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Case Wall Pressures in a Multistage Axial Compressor with Tip Clearance Variation

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Large cyclical pressure excursions from stator-averaged values are observed on the suction side of a rotor blade at the case wall in a multistage compressor as the rotor moves relative to the stator. The pressure changes correlate well with stator relative position, occur in the passage away from the blade, and are only slightly modified by tip clearance changes. Close to the entry and exit of the tip gap, however, the pressures remain stable. The pressures near the gap are only slightly affected by stator proximity and primarily vary with tip gap height.

Nomenclature

b = blade height
C_p = pressure coefficient, $(p - p_{\text{ref}})/(0.5\rho U_t^2)$
c = blade chord at tip
e = clearance gap
p = static pressure (+pressure; −suction side)
t_u = blade tangential thickness at tip
U_t = blade tangential or whirl velocity at tip
W_t = relative velocity at blade tip
Φ = passage average flow coefficient, V_{axial}/U_t
Ω = rotor angular velocity

Introduction

RESULTS from a program of flow measurements in a low-speed multistage axial compressor are presented. The flow was measured as blade tip clearance was changed. Comparisons of the pressure patterns in the rotor, when the influence of the stator was averaged across the rotor, were described in a previous paper by Moyle et al.[1] This article examines the flow in the rotor with respect to the location of the upstream stator blades. In the rotor frame, the stator blades move across, or relative to, the rotor passage. The measurements were made to investigate the flow effects of tip clearance changes on axial compressor performance.

Previous studies of rotor–stator blade interaction, while complete in many respects, traditionally have focused on pressure loading at the blade surfaces away from the case walls. Consequently, the influence of the stator's presence on the tip and case wall local flow of the rotor has not been explored in great detail. In addition, the influence of a clearance change on the rotor flow has typically been inferred from changes in the flow at the rotor exit (e.g., Dring[2] and to a lesser extent Lakshminarayana et al.[3]). Consequently, in-passage flow development characteristics near the case wall as a function of clearance have been unknown until recently. Recent studies have obtained much more data in the passage and closer to the tip gap.[4–7] The present study provides additional high-response pressure data in the tip gap, relative to the upstream stator, over a range of flow and tip gap conditions.

The primary data obtained in the study were case wall pressures surveyed at gap-to-blade height ratios e/b of 0.0035 and 0.006 in a casing that was only 0.0002 of blade height out of round. The flow in the rotor was measured for these clearances at peak power input, near design, and at an open throttle condition on a constant speed throttle line.

Apparatus and Measurements

The compressor test section had a tip radius of 457 mm (18 in.) and a hub-to-tip ratio of 0.6. The test facility permits three stages with inlet and exit vanes to be tested. For the experiments described, the test section was fitted with an inlet guide vane row of 30 blades, two 30 rotor/32 stator stages in the second- and third-stage positions, and a 30-blade exit guide vane row as shown in Fig. 1. The first stage position was left vacant to allow circumferential survey access upstream of the rotor. The compressor rotational speed was 1610 rpm with a tip speed of 77.08 m/s (252.9 ft/s). Screens or plates of different flow resistance were placed in a throttle housing two duct diameters upstream of the test section to vary the airflow rate.

Stage Design

The blade whirl distribution at design was of the solid body type with a symmetric velocity diagram. Profiles for the rotor and stator were developed from a circular arc camber line and a modified C-4 thickness distribution. Details of the blading design are provided in Ref. 1. The Reynolds number $(\rho c U_t/\mu)$ was 4.2×10^5 at the tip for the design flow condition and the blade AR was 2.25.

Blade-to-Blade Survey Procedure

High-frequency response sensors and a timing signal from the compressor shaft permitted measurement of the flow relative to the instantaneous position of the rotor blade. The rotor tip could be located to within 0.005 of the rotor pitch and the data system could sample at intervals smaller than 0.001 of pitch. By moving a sensor on the case wall circumferentially or axially relative to the upstream stator, stator relative quantity distributions were determined in the rotor. A contoured plate fixture, which permitted this type of survey, was developed for the compressor case. The plate, positioned flush with the wall above the second rotor, carried a matrix of 6 × 5 holes spanning the axial chord and one stator pitch, respectively. The plate design prevented leakage into the flow region being sur-

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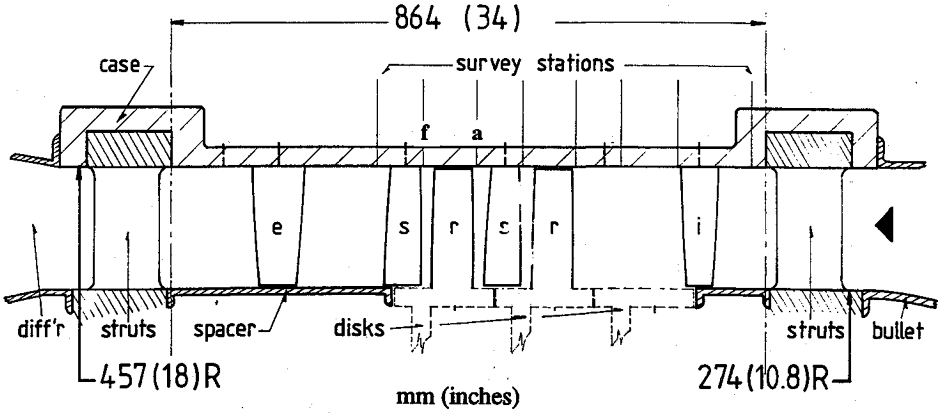


Fig. 1 Compressor test section showing the layout of blading in the two-stage build.

