ABSTRACT

Experimental investigations of flow fields and losses in an axial flow compressor stage were carried out. The stage has hub/tip ratio of 0.7. The design values of flow coefficient and pressure coefficient are 0.6 and 0.81, respectively. Aerodynamic performance was investigated for two principal configurations: i) axial flow stage with variable rotor blades, ii) axial flow stage with variable inlet guide and stator vanes. The most efficient volume flow rate regulation of the stage was with the application of variable rotor blades.

On the basis of experimental data an analysis of the origin of flow separation on the suction and pressure surfaces of rotor and stator blades was made with the use of simple design criteria. The unsteady flow of rotating stall type in the tested stage appeared after simultaneous occurrence of large stall regions in both rotor and stator blade rows. The existence of large stall regions in the IGV did not affect the rotating stall onset. At high values of the IGV stagger angle change (50 deg) pressure pulsations appeared due to the occurrence of stall.

NOMENCLATURE

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Presented at the International Gas Turbine & Aeroengine Congress & Exhibition
INTRODUCTION

The change of mass flow rate in the axial compressor of industrial type and gas turbine, running at constant revolutions, is carried out by stagger angle change of the inlet guide and stator vanes. Industrial fan is often regulated by rotor blades turning. Only some basic information of qualitative nature can be found in the available literature for a particular type of regulation, e.g. Horlock (1958), Wallis (1983), Bohl (1981) etc.

This paper deals with all the above mentioned ways of the change of mass flow rate in a low-speed axial stage of compressor type. The stage is aerodynamically highly loaded (Cyrus (1996), Cyrus and Kreuzer (1997)). Working conditions of blade elements including the onset of large areas of stalled flow were studied. The origins of unsteady flows in the stage were also investigated. Some partial results concerning 3D flow through the rear axial compressor stage with variable inlet guide and stator vanes have already been published (Cyrus, 1994).

CASE R

The experimental investigation of the stage blading was performed in two steps. The first was devoted to measurement of characteristics. The second concentrated on detailed flow field investigation by means of 5-hole conical probes with sensor diameter 2.5mm in the blading planes 01, 02, 03 and 04 (Fig.1). The measurements were carried out within one IGV and stator blade pitch at 16 - 20 circumferential points in planes 02, 03 and 04, respectively. The probes were located on 7 - 10 radial positions in all planes. The total temperature was measured in the planes before and behind the rotor row by thermocouples.

At the entrance of the test rig there is an inlet measuring nozzle to indicate the mass flow rate. At the outlet there is a radial diffuser whose movable rear wall makes it possible to change the aerodynamic resistance of the test rig. The rig is driven by a DC motor with a swinging stator to measure torque by weighing.

The experimental investigation of the stage blading was performed by means of unsteady pressure transducers placed in the planes before and behind of the rotor blade row on the casing. Two peripherally shifted pressure transducers were used in each plane.

The flow parameters obtained from the pressure probe and thermocouple data were averaged over a blade pitch. The mean axial velocity was determined by area averaging. The mean values of peripheral and radial velocity components, of total pressure and total temperature were obtained by mass averaging. The static temperature and pressure were then determined using fundamental relationships of fluid dynamics. The average value of flow angle was established on the basis of the area averaged axial and mass averaged peripheral velocity components.

Absolute measurement uncertainty of pressure was ± 5 Pa. The flow angles measured by 5-hole pressure probes were recorded with the accuracy ± 1 deg. The measurement uncertainty of temperature was estimated at ± 0.3 K.
BLADING CHARACTERISTICS

Comparison of fields of axial-flow stage characteristics for variable rotor blades (case R) and variable inlet guide vanes (case I) is shown in Fig. 2. It is worth noting that the stagger angle change of rotor blades causes greater flow rate increment through the stage than a stagger angle change of the inlet guide vanes. This could be derived from the velocity triangles and the basic equations of internal aerodynamics. In the case R, certain pressure increment was produced by the stage blading at the minimal flow rate values. This is very important for the implementation of industrial fans in processing lines of chemical and environmental arrangements. The adjustment $\Delta \gamma_R = +36^\circ$ is the maximal one. The blades touch already each other. The above described minimum flow rate conditions cannot be reached in the case of the stage blading with variable IGVs (Fig. 2).

