Darmstadt Rotor No. 2, II: Design of Leaning Rotor Blades

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For Darmstadt University of Technology's axial single-stage transonic compressor rig, a new three-dimensional aft-swept rotor was designed and manufactured at MTU Aero Engines in Munich, Germany. The application of carbon fiber-reinforced plastic made it possible to overcome structural constraints and therefore to further increase the amount of lean and sweep of the blade. The aim of the design was to improve the mechanical stability at operation that is close to stall.

To avoid the hazard of rubbing at the blade tip, which is found especially at off-design operating conditions close to the stability limit of the compression system, aft-sweep was introduced together with excessive backward lean.

This article reports an investigation of the impact of various amounts of lean on the aerodynamic behavior of the compressor stage on the basis of steady-state Navier-Stokes simulations. The results indicate that high backward lean promotes an undesirable redistribution of mass flow and gives rise to a basic change in the shock pattern, whereas a forward-leaning geometry results in the development of a highly back-swept shock front. However, the disadvantage is a decrease in shock strength and efficiency.

Keywords Bow shock, Rotor design lean, Sweep, Transonic compressor

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has been operating the high-speed transonic compressor rig—which represents a typical front-stage of a commercial turbofan, high-pressure compressor—since the early 1990s. The main purpose of the test stand is to establish a database for computational fluid dynamics (CFD) code validation as well as to serve as a test bed for new materials.

The initial rotor no. 1 was designed as a bladed disk (Blisk) made of titanium with conventionally stacked blade sections. The encouraging experience with vanes made of composites led to the use of carbon fiber-reinforced plastic (CFRP) to construct a high-speed compressor rotor with swept blades. The material's obvious benefits—reduced density combined with strength similar to that of titanium alloys—allowed for rotor blade geometries with amounts of sweep and lean that are not possible in a titanium-based design.

Because of lack of experience in predicting blade tip bending, especially under unsteady aerodynamic load at stall, an aft-swept design was chosen for Rotor No. 2 so as to prevent severe damage in case of rubbing. Rig tests partly presented by Kablitz and colleagues (2002) successfully demonstrated reliable operation of the rotor up to the highest speeds. Because the main focus of the project was to gain experience in building three-dimensional rotor blade geometries using CFRP, the high amount of lean was chosen independent of aerodynamic necessity. The results of the measurements (Kablitz et al., 2002) provided a reason to investigate the aerodynamic impact of significant variations in lean.

ROTOR GEOMETRIES INVESTIGATED

Darmstadt Rotor No. 2

Details concerning the mechanical design and the manufacturing can be found in the work of Pirker and colleagues (2002) and Blaha and colleagues (2000). The aerodynamic design intent for Rotor No. 2 was to influence the shock structure by sweeping the leading edge aft so as to minimize loss. Tip speed, mass flow rate, and pressure ratio at the design-point are the

FIGURE 1
Meridional view of Rotor No. 2.

same as for Rotor No. 1. To ensure the option of traverse probes between rotor and stator, the axial dimension of the rotor had to be held almost constant (Fig. 1), so most of the aft-sweep had to be introduced by aft-lean (Fig. 2a). Figure 3 lists the major design parameters of the compressor utilizing Rotor No. 2. Further information on the aerodynamic design and the tools used for it can be found in the paper by Blaha and colleagues (2000).

For the analysis presented in this article, two variations in the lean of the Rotor No. 2 blade geometry were generated. In the first step, Rotor No. 2's defining airfoil cross-sections (Fig. 2a) were radially stacked (Fig. 2b), and in the second step, they were inversely stacked (Fig. 2c), leading to a forward-leaning blade geometry with the same radial distribution but inverted-leaning angles (Fig. 4). No attempt was made to modify the stagger or airfoil parameters to take into account the altered flow field that had been caused by the heavily varied lean.

COMPUTATIONS

For the three rotor geometries, steady-state Reynolds-averaged Navier-Stokes (RANS) computations at design speed were conducted.

FIGURE 2
Cross-sections of the rotor blade. Lean variations: (a) Rotor No. 2 (aft-leaning); (b) radially stacked rotor; (c) forward-leaning rotor.

