SEPARATED FLOWS IN AXIAL FLOW COMPRESSOR WITH VARIABLE STATOR VANES AT POSITIVE INCIDENCE ANGLES

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ABSTRACT

A detailed investigation of three-dimensional flow was carried out in a low speed rear axial compressor stage with the change of the stator blade row setting. The stator blade stagger change was in the range of (-14) - (23) degree. Measurements were performed by means of both stationary and rotating pressure probes at seven working points. The origin of large regions of separated flow in blade rows at positive incidence angles was analysed with the use of the spanwise diffusion factor distribution. These areas in the rotor and stator rows originated as the diffusion factor exceeded the critical value D = 0.6 within (1/4 - 1/3) of the blade height near one end-wall. The rotating stall in compressor stage arose when large regions of separated flow occurred simultaneously in both rotor and stator blade rows.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>AR</td>
<td>aspect ratio $AR = h/c$</td>
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<tr>
<td>A</td>
<td>flow area</td>
</tr>
<tr>
<td>c</td>
<td>blade chord</td>
</tr>
<tr>
<td>$C_p$</td>
<td>relative total pressure coefficient</td>
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<tr>
<td>$C_T$</td>
<td>total pressure coefficient</td>
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<tr>
<td>D</td>
<td>diffusion factor</td>
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<tr>
<td>f</td>
<td>frequency</td>
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<tr>
<td>h</td>
<td>blade height</td>
</tr>
<tr>
<td>i</td>
<td>incidence angle</td>
</tr>
<tr>
<td>m</td>
<td>mass flow rate</td>
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<tr>
<td>P</td>
<td>total pressure</td>
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<tr>
<td>p</td>
<td>static pressure</td>
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<tr>
<td>Q</td>
<td>dynamic head</td>
</tr>
<tr>
<td>$Q_u$</td>
<td>dynamic head based on mean radius wheel</td>
</tr>
<tr>
<td>$Q_w$</td>
<td>flow rate determined from nozzle data</td>
</tr>
<tr>
<td>$s$</td>
<td>blade pitch</td>
</tr>
<tr>
<td>$T_u$</td>
<td>turbulence intensity</td>
</tr>
<tr>
<td>U</td>
<td>wheel speed</td>
</tr>
<tr>
<td>w</td>
<td>velocity</td>
</tr>
<tr>
<td>y</td>
<td>peripheral coordinate</td>
</tr>
<tr>
<td>z</td>
<td>coordinate normal to the end-wall, measured from hub</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>flow angle (from axial)</td>
</tr>
<tr>
<td>$\gamma$</td>
<td>stagger angle (from axial)</td>
</tr>
<tr>
<td>$\nu$</td>
<td>kinematic viscosity</td>
</tr>
<tr>
<td>h</td>
<td>fluid density</td>
</tr>
<tr>
<td>$\eta$</td>
<td>isentropic efficiency</td>
</tr>
<tr>
<td>$\psi$</td>
<td>pressure coefficient $\psi = (P_p - P_i)/Q_u$</td>
</tr>
<tr>
<td>$\phi$</td>
<td>flow coefficient $\phi = Q_u/(AU_n)$</td>
</tr>
</tbody>
</table>
| $\omega$ | loss coefficient: rotor $\omega_r = (P_{r2} - P_{r1})/Q_r$;
|        | stator $\omega_s = (P_{s2} - P_{s1})/Q_s$;
|        | IGV $\omega_{IGV} = (P_{IGV} - P_{IGV1})/Q_{IGV}$ |

SUBSCRIPTS

- ax     | axial |
- d      | design |
- h      | hub |
- IGV    | inlet guide vane |
- m      | at the mid-span |
- n      | nozzle |
- R      | rotor |
- r      | relative to rotor blade row |
- ref    | reference |
- S      | stator |
- t      | tip |
- 0,1,2,3 | planes of compressor stage (Fig.1) |

