COMPRESSOR DISCHARGE PIPE FAILURE INVESTIGATION WITH A REVIEW OF SURGE, ROTATING STALL, AND PIPING RESONANCE

by

Frank Kushner

Senior Consulting Engineer

Elliott Company

Jeannette, Pennsylvania

Doug Walker

Senior Reliability Specialist

The Dow Chemical Company

Plaquemine, Louisiana

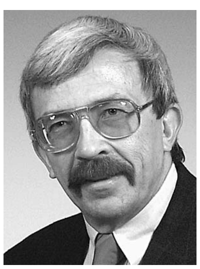
and

William C. Hohlweg

Senior Engineer

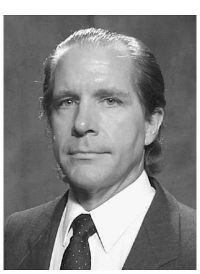
Elliott Company

Jeannette, Pennsylvania



Frank Kushner is a Senior Consulting Engineer for dynamics and acoustics testing at Elliott Company, in Jeannette, Pennsylvania. He has 33 years' experience with industrial turbomachinery, and four years' previous experience with the combustion section development group at Pratt & Whitney Aircraft. He is a previous author for the Ninth, Twenty-Fifth, and Twenty-Ninth Turbomachinery Symposia, as well as for ASME.

After obtaining a BSME degree (1965) from Indiana Institute of Technology, Mr. Kushner received his MSME degree (1968) from Rensselaer Polytechnic Institute. He is a registered Professional Engineer in the State of Pennsylvania, and a member of ASME and the Vibration Institute. Mr. Kushner holds patents for a blade damping mechanism, and also for a method to prevent one-cell rotating stall in centrifugal compressors.



Doug Walker is a Senior Reliability Specialist at The Dow Chemical Company, in Plaquemine, Louisiana. He has 23 years' experience with turbomachinery operations, maintenance, and reliability.

Mr. Walker holds a BSME degree (1977) and an MBA (1978) from the University of Central Florida.

William C. Hohlweg is a Senior Engineer in the Advanced Technology Department at Elliott Company, in Jeannette, Pennsylvania. He is the Supervisor of the Aerodynamics Group and is responsible for design and development of centrifugal compressor and axial turbine staging. This includes single-stage testing, ongoing improvement of the application computer programs for the multistage product lines, and aero performance consultation. He has been with Elliott for 26 years, and has specialized in centrifugal compressor performance both in the Development and Product Engineering departments. Prior to that, he was employed at Ford Motor Company and NASA Langley Research Center.

Mr. Hohlweg received his B.S. degree (Aerospace Engineering, 1971) from the Pennsylvania State University and an M.S. degree (Flight Sciences, 1975) from George Washington University. He has authored or coauthored eight technical publications for ASME, NASA, and I Mech E.

ABSTRACT

Rotating stall, even with multiple cells, can contribute to piping excitation due to acoustic energy traveling down the pipe as a plane wave. The main objective of this paper is to describe causes and solution of fatigue cracks in discharge piping due to two forms of acoustic energy, with occasional direct surge loading of a 50,000 hp centrifugal compressor. Described are transverse modes of the gas inside pipes, as well as structural mode interaction of pipe shell

modes. Also reviewed are frequency and force evaluation for two other cases of one-cell, rotating stall due to impeller eye incidence that caused rotor vibration in high-pressure centrifugal compressors.

Single-stage rig tests, correlated with analytical results including computational fluid dynamics (CFD), are essential to validate aerodynamic design calculations for new impeller stages. Some typical test results are shown to confirm why a design not only is accepted, but also can occasionally be rejected.

INTRODUCTION

It is extremely important in design of compressors that rotating stall will not occur except just before surge of the compressor system. If not, a redesign is sometimes the only recourse as shown by Sorokes and Trevaskis (1997) and by Fulton and Blair (1995). A nonrotating stall due to adverse pressure gradients in some component will often initiate compressor system surge. However, sometimes the stall forms rotating cells during or prior to surge; it is presently impossible to avoid rotating stall just prior to surge for all designs. What must be avoided is one-cell rotating stall with high amplitudes of unbalanced pressure in the normal operating flow regime greater than the surge control line. For extremely high-pressure centrifugal compressors, setting the surge control line with persistent one-cell rotating stall excitation could also present problems with high rotor vibration (Kushner, 1996).

One-cell rotating stall is the only type that will give significant unbalanced gas loads on the rotorbearing system. The unbalanced pressure forces, schematically shown in Figure 1, give a forced excitation to the rotorbearing system. There will be net unbalanced fluctuating radial loads on the rotor such that shaft vibration probes would show a response at a subsynchronous frequency equal to rotating speed of stall cells. It gives a forced response, not an instability at a natural frequency as would occur from aerodynamic cross-coupling forces that overcome positive damping. One-cell rotating effects are similar to mass unbalance:

- Excitation is a function of pressure instead of mass times speed squared;
- Response is dependent on speed, location of the impeller stage with stall, and rotorbearing properties.

Experience has been that almost every compressor can accommodate one-cell stall if it occurs at incipient surge or during surge flow reversals.

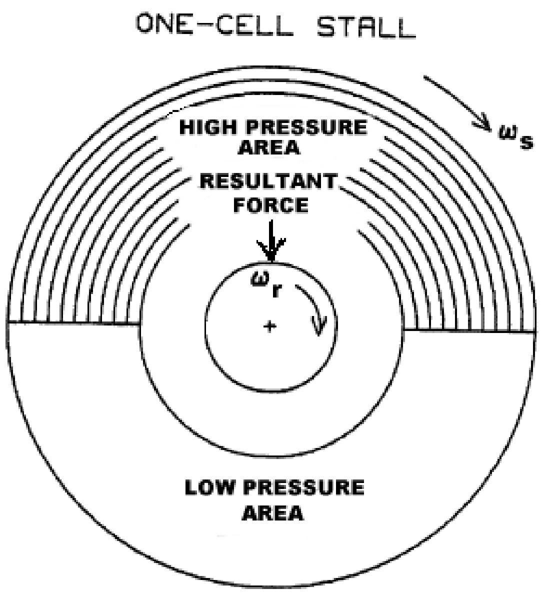


Figure 1. One-Cell Rotating Stall Schematic with Unbalanced Forces; ω_r is Forward Rotor Speed and ω_s is Forward Stall Speed.

When higher numbers of cells occur, such as two-cells shown in Figure 2, the rotor will only respond with minor amplitudes. There is a concern for axial compressors and open-impellers in centrifugal compressors for blade excitation from multicell rotating stall (Kushner, 1996). However, there is generally no concern when using covered impellers, since natural frequencies are so high, as shown by Kushner (2000). Also, Borer, et al. (1997), concluded that although rotating stall was measured during testing, it was not found to be the cause of covered impeller fatigue failures.

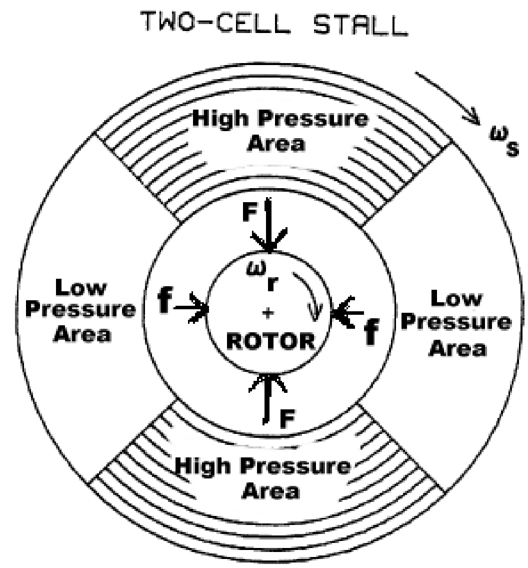


Figure 2. Two-Cell Rotating Stall Schematic with Balanced Forces; ω_r is Forward Rotor Speed and ω_s is Forward Stall Speed.

For axial compressors in gas turbines, compressor flow stability is extremely important as emphasized by Mazzawy (1980). Oscillations can result in severe damage to the mechanical components, produce unsteady thrust loads, or cause ingestion of combustion gases into the inlet. For jet engines, reversed flow through the compressor is taboo as it can extinguish combustion burners, causing total loss of power. The documentary film by Boeing (1993) of their Model 777 passenger airliner showed a dramatic engine flameout and reignition at takeoff during engine testing. Compressor surge at takeoff was reported to be quickly resolved; with structural modifications to the inlet casing by Pratt & Whitney Aircraft Company to prevent excessive rotor tip to casing clearance. In industrial axial compressors, cantilevered blading can withstand a very limited amount of surging; thus, automatic antisurge controls are used for all process compressors. For centrifugal compressors, the system determines the need for automatic controls, which are standard for integrally geared plant air compressors and critical process units with flow range that require high turndown. Some units such as hydrogen recycle compressors can employ simple manual controls, giving potential for surge only during unplanned trip-out due to system upsets.

Another concern for compressor systems is resonance of natural frequencies of piping systems. High frequencies, especially from rotating blade passing, are far above longitudinal modes in the gas and low-frequency bending modes for pipe runs. However, blade-passing frequency, especially from centrifugal compressors, can cause resonance of interior transverse gas modes and/or modes of the pipe wall as first emphasized by Seebold (1972). A high piping noise level at impeller blade-passing frequency is usually not a reliability concern, but rather one where ambient noise levels must be limited. The specific case of piping fatigue, to be detailed later in this paper, is different. Rotating stall, near surge, generated acoustic excitation that not only added to vibration, but also modified vibration energy at high frequencies.

