ACOUSTIC FATIGUE INVOLVING LARGE TURBOCOMPRESSORS AND PRESSURE REDUCTION SYSTEMS

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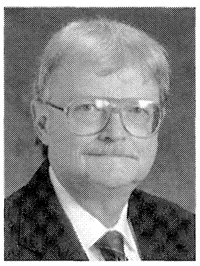

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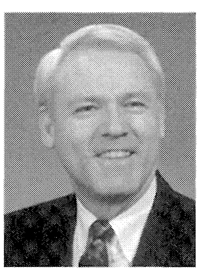

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ABSTRACT

Much work has been done over the years to predict the potential for acoustic fatigue in piping systems associated with pressure reducing systems in the design stage. However, an easily usable screening criteria for assessing the potential for acoustic fatigue for operating installations has not been readily available to industry at large. Based upon theory and empirical measurements, a near-field sound pressure level screening method has been developed for evaluating piping systems. A case history will be presented that will incorporate the use of the near-field noise screening criteria for risk assessment in an existing installation. It is hoped that the use of this near-field noise screening criteria by industry will enable users to assess the risk of acoustic fatigue, which will lead to safer systems and help to determine the need for additional measurements and analysis.

INTRODUCTION

Fatigue failures accompanied by high broad-band noise levels can occur in many types of turbocompressor and pressure reduction systems. These problems differ from most pulsation-induced vibration fatigue failures by virtue of the fact that there is not just one predominant pulsation or vibration frequency, but broad-band excitation and mechanical response up to 5 kilohertz (kHz) in frequency.

Acoustic fatigue is usually associated with large, high flowrate turbocompressor discharge piping and pressure reducing systems, such as throttle control valves, safety relief valves, surge control valves, and flare lines. Small attached piping, valves, nipples, etc., usually fatigue first. In the worst case scenario, the main piping can fail, greatly affecting safety and reliability.

Effective analysis and solution in the operating and design stages of turbocompressor and associated piping systems relies on characterization of the system in question by measurement and computation. Experience has shown that the risk of acoustic fatigue is highly dependent upon fluid/structural energy coupling mechanisms and damping.

By combining noise measurements along with dynamic strain measurements, it has been possible to develop a screening criteria based upon near-field noise levels to determine the risk of acoustic fatigue damage. This empirical criterion was developed from hundreds of noise, vibration, and strain measurements, which was then extended into an analytical capability for predicting the potential for acoustic fatigue failure in the design stage. A description and presentation of the analytical method will not be presented here.

Flow-induced noise and vibration problems can be broken down into two different types. The first type normally involves a coincidence of vortex shedding with an acoustic resonance of the same frequency. The result is an amplification of the dynamic pressure fluctuations induced by vortex shedding. This type of flow-induced noise and vibration problem is characterized by easily distinguishable, relatively pure pipe tones. If the pulsation energy is severe, excessive excitation of piping mechanical shell or beam-type resonances can occur, resulting in fatigue failure (Jungbauer and Eckhardt, 1997). Given the fact that these types of flow-induced noise and vibration problems are more easily defined with regard to frequency, location, specific failure sites, and dynamic strain amplitudes, the potential for fatigue failure can be more easily determined.

The second type of flow-induced problem that will be presented here involves a more general mechanism involving high velocity flows and dynamic pressure in the piping system. Both pressure and velocity components couple into the mechanical system to produce piping vibration. The pressure pulsations are generally broad-band in nature and normally extend up to 5 kHz in frequency. Hence, the piping vibrations usually involve mechanical shell modes along with the lower frequency beam modes. These various individual modes do not become excited to high amplitude. In actuality, the vibration spectrum consists of a broad distribution of many frequencies with only a few distinguishable peaks, as shown in Figure 1. When dynamic strain data are obtained at critical locations in the piping system, the spectrum of the overall strain response mirrors the broad-band nature of the pressure pulsations and vibrations. Fortunately, the overall strain amplitude can be compared with allowable values for determining the potential for piping fatigue failure.

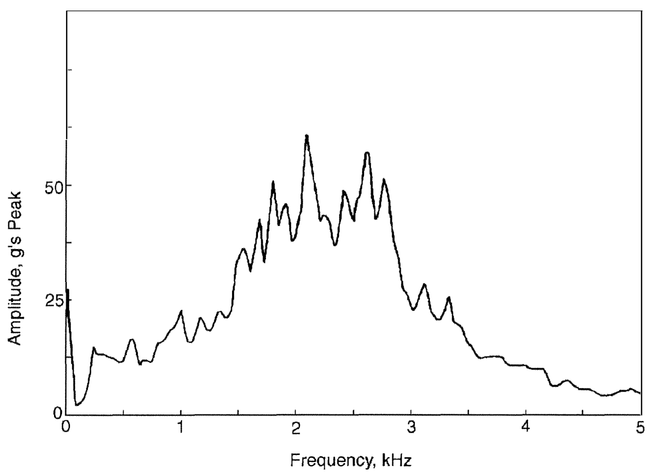

Figure 1. Vibration Acceleration Spectrum of FCCU Regenerator Piping.

Given the broad-band nature of the excitation and the fact that no reliable criteria existed, work has been performed to develop an easily used method for assessing the reliability of piping systems operating with high turbulence excitation sources. By obtaining data on literally hundreds of systems experiencing different levels of fatigue failures, a noise screening criterion has been developed that should be of significant value in evaluating operating systems for safety and reliability. This screening criterion, in conjunction with dynamic strain measurements when indicated by the criterion, is a valuable tool for establishing the need for appropriate engineering changes.

