Effects of Compressor Hub Treatment on Stator Stall and Pressure Rise

P Cheng,* M E Prell,†E M Greitzer,‡ and C S Tan§
Massachusetts Institute of Technology, Cambridge, Massachusetts

An investigation has been conducted of the changes in stator stall margin and performance which occur due to a slotted hub treatment rotating beneath an axial compressor stator row Two different stator rows (one with low stagger and one with high stagger) were tested, both with solid wall and hub treatment The experiments with the high stagger blading showed, for the first time, that hub treatment was very effective in delaying the onset of stator stall and increasing the peak stator static pressure rise For this blading measurements of the stator exit flowfield appear to indicate that the inception of stall was associated with a wall stall With the treatment, the large blockage in the hub region was decreased greatly and, near the hub, the total pressure at the stator exit was found to be higher than that at stator inlet These results support the hypothesis that endwall treatment is ef fective when the type of stall that occurs is wall stall

Nomenclature

C	= absolute velocity
C_x	= axial velocity
D factor	= diffusion factor, $1-(W_2/W_1)+(\Delta W_\theta/2\sigma W_1)$
HT	= hub treatment; notation used in figures
P	= static pressure
ΔP	= static pressure rise across blade or vane row
P_t	= total pressure
ΔP_t	= total pressure difference across blade or vane row
r	= radial coordinate
Δr	= streamline shift
RSH	= radially shifted rotor exit profile; notation used in figures
RTE	= rotor exit station (see Fig 1)
STE	= stator exit station (see Fig 1)
SW	= solid wall (no treatment); notation used in figures
U	= mean blade speed
W	= relative velocity
x	= axial coordinate
α	= absolute air angle measured from axial direction
θ	= tangential coordinate
ρ	= density
σ	= solidity, chord/pitch (gap)
ϕ	= flow coefficient C_x/U
$\bar{\omega}$	= total pressure change coefficient (across blade or vane row) $=\Delta P_t/\frac{1}{2}\rho W_1^2$

Subscripts

ex	= measured at compressor discharge
in	= measured at compressor inlet
m	= midspan
ref	= reference
1	= blade or vane row inlet
2	= blade or vane row exit

Received July 24 1983; revision received Jan 24 1984

*Graduate Research Assistant Department of Aeronautics and Astronautics; presently employed at Detroit Diesel Allison Division General Motors Indianapolis Ind

†Graduate Research Assistant Department of Aeronautics and Astronautics; presently employed with Aircraft Engine Group General Electric Company Lynn Mass

‡Associate Professor, Department of Aeronautics and Astronautics Associate Fellow AIAA

§Research Associate Department of Aeronautics and Astronautics

Introduction

FOR some years, it has been known that the application of grooves or slots over the rotor tips in the casing of an axial flow compresor can have a large effect on compressor stall margin and performance Experimental observations[1 2] have also shown that casing treatment can substantially reduce the endwall boundary layer blockage in the rotor tip region It has been hypothesized that this blockage reduction results from strong flow injection out of the treatment, so that the low momentum flow in the endwall region is energized due to injection of high momentum fluid

It also has been found that one can often (although not always) make a simple but useful distinction between two types of compressor stall One of these termed "blade stall " is a roughly two dimensional type of stall where a significant portion of the blade span has a large wake due to substantial thickening or separation of the suction surface boundary layer The other, termed "wall stall " is associated with the endwall boundary layer separation From the results of Refs 1 and 2 it appears that casing treatment will be effective only when the type of stall is wall stall

A third feature, which has emerged not only from Refs 1, 2 but from flow visualization reported in Ref 3 is that relative motion between endwall and blade is an important element for the success of the grooves in reducing the blockage In view of this, it is natural to ask whether a rotating "hub treatment" below a row of cantilevered stator blades would also be effective in improving the stall margin If this is so, the geometry would be more amenable to in vestigation of the details of flow in the blade passages, as well as provide a different perspective from which to view the fluid dynamic phenomena Hub treatments have in fact, been tested previously[4 5] However there does not appear to be any work reported in which the use of this type of geometry has shown an increase in stall margin, i e a shift in the onset point of rotating stall The present work examines this topic, specifically the effect of hub treatment on the stall margin and performance of a "hub critical" stator

The preceding comments concerning the applications of a grooved casing treatment imply that, for the demonstration of a change in stall point (between operation with a *solid wall* and operation with *hub treatment*) the mode of stall that is exhibited by the blade row should be wall stall rather than blade stall Consequently two different stator rows were tested 'blade stall'' and ' wall stall'' configurations As will be seen it was found that for the former the application of hub treatment had little effect on stator performance For the latter however the hub treatment improved the compressor performance delaying the inception of rotating stall and markedly increasing the peak stator static pressure rise

