected the change in efficiency relative to the efficiency of the uncooled blade row. The output energy from the blade was greater for coolant rows discharging to local pressures higher than blade-row exit static pressures than for those discharging to local pressures lower than blade-row exit static pressure. Multirow efficiencies predicted from single-row experimental efficiencies agreed with multirow experimental results.

The subject stator blade had the same number, location, and ejection angle of coolant holes but half the hole diameter of the blade of references 13 and 14. Otherwise, the blades were the same. The purpose of the investigation is to determine the performance of the smaller coolant hole blade and to compare its performance with that of the larger coolant hole blade. The testing was conducted in a two-dimensional cascade. The temperatures of the primary and coolant air were nearly the same. The nominal ideal primary-air critical velocity ratio at the blade row exit was 0.65 only, since the investigations of references 13 and 14 showed that performance was only slightly affected by critical velocity ratio in the range of 0.5 to 0.8.

The results are reported in terms of fractional change in primary efficiency compared with the efficiency of the noncooled blade row. Primary efficiency is defined as the ratio of the actual kinetic energy of the total flow (primary plus coolant) to the ideal kinetic energy of the primary flow only. To determine if coolant ejected from upstream rows of holes affects the output energy of coolant from downstream rows, the experimentally determined changes in primary efficiency for multiple rows are compared with predictions obtained from addition of the single-row experimental results. Also, the results of the investigation of references 13 and 14 are compared with the results of this investigation to determine the effect on stator blade performance of halving the coolant hole diameters while maintaining the number and location of coolant holes.

SYMBOLS

k_p	coolant pressure coefficient, $(p'_c - p_m)/(p'_{p,0} - p_m)$
L	coolant hole length, cm; in.
L_{pr}	pressure-surface length from leading edge to trailing edge (fig. 2), cm; in.
L_s	suction-surface length from leading edge to trailing edge (fig. 2), cm; in.
p	absolute pressure, N/cm^2; psia
V	absolute velocity, m/sec; ft/sec
w	mass flow rate, kg/sec; lbm/sec
x	local position along blade surface from leading edge (fig. 2), cm; in.
y	coolant fraction, w_c/w_p
β	angle between coolant hole axis and local blade surface tangent in plane parallel to end wall surface, deg
η_o	blade-row efficiency with no coolant flow
η_p	primary efficiency, ratio of kinetic energy of total flow to ideal kinetic energy of primary flow only

Subscripts:

c	coolant flow
cr	conditions at Mach 1
id	ideal quantity corresponding to isentropic process
m	station at blade exit where flow conditions are assumed to be uniform
o	conditions with no coolant flow
p	primary flow
0	station at blade row inlet

Superscript:

'	total-state conditions

APPARATUS, INSTRUMENTATION, AND PROCEDURE

Blading

Photographs of the test blade showing the 12 rows of coolant holes - 6 rows on the pressure surface and 6 rows on the suction surface - are presented in figure 1. The blade is hollow and untwisted and of constant cross section. The blade profile corresponds to the mean section profile of the stator blade of reference 15, in which report the blade is described in detail. Significant dimensions of the blade are as follows: span, 10.16 centimeters (4.0 in.); chord, 5.74 centimeters (2.26 in.); pitch, 4.14 centimeters (1.63 in.).

The profile of the subject blade and the location, geometry, and numbering system of the coolant rows are shown in figure 2 and table I. (The symbols used in table I are illustrated in fig. 2. All symbols are defined in SYMBOLS section.) The axes of all coolant holes are parallel to the planes of the blade end walls. The diameter and pitch of the coolant holes in all rows are 0.038 centimeter (0.015 in.) and 0.114 centimeter (0.045 in.), respectively.

The tested blade had the same number and location of coolant holes, but the coolant hole diameters were one half that of the blade of references 13 and 14. Otherwise the blades were the same.

Cascade

The blade was tested in the simple two-dimensional cascade shown in figure 3. There are 12 blades in the cascade. However, only the three blades near the center are cooled. Other details of the cascade are described in reference 16. Primary (atmospheric) air enters the cascade inlet shown on the right in figure 3, and coolant air enters the inside of the three hollow blades near the center of the cascade through the coolant manifold and associated piping. The survey probe actuator indicated in figure 3 operates a slide in which a multipurpose survey probe is mounted downstream of the blades. The coolant and primary flow passing through the blades is discharged from the cascade through exhaust piping attached to the circular base of the cascade.

Instrumentation

A calibrated, multipurpose survey probe located downstream of the blades of the type shown in figure 4 was used to determine the flow angle, the static pressure, and the loss in total pressure across the blade row. (A detailed description of this probe is given in ref. 16). Coolant total pressure p_c' inside the test blade was measured with a total-pressure probe. The sensing element of the probe was located 2.54 centimeters (1.0 in.) from the blade end wall on the coolant manifold side. The circular sensing end of the probe faced the coolant flow entering the blade so that the total pressure inside the blade was measured as accurately as was practicable.

Coolant flow was measured by using calibrated sharp-edged orifice plates of various sizes located in an orifice run either 2.54 or 5.08 centimeters (1 or 2 in.) in diameter. The orifice runs, including instrumentation and orifice plates, all conformed to ASME specifications. All pressure data taken during survey tests were measured by using calibrated strain-gage transducers.

Test Procedure

For testing, three blades of the same profile with the same row or rows of coolant holes open were installed near the center of the cascade. Coolant air was then supplied to these blades only. Data were taken for only the middle blade 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, data were taken at the midspan of the blade only.

In order 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 airflow is provided by the laboratory combustion air system. Desired coolant flow rates were obtained by first setting the upstream orifice pressure with an upstream pressure regulator and then setting the pressure ratio across the orifice plate by regulating a throttling valve downstream of the orifice plate.

Before the blade survey testing was started, blade surface static pressures were determined from manometer board readings.

In order to conduct survey tests, the desired primary-air critical velocity ratio and coolant fraction were established by regulating the primary and coolant flow control valves. A survey was then made with the multipurpose probe across one blade pitch of the middle test blade to determine the downstream flow conditions. During the survey, all data, including survey data and coolant flow data, 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 feedback from the laboratory data processing center.

Separate tests were first made with coolant ejection from each of the 12 single rows of coolant holes. Next the effect of multirow coolant ejection from the suction and pressure surfaces were individually determined. In the multirow tests the combinations of rows considered were the two rows nearest the leading edge, the three rows nearest the leading edge, etc., until all six rows on the separate surfaces were included. Finally tests were conducted to determine the effect on stator blade performance of full film cooling from all 12 coolant rows. The multiple coolant row tests are listed in table II.

The survey investigations of both single-row and multirow coolant discharge were conducted at a nominal ideal primary-air exit critical velocity ratio $(V/V_{cr})_{p,id,m}$ of 0.65 only, since previous investigations (e.g., refs. 13 and 14) have shown that the influence of coolant flow on stator performance is only slightly affected by primary-air critical velocity ratios between 0.5 and 0.8.

For the single-row tests the range of coolant fractions investigated was from zero to about 0.01. For the multirow tests the range of coolant fractions investigated varied for the following reasons: With multirow discharge the minimum practical coolant fraction occurs when the total pressure inside the blade p'_c is a little higher than the blade surface pressure p_s of the coolant row nearest the leading edge (highest local static pressure) of the blade. If the total pressure inside the blade is lower, primary air flows abnormally into the interior of the blade through the row nearest the leading edge and out coolant rows farther downstream. The minimum coolant fractions for multirow ejection were determined by this consideration. The maximum coolant fractions were determined by the number of coolant rows open with the absolute total pressure inside the blade limited to about 14.9 newtons per square centimeters (21.6 psia).

ANALYSES

The analyses are presented in four parts. The method of computing test results is presented first. The reasons for reporting results in terms of primary efficiency is discussed second. The method of comparing efficiency results for the blades with different size holes is discussed third and the method of predicting efficiency results from single row data is presented last.

Computation of Test Results

The general procedure for computing the test results was as follows: Coolant fractions were computed by using 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. These local values were next integrated at the measuring station. Then, with conservation of mass and momentum assumed, the integrated values at the measuring station were equated to the same quantities at the hypothetical aftermixed downstream station. These equations were then solved simultaneously to obtain the aftermixed flow conditions. (Equations for the survey data calculation procedure may be found in ref. 17.) With the aftermixed flow conditions known, the primary efficiency, as well as other results of interest, could be computed at fully mixed flow conditions.

Efficiency and Pressure Coefficient Definitions

There are a number of efficiency expressions commonly used to describe the performance of high-temperature turbines requiring coolant. For cold aerodynamic tests with no internal inserts to duplicate actual hot-engine heat transfer or pressure drop processes, the selection becomes arbitrary. The major parameter studied in the subject aerodynamic tests was the effect of ejected coolant on the output kinetic energy of the combined flow (primary plus coolant). Therefore, primary efficiency was selected as the most direct form of efficiency to investigate changes in output energy as affected by adding coolant. Primary efficiency relates the actual energy of the total flow to the ideal energy of only the primary flow and is expressed as

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

and in terms of isolated flows is equivalent to

$$\eta_p = \frac{w_p V_{p,m}^2 + w_c V_{c,m}^2}{w_p V_{p,id,m}^2} \tag{2}$$

where $w_p V_{p,m}^2 / w_p V_{p,id,m}^2$ is the efficiency of the primary flow. Equation (1) was used to compute experimental results. Thermodynamic efficiency is the same as primary efficiency except that the ideal energy of the coolant flow is included in the denominator.

$$\eta_{th} = \frac{w_p V_{p,m}^2 + w_c V_{c,m}^2}{w_p V_{p,id,m}^2 + w_c V_{c,id,m}^2} \tag{3}$$

A coolant pressure coefficient is used to show the relative pressure drops between the coolant flow and the primary flow

$$k_p = \frac{p_c' - p_m}{p_{p,0}' - p_m} \tag{4}$$

For cold air tests with equal inlet temperatures, this pressure coefficient also indicates the relative ideal specific energies between the coolant and primary flows.

