Impeller Rotating Stall as a Trigger for the Transition from Mild to Deep Surge in a Subsonic Centrifugal Compressor

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ABSTRACT
The present paper concerns the transition from mild to deep surge in a single stage centrifugal compressor using a vaned diffuser. Time-resolved measurements of the mass flow rate and the static pressures at various locations of the compressor were analyzed for different diffuser geometries and different operating points in the compressor map.

When the throttle valve was gradually closed at an impeller tip Mach number (Mu) above 0.4, the compressor showed first mild and then deep surge whereas at Mu=0.4 rotating stall was the dominant instability. This single-cell rotating stall was identified to be caused by the impeller.

During mild surge at higher Mach numbers the instantaneous flow and pressure traces showed that the overall flow at the stage inlet intermittently dropped below the critical value associated with the occurrence of impeller rotating stall. Rotating stall appeared for a while but vanished as soon as the flow increased again. With further throttling, however, a threshold was reached at which rotating stall triggered deep surge. The results show that triggering only occurred if the flow deficiency causing rotating stall persisted long enough to permit the stall cell to make at least one or two revolutions.

INTRODUCTION
About a decade ago, in a comprehensive review of the instability phenomena associated with compressor systems Greitzer (1981) discussed the observations and hypotheses concerned with centrifugal compressor stages. The existence of low-frequency flow oscillations in the system, termed "mild surge", had already been recognized as a key factor in triggering large-amplitude oscillations involving backward flow through the stage, or "deep surge" (Toyama et al., 1977). However the question was still open which element of the stage was responsible for flow break-down and by what physical mechanism. For moderate to high pressure centrifugal compressors usually the vaned diffuser was made responsible for the break-down, but it was open to debate if and when diffuser rotating stall played a role. Since that time important refinements have been made.

Recent investigations in turbomachinery show the potential of active control systems for surge suppression and hence for operating range extension (Epstein et al., 1989, Ffowcs Williams and Huang, 1989, Gysling et al., 1990, Pinsley et al., 1990, Simon et al., 1992, Ffowcs Williams et al., 1992). Although preliminary tests in laboratories and recently in an auxiliary power unit (Ffowcs Williams et al., 1992) are encouraging, detailed studies of the effects leading to aerodynamic instabilities are still of interest. Simon et al. (1992) show how active control applied to compression systems promises to be a practical technique, provided the B-parameter as defined by Greitzer (Greitzer, 1981) is small and the compressor slope not steep. The only exception is a close-coupled valve as actuator which means however always a loss in efficiency.) Centrifugal compressors for industrial application are usually associated with greater values for B. The latter authors further elucidate that the ability to control the system depends on the disturbance driving the system to instability. Therefore an improvement of the operating range...
without knowledge of the type and behavior of the unstable flow is difficult to realize.

Different factors can be responsible for the transition from mild to deep surge. Toyama et al. (1977) for example suppose that a critical value for the diffuser-inlet recovery is reached and thereby the flow break-down triggered. Fink et al. (1991) postulate that the mild surge fluctuations influence a stationary impeller stalling caused by the tongue of the scroll. The reduction in pressure rise promotes surge.

Although several authors believe that rotating stall cannot be the major mechanism for the beginning of surge due to the greatly differing frequencies between rotating stall and surge, it is well known that the loss of pressure rise during rotating stall can cause overall flow reversal (Emmons et al., 1959, Rodgers, 1978).

In this paper it will be shown how intermittent mass flow reduction during mild surge provokes impeller rotating stall and then triggers the transition to deep surge. If the time during impeller stall is long enough the loss of pressure rise allows overall flow reversal and hence deep surge is initiated. While operation in deep surge is strictly prohibited in many applications an excursion into a mild surge region for a short time (e.g. during transient processes) is tolerable as long as no deep surge follows. Therefore the excited rotating stall has a key function in setting the limits of useful operating range.

