THE INFLUENCE OF TIP CLEARANCE ON STALL LIMITS OF A RECTILINEAR CASCADE OF COMPRESSOR BLADES

GHASSAN R. KHABBAZ

August 1959

GAS TURBINE LABORATORY
MASSACHUSETTS INSTITUTE OF TECHNOLOGY
CAMBRIDGE, MASSACHUSETTS
THE INFLUENCE OF TIP CLEARANCE ON STALL LIMITS OF
A RECTILINEAR CASCADE OF COMPRESSOR BLADES

by

GHASSAN R. KHABBAZ

Under the sponsorship of:

General Electric Company
Allison Division of General Motors Corporation
Westinghouse Electric Corporation

Gas Turbine Laboratory
Report Number 54
August
1959

Massachusetts Institute of Technology
ACKNOWLEDGEMENTS

The author would like to express his gratitude to all those who sponsored, guided and helped in executing the work.

The author is deeply indebted to Professor E. S. Taylor, Director of the Gas Turbine Laboratory, and to Professor Y. Senoo whose many valuable suggestions initiated new approaches. Without their continual guidance, the present work could not have been accomplished.

The author wishes to thank Messrs. Dalton Baugh, Basil Kean and other members of the Gas Turbine Laboratory, and Messrs. Edward Gugger and Paul Wassmouth of the Machine Shop who helped in constructing and assembling different parts of the apparatus.

Lastly but not least special thanks are owed Natalie Appleton who did the typing of this report.
An experimental study of the influence of tip clearance on the stall limits of compressor blades was conducted on a two dimensional rectilinear cascade. By using the mirror and image technique the end wall boundary layer in the gap was dispensed with. The clearance was found to relieve the pressure gradient and to retard stalling. The loading on the blade near the slot was found to be higher than that at a distance further along.
TABLE OF CONTENTS

ACKNOWLEDGEMENTS

ABSTRACT

LIST OF SYMBOLS

I INTRODUCTION

II EXPERIMENTAL PROGRAM

1. Apparatus and Instrumentation

2. Experimental Procedure

3. Presentation of Data

III CONCLUSIONS

IV APPENDICES

A Relation between Circulation around the Airfoil and Pressure at the Leading Edge on the Suction Surface

B Theoretical Model

REFERENCES
LIST OF SYMBOLS

\begin{align*}
\text{p} & : \text{pressure} \\
\text{V} & : \text{velocity} \\
\rho & : \text{density} \\
\xi & : \frac{\rho_{l} - \rho_{0}}{\rho_{0} \nu^{2}} \\
c & : \text{chord length} \\
e & : \text{slot size} \\
\tau_{l} & : \text{local} \\
\infty & : \text{far upstream of cascade}
\end{align*}
INTRODUCTION

Some work in the past decade has been done in an attempt to study the influence of tip clearance on the behaviour of turbomachines as that of Ruden (1) in 1944 who found that a fan efficiency decreases linearly with increase in clearance gap over the entire range of throttling coefficients he used.

The tip clearance is the gap that exists between the blades of axial turbomachines and the casing. The difference between this gap and that of the shroud should be noted and here only the former shall be discussed.

In the past five years a considerable amount of work has been done in the Gas Turbine Laboratory of MIT to study more thoroughly this part of the turbomachine. Ryan and Ohashi (2) in 1955 found that there is an optimum clearance gap for the performance of the fan which they
tested and this gap is not the minimum. A report on the influence of tip clearance on boundary layer flow was published by Dean (3). In this report nearly all the effects of tip clearance were attributed to the behaviour of the end wall boundary layer. All the work carried in this lab dealt only with compressor rather than turbine blading.

The influence of tip clearance is due to two separate criteria:

1) The change of flow direction due to the slot. Here it is assumed that there is no boundary layer on the end wall.

2) The effect of the interaction between the flow through the gap and the end wall boundary layer.

Up until now all the work carried on in this field dealt with studying the results of the combined causes. It is believed that to understand the problem more thoroughly it might be better to break it into its different parts and study each separately. Here an attempt of this sort is made, and the end wall boundary layer was dispensed with. The flow in the cascade near the end wall has been studied by many such as Hawthorne and Soderberg. It was found that the end wall boundary layer causes a secondary flow in the passage from the pressure to the suction side on the end wall and a skewing in the flow direction of the main flow near the end wall.

