LOW FREQUENCY VIBRATIONS AT CENTRIFUGAL PLANTS

by

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ABSTRACT

Low frequency vibrations at centrifugal compressor and pump installations are usually discarded as unlikely. However, severe acoustically excited vibrations at frequencies as low as 5 or 10 Hz have been observed at a number of plants. The paper discusses some of the excitation mechanisms and techniques available for their control. Several field study examples illustrate typical problems and their solutions.

INTRODUCTION

In the design of centrifugal plants, one has learned to give careful consideration to the analysis of rotor-dynamics and perhaps to the natural frequencies of the unit support system, in addition to the static structural and operational considerations. Yet, satisfactory unit operation does not automatically mean satisfactory plant operation. The missing link, of course, is the piping system which may have vibration problems and even experience fatigue failures while the unit is operating satisfactorily. One of the drastic examples in the author's experience was a fatigue failure in a unit minimum flow bypass loop in a 30,000 hp plant refrigeration system. This piping failure caused complete plant shutdown and over $300,000 loss in production in the three days it took to have the problem corrected and to bring the plant back on line.

The major portion of dynamic energy generated by a centrifugal compressor or pump is at the blade passing frequency which is typically in excess of 1,000 Hz. Except for sample tests, and drain connections which are known to fatigue at these high frequencies, the usual consequence is piping radiated noise. In severe cases, however, particularly associated with high flow velocities, main gas line failures can also occur.

In addition to these high frequency pulsations, the unit also generates broad band turbulence which can excite low frequency acoustical resonances in the system. Piping resonant surge can also provide significant low frequency excitation and, in addition, flow excited turbulence at restrictions, obstructions, or junctions can also be a source of low frequency pulsations. It is therefore not surprising to find low frequency vibrations (typically below running speed of the unit) at centrifugal plants. As will be shown later in field examples, the amplitudes of such vibrations can be in the 15-30 mil (0, 38 - 0, 76 mm) p-p category in the 10-60 Hz frequency range with as much as 70 mils p-p (1,78 mm) recorded on one occasion. The percentage of centrifugal plants with objectionable low frequency vibration problems is small; however, it is higher than generally assumed since a number of such problems remain unrecognized and blamed on other sources.

SOURCE OF LOW FREQUENCY EXCITATION

Deep machine surge is a strong source of low frequency energy. However, it is outside of the intended operating range of the unit while our concern is with the low frequency pulsation excitation mechanisms which exist within the normal operating range of centrifugal units. The most frequent of such sources are flow induced pulsations excited by some type of vortex shedding action at piping junctions or other discontinuities in the flow stream. The energy available from such sources is small and the excitation frequencies are within a fairly narrow range. Nevertheless, if a piping response is present coincident with the frequency of excitation, pulsation amplitudes can build up to fairly high levels. In addition, the compressor can also be a determining factor since it can act as a pulsation amplifier under some circumstances.

Most industrial compressors have low Mach numbers in compressor passages and consequently do not have a major influence on the acoustic modes that could build up to significant amplitudes. Rather, it is the components in the piping system which have the greatest influence over the types of acoustic modes that might be present in a centrifugal compressor installation. However, the compressor does play a major role as a means of amplifying any pulsations which might occur due to any particular excitation source or some random excitation due to flow turbulence.

The process of pulsation amplification by the compressor is illustrated in Figure 1 and discussed below. The damping energy shown in Figure 1 is related to additional pressure loss which occurs under pulsating flow conditions. As compared to steady flow pressure loss, it increases as pulsation amplitude increases. Theoretical analysis of this phenomenon indicates that the damping energy is proportional to pulsation amplitude