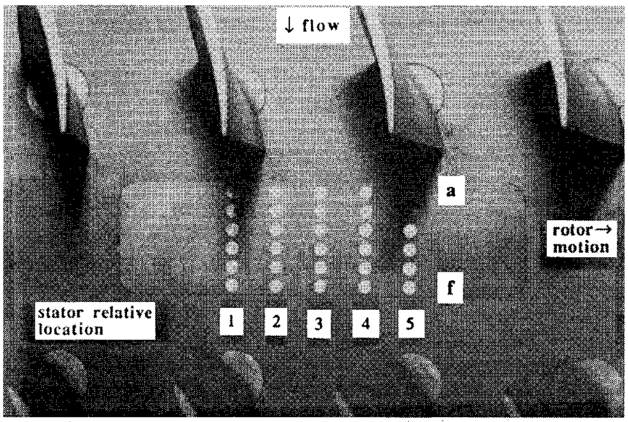


Fig. 2 Survey plate in the test section wall used for wall pressure measurements under the rotor. The relative location of stator blades is shown with the rotor removed.

veyed and its position relative to the stator blades is shown in Fig. 2.

Case Wall Static Pressure Surveys

Static pressure surveys at the case wall, under and between the rotor blades, were made by moving an instrumented plug to hole locations on the plate discussed earlier. The plug was fitted with a centrally located, flush-mounted Kulite® XCS-093-1D high-response pressure transducer. One transducer was used to measure all of the wall pressure data presented. Systematic error in the data was thus minimized by having the same calibration and sensor characteristics for all of the readings. To ensure the sensor's fidelity, measurements of the same flow condition were made with different orientations of the probe, depths of immersion, and with selective covering of the sensor's protective screen. The output signal spectrum was found to be free of unacceptable resonance. The probe resolved to within 0.01 of spacing and had a frequency response of 100 kHz compared to the blade passing frequency of 805 Hz. Such resolution was more than adequate for interpretation of the results presented. The calibration, response, and verification tests are described in detail in Ref. 7, Sec. A.5.

Data Acquisition and Reduction

The rotor blade spacing was surveyed in increments of 1%. Two passages were routinely sampled after confirming the periodicity from samples of three and five passages. A pressure coefficient C_p, defined as the difference in wall pressure from a pressure datum divided by the wheel dynamic pressure at the tip ($0.5\rho U_t^2$), was used to normalize the data. Unless otherwise noted, a contour interval of 4% of C_p was used for plotting. This interval is equivalent to 6% of a C_p based on the tip relative velocity W_t of the velocity diagram at design conditions. The measurement uncertainty in C_p was less than 0.4%. Each of the five circumferential positions on the wall plate was surveyed at the six axial stations and the interpolated data were stored in (60×200) matrices for plotting and data reduction. When surfaces or contour maps are examined to resolve flow features smaller than one-sixth of the axial chord, it is advisable to recall that the picture is based on interpolation of a 6×200 data point set.

Test Conditions

For each clearance condition, $e/b = 0.0035$ and 0.0060, the flow was measured at flow coefficients of 0.60, 0.64, and 0.68. The flow coefficient was derived from test section mass flow, tip speed, compressor face density, and test section area and reflects the following conditions: $\Phi = 0.64$ was the design flow coefficient, $\Phi = 0.60$ was slightly below maximum power input to the flow, and $\Phi = 0.68$ was an open throttle condition.

Flow in the Tip/Wall Corner

A general summary of the average flow structure near the blade tip and case wall corner form is shown in Fig. 3. A surface view (Fig. 3a) of the wall pressure averaged with respect to the stator[1] shows the main features of the flow. The (leftmost) high-pressure ridge corresponds to the position of the pressure side of rotor blade 2. The pressure magnitude was typically uniform from leading to trailing edge on the blade's pressure side and a broad, gradually rising plateau of high pressure advanced in front of the blade. The main features of the flow are shown in contour form in schematic (Fig. 3b). A pressure depression or basin (A) sat roughly 15% of chord aft of the leading edge and covered about 20% of the spacing from the blade suction side. The central region of this depression invariably included the lowest pressure in the whole passage distribution. Two pressure gullies were also persistent features of the flow. Minimum pressure conditions at all axial stations corresponded to the gully marked (B) on the surface. The gully marked (C) ran diagonally across the passage from the blade suction-side leading edge toward the pressure side trailing edge. This diagonal (C) gully was a local minimum in the otherwise monotonic pressure rise toward the pressure side of the passage. The stator average pressure patterns and key features were found to be very repeatable from blades 1-2 to 2-3 to 3-4. This was consistent with the expected periodicity of the flow.