GENERAL BACKGROUND

In the 1960s and 1970s, work was done to relate in-pipe sound power levels to documented fatigue failures of a number of piping systems involving piping sizes normally found in most piping installations. This work also included a limited number of systems with piping in excess of 48 inches O.D. (Carucci and Mueller, 1982). However, this method was not easy to employ by personnel required to evaluate systems that might be subject to failure in the startup or operating stages of plant facilities.

As a consequence of needing a reliable criterion for assessing the severity of acoustic-induced vibration (AIV) due to broad-band sources, a number of focused projects were conducted for industry in the 1970s and 1980s. During this work, it was possible to perform field measurements of external near-field sound pressure levels, internal pressure pulsations, and pipe wall dynamic strain. From these measurements, AIV problems were diagnosed and solutions developed for reducing dynamic strain to acceptable levels. Just as important, it was possible to obtain correlations involving systems, based upon three levels of severity. These levels were:

- No fatigue failures.
- Occasional fatigue failures.
- Persistent and numerous fatigue failures.

THEORETICAL BACKGROUND

Based upon theory and experience, acoustic fatigue involving systems in the oil, gas, chemical, and refining industries is usually associated with large, high flowrate turbocompressor discharge piping and pressure reducing systems. Pressure reducing systems involve recycle valves, throttle control valves, safety relief valves, and flare line systems. In most of the instances studied, small attached piping, valves nipples, etc., usually fatigue first. Few cases occur where the main piping fails. However, the possibility of main piping failures increases significantly with increasing pipe diameter and decreasing wall thickness. Similarly, high strength, thin wall piping systems (i.e., duplex stainless steels) are more prone to main wall fatigue failure.

Experience has shown that the highest flow-induced noise levels associated with centrifugal compressors involve high capacity machines. Examples include MCR compressors found in LNG facilities and cracked gas compressors typically found in refineries and chemical plants. Figure 2 presents a typical octave band noise analysis of a cracked gas compressor second stage discharge piping. These systems normally involve low pressures and high flowrates in the first and second stage discharge piping systems. The piping is of large diameter and relatively thin-walled, when compared with other types of process gas compressors. These factors result in increased radiated noise and broad-band pipe wall vibrations that can lead to fatigue of attached piping elements.

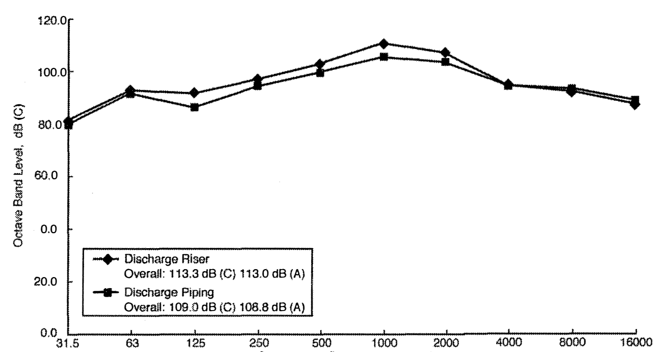

Figure 2. Octave Band Analysis Data Second Stage Near-Field Noise Levels.

By far, the most common broad-band flow-induced problems are associated with various types of pressure reducing systems. The dominant energy source in pressure reducing systems is due to turbulence generated as the confined jet expands toward the downstream piping system. The resulting noise and vibration can range from rather benign levels to those that are catastrophic.

Centrifugal Turbomachinery Noise Generation

Noise generation in turbomachinery is complex and depends upon many factors. However, it can be broadly generalized into two types. The first type involves relatively pure tones occurring at blade pass frequency (BPF) and its multiples. The second type involves broad-band noise sources. The actual mechanisms involved in both pure tone and broad-band noise generation have been receiving increased attention over recent years (Tetu and McLaughlin, 1995). Noise levels are greatly influenced by impeller and diffuser design.

The following are specific generating mechanisms for a centrifugal compressor that can be grouped under the major heading of interaction and flow:

- Interaction sources
  - Interaction of impeller outlet flow with diffuser
  - Rotating pressure field at the inlet to the impeller
  - Rotating pressure field at the impeller discharge
- Flow sources
  - Jet noise at impeller outlet
  - Turbulence noise in all parts of the compressor

This list does not include all sources, but does include the most likely to cause the preponderance of noise in the system. A ranking of the various noise sources indicates that they will have the following relative severity listed in descending order:

- Impeller-diffuser interaction (vaned diffuser)
- Rotating pressure field at impeller outlet
- Rotating pressure field at inlet
- Flow and turbulence noise

Most field evaluation studies evolve into two types of problems. The first is one in which the system performance is not up to par, and the other is one in which the system reliability is in question. The majority of the problems are in the latter group. The system reliability does not necessarily mean only the structural components, but can include personnel hearing damage due to the noise from the units. The majority of the problems are concerned with the structural integrity of the mechanical components, the pipe, and/or the compressor.

Pressure Reducing Systems Noise Generation

Gas pressure reducing systems are known to cause severe piping vibrations and fatigue failures after a few days or few hours of operation. Over the last 10 to 15 years, larger and larger systems have been, or are being, designed to capitalize on economies of scale. As systems have grown larger, noise and vibration problems are affecting piping safety and reliability.

Pressure reducing systems generate pressure pulsations, otherwise known as acoustic energy, caused by the fluid turbulence induced by the flow restriction in the valve or other restricting devices in the system. In valves, acoustic energy is generated by high-velocity gas impingement on pipe walls, turbulent mixing, and, in the case of choked flow, shock waves in the downstream system. Choked flow occurs when the ratio of upstream to downstream pressure at the pressure reducing valve reaches a critical value, sufficient to result in sonic flow at the valve.