Pressure ratio	1.5
Corrected mass flow rate	16.0 kg/sec
Corrected tip speed	398 m/sec
Relative rotor inlet Mach number at tip	1.35
Relative rotor inlet Mach number at hub	0.70
Shaft speed	20,000 rpm
Tip diameter	0.38 m
Number of blades	16
Inlet hub-to-tip ratio	0.51
Mean aspect ratio of Rotor No. 2	0.98
Rotor No. 2 solidity (hub / mid / tip)	1.7 / 1.4 / 1.1

FIGURE 3
Design parameters of the compressor.

Solver

Trace_S, an explicit cell-centered, second-order-accurate, finite-volume scheme was used to solve the RANS equations. To speed up convergence to steady-state, local time-stepping, residual smoothing, and successive mesh refinement were applied (Fritsch et al., 1997).

A high Reynolds κ - ϵ model with wall functions was used. All simulations were run in fully turbulent mode. International Standard Atmosphere standard conditions with purely axial flow were assumed at the inflow. A rather pronounced incoming casing boundary layer was prescribed by a total pressure profile. The boundary layer at the hub was neglected because of the strong acceleration over the spinner. The measured turbulence intensity of 1% was prescribed at the inflow boundary. At the

FIGURE 4
Leaning angles of the investigated geometries.

FIGURE 5

Typical process to convergence.

outlet, a radial equilibrium was assumed and the average static pressure was fixed at mid-span.

Every computation ran for 6000 time steps, which resulted in an execution time of approximately 24 (Central Processing Unit) h on a Silicon Graphics, Inc., Octane. The achieved state of convergence can be deduced from Figure 5, which may be taken as being representative of all the calculations discussed in this article.

Grids

An identical block-structured O–H grid was used, as is summarized in Figure 6, and it adds up to a total of 630,000 nodes. The nominal clearance amounts to 1 mm (1% channel height) at the rotor tip gap (Fig. 7).

The relative motion of the rotor and casing was included, but the cavity between rotor and the stator at the hub was neglected. The grid density was refined at the hub and casing so as to increase the resolution of the boundary layers as well as the passage shock (see Figs. 7–9).

Number of Nodes:	Axial	Pitchwise	Radial
Rotor H-grid	104	32	64
Upstream from rotor	28		
In rotor passage	64		
Downstream from rotor	12		
Stator H-grid	104	32	64
Upstream from stator	12		
In stator passage	64		
Downstream from stator	28		
Rotor O-grid	128	8	64
Stator O-grid	128	8	64
O-grid in tip-gap	128	6	8

FIGURE 6

Grid topology.

FIGURE 7

Grid at tipgap.

ANALYSIS

In Figure 10 the computed stage-performances of the three investigated rotor-blade geometries are displayed together with the measured speed line of Rotor No. 2 at design speed. Details regarding the experimental setup can be found in Kablitz and colleagues (2002). The calculations conducted for the Rotor No. 2 geometry conformed closely with the experiment. For the radially and the inversely stacked rotors, additional simulations were run to determine the characteristics of their speed lines.

The predicted pressure ratio of the radially stacked rotor was slightly higher than that of Rotor No. 2, whereas the forward-leaning rotor generated a significantly lower pressure rise. The peak efficiency of the forward-leaning rotor appears to be similar to that of Rotor No. 2, although the corresponding mass flow rate is lower. The simulation of the radially stacked rotor predicts an isentropic efficiency that is approximately 0.5% higher than that of Rotor No. 2. The following discussion compares computations that have a prescribed static backpressure of 1.22×10^5 pascals, leading to a total pressure ratio of about 1.43.

Figure 11 shows the radial distribution of the stage's total pressure ratio. As expected, based on the stage performance

FIGURE 8

Blade-to-blade grid.

FIGURE 9
Meridional grid.

results, the radially stacked rotor delivered roughly the same radial pressure rise as Rotor No. 2 in terms of plot magnification. In the forward-leaning rotor, an improvement in pressure rise was observed from the hub to the mid-span. At the outer span, however, its level fell far below that of Rotor No. 2. In addition, the rotor loss coefficient (Fig. 12) increased between 30% and 90% in channel height and in the tip-clearance vortex region in the forward-leaning rotor, whereas the tip leakage losses decreased.

FIGURE 10
Stage performance at design speed.

FIGURE 11
Radial distribution of the stage's total pressure ratios.

As expected on the basis of theory (Denton et al., 1999; Hah et al., 1998; Wadia et al., 1998), the altered pressure field between the rotor blades, which is caused by lean, led to a shift in mass flow toward the hub with increased aft-lean (Fig. 13), resulting in a relative reduction in velocity at the aft-leaning rotor's blade tip and thereby in increased incidence.

