ROTATING STALL IN A SINGLE STAGE AXIAL FLOW COMPRESSOR

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Abstract
This paper describes the results on an experimental investigation of rotating stall flow inside a single stage axial flow compressor. Tests were carried out in two steps. First, measurements were taken to investigate the transition process into rotating stall. The compressor starts into rotating stall via the "modal route" with a single rotating stall cell. Further throttling yields to a two-cell shape followed by a significant outlet pressure drop. Both transition processes are discussed in detail. Results from the Moore-Greitzer theory are compared with measured data.

In a second step measurements were taken to determine the three-dimensional unsteady structure of a full developed rotating-stall cell. Based on unsteady total pressure and three-dimensional hot-wire data, the structure of a rotating stall cell could be resolved in detail upstream and downstream the rotor. A typical part-span stall was found. By inserting the measured data into the Euler-equations convective and unsteady effects on the pressure fluctuations can be isolated. A dependence between the radial flow inside the stall cell and the unsteady flow accelerations was found.

Nomenclature
\( c_r \) radial velocity
\( c_{\varphi} \) circumferential velocity
\( c_t \) axial velocity
\( p \) static pressure
\( p_t \) total pressure
\( r \) radius
\( t \) time
\( U_m \) mean rotor speed at midspan
\( U_{hw} \) hot-wire anemometer output voltage
\( \rho \) density
\( \varphi \) mass flow rate ( \( c_r / U \) )
\( \psi_t \) total pressure coefficient ( \( 2\Delta p/\rho U^2 \) )

\( \omega \) vortex vector
LE leading edge of the stall cell
TE trailing edge of the stall cell

Index
* dimensionless value

Introduction
Compressor performance is limited by many factors, e.g. system-instabilities, rotating stall and surge. Although rotating stall was first observed by the group developing centrifugal compressors for the Whittle turbojet in 1938 (see Chesire L.J.) it is yet not fully understood. In the past a rotating-stall cell was assumed to consist of a region of very low or zero through-flow velocity surrounded on both sides by streamlines of un-stalled flow (Emmons et al. 1955). Day (1976) did a remarkable step towards a better understanding of a rotating-stall cell describing the stall cell to be in mass exchange with the surrounding flow.

Within the last 20 years there have been many experimental investigations on stalled compressors (Tanaka and Murata 1975, Day and Cumpsty 1978, Breugelmans et al. 1983 and Giannissis et al. 1988). The results indicate that the structure of the rotating-stall cell has many different appearances. Rotating stall is classified in part-span and full-span stall. Both types of rotating stall vary in a wide range according to the compressor used for the investigations.

The research groups concentrated their main effort on the detection of stall precursor events. They want build up an active control system (Day 1991 and Paduano et al. 1991) or a long term monitoring system to avoid rotating stall (Honen and Gallus 1993). Nevertheless, there is a need to understand the basic effects taking place inside a rotating stall cell.

The experimental investigations described in this paper were carried out in a single stage axial flow research compressor. The
compressor shows a typical part-span rotating stall. One purpose of this work is to study the mechanisms leading to rotating stall. Another purpose is the investigation of the detailed structure of a full developed stall cell. Detailed measurements were taken upstream and downstream the rotor (see plane 1 to 3 in Fig 2). With the acquired data it is possible to describe the structure of a part-span rotating stall cell.

FIG. 1: TEST COMPRESSOR

FIG. 2: CROSS SECTION THROUGH THE BLADING

TABLE 1: OPERATING POINTS AND CORRESPONDING STALL CELL CONFIGURATIONS

Experimental Facility and Techniques of Measurement

Ambient air enters the inlet pipe of the research compressor 9.5 m ahead of the rotor (Fig. 1). A venturi nozzle, which is located 1.5 m upstream the test section, measures the mass flow. A five hundred kW motor with speed control drives the rotor. The tests were performed from 2000 to 6000 rpm. Figure 2 gives additional information of the compressor. The observed effects are similar within the adjusted speeds. Results obtained at 3000 rpm are shown and discussed in this paper. Table 1 shows a summary of the operating points. More data is available on request from the authors.

Single hot-wire, triple-hot-wire and unsteady pneumatic pressure probes were used. An eight-channel transient-recorder linked to the processing computer digitized and sampled the data. Accuracy obtained is estimated to be about 2 percent in the un stalled flow and about 5 percent inside a stall cell.