The dominant energy source in high pressure relief systems is due to the turbulence generated as the confined jet expands from the valve outlet, narrows through the vena contracta, then expands toward the downstream piping system. Table 1 presents the four regions over which the turbulence process occurs.

Table 1. Turbulence Process Regions.

Region	Region Characteristics
Instability	Unstable shear layers result in generation of periodic vortices
Mixing	Vortex interaction dispersing energy in central cone
Transition	Flow interaction and shear becomes less dominant
Fully Developed	Development turbulence dominated by boundary constraints

The last region, "Fully Developed," brings into play the boundary effects of a confined jet when compared with a free jet.

Even though turbulent jet energy is the dominant source of acoustically-induced vibration, there are at least five other potential contributors that are crucial for an overall understanding of the nature of AIV, as follows:

- *Impinging jet instability*—The impinging jet resonance can occur due to a major energy reflection that occurs in the immediate vicinity (about nine pipe diameters) of the jet. The jet resonance may not be as strong and dominating as the pure impinging jet, but is still significant if coincident with the acoustical environment. Acoustical amplification can significantly amplify the process, producing a significant increase in focused energy.
- *Jet energy transfer*—The energy potential is derived from the choking flow at the vena contracta of the valve. This energy potential is best described as the mechanical stream energy. If the upstream pressure increases, the mass flow will increase, but the velocity at the vena contracta will remain sonically choked. Therefore, higher pressure ratios can, and do, exist across a high pressure relief valve. It is important to note that acoustical efficiency increases rapidly from 7×10^{-5} at a total pressure ratio of about 1.85, to approximately 3×10^{-3} at a total pressure ratio of 2.8, or two orders of magnitude.
- *Acoustical amplification*—The broad frequency distribution of energy, due to shear and geometry-related sources, enters a cylindrical shell in which transverse or radial acoustic standing waves predominate and cause amplification at specific frequencies.
- *Mechanical shell amplification*—Just as the acoustical environment tends to select and amplify preferred frequencies, the acoustical pressure exerted on the pipe wall causes random excitation of the mechanical shell natural frequencies and any smaller piping elements attached to the pipe wall. The pipe wall, in turn, radiates noise to the environment just as an audio speaker would.

Development of Noise Screening Criteria

In the late 1960s and continuing into the early 1980s, attempts were made to develop a methodology for predicting the potential for acoustic fatigue of piping, using sound pressure level and sound power measurements and predictions. Carucci and Mueller (1982) formulated a "design limit" based upon empirical correlations of actual operating experiences, with pressure reducing systems that had or had not experienced failures. These criteria were based upon a limited number of carbon steel piping systems ranging from about 10 inches to 36 inches in diameter. The recommended design limit was based upon in-pipe sound power calculated from sound pressure level measurements or design calculations for noise predictions. While a step forward, the methodology was not readily available and was difficult to apply.

On the other hand, Fagerland (1986) did further work, based upon tests conducted on three compressor installations using strain measurement techniques developed by Southwest Research Institute (Wachel and Bates, 1976). Using his method, measured sound pressure levels and strain amplitudes were obtained. These data were then used to back-calculate predicted noise levels and strain amplitudes. However, these data were again based upon a limited number of samples, typically X-strong pipe. The results of Fagerland's work gave a recommended sound pressure level limit of 120 dB at the pipe wall. When adjusted for each pipe size at a standard one meter distance, the resulting guideline was reduced to a sound pressure level of 110 to 115 dB as shown in Figure 3.

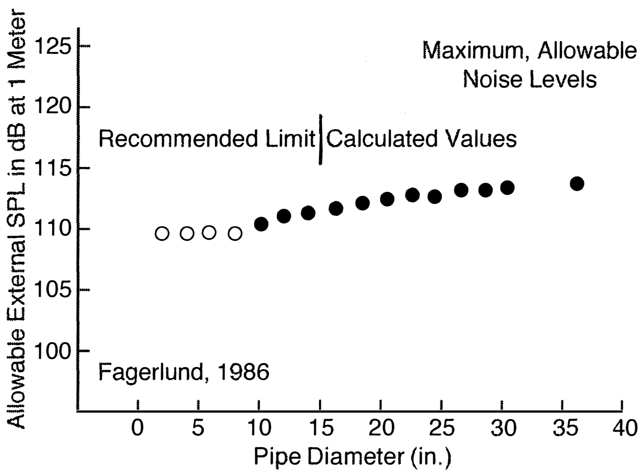

Figure 3. Developed Guideline Limits for External Sound Pressure Levels Versus Pipe Diameter.

Subsequently, based upon numerous correlations involving simultaneous measurements of near field sound pressure levels correlated with dynamic strain measurements, it appeared that the limit of 120dB at the pipe wall was overly conservative and additional work in the development of a general noise screening criterion for assessing piping reliability was justified.

Vibration and Noise Screening Criteria

In assessing the severity of general beam-type piping vibrations, the screening criteria given in Figure 4 was developed in the 1970s. These criteria provided a preliminary method for assessing vibration severity and are based upon empirical and analytical data obtained during the course of vibration studies in conjunction with dynamic strain measurement. This screening capability has proved to be effective, when expanded to include stress calculations and/or strain measurements in critical applications.