**NOMENCLATURE**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>L</td>
<td>diffuser channel length</td>
</tr>
<tr>
<td>m</td>
<td>mass flow rate</td>
</tr>
<tr>
<td>n_vane</td>
<td>number of diffuser vanes</td>
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<tr>
<td>p°</td>
<td>stagnation pressure</td>
</tr>
<tr>
<td>r</td>
<td>radius</td>
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<td>u_2</td>
<td>impeller tip speed</td>
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<tr>
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<td>flow coefficient</td>
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<tr>
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<tr>
<td>η</td>
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<td>ρ</td>
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<tr>
<td>ω</td>
<td>angular velocity</td>
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**Subscripts**

1. impeller inlet
2. impeller outlet
3. diffuser channel leading edge
4. diffuser channel exit
c. diffuser throat

**TEST RIG AND INSTRUMENTATION**

**Test Rig**

Measurements were carried out on a single-stage centrifugal compressor using air. A sketch of the test rig is shown in Fig 1. It is configured either as a closed loop or as an open suction line. During operation in the closed loop mode the pressure in the suction pipe was set to about atmospheric pressure. The impeller is driven by a 440 kW DC motor coupled to a two-stage gear box. The maximum rotational speed is limited by the shaft seal to 22,000 rpm. The air enters axially (through a flow straightener and a screen mounted in a long suction pipe),
passes through the impeller and the vaned radial diffuser and is discharged through a pipe toward the heat exchanger.

In these investigations an unshrouded industrial centrifugal impeller from aluminium was used (see Fig 2). The features of the impeller are listed below:

- inducer tip diameter: \(d_1\) 175 mm
- exit diameter: \(d_2 = 2r_2\) 280 mm
- exit height: \(b\) 17 mm
- exit blade angle 60° relative to the tangential direction (30° back lean)
- number of main blades: 11
- number of splitter blades: 11
- blade clearance width: 0.6 mm

Several vaned diffusers were used in this programme. The diffuser width \(b\) was constant and equal to \(b/d_2 = 0.06\). The vanes were simple circular arc profiles of a chord length of 96.5 mm and of constant thickness (3.1 mm) with their rounded leading edges at a radius of \(r/r_2 = 1.16\). The vanes could be set at different angles. In these investigations the camberline angle at diffuser channel entry was set to \(\alpha_3 = 15°, 25°\) or 30° relative to the tangential direction. The number of vanes was usually 24 but the removal of every second vane allowed to study the influence of a reduced diffuser solidity.

The main geometric data as the area ratio (AR), the length to width ratio (LWR) and the aspect ratio (AS) are listed in Table I. The narrowest location is taken as diffuser channel entrance.

<table>
<thead>
<tr>
<th>(n_{\text{vane}})</th>
<th>(\alpha_3)</th>
<th>AR</th>
<th>LWR</th>
<th>AS</th>
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<td>1.79</td>
<td>3.5</td>
<td>0.77</td>
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</table>

Tab. 1: Diffuser geometries used

**Instrumentation**

Conventional wall pressure taps and temperature probes in the suction pipe and in the outlet tube were used to determine the overall performance of the stage. The time mean mass flow rate was measured by a standard nozzle and is therefore estimated to be accurate to within 1.0 percent. The rotational speed of the impeller was sensed by an inductance probe near the gear box shaft. All signals (pressure transducers, temperature sensors, inductance probe etc.) were automatically monitored and converted to physical units by a Micro Vax computer.

Many instantaneous measurement devices were used in the test rig. The location of these devices is indicated in Fig. 3. High frequency response wall pressure transducers mounted in the diffuser entry region (PDE1 and PDE2) were used to detect instabilities such as rotating stall or surge. They are located at the same radius ratio \(r/r_2 = 1.05\) but with a 90° difference in circumferential position. Pressure transducers at the inlet and at the exit of the stage
(PVV9 and PNV7) indicated the instantaneous pressure rise of the stage during mild or deep surge.