The influence of tip clearance on the behaviour of turbomachines are manifested in many ways for example a change in efficiency, a variation in loading of the blade for a fixed angle of incidence and stalling of blades at different loadings. The present work will concern itself mainly with one of these aspects namely the effect on stalling limits. Stalling was studied rather than other effects because it offers serious limitations to the compressor operation due to decrease in efficiency.
EXPERIMENTAL PROGRAM

I Apparatus and Instrumentation

The experiments were carried on in a low speed wind tunnel cascade having compressor blading. This enables one to carry on the experiments more quickly and easily than in an actual machine having rotating blades. Moreover three dimensional effects due to the end wall boundary layer can be eliminated.

The available wind tunnel Fig. 1(b) had means of varying the inlet angle very easily by rotating the whole cascade around a center line passing through the leading edge of the end blade. A diffuser is used to adjust the inlet flow parallel to the vertical wall for different cascade angles. A detailed description of this wind tunnel is given in (4). The cascade consisted of nine NACA 65-410 blades with a chord length of 4.875" and 20" span. The stagger angle was 45° and the solidity \( \frac{\text{chord}}{\text{spacing}} \) was 1. The inlet velocity was 93.6 ft/sec giving a Reynolds number of \( 2.4 \times 10^5 \) based on chord length.

As was mentioned in the introduction, a blade with no end wall (casing) boundary layer is to be investigated. The principle of mirror and image was used to accomplish this. A gap was cut in the middle of the blade as shown in Fig. II-1. This gives a blade, its clearance and the image of the whole picture. Thus a fictitious

\[ \text{FLOW} \]

\[ \frac{\text{BLADE}}{e} \]

\[ \frac{\text{GAP}}{f} \]

Fig. II - 1
wall without boundary layer can be imagined at the middle of the gap.

The available wind tunnel discussed in (4) had no end wall boundary layer suction. When an attempt was made to study the stall in the vicinity of the slot, the blade was found to stall first near the end wall and thus the flow converged towards the center of the blade, changing the two dimensional pattern and preventing stalling at the middle of the blade. This is to be expected since the flow in the boundary layer of the two end walls has lower energy than the main stream and thus less ability to overcome an adverse pressure gradient. Moreover the first blade of the cascade which had its suction side coinciding with one of the vertical walls was receiving low energy fluid also and so stalled before the rest of the blades, changing the flow pattern. To overcome these difficulties it was decided to install boundary layer suction.

On each of the end walls the boundary layer was sucked from two places. First from a slot three inches wide and about a chord length upstream from the leading edge of the blades. An ordinary window screen was laid on top of the suction duct and about 3/4 inches from the bottom of the end wall surface (Fig. 2a). Its original purpose was to avoid objects from falling into the duct but later on an important use was made of it as shall be seen. Of course one difficulty is sure to arise, namely, getting uniform suction. This was met by a combination of ways: flaring the suction duct, installing a diffuser in the middle of it and last by trial and error, putting tape over parts of the window screen where suction was more than in the other parts. A uniform velocity equal to 0.9 that of the main stream velocity could be obtained about 0.0625 inches from the wall just behind the slot.
At larger air inlet angles the width of the test section decreases while the suction area inside the test section did not vary. This had no measurable effect on boundary layer thickness. Therefore the suction was not varied with angle of attack during the experiment.

A porous wall was put above and below the blades to suck the boundary layer that is formed along the wall between the upstream suction slot and the exit of the cascade. The suction from the porous wall did not cover all the blades, but due to construction difficulties was placed only at the ends of the middle nine blades. The porous wall Fig. 2b was made of a layer of fine screen, a piece of cloth made of plain weave cotton 78 x 74/in., another layer of fine screen and a coarse screen made of perforated metal 1/8 in. dia. of holes, 3/16 in. spacing giving 40% open area. The coarse screen acted as a support for the porous wall. The porosity of the cloth was determined by the requirement that the suction pressure in the duct be lower than the lowest pressure on the blade so as to have all the flow outward and yet not to draw an unnecessary large quantity of air. The fine screens were used to hold the cloth in place.

On the vertical wall one 2 1/2 in. wide slot was placed upstream of the leading edge of the blade. The size of the duct was small and it was found advisable to flare and to use a diffuser in the middle of the duct in order to insure uniform suction.