Stator Relative Behavior of the Wall Pressure

The characteristic flow features of Fig. 3 were developed from circumferential averages of all survey positions under the rotor at six axial stations. A single row of holes in the axial direction could also provide a flow picture. Such a survey of

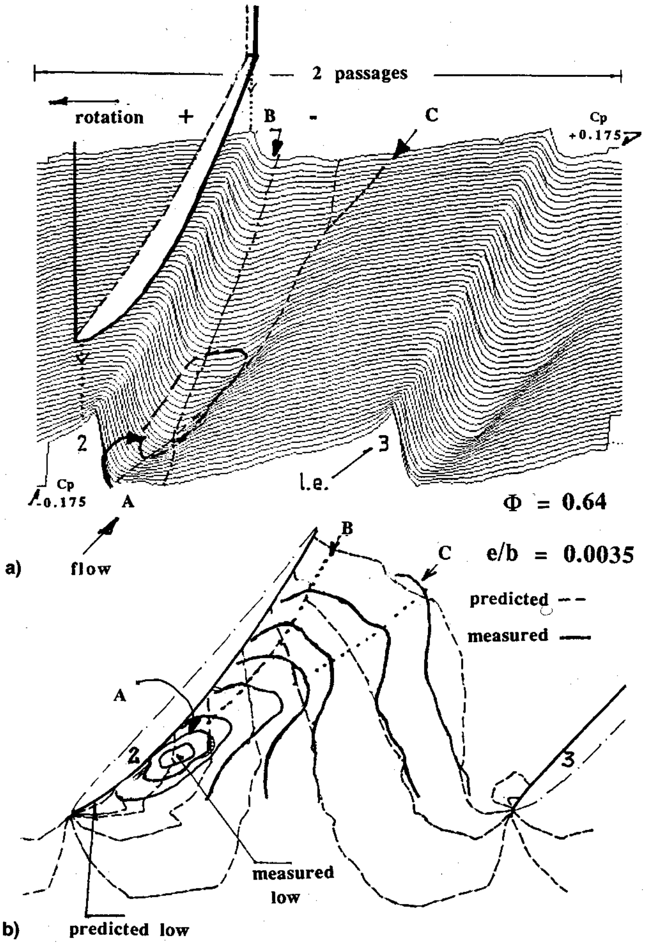


Fig. 3 a) Surface plot of the stator average pressure distribution on the annulus wall and b) schematic of main features of the flow at the tip on the suction side of the passage.[1]

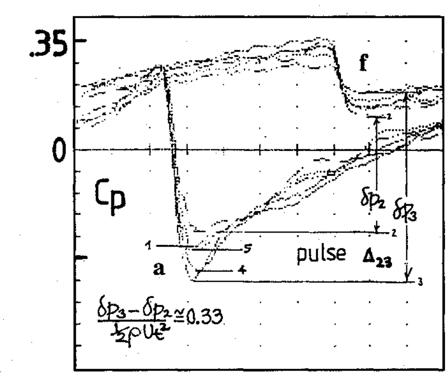


Fig. 4 Pressure surveys across the rotor passage for stator relative circumferential positions 1–5 at axial stations a and f.

a single axial row provided a complete picture of the flow in the rotor with respect to or relative to the stator, as the rotor moved across the pitch of the stator. Comparison of stator relative pressure distributions from several axial rows revealed the behavior of the flow in the rotor's tip region with respect to the stator pressure field and wake.

Samples could also be taken from a series of circumferential holes at one axial station. A comparison of stator relative circumferential positions 1–5 at axial stations a and f only, are shown in Fig. 4 to familiarize the reader with the data. Comparing pressures across the rotor passage, it can be seen that the pressure distribution and levels varied substantially (up to 25–30% of the pressure change across the rotor blade) in the region near the rotor's suction side. The variation depended on the stator relative position of the sensor (and thus the rotor). The trace for the sensor at circumferential position 1 repeated itself at position 5. This confirmed the periodicity with respect to the stator spacing. In contrast, the pressure trace and level on the pressure side of the blade was roughly constant, regardless of sensor position. The data also showed that pressure differentials across the tip, and some distance out into the passage, continued to fluctuate by as much as 30% of the average along the whole rotor chord.

Radiative Pattern of Wake Cutting Disturbances

By correlating the position of the rotor and the stator, it was found that the pressure across the tip region undergoes a very rapid change from the minimum differential to the maximum differential as the rotor passed out of the stator shadow (i.e., the approximate position of the wake of the stator). This rapid pressure change near the blade was strongest at axial station a and coincided with the rotor passing between stator relative circumferential positions 2 and 3. The abrupt pressure change also radiated from the leading edge into the passage in the form of pressure ridges and peaks. The wave-like nature of the radiative pattern in the passage is shown in Fig. 5. These waves (as much as 10% higher than average) were superimposed on the average pressure distribution in the passage and created a pattern of pressure nodes that can also be seen in the overlaid pressure traces of Fig. 4 (see Fig. 3a for orientation).

