DYNAMIC DATA ANALYSIS OF COMPRESSOR ROTATING STALL

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ABSTRACT

Techniques for rotating stall verification and analysis of detrimental effects are described. Majority of dynamic pressure data presented herein are from centrifugal compressors, including one unit that required field modification. Data from dynamic pressure probes, along with rotor vibration are scrutinized for two sources of rotating stall, the impeller itself and the diffuser.

In many cases, the number of stall cells is greater than one, which is verified to have negligible reaction forces on the rotor-bearing system. However, when there is one stall cell per 360 degrees, unbalanced loads can be greater than the machine can withstand, thereby dictating design alterations. Use of dynamic pressure probes is shown to be highly beneficial for more complete analysis.

Various techniques for changing rotating stall characteristics, including forcing stall into a multiple cell structure, are discussed. Results are given for some methods, and others having potential are also described, with suggestions for testing verification. The severity of rotating stall is another issue to be addressed, relating to when economics outweigh the need for change.

Axial compressor data are also presented, with somewhat different analyses, especially on blading. For a unit in FCC service, dynamic pressure probe data are presented for both shop tests and for field surge line definition. Pressure pulsations are shown to provide an excellent precursor to full surge flow reversal, which can be much more detrimental if not properly controlled.

INTRODUCTION

Requirements for comprehensive data analysis of machinery typically include the three major parameters of amplitude, frequency, and phase angle. Accurate conclusions can often be made with low risk for rotating stall if phase angle data are not available, as those made by Fulton and Blair [1]. Another example is subsynchronous rotor vibration at a natural frequency due to change in variables such as aerodynamic cross coupling, described in numerous publications. As in synchronous rotor vibration, phase angle for compressor rotating stall can be a determining factor to solve problems or test new designs. Multiple dynamic pressure probes can be used to ascertain relative phase angles and inherent stall propagating speed, as explained in the Appendix.

Test cases such as those described herein can offer updated trend analyses to preclude the need for pressure transducers in some cases of problem solving. By contrast, the need for them in evaluations of new designs is highlighted. These data can parallel computational fluid dynamics (CFD), which requires accurate boundary conditions, confirmed with testing [2].

Results will be given herein for several cases of compressor dynamic pressure measurements:

- An open inducer centrifugal impeller, which was shown to be free of rotating stall at surge inception, thus verifying that blade excitation was due to surge
- Single-stage test rig data on a covered impeller to verify the cause of rotor vibration on a high pressure reinjection compressor
- Two impellers developed for aerodynamic performance improvements that were shown to have unacceptable stall characteristics showing need for modifications
- An axial compressor for which surge control line could be defined with increased confidence, especially relative to rotating blade reliability

Severity of rotating stall can also be evaluated; a method is given herein to eliminate concerns for severe problems, and theory is presented for future test verifications of another method. Continued use of techniques described can also assist in advancements in surge control for reliability and cost savings.

OPEN INDUCER CENTRIFUGAL IMPELLER

A field failure of blading on a large combustion air compressor was analyzed to show that the most probable cause of high cycle fatigue was continuous surge. The compressor has a single stage, employing an open inlet, with a design pressure ratio near 2.0 and tip speed near 1100 ft/sec. Failures occurred shortly after commissioning, during which questionable flow and pressure fluctuation sensing was utilized in defining incorrect setpoints for automatic surge controls. A cross section shown in Figure 1 gives initial location of dynamic pressure probes for failure analysis. Probes were used for shop tests completed on both an original blade design and redesigned impeller, and also during recommissioning in the field. It was shown that redesign may not have been necessary, but offered additional reliability.

In the meantime, an original spare rotor was used for production, but probes were installed for a more comprehensive definition of the surge control line. Probe installation was not optimum, but stall was found with a small increase in rotor vibration prior to surge, sufficient to obtain accurate setpoints. Subsequent test instrumentation for pressure pulsation included the following:

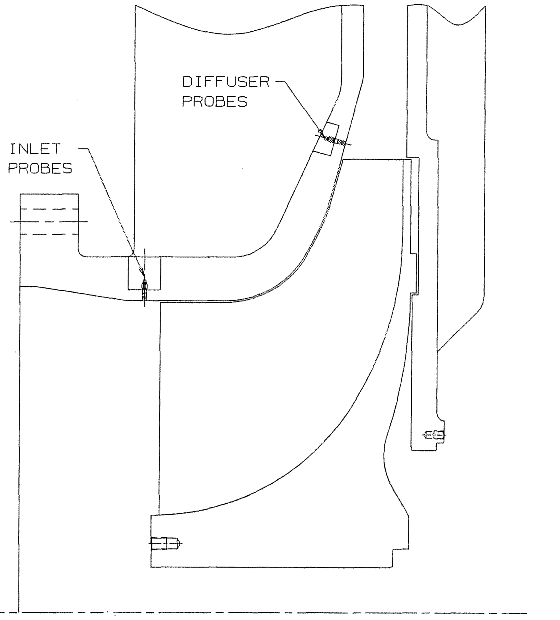

Figure 1. Cross Section of Compressor Showing Initial Locations of Pressure Probes.

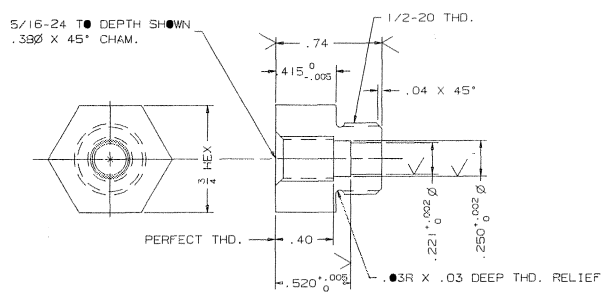

Figure 2. Adapter for Dynamic Pressure Probes

- Flush mounted, AC-coupled piezoelectric transducers with 0.5 Hz to > 10 KHz frequency response, two microsecond rise time, 100 mv/psi sensitivity, and high signal to noise ratios
- Adapters shown in Figure 2, along with seals and clamp nuts for the probes

These ensure no extraneous natural frequencies to contaminate data, either from gas modes or structural effects. In addition, multiple probe circumferential angles were utilized in case data indicated a high number of stall cells. See Appendix.

Example of pulsation data in the inlet is given in Figure 3, on the verge of surge. The component at 2.5 Hz only occurred just below peak discharge pressure, and was found to be purely axial, similar to a mild part volume surge with zero circumferential phase angle. Amplitudes were sufficiently high and electrically noise free that besides giving incipient surge points also provided an additional alarm signal for the control room. Eddy current probe and accelerometer monitors were modified in turn, including band pass filters.

Shop verification tests also involved checking the inlet just upstream of the inducer for instantaneous circumferential static pressure variations, in about 20 degree increments. Even without any blade-damaging type of rotating stall being found for the original and redesigned stronger impeller, the concern was that the discharge volute caused a serious maldistribution of loading at the compliant area of blading. Data shown in Table 1 verified that up to and at the incipient surge point, Fourier harmonic analysis gave negligible excitation in the frequency area of concern, above three

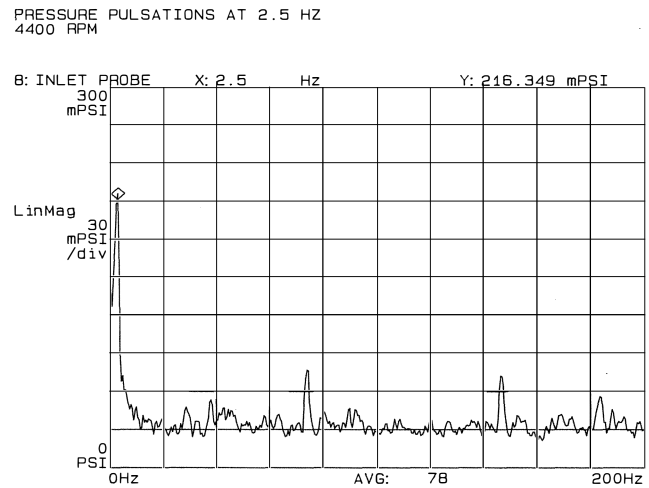

Figure 3. Spectrum Showing 2.5 Hz Pulsation for Open-Inducer Impeller.

times rotor speed. However, going lower in flow caused a large change in dynamic pressure as shown in Figure 4 at probes located upstream and above the inducer leading edge. Using various points that gave differences around the circumference, another Fourier analysis of the dynamic data gave an approximate force variation for flow reversal also shown in Table 1. This showed that the blading could be subjected to large numbers of severe cyclic forces for excessive running time immediately prior to or in surge. In order to give utmost reliability, impellers were designed with zig-zag damping pins (patent applied for) shown in Figure 5. These would overcome the most probable cause of failure, continuous surging, if it occurred again.

