PREDICTION AND MEASUREMENT OF ROTATING STALL CELLS IN AN AXIAL COMPRESSOR

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ABSTRACT

This paper presents a parallel numerical and experimental study of rotating stall cells in an axial compressor. Based on previous theoretical and experimental studies stressing the importance of fluid inertia and momentum exchange mechanisms in rotating stall, a numerical simulation using the Euler equations is conducted. Unsteady 2-D solutions of rotating stall behavior are obtained in a one-stage low subsonic axial compressor. The structure and speed of propagation of one fully developed rotating stall cell together with its associated unsteady static pressure and throughflow field distributions are presented. The numerical capture of a stalled flow region starting from a stable high-flow operating point with an axisymmetric flow distribution and evolving at a reduced mass flow operating point into a rotating stall pattern is also discussed.

The experimental data (flow visualization, time-averaged and unsteady row-by-row static pressure measurements) acquired in a four-stage water model of a subsonic axial compressor covers a complete characteristic line ranging from high mass flow in the stable regime to zero throughflow. Stall inception is presented together with clearly marked different operating zones within the unstable regime. For one operating point in the unstable regime, the speed of propagation of the cell as well as the static pressure spikes at the front and rear boundaries of the rotating stall cell are compared between computations, measurements and an idealized theory based on momentum exchange between blade rows entering and leaving the stalled cell. In addition, the time-evolution of the pressure trace at the rotor/stator interface is presented.

This study seems to support the assumption that the cell structure and general mechanism of full-span rotating stall propagation are essentially governed by inertial effects and momentum exchange between the sound and stalled flow at the cell edges.

NOMENCLATURE

Abbreviations
CFD computational fluid dynamics
IGV inlet guide vanes
RS rotating stall
SLOR successive line over-relaxation
stg stagnation point

Symbols
A apparent fluid inertia ratio, Eq. (4)
C absolute velocity
cx axial velocity
ΔCp pressure coefficient
ΔCp_ref pressure coefficient
h static enthalpy
M fluid mass contained in cascade channels
m mass flow through surface S
m, normalized mass flow
n normal vector
p static pressure
pa normalized static pressure
p0 stagnation pressure
S surface
T stagnation temperature
T rotor period
T<sub>RS</sub> rotating stall period
t time
t<sub>A</sub> time at CFD occurrence of compact cell
t<sub>B</sub> time at CFD occurrence of diffused cell
t<sub>C</sub> time occurrence during stall generation
U<sub>2</sub> rotational speed at Euler radius

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I. INTRODUCTION

It is well known that the stability of any compression system is affected as the mass flow rate is reduced from its nominal value beyond the so-called limit of dynamic stability. Close to this operating point large scale disturbances such as surge occur and rotating stall begins to appear in the compression system setting a performance boundary. Surge-like disturbances are primarily axi-symmetric and involve primarily variations in the mass flow through all the compression components. Rotating stall-like disturbances, on the other hand, show static pressure fields decaying away from the compressor and involve primarily circumferential flow distortions within the compressor itself. These produce a locally reduced throughflow in part of the compressor annulus while keeping the average mass flow and pressure rise fluctuations substantially below the levels encountered during surge. Rotating stall is characterized by cells of stalled fluid (with negative or nearly zero throughflow velocity) rotating around the annulus at a fraction of the rotor wheel speed. The scope of this paper is twofold. Firstly, to analyze the capability of a numerical procedure for unsteady inviscid flows to capture and sustain a physically meaningful fully developed rotating stall pattern. Secondly, to provide additional experimental data on rotating stall for a broad range of throughflows using a low-speed water model of a subsonic axial compressor.

Historically the analysis of the flow field beyond the stability limit in axial compressors has been based mainly on experimental observations and studies, see for example (Cumpsty, 1989). Also some approximate theoretical models derived from these observations are available, see for example the review of Longley (1993). Among them the model developed by Greitzer & Moore (1986) considers the stability of the whole compression system in which surge and rotating stall appear as eigenmodes.