Experiment Design

There are several fundamental criteria implied in the design of this experiment First it must be ensured that the stator hub endwall is stall limiting The design used is aimed at having the rotor (and inlet guide vane) act only as a "flow generator" for the stator Thus in choosing the blade setting angles, one must insure that the rotor has a low loading relative to the stator The requirement of high aerodynamic loading at the stator hub can be satisfied by using a rotor blade with low twist, which creates a low total pressure rise across the rotor hub relative to the tip The flow distribution resulting from this produces the desired stator hub loading

Another critical consideration concerns the question of blade and wall stall As discussed in Ref 2, a measure of the tendency toward blade stall is an airfoil loading parameter, such as for example, the diffusion factor Wall stall on the other hand is more appropriately characterized by a pressure rise parameter such as static pressure rise divided by inlet dynamic pressure [6] Whatever the precise parameters used, however the central concept in design of a wall stall configuration involves raising the wall loading relative to the peak attainable value (as well as to the blade loading), with the converse being true for a blade stall configuration

When the initial experiments were carried out in the present facility a severe restriction was that the stator discharge was to ambient pressure Hence, as described in Ref 7 we were forced to use stator blading of rather low stagger For this type of blading the diffusion factors tend to be comparatively high relative to the pressure rise parameter i e the tendency is toward blade stall (although this is countered somewhat by the effects of the reduced inlet boundary layer skew[6]) In accordance with this (as described in some detail in Ref 7) the type of stall encountered in this first set of experiments did appear to be blade stall rather than pure wall stall

The second set of experiments was carried out with an exhaust fan and it was thus possible to run with an increased stator stagger This had the effect of raising the pressure rise parameter (compared to the peak attainable value as defined in Ref 6) relative to the diffusion factor i e to promote wall stall (Note that this restaggering was done for the rotor as well, with a new set of inlet guide vanes being made for this configuration so that again the rotor was well away from stall and acted only as a flow generator) In addition, based on the information in Ref 6, to further increase the tendency to wall stall the stator hub clearance was doubled to 4% of the blade chord (or 4 5% of the staggered gap based on exit angle) In sum, this high stagger configuration was biased much more strongly toward wall stall, compared to the low stagger situation

Some details of the geometry of the two stator configurations are summarized in Table 1 In order to distinguish between the two they will be referred to below as the low stagger and high stagger stators respectively The blade section have constant chord and the total twist is 5 deg

Experimental Facility

The experiment was conducted on a single stage research compressor driven by a variable speed d c motor This is a 0 59 m diam single stage compressor with a hub to tip ratio of 0 75 It was modified for casing treatment as well as hub treatment and the former was used throughout all the tests since this tended to increase the hub loading A cross sectional

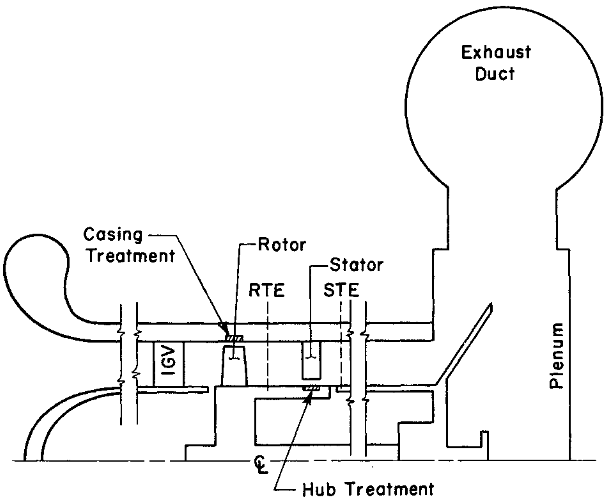

Fig. 1 Schematic of compressor cross section

schematic of the compressor showing the blading and treatment locations is given in Fig 1

Most of the measurements were taken at a blade speed of 75 m/s (at the mean radius) for the low stagger stator, and at 70 m/s for the high stagger stator The Reynolds number (at the stall point), based on blade chord, at the stator midspan was nominally $1\ 5\times10^5$ for the first set of experiments and $1\ 0\times10^5$ for the second Thus, runs also were made at speeds ranging from 55 to 95 m/s to check Reynolds number dependence; this was found to have only a slight influence on the observed changes due to the hub treatment, as discussed below