SUPERSCRIPTS

- $'$ | pitchwise averaged value |
- $\bar{}$ | average over entire measuring plane |
- $i$ | ideal |

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INTRODUCTION

The working range of a multistage axial compressor can be extended by restaggering of stator blades as well as by changing the rotational speed. To calculate the characteristics of the regulated compressor stage is however, not easy. The differences between computation and experiment rise with degree of change of stator blades stagger. This is due to the fact that the blade elements of rotor and stator rows work in off-design conditions. Therefore large regions of separated flow occur in the blade rows. There are several papers dealing with performance prediction of axial compressors working at off-design conditions e.g., Koch and Smith (1976), that support preceding statements.

A 3-D flow in an isolated rotor row in a lower than design flow rate was described e.g. by Dring et al. (1982). A similar paper on a stator row of a two-stage model compressor was presented by Joslyn and Dring (1985). On similar lines a detailed study of the origin of stall in a stator row with untwisted blades was investigated by Gallus et al. (1989). The SVUSS Prague examined the compressor stage with aspect ratio 2.0 over its whole working range (Cyrus, 1986). All indicated studies were concerned with 3-D flow mechanisms exhibiting stall in blade rows at the design stagger angle. Detailed data on the performance of a compressor stage with resettable stator blades are, however, missing in open literature. This led the SVUSS to undertake detailed measurements of flow fields in a real compressor stage with low blade aspect ratio (Cyrus 1988; Cyrus 1990) within IGV and stator blade setting change in range between (-14)° and (+23)°. This paper presents the results found on the performance of individual blade rows with positive incidence angles while special attention is drawn to the origin of stall.

TEST RIG AND MEASUREMENT TECHNIQUE

The low-speed test compressor consists of inlet guide vanes, rotor and stator blades (Fig.1). The outer and inner diameters of the stage are constant. The hub-tip ratio is 0.74. The stage binding has been designed on the assumption of uniform spanwise work distribution. Reynolds number, Re = \( \frac{w r}{\nu} \) was in the range Re = 300000 - 380000 during the flow investigations. Measurements were performed at a shaft speed of 2200 rpm.

### TABLE 1 Design stage geometry at mid-span

<table>
<thead>
<tr>
<th>IGV</th>
<th>Rotor</th>
<th>Stator</th>
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</thead>
<tbody>
<tr>
<td>blades</td>
<td>71</td>
<td>53</td>
</tr>
<tr>
<td>chord (mm)</td>
<td>40</td>
<td>51.9</td>
</tr>
<tr>
<td>camber (deg)</td>
<td>25.1</td>
<td>26.3</td>
</tr>
<tr>
<td>stagger (deg)</td>
<td>13.1</td>
<td>28.0</td>
</tr>
<tr>
<td>solidity</td>
<td>1.21</td>
<td>1.18</td>
</tr>
<tr>
<td>aspect ratio</td>
<td>1.3</td>
<td>1.0</td>
</tr>
<tr>
<td>thickness</td>
<td>0.11</td>
<td>0.082</td>
</tr>
<tr>
<td>chord ratio</td>
<td>0.8</td>
<td>0.5</td>
</tr>
</tbody>
</table>

Experimental investigations were made using a stationary five-hole conical probe, with head diameter of 2.2 mm, in measuring planes 0, 1, 2, 3 (Fig.1). The probe was adjusted to the flow direction. The total temperature was measured in the planes 1 and 2 by thermocouples. A rotating 5-hole conical probe having a head diameter of 2.5 mm was placed in the plane behind the rotor blade row. During operation, the rotating probe was moving peripherally to the rotor blades by means of a specially devised traversing mechanism (Cyrus, 1985). The flow parameters were determined using an indirect method, where the probe was not turned into the flow direction during traversing. At the rotor inlet measurements were also taken of the total pressure of the relative flow using a rotating total pressure probe shown in work (Cyrus, 1988). The measurements were carried out within two blade pitches at 35 to 40 circumferential points and on 11 to 12 radial locations (Fig.1). For close proximity to the end-wall the rotating total pressure probe was not used. The rotating system pressures were converted to the stationary system by a pressure transfer device (Cyrus, 1985).