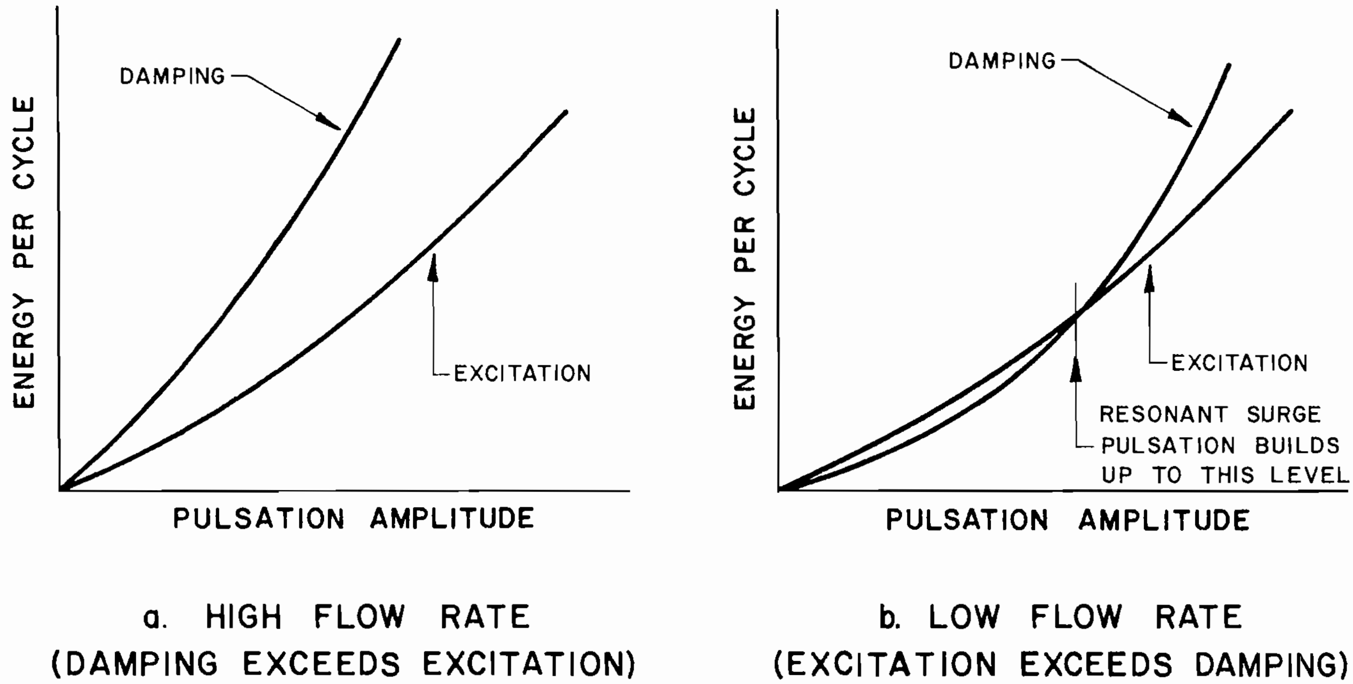

Figure 1. Excitation and Damping Energies as Related to Resonant Surge.

to some exponent greater than 2 but less than 3. Aso, the value of damping energy per acoustic cycle increases as flow rate increases. On the other hand, acoustic energy per cycle is added by the compressor under proper conditions. This excitation energy is proportional to the pressure amplitude to the second power.

Under relatively high flow rates the damping energy exceeds excitation energy at all possible pulsation levels as shown in Figure 1-a. Under these circumstances, if some type of random excitation did occur, the damping would always exceed excitation and the pulsation would damp out. However, as the flow rate is lowered, the damping energy for a given pulsation amplitude decreases while the capacity of the compressor to amplify pulsations increases as illustrated in Figure 1-b. In this case, excitation would exceed damping for low pulsation amplitude and pulsation would build up until damping and excitation were equal. This is the situation which occurs under surge conditions where surge is related to a piping resonance situation.

The ability of the compressor to amplify or energize a given standing wave pulsation depends on its impedance characteristics at the point it is operating and upon its position relative to the mode shape of that standing wave. Laboratory experience and field observations confirm that a compressor is most likely to excite a given resonance mode when it is located near a pressure node or a point of maximum velocity fluctuation. Laboratory work has further shown that the lower harmonics are most apt to be energized.

From the above discussion it is evident that the compressor location is an important parameter in determining which acoustic mode might be excited. It is not a means of eliminating surge possibility, but rather a means of eliminating one mode of surge pulsation in favor of another. While surge can not be completely eliminated by judicious compressor location and piping system design, proper compressor locations should reduce surge severity. The major value of this technique is for arriving at a system design which avoids highly responsive acoustic modes and avoids excitation of coincident mechanical resonant frequencies controlling thus compressor and piping vibrations.

Comparing resonance surge and machine surge, resonant surge occurs at a frequency which is associated with some acoustic resonance of the piping system usually above 1 Hz and flow perturbations experienced are usually not large enough to cause complete flow reversal. On the other hand, machine surge often occurs at frequencies below 1 Hz and may involve net reversal of flow during part of the surge cycle. Whether the surge is of the resonance type or the machine type depends entirely upon the match of the performance characteristics of the compressor and the acoustic characteristics of the system into which it is installed.

As stated earlier, the most frequently found source of low frequency pulsations at centrifugal plants is flow turbulence generation at obstructions and junctions in the piping system. The generated turbulence energy peaks over a relatively narrow frequency band known as Strouhal frequency (f_s) which is directly proportional to actual flow velocity in the main line and inversely proportional to the diameter of the side-branch line at a junction or diameter of the restriction in the main line. Laboratory tests at SwRI have shown best correlation when the coefficient of 0.5 is used to calculate the Strouhal frequency at a pipe junction:

$$f_s = 0.5\frac{V}{D}$$

where f_s = Strouhal frequency in Hz

V = Actual flow velocity in feet per second (or meters/second)

D = Side branch or restriction diameter in feet (or meters)

For example, if gas flows in a unit lateral at 30 fps (9.14 m/s), the Strouhal frequency of turbulence generated across a 15 inch I.D. (0.38 m) bypass line will be 24 Hz.

The energy and thus pulsation amplitudes at Strouhal frequency will be small (at least at typical flow velocities) unless they are amplified by acoustic piping resonances at or near Strouhal turbulence frequency. Acoustical amplification factors in most cases range from 5 to 100. This means pulsation amplitudes can be 100 times higher than they would have been otherwise; e.g., in cases when high Q acoustic resonance is present where turbulence is generated and coincident with Strouhal frequency. Thus, it is not surprising that in such cases high amplitude low frequency pulsations are found at centrifugal plants.