Comparison of Blading with Different Size Holes

The subject blade had the same number and location of coolant holes but half the hole diameter of the blading of references 13 and 14. Otherwise the blading was the same. As a consequence, for the same state conditions of coolant and primary flows, the fractional coolant flow with respect to the primary flow is about four times greater for the blade with the larger holes than for the blade with the smaller holes. Because the coolant fractions for the same state conditions of the flows are not the same, it was necessary to select appropriate correlating parameters for comparing the efficiencies of the two blades.

The analysis of reference 2 assumes that the percent change in output energy per percent of coolant flow is dependent on properties of the coolant such as velocity and density, and not on the amount of coolant. For the same pressure and temperature conditions for the two blades, the coolant properties at the coolant hole exits are essentially the same. Therefore, the parameters used to compare the two blades are the percent change in efficiency per percent coolant $(\Delta\eta_p/y\eta_0)$ at conditions of equal ideal specific energies (pressure coefficients k_p equal).

A significant state condition for comparing the efficiencies of the blades with different size holes is when the ideal specific energy of the coolant and primary flows are equal. For cold-air tests at equal inlet temperatures, this condition occurs when the pressure coefficient k_p equals unity (which is equivalent to the inlet total pressures of two flows being equal, i.e., $p_c' = p_{p,0}'$). For this condition the change in primary efficiency relative to the efficiency of the uncooled blade is caused by the coolant and represents that part of the output kinetic energy contributed by, or charged to, the coolant. In effect, it represents the efficiency of the coolant flow and is a useful measure for evaluating various coolant schemes for the same or different blades.

Prediction of Multirow Performance from Single-Row Data

A commonly used method of presenting the effect of coolant on efficiency for both single-row and multirow tests is to plot the fractional change in primary efficiency

$\Delta\eta_p/\eta_o$ against coolant fraction y, where $\Delta\eta_p = \eta_p - \eta_o$ and η_o is the efficiency of the blade row with no coolant.

In predicting the multirow efficiency from the single-row efficiency, it is assumed that the efficiency of the primary flow is unaffected by the coolant flow.

In adding the single-row data to predict multirow performance, the following assumptions were made concerning each given single-row condition applied to multirow conditions (see SYMBOLS for definitions):

(1) Constant w_c for same cavity pressure
(2) Constant $V_{c,m}$ (no change in loss)
(3) Constant $V_{p,id,m}$ (same setting condition)
(4) Constant $V_{p,m}$ (no change in efficiency of primary air)

With these assumptions, the change in efficiency for the multirow case in terms of single-row conditions is calculated using the method described in detail in reference 14.

RESULTS AND DISCUSSION

The results of an experimental investigation of the performance of a full-film-cooled turbine stator blade having 12 spanwise coolant rows with coolant hole diameters of 0.038 cm (0.015 in.) are reported. These results are compared with the results of a previously investigated full-film-cooled turbine stator blade (refs. 13 and 14). As mentioned previously, the percent change in efficiency per percent coolant flow $(\Delta\eta_p/\eta_o)/y$ is used as the basis of comparison between the two blades for conditions of equal coolant ideal specific energies. All results reported are for a nominal primary-air ideal critical velocity ratio of 0.65.

One of the principal factors which affects efficiency is the local static pressure at the coolant hole exit relative to the blade exit static pressure (e.g., refs. 1, 2, and 14). Figure 5 shows the experimentally determined static pressures at the exit of the 12 rows of coolant holes. The shaded area shown is that portion of the blade surface where the local static pressure must increase to the blade exit and is termed the "diffusion region." The unshaded area is the expansion region and is that portion of the blade surface where the local static pressure decreases to the blade exit condition. Figure 5 shows that rows 1 to 9 discharge in the expansion region of the blade surface and rows 10 to 12 discharge in the diffusion region of the blade surface. Figure 5 will be referred to frequently to explain reasons for different trends in results for coolant discharge from different blade rows.

The results are reported in four parts. The first part presents results for the blade with smaller coolant holes with ejection from each of the 12 coolant rows tested individually. The second part presents results for the blade with the smaller holes with ejection from various combinations of coolant rows, including full-film cooling. In the third part a comparison is presented of the efficiency results for the blades with different size coolant holes. The fourth part presents a comparison for the blade with the smaller coolant holes between experimental multirow efficiencies and predicted efficiency results obtained by adding single-row data.

Single-Row Ejection Results from Smaller Holes

Figure 6 presents the experimentally determined values of efficiency η_p as a function of coolant fraction y with ejection from each of the 12 single coolant rows tested individually. The vertical marks on figure 6 and subsequent curves is the coolant fraction when the primary and coolant flows have equal inlet pressures, $p'_c = p'_{p,0}$.

The variations in efficiency with coolant fraction are shown more clearly when normalized to each of their respective zero-coolant-flow cases, as shown in figure 7. When the inlet total pressure of the coolant and primary flows are the same, the coolant fractions for the different coolant rows increase, as would be expected, as the surface static pressure of different rows decrease (see fig. 5). In the range of coolant fractions up to about 0.5 percent, except for row 9, the increase in efficiency for a given coolant fraction is significantly greater for coolant rows that discharge in the expansion region (rows 1 to 8) than for rows that discharge in the diffusion region (rows 10 to 12).

With ejection from single rows in the expansion region, except row 9, the rate of increase in efficiency with coolant fraction is substantial, and the slope of the curve increases with cooling fraction for all coolant rows except 1 and 7. These trends can be explained as follows. Referring to table I, it will be noted that the ejection angles for rows 1 and 7 are normal to the blade surface, while the ejection angles for the other coolant rows are at acute angles to the blade surface. As the coolant fraction is increased for any coolant row, the dynamic head of the coolant flow at the coolant row exit increases. For coolant rows having acute ejection angles to the blade surface, a large portion of the dynamic head of the coolant is recovered as useful kinetic energy; while, for coolant rows ejecting normal to the blade surface, the dynamic head is not recoverable as useful kinetic energy. As a consequence, rows 1 and 7 have a constant rate of change in efficiency with coolant fraction, and rows 2 to 8 have an increasing rate of change of efficiency with coolant fraction.

With ejection from single coolant rows in the diffusion region of the blade surface, the efficiency first decreases at low values of coolant fraction and then gradually changes to a rapid rate of increase in efficiency at higher values of coolant fraction. This trend is due to the following: At lower values of coolant fraction, the energy of the dynamic head of the coolant that is recoverable is less than the energy required to diffuse the coolant flow to blade-row exit. Energy must, therefore, be extracted from the primary flow to diffuse the coolant flow to blade-row exit with a resulting decrease in primary efficiency. As the coolant fraction increases, the recoverable dynamic head of the coolant increases until its energy is greater than that required to diffuse the coolant flow to blade row exit. Therefore, at higher coolant fractions, energy is added to the primary flow with a resulting increase in efficiency. The reasons just given for different trends in single-row results will also pertain to trends of multirow test results, which will be discussed later.

Comparative results for the blade with the larger holes (ref. 14) showed similar trends to those of the blade with smaller holes excepting for coolant discharge from row 9. For this row the results for the smaller holes show a decrease in efficiency at lower coolant fractions and then an increase in efficiency as the coolant fraction is increased, while the results for the larger holes show an increase in efficiency for all values of coolant fraction. The reason for the different trends of behavior of the smaller and larger holes for this row is not understood. However, figure 5 shows that this row is located near the diffusion region on the suction surface. Perhaps in this region, the boundary-layer flow has a tendency to be unstable, so that the smaller momentum of the coolant flow from the smaller holes has a different effect than the larger coolant flow momentum from the larger holes.

When the ideal specific energy of the coolant and primary flows are equal ($p'_c = p'_{p,0}$), the average percent change in efficiency per percent coolant for rows discharging into the expansion region (rows 1 to 9) is about 0.75; for coolant rows discharging into the diffusion region, it is about 0.10. At the same conditions comparative changes for the larger hole blade are about 0.80 for discharge into the expansion region, and about 0.45 for discharge into the diffusion region. For these conditions, then, the utilization of coolant ideal energy for discharge into the expansion region is about the same for both blades. However, when the coolant is discharged into the diffusion region, the utilization of coolant ideal energy is significantly greater for the larger hole blade than for the smaller hole blade.

The reasons for the smaller increase in efficiency for the blade with the smaller coolant holes discharging into the diffusion region are unknown. Perhaps, in this region, the smaller momentum of the coolant from the smaller holes (resulting from the smaller mass flow of the coolant) has less tendency to accelerate the boundary layer than the larger momentum of the coolant from the larger holes.

Multirow Ejection Results from Smaller Holes

The results of this part are presented in three sections. First, the experimental efficiency results for ejection from various combinations of coolant rows on the pressure surface are presented. Next, similar results for ejection from the suction surface are presented. Last, full film cooling results are presented and compared with results for the pressure surface and the suction surface.