In the case I, the pressure ratio of blading does not increase for setting angle change of the inlet guide vanes $\Delta \gamma_{IGV} < -30^\circ$ (Fig. 2). The position of stability limit of the blading with the basic rotor blades stagger angle has not practically changed, if the inlet guide vanes were applied with $\Delta \gamma_{IGV} = 0$ deg.

Bladings characteristics are drawn in Fig. 2 for stage revolutions $n = 1500 \, \text{1/min}$ ($Re = 300 \, 000$). When stage revolutions were increased on $n = 2300 \, \text{1/min}$, blading efficiency increased by (1.2-1.5)% in the whole range of the flow coefficient for $\Delta \gamma_{IGV} = 0$ deg. At values $\Delta \gamma_{IGV} = \pm 20$ deg the differences were slightly lower. The positions of stability limits did not change within considered range of $Re$ number.

More detailed characteristics of blading are on Figs. 3a, b, and c for the stagger angle change of blades within (-20) deg to (+20) deg. Fig. 3a shows dependence of the pressure coefficient of the stage on the flow coefficient for the case R. The curves of constant values of isentropic efficiency are also drawn. It is evident, that decreasing or increasing stagger angle of rotor blades increases or decreases the flow rate through the stage, respectively. It is obvious, that the area with efficiency $\eta > 90 \, \%$ is relatively large. The maximal value of efficiency $\eta = 92 \, \%$ has been found.

In the case I, when the inlet guide vanes are used, the maximal value of efficiency is lower by 2% (Fig. 3b) when compared with the case R. This can be explained by decrease of $Re$ number and the IGVs energy loss. Also the area of high efficiency is considerably smaller. Some dependencies of the pressure coefficient on the flow coefficient of blading, when both inlet guide vanes and stator vanes are rearranged, are on Fig. 3c. The stagger angle change was carried out within the limits $\pm 20$ deg. It shows up very important, that the rearrangement of stator vanes does not practically influence the position of the stability limit. This is particularly noticeable for the stagger angle change of inlet guide vanes $\Delta \gamma_{IGV} = -20$ deg , when stagger angle change of stator vanes ranged within a wide interval: -20, 0, +20 deg and when the characteristics did change the shape essentially. It can be explained by a high value of design kinematic reaction of the tested stage ($\rho = 0.69$). The stator blade row is responsible for a small proportion of the stage pressure ratio. A more detailed explanation of some important phenomena in blading will be performed in the working conditions numbered 1-16 These are marked on the characteristics.

3D FLOW ANALYSIS IN AXIAL-FLOW STAGE BLADING

The aerodynamic loading of the blade row

In this paper we will apply simple criteria of cascade aerodynamic loading, used by compressor designers, in the description of 3D flow
in stage blading including the origin of larger regions of separated flow and rotating stall inception.

For the assessment of the aerodynamic loading of the two-dimensional blade cascade of the compressor type is generally used the definition of the diffusion factor $D$ according to Lieblein (1966). The flow on the suction part of the cascade blade profile is separated, if $D > 0.6$. We confined ourselves to modified definition of the diffusion factor $D$ for the case of compressor stage, where the change of meridional velocity and streamline position in blade row was considered. In the calculation of $D$ factor we used experimental data on 3D flow in measuring planes of tested stage blading. The spanwise distribution of axisymmetric streamlines in annulus area was determined with the use of continuity equation. Large areas of separated flow in the rotor and stator blade rows originated as the diffusion factor $D$ exceeded the critical value $D = 0.6$ within $(1/4 - 1/3)$ of the blade height near one end-wall. This conclusion proceeded from the analysis of a larger set of experimental data obtained in a few compressor stages (Cyrus (1986), Cyrus (1994)).