The pressure instrumentation consisted of 20 total pressure Kiel probes and 24 hub and casing static pressure taps Acquisition of the pressure data was done by a 48 channel scanivalve pressure scanner operated by a digital microcomputer A self nulling radial traverse mechanism was used to determine radial profiles of total pressure and flow angle and a hot wire anemometer was used to enable accurate determination of the stall point Further details of the facility are given in Refs 7 and 8

The data to be presented are nondimensionalized in a standard fashion: velocities are nondimensionalized by U, the mean blade speed and pressure differences are nondimensionalized by either $\frac{1}{2}\rho U^2$ or by the relevant stator inlet dynamic pressure Other quantities used for nondimensionalization or for reference levels are defined as they are introduced In the figures, SW is used to denote the solid wall and HT is used for hub treatment

Treatment Design

The hub treatment consists of axial slots skewed at a 60 deg angle to the radial direction These slots have been found to provide large increases in stall margin when used as a rotor casing treatment Although they also give some penalty in efficiency, their effectiveness in shifting the stall point suggested their use in the present set of experiments, which are aimed at developing an understanding of basic mechanisms rather than optimizing the design The slots rotate under the middle 90% of the stator for the high stagger stator case and under the middle 70% for the low stagger The slot spacing is such that the slotted area is twice that of the solid (land) area The slot aspect ratio (axial length/tangential width) is 2 0 and the radial depth is 30% of the axial length The actual dimensions and treatment geometry are shown in Fig 2 The hub treatment was constructed by machining a wide circumferential groove in the cylinder and glueing premachined plexiglass plates into the groove

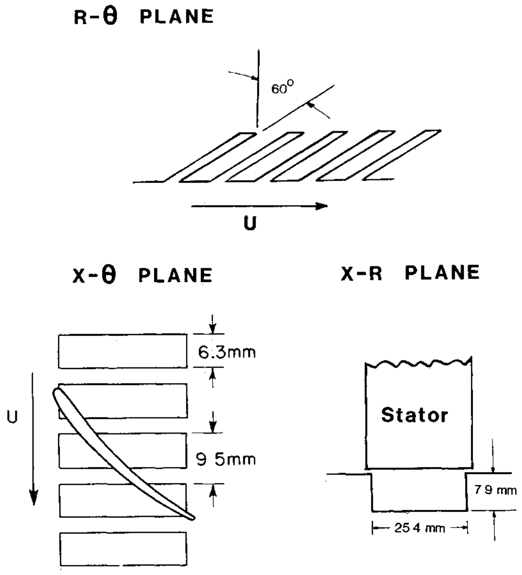

Fig 2 Hub treatment geometry (high stagger stator)

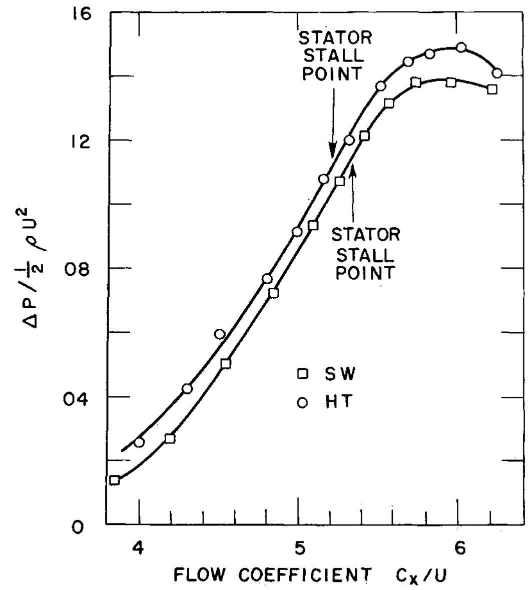

Fig 3 Stator static pressure rise characteristic (low stagger stators)

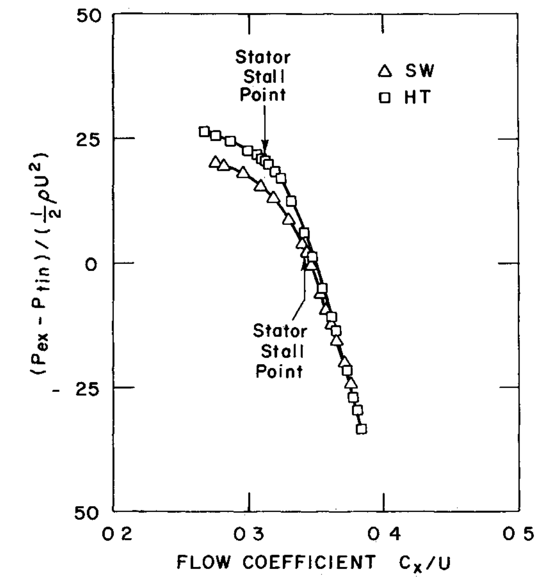

Fig 4 Exit static minus inlet total pressure rise stage characteristic (high stagger stators) with solid wall (SW) and hub treatment (HT)