Three problems had to be solved for a successful data acquisition and data evaluation. First it is not possible to operate the compressor for a longer time in the rotating stall regime. To perform a measurement the compressor had to be driven into rotating stall. The throttle was closed continuously. To acquire one data set took about five seconds. Afterwards the compressor was set to unstalled stable flow to avoid blade damage. As a stall-cell signal is not clear enough to serve as a high quality trigger signal (phase errors of about ten percent and more) the data was acquired without any trigger. The starting phase angle of a stall-cell signal is therefore arbitrary. A single hot-wire sensor was fixed near the casing at rotor inlet. This sensor gave a reference signal for all measurements taken within this investigation. The second problem is the flow angle variation at the boundaries of a stall cell. It is about three times larger than the range of the calibration of the hot-wire probes used. The flanks of the stall-cell signal are not regular. They have a small variation with time, which causes the third problem. The stall cell interacts with the rotor blades. The interaction between stall cell and rotor blade is large if the number of rotor blades is small. The different relative positions of rotor blades and stall cell cause a variable stall-cell size when a stall cell passes a stationary sensor. All problems could be solved by modifying the ensemble average method. Because of the complexity of this method we do not report about it here.

Stall inception

Figure 3 shows the compressor pressure rise characteristic at 3000 rpm. At the inception point of rotating stall the characteristic is of flat top type. When passing the compressor stability line, the stalling events can be classified into five different periods. The upper left time trace in Fig. 3 was taken by a triple hot wire, located at 10 percent span downstream the rotor. At the right hand the same signal is shown. The specific events are enlarged.

First the operating point of the highly throttled compressor is stable. The rotor blades are severely stalled at the tip. Close to the hub the flow is attached. For this case Fig. 4 shows the turbulent kinetic energy downstream the rotor. A tip leakage vortex cannot be seen. At all other operating points with higher flow rate the tip leakage vortex was visible. The contours in Fig. 4 indicate a typical blade stall. Reducing the mass flow rate the flow becomes unstable (period 2 in Fig. 3). Small periodic waves appear. They are amplified for about 20 revolutions. Finally a stable single-cell rotating stall is established (period 3,
1. STABLE ROTOR FLOW
2. ONSET OF ROTATING STALL
3. STABLE ONE CELL STALL
4. TRANSITION INTO THE TWO CELL SHAPE
5. STABLE TWO CELL STALL

FIG. 3: INCEPTION OF ROTATING STALL

\( \Psi_0 = 0.9 \). The single-cell stall collapses when the flow rate \( \phi \) is equal to 0.51 (period 4). A two-cell rotating stall appears. The pressure ratio is reduced and the compressor operates on its rotating stall characteristic (period 5, \( \Psi_0 = 0.7 \)).

To determine the number of stall cells three hot-wire sensors were placed around circumference. From the measured phase difference of the travelling stall cells, the number of stall cells is evaluated.

FIG. 4: TURBULENT KINETIC ENERGY NEAR STALL

Detailed hot-wire measurements upstream and downstream the rotor were performed to detect the spatial location of the stall inception. All sensors recorded simultaneously the onset of the periodic velocity fluctuations (period 2 in Fig. 3). No spatial location of the preferred stall inception was found. A periodic velocity perturbation is observed, as it is predicted by the theory of Moore and Greitzer (1986). When the compressor operates on its stability line, this perturbation grows without any changes in phase and frequency.

The tests of the stall inception have been repeated. It is
possible to estimate the process of a stall cell from the measured real-time traces (Fig. 5) (see also Garnier et al. 1991). Figure 6 shows a set of experimental results in comparison with a calculation based on the Moore-Greitzer theory. The nondimensionalized amplitude of a stall cell is plotted versus time. Experiment and theory are in good accordance. Figure 7 displays results from the application of the theory. The onset of stall could be modeled quite well. The calculated stall cell frequency fits the measured frequency within a range of ten percent. The parameters used for this calculation are listed in Fig. 6. The theory gave results for the one-cell configuration only. It was not possible to obtain reasonable results from the calculation for the two-cell rotating stall configuration.
The single-cell rotating stall is stable in a small range of compressor operation. If the mass flow rate is further decreased a stable two-cell rotating stall is formed. Figure 8 shows a characteristic time trace of two seconds. The corresponding enlargement is shown below. The dashed pointed box in the upper figure marks the time of transition. At both sides of this box either the single-cell or the two-cell form is stable. At the top of the figure the arrows mark the location of the peaks belonging to a single-cell form. The dashed lines mark the events of the two-cell stall. With time the circumferential size of the single-cell rotating stall are damped. The emerging cell of the two-cell stall is visible as a sharp peak. About two and a half cell rotations later a complementary stall cell appears. The growing process of both cells takes about seven to ten cell revolutions.