Some of the most severe pulsation buildups occur, however, when flow obstructions or similar turbulence sources exist in or at the compressor. Usually, high energy levels are associated with such turbulence resulting in high pulsation levels. In one such case, a peak to peak pulsation amplitude of 120 psi (8,33 bar) was recorded in the suction lateral with 843 psi (58, 5 bar) static pressure due to turbulence at certain variable guide vane settings. This level is much higher than the pulsation amplitudes normally found at reciprocating compressor installations.

PULSATION EFFECTS

Pulsations as such are not of concern except for the undesirable effects they produce. By far the most frequent consequences are the pulsation induced vibrations and ultimately the cyclic stresses they produce. The plant reliability chart shown in Figure 2 relates pulsation to vibration, to stress, and ultimately, to failure probability. It should be the guiding light to anyone involved with the design and evaluation of compressor and pump installations.

Of course, vibrations and the cyclic stresses they produce will be particularly severe if the mechanical resonances coincide with the acoustic resonances and they, in turn, coincide with the acoustic excitation frequencies. Considering that typical mechanical resonant amplification factors range from 5 to 50 and higher, the worst case may mean vibration levels 5,000 times higher than would have existed with adequate separation of acoustical and mechanical resonances from each other and from excitation frequencies.

The less recognized but equally important pulsation effects are those on the compressor unit. If the acoustically excited piping and foundation vibration are coincident with unit critical(s), excessive shaft vibrations with resulting bearing damage, etc., can occur. Even more important is pulsation excitation of unit torsional criticals since pulsations couple directly to compressor wheels putting torsional load on rotor-dynamic system(s) and have known to be the cause of shaft failures.

In addition, pulsation may have the effect of depressing the head, which in turn would depress the flow toward the surge point. In this indirect way, the presence of external pulsations may trigger surge prematurely. Furthermore, pulsations can cause significant errors in orifice low metering including introduction of uncertainties in unit surge control since the minimum flow bypass control is usually based on orifice flow data taken close to the unit. Finally, due to the nonlinear relationship between pressure and flow, pulsations will increase pressure drop in the system affecting thus the efficiency of plant operation.

In designing a centrifugal compressor (or pump) piping system there are some basic considerations which should be followed such as avoiding mechanical resonances coincident with unit running speed and the lowest critical(s). However, an acoustical system analysis is required establishing major excitation frequencies and acoustical resonances, including the effect of compressor, before mechanical system requirements can be fully defined and the system design developed accordingly.

PULSATION CONTROL TECHNIQUES

The continuous applied research programs carried out in the author's division for more than 20 years for some 48 companies comprising the so-called Pipeline and Compressor Research Council of the Southern Gas Association has produced the SGA-Dynamic Compressor System Simulator (SGA-Analog as it is commonly known) which is the worldwide accepted industry's standard for the design of reciprocating compressor and pump installations and the study of acoustic-fluid dynamic phenomenon usually in the 0-300 Hz frequency range. It is, in effect, a dynamic electrical model (Figure 3) duplicating the thermodynamic and fluid dynamic characteristics of the actual system in an analogous electrical model. Since it is a physical modeling technique (dynamic simulation) rather than a mathematical modeling (digital or analog computation) technique, the system to be studied can be easily assembled using basic building elements such as small plug-in sections duplicating usually one foot (0.3 m) of pipe of a given diameter and at a given velocity of sound in the system. Since

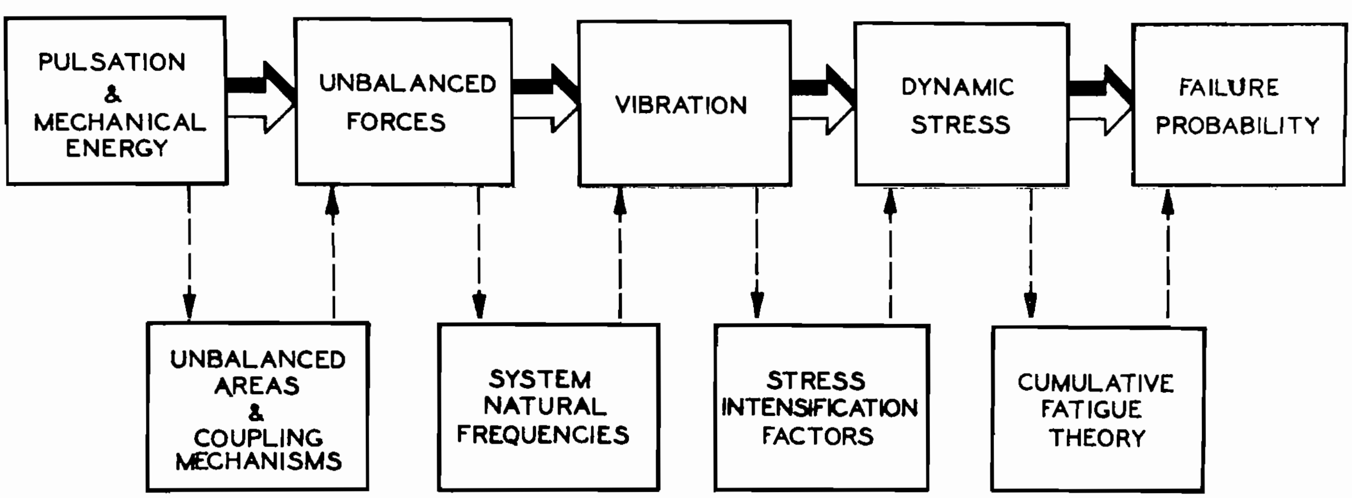

Figure 2. Flow Chart for Analysis of System Reliability.