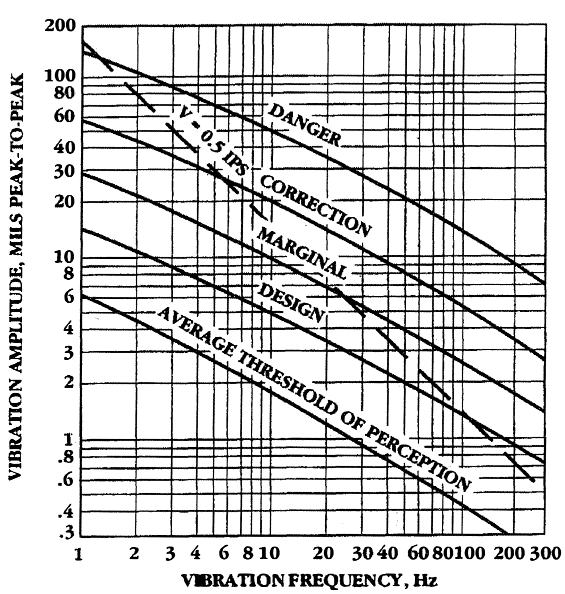

Figure 4. Screening Piping Vibration Severity Chart.

Based upon the validity of this method for correlating vibration data with strain information to develop the above vibration criteria, the same approach was used to develop the noise screening criteria. This work was based upon data taken on piping systems experiencing broad-band acoustical excitation from compressors and pressure reducing components. Figure 5 presents the resulting noise screening criteria, which are predicated upon near-field sound pressure levels of 124 dB or less as being acceptable for most systems. When sound pressure levels increase above this level, the probability of fatigue failure increases, especially in small diameter piping attachments. Above 130 dB, fatigue failure is highly probable and extreme caution should be exercised. However, these screening criteria merely indicate that alternative approaches should be used, i.e., strain measurements, in the final assessment.

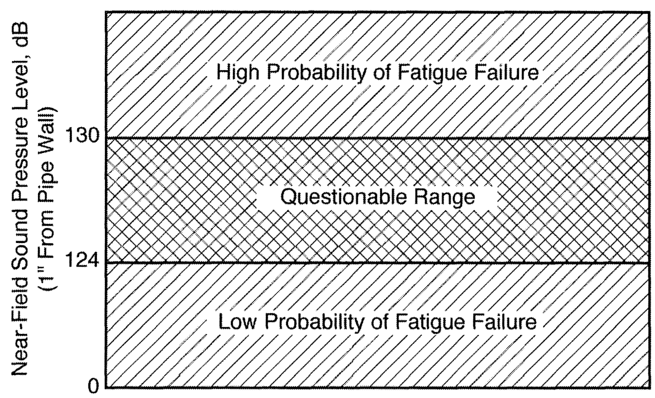

Figure 5. Noise Screening Sound Pressure Levels for Assessing Probability of Fatigue Failures of Piping Systems.

Dynamic Strain

Preceding discussions have mentioned the term "dynamic strain." At this time, it would be appropriate to discuss what dynamic strain is and how it is obtained.

Dynamic strain, as defined in Equation (1), is a direct measure of material deformation and is easily converted to dynamic stress, as defined in Equation (2). Equation (3) presents the dynamic stress derived from Equation (1) for a dynamic strain of 100 $\mu\epsilon$ peak-to-peak.

$$1 \text{ microstrain peak-to-peak} = 1\ \mu\epsilon\ (p-p) = 1\frac{\text{microinch}}{\text{inch}} = 1*10^{-6}\frac{\text{inch}}{\text{inch}} \quad (1)$$

$$\sigma_{p-p} \approx E*\mu\epsilon\ (p-p)*1*10^{-6}\frac{\text{inch}}{\text{inch}} \quad (2)$$

$$\sigma_{p-p} \approx 30*10^{6}\frac{\text{lb}}{\text{in}^2}*100\ \mu\epsilon\ (p-p)*1*10^{-6}\frac{\text{inch}}{\text{inch}}$$
$$\sigma_{p-p} \approx 3000\ \frac{\text{lb}}{\text{in}^2} \quad (3)$$

It should be noted that the dynamic strain amplitude of 100 $\mu\epsilon$ peak-to-peak and resulting stress of 3000 lbs/in^2 is the maximum acceptable amplitude for most carbon steel piping systems. This amplitude correlates with the maximum acceptable near-field sound pressure level screening criterion of 124 dB, given in Figure 5.

The measured stress can then be compared with the material fatigue data, with appropriate corrections reflecting the characteristics of the system being studied. When different material fatigue data are available, endurance limits must be corrected for various factors including:

- Surface effect.
- Size effect.
- Mean stress.

- Safety factor.
- Nominal stress intensifiers.

The resulting endurance limit can then be converted to a peak-to-peak dynamic strain amplitude and can be compared directly to strain measurements taken during field testing. These amplitudes can then be correlated with near-field sound pressure levels.

In order to obtain dynamic strain, straingauges are used. These consist of a foil grid with a polyimide plastic backing as the carrier, as shown in Figure 6. The gauge is attached to the piping where experience has shown dynamic strains will be the highest, such as shown in Figure 7. Gauges are attached to the piping using a special adhesive specifically for use with the gauge.

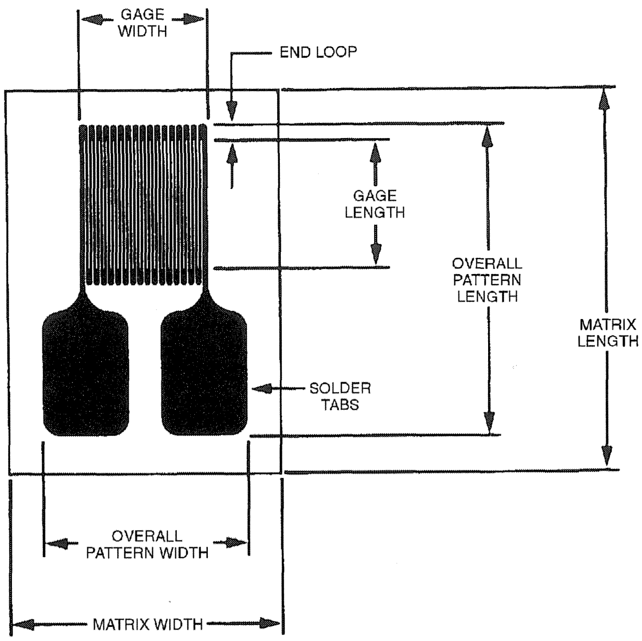

Figure 6. Gauge Nomenclature and Features of a Typical Foil Straingauge. (Courtesy of Measurement Group, Inc.)