While it is nowadays standard practice to use Computational Fluid Dynamics (CFD) tools to predict the flow within the stable operating range of a compressor, the CFD approach still needs to establish itself as a sound prediction method for operation in the unstable regime. To a lesser degree, rotating stall inception has been recently numerically investigated (Hendricks et al. 1996), (He, 1997). Propagating stall in isolated linear cascades using vortex methods has been studied, see for example (Speziale et al., 1986), (Nishizawa & Takata, 1994), and stall behavior in a 2-D rotor-stator system has been reported by Outa, Kato & Chiba (1994). The latter study focussed essentially on compressor performance and stall effect on blade loading. In this frame of research, numerical solutions for the flow in a one-stage axial compressor are presented and compared with experimental data for operation in the rotating stall regime. This study focuses on full-span, fully developed rotating stall with the intent of providing more insight into the mechanism for the propagation of the rotating stall cell. This is achieved by analyzing the circumferential variations in static pressure and throughflow velocity produced by a single rotating stall cell. The CFD simulation presents a 2-D inviscid unsteady solution for a one-stage low subsonic axial compressor bearing similarity with any of the stages of the four-stage water model of a subsonic axial compressor (Hof et al., 1996). The use of the Euler equations for capturing and propagating rotating stall cells is prompted by previous experimental and theoretical studies, (Cumpsty & Greitzer, 1981), (Longley, 1993), (Gyarmathy, 1996), which clearly indicate the crucial impact of fluid inertia and momentum exchange between the rotor and the stator rows for stall cell propagation. Furthermore, as shown in external and internal aerodynamics, (Rizzi, 1984), (Felici, 1992), the Euler equations allow the generation and capture of strong vortical flows as encountered during rotating stall. Because of the experimentally observed essentially axial-tangential nature of full-span rotating stall in low aspect ratio bladings, a 2-D numerical study is performed.

While the study of stall inception is of primary importance to dynamically control rotating stall and surge in order to extend the stable flow range of the compression system, see for example (Garnier et al., 1991), (Day, 1993), (Gysling & Greitzer, 1994), (Tryfonidis et al., 1994) there is still a need to understand the rotating stall dynamics as it often precedes surge. Hence, the intent here is rather to focus on the phenomenological understanding of rotating stall from a fluid dynamic point of view.

The water model permits an improved visualization compared to a similar air compressor, while retaining identical overall characteristic features. Compared to the air compressor, the water model presents a wider range of operation in the rotating stall regime without the appearance of surge, since no mass storage effects are present in this discharge system.

Based on specific parameters defining the stall cell such as cell width, speed of propagation and static pressure spikes at the cell edges, a cross comparison between the CFD prediction, a theory based on momentum exchange between the rotor and stator rows as well as experimentally...
obtained stall patterns is performed. The capture of the rotating stall pattern as part of the CFD solution by “jumping” from a stable operating point with an axisymmetric flow distribution and a constant throughflow to a reduced mass flow with a large circumferential disturbance travelling at reduced wheel speed is discussed. The analysis of stall inception is not an issue in this work. Due to the absence of viscous and 3-D effects in the numerical simulation, emphasis is given to the general mechanism of stall cell propagation and overall cell structure while overlooking detailed flow features.

II. MEASUREMENTS IN A FOUR-STAGE COMPRESSOR WATER MODEL

In an earlier investigation on centrifugal low-speed compressor rotating stall, a water model of the ETH Turbomachinery Laboratory single-stage centrifugal air compressor (using the principle of hydrodynamic analogy) has recently successfully been brought into operation (File et al., 1997). The aim was to study rotating stall through visualization (Gyarmathy et al., 1997). From the observed speed and structure of the single rotating stall cell, an idealized model for the propagation of a full-span rotating cell has been developed (Gyarmathy, 1996).

Based on this previous work, a water model of a non-existing four-stage low speed subsonic axial compressor has been constructed for studying stall patterns. In this Section, the water model and the experimental results are presented, comprising the test rig facility, the compressor characteristics, the RS inception process and the annulus blockage during RS for a broad range of throughflow coefficients. Furthermore, first results of flow visualization are shown.

II.1 Experimental facility

The closed loop water circuit with the whole “compression” (i.e., discharge) system is shown in Figure 1. The major components of the system are the axial compressor, the deaeration valve at the top, the throttle valve and the Venturi tube for mass flow measurements. Downstream of the throttle valve and upstream of the compressor inlet, flow straighteners and a stagnation chamber ensure axial inlet flow into the compressor’s IGV. Note that the loop does not provide for any elements allowing pneumatic storage effects, hence mild or deep surge in the compression system are excluded. Thus RS can be studied in the absence of these compressible effects.

The model, shown in Figure 2, consists in a four-stage axial compressor built with variable inlet guide vanes and repeating stages. All nine rows comprise 30 blades each. The design values include a hub flow coefficient of 0.45, a midspan degree of reaction of 61% and a midspan rotor speed of 2.1 m/s. The hub/tip radius ratio is 0.82 and the solidity of the stator and rotor is 1.39.

Figure 2: Axial compressor water model (cylindrical rotor in transparent cylindrical casing).

