HIGHLY LOADED TANDEM COMPRESSOR CASCADE WITH VARIABLE CAMBER AND STAGGER

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ABSTRACT

Highly loaded compressors, resulting from high pressure ratios and as few compressor stages as possible, need not only stators with variable stagger but furthermore with variable camber. By this means the blading can be adapted to the velocity triangles at off-design engine power setting leading to less sacrifice of the cycle efficiency and thus better engine power and improved fuel consumption.

Stators with variable stagger and camber can be accomplished by a tandem stator design where both stator blades can be adjusted independently from each other. In this investigation the usable range of such a tandem arrangement is measured and it is shown how the interference between the two blades can decrease losses in comparison with the two single blades of the tandem arrangement. Furthermore an inviscid flow calculation with a 2D-Euler-code shows the influence of this interference on the whole flow field.

NOMENCLATURE

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<td>a</td>
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<td>axial displacement</td>
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<td>c</td>
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<td>$\Delta p_{\text{DPT}}$</td>
<td>[hPa]</td>
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<th>Abbreviations</th>
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<tr>
<td>M</td>
<td>circumferential position of blade II at mid pitch of blade I</td>
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<td>part</td>
<td>portion of total pressure loss</td>
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INTRODUCTION

Modern jet engine designs are required to achieve improved specific thrust and efficiency. This can be accomplished by higher turbine inlet temperatures accompanied by increased pressure ratios of the core engine. The further requirement of improved thrust to weight ratios leads to designs with as few as possible compressor stages increasing the stage pressure ratio at the design point even more. However, this increased stage pressure ratio necessitates preventive measures at off-design power settings in order to guarantee the necessary compressor surge margin for the acceleration of the engine in the required time limit.

The problem of a decreased compressor surge margin at off-design power settings results from the fact that the annulus diagram is designed for the density increase in the compressor at design speed. The increased pressure ratio of the compressor however, leads to an increased compressibility effect on the behavior of the compressor at reduced rotational speed, i.e. the pressure decrease is even larger than the decrease of the mass flow rate at reduced rotational speed. This results in two important consequences:

1. The axial velocity in the rear stages of the compressor increases since the smaller density increase needs a larger cross-sectional area between hub and tip. This leads to the known effect that the rear stages of the compressor tend to block the flow connected with a throttling of the front stages of the compressor.

2. The velocity triangles of a compressor stage do not remain geometrically similar when reducing the rotational speed of the compressor thus changing the flow angles.

In order to explain this effect in detail the results of an investigation of an 11-stage compressor with a pressure ratio of 35 reported by Wunderwald (1989) were used to get the dependence of the axial velocity on the rotational speed in the first stage of this compressor (Fig. 1).

Starting with the velocity triangles at the design point this dependence is used to calculate the velocity triangles for any rotational speed. The basis of the following considerations is the fact that the absolute exit angle of a stator and the relative exit angle of a rotor remains constant within the safe working range of the blading. Using this assumption and the condition that the flow to the rotor of the second stage is free of incidence, the inlet and exit flow angles of the first stage stator are calculated for various rotational speeds and plotted in Fig. 1.

Let us assume now the simple case that the stator of the first stage is adjustable by changing the stagger angle while the rotor geometry is invariable. Adjusting the stator according to the required exit flow angle $\beta_e$ leads to an increasing incidence at the inlet of the stator with decreasing rotational speed, since the required deflection of the stator $\beta_s - \beta_e$ decreases with decreasing rotational speed. Thus a stator blade which incorporates as well a variation of the stagger angle as a change of the deflection by changing the camber could cope much better with the velocity triangles at lower rotational speeds than a stator where only the stagger angle can be varied. Since a solid blade cannot vary both stagger and camber, two stator blades have to be placed one behind the other, and both stator blades have to be adjustable. This tandem-stator design Fig. 2 is the basis of the following investigation but the question remains how the adjustment of the two stator blades can be accomplished by varying the stagger angle of the first blade $\beta_{s1}$ and the stagger angle of the second blade $\beta_{sII}$.

The stagger angle of the second stator blade II $\beta_{sII}$ is adjusted according to the required exit angle $\beta_e$. The already mentioned incidence with decreasing rotational speed is taken care of partially by the first and partially by the second stator blade by adjusting the stagger angle of the first blade $\beta_{s1}$ to a function of $\beta_{s1}$ and $\beta_{sII}$. This function has to take into consideration that the boundary layer builds up anew on the blade II which facilitates an increased blade loading of the tandem design compared to a single blade design. The special behavior of this tandem blade design is the basis for tests on the high-speed cascade wind tunnel and blade-to-blade calculations with a 2D-Euler-code.