DOCUMENTATION OF ROTATING STALL EFFECTS ON ROTOR VIBRATION

During investigation of rotorbearing instability on the 10,000 psi compressor for natural gas reinjection described by Geary, et al. (1976), there were some low amplitude odd frequencies for the original build. The instability problem from cross-coupling effects causing high response at the first rotorbearing mode frequency was solved, first with damper bearings and then with an improved, stiffer rotor design. The odd frequencies were also eliminated, likely due to stage design revisions after review of potential diffuser stall, characteristics of which were later published by Abdelhamid, et al. (1978). The customer had ordered a backup compressor from another manufacturer that did not have to be installed as the problem was solved. Testing of the backup unit was eventually reported by Ferrara (1977) to have odd response at three asynchronous frequencies with a suggestion that rotating stall was a likely cause. At that time, after review of other reports, it was concluded that only rotating stall of the one-cell variety could be the type to give significant rotor vibration. Then in 1993, an impeller design, with extensive operational history, experienced impeller stall that caused subsynchronous rotor vibration in a natural gas reinjection compressor described by Kushner (1996). Rotor vibration occurred at a frequency varying from 70 to 80 percent of rotating speed, only when approaching system surge flow. Because of the concern for higher vibration at surge causing a potentially damaging rotor rub when setting surge controls in the field, it was agreed to modify the rotor and its spare. Rotor subsynchronous vibration at part load is shown in Figure 3. Note that when the subsynchronous vibration initiates, it is at a frequency that is slightly less than operating speed (in line with the annotation "start" shown in Figure 3), then quickly decreases to a stable frequency near 80 percent of running speed. This symptom is a clue that the stall initiates within the impeller, i.e., first traveling at impeller speed. Response at the rotorbearing first critical frequency was extremely low; however, the forced excitation frequency was close enough to be somewhat amplified by the system natural frequency for the fairly stiff rotor. Future operation of the reinjection unit would also require a wide speed range so that subsynchronous frequency could often be close to the first rotorbearing mode.

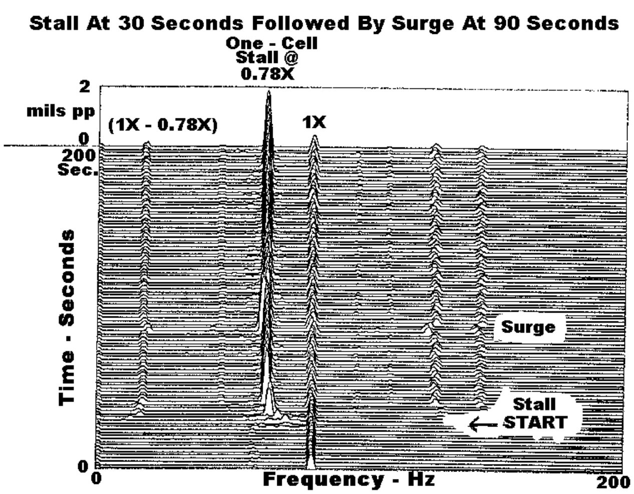


Figure 3. Cascade Plot of Gas Reinjection Compressor —Rotor Vibration with Amplitude Near 2.0 mils (Peak-to-Peak) for Subsynchronous Frequency at 0.78 ×.

The impeller design was modified while at the same time rig testing was used to confirm the cause of the problem. It was also shown in testing that a vaned diffuser would eliminate impeller stall due to initiation of stall in the diffuser before the impeller had an opportunity to stall. However, flow range with the vaned diffusers would have been too limited. Therefore, impeller blade leading edge angle was changed to reduce the positive incidence of low flow, and thus shift onset of impeller rotating stall closer to compressor system surge flow. This modification was expeditiously completed, allowing surge control line to be established in field tests, without tripping on high vibration.

A more recent similar case for a process compressor occurred where part load tests in the shop showed subsynchronous vibration near 85 percent speed. The rotating stall only occurred just prior to surge; so the risk was evaluated for field operation. Calculations were made for changes in the level of forced excitation to be expected for field conditions; based on a forcing function directly related to gas density times the square of speed. There is a similar comparison given by NASA (1995) for comparing vibrational power as a function of flow to the third power divided by mass, confirmed by test data on five rocket turbopumps. In addition, comparisons were made for forced response, by equating it to mass unbalance force for excitation frequency relative to rotorbearing first mode for test and field conditions. The proper decision that rotating stall was not a reliability problem was verified when antisurge controls were set with no problems in both the commissioning of three identical compressors, and also in their operation for the past four years.

ROTATING STALL EVALUATION IN DESIGNING CENTRIFUGAL COMPRESSORS

Centrifugal compressors are used in a wide variety of applications throughout the petrochemical industry. They are often chosen because of their mechanical ruggedness and reliability. Also, process gas compressors frequently require a wide operating flow range. This is needed to accommodate variations in process operation, field pressures and gas compositions, seasonal changes, and end user demand. To meet this need, designers today are employing the latest, state-of-the-art computational fluid dynamics (CFD) computer programs, as described by Hardin and Boal (1999). CFD is now used routinely to evaluate the compressor flow field with the goal of maximizing design point efficiency, and delaying the onset of flow separation at off-design, lower flowrates. For a new impeller design, at a given tip Mach number, blade geometries are specified along the hub and cover, from inlet to tip, with the purpose of controlling the rate of change of the suction and pressure side velocities (or static pressure). Also, the hub and cover contours are determined based on flowrate, pressure ratio, and stage spacing requirements. Contour curvature and inlet to tip area schedule must be closely examined in order to reduce or eliminate regions of flow reversal. CFD calculations are run on new geometries at design and several off-design flows with the intent of defining the performance qualities from maximum flow (choke) to surge flow. Since surge and rotating stall are unsteady, transient phenomena, CFD, and other more empirical 1-D analysis methods cannot directly predict the flowrate where surge or stall will occur. Still, they can often give good estimates and provide the designer guidance on when a design is approaching a rotating stall situation. Aungier (2000) provides a simple indicator of impeller inducer stall related to the flow diffusion between the inlet and blade passage throat. Likewise, by utilizing proper indicators for stage design, hazardous one-cell stall due to diffusers has been able to be avoided for flows above the desired surge control line, unlike the cases described by others, e.g., Ferrara (1977), Fulton and Blair (1995), and Kuzdzal, et al. (1994).

Also, surge is dependent on the specific system piping arrangement that is not normally modeled during the stage design process. However, as the compressor flowrate is reduced, CFD can indicate the growing presence of localized areas of low momentum fluid and regions of separated or reverse flow. These internal "pictures" of the compressor design have been successful in estimating where the minimum flowrate will occur, and the likelihood of rotating stall. Still, only detailed measurements during the prototype testing can determine the exact flowrate where rotating stall occurs, the number of rotating cells, their frequency, and the likely compressor component that initiated the stall. The location of full system surge and the peak of the head curve are also defined during the test. These data are then used to establish aerodynamic performance ratings, for families of similar stage designs that are applied in a wide range of industrial compressor applications. CFD analysis has helped to improve the probability of success for new designs, thus minimizing or eliminating any costly redesign activity.

In Figure 4, performance test results are compared for two stages designed for similar flow and head coefficients. Both had 3-D, ruled surface impeller blade geometry that attempted to control the hub to shroud incidence angle at the leading edge, and blade loading levels from inlet to tip. Stage A was designed without the benefit of CFD analysis but utilized proven, quasi-3D gaspath and empirical performance prediction programs. Stage B was designed with the same techniques as for stage A, but also benefitted from concurrent analysis of the impeller with a commercial CFD code. The test results show a clear difference in the location of the peak head and also the flow coefficient where rotating stall was measured (testing details discussed below). During a later postmortem evaluation of stage A, CFD analysis at the flow point where stall initiated on test, indicated a severe pattern of reverse flow along the shroud, suction side of the blade. Also, early signs of flow separation were evident near the design flow point. Had CFD been used during the design phase, it is obvious this stage would never have been built and tested in that configuration. Efforts would have been made to eliminate or minimize the reverse flow regions, especially at design flow. Figure 5 compares impeller A and B velocity maps at the stall flow point of each stage. Note the much larger recirculation zone along the shroud wall for the rejected stage. It is believed the higher rate of impeller diffusion and the tighter shroud curvature caused impeller A to stall much sooner than impeller B. Blade loading was not considered a factor since the A impeller, with 21 blades, had lower loading rates than the B impeller, with 15 blades.