II.2 Instrumentation

Figure 3 presents a schematic of the four-stage axial compressor. Time-mean and unsteady static pressure measurements are taken at the compressor inlet, outlet and in-between the blade rows using static pressure tabs and fast-response pressure transducers (500Hz), see markers 1-10 in Figure 3. An additional tab is placed upstream of the IGV (marker 0). The reference total pressure for the compressor is obtained from three Pitot probes equally distributed over the circumference and located upstream of the IGV. The time-averaged pressures are obtained by measuring the pressure difference between the wall static taps and the reference inlet total pressure. Hydrostatic effects are compensated. In order to better study the RS effects, the blade passing frequency is filtered out in the measured static pressure history plots.

Figure 3: Positions of measurements in axial compressor water model.
The flow visualizations are obtained by using polystyrene spheres with a diameter between 100 and 400 µm and a specific density of 1.05 kg/m³. Photographs from a CCD video camera (CF 16/2) are processed by digitalization and imaging software.

II.3 Compressor characteristic

In Figure 4, the averaged characteristic of the four stages is shown for a rotor speed of 120 rpm. Due to incompressibility and negligible Reynolds number effects the characteristic remains constant for different rotor speeds. In 1978 Day et al. proposed to divide a stall characteristic in two categories (Day et al., 1978). In "progressive stall", part-span stall is observed first, while in the "abrupt stall" characteristic full-span stall appears. The water model shows an abrupt behavior since after stall inception a full-span single cell RS is observed. The performance map for the entire range of flow coefficients can be structured into five different domains:

Region I includes the design point and is governed by stable flow from \( \bar{\varphi} = 0.55 \) to \( \bar{\varphi} = 0.358 \), the latter corresponding to the instability limit.

Region II (\( \bar{\varphi} = 0.358 \) to 0.31) covers the sudden jump to Region III when RS develops. No persistent operation is possible here. When the flow is increased to make RS disappear the jump is shifted to Region II*. A detailed description of such hysteresis is given by Day et al. (1978).

In Region III (\( \bar{\varphi} = 0.31 \) to 0.067) the compressor is operating in a single-cell full-span RS regime. The static enthalpy rise of the CFD solution for the single-stage compressor at RS conditions is also marked in Figure 4. As expected, due to the 2D inviscid nature of the CFD solution, the static enthalpy rise is higher than the measured data.

Region IV (\( \bar{\varphi} = 0.067 \) to 0.044) contains a second jump (and hysteresis) where the flow changes from RS operation to axisymmetric stall, in which a nearly 100% blockage of the annulus is observed.

Region V denotes an axisymmetric fully stalled flow over the entire annulus. This branch is similar to the unstalled characteristic branch, where the rise in static pressure is increasing when closing the throttle valve. Visualization of this flow indicates that the structure of the full stall flow is similar in nature to the flow within the RS cell.

II.4 Rotating stall inception

Currently, rotating stall inception of an air compressor is divided into spike-type stall inception and stall inception due to modal activity, as discussed in Camp & Day (1997). The spike-type disturbances are characterized by sharp peaks in the history plot, which rotate with about 70% to 80% of rotor speed. Modal perturbations, on the other hand, revolve slowly up to 50% of rotor speed and appear as gentle waves. In Figure 5 the history plot of the pressure transducer at position 5 during RS inception at a constant throttle valve position is shown. The roman numerals (I to III) correspond to the regions defined in Figure 4. Pre-stall spikes propagating at about 74% of rotor speed are observed for about 6 rotor revolutions before the emergence of a fully developed full-span single cell RS propagating at 54% of rotor speed. Notice that the RS frequency varies from 53% to 56% of speed in Region III.

The increase in time-averaged static pressure from sound flow to RS observed in Figure 5 can be explained from the implementation of the data acquisition system. The reference pressure for the static pressure transducer is the environmental air pressure. The rise in pressure across the stages from positions 6 to 10 (Figure 3) is subtracted from the geodesic height from position 6 up to the deaeration valve on top of the test rig. Consequently, the drop in pressure from sound to stall flow is measured as a pressure rise in Figure 5 during RS inception.

II.5 Annulus blockage

In (Day et al., 1978) a prediction of compressor performance is presented. It is suggested, that for abrupt stall characteristics a minimal blockage factor \( \lambda \) of 0.3 is necessary and that in case of lower blockage factors, i.e. higher throughflows, part-span stall would appear. This minimum blockage value at RS inception has then to be found in our water model, since no part-span stall has been observed. This is verified in Figure 6 where the experimental blockage factor \( \lambda \) is plotted versus the flow coefficient. The blockage ranges from 0.3 at RS inception to 0.85 where full annulus stall is developed.