The flow parameters obtained from the pressure probe and thermocouple data were averaged over a blade pitch. The averaging procedure is described in paper (Cyrus 1985, 1988).

In our investigation we modelled the working conditions of a rear axial compressor stage with the use of special designed screen (Fig.2) and lengthened annulus (Cyrus, 1988). We created inlet velocity profile (Fig. 11) to represent the limit of possible distortion, which may be important in assessment of rear stage performance. The turbulence intensity reached the value of (6-7) % outside the wake of inlet guide vanes as can be seen in Fig.2. It was determined by means of the hot-wire probe.
In order to investigate the three-dimensional flow in our axial compressor stage with inlet flow field modelling it was necessary to connect serially an axial flow fan.

Absolute measurement uncertainty of pressure was ± 0.4 % of dynamic pressure \( Q \), based on wheel speed \( U_w \). The flow angles measured by the stationary and rotating probe were recorded with accuracies of ± 0.7 deg and ± 1 deg respectively. The measurement uncertainty for temperature was estimated at ± 0.3 K.

**DISCUSSION OF THE RESULTS**

Fig. 3 shows the dependence of the relative pressure coefficient \( \Psi / \Psi_t \) on the flow coefficient \( \phi \). It also shows the isentropic efficiency curves. We determined stage characteristics on the basis of the total pressure investigations in the planes 1 and 4. Fig. 3b implies that the stage was working at a high efficiency (88-90) % in a wide operating range. It indicates the points where a detailed flow fields investigation was carried out. Point 1 is a design point. Points 2, 4, 5 and 7 are in close proximity to working stability limit. Points 3 and 6 are in the area of high efficiency values of the stage. In operation state 3 and/or 6 the stage was working with top value of efficiency and/or pressure coefficient. The subsequent text will deal with the analysis of flow in individual blade rows.

**The inlet guide vanes**

Fig. 4 shows the dependence of the incidence angle of inlet guide vanes on the radius for different vanes adjustments \( \Delta y = -8^\circ, 0^\circ, +15^\circ \) and \(+23^\circ\). We have tried to apply Ainley and Mathieson's method (1951) used in the case of turbine cascades so as to determine the origin of stall on the blade profile. In terms of this method the stall should arise on the suction side at \( \Delta y = +15^\circ \) and \(+23^\circ\). Fig. 4 shows the spanwise distribution of \( i^\prime \) for \( \Delta y = +15^\circ \). It follows \( i^\prime < i_{0,0} \). These theoretical conclusions are substantiated by the dependence of the loss coefficient (Fig.5) on the radius. In the case \( \Delta y = +15^\circ \) and \(+23^\circ\) the flow losses are rising near the casing as a result of stall.
The operating conditions of rotor blade elements are described in Fig. 6 which shows the distribution of the incidence angle along the blade height for typical working states. In operation points 1 and 3 in the mid-span the incidence angle values are approaching the reference values of Lieblein (1965). The greatest differences in incidence angles $i - 1$ are found in operating states 4, 5 - for stator blades setting change $\Delta \gamma = +15^\circ$ and $+ 23^\circ$ in the region near the casing. Consequently, most areas of stall can be expected to occur here.

Fig. 7 a, b shows the contours of the $C_p$ coefficient in the plane behind the rotor row for two typical operating states (1, 2 - Fig. 3). In the design value of the flow coefficient $P = 0.74$ (state 1) larger areas of low energy fluid appear in the wake while the top values of the pressure coefficient $C_p$ occur in the corner formed by the suction side of the profile and the hub (Fig. 7a). Here, it comes to the stall resulting in high losses. The corner area receives the low energy fluid from the inlet end-wall boundary layer as a result of pressure gradient between the pressure and suction sides of the adjacent blades. In case of our distorted inlet velocity profile (Fig. 11a) it obviously comes to radial transport of low energy fluid from the corner area to the mid-span as explained in Cyrus's publication (1968). That is why the high loss values are not confined only to the corner area. In addition, it is possible to observe a low total pressure area near the casing resulting from tip clearance flow and relative movement of the end wall and the blade tip.