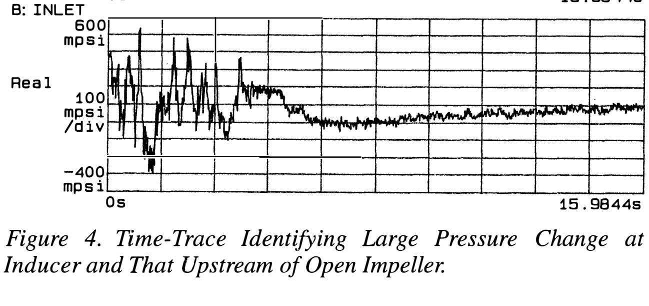

Figure 4. Time-Trace Identifying Large Pressure Change at Inducer and That Upstream of Open Impeller.

REINJECTION COMPRESSOR IMPELLER TESTING

An impeller design with extensive experience produced subsynchronous response at a frequency varying from 70 to 80 percent of rotating speed. This occurred only when approaching the surge flow points, similar to that described by Fulton and Blair [1]. Difference evaluation did not show the cause, so there was impetus to do a special rig test. Moreover, this was the first time the manufacturer experienced subsynchronous vibration on a unit with these characteristics. In fact, there were also never any known cases of

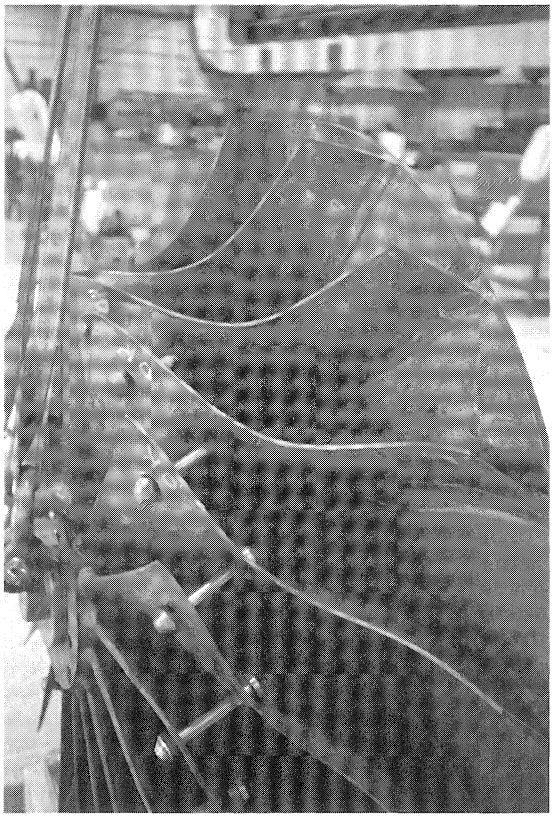

Figure 5. Open-Inducer Impeller with Zig-Zag Damping Pins.

Table 1. Calculated Harmonic Excitation Forces on Open-Inducer Blades.

Implied Force Variation From Average (% - Peak to Peak)			
Speed Harmonic	**Using Static Probes**		**Using Dynamic Probes**
	At Design Flow	**With Inlet Stall**	**With Flow Reversal**
1	**2.28**	**6.48**	**5.8**
2	**1.86**	**6.27**	**4.8**
3	**0.47**	**1.92**	**14.4**
4	**0.30**	**0.50**	**3.2**
5	**0.34**	**0.85**	**4.6**
6	**0.23**	**0.81**	**3.1**
7	**0.11**	**0.62**	**0.4**
8	**0.10**	**0.87**	**1.0**

low-frequency stall similar to others' reported experiences [1, 3, 4, and 5]. However, potentially damaging amplitudes described by Lynghjem, et al., [6] required the design to be modified. Rotor subsynchronous vibration at part load is shown in Figure 6. Response at rotor/bearing first critical frequency was extremely low, so that this was concluded to be a forced response, most likely one-cell rotating stall. Note that the subsynchronous component starts just below rotor speed, then decreases in frequency. This characteristic will be further expounded on. Field tests showed that vibration levels might be harmful especially if automatic controls failed to keep flow above that where forced vibrations occurred. Since bearing modifications including damper bearings were analytically shown to give negligible benefits for vibration amplitude, the impeller design was modified. Verification of the rotor/bearing design was also shown by insignificant amplitude at natural frequencies over the entire performance map from aerodynamic or seal cross coupling effects.

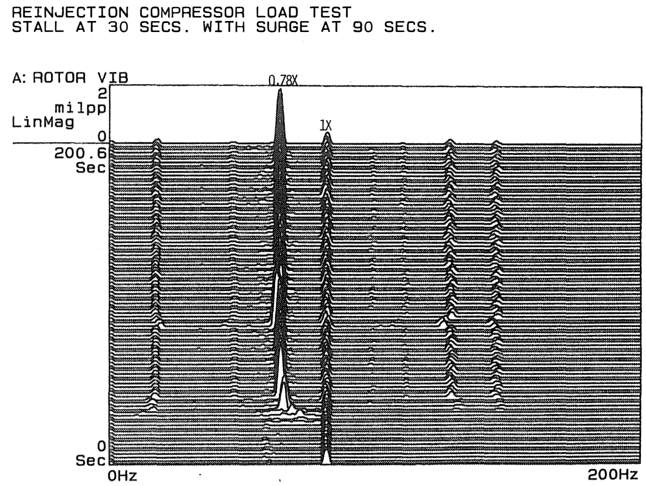

Figure 6. Waterfall of Rotor Vibration with Subsynchronous Component for Reinjection Compressor.

While redesign was done and impellers were being manufactured, a closed-loop rig test was done for the original stage design. It was recognized from previous analyses that the cause was most likely a one-cell rotating stall due to an impeller, [7] thru [9].

However, the rig-tested impeller always exhibited three or two cells. Dynamic pressure probes were installed in both the inlet and diffuser near the impeller. Response at blade passing frequency (BPF) for normal flow is shown in Figure 7, while pulsation amplitude during three-cell stall is in Figure 8. Rotor response changes for the rig were negligible confirming that forces are balanced from the peak pulsations, 120 degrees apart. Time traces for phase angle calculation are given in Figures 9 and 10. A summary is given in Table 2 for one of the test sequences, as air equivalent tip speed was increased.