A stable single cell rotating stall can only be achieved in a small range of compressor operation. Stall events were recorded at 25 spanwise radial positions. It was not possible to set the operating point with the necessary repeating accuracy for all radial positions. If the stability point of the single-cell rotating stall was not found correctly the stall cells are unstable. We observed by detailed real-time data analysis a phase-shift in the single-cell signal. Due to an unknown distortion a two-cell stall is formed and damps the single cell rotating stall (Fig. 9). The two-cell type is not stable. A new single-cell rotating stall appears. The phase difference between the old and the new single-cell stall was found to be arbitrary. This procedure takes place within 10 to 30 rotor revolutions. The arrows indicate the position of the peaks of the single-cell stall signals (Fig. 9). They are traced from the left for the old stall cell. For the new stall cell the arrows are traced back from the right where the new stall cell is stable. Above the arrows a cross marks whether a cell was found or not. The position of the arrows relative to each other shows the phase shift. Inside the transition region an intermittent two-cell rotating stall is formed. The data shown here indicates on the left and on the right hand of the time window stable single-cell rotating stall. In the transition region the amplitudes of both single-stall cells were recorded. They coexist for this time. A simple model shown in Fig. 10 illustrates this process.

The timetrace in Fig. 9 was selected to demonstrate the phase shift of the single cell shape. The transition zone is quite small and noisy. Nevertheless, the two cell shape could be stable for a longer term. Then this signal triggers an oscilloscope. Fig. 11 shows a series of FFT-spectras. The FFT data originate from one single time trace (digitized into 32768 data words). Thus it is possible to calculate 32 instantaneous FFT spectras with a length of 512 frequency points each. This corresponds to 1024 points in the time domain. The lower spectrum displays the corresponding 32-average FFT-spectrum. Both frequencies, one-cell and two-cell rotating stall, are detected. The upper spectras are instantaneous FFT spectras. The reason of the presence of both frequencies in the averaged spectrum becomes obvious. The one cell stall frequency is the dominant frequency in the 17th and 19th spectrum. The 18th spectrum indicates the existence of a two cell rotating stall while the one cell shape is less pronounced.

**Structure of a Stall Cell**

To investigate the detailed structure of the different rotating stall patterns radial measurements were carried out upstream and...
downstream the rotor. The relative velocity vectors for both stall cell types are shown at five spanwise locations in Fig. 12 and Fig. 13 respectively. The profiles represent the complete blade row. In Fig. 12 the wavy nature of the single-cell stall is clearly visible. The length of the relative velocity vectors is harmonic with circumference at the tip. No flow angle variations take place. At midspan only a minor flow unsteadyness was measured. Near the hub there is a slight increase of the velocity fluctuations. Figure 13 emphasizes the flow inside a two-cell configuration. Inside a stall cell the flow close to the rotor tip is reversed. At the trailing edge of the stall cell the flow recovers. The axial velocity rises but the velocity vectors are overturned. The recovery process is finished when the outlet flow angle reaches design angle. Then the flow decelerates continuously until flow reversal takes place inside the next stall cell. This mechanism can be followed from tip to midspan. At the hub the flow accelerates inside the stall cell. Between two stall cells the velocity becomes very small.

When the rotor operates stable at $\varphi \geq 0.53$ a strong secondary flow vortex exists close to the hub. This vortex rolls up in the corner between rotor blade suction-side and rotor hub. Therefore, no corner separation is observed but strong flow diffusion takes place (Poensgen 1991). This effect explains the low through flow near the hub between two stall events.