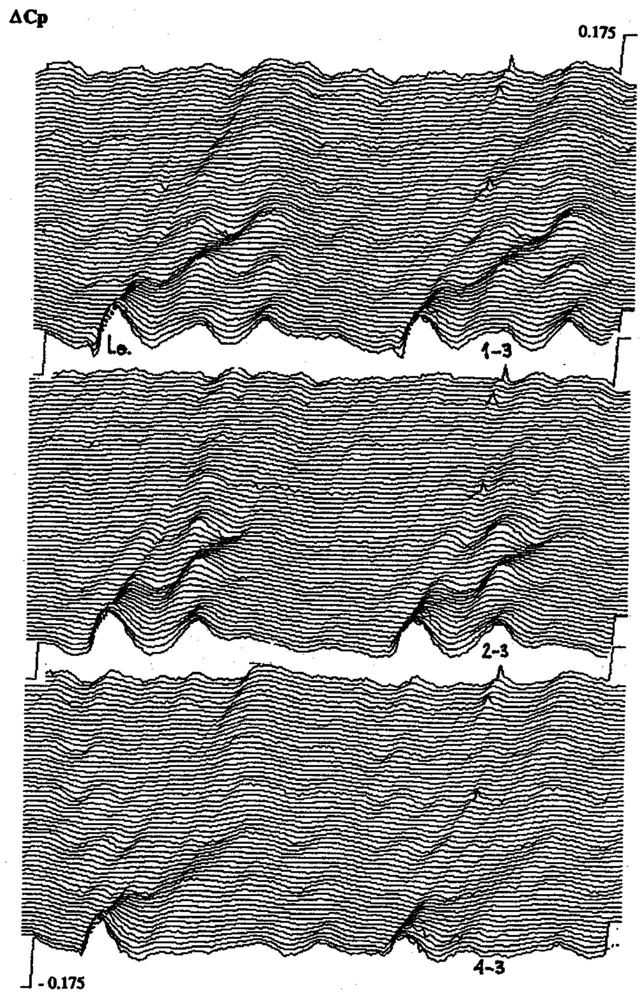


Fig. 5 Pressure waves radiating from the rotor leading edge (l.e.). The surface was created by subtracting the pressure distribution of position 3 from other positions to show the stator relative effect.

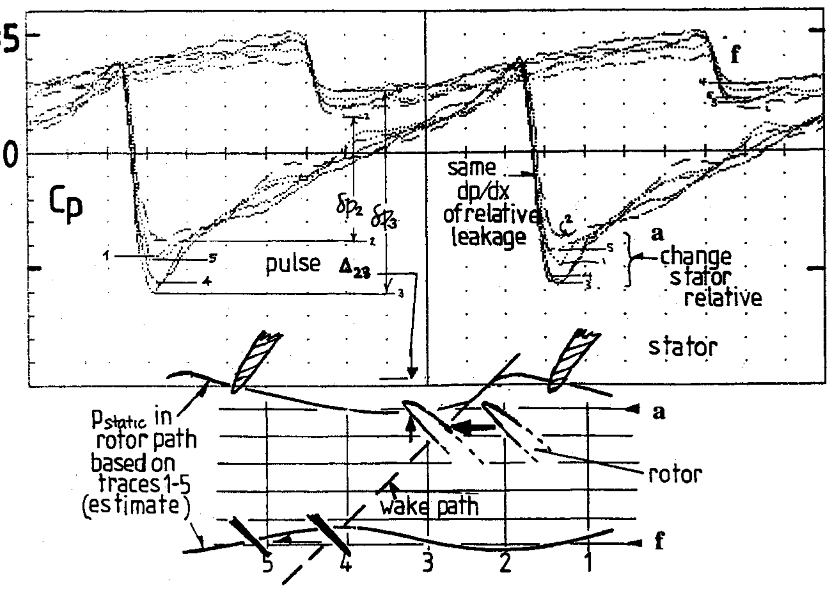
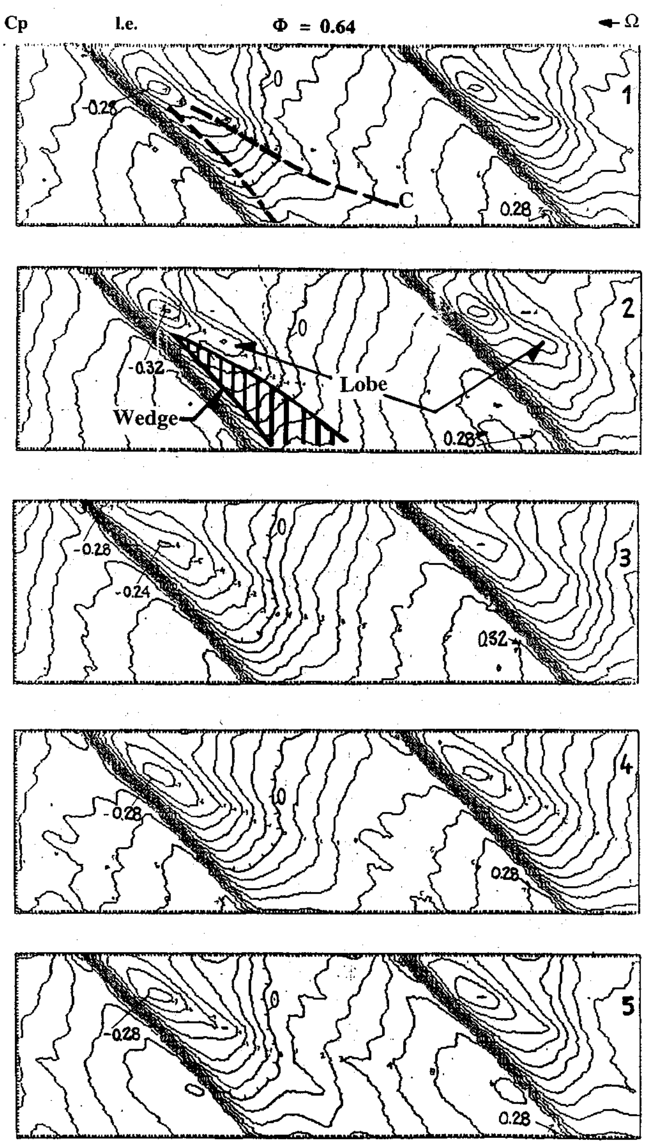