Although at this point the flow in the test section was found to be as nearly two dimensional as one could expect, the flow over the end walls had still a lower velocity than the main stream causing the blades to stall at the ends first. To overcome this, a way of absorbing some of the energy in the main stream had to be devised. Bars running across the width of the test section and covering the middle part of the span were
tried with little success. Then screens of different heights and porosity were used instead. Stalling of the middle part of the blades before the ends was promoted by the use of the hardware screen of 1/4 in. mesh. This screen covered the entire cross-section except for 2 1/2 in. from each end wall. It was located at 24 in. from the cascade. The flow in the central part of the cross-section 12 inches height by 3 blade passages width was found to be two dimensional with a maximum deviation 1° in yaw angle measured upstream of the cascade and 0.5° in pitch angle measured 1/8" downstream of the cascade. Fig. 6 shows two surface pressure distribution curves. They represent the pressure distribution at 1/4 in. and 4 11/16 in. from the midspan respectively. They agree fairly well proving that the flow in that section is two dimensional.

Blades

Two MACA 65-410 brass blades were made (Fig. 3) with static pressure taps on them to measure the surface static pressure distribution along the chord at different spans. All of the pressure taps were located on the lower half of the blades. Pressure tubing ran along the lower half of the span inside the blades and the static pressure taps were drilled into them through the surface of the blades. The two blades that had the static pressure taps were mounted beside each other in the cascade in such a way that the surface on which the pressure taps were drilled formed one blade passage. Due to the fact that the flow is two dimensional the loading in one passage is equivalent to that on one blade. Whenever a pressure distribution was to be recorded at a certain row, all the other rows were closed by scotch tape.

The pressure tubes on the blades were connected by rubber flexible tubes to an inclined manometer filled with alcohol giving a ratio of 5:1" of water. The accuracy to which the manometer could be read is 0.03 inches.
Probes

For measuring stagnation pressure, yaw angle, and pitch angle five hole probes (Fig. 4) were used. For static pressures a sphere probe and pitot static tubes were used. While adjusting the boundary layer suction, a kiel probe was used for measuring the stagnation pressure because it is insensitive to stream directions up to $\pm 40^\circ$.

The stream pressures were fed to a D.C. calibrator through a transducer. The calibrator was made of the balancing bridge type manufactured by Dynamic Instrument Company, Cambridge, Mass.

Traversing Rigs

The upstream traversing rig was a simple sliding bar located at a chord length upstream of the cascade. It allowed the probe to be traversed in one plane. The downstream traversing rig was more complicated as it allowed the probe to move in three dimensions. The probe could be traversed in steps of 0.002" behind the blades.
2. Experimental Procedure

The static pressure distribution was measured along the blade surface for each row at different angles of attack in the vicinity of stall. The velocity far upstream from the cascade at different cascade angles was kept constant by bleeding more or less air from the outlet of the tunnel fan. The velocity was determined by means of reading the difference in static pressure between the plenum chamber (stagnation) and a position just upstream of the cascade. The cascade angle was determined by a scale fixed directly to the cascade support. For each slot the whole procedure was repeated. The first slot was cut at a distance of 1/16 inches from the first row of blades. After that the gap was made bigger by cutting more from the upper half which had no pressure taps. The slot was cut in only the blades having static pressure taps and one on each side, Fig. 5. This gave satisfactory results as indicated from the yaw angles measured at 1/8 in. downstream of the cascade.
3. Presentation of Data

Fig. 7 shows a plot of the static pressure distribution on the blade surface vs. the angle of attack for one row of taps. From the above curves it is very difficult to say at which angle the blade is stalled. A study of carbon black traces was made. This is carried on by painting the suction surface of the blade with lamp black dissolved in kerosene and then running the fan of the wind tunnel. The pattern of streamline is shown as traces as the lamp black solution dries up. This allows one to detect if back flow occurred. The carbon black study revealed that the range of the angles of attack in Fig. 7 covered that at which the blades stalled. This was supplemented by a study of threads glued to the suction surface of the blades which also enables one to find if any back flow takes place. Although both the carbon black traces and tufts indicate if a blade is stalled, they fail to detect within one degree the angle at which stall occurs. This uncertainty arises from the definition of stall.