In some cases we will consider "stalling" incidence spanwise distributions after Miller and Wasdell (1987). This criterion of flow separation origin on the suction blade surface was derived from the compressor cascade data. At high blade stagger angles the method gives unrealistic predictions.

For the prediction of rotating stall onset and maximum loading capacity Koch (1981), Zika (1985) and Schweitzer et al (1984) used an analog of compressor cascade with straight diffuser.

**Stage blading with variable rotor blades - case R**

Figs 4 and 5 show the distributions of the incidence angles along the radii for the rotor and stator blade rows with the basic rotor blades setting ($\Delta \gamma_R = 0$ deg) and with the adjustments of the rotor blades ($\Delta \gamma_R = -20$ deg, $+20$ deg). The curves are plotted for the maximum and

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**Fig. 3b** Characteristics of stage with variable IGVs

**Fig. 3c** Characteristics of stage with variable IGVs and stator vanes

**Fig. 4** Spanwise distributions of rotor incidence angle
In the following text we will deal with the 3D flow in two typical working conditions shown in the field of characteristics (Fig. 3a): condition No. 1 - $\Delta \gamma_R = 0$ deg, the design point ($\phi = 0.59$), condition No. 2 - $\Delta \gamma_R = +10$ deg, near stability limit ($\phi = 0.39$).

Fig. 6 Spanwise distributions of rotor loss coefficient

The rotor blade row. Fig. 6 shows the distribution of the loss coefficient along the height of the rotor blade for two typical working conditions. For the design condition (point No. 1) the loss coefficient reaches, in the middle part of the blade height, values valid for two-dimensional cascades. At the end-walls, the losses increase due to secondary flow. Near the hub the separation of the flow occurs in the corner formed by the suction side of the profile and the hub. The corner area receives the low energy fluid from the inlet boundary layer as a result of pressure gradient between the pressure and suction sides of adjacent blades. Near the casing, the tip clearance flow occurs. This description of 3D flow mechanism is compatible with literature.

Fig. 6 shows also the zone, where all curves of the loss coefficient $\zeta_R$ are located for the working conditions of the stage blading near the...
stability limit (points No 2,4,5,6 and 7 - Fig. 3a). Plotted in the zone is the dependence corresponding to state No 2. The losses are high for both annular walls due to the flow separation on the suction side of the blade profile. Therefore a distortion of the velocity profile occurred in the plane behind this rotor row (Fig. 8). Marked analogically in Fig. 7 is the zone, where the curves of the diffusion factors are located, which are valid for the points near the stability limit. With these distributions, for the height of the blade z/h of minimum z/h = 0.25 - 0.3, the critical value D = 0.6 is exceeded. Consequently, in the rotor row the conditions were created for the origin of a larger area of flow separation which is documented by the measured energy losses of the flow. Similarly, values of incidence angle iR are higher than stalling incidence values iR*. It can be seen e.g. from Fig. 4a for the case of iR* = 0. The stator blade row. Fig. 9 shows the spanwise distributions of stator loss coefficient for three typical working conditions. The loss coefficient was determined from the stationary pressure probes data. The data of 5-hole conical probe, located behind the rotor blade row, were affected by strong flow pulsations resulting from the relative movement of rotor blade wakes and probe sensor. We tried to correct this effect in the evaluation of the stator loss coefficient in the mid-span region. The pressure \( p_{21} \) was determined from the pitchwise total pressure distribution \( p_{21} (r, \phi) \). We supposed that the flow outside blade wakes can be considered as free of energy losses. Details of flow pulsation effects on pressure probe can be found in the work of Cyrus (1988).

In the design condition (point No 1) the energy losses of the flow in the mid-span region are slightly higher than the losses corresponding to the two-dimensional cascades. However, we considered the experimental points whose evaluation respected the effect of the flow pulsations on probe data (flagged circles in Fig. 9). At both end walls the losses increase due to the existence of the separated flow in the corners. The aerodynamic loading of the stator blade elements is relatively high, but subcritical one.