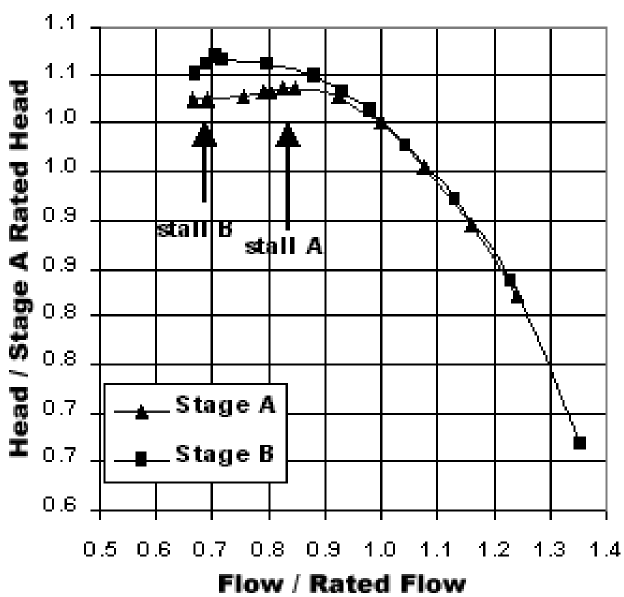


Figure 4. Tested Headrise for Rejected Stage A and Accepted Stage B at Tip Mach Number = 0.7.

The widest flow range for a centrifugal compressor stage, without any variable control devices, is achieved with the vaneless diffuser. There is generally an optimum range of diffuser flow angles that are targeted for the design point in order to achieve a

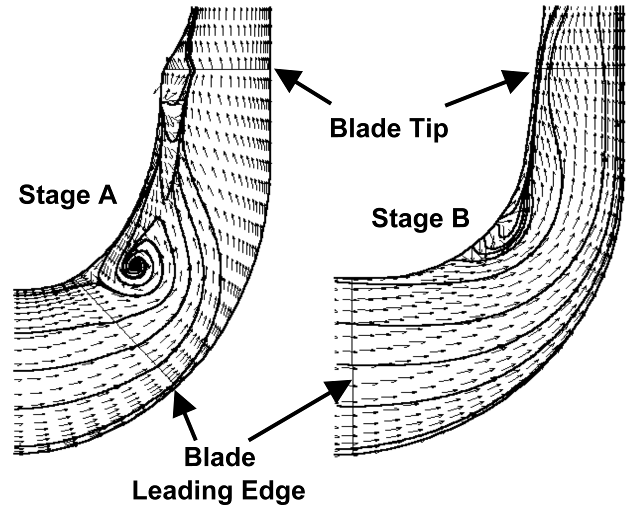


Figure 5. CFD Velocity Vector Comparison at Measured Stall Flow.

balance between good efficiency and maximum surge to choke range. Flow angles of 30 to 35 degrees (from tangent) at the diffuser exit are typically used that allow for at least 30 percent stability range for many process gas applications. With this criterion, one-cell diffuser stall is normally avoided for the stage design speed. However, it is not uncommon for multicell diffuser stalls to appear prior to the peak of the head curve that have no discernable effect on the overall stage performance. Sorokes and Trevaskis (1997) provided a good review of the various published methods to estimate the onset of diffuser stall.

Stage performance must always be checked at the higher, off-design speeds, to ensure there is an adequate flow margin to surge. For a given stage design operating at much higher impeller tip Mach numbers, the diffuser flow angle will decrease as speed is increased. This can result in the diffuser stalling before the impeller. In this situation, the stage is often sized with a narrowed diffuser width that boosts the flow angle back up to the levels near the original design speed. This helps to maximize the flow range to surge and also preserve the pressure rise to surge. This procedure will move the diffuser stall point to lower flows such that often the impeller will initiate the stall first. Of course if the impeller is determined, by test measurements, to already initiate the rotating stall at higher Mach numbers, then narrowing the diffuser may have little or no effect on the surge margin.

As mentioned above, experimental data are required to accurately assess the rotating stall phenomena in each new stage design. During these tests, proper measurements of frequency response and phase differences of three dynamic pressure transducers are used to accurately define when a rotating stall pattern occurs. These probes are located in the diffuser passage, on one wall, just downstream of the impeller discharge, and sometimes at the impeller eye as in Figure 6. Frequency analysis can determine the number of stall cells and also the likelihood whether the rotating stall is caused by the impeller or diffuser, or due to an interaction of both. Frigne and Van den Braembussche (1984) first described the unique differences in the rotating stall frequency range for the impeller and diffuser. Van den Braembussche (1984) gives additional pioneering summaries of rig test results. Some characteristics for more modern designs are shown by Kushner (1996); included is a detailed procedure for phase analysis. Stall caused by the diffuser usually gives very low stall speeds, in the range of 3 to 20 percent of rotor speed. Diffuser-impeller interaction generally gives higher speeds in the range of 30 to 60 percent, and impeller stall at or near blade leading edges gives high speeds in the range of 60 to 95 percent of rotor speed. Evidence of the component that is causing the stall is usually given by changes in rotating speed of the stall as flow is reduced further past the initiation point. Typically, stall speed increases for diffuser stall as flow further reduces, and vice-versa for impeller blade incidence stall. It should be noted that these characteristics are typical of acceptable designs from this OEM; others could have somewhat different characteristics, e.g., refer to Marshall and Sorokes (2000). Particular attention is paid to any occurrence of one-cell rotating stall patterns; number of stall cells also can change with either flow or speed variations. One-cell rotating stall is the only pattern capable of exciting a rotor by the unbalanced aerodynamic force. Thus, during these tests, the bearing vibration probe measurements are also monitored and recorded for any sign of increased rotor displacement.

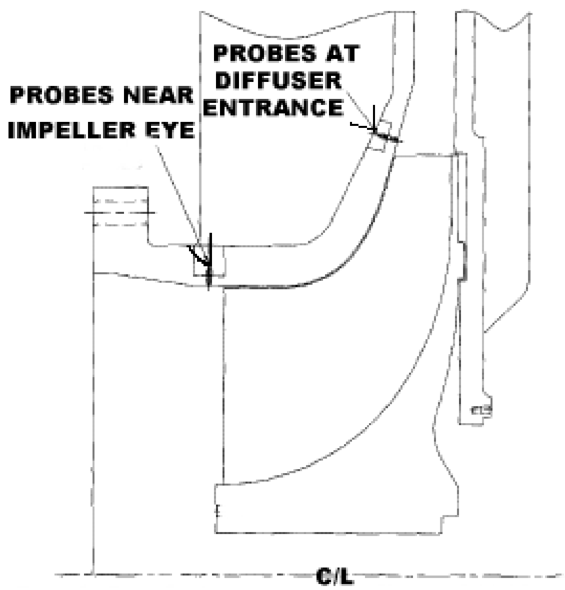


Figure 6. Schematic Showing Radial Location of Dynamic Pressure Probes; Typically Three Probes at Each Location at 0, 45, and 60 Degrees for Phase Reference.

Results of the dynamic pressure transducer measurements for the test stages discussed above indicated that both impellers exhibited one-cell rotating stalls. In impeller A, at design tip speed, one-cell stall was verified by phase analysis, and indicated the cell was rotating at 65 percent of the compressor rotor speed. This confirmed that the stall initiated within the impeller. In Figure 7, the pressure pulsation frequency at 50 Hz was indicated first; then as the flow was lowered, another pulsation at 4.5 Hz appeared. The second pulsation was due to the diffuser, which had a two-cell rotating stall at 4.5 Hz, traveling near 3 percent of rotor speed. Note the significant rotor vibration at 50 Hz, but negligible response at 4.5 Hz frequency. CFD analysis after the test had shown that the stall initiated near the middle of the meridional passage and propagated back toward the eye as flow was reduced. This may be why the stall speed was lower than the 80 to 95 percent impeller stall that is typically found when stall is initiated at the blade leading edge. During tests for accepted Stage B, at design tip speed, the dynamic pressure probes showed there was a two-cell rotating diffuser stall that developed, immediately followed by a one-cell impeller stall with frequencies as shown in Figure 8. The two-cell diffuser stall had a frequency of 9 Hz or 0.06 times speed, and thus was rotating near 3 percent of rotor speed (0.06 × / 2 = 0.03 ×). The impeller one-cell stall, with frequency equal to 139 Hz or 0.83 times speed, was rotating at 83 percent of rotor speed. There was no significant rotor vibration at the corresponding frequency of pressure pulsation at 9 Hz, whereas the one-cell stall did show a mild response. At higher impeller tip Mach numbers; there was only multicell diffuser stall just before surge. Surge is the condition of full flow reversal that was proven by the fact that pressure pulsations at surge frequency were in phase for all three dynamic pressure probes.

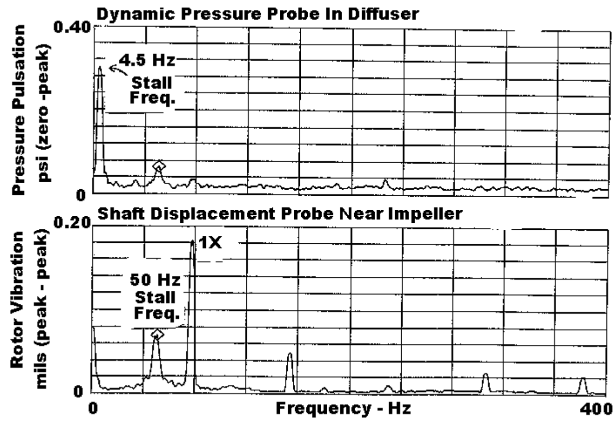


Figure 7. Pressure Pulsation and Rotor Vibration Frequency Spectrum after Peak of Head Curve for Stage A.