Experimental Results

In this section we first show the effects of hub treatment on stall margin and compressor speedline characteristics, compare the untreated and treated traverse data and then discuss the results

Low Stagger Stator

For the low stagger blading, the application of hub treatment to the compressor resulted in only a small reduction of the stalling flow coefficient (approximately 2%) and an increase in the stator pressure rise characteristic of ap proximately 5% at the peak The stator pressure rise characteristic from this test as determined from casing wall static taps just ahead of and just behind the stator, is shown in Fig 3. The vertical axis is nondimensionalized wall static pressure rise across the stator and the horizontal axis is the axial velocity parameter (C_x/U) based on mean inlet axial velocity The stall points, i e the points of inception of rotating stall in the stator are also indicated (For reference the rotor stall point was far below these at $C_x/U \simeq 0\ 35$) Rotating stall onset clearly was detectable on an oscilloscope which displayed the response of a hot wire anemometer at stator exit

Although there is an increased static pressure rise across the stator due to the hub treatment at all flow coefficients tested the increase is much less than the large increases that have been experienced with casing treatment One additional aspect described in Ref 7, is that there was a decrease of stator deviation angle and blockage with treatment compared to the untreated case and presumably this improvement is the reason for the increase in treated stator static pressure rise However analysis of the stator wakes showed that a thick wake was present from hub to midspan both with and without treatment This data, as well as other supporting evidence given in Ref 7 implied that the inception of stall was associated with a blade stall type of separation

High Stagger Stator

For the high stagger stator blading the application of hub treatment resulted in a much more substantial change The flow coefficient at stall onset was decreased by 10% and the peak stator static pressure rise was increased by over 50% Figure 4 shows a comparison of the exit static minus inlet total pressure rise characteristic $[(P_{ex} - P_{t_{in}}/(\rho U^2/2)$ vs $(C_x/U)]$ for the complete stage with and without the hub treatment The stator stall points, also indicated here are at $C_x/U = 0\ 344$ and 0 312 for the solid wall and hub treatment, respectively, with an uncertainty in the relative measurements of approximately 1% Note that, although the C_x/U values may seem somewhat low, it should be emphasized that this is due to the high rotor stagger and the stagger and flow angles associated with the stator are representative of those en countered in modern compressors Due to the large pressure rise in the rotor the improvement in stator performance is not really brought out in Fig 4 However, the figure does show that the overall stage characteristic is negative and the rotor is operating far from stall, as should be the case in this ex periment

A comparison of the stator static pressure rise characteristic is shown in Fig 5, which shows $\Delta P_{stator}/(\rho U^2/2)$ plotted vs C_x/U for the solid wall (SW) and the hub treatment (HT) It is evident that the inception of stall has been delayed and the static pressure rise has been increased markedly due to the hub treatment

These stator speed lines were checked several times with and without treatment It was found that there was some slight deterioration in performance over long test times (due

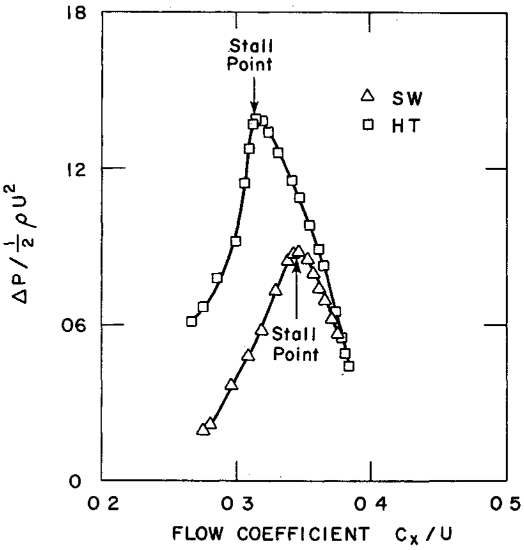

Fig 5 Stator static pressure rise characteristic (high stagger stators) with solid wall (SW) and hub treatment (HT)