Fig. 6 Pressures at positions 1–5 for stations a and f showing the path of the stator wake deduced from the data.



interval of Cp contour = 0.04

Fig. 7 Pressure contours in the rotor for five stator relative positions (e/b = 0.0035).

Wake Convection Through the Rotor

Figure 6 shows the pressure distributions at axial stations a and f for stator relative positions 1–5. It can be seen that the pressure continued to fluctuate along the blade chord in the region near the suction side. The form of the distributions was well correlated, from leading to trailing edge, with the path of a convected stator shadow (or wake) through the rotor. The pressure difference across the rotor blade was at a maximum approximately 20% from the blade's leading edge and this axial station showed large fluctuations on the suction side. The fluctuations then diminished in amplitude as the pressure differential between the pressure and suction side decreased along the blade chord toward the trailing edge.

Higher Pressure Movement Along Suction Side

Figure 7 presents contour plots of pressure in the passage from surveys of five stator relative positions. The contours of higher pressure move cyclically beside the blade with stator relative position. They move in a wedge-like zone between the suction side or low pressure (gully B) and the pressure contour of the cross passage filament (gully C).

Note that the outboard side of the pressure contour of gully C changes only slightly throughout the cycle of one stator pitch. The gullies are shown in Fig. 3. When the high-pressure wedge moves farthest upstream, the contour of gully C becomes a long lobe of low pressure extending diagonally out into the passage (see position 2). The change to a lobe shape did not extend the low pressure of gully C from its average position, however. Oscilloscope traces at the axial station cutting the lobe showed distinct saddles (low–high–low) in the pressure as the high-pressure wedge crossed the probe location. The saddle was most pronounced when the high-pressure wedge was farthest forward and provided the strongest suction side diffusion.

Pressures at Entry and Exit of the Tip Gap

In contrast to the passage flow, the pressure distributions and levels at the entry and exit of the tip gap showed only slight changes with stator relative position. The marked difference between tip local flow and passage flow behavior is described in Table 1. The tip local wall pressure distributions and levels are compared in more detail in the following section, which addresses the wall pressure behavior with changes in clearance.

Table 1 Flow feature dependence on stator relative position

Strong correlation with stator relative position	Weak correlation with stator relative position
Pressures in the passage on the suction side of the blade outboard of the minimum pressure line (gully B)	Pressure levels and distribution at entry and exit of the tip gap (inside gully B)
High-pressure wedge excursion fore and aft along suction side of blade; radiative pattern corresponding to wake cutting by the rotor	Direction and length of the low-pressure region (gully C) under the cross passage filament

Change of the Tip Flow Structure with Clearance Variation

The effect of tip clearance changes on the near-wall flow of the rotor was determined by comparing passage surveys, at a particular stator relative position, after each clearance change. It should be noted that results for all flow coefficients were similar to those found for the design flow condition discussed next.

Stator Relative Behavior of the Wall Pressure

Figure 8 shows contours of the difference in wall pressure (i.e., contours of ΔP) between two clearance cases at five stator relative positions for the design flow condition. The variation in the difference in pressure can be seen to be relatively small over most of the passage. Large changes only occur under the blade tip itself. The general form of the contours of the difference in pressure were also consistent, with only minor variation, from one to another of the stator relative positions of the sensor. It was inferred from these data that a change in clearance has little effect on the stator relative flow, even though appreciable changes in wall pressure occur from one stator relative position to another.