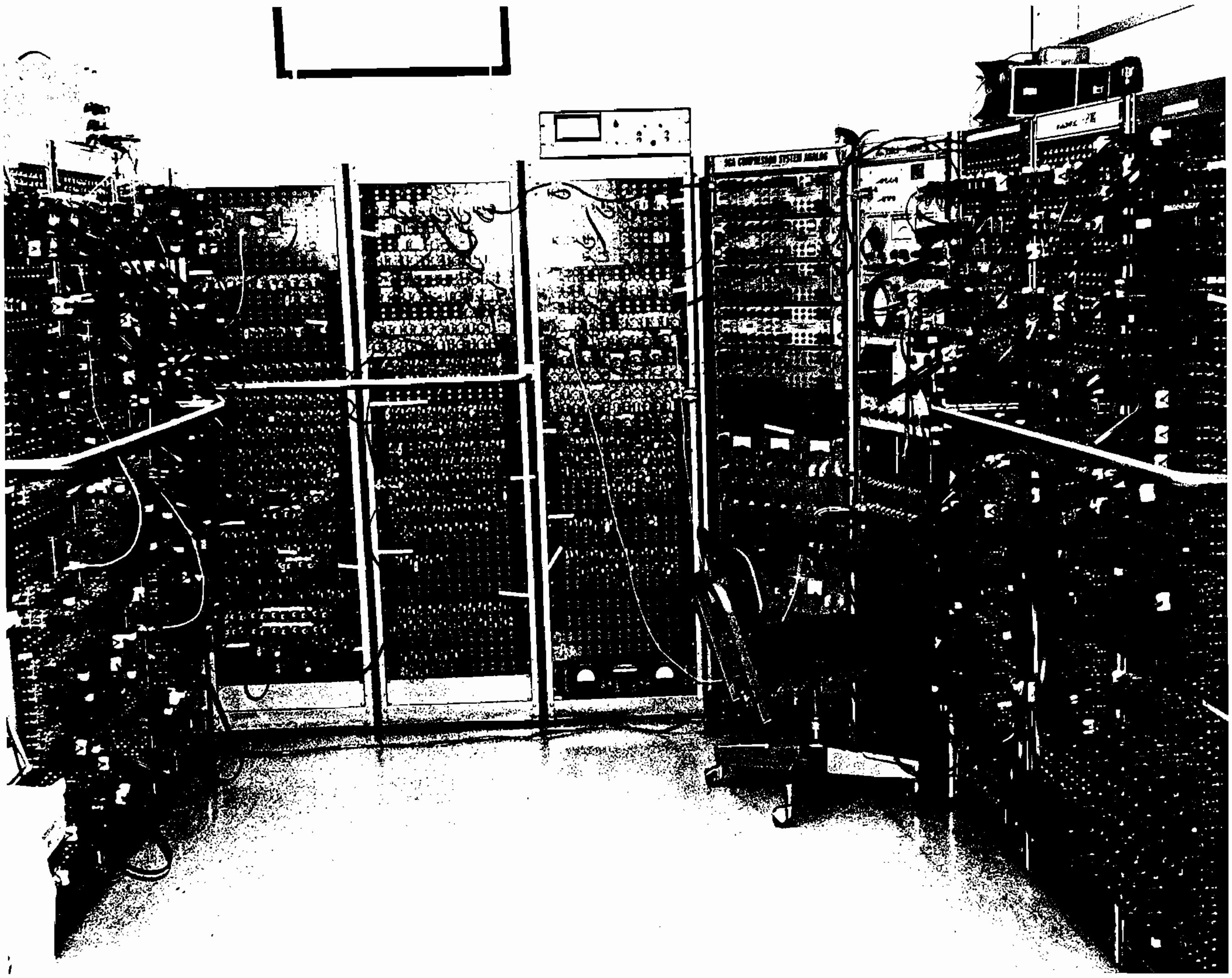

Figure 3. SGA-Compressor System Simulator with a Centrifugal Plant Study in Progress.

all simulator elements used are analogous to corresponding elements in the actual system, the transfer functions between elements or sections are automatically correct and the dynamic interaction between compressor, piping, and other system components is automatically taking place. Several thousand worldwide plant design and evaluation studies conducted over the past fifteen years stand as a testimonial to the validity, accuracy and dependability of this technique.

The first use of the SGA Compressor System Simulator for study of acoustic response of centrifugal plant piping was carried out over 5 years ago and several dozen such studies have been carried out to date. The basic handicap in the past was the lack of a centrifugal compressor analog necessitating the representation of compressor effect with some assumed impedance termination. This problem was alleviated with the design of the centrifugal compressor impedance analog which became operational in the early part of 1975 (Figure 4). As it is used today, the centrifugal analog properly duplicates the impedance of the compressor at any point of operation. It permits thus, determination of total acoustical system characteristics including the effect of compressor on acoustical responses in the system and resonant buildups at any point in the piping.

In addition to overall broad-band acoustical response analysis, the model can also be excited at specific calculated Strouhal frequencies at the points in the system where such turbulence is occurring. Of course, if external low frequency pulsation sources are present, such as series or parallel operation with reciprocating units, they can and should be duplicated in such studies.