The experimental blockage was obtained from the wall pressure traces as

\[
\lambda = \frac{\text{time between suction peak and next pressure peak}}{\text{time between two suction peaks}}
\]
As expected and described in (Day et al., 1978), the blockage factor is a linear function of the flow coefficient. Following Day, it can also be calculated from the performance map as

\[ \lambda_b = \frac{\Phi_{\text{unstalled}} - \Phi_{RS}}{\Phi_{\text{unstalled}}} \]  

(2)

This correlation obtained from the characteristic in Figure 4 is shown in Figure 6 together with the measured data. Clearly the measured blockage of the annulus follows this model for Region III. In addition, the computed result is marked, and will be discussed later in Section IV.

The shift between \( \lambda \) and \( \lambda_b \) is due to the use of slightly different definitions.

![Figure 6: Blockage factor as a function of flow coefficient.](Image)

**II.6 Unsteady static pressure measurements**

The structure of the RS cell for two flow coefficient values as measured at two axial positions in the water model are discussed in this Section. Measured time traces of casing wall static pressure in the interspace rotor/stator for high and low \( \Phi \) within Region III (Fig. 4) at positions 6 and 8 are shown in Fig. 7a, b, c & d, respectively. For \( \Phi = 0.31 \), the pressure traces show a single RS cell propagating at 54% of rotor speed and blocking about 33% of the annulus. The cell front and rear boundaries are associated with low and high pressure peaks, respectively. As explained later in Section IV these peaks mark the momentum exchange between the sound flow and the stagnant flow in the RS cell. Compared to the abrupt pressure change within the RS cell, a mild circumferential pressure gradient is present in the sound flow region.

At a low flow coefficient of \( \Phi = 0.075 \) the cell covers 79% of the annulus and propagates at 55% of rotor speed. Within the small portion of sound flow remaining present in the annulus the static pressure gradient is large but nearly constant, while within the RS the pressure is increasing from the front to the rear boundaries.

The pressure distribution within the RS cell itself seems to depend on the size of the cell, i.e., on the flow coefficient. The RS cell structures in the different stages are similar in nature regarding the cell speed and blockage factor. An increase in the amplitude of the pressure spikes at the cell front and rear boundaries is observed from the first to the last stage.

![Figure 7: Wall pressure trace between rotor and stator of](Image)

a) stage 3 (pos. 6) during RS operation at \( \Phi = 0.31 \)
b) stage 4 (pos. 8) during RS operation at \( \Phi = 0.31 \)
c) stage 3 (pos. 6) during RS operation at \( \Phi = 0.075 \)
d) stage 4 (pos. 8) during RS operation at \( \Phi = 0.075 \).
III. NUMERICAL PROCEDURE

The numerical procedure uses a multi-block grid generator MELLIP (Saxer-Felici, 1996) operating in conjunction with an unstructured flow solver called MULTI2 (Saxer, 1992).

III.1 Grid generator and flow solver

MELLIP solves the Poisson equations in two dimensions in the axial-tangential plane for different spanwise locations (here at the Euler radius only). An iterative SLOR technique is applied on blocks of C and H grid types which boundaries are allowed to move, resulting in an overall unstructured smooth grid. The source terms are calibrated to control spacing and orthogonality at the blade surfaces (Steger & Sorenson, 1979). MULTI2 solves the time-dependent Euler equations with an explicit, finite-volume, node-based Lax-Wendroff type algorithm (Ni, 1981) which has been extended to 3-D unstructured meshes and hexahedral cells (Saxer, 1992). For this particular study, the 3-D solver is used in 2-D mode in which three 2-D meshes are piled up to form a two-cell-height 3-D control volume. MULTI2 is capable of solving the time-dependent as well as the steady-state interaction in a rotor/stator configuration. A combined second- and fourth-difference numerical smoothing consistent with the second-order accuracy (both in space and time) of the discretisation scheme is added to prevent high frequency oscillations in the solution and to capture shock waves. The fourth-difference smoothing, an extension of the method proposed by Holmes & Connell (1989), ensures second-order accuracy in shock-free regions even on distorted grids, a desirable feature when studying the evolution of flow distortions or secondary flows. A non-linear second-difference operator allows to capture shocks with an artificial bulk viscosity parameter tailored by the local flow divergence and Mach number in order to avoid large shock overshoots. Due to the low subsonic case studied here, the second-difference smoothing has been turned off.