Ejection from pressure surface. - Figure 8 presents the variations in efficiency with coolant fraction for discharge from the various combinations of rows, and figure 9 compares the fractional variations in efficiency with coolant fraction for discharge from the various combinations of coolant rows. The multirow results in figures 8 and 9 are similar to the single-row results in figures 6 and 7 in that there is a substantial increase in efficiency for all values of coolant fraction. Figure 9 shows that the results for all combinations of rows tested fall within a narrow band of about 1/4 percent difference in efficiency. For given coolant fractions below about 0.025, there is, in general, a slight decrease in efficiency when 5 and 6 coolant rows are open compared with when 2 to 4 rows are open. This might be expected since, for the same coolant fraction, discharge from additional coolant rows requires lower coolant cavity pressures and ideal coolant specific energies. This then, results in lower output energy for discharge from more holes. This trend does not hold for coolant fractions above 0.025, where the increase in efficiency for the five-row configuration is greater than for the four-row configuration. The reason for this inconsistency is not understood. However, considering the accuracy limitations, the same observation is made for the smaller hole blade and larger hole blade (ref. 13); that is, for coolant discharge from the pressure surface, the output energy is essentially independent of the number or location of coolant holes.

When the ideal specific energies of the coolant and primary flows are equal ($p'_c = p'_{p,0}$), data from figure 9 shows that the percent change in efficiency per percent coolant flows varies from about 0.8 to 1.0 for pressure surface ejection. For the same conditions comparative results for the blade with larger holes (ref. 13) are closely the same as those of the blade with smaller coolant holes.

Ejection from suction surface. - Figure 10 presents the variations in efficiency with coolant fraction for discharge from suction-surface coolant holes, and figure 11 compares the fractional variations in efficiency with coolant fraction. They show an increase in efficiency with increasing coolant fraction for all combinations of blade rows.

Figure 11 shows that for a given coolant fraction there is a trend of decreasing efficiency as an increasing number of coolant rows, starting at the blade leading edge, are opened. A similar trend was found for the blade with the larger coolant holes (ref. 14). The major reason that efficiency drops off when more rows are opened (starting from the leading edge) is because of the low output from coolant discharged into the diffusing region of the blade (rows 10 to 12). First, the efficiency would decrease as the latter rows are opened, if the total coolant flow were equally split among the rows, because of the low efficiency in the diffusing region. Second, the decrease is compounded because more coolant is ejected from rows in the diffusion region due to the lower static pressures at the hole exits (see fig. 5).

The results in figure 11 show that, when the total pressure of the coolant in the blade cavity and of the inlet primary flow are equal, as an increasing number of coolant rows are opened, the percent change in efficiency per percent coolant flow decreases from about 0.80 for discharge from rows 7 to 8 to about 0.35 for discharge from rows 7 to 12. Comparative results for the blading with the large holes are about 0.75 for coolant discharge from rows 7 and 8 and about 0.45 percent for rows 7 to 12.

Full-film cooling. - The experimental variation in primary efficiency with coolant flow for the stator blade with full-film cooling is shown in figure 12. The results show a trend of increasing primary efficiency with increasing coolant fraction. A similar trend was found for the larger hole blade.

A comparison of the fractional changes in primary efficiency for coolant discharge from the six rows on the pressure surface (full pressure surface discharge), the six rows on the suction surface (full suction surface discharge), and full-film cooling is made in figure 13. A significant comparison to be made (fig. 13) is when the ideal specific energy of the coolant and primary air are equal for all three configurations ($p'_c = p'_{p,0}$). These are indicated by vertical lines on the figure. As discussed in ANALYSES, for this condition the change in primary efficiency relative to the efficiency of the uncooled blade is caused by the coolant and represents that part of the output kinetic energy contributed by, or charged to, the coolant. In effect, it represents the efficiency of the coolant air and is a useful measure for evaluating various coolant schemes for the same or different blades. For this condition data from figure 13 shows that the utilization of coolant ideal specific energy was about 0.80 for pressure surface ejection, 0.35 for suction surface ejection, and 0.45 for full-film cooling.

Comparison of Efficiency with Blade Having Larger Holes

The results of this part of the report are presented in three sections. The first part compares the results for single and multirow coolant ejection from the pressure surface of the blade. The second part compares the results for single and multirow coolant ejection from the suction surface of the blade. The last part compares the results for full film cooled discharge. All comparisons are made in terms of percent change in efficiency per percent coolant as a function of coolant pressure coefficient.

Ejection from pressure surface. - The comparative results for coolant discharge from the blade pressure surface for the two blades with different size coolant holes are shown in figures 14 and 15. Figure 14 presents results for discharge from single coolant rows and figure 15 presents results for discharge from multiple coolant rows.

Concerning the results in figure 14 for single-row discharge, for some of the lower values of coolant pressure coefficients presented, the corresponding values of coolant fraction are around 0.1 percent. At this low coolant rate an efficiency difference of 0.025 percent results in a difference in percent change in efficiency per percent coolant flow of 0.25. Because of this sensitivity, a difference in percent change in efficiency per percent coolant flow of 0.25 between the two blades is considered very good agreement in the lower range of pressure coefficients.

The comparative results in figure 14 for discharge from single coolant rows are closely the same for all rows except row 6. It may be, as shown in reference 12 for this row, that the lower percent change in efficiency per percent coolant flow for the larger hole blade results from the fact that the slope of change in efficiency for the blade with the larger holes appeared inconsistently low.

The comparative results in figure 15 for multiple coolant row discharge from the pressure surface are closely the same for all combinations of rows except rows 1 to 6 in the upper range of coolant pressure coefficients, where the maximum difference in percent change in output per percent coolant flow is about 0.15 greater for the blade with the larger holes. The reason for the difference in results for rows 1 to 6 is not understood.

Ejection from suction surface. - Figure 16 presents comparative results for discharge from single coolant rows on the suction surface, and figure 17 presents similar results for discharge from multiple coolant rows on the suction surface.

The comparative results in figure 16 for single-row discharge for the two blades are in good agreement for coolant discharge from rows 7 and 8 which eject well up in the expansion region of the blade surface. The percent changes in efficiency per percent coolant flow for rows 9 to 12, which discharge in or near the diffusion region, are considerably less for the blade with the smaller holes, the maximum difference being about 0.75. The reason for the smaller percent change in efficiency per percent coolant flow might be due to the smaller momentum of the coolant which has less of a tendency to accelerate the boundary layer.

The comparative results shown in figure 17 for multiple row ejection for the two blades are similar to the comparative single-row discharge results of figure 16. For the combinations of coolant rows in the expansion region, the results for the two blades are about the same. For the combinations with some rows in the diffusion region, the percent change in efficiency per percent coolant flow for the blade with the smaller holes is as much as 0.25 less than for the blade with the larger holes. Thus the comparative single-row results are reflected in the comparative multiple row results.

Full-film cooling. - The comparative full film cooling results in figure 18 show that the percent change in efficiency per percent coolant flow is from 0.1 to 0.2 less for the blade with the smaller holes than that of the larger holes. This result would be expected, since both single and multiple row results showed lower values for the blade with the smaller coolant holes. When the ideal specific energies of the coolant and primary flows are equal ($k_p = 1.0$), figure 18 shows that the blade with larger holes utilized about 65 percent of the available energy of the coolant compared with about 45 percent for the blade with smaller holes.

Comparison of Additive Single-Row Data With Multirow Data

To determine if coolant flow ejected from upstream coolant rows affects the performance of coolant flow ejected from succeeding downstream rows, the efficiency results of the single-row tests were added and compared with the efficiency results for the multiple row tests by using the method described in ANALYSES section. To do this requires knowledge of both the effect of coolant total pressure on coolant flow and the effect of coolant flow on primary flow. The effect of coolant total pressure drop on the coolant flow is presented first. Then the effect of coolant flow on primary flow is shown. Finally, the multiple-row efficiency results obtained from single-row efficiency results are compared with the experimental multiple row efficiency results.

Effect of coolant flow pressure drop on coolant flow. - In figure 19 the fractional coolant flow y for the individual coolant rows is plotted as a function of the coolant pressure coefficient k_p. The results in figure 19 show an increase in coolant fraction with increasing k_p for all blade rows, as expected, since with increasing k_p the total pressure and pressure drop of the coolant flow through the rows increase. In the lower range of coolant pressure coefficients, the coolant fraction for the different rows increase with decreasing blade surface pressure (fig. 5). This occurs, of course, because of the larger pressure drop across the coolant rows with the lower surface pressures. The results also show that for constant k_p the average coolant fraction for pressure surface discharge is less than for suction surface discharge. This result is caused by the blade surface pressure on the pressure surface being higher than that of the suction surface (fig. 5).

Effect of coolant flow on primary flow. - The reduction in primary flow resulting from single and multiple coolant row discharge is shown in figure 20. The curve shown is an average curve for all data points where the maximum deviation of any data point was less than 1/2 percent from the average curve.

The reduction in primary flow due to coolant flow was significant, amounting to roughly 0.8 percent in primary flow for 1.0 percent coolant flow. Comparative trends of reduction in primary flow due to coolant discharge were similar for the blade with the larger coolant holes although the magnitude of reduction in primary flow was generally slightly larger for the same coolant fraction.

Comparison of predicted and experimental multirow performance. - The experimental and predicted fractional variations in primary efficiency for multiple coolant row discharge are compared in figure 21 as a function of coolant fraction.

Figures 21(a) to (e) compare the experimental and predicted efficiencies for multiple coolant row discharge from the blade pressure surface. Very good agreement is shown, the difference between the experimental and predicted variations in efficiencies generally being less than 0.001.