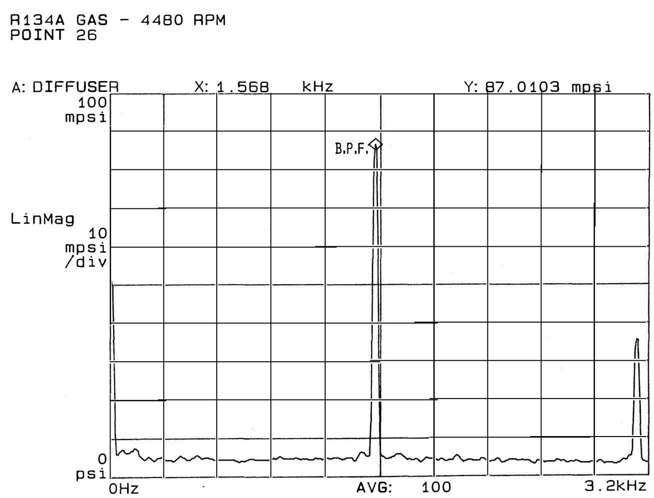

Figure 7. Normal Spectrum for Reinjection Compressor Impeller.

The rig test did give evidence that the impeller was the cause of rotating stall. Using an abradable seal at the impeller eye did not

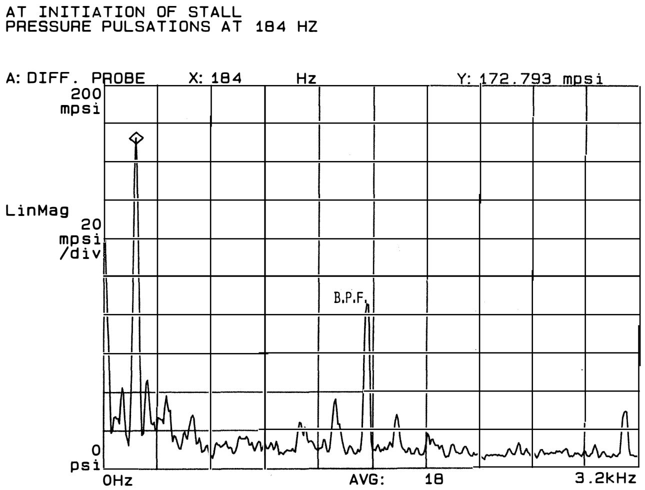

Figure 8. Spectrum for Rotating Stall of Reinjection Compressor Impeller.

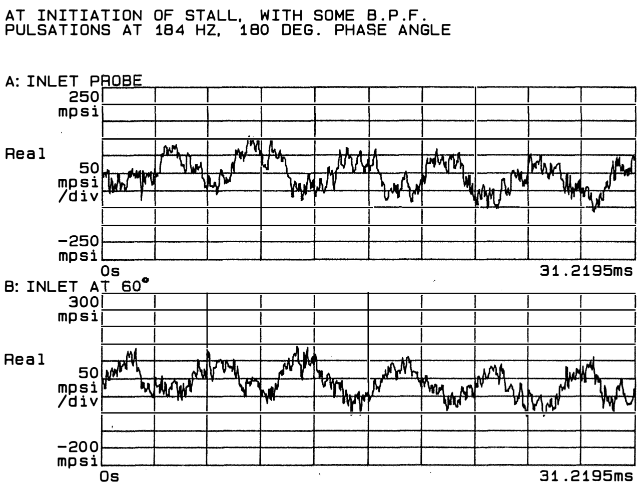

Figure 9. Three-Cell Stall Time-Trace at Inlet of Reinjection Impeller; Probes 60 Degrees apart.

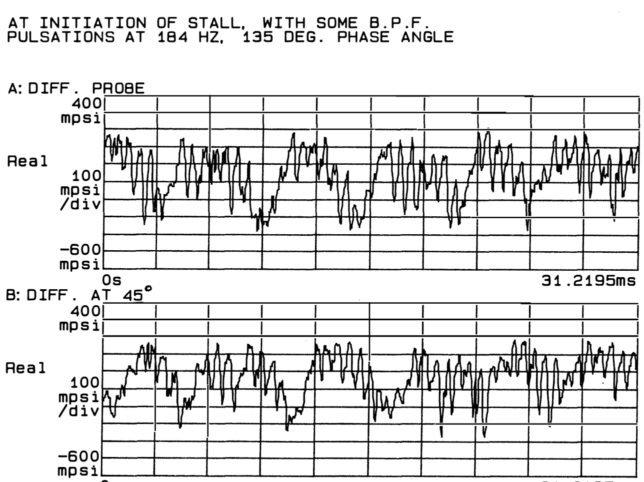

Figure 10. Three-Cell Stall Time-Trace at Diffuser of Reinjection Impeller; Probes 45 Degrees apart.

Table 2. Test Variation for Impeller Stall.

REINJECTION COMPRESSOR IMPELLER				
RIG ROTATING STALL DATA				
EQUIVALENT TIP SPEED	STALL FREQUENCY	NO. CELLS	STALL SPEED	STALL SPEED/ ROTOR SPEED
FT/SEC	HZ		HZ	PERCENT
300	143	3	47.7	85
400	191	3	63.7	85
500	228	3	76.	81
600	258	3	86.	77
STALL COULD ONLY BE HELD IN WITHOUT SURGE AT 300 AND 400 FT/SEC.				
STALL COULD NOT BE FOUND AT 720 FT/SEC.				

change characteristics. There were some small stator differences in the rig as compared to the compressor, which was also inspected to be as designed and not much different than previous compressors with similar staging. However, the differences do remain as a likely cause for why the actual compressor had one cell, rather than two or three cells as in the rig. The most likely cause of one-cell rotating stall was deduced to be due to a predominant one per revolution variation in the impeller itself. Note that the frequency relative to a stationary point was decreasing as the flow was decreasing. In other words, initially the rotating stall frequency relative to impeller blades is zero, then increased to near 20 percent of speed. Relative change in frequency correlated with compressor surge testing, Figure 6. The rig-tested impeller supports this hypotheses for the three and two-cell stall; data such as in Figure 11 showed a higher amplitude at two and three per revolution than at one times speed. More support is found in [7], where in Figure 6 of their tests showed a four per revolution predominant peak, and typically four stall cells for inducer stall, even with intentional nonuniform stationary sources.

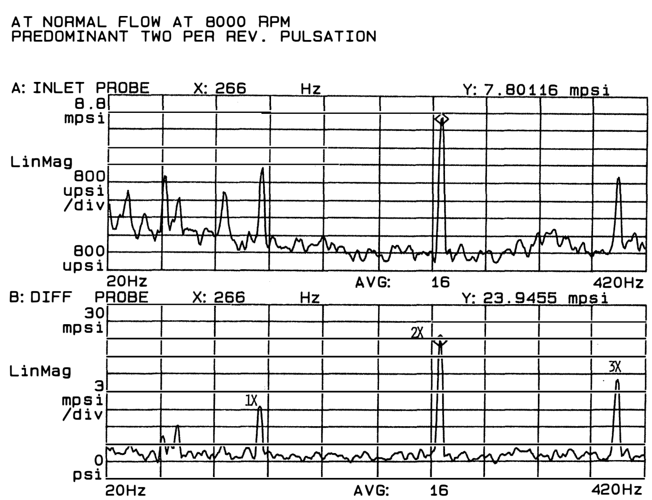

Figure 11. Typical Low-Frequency Spectra for Reinjection Compressor Impeller.

Rig tests could not produce one-cell stall by changing size or rate of discharge valve closure, nor did variations in diffuser width, eye seal, gas composition, or pressure. A last test with a vaned diffuser design eliminated impeller rotating stall, but gave an unacceptable flow range.

Incidence angle changes at the impeller inlet analytically forced rotating stall to a much lower flow, below that of previous experience with actual surge. Gas production plans for the compressor did not require rig confirmation, as critical-path delivery of new impellers was almost complete for commissioning. In addition, it still could have been two-cell or three-cell stall in the rig. Surge and rotating stall occurred in the field just about simultaneously only during control line definition with rotor vibration reduced to acceptable levels, precluding bearing or other damage.