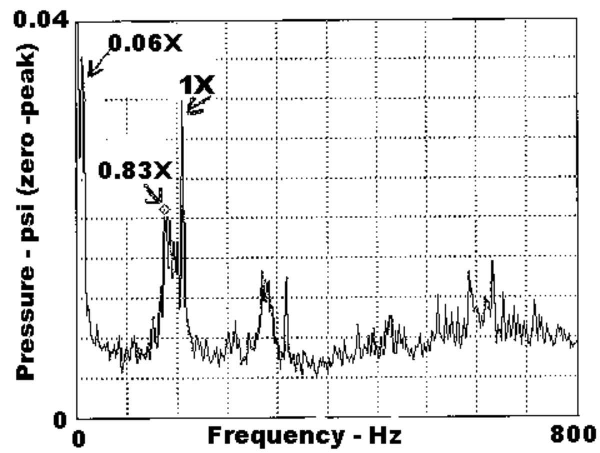


Figure 8. Pressure Pulsation—Frequency Spectrum Analysis at Peak of Head Curve for Stage B.

DOCUMENTATION OF SURGE PULSATIONS

It is well known that compressor surge should be avoided as much as possible, and also that surge control can often be difficult as shown by Locke (1984). Actual data to show the severity of surge is herein presented for both an axial compressor and a multistage centrifugal unit. The pulsations in Figure 9 were for an axial compressor under test with inlet pressure (and power) about one-third of field conditions (25,000 hp and 3.6 pressure ratio). The extremely high pulsations at the front end are especially conducive to breaking blades or stator vanes and damaging variable linkage mechanisms with accumulation of enough cycles. The pulsation pattern certainly is not a sine wave; so besides impact-type forces that likely include shock waves, there will be harmonic content to excite blades especially if there are multiple surge pulses. Thus antisurge controls for this unit in the field used a quick-acting blowoff valve. In fact, assistance was given during commissioning, using dynamic pressure probes near blade tips, in order to avoid surge during the surge control line determination. The blowoff valve was very slowly closed, using 0.10 percent increments. Then the valve was opened as soon as there was any sign of rotating stall, for which data were used from shop tests where it had been shown to only occur immediately before surge. It was decided by the user not to subject the compressor or the piping to severe pulsations even during commissioning. Linden and Parker (1996) describe a similar field test.

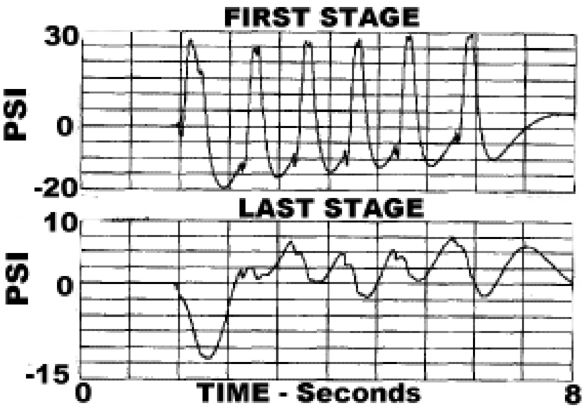


Figure 9. Simultaneous Pressure Pulsations During Part-Load Testing at Front and Rear Stages for a 14-Stage Air Axial Compressor.

In the early 1970s four duplicate 30,000 hp barrel compressors for mixed-refrigerant gas were experiencing numerous failures of covered impellers at a natural gas liquefaction plant. The cause of the failures was eventually found after using a task force including many consultants. One renowned consultant, who gives a description of system surge in Den Hartog (1954), professed that it was the most difficult problem that he ever worked on. Many believed that the cause was excessive surging, but strain gauges on the impellers eventually proved the direct cause was liquid ingestion (refer to Kushner et al., 2000; and Kushner, 2000). However, surging could have assisted in crack propagation, at least initially when failures were more frequent and surging lasted for long periods (reportedly up to 20 minutes). Strain gauge data verified that intentional surging only slightly excited impeller natural frequencies but did give transients for direct flow reversal, with corresponding pressure and thermal loads (Figures 10, 11, 12). Note that the data are for nine consecutive surge pulses close to two seconds apart, and then about 100 seconds of recovery followed by 11 additional flow reversals. In addition, centrifugal stresses were also part of the changes in impeller stress variations shown in Figure 13. This was due to the wide change in the steam turbine driver speed from 6400 to 5800 rpm, which during surge reverse flow caused immense torque reversals.

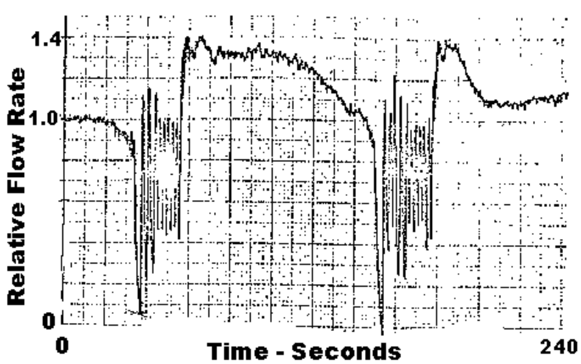


Figure 10. Multiple Surges of Mixed Refrigerant Compressor Showing Gas Flow Variations.

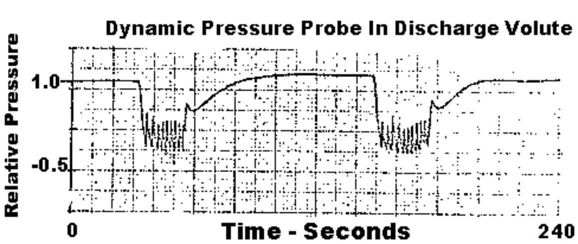


Figure 11. Multiple Surges of Mixed Refrigerant Compressor Showing Pressure Pulsations in Discharge Volute.

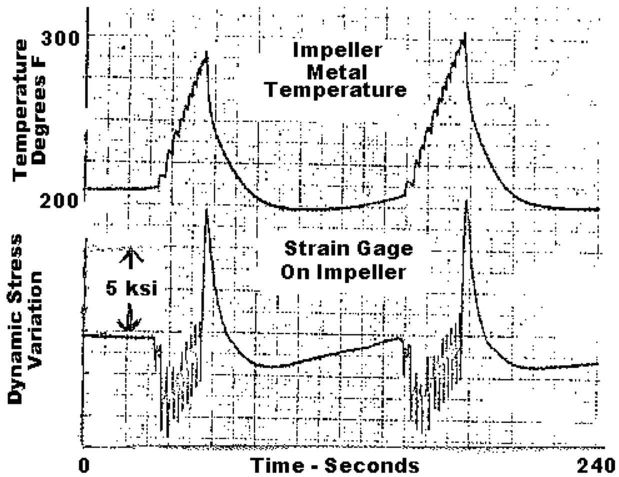


Figure 12. Multiple Surges of Mixed Refrigerant Compressor Showing Last Stage Metal Temperature and Dynamic Stress near Crack Initiation Point.

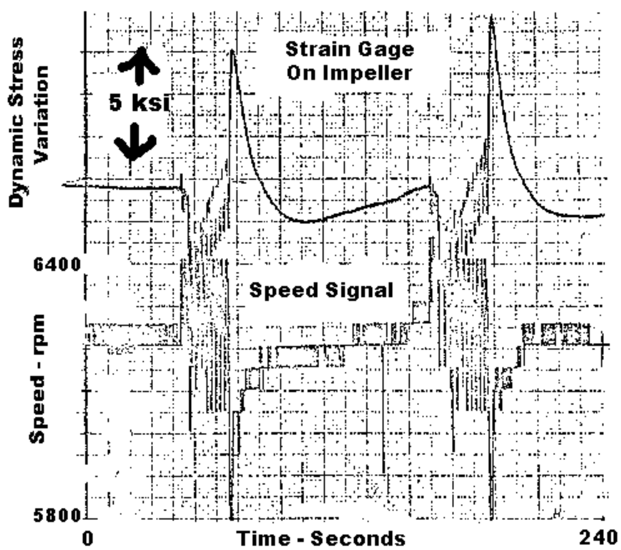


Figure 13. Multiple Surges of Mixed Refrigerant Compressor Showing Last Stage Dynamic Stress near Crack Initiation Point and Rotor Speed Variations.

REVIEW OF CONCERNS FOR PIPING RESONANCE

For a well-designed piping system for centrifugal compressors, there is low likelihood of piping fatigue failures. Energy levels are much lower than for other compressors, especially compared to screw compressors. Reciprocating compressors have very low frequencies where longitudinal modes of the gas must be evaluated, with often a need for pulsation dampers. For properly designed centrifugal systems, normal turbulence-induced vibration cannot be avoided, but is easily accommodated in the system. Noise related excitation due to vortex mechanisms are well within limits; while the most important noise source, rotating blade-passing frequency, normally does not give cause for concern.

For a particular design, noise level at blade-passing frequency increases as mass flowrate, power, and impeller tip Mach number increase, and also varies along the operating map due to incidence angle and wake variations. A vaneless diffuser stage will also typically be much lower in noise than vaned since there is only interaction with inlet vanes and casing walls and not diffuser vanes; refer to data by Motriuk and Harvey (1998). Of course, the fundamental design procedure must not use equal numbers of stationary and rotating vanes, as lining up of wakes would give the effect used in some emergency sirens and could excite different disk vibratory modes. It has been known at least since the early 1970s that part of the reason for the wide disparity in noise levels near centrifugal compressor piping is piping gas mode resonance with rotating blade-passing frequency. Since review of 1974 API Task Force data given by Stein (1980), many more systems have piping treated due to hearing conservation, residential noise limits, and also because of a steady increase in design speeds and power per compressor.