The comparison of experimental and predicted efficiencies for multiple coolant row discharge from the blade suction surface are shown in figures 21(f) to (j). The agreement of the comparative efficiencies are considered fairly good. The differences between predicted and experimental changes in efficiency are shown to range from 0.001 to 0.004 for the various combinations of rows, the predicted efficiency differences being lower than the experimental for all cases.

The comparative experimental and predicted variations in efficiency for full film cooling are presented in figure 21(k). The agreement of the comparative efficiency variations is considered good, the differences being generally less than 0.003 over the range of coolant flows investigated.

For the blade with the larger coolant holes (refs. 13 and 14), the fractional variations in efficiency predicted from experimental single-row results are less than 0.01 different than the experimental efficiency for all combinations of rows tested. Therefore, the results of the two investigations strongly indicate that coolant ejected from upstream rows of holes does not significantly affect the output of coolant ejected from rows farther downstream.

SUMMARY OF RESULTS

The subject experimental investigation concerns the effect on turbine stator blade performance of coolant ejection for a full-film-cooled stator blade having 12 spanwise rows of coolant holes. The blade was identical in all respects to that of references 14 and 15 except the coolant hole diameters were 0.038 centimeter (0.015 in.) instead of 0.076 centimeter (0.030 in.).

The tests were conducted in a two-dimensional cascade at a nominal ideal primary-air critical velocity ratio of 0.65 with the coolant and primary air temperatures essentially equal to atmospheric.

The results of the investigation include the effect on stator blade performance of coolant discharge for the following cases: (a) single-row discharge from each of the 12 coolant rows, (b) multiple-row discharge from various combination of coolant rows on the blade pressure surface, (c) multiple-row discharge from various combinations of coolant rows on the blade suction surface, and (d) full film cooling discharge.

In addition to the other reported results for the blade with the smaller holes, the performance results of the single-row tests were added and compared with the multirow test results to ascertain if coolant flow ejected from upstream rows of holes affect the performance of coolant flow ejected from succeeding rows downstream.

The results are summarized as follows:

1. Full film cooling results show that the blade with the larger holes utilized the ideal specific energy of the coolant flow more efficiently than the blade with the smaller coolant holes. For instance, when the ideal specific energies of the coolant and primary flows are equal for both blades, the percent increase in primary efficiency per percent coolant flow is about 0.65 compared with about 0.45 for the blade with the smaller holes. This result is attributed to significantly better utilization of the coolant flow ideal specific energy by the blade with the larger coolant holes than by the blade with the smaller holes for coolant flow discharge in the diffusion region of the blade surface. This was indicated by single-row ejection results.

2. At the same state conditions of the coolant and primary flows, the blade with larger holes utilized about 45 percent of the available energy of the coolant for single-row ejection into the diffusing area compared with about 10 percent for the blade with smaller holes. For the remainder of the blade surface they both resulted in about 80 percent utilization of coolant available energy.

3. As with the larger hole blade, the results of the investigation of the smaller hole blade show that changes in experimentally determined values of primary efficiency for multiple coolant row discharge can be well predicted by properly adding the changes in primary efficiency for single coolant row discharge. Apparently, coolant flow from upstream rows of holes does not significantly affect the output of coolant flow from downstream rows of holes.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, June 4, 1976,
505-04.

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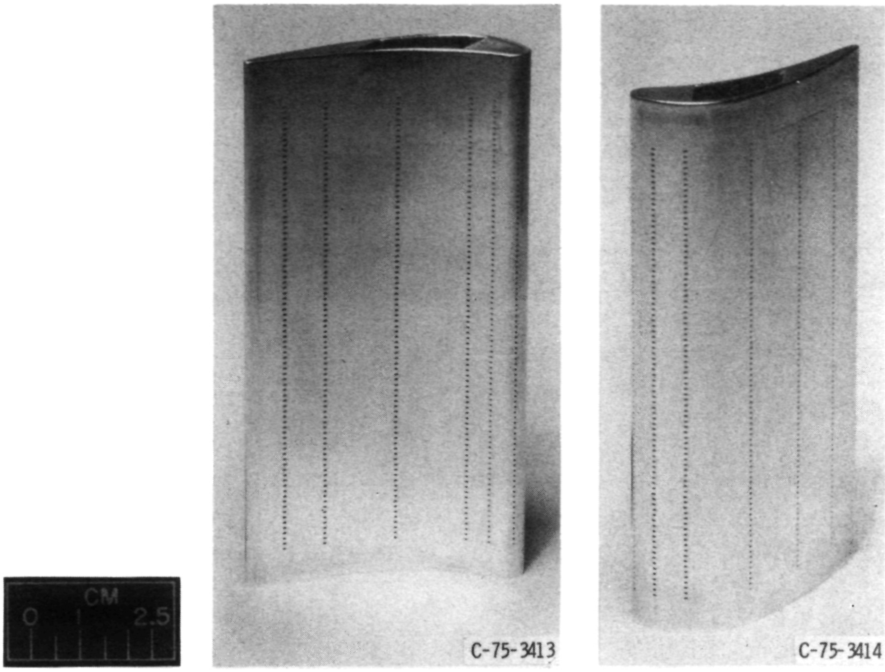
TABLE I. - COOLANT HOLE DATA

Coolant row	Percent of blade surface length, x/L_{pr} or x/L_s	Angle between coolant hole axis and local blade surface tangent in plane parallel to blade end surfaces, β, deg	Length-diameter ratio of coolant hole, L/D
1	3.5	90	4.4
2	12	34	7.4
3	20	33	6.6
4	45	35	
5	70	33	
6	85	34	
7	3.5	90	4.4
8	10.5	36	7.4
9	20	39	9.0
10	40	38	8.0
11	60	38	7.6
12	80	35	7.6

TABLE II. - LISTING OF MULTIROW COOLANT TESTS

Multirow configurations tested	Coolant rows included
1	1, 2
2	1, 2, 3
3	1, 2, 3, 4
4	1, 2, 3, 4, 5
5	1, 2, 3, 4, 5, 6
6	7, 8
7	7, 8, 9
8	7, 8, 9, 10
9	7, 8, 9, 10, 11
10	7, 8, 9, 10, 11, 12
11	1 to 12 (full film)



(a) Pressure-surface view. (b) Suction-surface view.

Figure 1. - Tested stator blade.

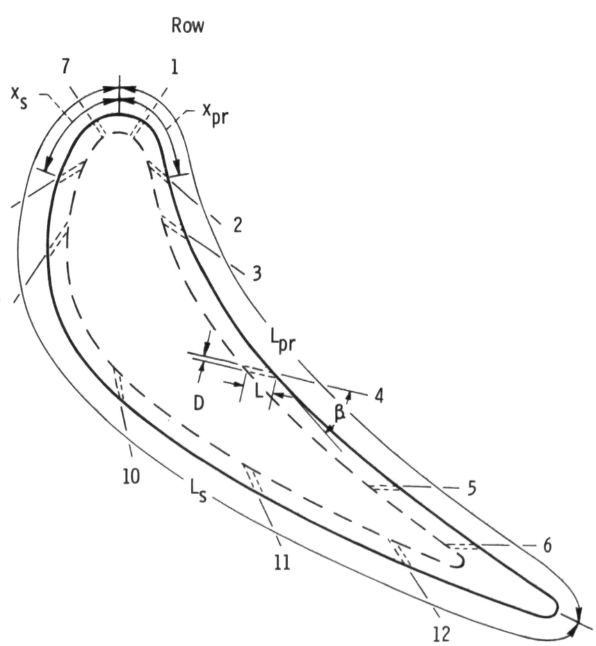


Figure 2. - Cross-sectional sketch of cooled stator blade.

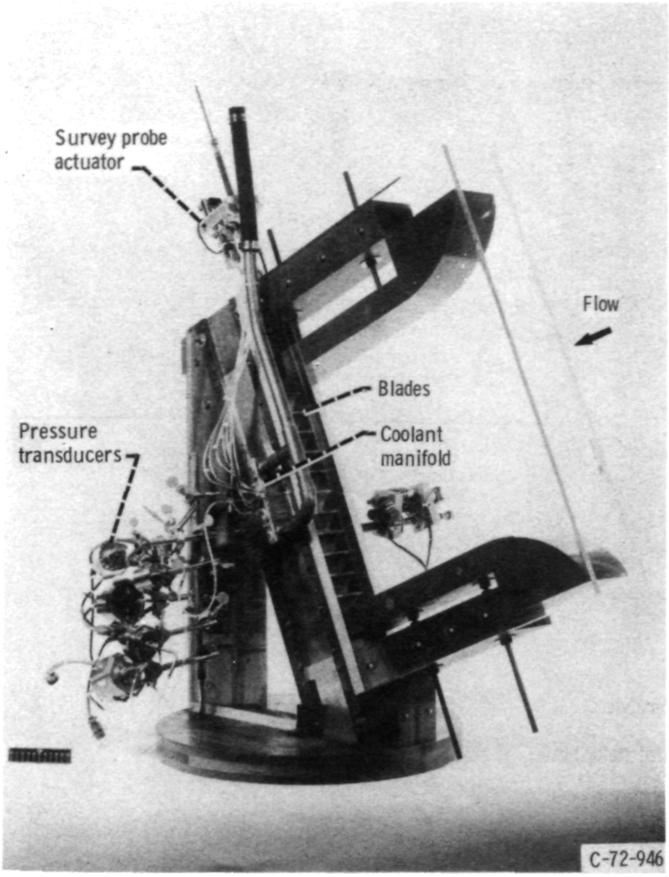


Figure 3. - Stator blade cascade.