Time dependent mass flow was detected in two different ways: During rotating stall the local flow angle and dynamic head were determined with the aid of PDE1 sensing the wall pressure and of a cylindrical single-hole probe measuring a pressure signal behind the impeller exit in the middle of the diffuser walls. The probe includes an embedded sensor as described in Gyarmathy et al., 1991. With the knowledge of the time mean flow angle found by a pneumatic yaw meter three readings over a number of rotating stall cycles were taken by this single-hole probe: one by aligning the hole to this angle and two others by turning it 45° clockwise and anticlockwise. Using the probe calibration characteristics the instantaneous flow angle and the velocity could be calculated from the static pressure and the recorded pressures measured by the hole in these positions.

During operating points with mild or deep surge the time dependent mass flow through the stage was estimated by a double face Prandtl probe as sketched in Fig. 4.

Stagnation temperature at impeller inlet and exit during rotating stall were monitored by a thermocouple. The chromel-alumel wires had a thickness of 20 µm and hence the response time was lower than 40 Hz required to monitor rotating stall. At the impeller entry further measurements could be taken with a hot wire anemometer using a low overheat ratio, so that changes in temperature could be sensed (cf. Manca et al., 1988). These measurements allowed the determination of the phase shift and the amplitude damping between real flow and thermocouple signal. Since at the impeller exit no such comparison could be done due to the lack of space, the phase shift and amplitude damping were calculated from the results found at the impeller entry.

Signal Processing

The pressure traces were stored on a magnetic tape and for further processing low-pass filtered below the blade passing frequency and then digitized. The type of the instability was identified by processing the pressure signals by a two-channel frequency analyser. To distinguish the instabilities the following criteria were used:

- A phase shift between the pressure traces from PDE1 and PDE2 reveals rotating stall.
- Synchronous pressure fluctuations at PDE1 and PDE2 were identified as mild or deep surge.
- Mild and deep surge were distinguished by use of the double-face Prandtl probe detecting reverse flow.

The measurements done during rotating stall were recorded by the two-channel frequency analyser over several rotating stall periods. Afterwards a phase-locked averaging (locked to the stall cycle) of all signals was performed on a Vax computer.

![Fig. 3: Location of the instantaneous measurement devices](image)

![Fig. 4: Double-face Prandtl probe](image)
the topic of the present paper. We just point out some results which can be seen from the compressor maps and from the frequency analyses.

Frequency analysis results are plotted over the flow coefficient $\phi$ for three diffuser geometries tested in the closed-loop mode at the same impeller speed ($\text{Mu} = 0.4$) in Fig. 6a-c). In this figures two different kinds of information are plotted over the mass flow rate $\phi$. The first is the continuous "throttle" line starting at some $\phi$ value at time zero (right ordinate) and indicating the reduction of $\phi$ by throttling as time proceeds. The second information refers to the black patches and is associated with the frequency and phase lag of any fluctuations recorded during the throttling process. These patches are formed by arrays of small T-shaped strokes, each of them representing the result of a Fourier analysis carried out during a time period of 1 second. The strokes indicate at what throttle setting (i.e. $\phi$) the analysis was made, what harmonic frequencies were detected (left ordinate) and what phase shift was found between the two transducers. The phase shift is represented by the orientation of the T-shaped stroke (see window). Since PDE1 and PDE2 are positioned $90^\circ$ apart a $90^\circ$ shift between their signals corresponds to a single rotating stall cell, a $180^\circ$ shift to two cells and a $270^\circ$ shift to three cells. Zero phase shift corresponds to an axisymmetric flow fluctuation due to mild or deep surge.

The horizontal bonds of $0^\circ$ phase shift appearing at low frequencies indicate that synchronous fluctuations occur at 4.5 and 19 Hertz (including higher harmonics at 9, 40 and 60 Hertz), these are associated with deep and mild surge, respectively. The large patches of $90^\circ$ phase shift above 40 Hertz are due to rotating stall with a single cell. This rotating stall appeared a $\phi = 0.028$ regardless of the diffuser used.