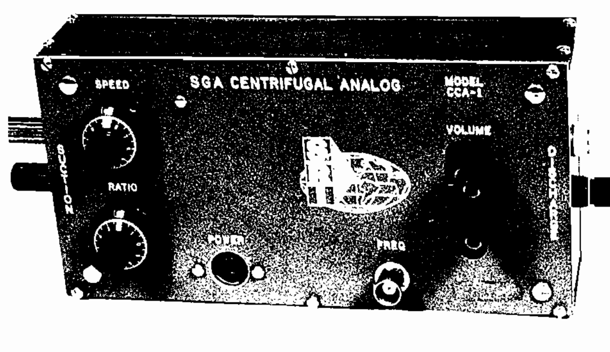

Figure 4. Current Model of Centrifugal Compressor Impedance Analog.

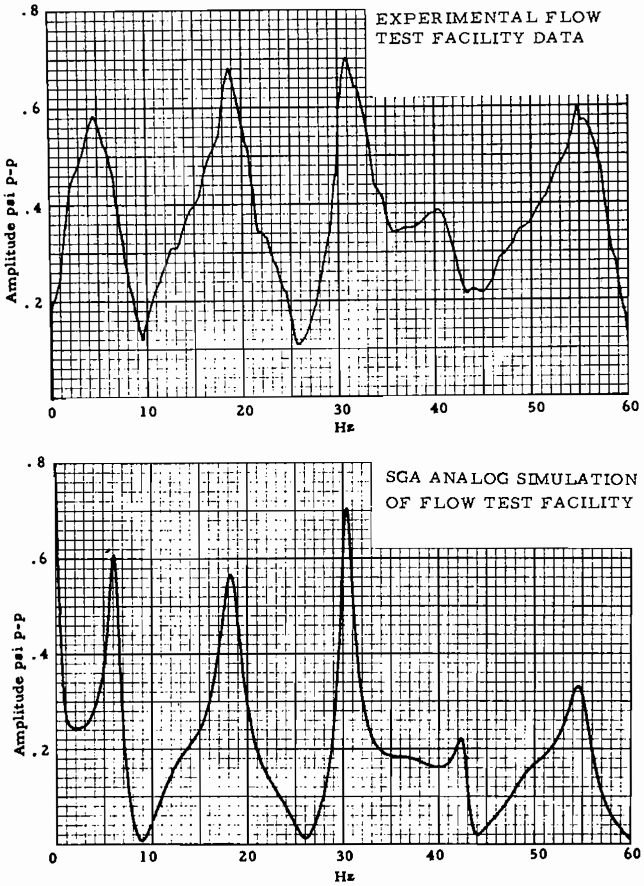

Figure 5. Comparison of Flow Test Facility and the SGA-Analog Simulator of the Facility Using Centrifugal Compressor Analog.

The validity and adequacy of the present model of the centrifugal compressor analog has been verified both in lab and field studies. The laboratory tests were conducted at SwRI's flow test facility consisting of a centrifugal compressor and flow modulator connected in series in a closed loop system. The accuracy of the compressor and piping system simulation is illustrated in Figure 5. Comparison of predicted acoustical responses with the pulsation data recorded in the field at a natural gas compressor installation is shown in Figure 6. While there were differences between field and lab system representation (two units running in series in the field while only one unit was duplicated in the lab), there is good correlation with the major responses recorded in the field. The successful results of previous studies further validate the effectiveness of the SGA-Dynamic Compressor Simulator as a design and evaluation tool for centrifugal compressor and pump installations.

VIBRATION CONTROL TECHNIQUES

Vibration control in the system requires not only the knowledge and control of acoustical excitation and acoustical resonances in the system, but also the determination of mechanical system response and the use of mechanical vibration control techniques. As mentioned earlier, the worst possible case would be the coincidence of mechanical natural frequencies with the acoustical resonant frequencies and the excitation frequencies. While this does not happen too frequently, when it does, excessive vibrations and fatigue failures are likely to result.

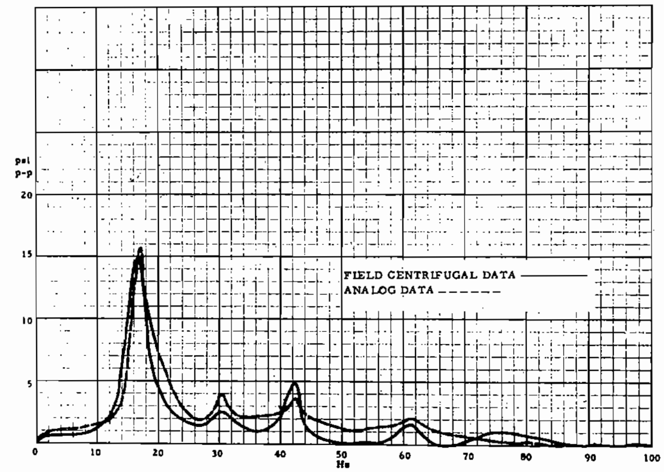

Figure 6. Comparison of Field and Analog Study Results Using Centrifugal Compressor Analog.