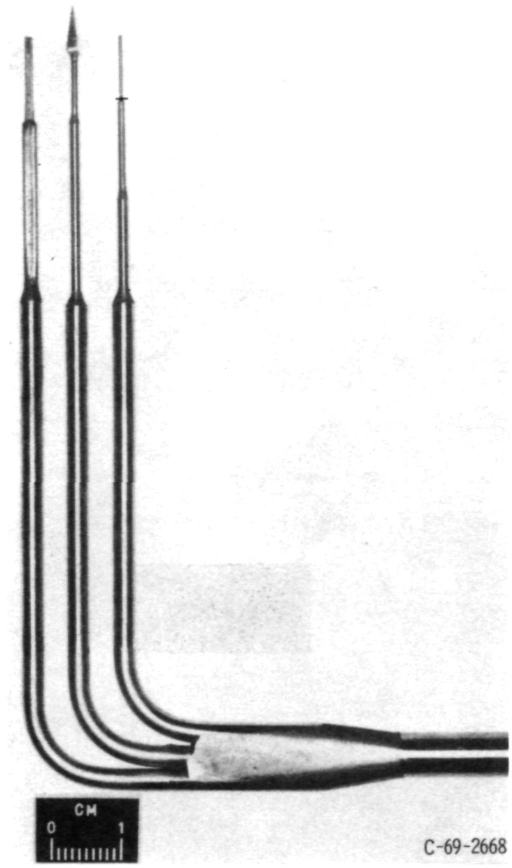


Figure 4. - Survey probe.

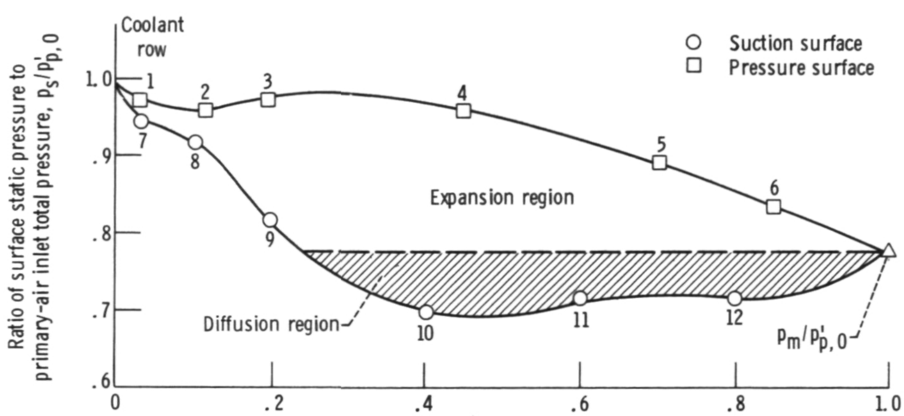


Figure 5. - Comparison of local coolant row static pressures on suction and pressure surfaces of blades for primary-air critical velocity ratio $(V/V_{cr})_{p,id,m}$ of 0.65.

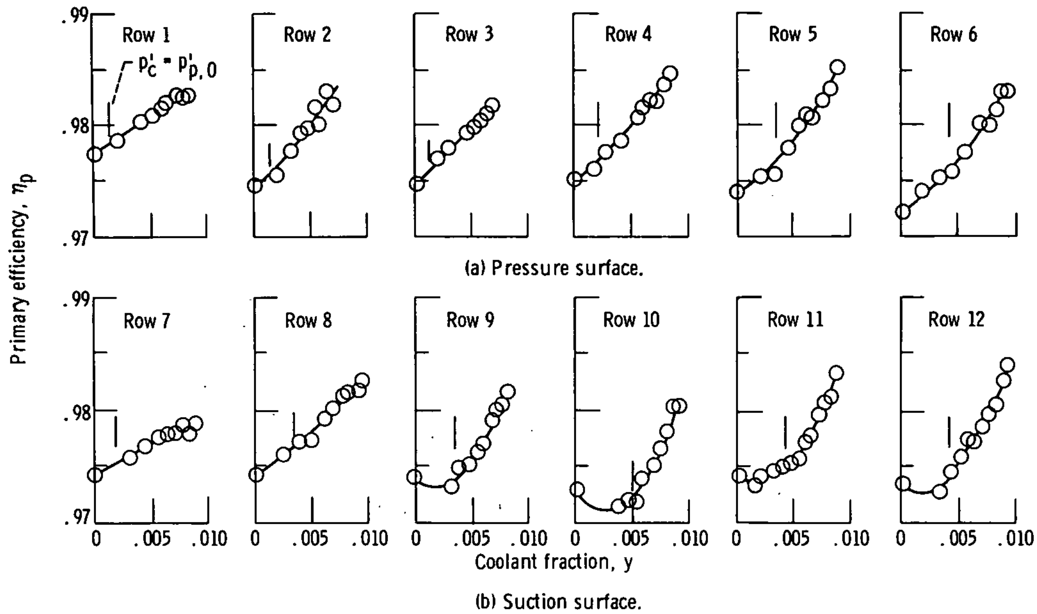


Figure 6. - Variation in efficiency with coolant flow for discharge from single rows.

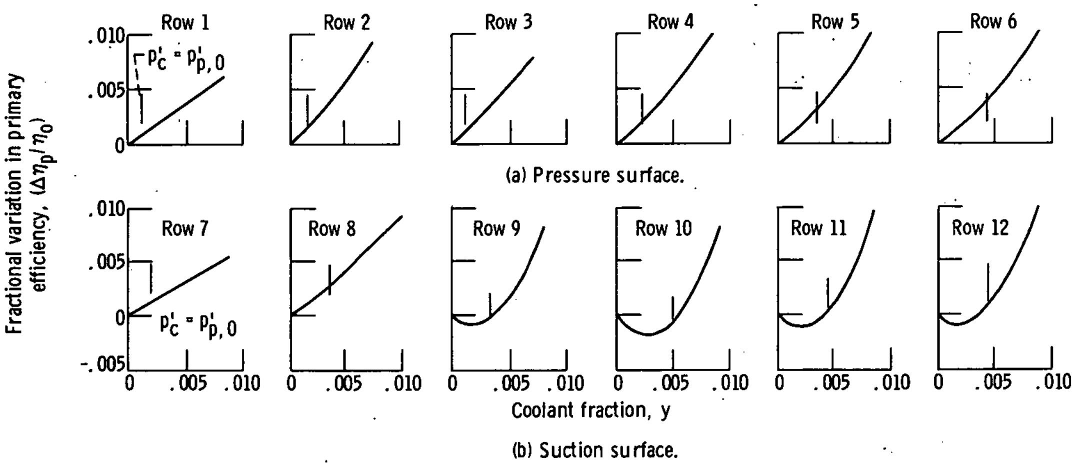
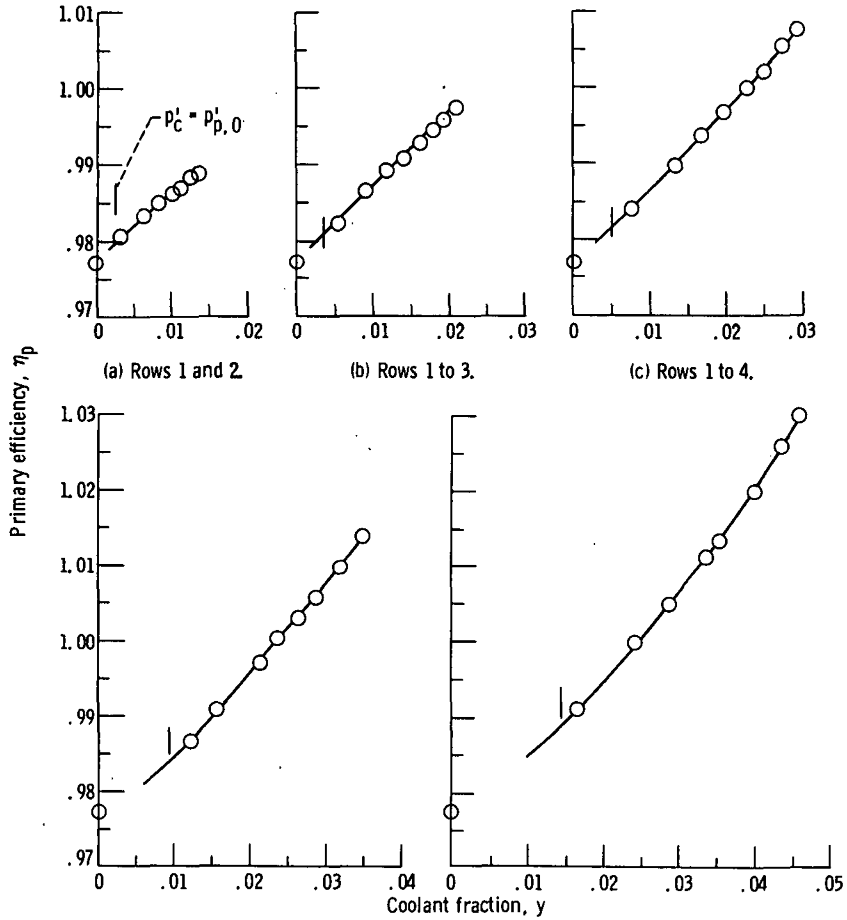


Figure 7. - Fractional variation in efficiency with coolant flow for discharge from single rows.