Seebold (1972) explains transverse modes (phase changes across the pipe wall) and gives an example showing at least a 10 dB increase at resonant peaks. Applications with lower mol weight gas applications have the best chance of resonance of the more-excitable modes due to higher sonic velocities. These applications typically are at lower power and run at lower tip Mach numbers, which helps to minimize concerns for resonance. Increase in ambient noise levels can still be accommodated with proper piping treatment (Stein, 1980; and Frank, 1995). In Figure 14, resonance is indicated whenever compressor blade-passing frequency intersects one of the modal frequency lines. A representative example possible for a light gas, centrifugal compressor is shown in Figure 15. There are also structural bending modes of the pipe wall that can be excited, but the chance of phase matching for both the gas and the wall modes is extremely small. This is due to typical optimum selection of compressor and piping design variables for aerodynamic and other structural considerations. Good descriptions of transverse modes are in Beranek (1971) and Seebold (1971) who also discuss structural modes of pipe walls. Jungbauer and Blodgett (1998) and Price and Smith (1999) provide descriptions and calculation procedures for piping resonance.

Normally, resonance of higher modes of the gas or structural modes of piping only gives slightly higher ambient noise levels without concern for potential failure. However, the recent trend is to use lower numbers of rotating blades, especially with high efficiency, 3-D full inducer impellers. This gives higher piping noise levels since lower, more excitable modes of the gas can be in resonance. This was the case for a methane gas compressor that required a dissipative silencer in the discharge pipe due to a residential noise problem at plant boundaries. There was also a fairly low blade-passing frequency for the propylene compressor, the main problem case in this paper described below. Along with the added consequence of rotating stall at the last stage of a centrifugal compressor, forces were sufficient to cause fatigue failures that had to be corrected.

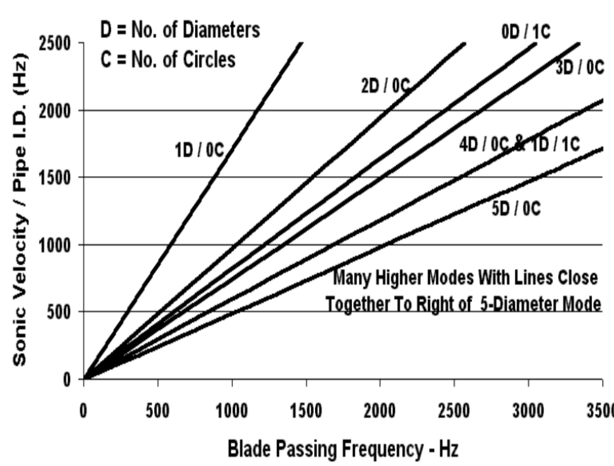


Figure 14. Lines of Resonance for Gas Transverse Modes Inside Long Straight Circular Pipes.

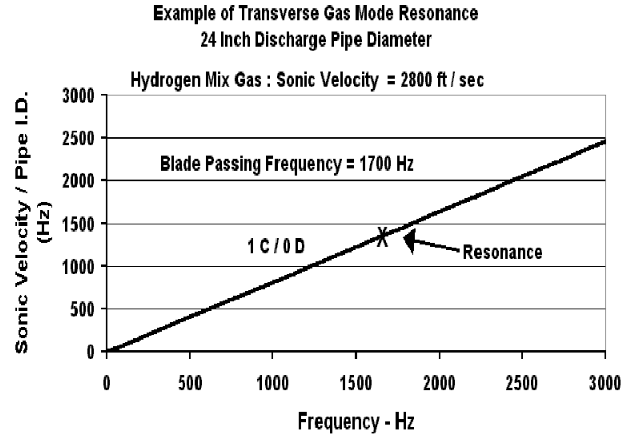


Figure 15. One-Circle Mode Resonance for Discharge Pipe from Compressor Handling Low Mol Weight Gas.

PROCESS COMPRESSOR PIPING FAILURE ANALYSIS AND CORRECTION

There have been some unusual piping vibration problems such as those reported by von Nimitz (1975). Two cases in that report were solved with structural modifications. Causes of vibration were not determined to be due to multicell rotating stall. However, they could have been based on the case reported herein for compressor discharge piping excitation that occurred for a 50,000 hp propylene refrigeration compressor in an ethylene plant. Normally, abnormal rotor vibration is at a subsynchronous frequency. In this case it was supersynchronous, near 55 Hz or 1.1 times rotor speed of 50 Hz (3000 rpm). Amplitude levels were very low on the rotor, less than 0.20 mils peak-to-peak, but the frequency gave evidence of rotating stall, one of the causes of discharge piping fatigue failures. Also, higher internal piping noise amplitudes at blade-passing frequency were found to be a direct cause of piping failure. These amplitudes were sometimes modulated by the 55 Hz acoustic waves resulting in even higher amplitudes. The original unit, commissioned in 1978, was driven by a two-shaft gas turbine (main driver, through a speed reduction gearbox), and a steam helper turbine at the opposite end. In 1992 the driver configuration was changed to a single 60,000 hp steam turbine. To accommodate this driver change, the drive-through shaft on the compressor was removed and the outboard bearing housing modified with an end cover. Although the rotor was switched, no gas path modifications were made. The machine operated successfully in this configuration. However, in 1994 the first piping failure occurred on a short piece of bleed piping downstream of a block valve; failure analysis determined the cause to be fatigue. In 1995 the second failure occurred on a piece of instrument tubing just downstream of a small instrument tap. A third failure occurred in the 30 inch discharge line in the heat-affected zone of a welded-in stub connection. A consulting firm conducted an analysis of the system and concluded that main line lateral vibrations caused the problems and, in order to stop the failures, recommended improvements in the piping restraint system. Late in 1996 a new pipe support system was designed and installed, and the compressor was modified (rerated) for flowrate changes and efficiency gains.

After the rerate, noise levels went up considerably resulting in overall levels near 118 dBC at the discharge pipe. It should be noted that this value is somewhat below the “questionable” level and below the “danger” level shown in a noise-screening plot recommended by Jungbauer and Blodgett (1998). Soon after restart in December 1996, an instrument pressure sensor failed due to excessive vibration. In January 1997, a crack developed in the 30 inch discharge line in the heat-affected zone of a lifting lug attachment. In June 1997, two more leaks occurred at which point all appendages were "leak clamped" as a preventive measure to stop vibration failures. In 1997, extensive pressure pulsation and vibration data were recorded showing that vibration severity greatly increased as flow was reduced to points approaching predicted surge flowrate. In fact, the component near 55 Hz was completely eliminated at higher flows above the surge control line shown in Figure 16. In Figure 17 a cascade plot for a shaft vibration probe shows that the 60 to 53 Hz (1.1 times speed) supersynchronous vibration was present at the first data points but then disappeared. This was at higher flows to the right of surge control line (manual control for testing) shown in Figure 16. Supersynchronous vibration from 60 down to 53 Hz was also recorded for alternating stress using strain gauges such as on a drain line shown in Figure 18.

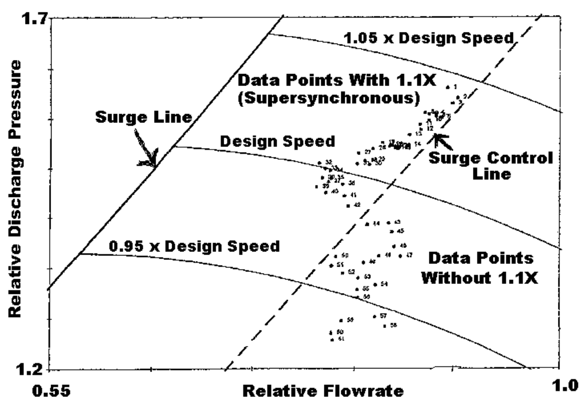


Figure 16. Test Data Points for Propylene Compressor as Speed Was Lowered for Lower Flowrates in Last, High-Pressure Section.

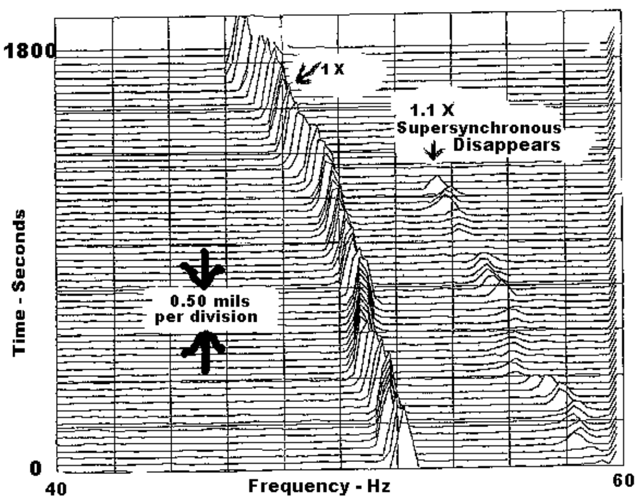


Figure 17. Cascade Plot for Shaft Vibration While Taking Data Points Shown in Figure 16.