It is remarkable that in respect to the radially stacked blading, the magnitude of the decrease in velocity at mid-span of the forward-leaning rotor exceeded the increase in velocity of the

FIGURE 12
Radial distribution of rotor's loss coefficient.

FIGURE 13

Radial distribution of the absolute Mach number at the rotor inlet.

aft-leaning rotor. In contrast to those findings was the absolute Mach number at the rotor exit, as shown in Figure 14. There, an opposite redistribution of the mass flow occurred. Also, the transported mass flow declined toward the tip in all investigated rotor geometries, corresponding to the evenly distributed total pressure ratio along the span, as shown in Figure 11.

FIGURE 14

Radial distribution of the absolute Mach number at the rotor exit.

Rotor No. 2 itself, as described by Blaha (2000), suffered from a work deficiency at the tip, which was attributed to the radial flow migration which, in turn, was due to the aft-lean. The anticipated advantageous effect of the forward-leaning rotor on the radial distribution of the mass flow seemed to be overcompensated for by the nonuniform, radial work input and by the deflection resulting from the changed shock pattern.

Figures 15, 16, and 17 show the relative, isentropic profile Mach number at 90%, 52%, and 10% channel height, respectively. As noted earlier, the reduced axial velocity at the tip led to an increased incidence.

It can be seen in Figure 15 that the shock became weaker and its location moved farther downstream with increased forward lean. Additionally, the preshock Mach number tended to rise. The remarkably high level of the isentropic Mach number at the trailing edge of all three rotor geometries, noticeably supersonic at the forward-leaning blading, explained the work deficiency.

Figure 16 illustrates that the higher mass flow in the lower half of the aft-leaning rotor's passage promoted a double-shock system. Because of the decreased rotor inlet Mach number at the mid-span, the forward-leaning rotor's shock strength was reduced, and the flow did not expand to a second shock. Near the hub (see Fig. 17), the shock was more intense and was located farther upstream with increased forward lean.

Considering the contour plots of the suction-side isentropic Mach numbers (Fig. 18), Rotor No. 2's two shock fronts, whereas

FIGURE 15

Relative isentropic Mach number at 90% channel height.

FIGURE 16

Relative isentropic Mach number at 52% channel height.

FIGURE 17

Relative isentropic Mach number at 10% channel height.

FIGURE 18

Suction side isentropic Mach number contours at $Pi = 1.43$.

almost radial, were slightly curved upstream. The increased lean promoted reduced strength and decreased inclination of the shock front at the tip, leading to a heavily aft-swept shock at the forward-leaning rotor. Figure 19 quantifies the sweep of the shock fronts on the suction surfaces, giving the measured angles relative to the x axis in Figure 18.

	Hub	Tip
Rotor No. 2 (aft-leaning)	70°–85°	76°–90°
Radially stacked rotor	83°	72°
Forward-leaning rotor	–8°–80°	45°–80°

FIGURE 19
Sweep-angles the shock front.

FIGURE 20
Isentropic Mach number near trailing edge, at axial plane as seen in Figure 21 (forward-leaning rotor).

FIGURE 21
Streamlines at mid pitch (forward-leaning rotor).

As pointed out by Hah and colleagues (1998), the shock does not intersect the casing obliquely. Instead, it turns normal, as can be seen in Figure 20. Streamlines shown for the forward-leaning rotor at mid pitch (Fig. 21) are projected onto the meridional plane. They illustrate the mass flow's deflection in the rotor passage, which was pronounced at the aft-swept shock above the mid span. The forward-leaning rotor demonstrates the possibility of an enormous shock inclination, which is desired by many designers. The disadvantage of reduced shock strength and the resulting redistributed mass flow is that they destroy the efficiency benefits previously gained from the improved shock situation.

CONCLUSIONS

This numerical investigation of three-dimensional blade geometries in a high-speed transonic compressor rotor provides some insight into the ways in which lean controls the work input and the radial distribution of the mass flow.

It has been observed that a highly oblique shock front on the meridional plane is possible, but it has the disadvantage of forcing a strong radial migration of the flow to the hub, which in turn affects the efficiency gained from the shock's inclination. Further improvements can be expected from the optimization of the profile shapes certain three-dimensional blade applications.

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