Conclusions drawn from Figs. 5 and 6 (and from analogous of Fig. 6 pertaining to higher $\text{Mu}$ values) can be summarized as follows:

- At those speed lines where mild surge was the first detected instability, the compressor system became unstable at the speed line maximum.
- Rotating stall in stable operation was detected for the lower speed lines; at the higher speed lines rotating stall occurred only in combination with surge.
- The rotating stall consists of a single stall cell. The rotating stall frequency was proportionally changed by impeller speed. At $\text{Mu} = 0.4$ (42 Hertz) the stall cell propagated in the direction of the rotation of the impeller with a relative speed $\omega_s/\omega_{	ext{Rotor}} = 27\%$.
- Impeller speed did not noticeably influence the frequency of mild or deep surge.

In the open loop mode only one configuration was investigated, the one with the standard diffuser (24 vanes, $25^\circ$). Instead of a broad mild surge region the compressor ran into deep surge as soon as the pressure maximum was reached, i.e. after only a few cycles of mild surge. The frequencies of mild and deep surge decreased compared to the closed loop mode, being 10 and 1.6 Hertz, respectively.

**IMPELLER ROTATING STALL AT $\text{Mu} = 0.4$**

Before any hypothesis concerning the transition from mild to deep surge can be established, one has to determine whether the rotating stall is caused by the impeller or by the diffuser. Three observations point to the impeller.

![Fig. 7. Dimensionless pressure rise versus flow rate for different diffusers ($\text{Mu} = 0.4$)](image-url)
element for the onset of rotating stall. As a third evidence, Fig. 7 shows that the dimensionless pressure rise calculated from the stagnation pressures - falls dramatically as the flow is reduced below $\phi = 0.028$. Such reduction can only be explained by failure of the energy-input, i.e. break-down of the flow in the impeller.

Further measurements were made to identify the impeller as the responsible stage element for rotating stall, involving the work factor of the stage. For this first let us make some simplified assumptions. With a $90^\circ$ impeller any change in work factor $\lambda$ would be a consequence of a change in slip factor since the ideal work input at constant speed is independent of mass flow. If one measured a fluctuating work factor during rotating stall it would be caused by rotating stall in the impeller which makes the slip factor rise up and down as a consequence of the separations (cf. Kämmer, 1984). A rotating stall in the diffuser only would just act on the local mass flow but would not affect the work factor.

A different situation is found in an impeller with back-leaning blades like the present one. Here the work factor directly depends on the mass flow. A change in work factor during rotating stall is therefore not a sufficient proof for impeller rotating stall. But at a given difference between maximum and minimum in local flow rate during rotating stall the influence on the work factor is much greater if this change is enhanced by separations in the impeller than it is if the change is caused due to variations of the flow rate due to rotating stall in the diffuser. These two situations are sketched in Fig. 8.

The instantaneous work factor $\lambda$ was determined by measuring the fluctuations of stagnation temperature and stagnation pressure at the entry and the exit of the impeller. Fig. 9 shows the trace of the work factor $\lambda$, the total pressure ratio $\pi$, the efficiency $\eta$ and the local flow rate $\phi$ during two periods of rotating stall. They are calculated as follows:

$$\lambda = \frac{c_p}{u_2} (T^o_2 - T^o_1)$$

$$\pi = \frac{p^o_2}{p^o_1}$$

$$\eta = \frac{c_p T^o_1 \left( \frac{p^o_2}{p^o_1} \right)^{\frac{k-1}{k}}}{c_p (T^o_2 - T^o_1)}$$

$$\phi = \frac{\dot{m}(t)}{\rho_1 D_2^2 u_2}$$

The change in work factor $\Delta \lambda$ is as high as 20%. If one determines the change in work factor expected as a consequence of the change in mass flow using the Wiesner slip factor (Wiesner, 1967) one would expect a $\Delta \lambda$ of a mere 4 - 5 %. Therefore it is obvious that the impeller as the energy feeding element is impaired by the rotating stall.