MID-FLOW COVERED IMPELLER RIG TEST

Following design, a rig confirmation test was done on an impeller with a change in blade profiles to obtain efficiency improvements. Again dynamic pressure probes were used in both the inlet and diffuser to assist in surge/stall definition. The same rig was utilized as for the reinjection impeller test described above, so that small nonuniform flow from stationary components should give similar effects. At tip speeds equivalent to operation with air below 1000 ft/sec, a one-cell rotating stall was found prior to surge. This stall was confirmed by CFD analysis to be due to the impeller. Near 1000 ft/sec equivalent speed, the same stall propagating at 65 percent of rotor speed was quickly followed by a diffuser stall, traveling near three percent speed, but with two cells. Rotor vibration shown in bottom trace of Figure 12 only increased for the one-cell stall (at 50 Hz), but not for the two-cell stall (at 4.5 Hz). Away from stall, pulsation at once per revolution was predominant, giving an indication of one-cell tendency. This is in sharp contrast to the previous rig test for the reinjection compressor which did not have one-cell stall. As rotor speed increased, only a strong two-cell diffuser stall without rotor vibration was found. The frequency increased as flow was reduced, contrary to that for impeller stall.

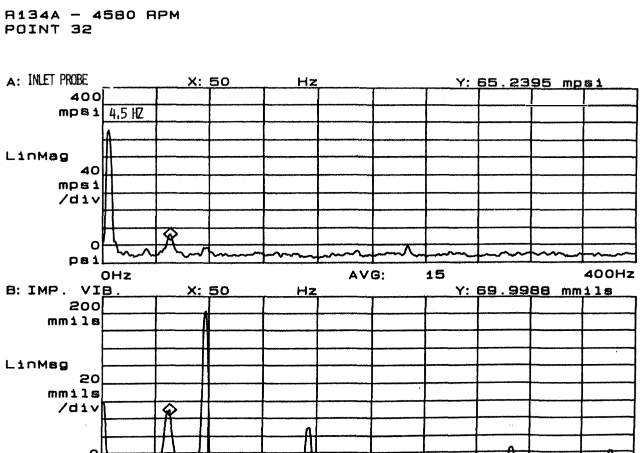

Figure 12. Pulsation and Vibration Spectra During Stall of Mid-Flow Impeller and Diffuser.

The same impeller was then tested with a vaned diffuser, which sometimes reduces the flow range of an impeller design. Flow at diffuser stall was higher, i.e., above that at which impeller stall had occurred. Had this been the only test, impeller stall would not have been discovered. This diffuser stall propagated faster than for the vaneless diffuser, probably due to the presence of diffuser vanes, in the range of 14 to 17 percent of rotating speed. Stall propagating speed increased as flow decreased. The number of stall cells was either three or two, with negligible effect on rotor vibration.

The vaned diffuser was removed and a design utilizing two stabilizer vanes in the diffuser was tested. The vanes were shaped to only guide the flow in the diffuser at the calculated log-spiral flow angles at the design flow point. When off design, the vanes would be at a different angle than the flow, but effects on efficiency were expected to be small, since there were only two vanes, 180 degrees apart. Retesting verified efficiency changes at normal flows were small, but effect on rotating stall was dramatic. One-cell impeller stall at 65 to 70 percent of running speed was eliminated, along with rotor subsynchronous vibration. At low flow, stall frequency near 80 Hz was mild. A waterfall plot is shown in Figure 13, where two cells were verified by phase angle measurement, giving a propagating speed near 30 percent of rotor speed.

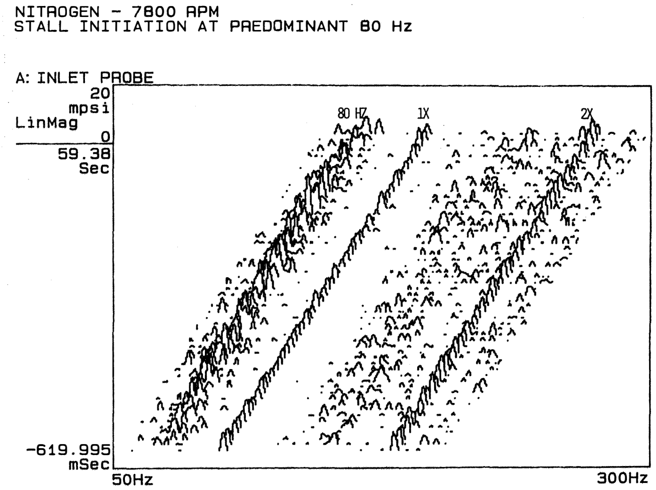

Figure 13. Initiation of Stall for Mid-Flow Impeller with Stabilizer Vanes.

MIXED FLOW COVERED IMPELLER RIG TEST

A comprehensive description of this impeller and rotating rig test is given by Hohlweg and Boal [2] including CFD analysis and aerodynamic testing. Additional information is given herein to expand on stall measurement and subsequent mitigation.

Original Design

At all equivalent tip speeds up to 900 ft/sec, a one-cell rotating stall causing subsynchronous rotor vibration was found, see Figure 14. Stall propagating speed range was 75 to 80 percent of rotor speed, implying the impeller to be the likely source. This conclusion was supported by CFD analysis, e.g, Figure 15, discussed in [2]. Pressure peak pulsation levels shown in Figures 16 & 17 were high, on the order of five percent of average pressure. Since pulsations were somewhat distorted, the discharge volute could have been the cause. The inlet was ruled out by using a flow straightener downstream of the guide vanes in the inlet connection. Tests were also made with a wider diffuser, and with a change in impeller axial position, giving the same results.

Analysis also showed that there was a definite large component at one times rotor speed, as compared to higher harmonics, e.g., Figure 18. As flow was decreased towards stall, the one per rev component was usually not as significant by comparison, especially at the inlet, but the one-cell stall persisted. Onset of stall shown in Figure 19 was sudden, typically building up to full amplitude in several cycles, less than 30 milliseconds.

Stabilizer Vane Tests

With success found for the mid-flow impeller, stabilizer vanes were again designed and tested. In this case since a volute was being utilized, one of the vanes was used to guide the flow to the volute tongue, and the second vane was located 180 degrees

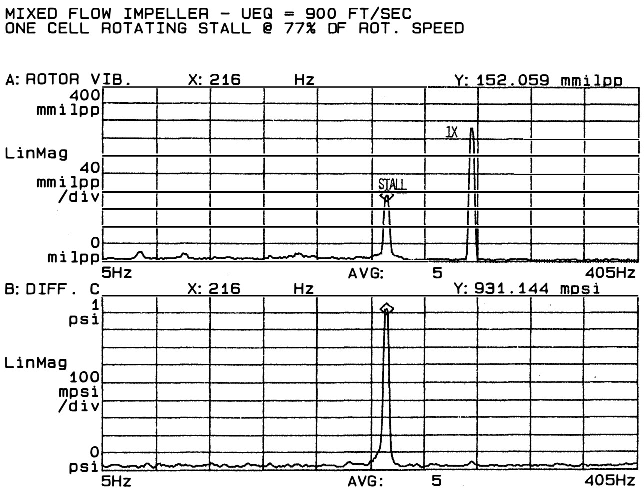

Figure 14. Spectra Showing Rotor Vibration and Diffuser Pulsations.

