NUMERICAL EFFORTS OF AERODYNAMIC RE-DESIGN IN A SINGLE-STAGE TRANSONIC AXIAL COMpressor
PART 1 : STATOR AXIAL COMPRESSOR

ABSTRACT
In many aerodynamic design parameters for the axial-flow compressor, three variables of tailored blading, blade lean and sweep were considered in the re-design efforts of a transonic single stage which had been designed in 1960’s NASA public domains. As Part 1, the re-design was limited to the stator vane only. For the original MCA (Multiple Circular Arc) blading, which had been applied at all radii, the CDA (Controlled Diffusion Airfoil) blading was introduced at midspan as the first variant, and the endwalls of hub and casing (or tip) were replaced with the DCA (Double Circular Arc) blading for the second variant. Aerodynamic performance was predicted through a series of CFD analysis at design speed, and the best aerodynamic improvement, in terms of pressure ratio/efficiency and operability, was found in the first variant of tailored blading. It was selected as a baseline for the next design efforts with blade lean, sweep and both combined. Among 12 variants, a case of positively and mildly leaned blades was found the most attractive one, relative to the original design, providing benefits of an 1.0% increase of pressure ratio at design flow, an 1.7% increase of efficiency at design flow, a 10.5% increase of the surge margin and a 32.3% increase of the choke margin.

INTRODUCTION
A constant need for higher efficiency, lower fuel consumption and cost reduction in gas turbine engines leads to increased pressure ratios per stage, resulting in higher stage aerodynamic loading and transonic flow compressor design. Total pressure loss of shock waves and subsequent losses caused by shock-induced boundary layer separation in blade rows are one of main targets to reduce in aerodynamic design. Modern aerodynamic blade element design of transonic axial-flow compressors requires consideration of additional features, such as tailored blading, lean and sweep, to reduce losses which develop from passage shocks at high Mach numbers, significant growth of boundary layer behind shocks, and higher blade loadings.

Blading design (or blade camber and thickness distribution design) aims for flow fields with minimal losses and wider off-design ranges. In the past, with the aid of empirical data built on numerous cascade tests, a set of profile families, such as the NACA65, the NGTE, and the DCA blading, was generally used. As inlet Mach number was increased further in high-subsonic and transonic applications, the MCA profile was frequently introduced in order to cope better with passage shock positions. For a direct control of diffusion rates by lowering peak Mach numbers on the suction surface, the CDA blading [1][2] was developed leading to improvements of efficiency and operability especially in two-dimensional cascade tests. However, as Behlke, R.F. [3] concluded, the CDA ability is limited when applied to three-dimensional compressor blading because of effects imposed by complex endwall flow fields. The limitation was also confirmed by Gelder, T.F. et al. [4] where CDA stator losses were found much higher than DCA counterparts near hub region because of increased endwall effects. It is obvious that two-dimensional CDA design philosophy cannot be directly applied to near-endwall regions, but works in the midspan core region.

The influence of blade sweep and/or lean on the aerodynamics of axial-flow compressors has been widely investigated in the literature. Wadia, A.R. et al. [5] tested aft-swept (or backward-swept) rotors, and found smaller stall margins than the baseline unswept rotor. However, an improved efficiency and a better stall margin were found in forward-
swept rotors. The same conclusion was drawn in Christoph, B. et al. study [6] where an aft-swept transonic rotor was numerically analyzed, and its stall margin was reduced while similar peak efficiency was maintained. Xing, X.Q. et al. [7] found no performance benefit from an aft-swept rotor but substantial gains in a stall margin and peak efficiency from a forward-swept rotor in the numerical analysis. Despite still remaining controversial, some performance improvement is generally expected with forward sweep because of reduced axial flow diffusion and more uniform spanwise loadings. Wadia, A.R. et al. [8] even showed that inlet distortion sensitivity was halved with the forward swept rotor in the front stage relative to the radial counterpart. The forward swept blades also demonstrated improvements in tip clearance sensitivity in McNulty, G.S. et al. work [9]. It was found that forward sweep causes a spanwise redistribution of flow toward blade tip and reduces tip loading in terms of the static pressure coefficient.

As for blade lean, LeJambre, C.R. et al. [10] found that bowed stators together with rotor blades with contoured endwalls numerically and experimentally demonstrated higher efficiency and improved stall margin in a multistage compressor. Weingold, H.D. et al. [11] showed in the numerical analysis that bowed stators generate radial forces on flow fields which reduce diffusion rates in the suction surface corner and substantially delay or eliminate the formation of corner separation. Fischer, A. et al. [12] investigated the effect of strongly bowed stator vanes on flow fields in a 4-stage high-speed axial compressor numerically and experimentally. The reduction of hub corner separation in the strongly bowed vanes was found to lead to performance improvements when the compressor was highly loaded beyond a peak pressure ratio point.