Wake Convection Through the Rotor

It can also be seen from Fig. 8 that the slight changes that are observed in passage pressure levels lie diagonally along the path of gully C. There is no evidence of any pressure feature correlating with the approximate position of the stator wake in the contour plots. This indicated the stator wake, which acts as a localized pressure difference when convected across the blade, was not affecting the tip flow differently from one clearance to another.

High-Pressure Excursions Along Suction Side

Evidence of the high-pressure wedge motion fore and aft along the blade suction side was also negligible. This suggested that the wedge motion was not coupled to the tip clearance size and was primarily dependent on the stator position.

Pressures at Entry and Exit of the Tip Gap

The changes under the blade, evident in Fig. 8, are compared using blade pressure distributions and tangential loading for three clearances at two stator relative positions (2 and 3) in Fig. 9. Positions 2 and 3 correspond to the blade leading edge cutting the stator wake (or shadow) region. In these plots the data are overlaid and the differences shaded. Data for a test case with a rubbing seal on the pressure side of one blade, at the design flow coefficient condition, are used to show the zero clearance case. As the clearance enlarged, the shape of the pressure distribution and the pressure levels change significantly with clearance. However, the shape changes little with stator relative position of the sensor, and thus rotor, except in the first 10% of axial chord. A change in loading is noted over this zone of the axial chord, but, as a proportion of the total loading, the change is seen to be small. Changes in the tip pressure distribution and loading with stator relative position

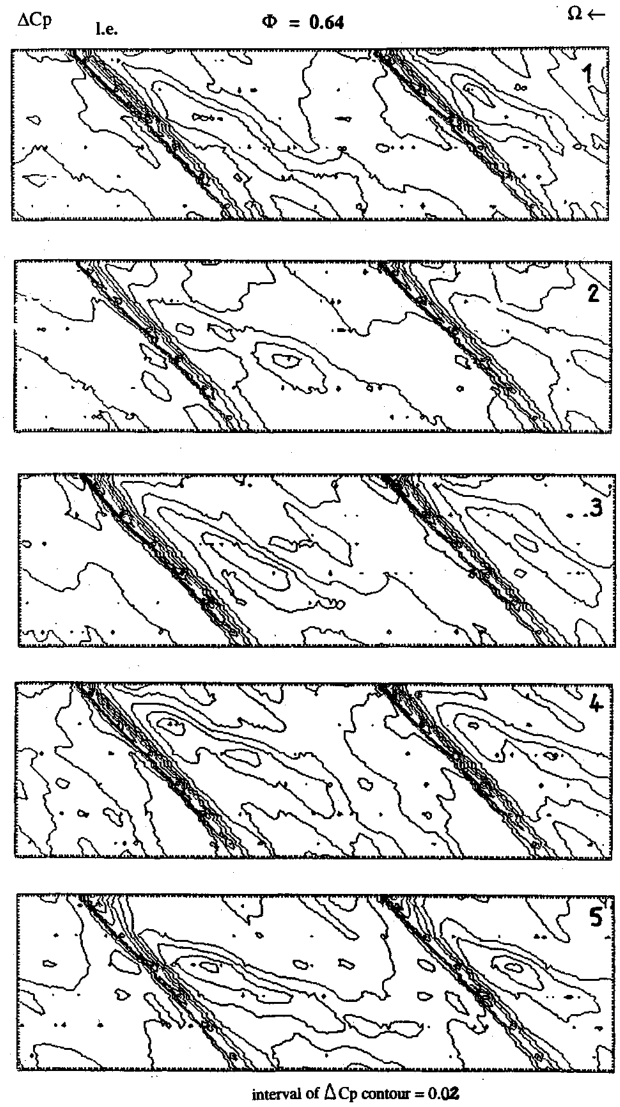


Fig. 8 Contours of the difference in pressure between clearances of e/b = 0.006 and 0.0035 for five stator relative positions. Differentials are small in the passage compared to the zone under the blade.

because of a clearance change were similar at other flow coefficients. This suggested the stage's mean line incidence, which varied from -1.25 to $+3.5$ deg, had minimal influence on the tip flow near the blade surfaces. This was to be expected considering the insensitivity of the tip local conditions to stator proximity.

Correlation of the Blade Loading with Clearance

A decoupling of the pressure changes near the blade tip (due to clearance variation) from the pressure changes in the passage (due to stator relative motion) was apparent in all of the data. The dependence of the tip local pressures on gap height could be further quantified as shown in Fig. 10. Along 80% of the axial chord, from the leading edge, it was found that the tip loading of the blade divided by the tangential thickness t_u of the blade at the tip was a constant. The constant ratio did not hold when the blade became thin, in the last 20% of chord near the trailing edge. The magnitude of the ratio varied nonlinearly (as shown in Fig. 10) with gap height, but correlated with a shear layer thickness[7] in the tip gap.