What does one mean by stall? Some back flow can be allowed to occur on the blade suction surface and yet one says the blade is unstalled. That is why the above two mentioned methods namely carbon black and tufts fail to define sharply the angle at which stall occurs. The term stall is usually referred to in a situation where this back flow has reached such a magnitude as to decrease the efficiency of the compressor causing unsteady flow and decreases in pressure rise across the compressor. For experiments in a cascade stall can be defined in a number of ways: when the drag coefficient becomes 1.5 to 2 that of the minimum drag coefficient, or at the peak of the curve of the turning angle vs. the incident angle, $\theta$, or that of circulation vs. the angle of attack.
Let one now plot the static pressure at a point near the leading edge on the suction surface vs. the angle of attack. A cross plot of Fig. 7 is shown in Fig. 8a. Three pressure taps on the suction surface at e/c 2.5, 5 and 7.5% from the leading edge are chosen respectively as representative points. One finds that the peaks of these curves occur in the vicinity of stall. A deeper look reveals that these peaks should correspond to the angle at which the blade stalls. (Appendix A.) This is true because the difference between $p_\infty$ and the static pressure at a point on the suction surface near the leading edge is inversely proportional to the square of circulation. Moreover the angle at which stall occurred as determined by this last method checks very well with NACA data (5). This last method of detecting the angle of stall was used. Three pressure taps are used to get a better correlation. On each of the figures of Fig. 8 is shown the static pressure for the three taps vs. the angle of attack for different span locations for a certain gap size. All the pressures are plotted in the non-dimensional form

$$S = \frac{P_\infty - P_e}{\frac{1}{2} \rho V^2}$$

Fig. 9 shows the angle at which stall occurs for different span locations for each slot.

The surface pressure distributions are plotted for each slot at two different angles of attack, one away from that of stall and the other near it. Each (Fig. 10) contains a pressure distribution at 2 11/16" away from the edge of the slot and two or three others near it. Here too, the pressure is plotted in the non-dimensional form $S$.

The size of the gap is presented as a ratio of the width of the slot divided by the chord length e/c. Since in this investigation the
mirror image method was used, it should be noted that in actual turbo-
machines only half of this ratio is equivalent to the clearance gap.

The angles are plotted as cascade angles which are equal to the
angle of attack plus the stagger angle.

CONCLUSIONS

From Fig. 9 it can be concluded:
1) That the effect of the clearance gap is not harmful as far as stalling
is concerned because near the tip stall occurred at a higher angle of at-
tack than that for the rest of the blade.
2) The influence of the clearance gap is concentrated in a very small re-
gion near the tip. Its magnitude is of the order of 5% of the chord. The
length of this region varies slightly with the size of the gap.
3) As the gap size is increased the region that can stand the highest
angle of attack before stalling moves slightly away from the tip.
4) As the gap size is increased the blade as a whole stalls at slightly
higher angles of attack.

Fig. 8 shows that near the edge of the slot where the gap is
effective stall is sudden at small clearance. This is clear from the
sharp discontinuity in the curves. But stall is gradual with large clear-
ances.

Looking at the surface pressure distribution curves (Fig. 10)
the following can be concluded as the gap size is increased,
1) The point of minimum static pressure moves away from the leading edge,
2) The loading on the blade near the slot is increased, while at a further
distance from the slot it remains unchanged.

3) The shape of the static pressure distribution in the vicinity of the slot changes radically especially on the suction surface.
Relation between Circulation around the Airfoil and Pressure at the Leading Edge on the Suction Surface

The easiest method of understanding the relation between circulation around the airfoil and the static pressure at a point near the leading edge of the airfoil on the suction side is to look at a cylinder moving in a fluid transversely and at the same time having circulation around it.

The stream function representing the flow pattern is a combination of a circular cylinder moving transversely and circulation about a vortex core of strength $\Gamma$.

\[
\psi = \psi_{cylinder} + \psi_{vortex
core} = U \left( r - \frac{a^2}{r} \right) \sin \theta - \frac{\Gamma}{2\pi} \ln r
\]
From Bernoulli

\[ p_L = \text{constant} - \frac{p}{2} v^2 \]

\[ v_L = v_r + v_\theta = \left( \frac{\partial \psi}{\partial \theta} \right) + \left( -\frac{\partial \psi}{\partial r} \right) \]

\[ p_{\text{on surface of cylinder}} = \text{Const.} - \frac{p}{2} \left[ 4 U^2 \sin \theta - \frac{2 T U}{\pi a} \sin \theta + \frac{T^2}{4 \pi a^2} \right] \]

In the vicinity of A:\[ \theta = \pi \quad \therefore \sin \theta = 0 \]