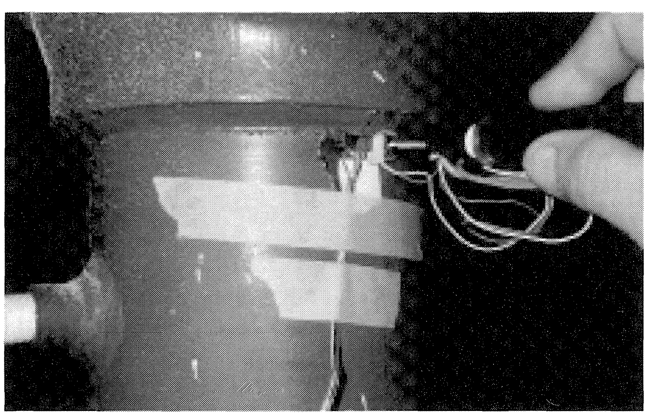

Figure 7. Typical Strain Gauge Location for Measuring Maximum Strain Amplitudes.

When the material deforms, the deformation is also experienced by the gauge. This results in minute changes in gauge resistance due to the Poisson effect. By using a wheat stone bridge with the proper electronics and calibrations, it is possible to relate the voltage change to a strain amplitude and then to stress. Hence, the straingauge and the data obtained from it are exceedingly valuable tools for assessing safety and reliability of critical piping systems.

Mechanical Resonances

There are two types of mechanical responses that greatly influence the severity of piping vibrations. These resonances can be loosely grouped into beam-type and shell-type resonances.

- *Beam-type resonances*—A piping span between two restrained points will vibrate at specific frequencies when excited. Each vibration frequency is associated with a precise deflection or mode shape. These natural mechanical frequencies are strongly influenced by the boundary or end conditions and concentrated masses.

It is important to note that the natural mechanical frequency of the piping is not always the vibration frequency. If the pulsation frequency and natural mechanical frequencies are not the same, the piping can still vibrate. This forced nonresonant vibration requires more pulsation energy to obtain high amplitudes than if the pulsation and mechanical response frequencies were coincident and resonant.

Beam-type piping vibration frequencies are significant up to 200 Hz. Most fatigue failure problems occur between 25 and 75 Hz.

- *Shell-type resonances*—Piping composed of cylindrical shells also exhibits mechanical resonant frequency responses. The resonant mechanical frequencies are dependent upon pipe diameter, wall thickness, and segment length. The node locations for particular shell modes are a function of the boundary conditions. Figure 8 presents several of the lower frequency shell mode shapes. These modes are typically in excess of 200 Hz. Shell modes can be excited by acoustic energy in the same manner as beam-type piping mechanical resonances. If small diameter piping components are attached to the vibrating shell, fatigue failures can occur. Similarly, small diameter, short piping stubs can have their resonant mechanical frequencies in the same range as the lowest shell modes, resulting in a coincidence of the two with significant impact on fatigue life.

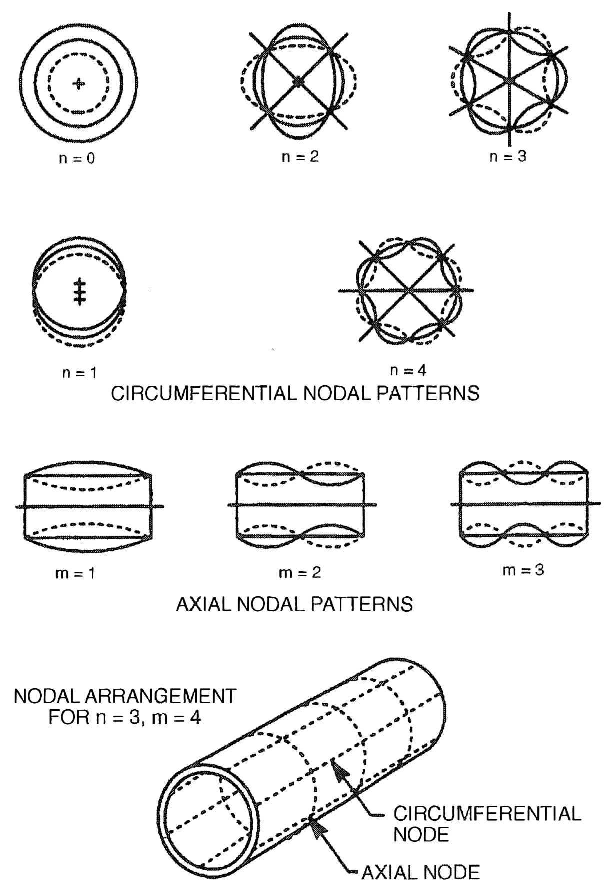

Figure 8. Cylindrical Shell Vibration Mode Shapes.

CASE HISTORY

Having discussed the characteristics of broad-band flow-induced noise and vibration, its generation in compressor and pressure reducing systems, development of a noise screening criteria, and final reliability assessment via dynamic strain measurements, a case history of an actual application can be presented. The following case history demonstrates the correlation that was obtained between screening sound pressure level readings and dynamic strain measurements.