Various analytical computer techniques and graphic, nomogram methods have been developed for calculation of the mechanical response of various piping configurations. The problem in calculating mechanical natural frequencies is usually not the mathematical procedures used, but rather the interpretation of end conditions, branch connection flexibilities and piping supports.

In real life, end conditions are seldom the classical text book terminations such as fixed, simple supported, free, etc. More likely, one will find welded, clamped, three-dimensional welded or clamped, change of plane and similar terminations which require extensive field and experimental test data for proper interpretation. For example, a pipe section terminating in a welded connection on one end and a pipe clamp on the other will have a mechanical resonant frequency approximately 20 percent lower than the same section with ideal fixed-fixed termination. In addition, the pipe will be a continuous beam rather than cut off at the knife edges. It may also have axial loads due to thermal expansions affecting its natural frequency. The concentrated weights will be distributed over some finite length — all of which increase the difficulty of mechanical system response evaluation.

Interpretation of branch connection flexibility is also essential in mechanical natural frequency calculations. When a branch connection bends the shell of the line to which it is attached will rotate (bend) decreasing system stiffness and thus lowering the mechanical natural frequency (Figure 7). To acquire this knowledge, SwRI has carried out a six year experimental program supported by use of finite element analysis techniques. This effort generated in-plane, out-of-plane and torsional flexibility and stress intensification factors for all typical branch connections. This information is not only required for mechanical response calculations but is also essential for obtaining realistic stress level predictions in thermal flexibility studies of compressor piping systems.

The effect of pipe supports remains perhaps the least precise and most dependent on extensive field experiences. To illustrate the problem a concrete pier with a clamp may be a stiffness or mass addition depending whether the pier is stationary or moving with the pipe. Field installation practices, soil characteristics, weather changes, clamp main-

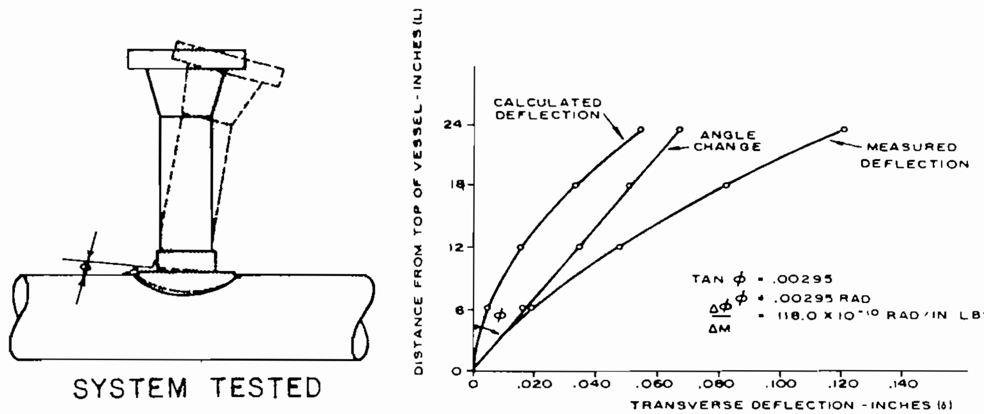

Figure 7. Calculated vs Measured Deflections of a Branch Connection.

tenance practices and other intangibles will affect the response of pipe supports. Progress has been made in recent years toward design of predictable and dependable (over a period of time) pipe supports but still more remains to be done.

CONCLUSIONS

Several mechanisms of low frequency excitation exist in centrifugal compressor installations of which flow turbulence excitation is most frequently found. In cases where acoustical and mechanical resonances coincide causing pulsation amplitude buildups, excessive vibrations frequently occur. In addition to piping vibrations, excessive torsional and lateral vibrations can also be caused by pulsations amplified by acoustic resonances in the system. To control these vibrations, an analysis and control of both the acoustical and mechanical system responses is required.

The use of the SGA Compressor System Simulator (SGA-Analog) with the recently developed Centrifugal Compressor Analog permits the study of acoustical system characteristics and optimization of acoustical design requirement, while the use of three-dimensional eigen matrix, eigen value techniques permits the determination of mechanical system characteristics and forced vibration responses. Knowledge based on experimental studies and field experience is essential in these analyses for interpretation of end conditions, branch connections, and piping supports if realistic results are to be obtained in mechanical system analysis.

Such techniques have been successfully used in a number of new plant designs as well as to solve problems at existing installations. The number of centrifugal installations with low frequency vibration problems is actually greater than generally assumed since some problems are being blamed on other causes, while others are not severe enough to be reported.

Further research efforts will advance the state of the art but the techniques available today are already capable of reducing the probability of low frequency problems by an order of magnitude. Thus, an acoustical and mechanical analysis of proposed centrifugal compressor or pump installation design can indeed be the ounce of prevention that eliminated the need for a pound of cure after the plant is put into operation.

FIELD CASE HISTORIES

Four field case histories have been selected to illustrate and amplify the comments made in the body of this paper. They include a gas compressor installation in Germany, turbine steam piping at a power plant in the U. S., complex piping system of a processing plant on the Gulf Coast and a natural gas plant in England. While these are but a few of the cases studied, they are typical of the problems encountered and the types of remedies used for their solution.