Hence the term containing \( T^2 \) is more dominant than the term containing \( T \). Therefore the static pressure is expected to drop with increase in circulation. From the above consideration it is clear that the static pressure at a point on the upper semi circumference in the vicinity of "A" is inversely proportional to the square of circulation. By conformal mapping the surface on the upper semi circumference in the vicinity of "A" plots as the suction surface near the leading edge of an airfoil. Therefore all the above conclusions could be applied also to the surface near the leading edge on the suction side, namely the static pressure is inversely proportional to the square of circulation. The curves of the study of the leading edge of turbine blades by Dr. S. Kubota show that a relation \( p = p_o \cdot k \alpha^2 \) can be written near the leading edge on the suction surface where: \( p = \) static pressure; \( p_o \) and \( k \) are constants, and \( \alpha \) angle of attack. The data of Dr. S. Kubota verifies the conclusions made above.
If one plots negative static pressures vs. the angle of attack, one finds that the peak of this curve occurs at the same angle of attack as that of the circulation vs. the angle of attack. Therefore the angle at which the peak of the former curve occurs can be called the stall angle.
APPENDIX B

Theoretical Model

Consider a rectilinear cascade in which the flow far upstream is two dimensional. The blades in the cascade act like airfoils with infinite span causing uniform loading and turning angle along the span. The bound vortex can be assumed to disappear in the end walls. If a gap is cut along the chord of the blade no vortex lines can cross it, in case of viscous flow the gap should be of a relatively large size.

The fluid passing through the gap has a smaller pressure ratio than that passing through the rest of the blade, because it does not turn. In case of viscous flow with a small gap some of the bound vortex can be transported along the gap by the means of viscosity. The fluid in such a case turns through an angle that is smaller than in the case where no gap exists.
The trailing vortex sheet from the tip induces a spanwise flow along the surface of the blades, towards the tip on the pressure surface and away on the suction.

The trailing vortex causes a downwash on the blade which reduces the incidence angle. Moreover the gap relieves the pressure on the suction surface and hence the pressure gradient would not be as steep as in the absence of the gap. One would expect that the combined influence of the above mentioned effects would cause the blade to stall at higher angles of attack. This has been observed by experiments. But one also expects that the same effects would tend to reduce the loading at the tip. This is contrary to what is observed in the experiments carried on for this report and those reported in Ref. 3. To explain this behaviour one should approach this problem from a slightly different angle.

The loading on the blade at any spanwise location is equal to the change of momentum of the fluid passing through that section. From
the previous analysis one can conclude that the turning angle in the vicinity of the tip is smaller than that at other parts of the blade. Nothing as yet has been said of the velocities at different parts.

![Diagram](image)

The pressure upstream of the cascade is low \( \rho_L \) and downstream high \( \rho_H \) because the cascade has compressor blades. In the vicinity of the slot one expects the pressure difference between the two sides of the cascade to be less than that far from the slot as explained previously. Let one assume the pressure upstream of the cascade in the vicinity of the slot to be.

Take a control volume as shown in the Figure above by the dotted line. From the momentum equation:

\[
-A(\rho_L + \epsilon) + A \rho_L = -A \epsilon = \text{Net Momentum Flux Out}
\]

This shows that the momentum flux into the control volume is larger than that out. No flow can pass through the upper surface because it is a surface of symmetry. If no flow passes through the lower surface, \( v_1 = v_0 \) from continuity. This is contrary to what one finds from the momentum equation; \( v_1 \) should be greater than \( v_0 \). Thus fluid has to leave the cascade by passing through the blade. Due to the fact that the flow is two
dimensional away from the slot, more fluid passes per unit length along the span through the blade near the slot than at a distance from the slot. This means that the axial velocity should be higher near the slot, and hence the total change in momentum might be higher although the turning angle is smaller.
REFERENCES

1. Ruden, P.,

2. Ryan, F. J. and Ohashi, H.,

3. Dean R.,

4. Montgomery, S. R.,

5. Herrig, L. J., Emery, J. C. and Erwin, J. R.,

6. Erwin, J. R. and Emery, J. C.,

7. Hutton, S. P.,

8. Hunsaker, J. C. and Rightmire, B. G.,

9. Durand, W. F.,
10. Pope, A.,


11. Prandtl, L. and Tietjens, O. G.,


12. Kubota, S.,

FIG. 1-a OVERALL VIEW OF TEST SECTION
FIG. 1-b SECTIONAL VIEW OF TEST SECTION
FIG. 2a SKETCH SHOWING CONSTRUCTION OF SUCTION DUCT UPSTREAM OF CASCADE