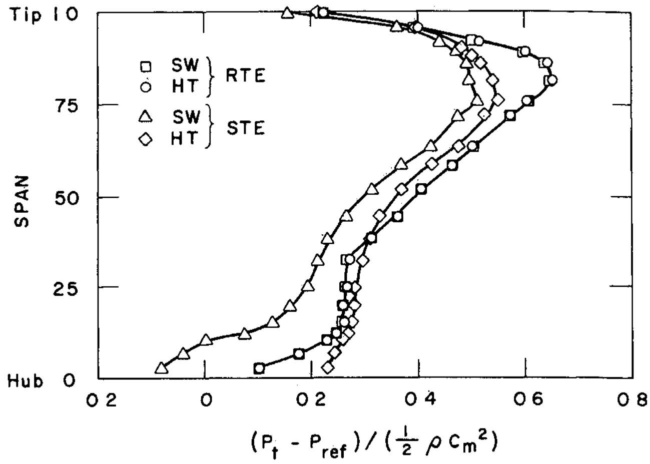

Fig 6 Stator inlet and exit total pressure distributions with solid wall (SW) and hub treatment (HT)

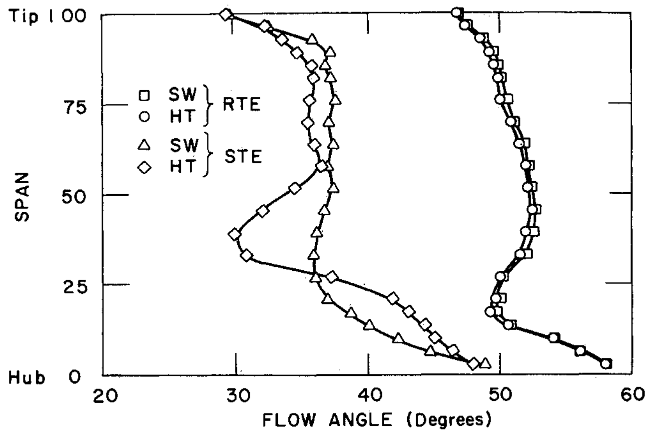

Fig 7 Stator inlet and exit flow angle distributions with solid wall (SW) and hub treatment (HT)

presumably to the accumulation of dirt, which could be seen to build up on the compressor blades) However, the maximum difference found due to this effect was a flow coefficient shift with both solid wall and hub treatment of 2%, i.e the changes were an order of magnitude less than the differences between the solid wall and hub treatment configurations The data presented are representative of the compressor in a clean condition

Radial Traverse Total Pressure Data

In order to examine the features of the flow in more detail, radial traverses were taken with solid wall and with hub treatment at $C_x/U=0\ 36$, which corresponds to a "near stall" point with the solid wall A comparison of the radial distribution of the (arithmetic) pitch averaged total pressure at rotor and stator exit for both hub treatment and solid wall, at this near stall point is presented in Fig 6 The symbols RTE and STE refer to the locations downstream of the rotor and stator respectively, as shown in Fig 1, and again, SW denotes solid wall and HT denotes hub treatment In the figure the vertical axis is radial location (percent span) and the horizontal axis is the nondimensionalized total pressure The refererence pressure is ambient (compressor inlet total pressure) and the dynamic pressure used for non dimensionalization is based on the midspan stator inlet velocity The reason for this latter convention is that due to the very nonuniform total pressure profile (set up, as stated to load the stator hub) there is no one "freestream" value of the stator inlet dynamic pressure; however, the midspan quantity is a convenient representative value

In examining the stator exit traverses it should be men tioned that there was an axial gap of 3 mm behind the stator (see Fig 1) between the rotating hub piece and the stationary annulus, with a dead ended cavity below Although this may affect the absolute values of total pressure and flow angle very near the hub, it is to be emphasized that it is the comparisons *between* solid wall and hub treatment that are of most interest here and the gap geometry is the same for both of these The comparisons should thus be valid, especially since the dif ferences are quite substantial

It can be seen in Fig 6 that the rotor exit total pressure profiles are nearly identical so that for both smooth wall and hub treatment the (stator) inlet conditions are the same However, the two stator exit profiles are quite different With the solid wall there is a large total pressure defect (and also a large blockage) near the stator hub In contrast, with the treatment the stator exit total pressure is higher than the rotor exit total pressure near the hub It is evident that work has been done on the flow in the endwall region due to the fluid injected from the grooves and that the heavy loss (and large blockage) in the hub region has been reduced (Note that for this set of experiments the flow is essentially in compressible so there is no net flow out of the treatment. However since the flow entering and leaving the grooves has very different velocity, there can be a net momentum flux out of the grooves)