rotating stall in the diffuser

rotating stall in the impeller

slip factor $f$

$\phi \rightarrow \lambda$

Fig. 8: Work factor changes distinguishing impeller and diffuser rotating stall

1Due to the subdivision proposed by Frigne and Van Den Braembussche (1983) the rotating stall encountered here must be a progressive one (Group III: flow separation in the impeller) although its propagation speed points to an abrupt one (Group II: interaction between impeller and diffuser).
HYPOThESIS ABOUT THE TRANSITION FROM MILD TO DEEP SURGE

With the knowledge that the detected rotating stall with 42 Hertz is caused by the impeller - supposedly by incidence - a possible explanation for the transition from mild to deep surge can be given. The true behaviour of an unsteady inducer/impeller flow field is not yet sufficiently understood. Therefore an approach using quasi-steady data shall be attempted. This is justified by the fact that the mass flow pulsations are much slower (typically 7 to 18 times) than the transit time of a fluid particle through a rotor channel.

During mild surge one observes that the time mean flow coefficient \( \phi \) as measured by time independent devices is much higher than the flow rate labeled \( \phi_{RS} \) at which rotating stall begins under steady flow conditions. The latter limit could be observed during low-speed tests, e.g. for \( \mu = 0.4 \). However, during a mild surge cycle the time-resolved value of the flow rate \( \phi(t) \) can intermittently reach or even undershoot the value \( \phi_{RS} \). The question is whether this really happens and if there exists a threshold time during which \( \phi(t) \) can be less than \( \phi_{RS} \) without causing a subsequent pressure drop in the impeller, i.e. without triggering deep surge.

In Fig. 10 we consider operation at constant speed with mild surge during a gradual closing of the throttle valve. Measurements show that the flow coefficient \( \phi(t) \) varies sinusoidally around the decreasing mean value \( \phi \) dictated by the valve. The frequency \( f_{MS} \) is constant and the peak-to-peak amplitude \( \Delta \phi_{TT} \) may stay constant or increase in time. At some time the momentary minimum flow rate will fall below the stall threshold, making \( \phi(t) < \phi_{RS} \) for the first time. Here rotating stall is not yet necessarily triggered. During the next cycles stall will occur at first for a short, then for a longer time. Once rotating stall is established for a time long enough, the system will respond with backflow, i.e. a deep surge pulse.

Let us denote by \( T^* \) and \( \Delta \phi^* \) the critical undershoot time and the critical flow deficiency, the term "critical" meaning the final event leading to flow break-down. This will enable us to analyze the experimental data in search of a flow break-down criterion on the basis of undershoot time \( T^* \) or flow deficiency \( \Delta \phi^* \) or the product \( \Delta \phi^* T^* \) which is a measure for the energy available for causing flow break-down. The undershoot time \( T^* \) will be set in relation to either the time of one rotor revolution \( T_{Rotor} \) or the time a fluid particle needs to pass the impeller flow channel \( T_{Trans} \) or the time a rotating stall cell needs to run once around the annulus \( T_{RS} \). Thus we get the following parameters to formulate flow break-down criteria:

\[
\begin{array}{ccc}
\text{parameter} & a) & b) & c) \\
\hline
I & \frac{T^*}{T_{Rotor}} & \frac{T^*}{T_{Trans}} & \frac{T^*}{T_{RS}} \\
II & \Delta \phi^* & & \\
III & \frac{\Delta \phi^* T^*}{T_{Rotor}} & \frac{\Delta \phi^* T^*}{T_{Trans}} & \frac{\Delta \phi^* T^*}{T_{RS}} \\
\end{array}
\]

Tab. II: Definition of various parameters to formulate flow break-down criteria

From the time averaged measurements the flow coefficient for rotating stall onset (measured at \( \mu = 0.4 \) \( \phi_{RS} \) and the one for the beginning of deep surge \( \phi_{DS} \) are...
known. From the instantaneous measurements one gets \( f_{RS}, f_{MS} \) and \( \Delta \phi_{TT} \). So \( T^* \) and \( \Delta \phi^* \) can be calculated. We find for \( T^* \) with the assumption of a sinusoidal flow fluctuation:

\[
T^* = \frac{1}{f_{MS}} \left( 1 - \frac{1}{\pi} \arcsin \left( \frac{2}{\Delta \phi_{TT}} (\phi_{DS} - \phi_{RS}) \right) \right)
\]