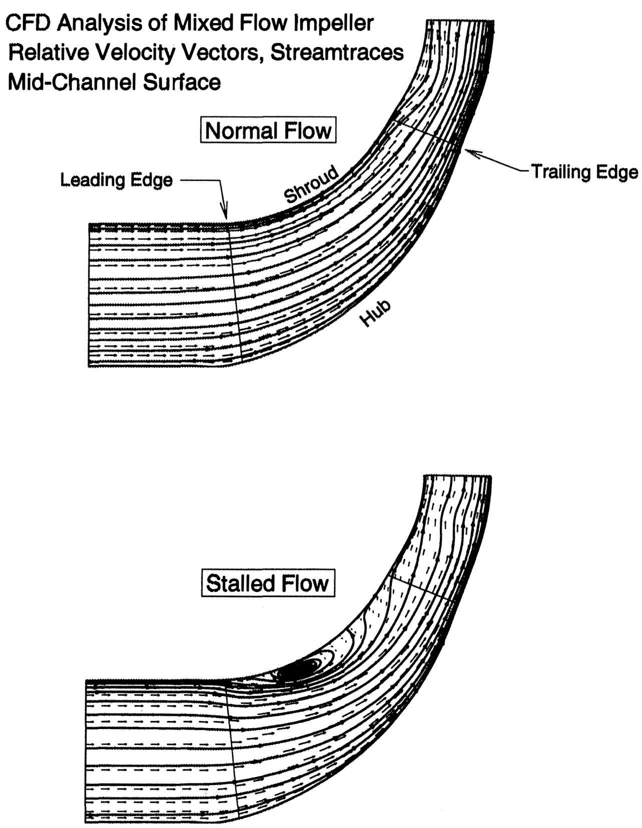

Figure 15. CFD Analysis Showing Relative Velocity Streamlines at Mid-Channel.

opposite. Test results were successful in that rotating pressure pulsation at normal speeds was greatly reduced, on the order of 80 percent, and surge occurred just about simultaneously. In fact, near optimum design speed, surge frequency near 11 Hz was found prior to rotating stall as shown by the peak-hold plots in Figure 20. In this case, stabilizer vanes did not fully overcome the one-cell pattern. In addition, the once per revolution pressure pulsation was still dominant, implying that the impeller effects were controlling. Rotor vibration was also greatly reduced, comparing Figure 21 to Figure 14. When stall frequency occurred, it was predominantly on only part of the 10 to 11 Hz, low frequency surge pulse cycle as

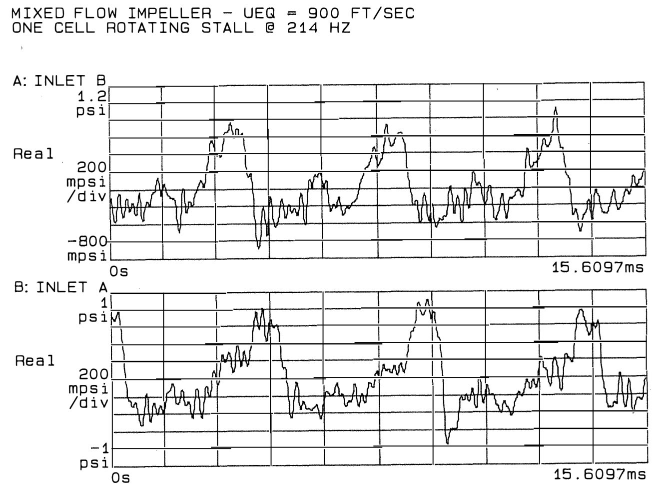

Figure 16. Time-Trace for Inlet Pulsations at Stall.

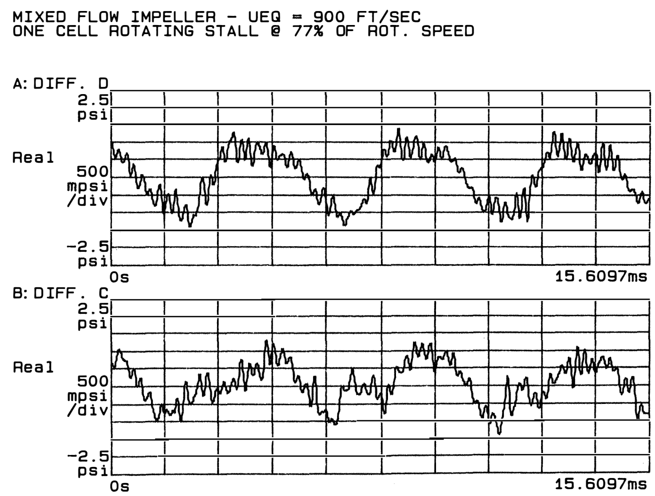

Figure 17. Time-Trace for Diffuser Pulsations at Stall.

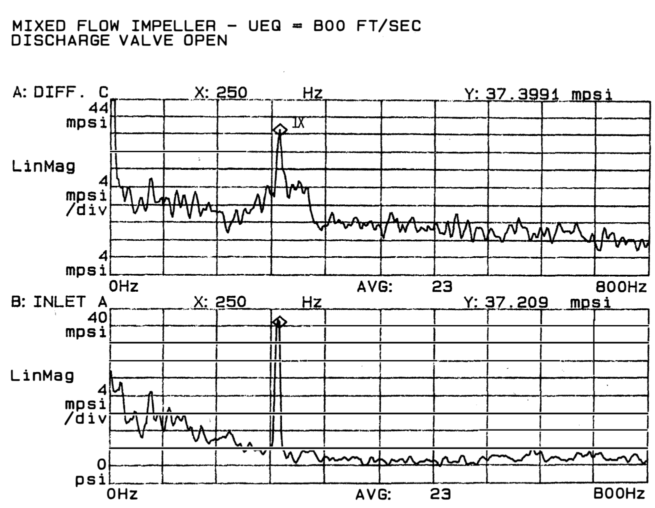

Figure 18. Normal Spectra at Diffuser and Inlet.

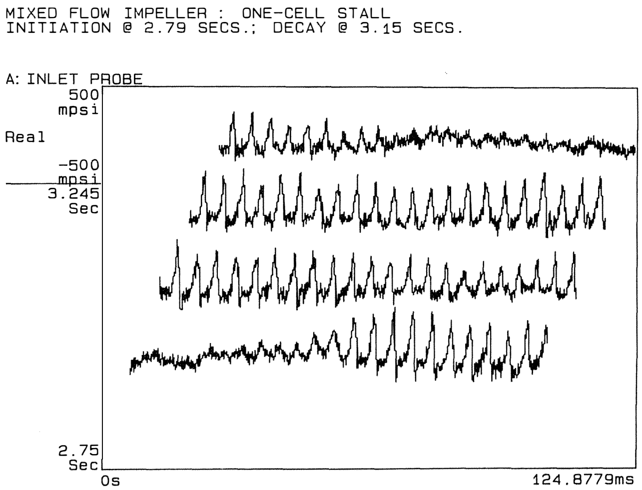

Figure 19. Time-Trace Waterfall for Onset of Rotating Stall.

shown in Figure 22. The amplitude at stall frequency was also broad-banded especially in the normal speed range. Note that the surge pulses in Figure 22 are in phase, verifying an axial motion. It was hoped that the stabilizer vanes would also at least give the same efficiency as without them, due to some guiding into the volute. Results showed an actual performance improvement so that vanes can be useful even if rotating stall is eliminated by other means.

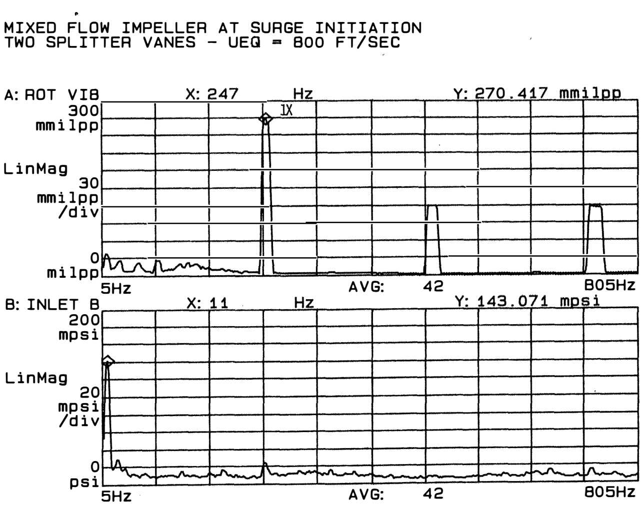

Figure 20. Peak-Hold Spectra Prior to Rotating Stall.