The 1.1 times speed component was also recorded on the compressor balance piston line, which was subsequently modified in 1999 with midspan supports. Pressure pulsations as high as 4 psi (peak-to-peak) were measured at 55 Hz in the discharge piping downstream of an opened block valve. For an acoustic wave, this level would be equal to 175 dB sound pressure level, which is high, especially since there should be dissipation due to the very long wavelength. Sound waves at low frequencies travel from the compressor inside the pipe as a plane wave, so there should be attenuation especially at bends. By contrast, typical high frequency noise at blade-passing frequency near 750 Hz travels down the pipe with little attenuation. The wavelength is less than

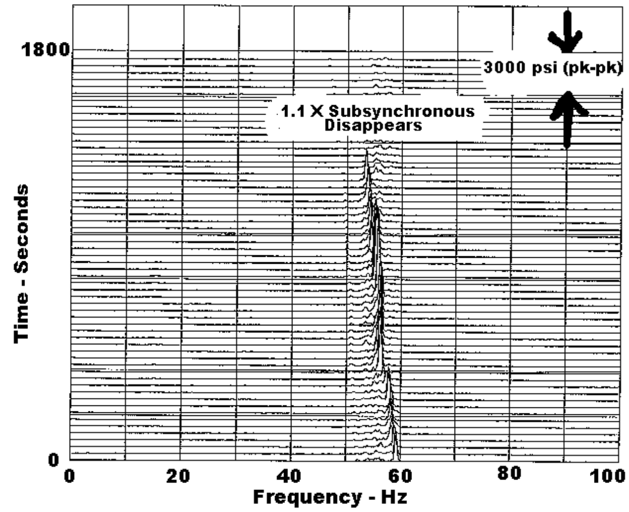


Figure 18. Cascade Plot for Strain gauge on Valve Drain Line While Taking Data Points Shown in Figure 16.

that at the cutoff frequency, given by the frequency for one-diameter transverse mode of the gas as reviewed above. Excitation at 55 Hz was also close to the 2-diameter structural mode frequency of 42 Hz, calculated for the discharge pipe, but the plane wave would not easily excite a mode with phase changes around the circumference.

For data in Figures 17 and 18, as the frequency varied from 59 to 53 Hz, the ratio to speed varied somewhat, from 1.08 to 1.12. However, the flowrate was also reduced by nearly the same ratio. Thus another test at two different constant speeds was done in August 1998. Direct flow-related peaks generally are due to a vortex shedding mechanism; as the flow changes, the frequency changes proportionally. The Strouhal number stays about the same for the same range of Reynolds numbers. Figure 19 indicates the frequency stayed near the 1.1 times speed level for a constant speed and varied flow. Normal frequency spectrums for discharge pipe noise show a pure tone at blade-passing frequency. Additional evidence that rotating stall was the cause of the supersynchronous component was the modulation of "normal" blade-passing frequency of 15 times speed for the last stage impeller by 55 Hz, which is 1.1 times rotor speed. In Figure 20, pipe wall vibration spectrum includes modulation of pressure pulsations to give a component at 13.9 times speed, because $15.0 \times - 1.1 \times = 13.9 \times$. Measurements of pressure pulsations recorded inside the piping in the blade-passing frequency range (600 to 800 Hz) reached 40 psi (peak-to-peak), equivalent to 193 dB, in Figure 21.

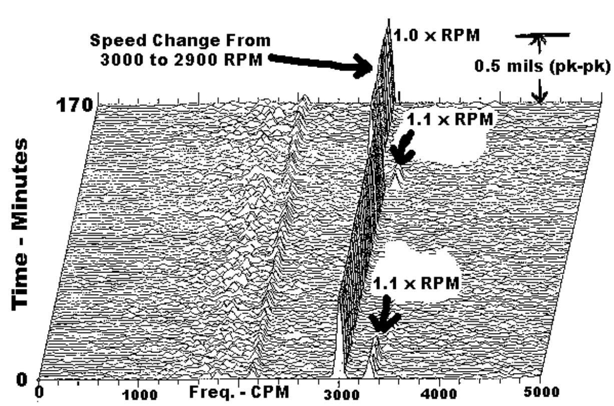


Figure 19. Rotor Response for Propylene Compressor During Test with Varying Flow at Two Speeds.

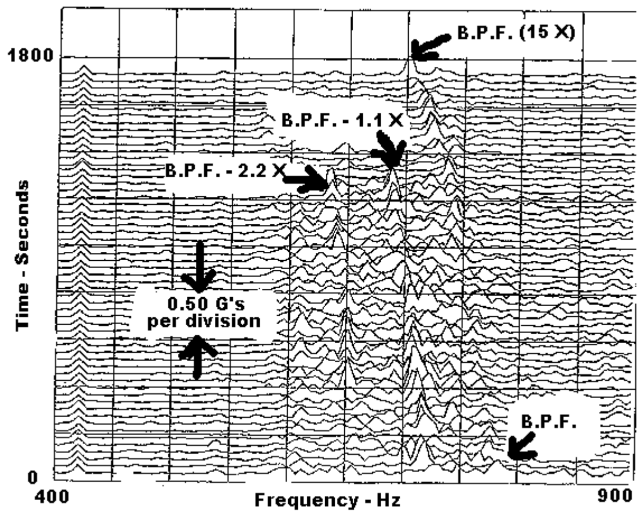


Figure 20. Cascade Plot for Discharge Pipe Wall Vibration While Taking Data Points Shown in Figure 16.

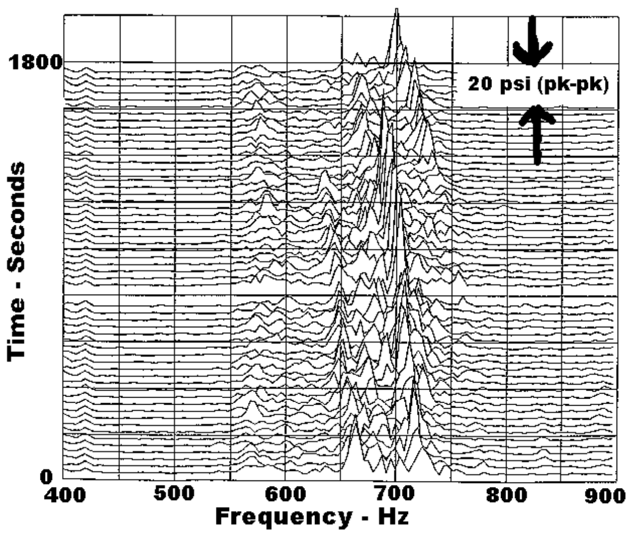


Figure 21. Cascade Plot for Pressure Pulsations Inside Discharge Pipe While Taking Data Points Shown in Figure 16.

Excitation at 55 Hz and 600 to 800 Hz was well below the ring frequency of the pipe, which is the mode where a wavelength of sound in the wall is equal to the circumference. For a pipe, there also can be a coincidence frequency (Walter, et al., 1979) where pipe wall sound waves can match those for both the gas inside and the ambient air outside, but this criterion also was not met. However, calculations did show that high noise level in the 600 to 800 Hz range was due to amplification at gas resonance of transverse modes, the 5-diameter mode or the 2-diameter/one-circle mode. Resonance at 13.9 times speed is shown by strain gauge data in Figure 22. Inside the compressor discharge nozzle, maximum pulsation levels were near 0.50 psi (161 dB). In addition, blade-passing frequency was close to the seven-diameter mode of the pipe wall, further adding to overall vibration. Measurements also showed distinct amplitude peaks at different times verifying resonance of both gas and pipe wall modes. There is another component at 12.8 times speed ($15 \times - 2.2 \times = 12.8 \times$) that implies the 55 Hz component did not have a pure cosine wave character—typical of most rotating stall patterns. At a stationary point near the tip of the last-stage, 15-bladed impeller, there is a forward rate of blade tip wakes at 15 times speed. For a rotating stall traveling at a lower speed than the rotor, but still forward, it would in effect give a difference frequency rather than a summation of the two acoustic frequencies. Since the stall frequency is higher than one times speed, the number of cells had to be greater than one, as one-cell rotating stalls have always been subsynchronous in nature. There could have been two or three cells; higher numbers are possible but rare for modern centrifugal stages. With three cells, the stall would be traveling near 37 percent of rotor speed; with two cells the speed would be faster, near 55 percent of rotor speed. (Stall speed is found by dividing its frequency by the number of stall cells). A ratio of 37 to 55 percent for stall to rotor speed indicates that the stall was likely due to the impeller interacting with the diffuser.

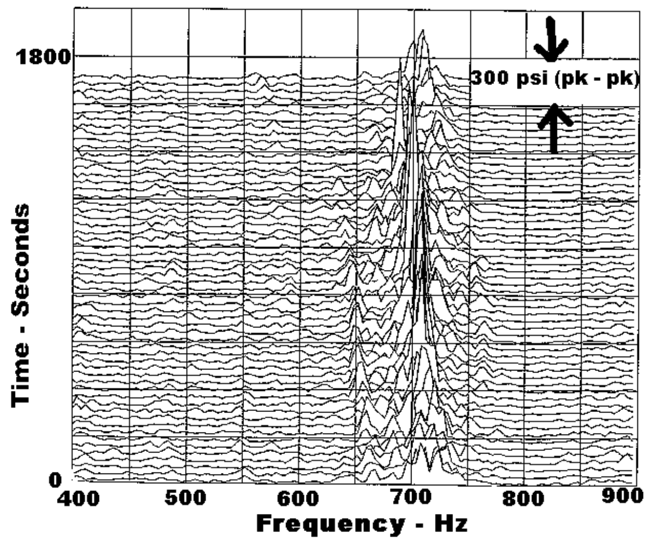


Figure 22. Cascade Plot for Strain Gauge on Drain Line of Discharge Pipe While Taking Data Points Shown in Figure 16.