Another feature to be noted in Fig 6 is that even at midspan the treated stator exit total pressure with hub treatment apparently is higher than that which occurs with the solid wall One would not expect the mixing effects (i e , shear forces) to extend to this radial station from the endwall region In fact the reason for this total pressure change is that there is a large blockage at the hub region with the solid wall and the flow near the hub is forced to move outwards In consequence there is a considerable radial streamline shift that occurs between the RTE and STE axial locations for this con figuration This radial shift (convection) of the nonuniform total pressure profile causes the apparent total pressure change in the midspan region, as discussed below

Flow Angle

Figure 7 presents a comparison of the absolute flow angle α measured from the axial direction The figure shows the midpitch radial distribution of angles with smooth wall and hub treatment at $C_x/U=0\ 36$ It is seen that again the two rotor exit profiles are very similar, the difference between the two being within 1/2 deg at all radii In contrast, the two

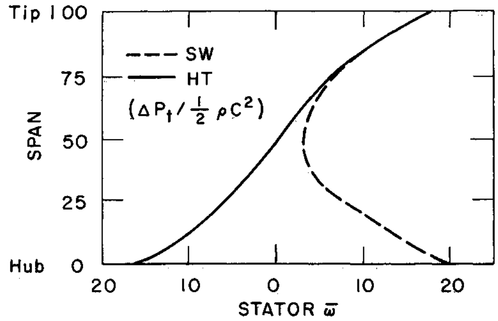

Fig 8 Nondimensional total pressure change across stator ($\bar{\omega}$)

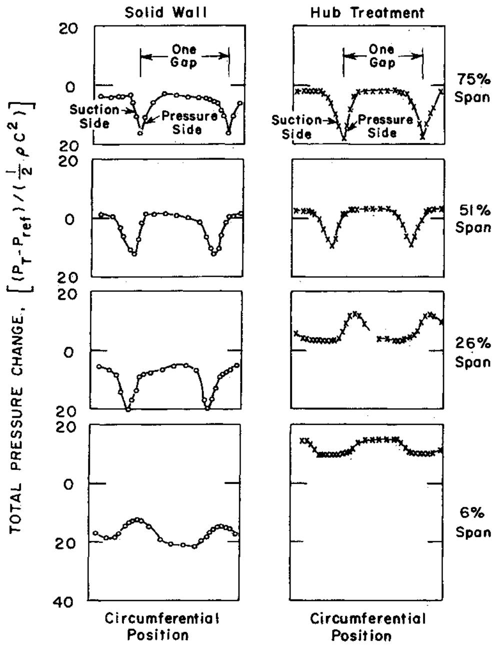

Fig 9 Circumferential total pressure distribution at stator exit

stator exit profiles are quite different With the treatment there is an increase in angle very near the hub but a decrease in angle i e , an increase in turning, in the midspan region due to the treatment; similar results were obtained in the low stagger case As discussed in Ref 3 this suggests that a radial flow along the blades is created by the treatment

Total Pressure Changes Across the Stator

The effect of hub treatment on the total pressure change across the stator is shown in Fig 8, which is also at $C_x/U = 0\ 36$ We have defined the total pressure change coefficient $\bar{\omega}$ in a conventional manner; (pitch averaged) total pressure change across blade divided by local inlet dynamic head This quantity has been calculated using the measured data and an axisymmetric (streamline curvature) data analysis program [9] For the solid wall the distribution of this total pressure change is in accord with what one would expect, with low losses near the midspan region and higher values at the two endwall regions With treatment, however, the total pressure change coefficient becomes negative from midspan to hub endwall, showing clearly that the rotating hub is doing a significant amount of work on the flow

Circumferential Traverses

A more detailed way in which to examine the effect of treatment on the flowfield is to compare the circumferential distribution of total pressure at the STE station for the solid wall and hub treatment This comparison is made in Fig 9, which shows plots of total pressure change across the stator