(a) Rows 1 and 2. (b) Rows 1 to 3. (c) Rows 1 to 4.

(d) Rows 1 to 5. (e) Rows 1 to 6.

Figure 8. - Variation in efficiency with coolant flow for pressure surface discharge from multirows.

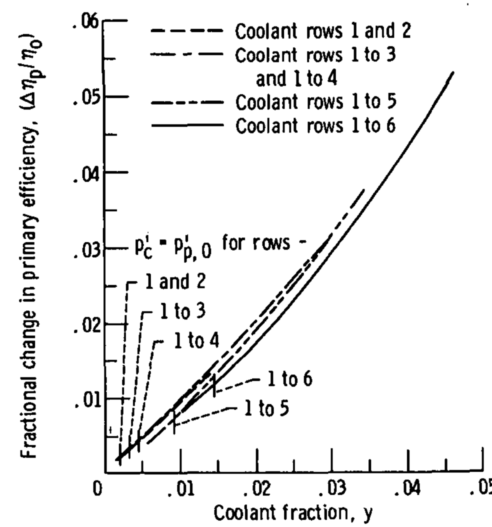


Figure 9. - Comparison of fractional variation in efficiency with coolant flow for discharge from combinations of rows on pressure surface.

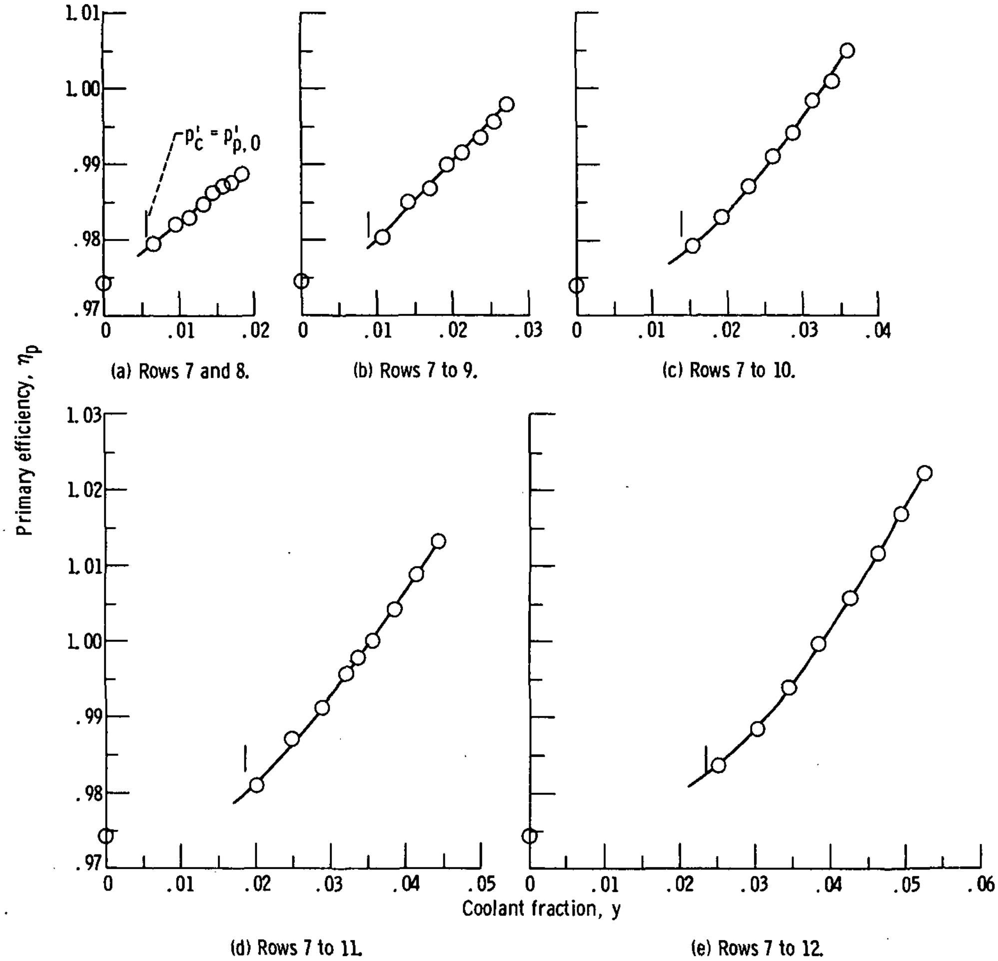


Figure 10. - Variation in efficiency with coolant flow for suction surface discharge from multirows.

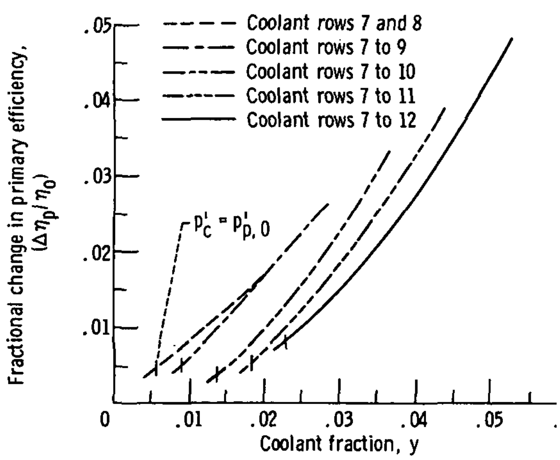


Figure 11. - Comparison of fractional variation in efficiency with coolant flow for discharge from combinations of coolant rows on suction surface.

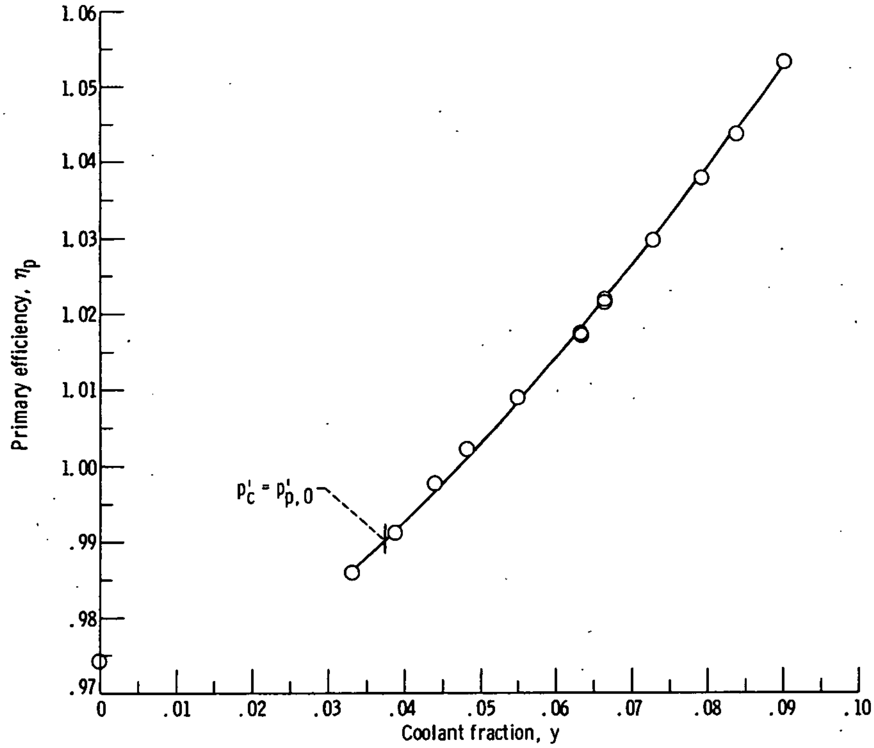


Figure 12. - Variation in efficiency with coolant flow for full film cooling (rows 1 to 12).

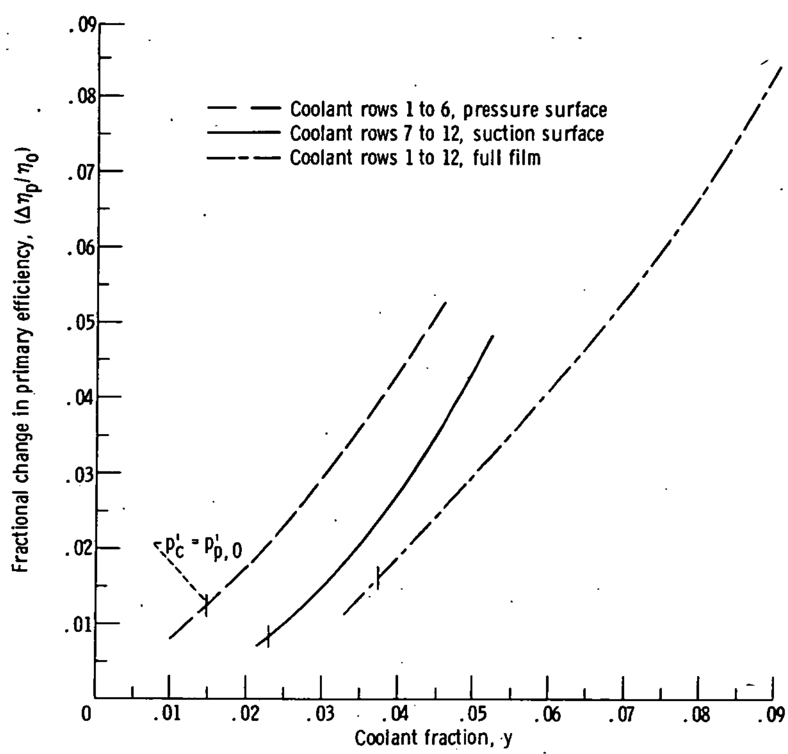


Figure 13. - Comparison of fractional variation in efficiency with coolant flow for coolant discharge from pressure surface, suction surface, and full film cooling.