Figure 14 shows the axial velocity distribution downstream the rotor. The data shown here is a view from downstream. In the absolute frame of reference the stall cells rotate in clockwise direction. To give an orientation the circumferential locations are labeled like a compass. Reversed flow occurs near the casing (east and west direction) while the through flow rate at the hub is larger than design. Figure 15 displays for the same stall cells...
the distribution of the circumferential velocity. Comparing both figures the shape of the stall cell is different. The axial velocity distribution shows part-span stall, but the circumferential velocity distribution indicates "full-span" stall. The trailing edges of both stall cells are located horizontally (west-east axis). Starting north and following the contours in the rotational sense the circumferential velocities are increasing. This indicates a large blade loading. Inside the stall cell there is a drop of the circumferential velocity which signifies the collapse of the profile flow.

Figure 16 summarizes the effects downstream the rotor. The stall cell leading edge is not a sharp line as observed in compressors operating in full span stall. The axial flow in the annulus can be classified in fully reversed, partially reversed, unstalled, accelerated and reentered flow. The reversed flow inside the stall cell leaves the rotor at the front. Here the circumferential velocity of the reversed flow is larger then the speed of blade rotation. Therefore, the stalled flow reenters the blade passage at the stall cell's trailing edge.

Figure 17 illustrates the contours of the total pressure upstream the rotor in the absolute frame of reference. The areas of reversed flow are located near casing. Here the absolute circumferential velocity is larger then blade speed. These large circumferential velocities produce high total pressures. Near midspan a total pressure drop is observed. This pressure loss is produced when the reversed axial velocity and the incoming velocity are dissipated to 100 percent near casing. Figure 18 outlines this process. The area of the mixing process is cut by the plane of probe traversing. Due to the blockage of the reversed flow the incoming flow is forced towards the hub. The low momentum flow is transported along the radial shifted streamlines into the plane of measurement. Therefore, the measured total pressure drop results from transport of low momentum fluid. Figure 19 shows the total pressure contours upstream the rotor in the stall cell frame of reference. Two areas of low total pressure exist inside the stall cell. They are marked by the shadowed areas. In the outer area flow reversal takes place. The low total pressures near midspan are caused by flow mixing. If the flow upstream the rotor is assumed do be irrotational then the total pressure relative to the stall cell is constant. As Fig. 19 indicates, the flow upstream the rotor is not irrotational because the plane of measurement is not far enough upstream. It cuts the zone of flow mixing.

Figure 20 shows enlarged the circumferential velocities upstream the rotor inside the stall cell. The corresponding secondary flow vectors are superimposed. In this case secondary flow is defined to be the deviation from the axial inlet flow. These vectors denote a strong vortex whose vector is normal to the blade suction side. Its center is near midspan. This vortex can be divided into its circumferential and axial components. The influence of its circumferential component is sketched in Fig. 21. Inside the rotor blade passage the flow separates. This separated flow is centrifuged and forced to the leading edge by the adverse pressure gradient. The incoming flow is blocked at the casing. It moves further towards the hub. The influence of the axial component of the vortex vector yields to high
circumferential velocities of the reversed flow. Near the hub the flow enters the rotor blade passage with a positive incidence (Fig. 22). Strong radial flow was detected. The radial velocities are within a range from -40 percent to 40 percent of the axial velocity.

Evaluating the unsteady pressure measurements upstream (also downstream) the rotor, the radial equilibrium is satisfied in the undisturbed regions between the two stall cells. The meridional streamline curvature was neglected in this estimate. Inside a stall cell the simplified radial equilibrium was not fulfilled either downstream or upstream the rotor. This indicates unsteady flow due to violation of the radial pressure balance.

To get a better understanding of the balancing forces inside the stall cell the circumferential component of the Euler equation was evaluated:

\[
\frac{\partial c_x}{\partial t} + c_r \frac{\partial c_x}{\partial r} + c_r \frac{\partial c_x}{\partial \phi} + c_r \frac{\partial c_x}{\partial z} + c_x \frac{\partial c_x}{\partial r} = -\frac{\partial p}{\rho \partial \phi}
\]