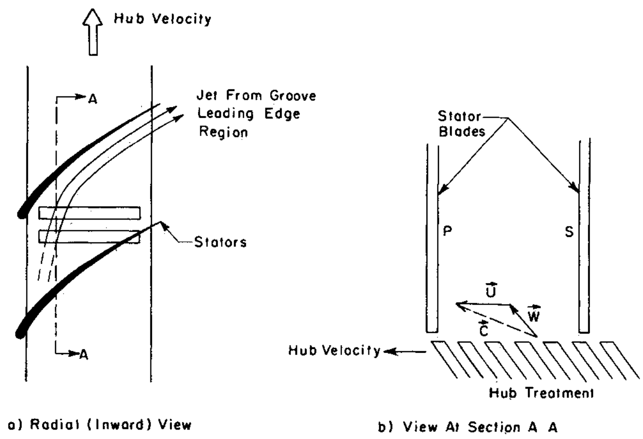

Fig 10 Conceptual picture of jet from groove leading-edge region

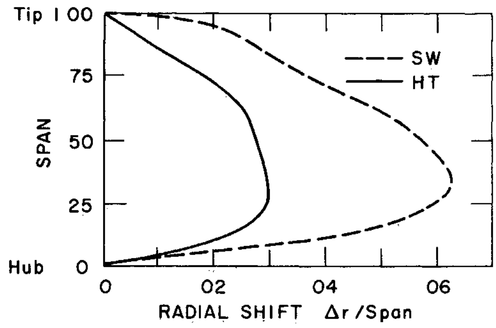

Fig 11 Radial streamline shift across stators (from data reduction using streamline curvature calculation)

blade row (divided by local inlet dynamic head) vs circumferential position at several different span locations The total pressure used for reference in this figure is the rotor exit total pressure which has been convected to the appropriate stator exit radial position using the data reduction program [9] The dynamic pressure used for nondimensionalization is based upon the local stator inlet velocity

At the midspan region and outboard the wake distributions are similar for the solid wall and the treatment although the magnitudes are slightly different At the inboard stations, however, there are major differences between the two We consider the 6% location first Although the reference dynamic pressures at inlet are *slightly* different, between the solid wall and the hub treatment, these small differences are far overshadowed by the differences in the wake profiles For the solid wall the wake is very deep with large loss With hub treatment the wake profile is flatter and, more important the total pressure is higher than the rotor inlet total pressure at all circumferential positions

Further evidence of the differences between solid wall and hub treatment is seen in the comparison of the wake distribution at the 26% span location With the solid wall the wakes are fairly thin, and this is viewed as one indication that the onset of stall is due to a wall stall at the hub Examinations of blade element data[8] also support this view With the hub treatment however, there is a "spike" in total pressure near the pressure surface side of the passage We take this to imply the indication of a *jet* of high total pressure fluid This is consistent with the idea of strong flow injection out of the slot leading edge region, impingement on the pressure surface and consequent rearward deflection

A tentative conceptual picture of the situation is shown in Fig 10 The left hand side of the figure shows a sketch of the jet from the groove leading edge region The right hand side gives a sectioned view looking downstream into the rotor

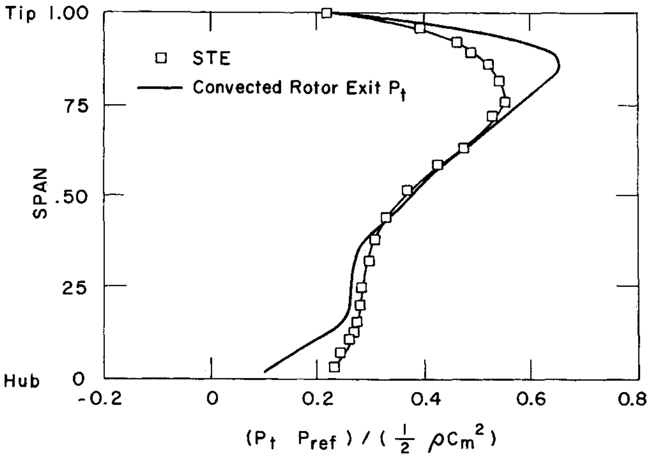

Fig. 12 Convected rotor exit and (midpitch) stator exit total pressure distributions (convection of inlet profile based on streamline curvature calculations).

passage, and shows absolute and relative velocity vectors. It can be seen that if the relative velocities out of the grooves are on the same order as other representative fluid velocities in the stator passage, the absolute velocities can be considerably higher than wheel speed.

Radial Flow Shifts Across the Stator

Previously we have discussed the differences in radial streamline shifts for the two configurations. The radial streamline shifts derived using the data reduction program are presented in Fig. 11. In the figure the streamline shift between the RTE and STE axial locations, divided by span, is plotted as a function of radius. For the untreated case there is a large streamline shift due to the blockage occurring near the hub, while for the treated case the shift is smaller because of the reduced blockage.