III.2 Computational domain and boundary conditions

In order to avoid excessive computational time, and in contrast to the experimental domain, the CFD work was performed on a single compressor stage. Also, the circumferential domain determined by the boundary conditions has been reduced to 15 blades (test rig compressor: 30 blades). Long inlet and exit domains were specified for enabling time-independent inlet and exit boundary conditions to be used despite the large blockage to be expected by rotating stall. The computational grid is presented in Figure 8 with the location of the rotor inlet, rotor/stator interface and stator exit (stations 1, 2 and 3, respectively). The block-structure can be seen in a blow-up of the near-blade region. The blades disposition used in the simulation corresponds to a cut at the (RMS) Euler radius of any of the repeating stages of the four-stage water model described in Section II and in (Hof et. al., 1996). The grid in domain 1 to 2 is fixed to the moving rotor blades and the Euler equations are solved in the relative frame of reference. In domain 2 to 3 the grid is stationary and absolute flow variables are used. With respect to the overall goal of assessing the determining effects in the propagation of full-span rotating stall, the grid resolution, which is standard for an Euler simulation, seems adequate. As described in detail in Saxer & Giles (1993) and Saxer (1992) non-reflecting boundary conditions are used. By adopting long inlet and exit farfields and by applying steady-state non-reflecting boundary conditions based on circumferentially averaged quantities at the inlet and the exit, any uncertainty in modeling the physically correct unsteadiness reaching the farfield boundaries should be minimized. At the rotor inlet, the average entropy (or stagnation pressure), stagnation temperature, and tangential flow angle are set in the absolute frame of reference and correspond to $\rho_0 = 1$ bar, $T_0 = 309°K$, $\alpha_1 = -31.3°$. This is in contrast to calculations performed exclusively in the stable branch of the characteristic, where it is standard practice to set the rothalpy and relative flow angles at the rotor inlet. The circumferentially averaged

![Figure 8: Computational grid (15 channels, 3 mesh planes, 38475 nodes per mesh plane) with stations 1, 2 and 3 and blow-up of near-blade region.](http://gasturbinespower.asmedigitalcollection.asme.org/GT/proceedings-pdf/GT1998/78620/V001T01A020/2410101/v001t01a020-98-gt-067.pdf)
conditions allow local flow adjustments due to potential effects. At the exit, the average static pressure is prescribed in order to reach the desired throughflow coefficient. At the rotor/stator interface an unsteady numerically non-reflecting procedure based on the local characteristics variables ensures a physically consistent boundary condition, see Saxer (1992), even in the presence of strong backflow.

A mass flux condition is enforced at the pseudo hub and tip endwalls as well as on the stator and rotor blades. Also, the repeating, i.e. periodic condition is applied between the upper and lower boundaries of the multiple blade passages domain.

III.3 Validity of the computational solution

As MULTI2 has not been specifically designed for the low Mach number flows encountered in this study, i.e. inlet axial Mach number of 0.11 at stall conditions, its capability is additionally tested against an incompressible solution. Figure 9 shows the computed and analytical pressure coefficient distributions on the blade in the Gostelow cascade, (Gostelow, 1984) where a conformal mapping transformation has been used to derive the 2D analytic incompressible solution. The computed maximum Mach number is approximately 0.182, so that the compressibility effects are very small. The agreement is very good.

![Figure 9](https://example.com/fig9.png)

Figure 9: Computed and analytical pressure coefficient distributions on the blade in the Gostelow cascade.

Another important issue is the time-accuracy of the computed unsteady solution and its repeatability under stall conditions. The time traces of the inlet, rotor exit, stator inlet and exit surface-integrated mass flows are shown in Figure 10 for twelve rotor revolutions in the stall regime. For each station, a periodicity in mass flow fluctuations is observed. Notice that the rotor exit and stator inlet curves overlap, showing the conservation of mass across the interface. The blade passing frequency is also clearly visible. The phase shifts and amplitude differences are due to compressibility effects expressed by

$$\int \frac{\partial P}{\partial t} dV$$

This value is indicated on Figure 10 for $V = V_{2,3}$ (computational volume from interface to stator exit). Inlet and exit mass flow fluctuations of about 15% of the mean value indicate significant surge-like pulsations superimposed upon the rotating stall phenomenon.

The instantaneous error in the continuity equation applied on the computational domain lies below 1% of the inlet mass flow.

![Figure 10](https://example.com/fig10.png)

Figure 10: Normalized mass flow iteration history during rotating stall, plotted for inlet (1), interface (2) and exit (3) surfaces.

III.4 Rotating stall generation

Two approaches have been tested to produce rotating stall starting from the axisymmetric unsteady solution in the unstalled branch of the characteristic. These are schematically depicted in Figure 11. Both of them lead to the same rotating stall pattern at the same throughflow conditions.

![Figure 11](https://example.com/fig11.png)

Figure 11: Two ways of generating rotating stall (round symbol) from stable CFD solutions (squares).