Fig. 6: Rotor incidence angles along span

Fig. 7: Rotor exit relative total pressure contours
In the operating state 2, which is, with the basic stator blade setting near the stability limit a larger stall area appeared on the suction side in the upper half of the rotor blade. This is evidenced by high values of the Cₚ pressure coefficient (Fig. 7b).

The dependences of the loss coefficient on radius are presented in Fig. 8. Loss coefficient was evaluated on the streamlines of the axisymmetric flow. The radial positions of the streamlines in the planes of the stage were calculated in terms of the continuity equation as valid for different stream tubes. The figure includes the curves of the loss coefficient for states in the vicinity of stage stability limit (states 2, 4, and 5), evaluated from the data of stationary probes. To make the picture clear the coefficient values derived from the data of rotating probes are indicated only for state 2. Besides, the data of the probe of total pressure located before the rotor blade row were corrected for the effect of flow pulsations (Cyrus, 1988). It should be added that in states 4 and 5 it was not possible to evaluate certain points measured in the rotor wake near the casing by the otherwise employed indirect method, as the flow had been strongly separated.

Negative values of loss coefficient evaluated from stationary probes data near hub in Fig. 8 for the state 2 can be explained by measurement errors and the flow unsteadiness effect on pressure probe data. This is supported by values of loss coefficient determined from rotating pressure probes data.

The spanwise loss coefficient distribution for state 3 is practically identical with the distribution valid for state 1, and is therefore not plotted. For state 1 the points of loss coefficient derived from data of the rotating probes are indicated on the assumption of amended effect of flow unsteadiness on probe data in the plane before the rotor row. Similarly, Fig. 8 shows the calculated curve of the loss coefficient for the reference state of the plane cascade according to Lieblein (1966). Obviously, in the midspan region the losses are approximately equal to the double value found in plane cascades in states 1, 3 and 6, when the incidence angles are in the vicinity of reference values. The explanation should be sought in radial transport of low-energy fluid, in our case the distorted inlet velocity field (Cyrus, 1988).

Fig. 8 shows high losses of energy flow in the upper half of the rotor blade resulting from the stall on the suction profile side in off-design states. Particularly extreme values were measured in states 4 and 5. The dependences of the loss coefficient on the radius indicate the 3-D nature of the stall. The cause of high energy flow loss resulting from the stall, whose extent rises from the hub to the casing, is a heavy aerodynamic loading. Its criterion was the diffusion factor D whose spanwise distribution for different states of the turbomachine can be seen in Fig. 9. To compute factor D we employed the flow properties on the streamlines of the axisymmetric flow.

In straight cascades there is marked stall on the blade suction side at positive incidence angles, if the diffusion factor exceeds D = 0.6. In the rotor and stator rows of an SVOSS compressor stage (Cyrus, 1986) there was
a large stall if the critical value $D = 0.6$ had been exceeded in the area where $z > h/3$. This low energy area covered almost half the channel near one end wall. The loss coefficient exceeded in this region the boundary $w = 0.08 - 0.1$, which is approximately the $(4+6)$ multiple of the design value of the loss coefficient. In our rotor blade row the stall origin condition had been fulfilled in all off-design states near stability limit; in states 4 and 5 even significantly. In state 5 the diffusion factor near the casing is as high as $D = 1.0$.