Gummer, V. et al. [13] applied 3D blading concept of combining blade sweep and lean to the engine section stator of an aviation engine fan, and showed the large hub corner stall could be alleviated by moving endwall low-energy fluid towards midspan. Denton, J.D. and Xu, L. [14] confirmed that blade lean and sweep had a significant benefit to stall margins, though their effects on efficiency and pressure ratio were shown remarkably small in the numerical study of a transonic fan. Gallimore, S.J. et al. [15][16] introduced 3D blade designs into multistage core compressors of an aviation engine with particular emphasis on the use of sweep and dihedral in rotor and stator design, and successfully showed performance improvements from both low-speed rig tests and high-speed engine tests. Jin, H. et al. [17] numerically studied the effects of forward sweep, lean and hub bending in a highly loaded transonic two-stage fan. The results indicated that the forward-swept rotor showed insignificant impact on the pressure ratio and the efficiency, but improved stall margin. The bowed stator design was found to increase all of the three performance parameters pronouncedly. Many studies are still available on the effects of blade sweep and/or lean on aerodynamic performance, but more systematic studies are hardly found in terms of aero design parameters.

The present paper is one of attempts of aerodynamic design studies aiming for performance improvements of a single-stage transonic axial-flow compressor, focused on a design parametric study in a systematic way, using the CFD approach. The effects of three design parameters, i.e., tailored blading, blade lean and sweep, were investigated for a stator which blades had been radially-stacked with the MCA blading. The stator row is only considered as the first part of the study. In total, 3 design variants of tailored blading, 2 variants of blade lean, 3 variants of blade sweep, and 2 variants of blade lean/sweep combined were investigated, and their design cases are summarized in Table 1. Due to the limited pages recommended, some cases which seem to draw less attention were not included in the paper.

**NOMENCLATURE**

LE : Leading Edge
TE : Trailing Edge
x,y,z : Cartesian coordinates ($\Delta x^2 = \Delta x^2 + \Delta y^2$)
m : Meridional coordinate ($\Delta m^2 = \Delta z^2 + \Delta r^2$)
LER : Leading-edge Re-camber
t : Blade thickness
$\beta$ : Blade camberline (metal) angle
$\theta$ : Tangential angle in cylindrical coordinates

**Subscript**
b : Blade
n : Normal
1 : Inlet
2 : Outlet

All metal and flow angles are measured from a meridional (or axial) reference.

**COMPRESSOR STAGE**

In 1969, there was an effort to develop aerodynamic design technology of highly-loaded high-Mach-number axial-flow compressor stages in NASA [18][19], executed by the Pratt & Whitney under Contract NAS3-10482, which geometry and test results are open to the public. It was selected as a baseline mostly because of no limits in sharing any findings including geometry and performance. A single compressor stage of 488 m/sec (1,600 ft/sec) tip speed and 838 mm (33
inch) rotor inlet tip diameter was designed to deliver a total-to-total pressure ratio of 1.936 with an adiabatic efficiency of 0.842 at design flow rate of 84.87 kg/sec (187.1 lb/sec) of air. At design point, rotor inlet relative Mach number was designed supersonic over nearly the entire span (1.61 at the rotor tip), while stator Mach number was subsonic across the span with a maximum of 0.89 at hub where the diffusion factor was expected to be 0.6. The stage was without inlet guide vanes, and no swirl was intended at stator exit at all radii. Both 30 rotor and 44 stator blades had radially-stacked MCA sections with a constant chord length from hub to casing. Three-dimensional shape and meridional flow path of the stage are shown in Fig.1. It was tested with uniform inlet flow, but design pressure ratio and efficiency were only met at about 97% of design flow rate at design speed in the test, as shown in Fig. 2. The inability of achieving design flow was probably caused by local choking near rotor hub[19]. For the present study, a new design flow rate was set to 83 kg/sec (183 lb/sec) which stays near the peak efficiency. As for the rotor, a basic rotor design without any slots was used here.