Discussion

The pressure patterns and results of the present study provide complementary information to the results of Lakshmi-

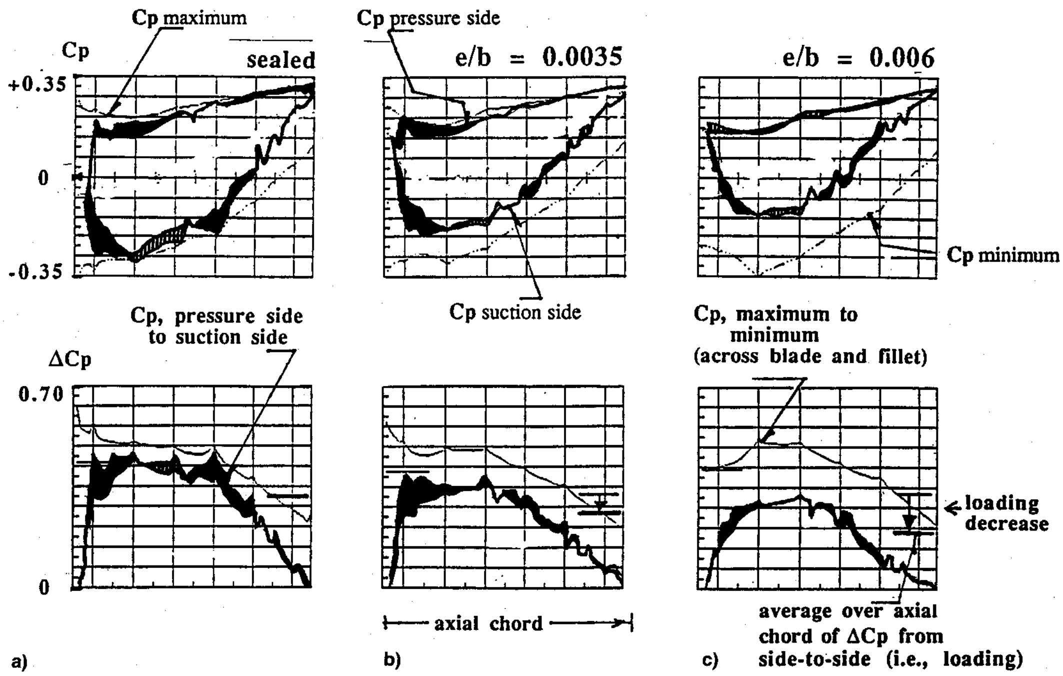


Fig. 9 Difference in rotor tip pressure distribution and loading between stator relative locations 2 and 3 (shown shaded) for a) sealed tip on blade 2, b) e/b = 0.0035 clearance, and c) e/b = 0.006 clearance for the design flow.

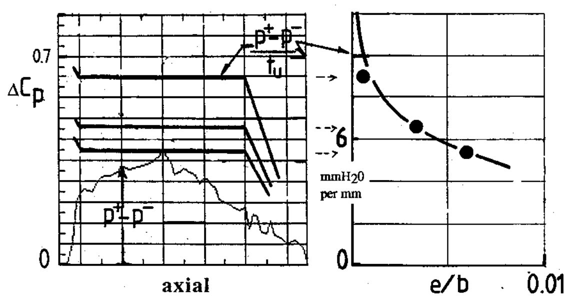


Fig. 10 Correlation of rotor tip loading divided by blade tangential thickness (the ratio is a constant for most of the axial chord) with three tip clearance conditions.

narayana et al.[5] and Stauter.[6] In those studies the rotor was preceded by a guide vane or stator row, respectively. In both studies the wall shear to centrifugal force ratio[8] was moderate to large and, respectively, lower and higher than in the present compressor. Both of these studies surveyed three velocity components close to the wall just downstream of the rotor. Stauter[6] also measured three velocity components in the rotor passage. His survey was made with a laser Doppler velocimetry beam at the midgap position of the stator (which is assumed to be equivalent to stator relative position 3 of the present study). Stauter had a nominal clearance level of 1.25% of chord. Lakshminarayana et al.[5] measured stator average data because they used on-rotor traversing pressure probes. They presented loss information at two stations, close and far, behind the rotor. Their nominal clearance was 2.27% of chord. Neither study varied clearance. The clearance range of the present study was 0 (rubbing seal), 0.8, and 1.35% of chord. The 1.35% case was comparable to Stauter's[5] clearance.

Both studies[5,6] identified a loss formation in the general vicinity of, or corresponding to, an extension of gully C of the present study. In both cases their loss formations were close to the wall; approximately 5–10% and 4% of blade height, respectively. This permits plausible comparison of their velocity data with the present wall pressure data. Stauter was able to show that a weak vortex existed in the flow. The vortex wandered in the direction of the passage and had largely disappeared at the rotor exit. Lakshminarayana et al.[5] concluded their observed low stagnation pressure region, at the rotor exit or beyond it, contained no evidence of vortical motion. Stauter found the vortex reappeared beyond the rotor exit. In the present study, the distinct lobe of low-pressure necking out into the passage at stator relative position 2 of Fig. 7 implied high local velocity near the wall. Stator relative position 2 is in the general vicinity of Stauter's stator relative survey location. If it is assumed that the lobe of Fig. 7 corresponds to Stauter's weak vortex, then it can be seen by inspection from Fig. 7 that the strength of this formation would be negligible on a stator average basis. This would be consistent with Lakshminarayana et al.'s[5] finding of no vorticity in the stator average data. In general, there was reasonable consistency between the studies when interpreted collectively. It should be noted that the behavior of gully C, weak as it was in the present study, was found to be largely decoupled from the pressure levels, in and near the tip gap. This was so when the clearance was changed and for all stator relative positions. Thus, it would be reasonable to project that increased vortical flow would be unlikely in the other two compressors at different (small) clearances.