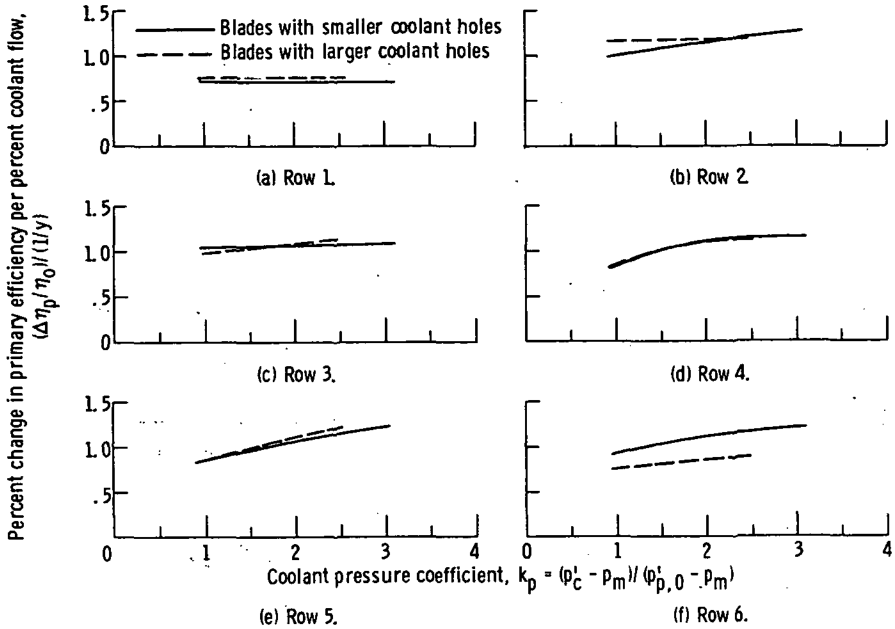


Figure 14. - Effect of coolant hole size on efficiency for single-row discharge from pressure surface.

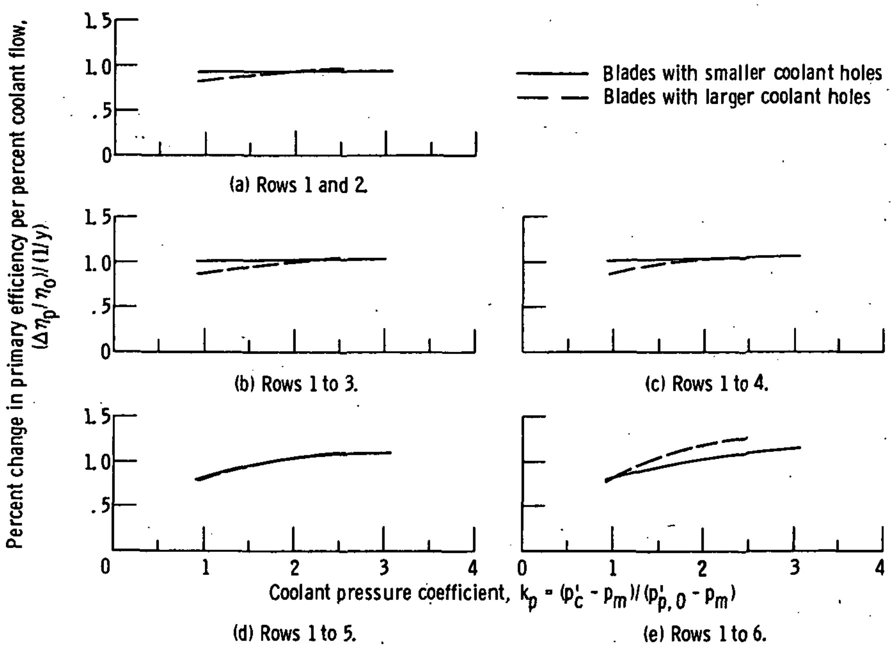


Figure 15. - Effect of coolant hole size on efficiency for multirow discharge from pressure surface.

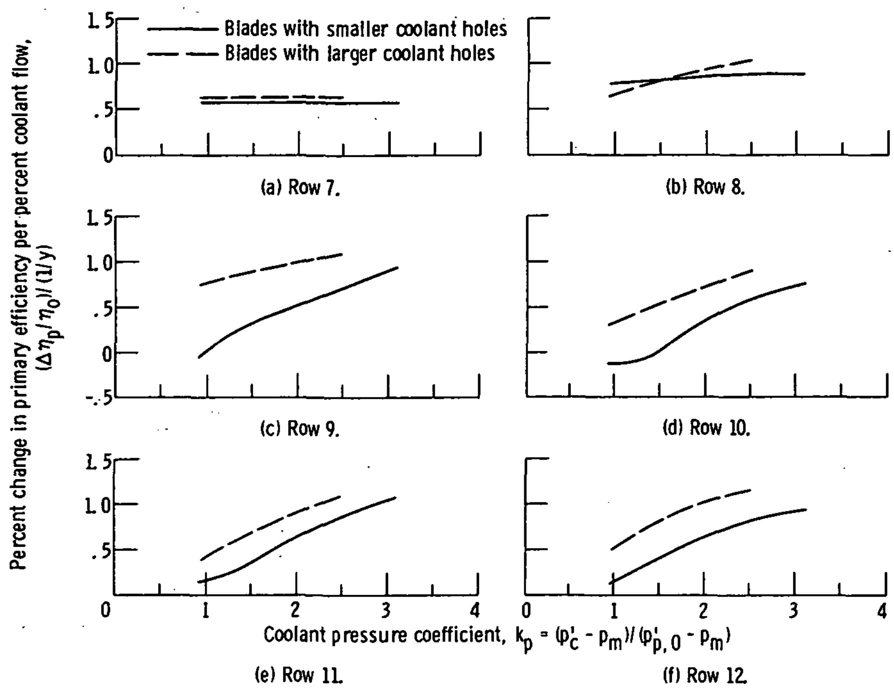


Figure 16. - Effect of coolant hole size on efficiency for single-row discharge from suction surface.

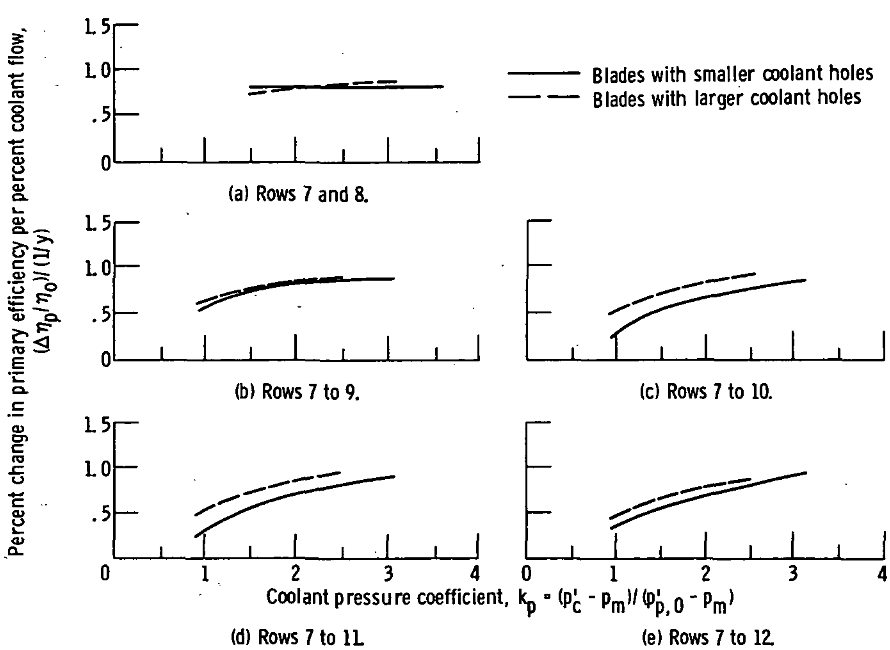


Figure 17. - Effect of coolant hole size on efficiency for multirow discharge from suction surface.

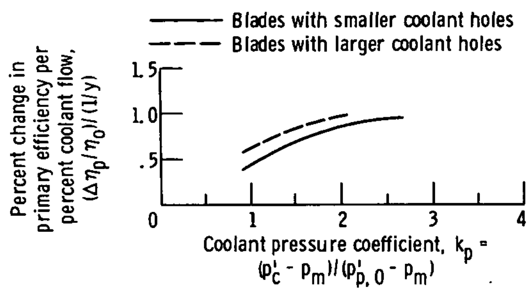


Figure 18. - Effect of coolant hole size on efficiency for full-film cooling (rows 1 to 12).