Case A — Excessive piping vibrations in a centrifugal gas compressor installation in Germany were observed in the 20-32 Hz frequency band when flow was changed from maximum to minimum. The maximum buildup occurred at 27 Hz at a low flow rate reaching 33 mils (0, 84 mm) peak to peak which is considered a level requiring immediate correction (Figure 8). The acoustical excitation was generated primarily in the compressor which was operating above the machine surge line but in the resonant surge condition. The maximum pulsation level of 7.8 psi (0,54 bar) was slightly less than 1 percent of static line pressure. However, because of several mechanical resonances in the bypass piping near 25-30 Hz excessive vibration of the recycle valve was occurring. The solution consisted in raising the mechanical natural frequency of the bypass line and increasing the minimum flow rate which was acceptable to the company.

Case B — Dangerously high vibrations were recorded in power plant steam piping to turbine. Field analysis indicated

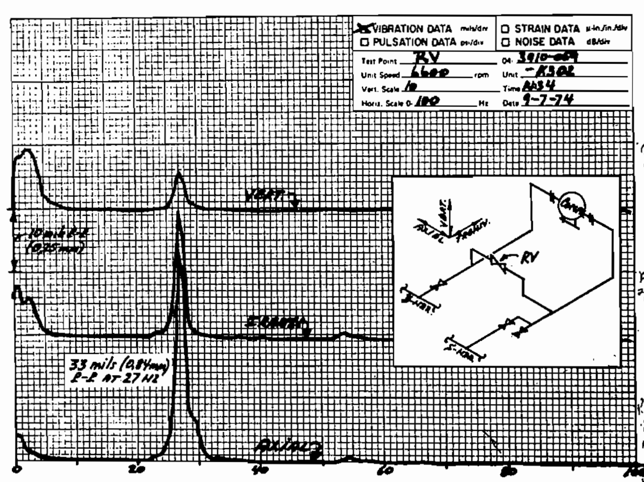

Figure 8. Field Vibration Analysis Data Recorded at a Gas Plant in Germany (Case A).

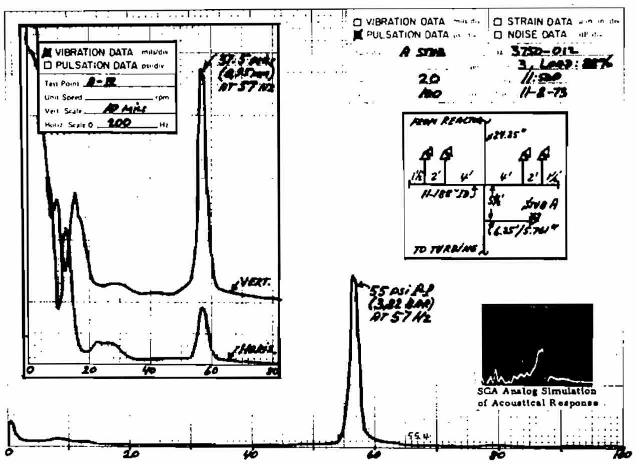

Figure 9. Frequency Analysis of Pulsations and Vibrations Recorded at a Power Plant (Case B).

that the flow induced turbulence at the safety valve header was the excitation source which was amplified by the acoustic resonance of the safety relief valve configurations (Figure 9). An acoustical response analysis on the SGA-Dynamic Compressor System Simulator (SGA-Analog) revealed strong acoustic resonance in the Safety Relief Valve system at 57 Hz as also shown in Figure 9. Furthermore, acoustical response data confirmed the field observed dependence of vibration buildup on the unit load (flow velocities).

Under the worst field conditions, the maximum pulsation and vibration levels reached 55 psi (3,82 bar) and 37.5 mils (0.95 mm) at 57 Hz, respectively. Since fatigue failures can occur under these conditions, customer was advised to avoid the critical flow range until solutions are developed. Considering that the acoustical resonances were responsible for the magnitude of pulsation buildup, several possible acoustical modifications were designed on the SGA Compressor Simulator and the problem was solved when one of the modifications was installed in the field.

Case C — Two large mezzanine mounted centrifugal compressor trains with rack mounted piping at a Gulf Coast gas processing plant were reported to have excessive low frequency vibrations. A field study revealed vibrations at 14 HZ reaching 30 mils (0,76 mm) p-p at several locations and a maximum of 70 mils (1,78 mm) at one point in the system. The recorded pulsation levels were also at 14 Hz with a maximum amplitude of 2.6 psi (0,18 bar) p-p (Figure 10).

These pulsation levels are not insignificant considering that these trains were operating from a vacuum to 185 psia (12,85 bar) in three stages of compression. Thus, even on the final discharge system they were 1.5 percent of static line pressure. Cursory evaluation did not identify the source(s) of acoustical excitation at 14 Hz and, consequently, an acoustical evaluation on the SGA-Compressor System Simulator was recommended. However, the company chose to proceed with mechanical means of vibration control. Carrying out various proposed modifications has reduced vibrations in the most critical areas eliminating thus the known safety hazards and leaving less critical areas as maintenance items.

Case D — Vibration at 16 Hz was severe enough at a natural gas plant in England to require operation with a partially opened bypass valve. Data taken by the company indicated buildup to 18 mils (0,46 mm) p-p at the discharge riser to compressor unit. Vibration levels were found to be very velocity dependent. They were changing from 2 to 18 mils with some 20 percent change in flow velocity and peaking at about 80 percent of maximum flow rate.