Stage blading with variable IGV - case I

The rotor blade row. In the case of the stage with variable inlet guide vanes, spanwise distribution of rotor incidence angle is plotted for the basic stagger angle of rotor and stator blades and for rearrangement of inlet guide vanes \( \Delta \gamma_{IGV} = -20 \) and +20 deg (Fig. 11). The curves are plotted for maximal and minimal values of the flow coefficient, similarly as for the stage with variable rotor blades (Fig. 4). All curves \( i_{k} \) lie in the marked strip for the stagger angle change \( \Delta \gamma_{IGV} = -30 \) to +50 deg under working conditions at the stability limit. Similarly, the values of diffusion factor \( D_{R} \) are placed in the marked
Case I

Point No. — 11 — 12

Points near stability limits

The critical value of diffusion factor $D = 0.6$ is exceeded in the upper narrow region (Fig. 12). The critical value of diffusion factor $D = 0.6$ is exceeded in the upper

**Fig. 10** Spanwise distributions of stator diffusion factor

**Fig. 11** Spanwise distributions of rotor incidence angle

This is why flow separation on the suction blade surface arises in this region. Distribution of the loss coefficient of the rotor row along the radius for states in vicinity of the stability limit is similar to that of the case R (Fig. 6) and therefore it is not presented.

**The stator blade row.** Distribution of stator incidence angle along the radius is plotted in Fig. 13. It is obvious, that with increasing change of stagger of inlet guide vanes the working zone of incidence angles $i_s$ shifts towards higher values. It means that the aerodynamic loading of the stator row increases. The curves of incidence angle show that the range of flow separation increases with increasing turning $\Delta \gamma_{IGV}$ at points near the stability limit. In the case of $\Delta \gamma_{IGV} = -20$ deg flow is separated only in the upper third of the stator blade near casing. Higher values of incidence angle in this region are caused by flow redistribution in the rotor row as a consequence of separation on the suction blade surface, as was already explained.

**The inlet guide vanes.** Fig. 14 shows dependence of the loss coefficient $\zeta_{IGV}$ on the IGV height for typical values of stagger angle change $\Delta \gamma_{IGV} = 0, 20, 40, \text{ and } 50$ deg. It is evident, that energy losses are relatively low up to the value $\Delta \gamma_{IGV} = 20$ deg. Exceeding this value large flow separation arises on vane surface, which implies also high energy losses. Therefore the stage pressure coefficient $\psi$ does not grow for $\Delta \gamma_{IGV} \leq 30$ deg (Fig. 2), as already mentioned. At the limit value $\Delta \gamma_{IGV} = 50$ deg, however, pressure pulsations of flow with frequency $f = 280$ Hz arise in the stable part of the characteristic. Dimensionless amplitude of static pressure pulsations $A_p/q_m = 0.08$ was found in the plane of the casing (Fig. 1b). However, the flow separation in the IGV has no influence on onset of rotating stall in stage blading. It is connected with the fact that the rearranged inlet guide vanes form turbine type cascades (Fig. 1b). The pressure pulsations arise evidently as a consequence of large region of flow separation on the IGV surface, when vortex systems are formed. A similar phenomenon was observed at the inlet guide vanes of rear stage of axial compressor (Cyrus, 1994).

**Stage blading with variable inlet guide and stator vanes - case I & S.**

Working conditions of rotor blade elements for adjusted stator vanes are not practically influenced by stagger angle change of the stator vanes in the investigated range $\pm 20$ deg. Therefore, the

incidence angles of rotor row $i_r$ for working points near the stability limit lie in the marked hatched region (Fig. 11).

If the stator vanes are turned by $\Delta \gamma_{IGV} = +20$ deg or $-20$ deg, the incidence angle $i_s$ decreases or increases, respectively, approximately by this value along the span when compared with the basic case. In Fig. 13 incidence angles $i_s$ are marked for points No. 15 and 16 at the stability limit. We used marks without connection by curve.