As starting point, the first technique uses the model of Day et al. (1978), which assumes that the axial velocity in the unstalled region is determined by the requirement for the unstalled section of the compressor to give the same pressure rise as in the stalled region. Then from the unstalled operating point, the rotor-relative inlet flow angle is abruptly increased to an average value corresponding to stall conditions. Once the stall process has developed, it is maintained by setting the inlet flow angle back to its constant absolute level, as given by the inlet guide vane. The second approach starts from the unstalled branch near the peak of the characteristic. With the inlet conditions fixed by the absolute-frame stagnation conditions and flow angle, converged solutions are obtained when increasing the static pressure stepwise. When the instability limit is reached, the solution begins to show a breakdown of the static pressure rise (stronger rise of the inlet pressure...
than the step imposed in the exit pressure). The pressure fluctuations exponentially grow over the whole flow field. This is linked to the generation of several recirculation regions within the stator passages during the first 20 rotor revolutions. These zones will evolve later into a single rotating stall cell. In order to get a stable periodic unsteady solution, the exit pressure has to be manually reduced to an appropriate value. This value can be estimated from experimentally measured stage performance characteristics or from the first approach. As an example, Figure 12 shows the static pressure time trace for the numerical probe #1 at the rotor/stator interface and illustrates the stall generation process in this second approach. The last upward step in exit pressure was taken at time $t/T = -10$, the manual reduction at $t/T = 10$.

IV. ANALYSIS OF STALL FLOW

For the 15 blade passages shown in Figure 8, the numerical solution presents a single stall cell at a throughflow coefficient $\phi$ of 0.25. The flow structure associated with the rotating stall is analyzed in this Section by using unsteady static pressure and throughflow distributions as well as flow visualization. At the interface between rotor and stator a comparison of numerical results with unsteady static pressure measurements at the same $\phi$ is shown. An estimate of the RS speed of propagation is conducted using the analytical model of Gyarmathy (1996) and compared to the experimental and numerical data.

IV.1 RS cell structure

The unsteady axial velocity fields, the static pressure fields and the instantaneous streamlines are shown in Figure 14a, b, c for two times ($t_A$ and $t_B$). This distinction is rendered necessary by the presence of significant non-damped mass flow fluctuations in the CFD solution. At time $t_A$ the mass flow through the stage is high, at $t_B$ it is low (see Figure 10). The top fields of Figure 14 show the instantaneous axial velocity contours (where $c_x < 0$) and the rotor/stator interface $c_x$ profile over the periphery. The middle fields (b) display the static pressure. For visualization and interpretation purposes a frame of reference rotating with the calculated cell speed is used. Hence the RS cell appears to be stationary while the rotor row is moving downward and the stator row upward.

In the numerical solution a single cell is observed propagating at a fluctuating velocity ranging from 61% to 66%. Associated with this change in propagation speed is a change in the structure of the cell, which evolves from a diffused to a compact form and back with a frequency dictated by the surge-like fluctuations shown in Figure 10. At time $t_A$ a compact RS cell is observed with strong backflow near its center and negative velocities of the order of the throughflow velocity. Larger regions of backflow are present in the rotor than in the stator, with a strong recirculation bubble forming ahead of the rotor (Figure 14c). This deflects the rotor-relative incoming flow to the sides of the RS cell. Looking at the static pressure field one notices that viewed from the RS cell, a drop in pressure occurs at the front (bottom side) boundary of the cell, while a pressure maximum ends it. The width of the cell as defined by the circumferential extent between the front and rear boundary peaks in static pressure (see lines F and R) corresponds to 33% of the periphery. On the other hand, the circumferential static pressure remains nearly constant in the sound flow region (except of course variations due to blade potential effects). The center of the large vortical bubble is marked by a depression. The width of the RS cell can also be calculated from the peripheral distance formed by the bisection of the overall mean throughflow and the local axial flow distribution as shown on Figure 14a.
Figure 14: a) Contours of negative axial velocity, b) static pressure contours, c) instantaneous cell-frame streamlines, for times $t=t_A$ and $t=t_B$ (rotating stall frame of reference). For a) profiles of axial velocity and for b) profiles of static pressure along the rotor/stator interface are also shown ($\phi=0.25$).
At time \( t_B \) a less compact, larger, RS cell is observed (Figure 14c) with near zero throughflow. While the RS cell trailing edge is clearly marked by an abrupt change in static pressure, the cell leading edge is somewhat diffused, with smoother changes in static pressure. The sound flow presents a "ramp" distribution of static pressure along the circumference.