FIG. 2b SKETCH SHOWING CONSTRUCTION OF POROUS WALL
FIG. 3 STATIC PRESSURE BLADES
FIG. 5 THE CASCADE WITH THE BLADES HAVING A GAP
Fig. 6 Static Pressure Distribution on Surface of Blade at Two Spanwise Locations
FIG. 7 STATIC PRESSURE DISTRIBUTION ALONG SURFACE OF BLADE FOR DIFFERENT CASCADE ANGLES WITH e/c = 0 AT 2 11/16" FROM SLOT
FIG. 8a) STATIC PRESSURE ON SUCTION SURFACE VS CASCADE ANGLE  \( e/c = 0\% \)
FIG. 8b) STATIC PRESSURE ON SUCTION SURFACE VS CASCADE ANGLE  $e/c = 0.615\%$
FIG. 8c) STATIC PRESSURE ON SUCTION SURFACE VS CASCADE ANGLE  \( e/c = 1.230\% \)
FIG. 8d) STATIC PRESSURE ON SUCTION SURFACE VS CASCADE ANGLE  e/c=1.949 %
FIG. 8e) STATIC PRESSURE ON SUCTION SURFACE VS CASCADE ANGLE \( e/c = 4.041\% \)
FIG. 8f) STATIC PRESSURE ON SUCTION SURFACE VS CASCADE ANGLE  \( e/c = 6.010\% \)
FIG. 8g) STATIC PRESSURE ON SUCTION SURFACE VS CASCADE ANGLE  e/c = 10.174 %
FIG. 8h) STATIC PRESSURE ON SUCTION SURFACE VS CASCADE ANGLE  e/c = 20.308%
FIG. 9 ANGLE OF STALL V.S. DISTANCE FROM SLOT FOR DIFFERENT CLEARANCES
FIG. 10-a STATIC PRESSURE DISTRIBUTION ON SURFACE OF BLADE
Cascade Angle = 59°
\( e/c = 1.949 \%
\)

- \( \circ \) - 1/16" from Slot
- \( \triangle \) - 3/16" from Slot
- \( \bullet \) - 211/16" from Slot

**FIG. 10-b** STATIC PRESSURE DISTRIBUTION ON SURFACE OF BLADE
FIG. 10- c  STATIC PRESSURE DISTRIBUTION ON SURFACE OF BLADE

Cascade Angle = 61.5
$e/c = 1.949\%$

- $1/16''$ from Slot
- $3/16''$ from Slot
- $21/16''$ from Slot

Percent Chord

-1.6 -1.2 -0.8 -0.4 0 0.4 0.8 1.2 1.6

$S$
FIG. 10-d  STATIC PRESSURE DISTRIBUTION ON SURFACE OF BLADE

Cascade Angle = 59°
e/c = 4.041%

- - 1/16" from Slot
\[\Delta\] 3/16" from Slot
\[\Delta\] 7/16" from Slot
- - 2 11/16" from Slot
Cascade Angle = 62°
e/c = 4.041% 

1/16" from Slot
3/16" from Slot
7/16" from Slot
2 11/16" from Slot

FIG. 10-e STATIC PRESSURE DISTRIBUTION ON SURFACE OF BLADE
FIG. 10- f  STATIC PRESSURE DISTRIBUTION ON SURFACE OF BLADE
FIG. 10-g  STATIC PRESSURE DISTRIBUTION ON SURFACE OF BLADE
Cascade Angle = 59°
\( e/c = 10.174\% \)

- 1/16" from Slot
- 3/16" from Slot
- 7/16" from Slot
- 2 11/16" from Slot

FIG. 10-h STATIC PRESSURE DISTRIBUTION ON SURFACE OF BLADE
Cascade Angle = 62
\( \text{e/c} = 10.174\% \)

- 1/16" from Slot
- 3/16" from Slot
- 7/16" from Slot
- 2 11/16" from Slot

FIG. 10-1  STATIC PRESSURE DISTRIBUTION ON SURFACE OF BLADE
FIG. 10-j  STATIC PRESSURE DISTRIBUTION ON SURFACE OF BLADE
FIG. 10-k THE STATIC PRESSURE DISTRIBUTION ALONG THE SURFACE OF BLADE

Cascade Angle = 62
\( e/c = 20.308 \% \)

- 1/16" from Slot
- 3/16" from Slot
- 7/16" from Slot
- 21/16" from Slot

Percent Chord