The rotor blade row gives the same total pressure increment for various values of stator vanes stagger. Therefore the differences of blading pressure coefficient $\psi$ are caused by different energy losses in the stator row.
First the aerodynamic characteristics for the IGVs setting angle change $\Delta \gamma_{IGV} = +20$ deg will be discussed. For $\Delta \gamma = +20$ deg the value of pressure coefficient $\psi$ increases or decreases for flow coefficient values $\phi > 0.53$ or $< 0.53$, respectively, (Fig. 3c) when compared with the case of zero stagger angle change of stator vanes $\Delta \gamma_S = 0$ deg. Near the stability limit energy losses in the stator row are higher for the case with the design setting angle ($\Delta \gamma_S = 0$ deg - point No.12) than for the case with restaggered stator vanes ($\Delta \gamma_S = +20$ deg - point No.16) due to higher incidence angles $\gamma_S$ and larger flow separation areas on the vane suction surface. This follows from Fig.13. At higher values of flow coefficient $\psi$ the different stator energy losses can be also explained by various values of stator incidence angles. If the stator vanes are turned by $\Delta \gamma_S = +20$ deg, the incidence angles will be rapidly decreased by the same value. Therefore a larger flow separation area on the stator vane pressure surface will appear in comparison with the case of basic stator vanes stagger angle ($\Delta \gamma_S = 0$ deg).

At the IGVs setting angle change $\Delta \gamma_{IGV} = -20$deg the rapid decrease of pressure coefficient $\psi$ with increasing flow coefficient $\phi$ can be observed in Fig.3c for the case of $\Delta \gamma_S = +20$ deg in comparison with the cases of $\Delta \gamma_S = 0$ and -20 deg. The high total pressure losses in the stator row at $\Delta \gamma_S = +20$deg are caused by a strong flow separation on the vane pressure surface. The stator incidence angles are namely very low. This can be illustrated by some experimental data obtained by means of 3D flow field investigations in stage measuring planes (Fig.1). At working condition $\phi = 0.73$ the value of incidence angle was $\gamma_S = -17$deg at the midspan (Fig.13) for basic stator vanes stagger ($\Delta \gamma_S = 0$ deg).

If the stator vanes were turned by $\Delta \gamma_S = +20$ deg, the incidence angles decreased by the same value ($\gamma_S = -37$ deg.). Similarly, the different values of pressure coefficient $\psi$ for working points No.11 ($\Delta \gamma_S = 0$ deg) and No.15 ($\Delta \gamma_S = -20$ deg) in Fig.3c can be explained with the help of stator incidence angle curves in Fig.13.

Larger areas of stalled flow in the rotor and stator row occur simultaneously at both working states No. 15 and 16 (Fig. 3c), which are in vicinity of the stability limit. This statement follows from experimental data, which were not shown in this paper.

**Nonsteady flows**

After crossing the stability limit in the direction of decreasing volume flow rate a rotating stall was found with typical relative frequency $f_{rot}/f_{rev} = 0.35$. The condition of rotating stall onset in blading of our stage in discussed cases R, I and I&S is simultaneous existence of a larger area of stalled flow on the suction blade surface of rotor and stator rows. It seems, that the main role in investigated blading play the flow conditions in the rotor blade row. In working points at the stability limit, stalled flow arose in the upper half of rotor row in all the three investigated cases. The critical value of the diffusion factor $D_s$ was exceeded in the range of 1/4 - 1/3 of annulus height near the casing.

The extent of stall in the stator row changed with value of adjustment of rotor, stator and inlet vanes. It is possible to estimate it according to distribution of aerodynamic loading criteria for compressor cascades along the radius, e.g. the diffusion factor in our case. The stalled flow in the inlet guide vanes did not influence the onset of rotating stall in stage.

The above mentioned conclusions are in agreement with the results of measurement of three-dimensional flow in one-stage low-speed stages - Cyrus (1986), McDougall. (1990) and Cyrus (1994), as far as the simultaneous existence of larger areas of stall in rotor and stator row of the stage just before the stability loss is concerned.