Problem

During the startup and commissioning of two gas processing compressors on an offshore platform, numerous failures of small diameter branch connections and fittings were experienced. One failure involved a crack at the toe of the weld of a two inch weldolet connecting a nitrogen (N_2) purge line to the main 16 inch suction line, which was downstream of the junction of the recycle piping with the main suction pipe. A second failure occurred at the attachment of a two inch emergency shutdown valve (ESDV) bypass line weldolet upstream from the first failure. Cracks were also found in the smaller nipples attached to the suction piping downstream of the recycle control valve. The compressors had operated with equivalent flow orifice plates in place of the recycle control valves, and with the design low noise valves installed.

As a consequence of the failures and indications of additional cracks revealed by NDT, it was decided to conduct a detailed study of the compressor suction piping by measuring pressure pulsation, vibration, noise, and dynamic strain. For the purposes of safety, initial tests were planned using N_2 as the process gas. Eventually, as test information was gathered and analyzed, actual hydrocarbons were to be introduced.

Objectives

- Obtain dynamic pressure pulsation data at select points in the recycle and suction piping systems of one train, using transducers mounted flush with the inside diameter of pipe wall.
- Collect comprehensive dynamic strain data at critical locations determined from previous operating experience.
- Conduct the tests with both an equivalent orifice and the recycle valve using N_2.
- Evaluate the test data obtained using N_2 and then introduce process gas, if preliminary results did not compromise safety.
- Conduct appropriate tests using the process gas with the recycle valve installed, which would further help to define the problem.
- Evaluate the results obtained and assess the cause and effect relationships.
- Recommend modifications or additional analysis for the elimination of the fatigue failures.

Test Setup

Testing was conducted using the following types of transducers (Figures 9 and 10 show test point locations):

- Two flush-mounted pressure pulsation transducers (P-1 and P-2)
- Two vibration transducers (XL-1 and XL-2)
- Nine dynamic straingauges (SG-1 through SG-9)
- Four near-field noise test points (SPL-1 through SPL-4)

Data were acquired during three primary test runs:

Nitrogen Gas Test

Test 1. 3.00 inch orifice installed in place of the recycle valve at 50 and 60 percent of compressor operating speed.

Test 2. Compressor recycle valve installed with valve fully open at 50, 60, 70, 80, 90, and 100 percent of compressor speed.

Process Gas Test

Test 3. Compressor recycle valve installed with the valve fully open at 50, 60, 70, 80, 90, and 100 percent compressor speed.

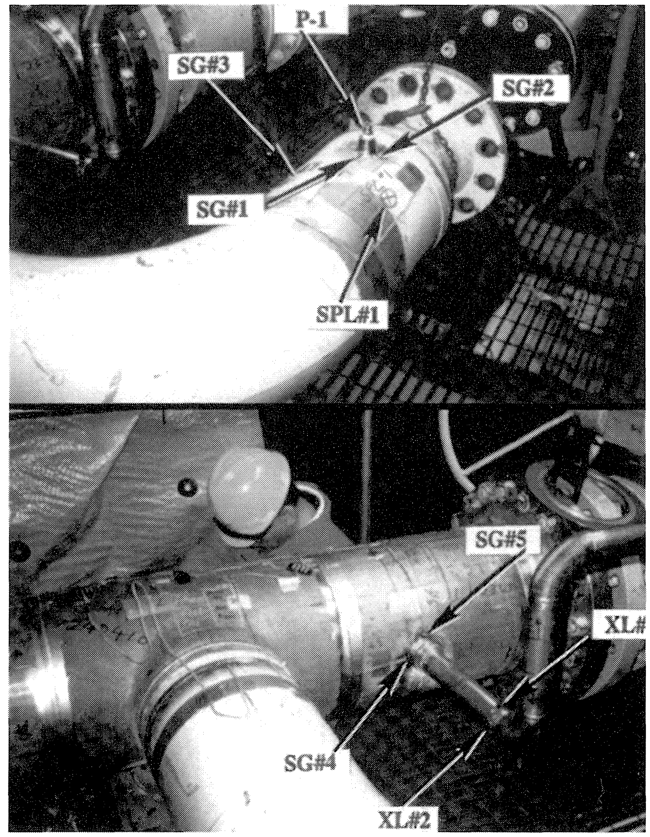

Figure 9. Pressure Pulsation, Vibration, Dynamic Strain, and Noise Test Point Locations.

Results

The results of Nitrogen Gas Test 1 are presented in Figure 11. The data presented gave the highest dynamic strain and sound pressure levels of the three test conditions studied.

Figure 11(a) presents the strain data taken on the small weldolet fitting just downstream of the 3.00 inch orifice, which was installed in lieu of the recycle valve. These data show that at 60 percent of compressor running speed, the overall dynamic strain was approximately 116 percent of the allowable amplitude. Spectrum analysis of the strain data indicated that the excitation was relatively broad-band, with no particular frequency being significantly higher than any other frequency.

The near-field noise levels for the two compressor speed conditions are shown in Figure 11(b). The noise screening criteria developed previously indicates that at the measured noise level of 124 dB(c), the overall dynamic strain due to broad-band turbulence should be at or near the upper limit of allowable strain. This is confirmed by the strain data in Figure 11(a).

Test 2 was then conducted (N_2 gas and recycle valve installed). The speed of the compressor was increased to 100 percent in 10 percent increments. The dynamic strain amplitudes at this test condition were 17 to 20 percent of the allowable levels for SG-1 and SG-2 (Figure 12(a)). This correlates with the data presented in Figure 12(b), which shows a maximum near-field noise level of 110 dB(c), which is well below the maximum acceptable screening noise level.