**NUMERICAL METHOD**

A single-stage steady-state CFD (Computational Fluid Dynamics) method [20][21][22], developed by the author and named CNSTURBO, was applied using the Reynolds-averaged compressible Navier-Stokes finite volume method with the 4-step Runge-Kutta time integration scheme and the $2^{nd}/4^{th}$-order artificial dissipation damping. The k-omega equations were implemented as a turbulence closure. H-type structured grids were generated in a multi-block system by solving elliptical Poisson grid equations. In general, about 400,000 to 500,000 nodes were used to build rotor and stator domains, respectively. One feature of the numerical grids used is shown in Fig.3. A design tip clearance of 1.27 mm (0.05 inch), corresponding to 0.3% of span at the rotor inlet, was modelled in the rotor, but no clearance was considered in the stator. The conventional mixing plane was applied to the interface between rotor and stator. Neither the fillet radius of blades on the endwalls nor the secondary flow channel through seals under the stator platform was included, because the fundamental blade passage was primarily concerned looking for aero design philosophy. At rotor upstream boundary, ambient uniform total pressure and total temperature were given with an axial flow direction, and at stator downstream a hub static pressure was specified to let the radial equilibrium equation determine spanwise static pressure distributions.

![Fig.2 Single-stage compressor characteristics](image-url)
The solution was regarded as converged when the normalized residual, a measure of local imbalance of continuity, momentum and energy in each conservative control volume, fell below $1.0 \times 10^{-5}$. It should be noted that the lowest mass flow in predicted compressor maps does not mean a true surge point because any reverse flows occurring at lower flows in the numerical computation become an obstacle to solution stability. The minimum steady flow with an acceptable tolerance of solution convergence would be considered close to the limit of operating ranges. Every CFD point represented by symbols on the compressor map has the identical set of exit boundary static pressures specified on the hub. Therefore at the same peak exit pressure if one case is converged but the other is not, the former is expected to have a smaller surge flow. Endwall velocities in figures to be presented hereinafter show non-zero values because they were simply extrapolated from interior cell-centered control volumes for the purpose of an easier post-processing.

CFD VALIDATION
In the stage overall characteristics of Fig.2, a good agreement to test data on both two speed lines of 100% and 70% of design speed was found. The 70% speedline was added to show how the code predicts off-design speed characteristics, despite only the 100% speedline will be considered in the present study. Isentropic efficiency was calculated using mass-averaged total pressure and mass-averaged total temperature at rotor inlet and stator exit locations. Air was assumed an ideal gas. Slightly higher choking flows were predicted, but those gaps could be blamed to either tested rotor geometry deviation from the design (possibly due to the existence of blade fillet radii) or numerical discrepancies in evaluating surface blockages. Near-surge flows at the 70% speed line were well located relative to test flows, while at the design speed a minimum converged flow was predicted higher than test data. One of the reasons of disagreements can be found from the fact that rotor incidence is suspected to be altered from test hardware, considering that there exists a small geometry discrepancy in the rotor inlet metal angle which was converted from two-dimensional radial section coordinates in the reference [18]. However, as shown in Fig.4, at a mass flow rate of 83 kg/sec (183 lb/sec) of design speed, spanwise blade-element performance has a good agreement with measured data at a flow point near design pressure ratio. Another acceptable agreement was found in the stator downstream wake of Fig.5 where tangential traverses of stator exit total pressure were taken at 90% span from hub near stall at design speed.

RESULTS AND DISCUSSION
Before starting main design improvements of the stator, a small change of inlet and exit metal angles of the original stator vane was made along the span. At the new design flow rate of 83 kg/s, which was reduced from the original design flow, a better efficiency was expected by reducing incidence (i.e., closing the blade inlet toward the circumferential direction) along the span. Exit metal angles were also modified to be rather linearly re-distributed along the span on purpose of providing a reference in future rotor design modifications which would be aiming for a zero exit swirl. Such a modified design was named “Original-Modified”, and its spanwise distributions of the blade camberline metal angle at the leading-edge and the trailing-edge are shown in Fig.6 against those of “Original”. Metal angle distributions will be discussed later. The conventional loss buckets, plotted for three...
spanwise regions from the present CFD outputs by taking area-averaged pressures and mass-averaged total pressures circumferentially, were shown in Fig.7. It was confirmed that the reduction of incidence in Original-Modified at design flow lowered the level of total pressure losses of the blade in the core region of 30% to 70% span from hub and in the near-casing region of 70% to 100% span, and slightly increased the surge/stall margin. No improvements were found in the near-hub region, which implies that more reduction of incidence would be workable, but not tried further.

\[ \beta_b = \frac{\beta_{b1} - \beta_b}{\beta_{b1} - \beta_{b2}}, \quad \bar{t}_n = \left\{ \frac{t_n - t_{n1}}{t_{n,\text{max}} - t_{n1}}, \frac{t_n - t_{n2}}{t_{n,\text{max}} - t_{n2}} \right\}. \]