AXIAL COMPRESSOR STALL MEASUREMENTS

Consideration of rotating stall has been much more crucial for industrial axial compressors, let alone those in jet engines. It is well known that blade fatigue can occur if natural frequencies are equal to a major harmonic of rotating stall frequency, relative to a blade. For example, with propagating stall speed of 33 percent of blade rotating speed, the sixth harmonic of one-cell stall frequency can excite a rotating blade if its fundamental mode is at four times rotor speed. Likewise, six stall cells at the same propagating speed and amplitude would give much more excitation, since the first harmonic would be the excitation source. Thus, it can be important to define number of stalls and severity, unless it is for an off design transient such as shutdown.

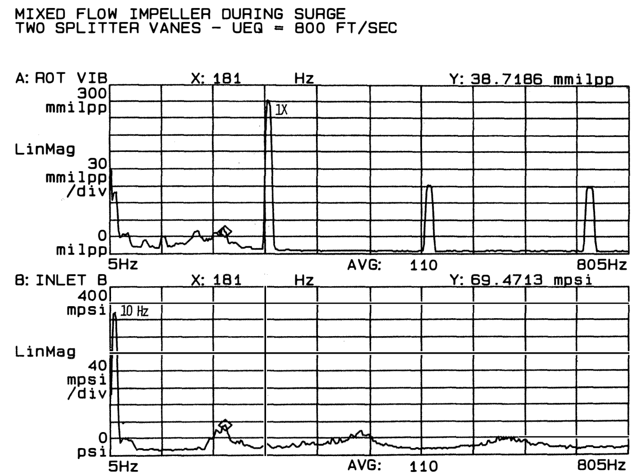

Figure 21. Peak-Hold Spectra with Stall Following Surge.

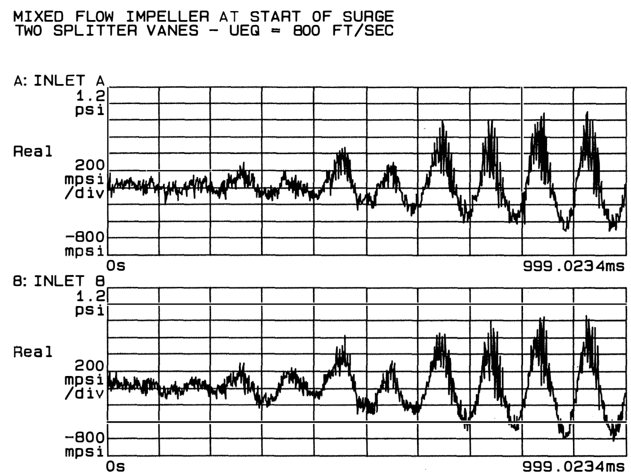

Figure 22. Time-Traces at Inlet Showing In-Phase 10 Hz Pulses.

Shop tests of initial axial compressors with controlled diffusion airfoils included dynamic pressure measurements. The only detrimental rotating stall that was found was during startup if the adjustable vanes were not fully closed, similar to older NACA 65 type blading. At normal speeds, setting the surge point showed no adverse effects of stall. An unusual case of a more recent unit for FCC service with a blank first row for future uprate was analyzed using dynamic pressure probes at rotor blade tips of various stages. Data shown in Figure 23 were extremely useful, as initial tests with fully open, adjustable guide vanes produced pulsations following surge recovery. Surge pressure is the low frequency trace, and rotating stall frequency is shown during time of recovery. This stall persisted after surge as the discharge valve was being open. Rotor vibration was not affected, as there were two stall cells traveling at 60 percent of rotor speed. As flow was somewhat higher than design specification, guide vane opening could be limited to a value that had a more *normal* surge, such as shown in Figure 24. A *normal* surge can produce extremely high pulsation, impact loads, shock waves, etc., so that occurrences must be limited using reliable anti-surge automatic controls.

The same probes were used for assistance in field commissioning with blow-off valve operation prior to production. The first point on the performance map gave a long surge pulse time and a single surge could be ascertained by stopping valve closure more quickly as compared to the shop test. Valve controls in the field gave ability to step the closure in 0.1 percent increments. Thus, a

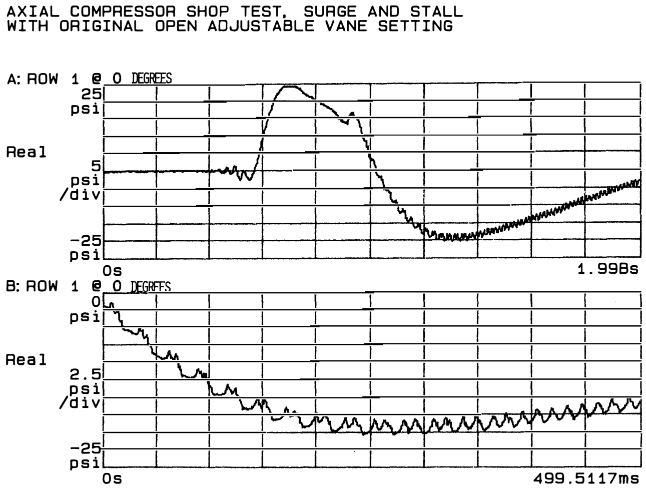

Figure 23. Time-Trace at First Row During Surge and Recovery.

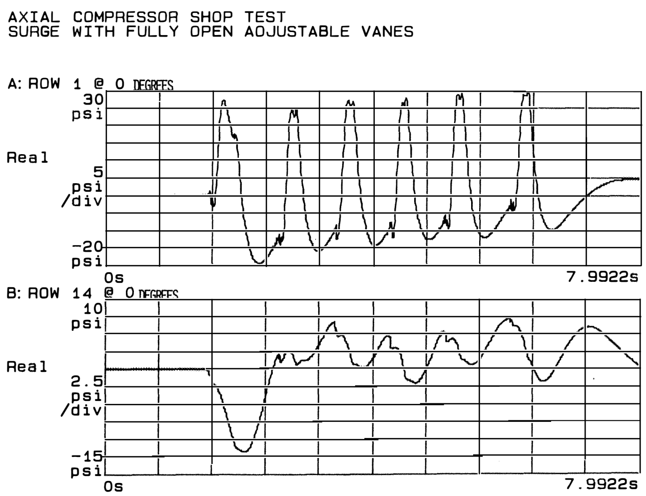

Figure 24. Time-Traces at First and Last Rows with Multiple Surges.

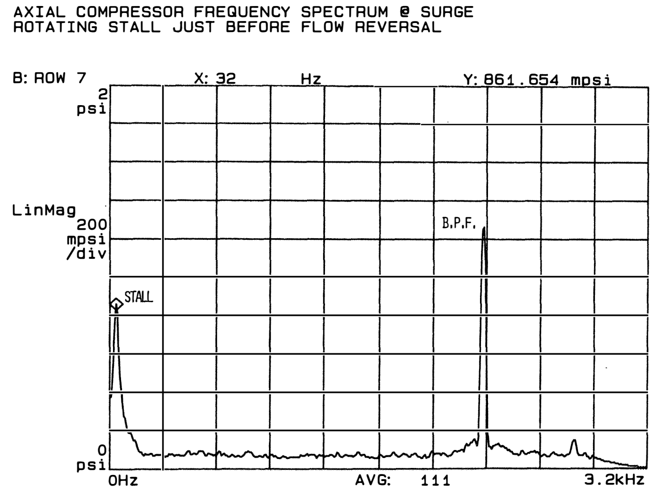

Figure 25. Spectra Showing One-Cell Pulsation at Middle Row.