Test 3 was performed with hydrocarbon gas under test conditions similar to those for Test 2. The dynamic strain amplitudes shown in Figure 13(a) are nearly identical to Test 2 data at 100 percent compressor speed for the SG-1 and SG-2 locations.

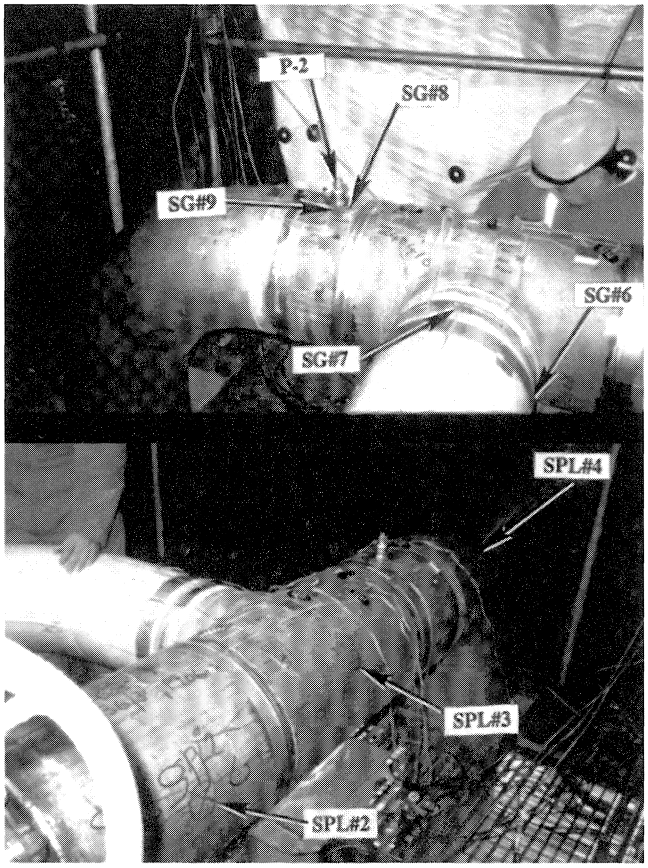

Figure 10. Pressure Pulsation, Vibration, Dynamic Strain, and Noise Test Point Locations.

Near-field noise levels were approximately 113 dB(c). This was also well below the maximum allowable amplitudes for both strain and noise.

CONCLUSIONS

The results of these tests confirmed the validity of the noise screening criteria when correlated with dynamic strain measurements. The testing confirmed that the use of the 3.00 inch orifice in lieu of the recycle valve generated excessive broad-band flow turbulence, noise, vibration, and dynamic strain. Installation of the recycle valve significantly reduced the generation of flow turbulence, resulting in reduced noise, vibration, and dynamic strain. The results of the testing allowed startup and normal operations to proceed.

SUMMARY

It has been demonstrated that near-field noise measurements can be used for assessing the severity of broad-band turbulence excitation of piping systems by using the screening criteria presented. The source of the noise can be either a turbocompressor system or a pressure reducing device. Use of these noise screening criteria can be of significant assistance to industry in avoiding potentially catastrophic failures if screening levels are exceeded. When levels are exceeded, further measurement and analysis is warranted.

NOMENCLATURE

E = Young's Modulus, lbs/in^2
$\mu\epsilon$ = Microstrain, microinch/inch
p-p = Peak-to-peak amplitude
σ = Stress, lbs/in^2

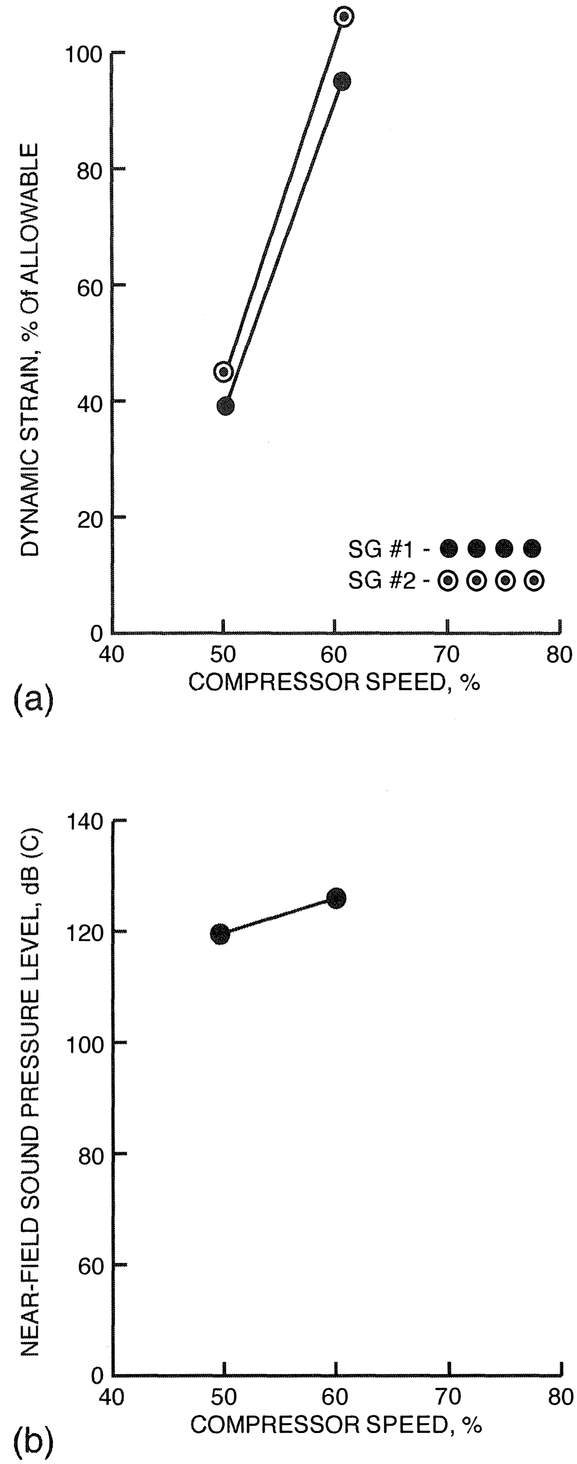

Figure 11. Test 1—Nitrogen Gas and Equivalent Orifice.