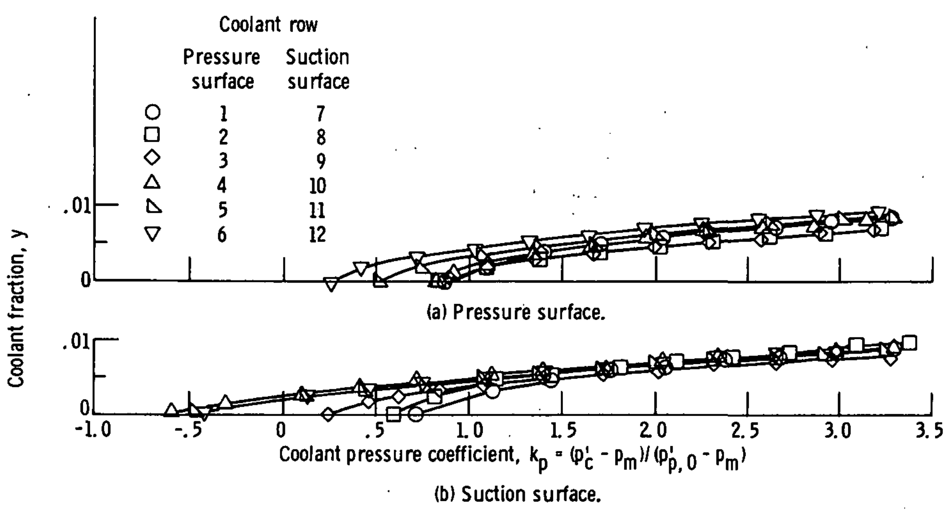


Figure 19. - Variation in coolant fraction with pressure coefficient for single-row discharge.

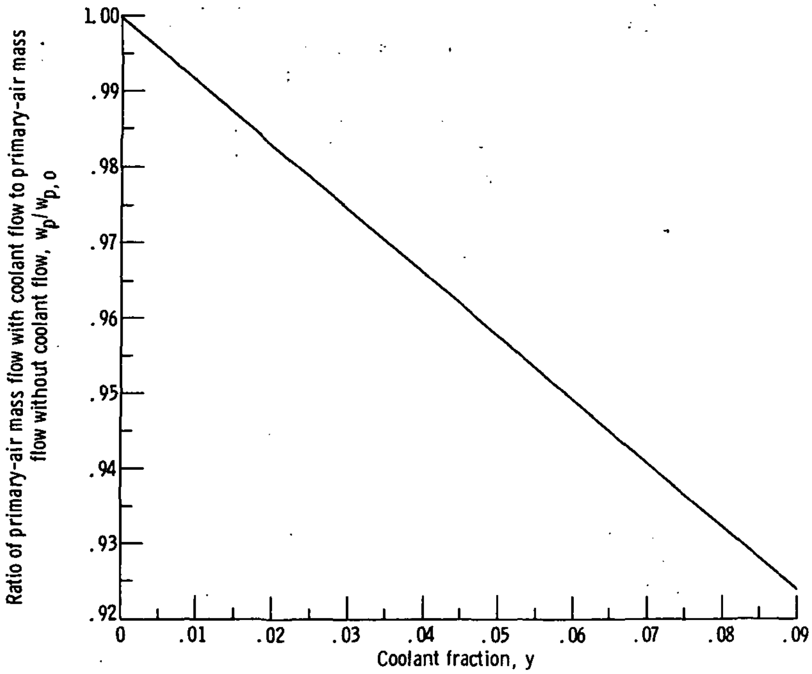


Figure 20. - Effect of coolant flow on primary-air mass flow for single and multiple coolant row discharge. (Data averaged for all test points).

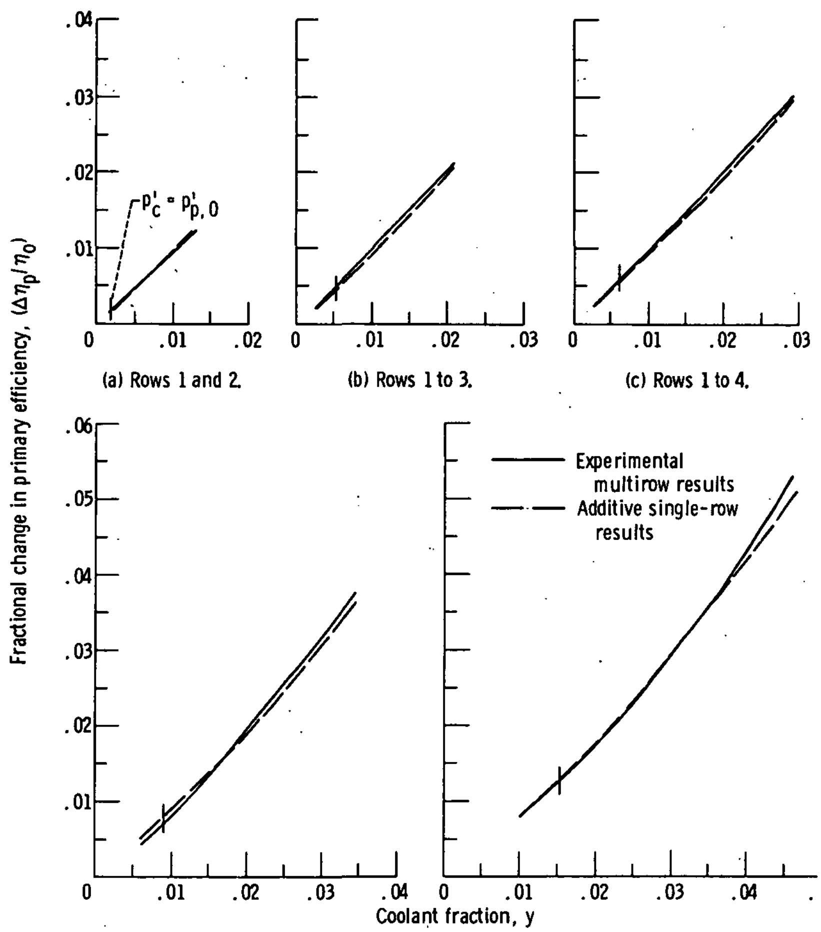


Figure 21. - Comparison of experimental multirow efficiency changes with changes predicted from additive single-row data.

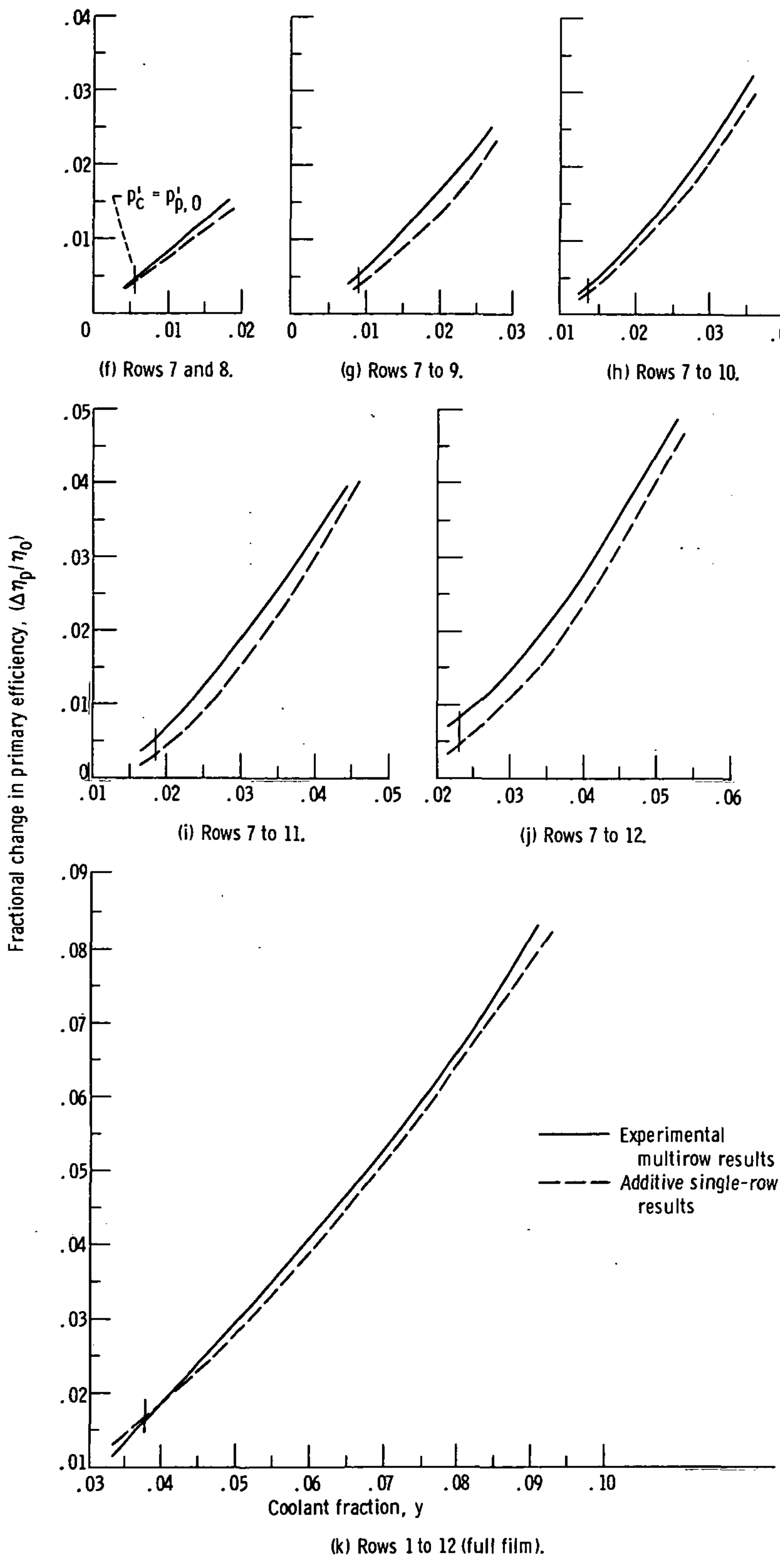


Figure 21. - Concluded.

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