Additional piping appendage braces and saddles were implemented, but other small failures still occurred between that time and the next plant outage in June 1999. After some further failures, a second consultant confirmed the discharge pipe gas transverse and structural modes to be in resonance with the blade-passing frequency, near 15 times speed. Noise levels were recorded up to 124 dBC and strain gauge data showed that there was still danger of fatigue.

In June 1999, all small appendages were removed near the compressor discharge. At the recommendation of another consulting firm, a flow splitter was installed in the first 30 ft of discharge piping, with the branch piping moved further downstream to where vibration and noise levels were lower. The x-shaped internal splitter plates were welded with much difficulty due to fit-up as well as access inside the 30 inch diameter pipe; using continuous fillet welds in 2 ft sections to facilitate the fabrication process. Splitters were used to divide the sound waves into four sections. On the discharge end of the splitter sections, there would thus be reduced levels and amplification, especially for the higher frequency waves that now would be closer to the cutoff frequency within the four three-sided passages formed by the splitter vanes. The compressor discharge nozzle gas velocity was near 85 fps, which is fairly low for turbulence excitation for the splitter vanes. The vanes had rounded leading edges and tapered trailing edges. The calculated first mode frequency for splitter plates was near 210 Hz, which was removed both from the 55 Hz rotating stall frequency and from calculated vortex shedding frequency. It was also much lower than the blade-passing frequency range of 600 to 800 Hz. Noise and vibration levels were greatly improved based on external vibration and dynamic strain readings showing from two to six times reductions at various locations. Over the next seven months there were about 12 short-duration system surges partly due to control problems including admission steam on the steam turbine drive. Thus the severity and duration of acoustic excitation were most likely even greater during this time period, and the surge pulsations surely would aggravate any crack initiation and/or propagation. Turbine control problems were rectified and shortly after, in February 2000, the plant had to take an emergency shutdown when the discharge piping in the area of the x-splitters cracked through. Fatigue cracks initiated in the heat-affected zone of the weld area of the splitter, at the toe of the welds that were not optimized nor stress-relieved, and propagated to give cracks in the outer pipe wall; cracks also were found in the vanes. The splitter section was removed and bare discharge pipe was once again installed, with all appendages removed for the first 50 ft.

After much discussion, a unanimous decision was made to improve the piping with both diameter and schedule changes, and also have support design changes and pipe appendages minimized. A final agreement of the parties was to use a completely different impeller at the last stage—a higher flow version of the original 27-bladed family. This choice was made knowing that the last impeller stage efficiency would be lower by about 4 percent, but overall compressor efficiency would only decrease by near 1 percent. While compressor modifications were in process, in May 2000, the plant took another emergency shutdown to repair a major crack in an 18 inch connection to the 30 inch discharge line. In August 2000, a third emergency shutdown took place to repair a crack on another 18 inch stub on the same 30 inch header. Fortunately this shutdown coincided with the delivery of the modified spare rotor and diaphragm. The compressor internals were also modified and sections of the discharge piping were changed to heavy wall pipe during the summer of 2000. From that point there have not been any further failures. In Figure 23 is shown the change for piping gas mode resonance points; the revised impeller is resonant with much higher modes, compared to the original with resonance for the 5-diameter mode or the 2-diameter/1-circle mode. Piping wall structural modes are also avoided for the revised pipe as shown in Figure 24. The original pipe had blade-passing frequency close to the 7-diameter mode, whereas the revised thicker pipe has much higher frequencies with margin from resonance. Note that there was a wider frequency range percentage for the original since blade passing was modulated by the 55 Hz component.

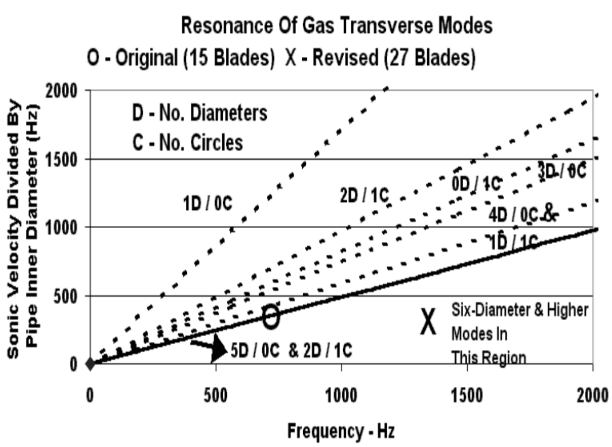


Figure 23. Propylene Compressor Discharge Pipe; Gas Mode Resonant Points for Original and Revised Impeller.

Options rejected prior to the final modifications for 2000 were:

• Utilize another more robust x-splitter, another baffle design, or a series of long tubes as described by Price and Smith (1999). There was general reluctance to use either method due to nonpredictability of results.

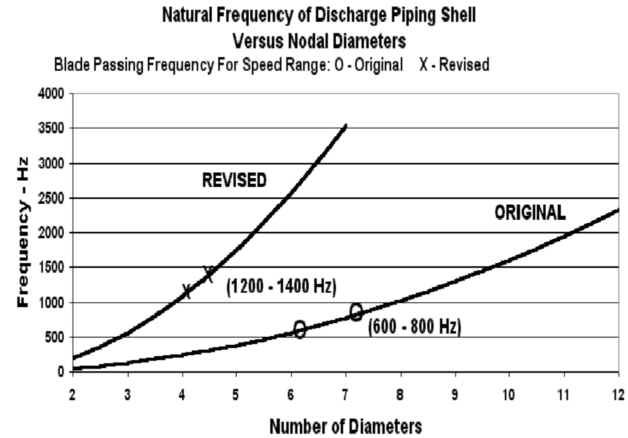


Figure 24. Comparison of Propylene Compressor Discharge Pipe Wall Structural Resonance.

• Install a dissipative silencer; but there was no plant experience for the gas type and flow conditions (gas properties dictate major changes in construction including internal insulation, as compared to air). Dissipative silencers also do not offer much reduction for low frequency components near 55 Hz—as can reactive mufflers.

• Adding external insulation and heavy cladding would have reduced overall piping vibration somewhat due to mass loading and damping, and greatly decreased ambient noise levels. But local levels could have still been high especially for 55 Hz for which lagging has little effect. The actual vibration of the pipe wall at intersections of branches and takeoffs still could be nearly as high as that without lagging. It also would limit routine inspection for cracks as dictated by past history of failures.

• With a more-narrow diffuser at the last stage, the stall/surge point could be moved further to give acceptable lower flow operation. However, data showed that the stall was not just due to a pure diffuser stall or impeller eye incidence, but involved an interaction with the diffuser. Also the transverse mode excitation at blade-passing frequency would still exist, and flowrates at or near unintended surge (as occurred for running with x-splitters) could still give high rotating stall pulsations.

• Add two close-clearance stabilizer vanes in the diffuser to break up the rotating stall, as described by test rig qualification given by Kushner (1996). The drawback was that acoustic energy at blade-passing frequency would most likely increase, so that high frequency gas and pipe wall modes could still have high amplitudes.

Although both consultants had recommended heavier wall pipe to reduce vibration, this was not initially done because the allowable nozzle loads would have been exceeded. The original nozzle loads were approximately 2.6 × NEMA SM23; already above the standard design criteria for nozzle loads on centrifugal compressors of 1.85 × NEMA specified in Appendix G of API 617 (1995). However, depending on load directions, much higher than 1.85 times NEMA can be acceptable. The original spring hanger on the discharge pipe was removed and a rigid support was installed to reduce pipe vibration. The added nozzle loads were evaluated for various designs that gave about twice the existing loads at compressor flanges. The review showed low stresses in casing nozzles, casing support keys and keybars, and also a calculated shaft end deflection of 0.90 mils, well within acceptability for the flexible coupling. Following installation of the revised rotor and improved piping as well as tuning of antisurge controls, strain gauge readings at the first elbow of the discharge pipe were reduced by a factor of 10, and the 55 Hz component was nonexistent.

CONCLUSIONS AND LESSONS LEARNED

- Significant vibration at supersynchronous frequencies that are not harmonics of speed is a sign of an unusual mechanism. As shown by the case described for discharge piping fatigue, rotating stall with multiple cells should be investigated to determine if it is the cause.

- The trend to use a lower number of rotating blades for centrifugal impellers will lead to higher noise levels. This is mainly due to greater likelihood of resonance of the lower, more excitable, transverse modes of the gas inside piping and shell modes of the pipe wall. Vaned-diffuser stages that also increase efficiency but generally reduce flow range also increase noise levels. Thus more acoustic insulation will typically be required for ambient noise limits. The possibility of also matching a structural mode of the pipe and gas transverse mode with the same mode shape should be reviewed and eliminated with piping diameter and/or thickness changes.

- Occurrence of significant failure(s) of piping appendages should lead to a complete system review. The appendage itself should first be reviewed to ensure it does not have matching natural frequency with compressor, valve or vortex generated frequencies.

- Piping thermal/pressure loads on compressor nozzles should be more accurately analyzed to permit stiffer, heavier-wall pipes to give somewhat more safety margin from transients whenever possible.