The unsteady static pressure recordings of five numerical probes located at the stator/rotor interface for twelve rotor revolutions are shown in Figure 15 together with the fixed probes location. The times \( t_A \) and \( t_B \) corresponding to the Figure 14 are marked. A clear rotating stall pattern is observed with a speed of propagation (measured from peak-to-peak) of about 64%. The cell in its compact form seems to propagate slightly faster than in its diffused form. The evolution from diffused back to compact is marked by a "bump" in static pressure occurring between peaks. Its frequency is about half of the cell propagation speed and is linked to the overall mass flow pulsations present in the compression system, as seen in Figure 10. In particular, a positive mass flow pulsation tends to render the cell more compact, while a negative pulsation tends to diffuse it. The blockage varies from 31% to 46% as measured in Fig. 14. The unsteady static pressure fluctuations ahead of the rotor are shown in Figure 16. The pressure trace of probe #5, for example, is fairly constant except for one short pressure drop in each cell revolution. This drop in static pressure corresponds to the vortical bubble generated ahead of the rotor in Figure 14c.

**IV.2 Comparison with idealized model**

In (Gyarmathy, 1996), a simple model to estimate the RS cell speed of propagation and the pressure spikes at the front and rear boundaries of the rotating stall cell has been derived. By considering the rotor and stator blade channels as one-dimensional counterrotating ducts exchanging momentum across the interspace, an expression for the \( \Omega_{RS} \) ratio of RS cell speed to rotor speed is obtained in terms of geometrical data. The model assumes an incompressible inviscid flow in a cascade of infinite number of blades and narrow channels without any diffusion. Viewed in the frame of reference of the rotating stall cell this process appears steady. At the front (lower) boundary of the cell the stator channels containing high velocity sound flow and the rotor channels with stalled flow steadily meet. The opposite effect takes place at the rear boundary, where stalled stator fluid emerging from the RS cell meets sound rotor fluid. The overall effect is a short duration momentum exchange between the fluid masses transported by the two rows. The net result is a depression at the front boundary, where the stator fluid decelerates, while a static pressure peak is produced at the rear boundary where the rotor flow is decelerated. This general behavior is clearly seen in Figure 14a, where the front (F) and rear (R) boundaries of the RS cell are marked. The actual local pressure gradient and peaks depend on the abruptness of the momentum exchange. At the front boundary of the cell the driving mechanism is the transfer of the stator fluid momentum by suction. At the cell rear boundary, the transfer of the rotor fluid momentum towards the stator occurs through a jet-type effect. Hence, the rear boundary is defined by a steeper pressure gradient than the front boundary, as also seen in the numerical results in Figure 14.

According to the analysis of the momentum exchange process at the cell boundaries, the following expression is found for the normalized RS cell speed of propagation.

\[
\Omega_{RS} = \frac{\omega_{RS}}{\omega} = \frac{1}{1 + A}
\]

(3)
where
\[ A = \mu \frac{\sin^2 \beta_m}{\sin^2 \alpha_m} \quad (4), \quad \mu = \frac{M_S}{M_R} \equiv \frac{x_S}{x_R} \quad (5) \]

\( A \) represents the relative inertia of the fluid contained in the stator and rotor channels. \( \mu \) is formed by the ratio of the circumferential mass entrained by the stator and rotor bladings, respectively. Here the blade channel flow angles \( \alpha_m \) and \( \beta_m \) are approximated by the rotor and stator stagger angles \( (51.2° \text{ and } 37.1°) \), respectively. According to the above formula, the cell rotational speed is 56% of the rotor speed, which compares to 54% to 56% in the experiments and to 64% for the computed solution. A possible explanation for the higher value found in the CFD solution is given by the fact that in the computed solution, the actual mass entrained through the stall cell is larger on the rotor side than on the stator side, see Figure 14. Thus \( \mu \) in the above Equation is effectively reduced bringing the corrected idealized model value to 65% for \( \mu = 0.9 \) (instead of \( \mu = 1.3 \)).

IV.3 Comparison with experimental data

All measurements in the four-stage water model and the computations in the one-stage compressor at a given flow coefficient of \( \varphi = 0.25 \), show a fully developed single RS cell. As seen in Figure 6, the computed blockage closely complies with the experimental curve. The normalized unsteady static pressure fluctuations measured at position 8, (i.e., in the rotor/stator interspace of the fourth stage), is shown in Figure 17 for twelve rotor revolutions. The shape of the pressure traces can be directly compared to the computed traces in Figure 15, for example to that of probe #5 (or any other one). Apart from the slightly higher frequency of the CFD trace, the resemblance is conspicuous. For example, the abrupt pressure rise at the cell rear boundary is present both in the experimental and computed solution. This has also been observed in (Cumpsty & Greitzer, 1981). Also the more gentle static pressure decrease in the sound flow region is captured numerically. The absence of a flat plateau effect in the circumferential pressure distribution within the sound flow suggests that the RS cell acts as an obstruction like a cylindrical solid body placed in