Except the axial derivation of the circumferential velocity all terms can be calculated from the measurements. In a first and rough estimation this term is neglected. Assuming the measured data to be homogeneous with circumference the unsteady term \( \frac{\partial c_x}{\partial t} \) can be calculated because all the other terms are known. The momentum equation is non-dimensionalized by the mean radius \( r \) and the rotational speed \( u \) at midspan. Results of this calculation are shown in Figs. 23 and 24. The contours of the
unsteady velocity distribution \( \frac{\partial c}{\partial t} \) and the radial convected momentum \( c_r \frac{\partial c}{\partial r} \) are shown. Both contours are nearly identical. To estimate the influence of these terms the maximum and minimum values are listed in Table 2. Each term of eq. 1 fluctuates between these limiting values. According to Table 2 the highest flow accelerations occur due to the term \( c_r \frac{\partial c}{\partial r} \), which is practically balanced by \( \frac{\partial c}{\partial t} \) in front of the rotor. The influence of the circumferential pressure gradient is smaller by a factor of four. The convective acceleration in the circumferential direction \( c_r \frac{\partial c}{\partial r} \) is about 15 times smaller then the unsteady term. \( c_r \frac{\partial c}{\partial r} \) can be neglected upstream and downstream the rotor.

![FIG. 23: FLOW ACCELERATION \( \frac{\partial c}{\partial t} \)](image)

Further throttling yields to a stable two-cell-stall shape. The two corresponding transients, from undisturbed flow into the single-cell form and the transition into a two-cell rotating stall, are physically different. In the first process the compressor stall comes up via the "modal route" (long length scale disturbance). A global small velocity perturbation is amplified. It forms a stable single-cell-stall configuration. The transition into the two-cell shape follows the short length scale model (Emmons 1955 and Day 1991). Small and sharp stall cells are detected. With time they grow in amplitude and circumferential size. Both types of flow disturbances (small and long scale disturbances) exist together for a short time.

As the relative velocity vectors demonstrate in Fig. 12 and Fig. 13 the compressor has two different types of stall cells. The single-cell configuration is of periodic wave type. No severe flow reversal is observed. The losses produced by the stall cell are low. The compressor still operates near peak pressure. When the compressor operates in two-cell rotating stall regime, large flow reversal is observed near the rotor tip. At the rotor inlet the reversed axial velocities reach up to \(-20\) m/s which is about half the undisturbed inlet velocity. The reversed flow produces a large area of mixing upstream the rotor generating high mixing losses. This leads to a lower total pressure as measured at the rotor inlet. The compressor operates at its stall characteristic with a lower pressure ratio.

Real-time analysis of the single-cell stall data showed an intermittent stalling behaviour of the compressor. In a period of 10 to 100 rotor revolutions, the single-cell configuration collapses and a two-cell stall is formed. This configuration is not stable and another single-cell stall is generated. All configurations coexist over a short time. But the difference in the phase between both stall cells was found to be arbitrary. It can be concluded that several waves may initially coexist travelling around the annulus. In the investigated compressor only one of them is amplified and

### TABLE 2: NON-DIMENSIONAL PEAK VALUES OF THE TERMS OF EQU.1

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<thead>
<tr>
<th>Value</th>
<th>Upstream</th>
<th>Downstream</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \frac{\partial c}{\partial t} )</td>
<td>-16.8</td>
<td>3.6</td>
</tr>
<tr>
<td>( c_r \frac{\partial c}{\partial r} )</td>
<td>-3.6</td>
<td>16.8</td>
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<tr>
<td>( \frac{\partial c}{\partial \varphi} )</td>
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<td>1.1</td>
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<tr>
<td>( \frac{\partial c}{\partial \varphi} )</td>
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<td>0.20</td>
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<tr>
<td>( \frac{\partial p}{\partial r} )</td>
<td>-4.0</td>
<td>2.5</td>
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**Discussion and Conclusions**

- The transition into rotating stall occurs within two steps. The first stall configuration originates from a global perturbation.
yields to a full developed rotating stall. In some data sets of stall inception this effect was also observed to occur in the pre-stall wave. 

- The highest values of the time dependent flow acceleration are detected in front of the rotor. The unsteady term of the inviscid circumferential momentum equation is practically balanced by the second term on the left side of equation 1. This indicates that the major flow unsteadiness occurs where radial flow is present in combination with a gradient of the circumferential velocity in the radial direction. This effect can also be observed downstream the rotor. The flow unsteadiness is lower there.

- It was not possible to evaluate the radial component of the inviscid momentum vector. The radial velocity could not be measured with the accuracy need to form the term \( c \frac{dc_\theta}{dr} \). The magnitude of the radial velocities was found to be very sensitive with the point of compressor operation. The circumferential derivatives of the velocity and the radial gradient of the circumferential velocity are much less sensitive. Only values obtained from the circumferential component of the Euler equation gave useful results.

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