Fig. 10 shows the dependence of the loss coefficient on the incidence angle of the rotor blade element for the mid-span. It indicates the points of the loss coefficient for states with stator blade adjustment $\alpha = 0$, $-8^\circ$ and $+15^\circ$ evaluated from data of rotating probes. The figure also displays the theoretical curve valid for the plane cascade. The theoretical value of the loss coefficient was obtained as the sum of the reference value of Lieblein (1966) and the off-design increment resulting from the plane cascade data as described by Citavy (1986). The theoretical curves of the rotor loss coefficient are also drawn in Fig. 8 for two off-design states 2 and 4. Comparing experiment and computation it transpires that the stall appears in the row blade element at incidence angles higher than that in a plane cascade of the same parameters. This results from different conditions in the inlet plane. In the stage there exist flow pulsations and a higher turbulence intensity compared with the plane cascade aerodynamic tunnel. A favourable factor preventing the stall is a low aspect ratio of the blades in our stage ($AR = 1.0$).

The spanwise distribution of axial velocity in three measurement planes (1, 2, 3) is shown in Fig. 11 for selected operating states (1, 2, 3, 4 and 5). In off-design operating states there was a redistribution of flow in the rotor blade row due to an extensive stall in the upper half of the annulus. The flow through the lower part of the annulus is increased on the expense of the upper part flow. This curbs the growth of the corner stall area near the hub. The distortion of the axial velocity pro-

**Stator blade row**

Fig. 12 shows the incidence angle distribution on the height of the stator blade row in typical operating conditions. It also indicates the curves of the reference incidence angles for different change values of the stator blades setting. In states 1 and 3 the incidence angles in the mid-span are lower than the reference ones. The curves of incidence angles in all states are symmetrical to
the stator row for two typical operating states 1 and 2. Here, another definition of the pressure coefficient was used than in the case of the rotor row, for the peripheral distribution of total pressure in the plane behind the rotor row is irregular due to dissipation of wakes of the inlet guide vanes.

In the stator blade row with the design value of the flow coefficient (state 1) the areas having low pressure coefficient ($C_T = 0.2-0.4$), i.e. high flow energy losses, appear in mid-span region just like in the rotor row (Fig. 11). This is due to radial transport of low-energy fluid from the corners near both end-walls were the stall originates.

Near the stability limit (state 2) an extensive stall arose especially in the upper half of the blade. This is evidenced by the $C_T$ pressure coefficient lines in Fig. 13 b. The low values of the coefficient $C_T = 0.4-0.5$ exist also near the hub. The extent of the stall is however curbed as a result of flow redistribution (Fig. 11).

**Fig. 13:** Stator exit total pressure contours

the mid-span excepting states 4 and 5 when the stage works in the vicinity of stability limit. This is associated with a marked redistribution of the flow resulting from extensive stall on the suction side of the rotor blade row (Fig. 11). In these two cases the incidence angle passes from the region of high positive values to that of the negative values near the hub.

**Fig. 14:** Stator visualization patterns
The above described flow in the stator row is supported by flow visualization results on the suction and pressure side of the blade (Fig. 14). The surface of the blade was covered by a polyethylene tape using a visualization mixture of oil and fine carbon particles. The traces of the flow reveal corner areas with stall. In the design value of the flow coefficient (state 1, Fig. 14 a) the areas are small. In state 2 they are significantly larger. The stall area in the corner is very large near the casing. It begins just on the leading edge; near the hub it appears approximately in the middle of the chord. The basic character of the stall in the stator blade row is similar in the remaining cases, so no other patterns will be presented.

The rise of large regions of stall in the stator row can be followed - the same way as in the rotor row on the spanwise distribution of the diffusion factor D. The curves are displayed in Fig. 15. It can be noticed that the curves of the diffusion factor in the upper half of the span for states near the stability limit lie in a narrow zone for every value of adjustment of the stator blades. The critical value $D = 0.6$ is surpassed in the extent $z/h > 0.25 - 0.33$. The flow is strongly separated as can be seen from the spanwise distributions of the loss coefficient $\omega_s$ in Fig. 16.