Discussion by the Authors

In this article we have examined the influence of an upstream stator on the local flow near the rotor tip and case wall. This subject has not been explored in the literature with detailed data over a clearance range. There are, of course, comprehensive rotor–stator interaction studies by several authors that discuss the performance of the blading across the span as interblade spacing changes. It was not, however, our intention that this article be perceived as a comprehensive study of rotor–stator interactions near case walls. Rather, our purpose was a description of what flow changes occur as the blade tip passes through the wakes of the upstream blading, at different clearances.

Our measurements show that a low-velocity zone, covering about 20% of the pitch, exits from the upstream stator row on its suction side (Fig. 6). This zone appears to be the only wake of significance in our data. Streak lines on the casing suggested this wake was capturing contributions from the upstream blade row and inlet guide vane (IGV), and could be considered an aggregate wake. Given this dominant wake (whatever its origin), the question of interest in our study was what did cutting this wake do to the conditions very close to the tip gap and could something be learned about tip clearance flows from this event. This article therefore establishes a baseline for the flow in Figs. 4–7 and then shows what changes occur in Figs. 8 and 9 as the tip clearance gap changes. The striking observation, shown in Fig. 9, is that although the blade surface and case wall pressure conditions are markedly different between blade positions 2 and 3 (see Figs. 6 and 7), the conditions at the gap entry and exit edges are only slightly different and become less so as the tip gap enlarges.

The decoupling of the pressures at the gap edges from changing flow conditions out in the passage undermines the notion that the tip leakage flow is driven through the gap by the pressure distribution of the nominally inviscid flow around the section profile of the blade tip. Figure 10 also shows the pressure difference across the tip gap, from blade edge-to-edge, was proportional (for most of the axial chord of the tip) to the tangential thickness of the section profile. The constant of proportionality also became much larger as the tip gap became smaller. This result indicates the tangential thickness of the blade's tip profile is determining the pressure difference across the tip gap. The dependence of the gap pressure on this thickness appears to be unrelated to the pressure distribution established by the flow in the passage above the tip gap and the passage flow pattern in general.

Our results can be compared, in terms of measurement sensitivity, but not by stage geometry, to those of Inoue and Kuroumaru.[4] Their geometry tested a rotor without an upstream stator or guide vanes. Contour patterns of their case wall pressure distinctly show the path of a well-developed tip vortex. Also, their lowest pressure at the wall lies inside the footprint of the blade tip for axial stations up to 90% of chord. This persisted until the gap became quite large. Our results, on the other hand, show little evidence of a vortex pattern in the wall pressure contours and the lowest pressures at the wall, for each axial station, lie about one blade thickness away from the suction side at even the smallest clearance gap. The lack of evidence of a tip vortex in our compressor is discussed in the context of compressors with upstream stators or IGVs. Our results are found to be consistent with observations, using different measurement methods by Stauter[6] and Lakshminarayana et al.[5]; their compressors were similar to ours. This finding was also consistent with projections from trends in tip vortex strength discussed in Moyle,[8] where many different compressors were compared.

Summary

The major observations are summarized as follows:

1) The stator relative position of the rotor strongly affected pressures on its suction side near the case wall. The pressure changes were most pronounced in the passage away from the blade edges. The changes correlated well with stator wake transport through the rotor. A clear rotor–stator pressure interaction pattern was superimposed on the average wall pressure distribution of the passage when the rotor cut the stator wake.

2) A cyclic advance of higher pressure on the suction side of the rotor toward the leading edge correlated well with stator relative position of the rotor. The higher pressure filled a wedge-like zone on the contour plots between the cross passage filament's path and the low-pressure path just outboard of the rotor's suction side. This meant diffusion levels at the wall in the passage changed independently of the pressures local to the tip.

3) Pressure levels at the inlet and outlet of the tip gap were only slightly affected by the stator relative position of the rotor and were primarily determined by the tip gap height. Flow in and very near the tip gap was decoupled from the significant pressure changes, due to stator position, in the passage flow of the rotor. For the range of clearances tested, changes in tip clearance gap also had minimal effect on the stator relative flow out in the passage of the rotor. This finding shows only a weak coupling between the tip leakage flow and the passage inviscid flow. The results suggest that flow changes in the two regions are independent, but not mutually exclusive.

Acknowledgments

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