The notation used for tailored blading in Fig.8 follows the sequence of three representative spanwise sections of “Hub-Midspan-Casing(or Tip)” for the three different blading designs which are MCA-MCA-MCA (i.e., Original-Modified), MCA-CDA-MCA and DCA-CDA-DCA. Total 21 sections along the span were used in the design of three dimensional blade shapes, and for any sections other than the three sections of the hub, midspan and casing, a smooth transition was applied with an appropriate interpolation. It needs to be noted that the three different blading designs have identical distributions of blade metal angles at both inlet and exit along the span (which are those of “Original-Modified” in Fig.6), while following the distributions of blade camber and thickness shown in Fig.8. They are all radially stacked along a series of CG (Center of Gravity) points on each section, i.e., without any lean or sweep of the blade. More details of the three different blading designs are described in Fig.9.

A stage compressor map, predicted by the present CFD, is shown in Fig.10 where the three different blading cases, i.e., MCA-MCA-MCA (Original Modified), MCA-CDA-MCA and DCA-CDA-DCA after LER1, plus the original design are compared together. As expected, Original-Modified was confirmed to show better performance than Original at the new design flow rate of 83 kg/sec, in terms of pressure ratio and efficiency, thanks to reduced losses from the improved incidences discussed with Fig.7. Closing the blade inlet of Original design helped extend the range of stable operation, from design flow to a flow limit close to surge, but the choke margin was reduced due to the change of throat area, especially from the change of inlet metal angles near hub. When two design groups with and without the CDA blading at midspan are compared, a clear benefit from with the CDA is observed showing an impressively elevated level of both pressure ratio and efficiency over the range and also providing wider

![Fig.7 Total pressure loss buckets](image_url)

![Fig.6 Blade inlet and exit angles at new design flow](image_url)
The case of DCA-CDA-DCA blading (Case no. 4 of Table 1) was found (but not shown here) short of the operating range as well as pressure ratio/efficiency at lower flows, relative to the case of MCA-CDA-MCA. It implies that there might be a need for shifting the design incidence of the latter by closing the blade inlet toward the circumferential direction more, especially near endwalls, in order to reach the map of MCA-CDA-MCA blading. Such an additional design effort is called “LER” (Leading Edge Re-camber), and was accordingly made with the spanwise distribution of blade inlet metal angle, LER1, of Fig. 6(a). Between DCA endwalls and MCA endwalls, it is hard to say which one is better in pressure ratio/efficiency performance near design flow, but there is a clear gap in the range of stable operation, showing that the case of MCA-CDA-MCA blading exceeds in operability.

Detailed investigation among the cases of tailored blading is available by looking into blade loadings in the spanwise direction. Fig. 11 shows static pressure distributions along blade pressure and suction surfaces at three selected spans at design flow for the three cases of blading. They were plotted for a normalized meridional distance from the leading-edge to the trailing-edge, instead of a normalized axial distance or chord length, because of a better way of expressing the present flow path which is not purely axial. In the 50% span (or midspan)
plot, a clear distinction of the style of blade loading between
the MCA (which is center-loaded) and the CDA (which is front-
loaded) blading can be found, and the present CDA blade
design looks acceptable judging from a smooth flow diffusion
on the suction surface although small wiggles were found near
the leading-edge (which were removed in the following design
improvements). At 10% span (near hub), the flow of Original-
Modified starts to separate from the suction surface near 40%
to 50% of total meridional length from the leading-edge, kept
stalled all the way downstream, leading to a lower level of the
overall pressure recovery. The flow separation was however
considerably decreased in the two design cases with CDA
blading at midspan, easily observed from the elevated level of a
static pressure recovery at the trailing-edge. Compared to that
of MCA endwalls, the case of DCA endwalls experiences local
flow reversals due to a significantly larger blade loading near
the leading-edge, even though it has a good incidence through
the LER adjustment. At 90% span (near casing), a separated flow region is found on the suction
surface starting from about 50% of total meridional length, and
flow stays stalled downstream. The two cases with CDA
blading at midspan help reduce the stalled flow near casing to
lift the level of the overall pressure recovery, but not so much
benefit as found near hub. The findings are also supported by
Mach contours near the suction surface, presented in Appendix.