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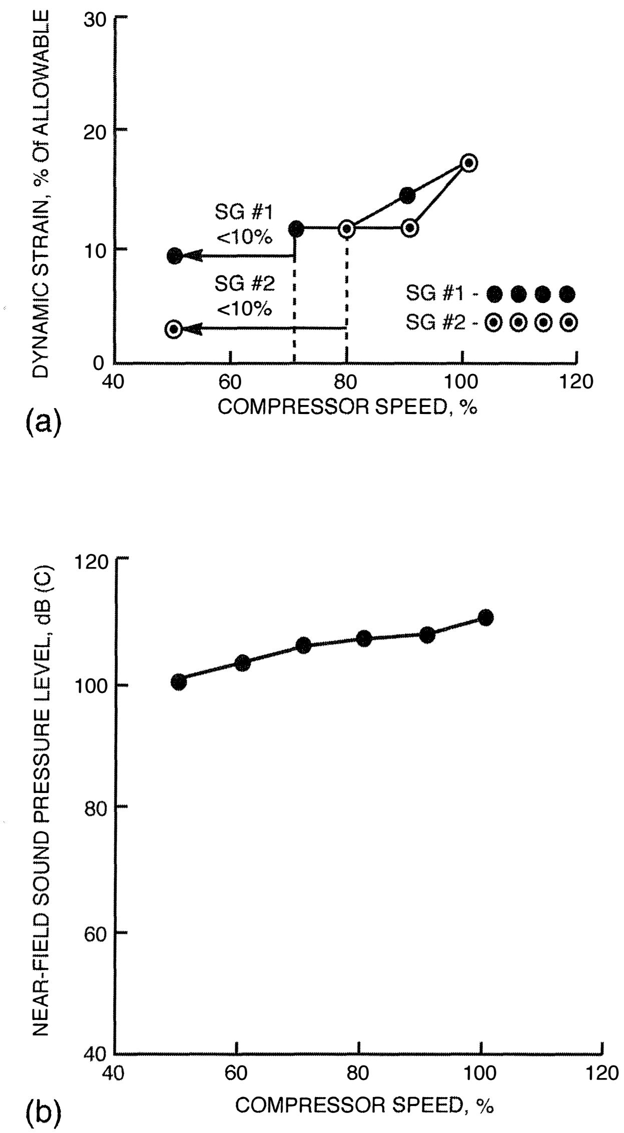

Figure 12. Test 2—Nitrogen Gas and Design Recycle Valve.

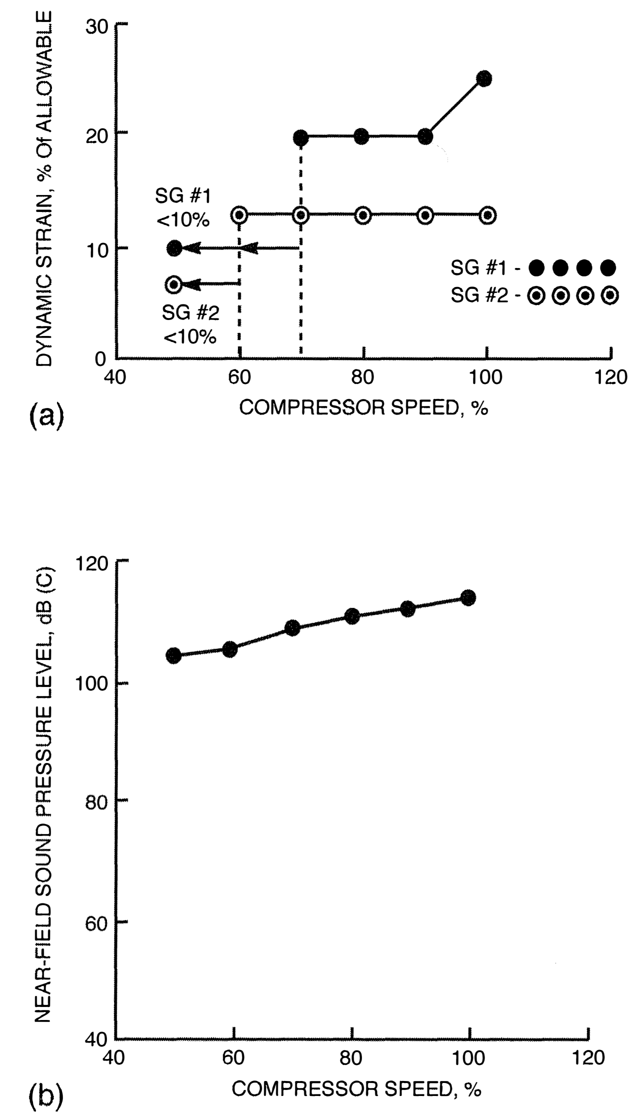

Figure 13. Test 3—Hydrocarbon Gas and Design Recycle Valve.

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