- General vibration and external noise standards for piping, e.g., those in Jungbauer and Blodgett (1998) are very useful but may not always give sufficient safety factors, especially when there is an unusual occurrence of more than one source of resonance that give additive dynamic stresses. Measurements should be made to find amplitude peaks at resonant points.

- Dissipative silencers can be applied to process compressors handling clean, dry gases, with consideration of gas property differences for noise absorption, and mechanical integrity that includes protection of insulation from erosion.

- Use of splitter blades or multiple tubes having a robust design inside main discharge piping, with stress-relieved welds, still has merit to reduce piping noise and vibration. A further review of acoustic modes of the gas within the formed passages with the vanes should be part of design review.

- There can be a rare application where excitation and response from one-cell rotating stall can be extreme at the surge point and potentially cause rubs, although the same is true for some rotor-bearing systems where the first system mode is excited by surge events. However, the trend to use abradable seals assists in reliability from these and other transients.

- Rotating stall in both axial and centrifugal industrial compressors can be presently accommodated in the vast majority of systems, as long as it occurs at or near surge—sufficiently away from the surge control line. If a rare case does produce a minor response during shop acceptance test; i.e., within subsynchronous amplitude limits, to the right of the surge control line, extensive analysis could prove that reliability is not affected (for low energy forces and/or nonresponsive rotorbearing systems).

- Excessive surging, especially for multistage, higher-pressure ratio and higher power units must be avoided. Unintentional surges should be limited to single pulse events using quick-acting valves. The optimum for large axial compressors is to use dynamic pressure probes for shop test stall/surge definition, with the probes also used for commissioning to set the surge control line.

- Large air or process gas axial compressor applications must have added scrutiny for stall and surge avoidance. Piping acoustic resonance has not been a problem since blade-passing frequencies are much higher than for centrifugal compressors; pipe treatment is still the norm for ambient noise limits.

- Continued research is needed to develop methods and designs that will produce multicell rather than one-cell rotating stall patterns in order to limit excitation, especially in high-pressure compressors with responsive rotors. Developments in active surge/stall control techniques may be needed for some designs, especially those with potentially damaging one-cell rotating stalls that may be unavoidable at flows very near surge.

- Until CFD techniques can accurately correlate with test data for prediction of flowrate at stall initiation and number of stall cells, prototype stage tests should be relied on to prevent vibration problems in normal operating regimes.

REFERENCES

Abdelhamid, A. N., Colwill, W. H., and Barrows, J. F., 1978, “Experimental Investigation of Unsteady Phenomena in Vaneless Radial Diffusers,” *ASME Journal of Engineering for Power*, *101*, (1), pp. 52-60.

API Standard 617, 1995, “Centrifugal Compressors for Petroleum, Chemical, and Gas Service Industries, Appendix G—Forces and Moments,” Sixth Edition, American Petroleum Institute, Washington, D.C., p. 95.

Aungier, R. H., 2000, *Centrifugal Compressors: A Strategy for Aerodynamic Design and Analysis*, New York, New York: ASME Press.

Beranek, L. L., 1971, *Noise and Vibration Control*, New York, New York: McGraw-Hill, Inc., pp. 214-218.

Boeing Aircraft Company, 1993, *777: First Flight*, PBS Home Video, Department of Public Broadcasting Service, Alexandria, Virginia.

Borer, C., Sorokes, J. M., McMahon, T., and Abraham, E., 1997, “An Assessment of the Forces Acting upon a Centrifugal Impeller Using Full Load, Full Pressure Hydrocarbon Testing,” *Proceedings of the Twenty-Sixth Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 111-121.

Den Hartog, J. P., 1954, *Mechanical Vibrations*, Fourth Edition, New York, New York: McGraw-Hill, pp. 291-292.

Ferrara, P. L., 1977, “Vibrations in Very High Pressure Centrifugal Compressors,” ASME Paper No.77-DET-15.

Frank, L., 1995, “Acoustical Laggings Applied to Turbocompressor Piping and Exhaust Ducts,” Eleventh Symposium on Industrial Applications of Gas Turbines, Banff, Canada.

Frigne, P., and Van den Braembussche, R., 1984, “Distinctions Between Different Types of Impeller and Diffuser Rotating Stall in a Centrifugal Compressor with Vaneless Diffuser,” ASME Paper No. 83-GT-61, *ASME Journal of Engineering Gas Turbine and Power*, *106*, (2), pp. 468-474.

Fulton, J. W., and Blair, W. G., 1995, “Experience with Empirical Criteria for Rotating Stall in Radial Vaneless Diffusers,” *Proceedings of the Twenty-Fourth Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 97-106.

Geary, C. H., Damratowski, L. P., and Seyer, C., 1976, “Design and Operation of World’s Highest Pressure Gas Injection Centrifugal Compressors,” Offshore Technology Conference, Paper No. OTC 2485.

Hardin, J. R., and Boal, C. F., 1999, “Using CFD to Improve Stall Margin,” *Proceedings of the ASME Fluids Engineering Division*, FED *250*, pp. 115-121.

Jungbauer, D. E., and Blodgett, L. E., 1998, "Acoustic Fatigue Involving Large Turbocompressors and Pressure Reduction Systems," *Proceedings of the Twenty-Seventh Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 111-118.

Kushner, F., 1996, "Dynamic Data Analysis of Compressor Rotating Stall," *Proceedings of the Twenty-Fifth Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 71-81.

Kushner, F., 2000, "Compressor Blade and Impeller Rotating Disk Vibration Avoidance Parameters," ASME Conference: Challenges and Goals in Industrial and Pipeline Compressors, Orlando, Florida, PID *5*, pp. 81-89.

Kushner, F., Richard, S. J., and Strickland, R. A., 2000, "Critical Review of Compressor Impeller Vibration Parameters for Failure Prevention," *Proceedings of the Twenty-Ninth Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 103-112.

Kuzdzal, M. J., Hustak, J. F., and Sorokes, J. M., 1994, "Identification and Resolution of Aerodynamically Induced Subsynchronous Vibration During Hydrocarbon Testing of a 34,000 HP Centrifugal Compressor," IFToMM, *Proceedings of the 4th International Conference on Rotordynamics*, Chicago, Illinois, pp. 143-151.

Linden, D. H., and Parker, C. A., 1996, "Surge Detection in an Industrial Axial Flow Compressor," *Proceedings of the Twenty-Fifth Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 83-88.

Locke, S. R., 1984, "An Empirical Solution to an Anti-Surge Control Problem," *Proceedings of the Thirteenth Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 51-58.

Marshall, D. F., and Sorokes, J. M., 2000, "A Review of Aerodynamically Induced Forces Acting on Centrifugal Compressors, and Resulting Vibration Characteristics of Rotors," *Proceedings of the Twenty-Ninth Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 263-280.

Mazzawy, R. S., 1980, "Surge-Induced Structural Loads in Gas Turbines," *ASME Journal of Engineering for Power*, *102*, pp. 162-168.

Motriuk, R. W., and Harvey, D. P., 1998, "Centrifugal Compressor Modifications and Their Effect on High-Frequency Pipe Wall Vibration," *ASME Journal of Pressure Vessel Technology*, *120*, pp. 276-282.

NASA (National Aeronautics and Space Administration), 1995, "Estimating Vibrational Powers of Parts in Fluid Machinery," NASA Tech Briefs, pp. 84-85.

Price, S. M., and Smith, D. R., 1999, "Sources and Remedies of High-Frequency Piping Vibration and Noise," *Proceedings of the Twenty-Eighth Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 189-212.

Seebold, J. G., 1971, "Valve Noise and Piping System Design," Flow Control: Control Valve Noise Section in Proceedings of the 1971 Symposium on Flow—Its Measurement and Control in Science and Industry, Instrument Society of America, Pittsburgh, Pennsylvania, pp. 1151-1159.

Seebold, J. G., 1972, "Resonance in Centrifugal Compressor Piping," ASME Paper No. 72-Pet-3.

Sorokes, J. M., and Trevaskis, K. G., 1997, "Recent Experiences with Rotating Stall in High Pressure Gas Re-Injection Compressors," ASME Paper 97-WA/PID-2.

Stein, T. N., March 10, 1980, "Analyzing and Controlling Noise in Process Plants," *Chemical Engineering Magazine*, pp. 129-137.

Walter, J. L., McDaniel, O. H., and Reethof, G., 1979, "Excitation of Cylindrical Shell Vibrations as a Result of Pipe-Wall-Acoustic Coincidence from Internal Sound Fields," ASME Paper 79-WA/DSC-25.

Van den Braembussche, R., 1984, "Surge and Stall in Centrifugal Compressors," VKI Lecture Series 1984-07-Flow in Centrifugal Compressors, Von Karman Institute for Fluid Dynamics, Rhode Saint Genese, Belgium.

von Nimitz, W. W., 1975, "Low Frequency Vibrations at Centrifugal Plants," *Proceedings of the Fourth Turbomachinery Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 47-54.

ACKNOWLEDGEMENTS

The authors recognize Elliott Company and the Dow Chemical Company for permission to publish the paper. Also gratitude is given to all who contributed, with a special thanks to: Mr. Ted Gresh of Elliott Company and Dr. Ralph Harris of Southwest Research Institute for data analysis; Mr. Charles Boal of Elliott Company for CFD analysis; Ms. Penny Baird of Elliott Company for preparation; and our monitor, Mr. Terryl Matthews of the Dow Chemical Company.