Fig.12 and Fig.13 present the spanwise distributions of
circumferentially-mass-averaged total pressure at stator inlet
and exit, and exit meridional velocity at design flow,
respectively, for the three tailored blading cases. Stator inlet
and exit locations are described as the position number 3 and 7
in Fig.1(b). Compared to Original-Modified, the two cases with
CDA blading at midspan showed a substantial recovery of total
pressure near hub at stator exit, which resulted in the significant
performance benefit observed in Fig.10. The increase of total
pressure near hub could be expected from the static pressure
recovery found at 10% span of Fig.11 which came from the
reduced flow separation. The energized flow near hub is then
confirmed with the increased meridional velocity found in
Fig.13 which happened in balance with the controlled diffusion
in the core region. It looks interesting that the effect of CDA
blading spread out toward about 10% and 90% span positions,
but most of the spanwise migration of flow momentum was
heading toward hub. A small amount of total pressure recovery
was found near casing in Fig.12, which would support the static
pressure recovery at 90% span in Fig.11.
As for incidence and deviation at design flow, Fig.6 needs to be revisited which also provides the spanwise distributions of blade metal angles and flow angles at stator inlet and exit for the tailored blading cases. Again, all cases, except the case of DCA-CDA-DCA after LER1, have the inlet metal angles of Original-Modified, while all cases are with exit metal angles of Original-Modified. The flow angles were circumferentially momentum-averaged from CFD outputs. Some negative incidence around 70% to 90% span of Original-Modified had intentionally been created to help mitigate the original flow separation on the suction surface near casing.

The introduction of CDA blading at midspan greatly reduced flow deviation, relative to Original-Modified, by the same amount from about 10% to 90% span, irrespective of endwall blading. That would be accredited to the reduction of exit meridional velocity observed in Fig.12, together with an increased exit circumferential velocity accordingly. A jump in flow deviation near hub was caused by the sharp increase in meridional velocity found in Fig.13. At midspan about -5° of an exit swirl was predicted which is far off from the design requirement of zero-swirl, but adjustment efforts will be made after the redesign of the rotor blade which will be the Part 2 study.

In the design efforts of tailored blading, the case of MCA-CDA-MCA was found the best performance provider in terms of pressure ratio, efficiency and operating range, even without going through the LER adjustment. The case of DCA-CDA-DCA after LER1 showed nearly the same performance, except for the range of stable operation which ran shorter. A good answer to why it faced an earlier stall would be said referring to Fig.11. At 10% span near hub, much larger blade loading was predicted near 20% streamwise position, despite still being at design flow. It tells that the DCA blading near endwalls, particularly near the stator hub, is subject to overloading due to earlier diffusion than the MCA blading and due to subsequent endwall effects. The case of MCA-CDA-MCA blading will be used as a baseline stator design for the effects of blade lean and/or sweep which will follow.

2. TANGENTIAL LEAN

In the present study, blade lean and sweep are defined relative to tangential and meridional (or axial if no changes in the radius) directions, for the sake of the author’s convenience in building a blade 3D geometry. Those definitions do not follow the “true” ones which are generally along a line normal to blade chord and along a blade chord line, respectively. Even though the present findings might not be all consistent with those of other studies based on the “true” definitions, they will not much deviate from information supporting an aerodynamic design methodology. Fig.14 describes all of stator design variants applied to investigate the effects of blade lean and sweep on aerodynamic performance.

![Fig.14 Stator vane shapes for the effects of blade lean and sweep](image)

![Fig.15 Stator blade lean distributions](image)
As the second concept of performance improvements of the stator vane, two types of blade positive lean, “Lean1” and “Lean2 (excluded here)”, were applied to the baseline (i.e., the MCA-CDA-MCA blading) which had been pre-selected in the study of tailored blading and named “Baseline” hereinafter. They were created by changing the stacking line in the tangential direction through shifting their CG (Center of Gravity) locations, relative to the midspan CG, along the span as shown in Fig.15. Lean1 has the effect of blade lean locally close to both hub and casing endwalls, while Lean2 is with much stronger effects of blade lean from midspan toward both endwalls. Positive lean means in the direction of positive lift of the airfoil. Any negative lean was not tried because it has been well reported that negative lean or dihedral designs bring a larger hub corner flow separation with higher losses and blockages[14]. All blade geometry, including inlet and exit blade metal angles, remained unchanged from Baseline, except for the stacking lines.

Fig.16 is a predicted stage compressor map showing the effect of blade lean between Baseline (with no lean) and Lean1 after LER2, where the LER process was added to Lean1 with LER2 of Fig.6(a) in order to cope with the changes of spanwise meridional velocity and flow angle profiles produced by blade lean. A significant benefit of blade positive lean, compared to Baseline, was observed in aerodynamic performance of pressure ratio and efficiency, but no benefits in higher flows close to choke. The range of stable operation from choke to near-surge looks almost the same in both cases, while Lean1 after LER2 shows higher levels of pressure ratio and efficiency except for higher flows.