(5)

and for \( \Delta \phi^* \):

\[
\Delta \phi^* = \frac{1}{2} \Delta \phi_{TT} + \phi_{RS} - \phi_{DS}
\]

(6)

As can be seen from Table 3 those parameters which are related to the an undershoot time \( T^* \) are most suitable as a flow break-down criterion, more suitable than the ones related to the flow deficiency \( \Delta \phi^* \) or to the product of flow deficiency and undershoot time, \( \Delta \phi^* T^* \). There is not much difference whether the undershoot time is related to the time of one rotor revolution \( T_{Rotor} \) or to the time a rotating stall cell needs to travel around the annulus \( T_{RS} \), since the rotating stall frequency itself is related to the rotor speed. It makes more physical sense if one considers the relation to \( T_{RS} \) because the rotating stall is responsible for the flow break-down. The relation \( T^*/T_{RS} \) is about 1.5, indicating that a rotating stall cell travels more than once around the annulus before deep surge is triggered. The flowfield downstream the impeller is consequently disturbed over the entire annulus. (Another criterion could be formulated by the ratio \( T^*/T_{RSi} \) where \( T_{RSi} = 1/(f_{Rotor} - f_{RS}) \) is the time in which the stall cell runs around relative to the rotor. \( T_{RSi} \) being about 2 to 2.5 times shorter than \( T_{RS} \), the ratio values are \( T^*/T_{RSi} = 3...5 \). The scatter is comparable to that of \( T^*/T_{RS} \).

The role of rotating stall can be seen more clearly in Fig. 11 which was measured in the open loop mode at \( \mu = 0.6 \). The trace PJ1 is the dynamic head sensed by the double Prandtl probe towards the normal flow direction. It is proportional to the square of the mass flow through the stage. As long as PJ1 is positive there is no global back flow. The dotted line in Fig. 11 represents the inferred value of the dynamic head corresponding to rotating stall inception in the impeller. It can be seen on the left that the mass flow begins to oscillate with about 10 Hertz. The amplitude of the oscillation grows rapidly. The first time the trace of PJ1 falls below the dotted line the static pressure at the impeller exit PDE1 exhibits a sharp peak. This peak corresponds to rotating stall since there is a phase shift between the peak sensed by PDE1 and the one sensed by PDE2 (not shown). After a single burst, rotating stall disappears as soon as the mass flow begins to rise. When the mass flow reaches the dotted line for a second time and falls below it, PDE1 shows more than one peak. Rotating stall is initiated now for a longer time. The time is long enough to enable global reverse flow i.e. deep surge. This can be seen as well from the trace of PJ1 which falls far below zero as from the locking-in of the traces PDE1 and PNV7 thus indicating, that there is no pressure rise through the diffuser and the scroll.

**EMPIRICAL TESTING**

Experimental verification was only possible for those configurations where the time resolved mass flow was measured either by the double Prandtl probe or by an additional pressure transducer for sensing the pressure difference over the flow nozzle. The latter is probably not accurate enough since an oscillating flow generates different pressure differences over a nozzle than does a steady one. But the relation between the pressure difference at the nozzle and the one sensed by the probe is known for different amplitudes of mass flow oscillation. So these data were taken to get an idea which criterion performs best.