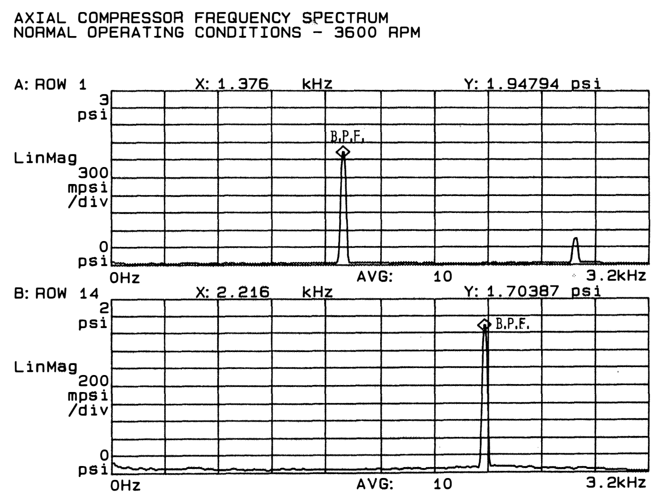

Figure 26. Normal Spectra at Front and Rear Rows.

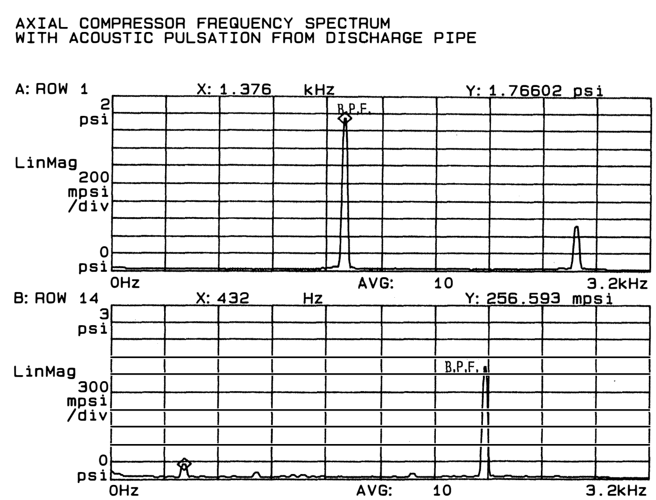

Figure 27. Spectra with 432 Hz Noise at Last Row.

trace of rotating stall could be found as shown in Figure 25 for open vane settings. Even though this was one cell, it was not detrimental, partly due to low sensitivity of the rotor/bearing system. Harmonics of stall frequency were also negligible in the range of frequencies that could excite blade natural frequencies. Away from surge points, pulsations were only at normal blade pass frequencies, Figure 26, and stall point settings gave optimum operational control line data.

An added value of dynamic pressure probes was that commissioning also showed an acoustic resonance. It was sensed as well with high levels at the discharge pipe and was audible, so that there was concern that it could be propagating upstream at a level that could excite the blades. Data verified frequency changed with speed of sound. Levels at the last rotating row plotted in Figure 27 showed that it was a standing acoustic wave at 432 Hz and could not cause blade excitation for the calculated number of waves. Amplitudes were much lower for the other probe in the last row, showing that a standing acoustic wave is the case, rather than some odd rotating stall.

DISCUSSION OF SEVERITY AND STALL MITIGATION

Relative to blade loading affects on fatigue, pulsation measurements can give values to be used for forced response analysis. Rotor vibration examples could also be used to determine whether calculations can be generated to set limits on nonsynchronous rotor vibration; as it is generally a forced, stable mechanism. For a given element, severity should be a function of flow raised to the third power; but for the same equivalent tip speed, would increase directly with power. For example, a reduced pressure performance test at correct equivalent tip speed showing a minor amplitude of 0.3 mil) $_{p-p}$ vibration could be as high as 1.2 mil)$_{p-p}$ for four times the flow at full field conditions, and still be acceptable if flow is to the left of the control line. Also, to be factored in would be the frequency experienced relative to the rotor/bearing system natural frequency. If test conditions give a frequency much different than projected field conditions, dynamic stiffness_would also be used to adjust calculations.

A pattern from centrifugal impeller testing emerged in that one-cell rotating stall correlated with indication that one per rev pulsation was higher than other harmonics.

Thus, a potential method is planned for testing wherein a multiple cell structure can be forced to occur. Hopefully, this idea, used for patent application, will compliment the use of stabilizer vanes and diffuser vanes.

CONCLUSIONS FROM TESTING

- Measurements using dynamic pressure probes are useful in identifying the true nature of stall inception, propagation and effects on system surge definition. Other flow and pressure instrumentation should be used in conjunction with audible noise, sensing of vibration, speed fluctuations, and load changes at peak head, etc.

- A volute stage can be properly designed to give negligible alternating blade loading on highly-loaded open inducer impellers, without damaging static pressure nonuniformity away from stall.

- Data show that centrifugal impeller stall frequency (propagating speed) decreased as flow is reduced, opposite to that for diffuser stall. Future tests should eventually better identify cause and effect relative to sources giving improved designs and easier modifications for problem-solving.

- Additional advances in CFD analytical techniques can be benchmarked using dynamic pressure probe data.

- Use of diffuser stabilizer vanes can eliminate concern for one-cell rotating stall caused by centrifugal impellers, and give efficiency improvements for volutes. Normal diffuser vane designs could also be used to change characteristics, generally only if flow range limitation is not a severe detriment.

- Data for centrifugal compressors also showed that predominate one per revolution pulsations may be an indicator of propensity for one-cell, in lieu of multiple-cell rotating stall. Only one-cell stall can give significant unbalanced loads to cause rotor vibration.

- Surge tests, especially for the axial compressor showed extreme pulsation levels, once again highlighting the concern for limiting occurrences. Adjusting discharge valve with dynamic pressure probes for surge point definition can be useful in defining true setpoints for automatic controls, minimizing number of severe surges. Setpoint data are best quickly logged and flow increased to remove any hysterisis.

- A procedure such as described for rotating stall severity calculations could save the industry from performing unnecessary full load tests for well-damped systems, along with identifying potential costly field problems.

- Dynamic pressure probes can be used for an added monitoring method for critical machinery; advances in data analysis should permit operation closer to surge, giving more flexibility and operational efficiency.

NOMENCLATURE

ω_r = rotor operating speed, Hz
ω_s = stall propagating speed in direction of rotor speed, Hz
n = number of rotating cells
f_s = first harmonic of stall frequency measured at a stationary point, Hz
f_r = first harmonic of stall frequency relative to each rotating blade, Hz
ϕ_s = measured phase angle between stall cells, degrees
ϕ_m = installed angle between stationary probes, degrees
α = tolerance on installed angle between probes, degrees
β = tolerance on phase angle measurement, degrees

APPENDIX

Aspects of Rotating Stall and Its Measurement

Compressor surge is the well known system instability succinctly described by Den Hartog [10], and later by Japikse [11]. Sometimes a more localized phenomenon, rotating stall, can be found just before or simultaneously with surge. Generally, stall is at the point where pressure rise peaks in one or more components and starts to give a positive slope on the system pressure *vs* flow curve. A sketch to indicate cells of low pressure alternating circumferentially is shown in Figure 28 with those cells higher than the average pressure. These revolve at ω_s, some fraction of rotor speed ω_r, almost always in the same direction from a stationary viewpoint. Relative to each rotating blade, the cells are propagating backwards at a speed of $(\omega_r - \omega_s)$. Thus sections of higher flow in turn provide more flow into blade areas that just had experienced a low flow condition. A steady state with this pattern can exist prior to in-phase reversal (surge). Each blade experiences a peak pressure at a rate of n multiplied by $(\omega_r - \omega_s)$; but any one-cell condition would give a net fluctuating force on the rotor at frequency of f_s.