To see more details of the effects of blade lean, meridional velocity distributions along the span were plotted in Fig.17 at design flow for total 7 streamwise sections whose positions were described inside. Relative to velocities of Baseline (No-Lean), the region near casing of the leaned blade gets fuller velocity profiles over all the streamwise positions. That is thanks to added radial force to the radial equilibrium of the flow between hub and casing, well known as the effect of blade lean, leaving the flow near hub weaker. Especially at stator exit (Position 7) a significant increase in the meridional velocity near casing was seen, which brings enhanced aerodynamic performance through suppressing flow separation.

Static pressure distributions around the blade surfaces at the three spanwise sections at design flow were presented in Fig.18. At 10% span near hub, the static pressure curves around the blade are all shifted upward, particularly more shifts of the suction surface, due to the spanwise migration of flow momentum toward casing. At 90% span near casing, compared to Baseline, the leaned blade loses static pressure on the suction surface from inlet down to around the mid-chord position.
because of fuller velocity profiles. The reduction of separated flow on the suction surface boosts the static pressure recovery at the trailing-edge. At midspan, quite smoother controlled-diffusion than Baseline is seen along the suction surface, cleaning up the wiggles.

In Fig.20 of spanwise total pressure distributions at stator inlet and exit, the total pressure drop across the stator was significantly improved from 70% span toward casing, thanks to the benefit of reduced flow separation by introducing the positive lean.

3. MERIDIONAL SWEEP

As the third concept of performance improvements of the stator vane, total three meridionally-swept blade designs, a forward sweep on casing (Sweep1), a backward sweep on casing (Sweep2), and a forward sweep on both casing and hub (Sweep3), were applied to a baseline which geometry was identical to Baseline (No-Lean), called Baseline (No-Sweep) here.

Those swept blades were created by changing the stacking line through shifting CG locations in the meridional direction along the span. The strengths of the sweep are described in Fig.21 where the meridional distance of CG at each spanwise section, relative to a normal to hub originating at CG, is plotted against the normalized span from hub. Even with no sweep, the plot of Baseline is curved backward toward casing because the normal
to hub CG is not purely radial in the present case. The stacking line of Sweep3 was a little pushed downstream in order to avoid any interference on hub with the rotor blade downstream in the grid generation. All blade geometry, such as inlet and exit blade metal angles, remained unchanged from Baseline, except for the stacking lines. Results of cases of Sweep2 and Sweep3 were not included here.

In general, sweeping blades causes a spanwise variation in induced velocity at the leading-edge because there are no blades above or below themselves. It brings the change of blade force and meridional velocity in both streamwise and spanwise direction, especially near the leading-edge. Therefore, Sweep1 needs the LER adjustment from Baseline metal angles for a better incidence control, leading to “Sweep1 after LER2”.

Fig.22  Effects of blade sweep on stage CFD characteristics

In the CFD overall compressor characteristics of Fig.22, compared to Baseline (No-Sweep), it showed the same performance gains of pressure ratio and efficiency, found in the case of blade lean of Fig.16, but failed to get an extended surge margin. Between Fig.16 and Fig.22, a very small performance reduction in pressure, efficiency and operability is observed in the blade sweep.

In Fig.23, the meridional velocity distributions in the spanwise and streamwise direction at design flow, Sweep1 after LER2 had fuller velocity profiles, compared to Baseline of Fig.17, in the front part of the blade near casing due to the forward sweep there. Endwall sections which are forward-swept experience no blades above or below them to maintain blade forces. Therefore blade loading near the leading-edge quickly drops to bring a local increase of velocity, and peak diffusion loadings are accordingly shifted downstream. In Fig.24 of blade loading plots at design flow, a sharp reduction of blade loading at the leading-edge and a rear-shift of the peak diffusion location were well observed at 90% span. The level of static pressure around the blade at 10% span was consistently moved upward, which implies a spanwise migration of flow momentum caused by the casing sweep. The 50% span plot represented a compensation for the changes of blade force near the endwalls. Quite smoother controlled-diffusion than Baseline is seen along the suction surface, cleaning up the wiggles, as seen in Fig.18.