The values obtained for the different parameters to formulate flow break-down criteria are listed in Table 3 for the different configurations. If the amplitude of the flow fluctuation near the beginning of deep surge, \( \Delta \phi_{TT} \), was not sufficiently steady, the largest amplitudes were taken to calculate the criteria. The bottom lines list the mean value \( \bar{X} \) and standard deviation \( S_X \) together with its relative value \( S_X / \bar{X} \).
CONCLUSIONS AND CONSEQUENCES FOR SURGE MARGIN IMPROVEMENT

The present experiments have shown rotating stall in the impeller to have a key function in triggering deep surge. Although the time-mean flow rate was far above the rotating stall limit, during mild surge the instantaneous flow rate intermittently dropped below this limit. Rotating stall appeared for a while but vanished as soon as the flow rate increased again. With further reduction of the time-mean flow rate, however, rotating stall could persist for a longer time period. If this time period exceeded a threshold value deep surge was triggered due to the break-down of the flow in the diffuser. In the investigated configurations this threshold was reached if the stall cell could make one and a half revolutions in the casing.

Although, in most centrifugal pumping systems, the transition from mild to deep surge is found to occur, as summarized by Greitzer (1981), either without noticeable rotating stall or with rotating stall in the (vaned) diffuser, the authors believe that mechanisms treated here are not a unique case. For example the data reported by Amann and Nordenson (1961) show a similar behaviour.2

If a transition from mild to deep surge is due to these mechanisms there are several possibilities to improve the surge margin.

The first group is directly aimed at reducing the mass flow at which impeller rotating stall is triggered. Apart from redesigning the impeller, rotating stall can be delayed by pre-swirl. This can be affected by various means like the use of an IGV, or tangential injection of fluid at impeller inlet, or the use of an impeller shroud which extends far in front of the impeller.

The second group of possibilities to improve the surge margin is deduced from Fig. 10. With a characteristic frequency of rotating stall $f_{RS}$, the question is whether the undershoot time is long enough to enable a developing rotating stall cell to exist during a sufficient number of revolutions. At a given time-mean flow rate the undershoot time increases both with increasing amplitudes of mass flow fluctuation, or with decreasing frequency of mild surge $f_{MS}$. As shown by stability theories the amplitude is determined by the overall system configuration and the slope of the speed line, being higher for systems with large volumes and for compressors delivering more pressure. The frequency $f_{MS}$ may be related to the Helmholtz frequency of the system or may be determined by a standing wave pattern. This means that a given stage exhibits different deep surge limits if used in different piping systems. A practical conclusion is that one has to check these influences before spending much time and cost on redesigning the stage for any particular application causing instability problems.

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Fig. 5: Compressor map for different diffusers
Fig. 6: Fourier analyses of the pressure signals PDE1 versus PDE2 for three different diffusers at $\text{Mu} = 0.4$
Table 3: Measured data and calculated parameters to formulate a flow break-down criterion

<table>
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<tr>
<th>Configuration</th>
<th>Flow Coeff. (at deep surge)</th>
<th>Time Scales</th>
<th>Parameter (understanding)</th>
<th>Parameter (amount)</th>
<th>Product Duration x Amount</th>
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<td>0.0524</td>
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<td>2.6 16.2 1.47 0.0162</td>
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<td>0.0790</td>
<td>354.4</td>
<td>19 107</td>
<td>2.5 16.2 1.47 0.0162</td>
</tr>
<tr>
<td>0.9 N12W25 closed</td>
<td>0.0570</td>
<td>0.0760</td>
<td>354.4</td>
<td>19 107</td>
<td>2.5 16.2 1.47 0.0162</td>
</tr>
<tr>
<td>0.4' N24W25 closed</td>
<td>0.0468</td>
<td>0.0524</td>
<td>296.1</td>
<td>19 75 2.7 11.8 4.17 5.35 1.12 0.0090</td>
<td>0.037</td>
</tr>
<tr>
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<td>19 107</td>
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</tr>
<tr>
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<td>345.4</td>
<td>19 107</td>
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<tr>
<td>0.9 N24W25 closed</td>
<td>0.0542</td>
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<td>354.4</td>
<td>19 107</td>
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<td>• hysteresis</td>
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</table>

Note: The table includes various parameters such as mean, standard deviation, and relative scatter, but the specific values are not provided in this excerpt. The table is structured to compare different configurations and their corresponding flow coefficients, time scales, and parameter estimations.