For stationary measurements in a casing near the rotating component (s), dynamic probes can be used to determine number of cells and relative amplitudes, including harmonics for waveforms, not purely sinusoidal. Usually, two probes are used with one giving a circumferential reference for phase angle measurements. If only one probe is used, a frequency of f_s is obtained, but the number of cells and their propagating speed can be in question.

A phase angle measurement value near zero would show an axial pulsation, which would indicate a local or full surge type flow reversal. Usually a phase angle of $\pm$ 10 degrees for the second probe can be easily detected even for low amplitudes using frequency response functions (FRFs) from two-channel fast Fourier transform (FFT) analyses. Typical projected results are shown in Table 3 for selected separation angles of the two probes. A separation much less than 45 degrees could give questionable data in that pulsation frequency could really be surge, near zero degrees. With consideration of accuracy, 45 degree probe separation, installed to ± 2.5 degrees, can determine one to five cells, as shown for Case B in Table 4 using Equation (1).

$$\phi_s = n(\phi_m \pm \alpha) \pm \beta \tag{1}$$

Other factors to consider:

- Measured amplitudes should also be about the same for each probe at a given location.

- Instrumentation, especially filters must be checked for phase measurement errors.

Following equations also apply:

$$\omega_s = f_s/n \tag{2}$$

$$f_r = n(\omega_r - \omega_s) \tag{3}$$

Examples 1 and 5 in Table 5 show that there is some small risk that instead of one cell, there are nine cells. For more accuracy, a

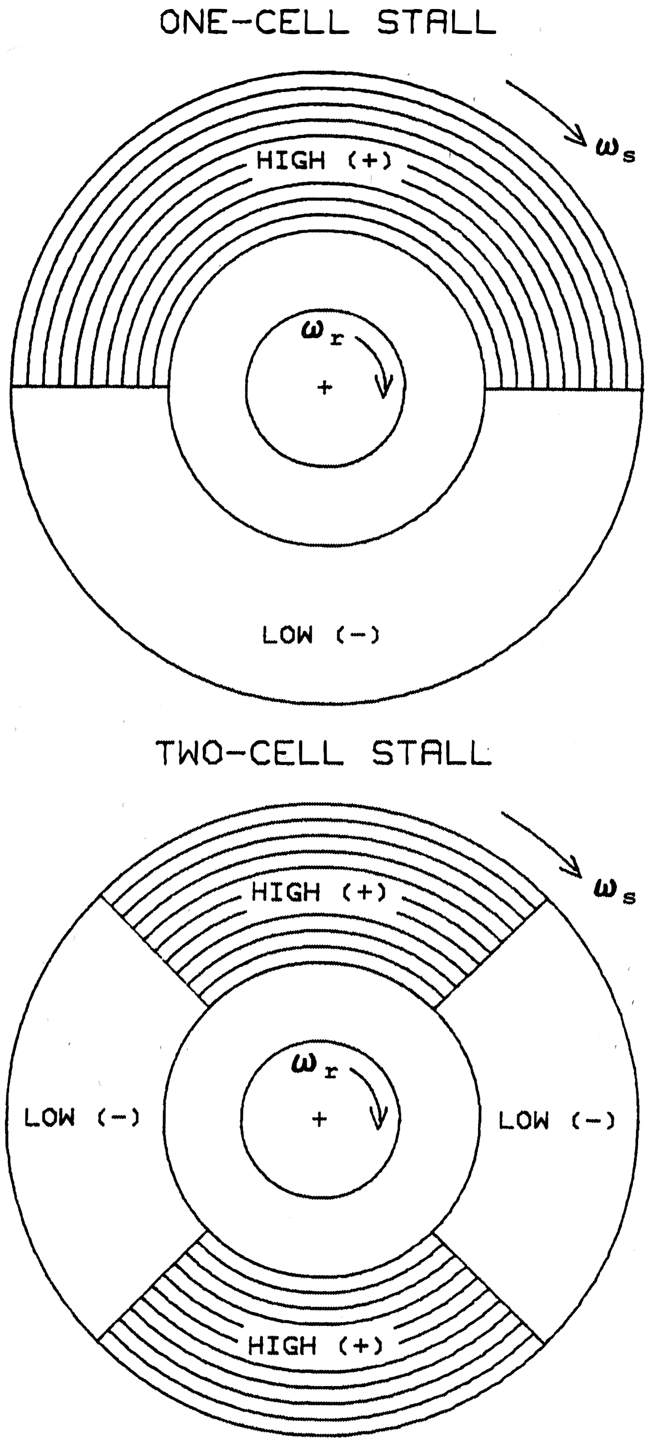

Figure 28. Sketch for One and Two Rotating Stall Cells.

third probe spaced 60 degrees from the reference probe gives a match if it shows near 60 degrees measurement needed for one cell; see bracketed values in Table 3. Nine stall cells would require this probe to indicate 180 degrees. However, rotor vibration also found during stall would add to certainty that there is only one stall cell, which gives unbalanced forces on the rotor as shown in Figure 28 at stall frequency, f_s. Likewise, if only one probe is used and the frequency is the same as rotor subsynchronous vibration, one cell can also be safely implied.

Table 3. Stall Number Determination for Separation Angle of Two Probes.

Probe Separation (degrees)	Measured Phase Angle (degrees)	Implied Number of Cells
180	180	1, 3, 5, 7, ...
	360	2, 4, 6, 8, ...
90	90	1, 5, 9, ...
	180	2, 6, 10,...
	270	3, 7, 11, ...
	360	4, 8, 12, ...
60	[60]	[1], 7, ...
	120	2, 8, ...
	180	3, 9, ...
	240	4, 10, ...
	300	5, 11, ...
	360	6, 12, ...
45	[45]	[1], 9, ...
	90	2, 10, ...
	135	3, 11, ...
	180	4, 12, ...
	225	5, 13, ...
	270	6, 14, ...
	315	7, 15, ...
	360	8, 16, ...

Table 4. Accuracy for Probe Spacing of 45 Degrees.

Range of Measured Phase Angle (degrees) Case A	Case B	Implied No. of Stall Cells
350 - 10	350 - 10	0
30 - 60	32.5 - 57.5	1
70 - 110	75 - 105	2
110 - 160	117.5 - 152.5	3
150 - 210	160 - 200	4
190 - 260	202.5 - 247.5	5
230 - 310	245 - 295	6
270 - 0	287.5 - 342.5	7
310 - 50	330 - 30	8
350 - 100	12.5 - 77.5	9
30 - 150	55 - 125	10

Case A $\alpha = 5°$; $\beta = 10$ **Degrees**
Case B $\alpha = 2.5°$; $\beta = 10$ **Degrees**

Examples 3 and 5 in Table 5 could show concern for an axial compressor or open (non-covered) inducer type impeller, e.g., the frequency relative to blading for Example 3 is approaching three per revolution; the second harmonic at 300 Hz could especially be near a fundamental blade mode. Excitation at rotating stall could add to other sources, especially critical if the blade mode is also at a harmonic of rotor speed. Again, more probes at different spacing could be necessary for convincing analysis.

Table 5. Examples of Rotating Stall Analysis.

	Two Probes 45 Degrees Apart				
Example No.	**1**	**2**	**3**	**4**	**5**
Rotor Operating Speed -Hz	**60**	**60**	**60**	**60**	**60**
Measured Frequency -Hz	**45**	**84**	**30**	**6**	**45**
Measured Phase Angle -Degs.	**45**	**90**	**135**	**45**	**45**
Number of Cells	**1**	**2**	**3**	**1**	**9**
Cell Rotating Speed -Hz	**45**	**42**	**10**	**6**	**5**
(Percent of Operating Speed)	**75**	**70**	**17**	**10**	**8**
Rotating Blade Excitation (1st Harmonic Frequency) -Hz	**15**	**36**	**150**	**54**	**495**

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