Fig.23 Meridional velocity distributions along the span at design flow at 7 streamwise positions with blade sweep

Fig.25 shows the effects of blade sweep on the exit flow angles along the span at design flow where a considerable reduction of separated flows near casing was brought by Sweep1 after LER2. The distribution also looks similar to that of blade lean in Fig.19, but shows a larger deviation close to the casing. Total pressure drops along the span across the swept stator blades were plotted in Fig.26 where big improvements in the forward sweep were found from 70% span thanks to reduced flow separation.

4. COMBINATION OF LEAN AND SWEEP

As the last concept of performance improvements of the stator vane, two more designs were tried by applying blade lean and sweep together. On the baseline design (i.e., MCA-CDA-MCA blading), Lean1 was combined with Sweep1 and Sweep3, respectively as described in Fig.27, expecting some augmented improvements in aerodynamic performance. As found before, they needed the LER adjustment once again from the LER2 to the LER3 of Fig.6(a), due to the 3D effects added, for better incidence distributions.
Fig. 2.4 Blade loading distributions at 3 spanwise sections at design flow with blade sweep.

Fig. 2.5 Effects of blade sweep on exit flow angle at design flow.

Fig. 2.6 Effects of blade sweep on total pressure at design flow.

Fig. 2.7 Stator vane shapes for the effects of combined blade lean and sweep.
They were then named “Lean1Sweep1 after LER3” and “Lean1Sweep3 after LER3” (excluded here), respectively. Fig. 28 presents the overall stage performance predictions near design flow, but surprisingly no further improvements were found in the combination. That is to say, Lean1Sweep1 after LER3 showed almost the same performance of pressure ratio, efficiency and operability as Lean1 after LER2. Fig. 29 presents the meridional velocity profiles in the spanwise and streamwise direction at design flow for the blade lean and sweep combined. Compared with the blade sweep only (Fig. 23), it is clearly observed that the spanwise flow migration from hub by the introduction of positive lean to the already-swept blade did not extend toward casing but only strengthened flows from 50% to 80% span, and near-hub velocities were reduced accordingly. Eventually the meridional velocity profiles returned close to those of Lean1 after LER2 (Fig. 17). In the blade loading distributions at design flow of Fig. 30, it is also confirmed that Lean1Sweep1 after LER3 has almost identical distributions to those of Lean1 after LER2. It appears that the effects of blade lean are more dominant in the combination design.

The current forward sweep, applied to the positively leaned stator blade, failed to bring additional improvements in aerodynamic performance. In other words, selecting Lean1 after LER2 as the final aero design could be more attractive in reducing any potential negative issues of structural integrity in terms of blade stress, vibration and manufacturability. If the final design selection, Lean1 after LER2, is stated with the “true” definitions of blade lean and sweep, the blade is mostly positively leaned and very slightly backward-swept at endwalls, referring to Fig. 31.

5. VANE COUNT

In the present re-design of the stator, no efforts of reducing blade counts were made in applying the changes of 3D shapes, because the diffusion factor at design flow already stayed around the limit of design recommendations. Fig. 32 shows the Lieblein diffusion factor using circumferentially-mass-averaged velocities, at design flow for the re-design variants. Total three groups of distributions were interestingly observed. The addition of CDA blading at midspan increased the diffusion factor in the core region up to about 0.45 from about 0.4 of Original-Modified. Applying the 3D shapes of blade lean and/or sweep raised it again up to about 0.5, no matter what types of design were concerned. Over-loaded endwalls were significantly alleviated with benefits from blade lean and/or sweep as discussed.

CONCLUSIONS

As one of fundamental research topics on aerodynamic design technology of axial-flow compressors, the parametric effects of tailored blading, blade lean and sweep were numerically investigated in a systematic way for a stator row of a transonic single stage which aims for a design pressure ratio of 1.936. Meaningful findings are summarized below.
Tailored Blading

In the first design variant for tailored blading among MCA-MCA-MCA (Original-Modified), MCA-CDA-MCA, and DCA-CDA-DCA blading,

- By introducing CDA blading at midspan of the radially-stacked original MCA vane, a significant increase of pressure ratio, efficiency and choke margin, and a small increase of surge margin were obtained.

- Most benefits from CDA blading at midspan are the reduction of meridional velocity in the core region thanks to controlled diffusion on the suction surface, and the subsequently balanced shift of flow momentum downward leading to the reduction of originally stalled flow near hub.