For prediction of the rotating stall onset there are semiempirc methods employing analogy of compressor cascade with diffuser: Koch (1981), Zia (1985) and Schweitzer et al. (1984). The methods start in a simplified way from the geometry of blade row on the mean diameter. The resulting dependence was obtained by correlating data of several dozens of compressor stages. This enhances practical importance of these procedures. The method devised by Koch (1981) respects the most important quantities influencing the flow stability. The author evaluates stalling pressure rise. He assumes that the rotating stall onset takes place in this point. This, however, is not true...
for some branches of characteristics field of our blading, that have positive slope of tangent at the stability limit without existence of rotating stall. Therefore, we chose the method by Zika (1985) for our purposes, whose assumptions satisfy fully the aerodynamic and geometric parameters of our blading.

\[
\text{INLET GUIDE VANES}
\]

\[
\text{loss coefficient, } \zeta_{\text{IGV}}
\]

Fig. 14 Spanwise distributions of IGVs loss coefficient

\[
\text{Fig. 15 Zika's correlation dependence}
\]

The main correlation dependence of the method is plotted in Fig. 15. Significance of particular parameters is clear from the figure. We have drawn points valid for our blading with variable rotor blades and inlet guide vanes within the range \( \Delta y = \pm 20 \text{ deg} \). These were obtained with the use of measured incidence angles in conditions near stability limits. At extreme values of adjustment \( \Delta y \), no detailed flow fields investigations were performed. Experimental points of our blading lie in a reasonable vicinity of correlation curve.

We add, that the method supposes the decisive role of the rotor row for the rotating stall onset. Therefore the method considers only the geometry of rotor blade element on the mean diameter. Further, it considers favorable influence of positive flow swirl at the stage inlet as a consequence of inlet guide vanes application. It corresponds to the case 1, e.g. with stagger angle change \( \Delta \alpha_{\text{IGV}} = +20 \text{ deg} \) (point in Fig. 15 marked by full square). The Zika's method introduces the correction \( \Delta(L/A_{\text{co}}) = 0.3 \), which shifts the point from the correlation line "b" in Fig. 15. It means that the application of the IGV did not cause the improvement of the working range of the investigated blading, as offered tested method (Zika, 1985).

In future precision of prediction of rotating stall onset in the axial compressor stage using CFD methods can be expected. Recently, theoretical results on the onset of flow instability in an isolated high speed rotor blade row (Ismael and He, 1997) have been published.

CONCLUSIONS

The main results of experimental investigation of aerodynamic performance in low-speed axial compressor stage using variable rotor, inlet guide and stator vanes can be summarized into the following points:

i) by turning rotor blades, it is possible to change more the value of volume flow rate of the stage than by a stagger angle change of inlet guide vanes. The first way of regulation shows high value of isentropic efficiency (90 %) in a large working area.

ii) as a consequence of blades adjustment, blade elements work often in off-design working conditions with flow separation on both suction and pressure sides of blades. The flow character is strictly three-dimensional. Onset of larger stalled flow region in blade row can be predicted fairly well employing simple designer criteria of aerodynamic loading of compressor cascades.

iii) flow separation arose in the upper half of rotor row on the suction side of profile in a close vicinity of the stability limit. Simultaneously, a larger area of separation arose in the stator row. Its extent increased with positive value of stagger angle change of the rotor row (case R) and of inlet guide vanes (case I). It seems that the rotor blade row plays the main role in rotating stall inception in our compressor stage blading.

iv) when the flow rate through stage was regulated by turning of inlet guide and stator vanes (case I&S), the position of the stability limit was not practically influenced by stagger angle change of stator vanes.

v) Zika's correlation method gives reasonable predictions of rotating stall onset in the investigated blading.

vi) pressure pulsations at frequency \( f = 280 \text{ Hz} \) arose for the extreme value of adjustment of inlet guide vanes 50 deg, caused evidently by large flow separation on the vanes surface. This separation did not influence the rotating stall onset in the blading.

In spite of enormous effort the conditions for rotating stall onset in the axial stage of compressor type have not been made fully clear yet. It is therefore necessary to continue in both theoretical and experimental research of axial compressors of different configurations.
ACKNOWLEDGEMENTS

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REFERENCES


APPENDIX - Stage blading parameters

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