![Figure 17: Wall pressure trace between stage 4 rotor and stator (pos. 8) during rotating stall operation at \( \varphi = 0.25 \).](image)

![Figure 18: Photograph of flow around stator trailing edge in stage 4 at \( \varphi = 0.25 \) (rotor speed of 60 rpm).](image)

![Figure 19: Computed instantaneous streamlines shown in absolute frame of reference at times \( t_A \) and \( t_B \) and \( \varphi = 0.25 \).](image)
a potential flow producing a stagnation effect, which deflects the incoming flow, as originally postulated by Emmons (1955). Hence ahead of the RS cell (in the circumferential direction), the cascade operates at incidences lower than nominal values, while behind the RS cell trailing edge the incidence of the sound flow is larger, see Figure 14c. The pressure bump present before every second pressure peak may be the result of an interference between acoustic pressure waves and the stall cell. Some weak signals of “bump” effects are also present in the experimental data. However, the modulation of the RS phenomenon at $\Omega_{stg}/2$ frequency as observed in the (slightly compressible) CFD solution is absent in the water experiments. 

The amplitude of the measured pressure spikes (from 0.3 to 0.5) seen in Figure 17 closely corresponds to the idealized value of 0.45 estimated from the idealized model. The momentum exchange in the CFD solution is seen to be stronger (0.6 to 1). 

A photograph of vortices formed near the trailing edge of the fourth-stage stator row is shown on Figure 18 for $\bar{\phi} = 0.25$. 

In Figure 19 the computed instantaneous streamlines, viewed in the fixed stator frame of reference, are plotted at the same flow coefficient and times $t_A$ and $t_B$. One should note that Figure 19 shows the same flow domain as Figure 14c, however, the instantaneous streamlines are plotted in different frames of reference. 

Stator trailing edge vortices, due to a slight back-flow pulse, clearly appear in the CFD solution. The stator vortices and the almost tangential flow motion behind the stator are also seen in the experiment (Figure 18). 

V. SUMMARY AND CONCLUSIONS 

A parallel numerical and experimental study of rotating stall in an axial compressor has been presented. The experimental work is based on measurements obtained in a four-stage water model of a subsonic compressor (repeating stages). The advantages of using the hydrodynamic analogy are: 
- improved visualization capabilities due to low shaft speed  
- the possibility of studying RS for a broad range of throughflows from stall inception to zero throughflow without surge since no mass storage effects are present in the compression system. 

The major experimental results are summarized as follows. From high to low flow coefficient ($0.55 > \bar{\phi} > 0.067$) the water model presents a characteristic typical for low-speed axial compressors with an unstalled branch, a hysteresis, and a rotating stall branch. For $\bar{\phi} < 0.067$ a second hysteresis is observed followed by a branch of negative slope near zero and at negative throughflow. In the rotating stall branch, a single fully-developed full-span stall cell is observed. Its speed of propagation ranges from 54% to 56% of the shaft speed. Consistently with previous experimental data and Day’s correlation for subsonic axial compressors, the blockage in the RS regime increases linearly from $\lambda = 0.3$ to $\lambda = 0.85$ with decreasing throughflow. Stall inception is of spike-type with pre-stall cells rotating at about 74% rotor speed. The shape of the pressure traces changes strongly as the blockage is increased. 

The CFD simulation presents a 2-D inviscid unsteady solution in the RS regime for a single-stage air compressor geometrically similar to the water model. Two ways of numerically generating RS have been described and lead to the same RS pattern. A numerically sustainable stall flow solution which includes one fully developed RS cell has been presented. Due to periodic low frequency mass flow pulsations associated with the axial length of the computational domain, the cell structure is not invariant in time. Its width (or blockage) varies periodically between 31% to 46%, and its speed of propagation varies from 61% to 66% of the rotor speed. Viewing CFD results in the frame of reference of the RS cell is particularly interesting to gain insight in the problem. 

The cell propagation speeds observed in the experiments and CFD can be explained with the help of an idealized theory based on momentum exchange between the rotor and stator rows at the RS cell front and rear boundaries. 

Due to differences between experimental and simulated compression systems (# of stages and medium), a strict quantitative comparison is not appropriate. In spite of this, the computed RS pattern (cell propagation speed, blockage, circumferential static pressure distribution and blade channel vortex formation) is consistent with the experimental data. Consequently it appears that the driving mechanisms for single cell full-span RS are essentially of 2-D inertial nature. 

However, in order to further substantiate the present approach, future work ought to step-wise remove the simplifications assumed in the current CFD model (2-D to 3-D, inviscid to viscous, single stage to multistage), thus revealing each contribution to the RS phenomenon. 

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