Comparison of the tests with earlier investigations Beelte (1979), Hebbel (1963), Linnemann (1964), Ohashi (1959), Pal...
Fig. 2 Tandem Cascade

(1965), Sieverding (1966), Staude (1975) with similar geometries is rather limited since these investigations did not consider bladings with variable camber and furthermore those tests were partly performed at low Mach-numbers equivalent to incompressible flow.

**2D-EULER COMPUTER CODE**

The computer code used for the blade to blade calculation was developed at the Institut für Strahlantriebe und Turbomaschinen of the RWTH-Aachen. This time-marching procedure for finite areas (McDonald, 1971) solves the two-dimensional Euler equations with the assumption of constant total enthalpy. Proceeding from the time-dependent conservation laws it can be used as well for subsonic flow as also for supersonic flow without making the change of the flow type necessary. The grid generation of this procedure, which until now was used essentially only for supersonic profiles (Kauke, 1986), had to be fit to the transonic blade in this investigation. The EULER-code itself is not modified. This grid was generated in such a manner that the grid lines \((J = \text{const})\), which describe the contour of blade II (C and D in Fig. 3) are identical from the origin of the calculation grid up to the leading edge of blade II. This facilitates the computation of the flow through two blades which as in this investigation are located in axial direction one behind the other with the further possibility of partial overlapping. A smaller grid size around the leading edge and the trailing edge of the blades helps to obtain a good resolution around those points with the used H-grid. This feature is rather important in order to resolve the gaps between the two blades as exact as possible when blade II is shifted with respect to blade I in circumferential direction. The disturbance of the total pressure on rounded leading and trailing edges as to the Euler-procedure in connection with the H-grid was accepted and not corrected. The inlet and boundary conditions for the calculation are \(p_t, T_t, p_2, \text{ and } \text{Ma}_l\) as a start value of the velocity at the inlet.

**EXPERIMENTAL INVESTIGATION**

The test facility consists of the enclosed high-speed cascade wind-tunnel, the external drive unit, and the data acquisition and reduction system including a 32 bit PE 3203 computer and suitable means for printing and plotting the results of the measurements.
High-Speed Cascade Wind-Tunnel

The High-Speed Cascade Wind-Tunnel Fig. 4 is contained in a tank which is 4m in diameter and 11.8m long. The wind-tunnel can operate continuously.

Mechanical energy is supplied from a 1.3 MW 3-phase motor which runs essentially on constant rpm. The adjustment on the desired rpm of the compressor drive is accomplished by a hydraulic coupling and a gear transmission. It's drive shaft enters the tank and drives the 6-stage axial compressor. The air is supplied from inside the tank. After leaving the compressor the air is cooled to a constant temperature of 40°C. After the cooling process the air flow passes the nozzle whose exit height can be varied from 250mm to 500mm by a width of 300mm, while a turbulence generator supplies the desired turbulence level of $T_u = 3.0\% - 4.3\%$ in the test section where the cascade is located. The air leaves the cascade into the tank loosing it’s kinetic energy thus rendering the wind-tunnel an open-loop system in an enclosed test facility.

The set-up of the wind-tunnel holds distinct advantages which otherwise could not be accomplished. By setting the exit area of the nozzle and the rpm of the compressor any desired Mach-number between 0.2 and 1.05 can be obtained in the test section. At the same time the pressure inside the tank can be varied between 0.05 and 1.2 bar setting the Re-number for a profile length of 1m between $10^4$ and $1.5\times10^5$. These two variations can be applied independently from each other so that the Mach-number and the Re-number setting is not coupled. This facilitates the independent variation of the most important parameters in the testing of cascades.

Built-up of the tandem-compressor cascade

The whole set-up of the tandem cascade arrangement from Fig. 2 and the nozzle of the high-speed cascade wind tunnel is shown in Fig. 5.

The blades were designed at MTU in connection with the development of a computer code for calculating the flow through tandem cascades assuming inviscid flow by Seelmeier (1987). Every blade row consists of 7 blades with a height of $h = 300\text{mm}$ (channel width). The chord length of the first blade row is $l_1 = 83.2\text{mm}$ while the chord length of the second row is $l_2 = 72.3\text{mm}$. The individual blades are attached to the bladeholders by rotatable mounts which enable a stepless
This is checked by the measurement of the static pressure in the $p_1$-plane and this static pressure should be constant over the whole cross section. The standard instrumentation at the cascade inlet comprises the measurement of the total temperature $T_1$ in the plenum with a Pt100-probe, the total pressure $p_{t1}$ with a pitot-probe and the static pressure $p_2$ by wall pressure taps in the $p_1$-plane. In the exit plane the wake is traversed with a wedge-probe within one pitch to determine the total pressure $p_{t2}$, the static pressure $p_2$, and the exit flow angle $B_2$. The distribution of the static pressure along the profile is measured by 12 pressure taps on the suction side and 12 pressure taps on the pressure side of two profiles of both blade rows as shown in Fig. 2. These blades enclose the flow channels of the blades for the wake measurement which are therefore not instrumented. A program package was installed on the multi user 32-bit Perkin-Elmer 3203 computer of the test facility for the control of the measurements and the acquisition of the data as well as the reduction of the data according to the exact momentum method (Amecke, 1967).