At the subsequent field study, the maximum vibration conditions could not be duplicated due to gas demands. However, real time signatures of pulsations and vibrations in the discharge riser at the compressor confirmed that the vibrations were being excited by pulsations in the line. Pulsation data (Figure 11) shows buildup to 5 psi (0,35 bar) p-p at 16 Hz when the unit is running around 5,000 rpm. As speed, and thus flow change, the pulsation levels drop off rapidly. Vibration data (figure 12) follows changes in pulsation levels at 16 Hz with the maximum recorded level of 6 mils (0,15 mm) p-p.

The problem was identified as Strouhal frequency turbulence generated at unit bypass junction and amplified by acoustic resonance of the system. It is interesting to observe that the mechanical resonance of the discharge riser was at 24 Hz thus vibrations at 16 Hz were forced vibrations (by some 4,500 lbs, or 2,000 kg, p-p shaking force). Vibration signature shown in

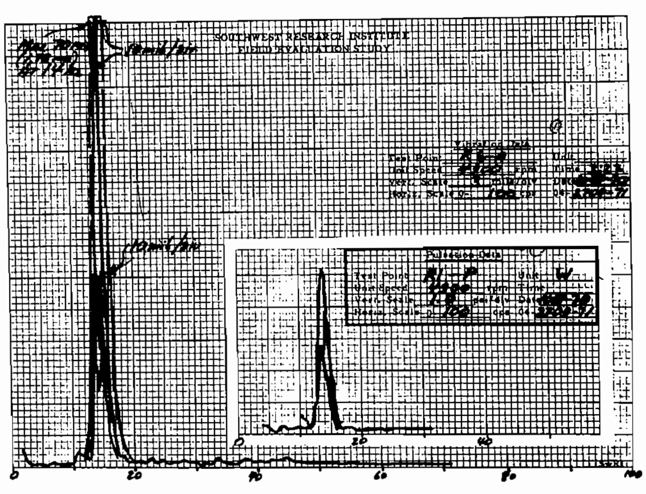

Figure 10. Field Pulsation and Vibration Data Obtained at Gulf Coast Processing Plant (Case C).

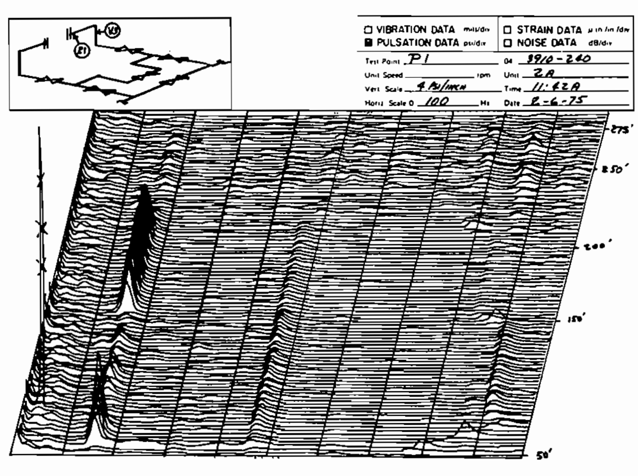

Figure 11. Characteristic Pulsation Signature Data Recorded at a Gas Plant in England (Case D).

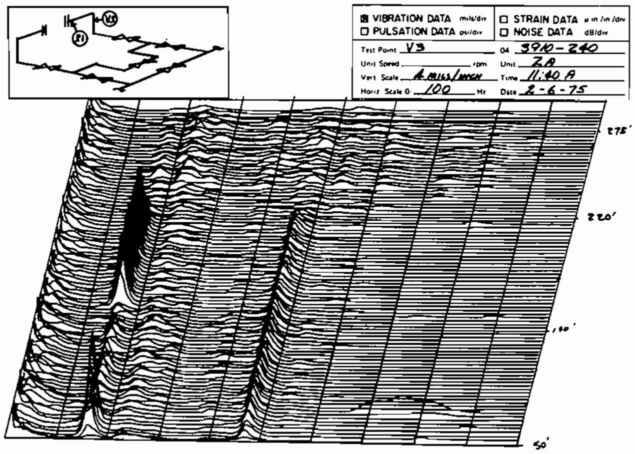

Figure 12. Vibration Signature Corresponding to Pulsation Signature Data in Figure 11 (Case D).

Figure 12 indicates some vibration at 24 Hz, even so there was hardly any acoustical excitation present at this frequency.

Since the vibrations were not occurring at a strong mechanical system resonance, mechanical modifications were not considered to be practical. With the help of the SGA-Compressor System Simulator, simple acoustical modifications were developed controlling acoustic resonant response at 16 Hz without increasing the response at 24 Hz. It is interesting to note that it was the total system response from suction header, though the compressor and to the discharge header that was responsible for acoustic resonance at 16 Hz rather than the bypass stub length.

The case histories presented illustrate not only the fact that low frequency pulsations and vibrations can exist at centrifugal compressor and pump installations, but also the magnitudes of vibration buildup which give concern for system reliability if such problems are left unchecked. Fortunately, acoustical and mechanical evaluation techniques are now available to alleviate the possibility of such low frequency problems at plant design stage and to determine effective and economical solutions if problems are discovered after plant startup.

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