Using this information about the radial flows we can now "close the loop" and present a rotor exit total pressure profile which has been convected according to the radial streamline shift shown in preceding figure. Figure 12 shows this shifted rotor total pressure profile and the measured midpitch stator total pressure profile for the treated hub configuration at $C_x/U=0.36$. The reference pressure used here is the ambient pressure, and the dynamic head for nondimensionalization is based on the midspan stator inlet velocity It is seen that in the neighborhood of the midspan the two profiles are closely the same. While this perhaps conveys no new phenomenological information compared to Figs. 8 and 9 it does indicate the consistency in the data analysis and the experimental results, since at midpitch the flow can be considered inviscid and the total pressure *should* be constant along a streamline. Near the hub the stator total pressure is higher than the shifted rotor total pressure; as described previously this is viewed as being due to the net momentum flux out of the treatment, which is seen as a key characteristic of hub treatment.

In addition to the data presented previously, other data of a diagnostic nature were also taken with this configuration to investigate the sensitivity of the results to Reynolds number. Over the range tested (0.75 to 1.25×10^5) there was little change (less than 3% *maximum*) in any of the measured performance parameters either for the solid wall or the hub treatment configuration. There was a (small) decrease in pressure rise at a given flow coefficient as the Reynolds number was decreased, as might be expected. However, since the difference between treated and untreated builds at *any* of the Reynolds numbers tested was an order of magnitude larger than this, Reynolds number effects on the overall phenomenon can be regarded as small.

Discussion

From the experimental results it appears that there is a strong similarity between hub treatment and casing treatment; flow injection out of the slots and consequent energizing of the endwall region and blockage reduction being a key element of the operation of both. Accompanied with the casing treatment results, it is also believed that the hypothesis of "blade stall" and "wall stall" can be a useful criterion for successful treatment application. From a more detailed fluid dynamic point of view, however, the basic mechanism for the operaton of endwall treatment is still not really understood in a quantitative manner. In this regard, the terms wall stall and blade stall are not really descriptive in any sort of fundamental fluid mechanic sense. It is thus important to quantify more precisely the features of the flow regimes that are associated with each type of stall onset process. The present set of experiments therefore can be viewed more as an overall examination of the problem and it is clear that questions, such as those posed in Ref. 1 about the nature of blade and wall stall, will require information on a more detailed level.

There is one further point that can be mentioned. As previously discussed, it is hypothesized that the delay in stall onset due to endwall treatment is associated with momentum transfer between slots and blade passage. This momentum transfer is viewed as reducing the blockage associated with the endwall region. We have explored this using a simple endwall boundary-layer flow model to examine the effect of casing/hub treatment on boundary-layer growth.[8] It is found that the momentum transfer at endwall due to flow in and out of the grooves has a much more powerful effect on the boundary-layer growth than the wall shear stress. The results indicate that the endwall treatment does have a very strong potential for decreasing the boundary-layer growth, although the effects as calculated with the simple model are larger than those observed in the limited data available.

Summary and Conclusions

1) An investigation has been carried out on the effects of a rotating grooved hub treatment on stator stall and performance. Two different sets of stators, a low stagger and a high stagger row, were tested—both with and without the axial skewed groove hub treatment.

2) It was found that for the low stagger row the application of hub treatment did not have a major effect on the stator performance. It also appeared that the type of stall encountered with this configuration was blade stall.

3) For the high stagger row the application of axial skewed slots to the rotating hub under a compressor stator resulted in a substantial change in both the stall onset point and peak static pressure rise of the stator. To the authors' knowledge this is the first time such results have been reported.

4) It is hypothesized that the success of the treatment was due to the suppression of wall stall at the hub; this being supported by hot wire surveys and circumferential traverses at stator exit. Accompanied with the results of the low stagger case, this seems to illustrate again that endwall treatment will be effective when the type of stall that occurs is wall stall.

5) It is seen that the large blockage at hub endwall region is removed when the treatment is applied. Not only did the high losses associated with this blockage decrease, but the total pressure actually increased across the stator near the hub region, because of the work done by the (treated) hub on the main flow

Acknowledgments

The authors would like to thank S. Akmandor for his help in improving the data reduction procedure. The authors are also grateful to C.C. Koch for useful suggestions concerning the experiment design. The comments of J.P Nikkanen, N.A. Cumpsty and Prof. F.E. Marble are also appreciated.

This work was supported by the Air Force Office of Scientific Research, Contract F49620 82 K 0002, Dr J D Wilson, Program Manager

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