MEASUREMENT PROGRAM

The extensive measurement program is displayed in Fig. 6 and can be split in two main parts. In the first part the characteristic of the tandem cascade:
- at various relative circumferential positions of blade II
  (s-Variation)
- at three combinations of the stagger angles
- and of the single blade row I is determined. In the second part the camber of the tandem arrangement is varied by a variation of the stagger angle of blade II $B_{II}$ at two constant conditions on blade I. Furthermore this $B_{s,II}$ Variations were also performed without blade row I.

These tests were also used in order to verify the blade to blade calculations with the 2D-Euler code.

RESULTS

Isentropic profile-Mach-number distribution and results for the s-variation

As an example for the influence of the interference between both blades the blade-Mach-number distribution from the tests and the calculations are displayed as a function of the axial chord Figs. 7a, 8a, 9a as well as the results of the flow field calculation Figs. 7b, 8b, 9b at design geometry and design inlet flow at three different relative circumferential positions $s$ of the blade II.

The following abbreviations are used:

PS: blade II near the pressure side of blade I
SS: blade II near the suction side of blade I
M: blade II at mid pitch of blade I
The agreement between the measured and the calculated profile-Mach-numbers is satisfactory as Figs. 7a, 8a, 9a show. The shock position and its extension can be reproduced exactly neither by calculation nor by measurement. The different profile-Mach-number at the leading edge of blade I is due to resolution of the applied H-grid. It can be concluded that the influence of the interference between both blades can be simulated very well by the calculation and the calculated field results can make clear the flow processes.

When the two blades I and II are located close together (position PS, SS) a gap arises. This gap is not optimized for flow acceleration in order to stabilize the boundary layer of blade II, but it results from certain combinations of the stagger angles of both blades and leads to an extensive deceleration of the flow close to the gap as Figs. 7 to 9 show. This effect is also found in the resulting profile-Mach-numbers on positions SS and PS contrary to position M where the influence of the interference between both blades is relatively weak.

At position SS, as Figs. 8 show, the speed on the suction side of blade I is noticeably lower so that at the trailing edge of the blade the flow is more decelerated on the suction side than on the pressure side. This leads to a change of the flow exit direction of blade I such that blade II experiences an inlet flow with positive incidence which is noticeable from the increased velocity on the suction side of blade II.

At position PS an essential unloading of blade II results from the extreme decrease of the speed level on the pressure side of blade I as Figs. 9 show.

### Total pressure losses

**Total pressure losses at design inlet flow for the s-variation.** The essential unloading of blade II at position PS leads to a strong decrease of the total pressure losses so that the lowest total pressure losses are experienced at this position as Fig. 10a shows for $B = 131.8^\circ$ and $s = 20.7\text{mm}$. At position SS blade II is heavier loaded, which leads to an increased total pressure loss for this blade. However the maximum Mach-number of blade I is reduced and thus it's shock losses in such a way that the total pressure losses are lower than for the reference location M. This is shown in Fig. 10a for $B = 131.8^\circ$ and $s = 38.8\text{mm}$. Position M with the weakest influence of the interference between both blades
leads to two separate wakes for blade I (Fig. 10b right wake) and blade II (Fig. 10b left wake) for a large variation of the inlet flow direction. Thus the total pressure losses can be split into the portion of blade I shown in Fig. 10a for $s = 71.7$-I and into the portion of blade II (Fig. 10a) for $s = 71.7$-II, by reducing the data from the measurement three times: both wakes and the single wake of blade I as well as of blade II. The portion of blade I predominates because of the prevailing shock losses due to the large supersonic flow field. The portion of blade II is relatively constant since this blade experiences an inlet flow which is close to design conditions. Thus the total pressure loss of blade II does not increase when the exit flow of blade I undergoes an insignificant change of the flow direction.

**Total pressure losses for design geometry and $B_1$-variation.** In addition to the total pressure losses at the design point also the losses for the $B_1$-variation are plotted in Fig.
10a. It is found that the loss behavior of position PS is as long favorable as the flow on the pressure side of blade I is strongly decelerated due to positive incidence and the accompanying unloading of blade II. This advantage diminishes if blade I is unloaded. At position SS an increased shock loss due to positive incidence is added to the losses arising from the higher loading of blade II. If the incidence is negative the exit flow direction of blade I changes even more. Thus blade II is further loaded and its losses increase beyond those obtained at position M.