- The case of DCA-CDA-DCA blading showed almost identical performance to the case of MCA-CDA-MCA blading in terms of pressure ratio and efficiency, but failed to attain as wide a range of stable operation, even after the LER adjustments. The loss of operability was mostly because of still higher blade loadings near hub.
Blade Lean
In the second design variant of tangentially leaned blades of Lean1 (which was mildly leaned at endwalls),
- Adding positive blade lean at both hub and casing to the radially-stacked MCA-CDA-MCA blading provided an increase in pressure ratio and efficiency at flows away from choke. Such a benefit results from getting fuller velocity profiles near casing thanks to spanwise flow migrations from hub triggered by the radial blade force of the positive lean.
- After the LER adjustments, which re-distributed incidences in the spanwise direction expecting a range extension, Lean1 after LER2 showed a significant increase of operability while maintaining the same level of pressure ratio and efficiency.

Blade Sweep
In the third design variant of the meridionally swept blades of Sweep1 (forward-swept at casing),
- Sweep1, with forward sweep at casing added to the radially-stacked MCA-CDA-MCA blading, provided almost the same level of benefit found from Lean1, thanks to fuller velocity profiles near casing caused by the change of blade loading in the chordwise direction.
- The benefits of forward sweep at casing result from an increase of meridional velocity near casing where blade loadings near the leading-edge quickly drop because the swept blade experiences no blades below to support blade forces. Peak diffusion loadings are accordingly shifted rearward.
- Blade loadings in Sweep1 compensate for balances of flow momentum in the regions where sweep exists or does not, resulting in an increase of blade loadings from midspan toward hub.
- Even after the LER adjustments, no more extension of operability was found in Sweep1, because of still higher blade loadings near endwalls.

All Combinations
In the last design variant of combination shapes, Lean1Sweep1 after LER3 was found acceptable in the aerodynamic performance point of view. However,
- Nearly identical performance to that of Lean1 after LER2 was predicted in terms of pressure ratio, efficiency and operability, which implies the effects of blade lean would be more dominant in the 3D combination design.
- Considering potential negative issues related to structural integrity, Lean1 after LER2 might be more attractive to a final aero design.

If the surge and choke margin are defined with mass flow rates at design speed as,

\[
\text{Surge Margin} = \frac{\dot{m}_{\text{design}} - \dot{m}_{\text{minimum}}}{\dot{m}_{\text{design}}} \times 100(\%)
\]

\[
\text{Choke Margin} = \frac{\dot{m}_{\text{choke}} - \dot{m}_{\text{design}}}{\dot{m}_{\text{design}}} \times 100(\%)
\]

the new design of Lean1 after LER2 was concluded as the best for the stator vane, predicting benefits of an 1.0% increase of pressure ratio, an 1.7% increase of efficiency at design flow, a 10.5% increase of the surge margin and a 32.3% increase of the choke margin, relative to Original-Modified.

In the present parametric study, as mentioned earlier, no stator clearances were considered. Depending on the gap size (% of span), hub blading and radial blade force might be influenced by mixed effects of leakage vortices, hub boundary layers and secondary flows near the hub. Knife seals under the platform were not considered, either. If included, both suction and blowing across the hub will affect blade loadings near the hub.

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REFERENCES

APPENDIX

Suction Surface Mach Contours

Mach contours near the suction surface of the stator provide straightforward insights on the progress of performance changes. Below are some of the design cases plotted at design flow together with relative Mach near the suction surface of the rotor. A typical corner stall is seen in the stator of Original-Modified with heavy flow separations at both endwalls and with midspan mildly stalled. In DCA-CDA-DCA after LER1, the CDA blading at midspan alters velocity distributions significantly removing the mild stall. The DCA blading at both endwalls mitigates the corner stall, and especially nearly removes hub stall. However, near the hub a local flow acceleration and a subsequent rapid diffusion happen due to an excessive front loading from the DCA blading. When hub loading is reduced by switching to the MCA blading in MCA-CDA-MCA, a smoother flow diffusion is obtained, but the corner stall near the casing is still active (because of very high flow turning requirements there). The introduction of moderate positive tangential lean near endwalls dramatically remove the corner stall near casing in Lean1 after LER2 thanks to flow migration toward casing. In alternative ways, a forward sweep on casing (Sweep1 after LER2) can be considered to suppress the corner stall near casing. Compared to Lean1 after LER2, it shows suction-side-flow-acceleration, resulting in increased blade loadings, toward the hub. A combination with Lean1 in Sweep1 after LER2, which is Lean1Sweep1 after LER3, puts blade loadings back to those of Lean1 after LER2.

The rotor undergoes a strong shock and boundary layer interaction near the suction surface from the tip down to about 25% span, which imposes a considerable portion of flow losses. The improvement efforts of the rotor will be made in Part 2 study.

Fig.A1 Mach contours near the suction surface of the rotor and stator cases at design flow