Fig. 16 shows the curves of the coefficient $\omega_s$ in whose evaluation the values of the total pressure behind the rotor row $P_\alpha$ were not corrected. To make the figure clear the points $\omega_s$ are marked only for state 1 where the effect of flow pulsations was respected at calculating the pressure $P_\alpha$ in terms of the method indicated in the study of Cyrus (1988). As in the case of the rotor blade row the values of loss coefficient $\omega_s$ were determined on the streamlines of the axisymmetric flow. Fig. 16 shows that in the upper half of the blade the distributions of the loss coefficient in states before the stability limit are very close to each other. This is in agreement with the spanwise distribution of the diffusion factor $D_\alpha$ (Fig. 15). High losses indicate the existence of a large stall regions in the corners.

The stage efficiency

With the basic stagger of stator blades and with the design value of the flow coefficient (state 1) the incidence angles in mid-span region of the rotor and stator blade rows are in the vicinity of reference values. It is alike in state 3 with rotor blade row ($\Delta \gamma = +15^\circ$) while the stage operates with maximum efficiency. In the stator row the incidence angles are in the area of negative values, the loss coefficient attaining values close to the minimum value. In both states 1 and 3 the efficiency reaches the top figures $\eta = (89 - 90)\%$ - Fig. 1b.
In working state 6 (\(\Delta \gamma = -8^\circ\)) with the maximum value of the pressure coefficient, the efficiency of the stage is lower \(\eta = 86\%\), since the difference between the incidence angles (\(\gamma_1\ldots\)) in both rows attains the values \(7^\circ \sim 9^\circ\). The losses, particularly in the stator blade row, are then higher than minimum.

The stage reached the lowest measured values of efficiency in states close to the stability limit where there are large stall areas in the blade rows accompanied with high flow losses and high energy losses (states 2, 4, 5 and 7). The efficiency ranged within \(\eta = (66 - 73)\%\). The efficiency of the stage also in the stator blade row. In the rotor bladings (Cyrus, 1993) the flow regularity non-steady flow phenomena of the rotor and stator blade rows, especially in positive sense, close to the stability limit. Higher aerodynamic losses have developed, which cannot survive in a state with design blade stagger. The above described results led to an acceptable hypothesis that the non-steady flow arises whenever conditions for the appearance of large regions of separated flow have been fulfilled simultaneously in the rotor and stator blade rows of compressor stage.

**CONCLUSION**

The main result of the investigation of 3-D flow in an axial compressor stage after change of stator blades setting within \(-14^\circ\) to \(+23^\circ\) can be summarized in the subsequent points:

1. The efficiency of the compressor stage attained high values \(\eta = (88 - 90)\%\) over a large working range.
2. On the suction side of the inlet guide vanes, a stall after adjustment \(\Delta \gamma = +15\^\circ\) and \(+23^\circ\).
3. Close to the stability limit a stall developed in the upper half of the rotor blades and near both end-walls of the stator blades for all examined values of stator blades adjustment, which was evidenced by high losses. The highest loss was found in the rotor blade row near casing \(w = 0.6\) for change of stator blades setting \(\Delta \gamma = +23^\circ\).
4. To rate the stall development on the suction side of blade profile at positive incidence angles a criterion can be used of cascade aerodynamic loading the diffusion factor. The large stall areas 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.
5. The nonsteady flow of rotating stall type in the compressor stage developed after simultaneous fulfillment of conditions valid for rise of large stall areas both in the rotor and stator rows.
6. The blade row stall at a positive incidence angle has a strictly 3-D character. That is why the prediction methods used for plane cascades cannot give satisfactory results in determining the aerodynamic performance of blade elements in high off-design conditions. Therefore empirical correlations should be applied.

The research of the 3-D flow in compressor stage blade rows should continue so as to be able to find reliable blade element performance computing methods to predict the characteristics of axial compressor in their off-design states. The results obtained in single compressor stages should be analysed in the case of their application in multistage turbocompressor computations.
REFERENCES


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