Fig. 9a $M_{a,x}$ of Position PS

Fig. 9b Flow Field Calculation of Position PS

Fig. 10a Loss coefficient at three circumferential displacements

Fig. 10b Wake measurement at design conditions

The knowledge gained up to now leads to the point that further investigations should be limited to position M due to the following reasons:

- the total pressure loss is lower for position M in a larger range of inlet flow angles than for the other positions.
- if the stagger angles of both blades are varied independently from each other, position M prevents the possibility of unwanted gaps or even contact between both blades.

**Total pressure losses for B1- variation at a reduced inlet Mach-number.** Fig. 11 shows the total pressure loss of the tandem cascade at an inlet Mach-number of 0.7 and a mass flow rate corresponding approximately to a compressor speed of 85% of the design speed.

Therefore the stagger angle of blade I was increased by 10° to a value of 129.2° according to the changed velocity triangle at the exit of the first rotor. In a test A the stagger angle of blade II remains at 105° and is increased in a test B by 20° to a value of 125° according to the velocity triangle at the inlet of the second rotor. The two combinations of the stagger angles lead to total pressure losses which are plotted in Fig. 11 for both blades (marked DPT), the portion of blade I (marked DPTI), and the portion of blade II (marked DPTII). It is found that the total pressure loss in A is somewhat higher because of the positive incidence of blade II of 10°. In B blade II has a negative incidence of 10° which leads to lower losses. In spite of the higher loading of blade I (βI = 137.8°, βII = 129.2°) a further decrease of the total pressure loss of blade II in B is anticipated since the lower deflection of blade I leads to a reduced incidence of blade II. The changing portion of blade I (DPTII) between A and B can only result from the interference between both blades since the stagger angle and inlet flow conditions of blade I are equal in both tests. The higher velocity peak on the suction side of blade II in test A decreases the possible deceleration on blade I and thus the portion of the losses of blade I. Comparing the losses of blade II with those of Fig. 10a it is found that also the portion of the losses of blade II are dependent on B. Since blade II has an incidence of 10° in both tests (positive in A and negative in B) it reacts very sensitively to a change in the flow exit angle of blading I and is thus also strongly dependent on B.

**Total pressure losses for a variation of the camber.** For a variation of the stagger angle of the second blade BII the resulting total pressure losses are shown in Fig. 12 for the constant condition of blade I: MaI = 0.7, BI = 137.8°, BII = 129.2° and in Fig. 13 for the constant condition of blade I: MaI = 0.836, BI = 131.8°, BII = 119.2°.

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**Fig. 11 Loss coefficient at two combinations of B**

**Fig. 12 Loss coefficient at B-variation and reduced Ma**

**Fig. 13 Loss coefficient at B-variation and design Ma**
However, not only the total pressure losses of the whole tandem cascade (marked TANDEM) are shown in both figures but also the portion of total pressure loss of cascade I (marked I part) and the portion of cascade II (marked II part) which are measured by traverses behind the tandem cascade. Furthermore, by running a test only with cascade I under the same condition as cascade I in the tandem arrangement the plotted results marked I single were obtained. Further tests only with cascade II at the inlet conditions equivalent to the exit conditions from the above mentioned test with cascade I results in the plotted curve marked II single. By comparing the various curves of Fig. 12 it can easily be deduced that only the influence of the interference between both blades effects the pressure losses of the tandem cascade in a positive manner. If the Mach-number and the camber are reduced below the design values, the total pressure loss of the tandem cascade (TANDEM) is significantly below the losses of the single cascades (I+II single). Fig. 13 shows results at the design condition. A slight advantage of the tandem cascade is still found especially if the camber is increased above the design value ($B_{II} > 10^5$). However the bigger advantage of the tandem blade results from the distribution of the losses of blade I and blade II. The losses of blade II in the tandem arrangement are lower than those of the single blade II. Thus the loading of blade II in the tandem arrangement can be increased so that at the design condition a higher deflection is possible by a reduction of $B_{s,II}$.

SUMMARY

In order to optimize the fuel consumption and thus the economy of a jet engine high pressure ratios are needed which, in connection with a reduction of the number of compressor stages, lead to heavily loaded compressor stages. However at off-design power settings this heavy loading results in such changes in the flow direction between stators and rotors that adjustable stators of the usual design cannot be employed in an optimal manner for increasing the surge margin of the compressor.

As an alternative design a tandem stator consisting of two adjustable blades one behind the other is introduced and its behavior is tested at the design condition and at off-design power settings. It is shown that the total pressure loss of this tandem cascade is different from the total pressure loss obtained when the two blades of the tandem arrangement are tested separately. This difference is attributed to the interference between the two blades. At off-design power settings this interference leads to a reduction of the total pressure losses in the decisive case of reduced blade camber but at the design point a higher deflection is enabled.

Comparisons between the tests and calculations with an Euler-code show that the influence of the interference is well represented by the calculations also at high Mach-numbers.

ACKNOWLEDGEMENT

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