Vibration Related Failures of Small-Bore Attachments

Brian C. Howes, M.Sc., P.Eng
Chief Engineer
bhowes@betamachinery.com

Chris B. Harper, P.Eng
Project Engineer
charper@betamachinery.com

Beta Machinery Analysis Ltd., Calgary, AB, Canada, T3C 0J7

ABSTRACT

Vibration can cause failures around reciprocating compressors. One major source of failures today is small-bore appendages found particularly on compressor station vessels and piping. The simplest solution to this class of problem is to avoid appendages. Unfortunately, avoidance is not always practical.

This discussion of the topic includes design considerations and diagnostic advice for as-built systems.

A discussion of vibration and stress guidelines for operating machines includes an assessment of an overall guideline versus single frequency guidelines.

INTRODUCTION

Small-bore appendages on reciprocating compressor systems cause serious problems with otherwise safe operations when they fail. Costs related to small-bore appendage failures not only include replacement and lost production costs, but also the cost of checking the replacement (vibration checks, dye penetrant check, and/or x-ray checks).

Small-bore appendage issues should be dealt with in the design stage and during start-up of the system. It is better to determine which appendages are likely to have problems and correct them than to wait for a failure to occur.

Two variables that contribute to small-bore appendage failure:

- appendage design and construction; and
- vibration of appendage attachment point (base motion).

There are two techniques for screening small-bore appendages for problems:

- measuring strains; and
- measuring vibrations.

When measuring vibrations, there are three issues to consider:

- whether to measure vibration at individual frequencies (frequency domain) or measure overall vibrations (time domain);
- whether to measure vibration velocity or displacement; and
- whether to measure absolute vibration or relative vibration.

All these issues will be discussed in this paper.

APPENDAGE DESIGN

Examples Of Small-Bore Attachments

The following photographs show some typical small-bore appendages (Figure 1, Figure 2, and Figure 3). The pictures reinforce how many small-bore appendages are typically found in a reciprocating compressor package.

Reference 4 indicates, and our experience confirms, that most small-bore appendage problems occur with:

- cantilever pipe with masses at the end (e.g., flanges, valves); and
- with long unsupported small-bore piping runs with and without large masses (e.g., valves).

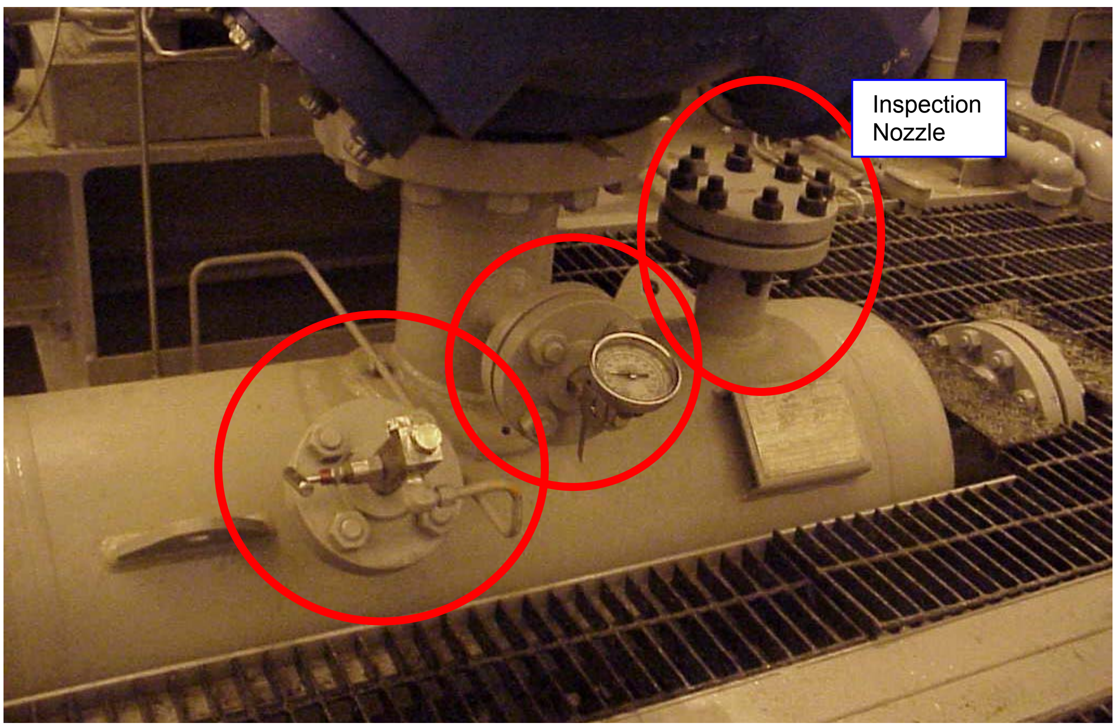

Figure 1. Discharge Bottle Small-Bore Appendages

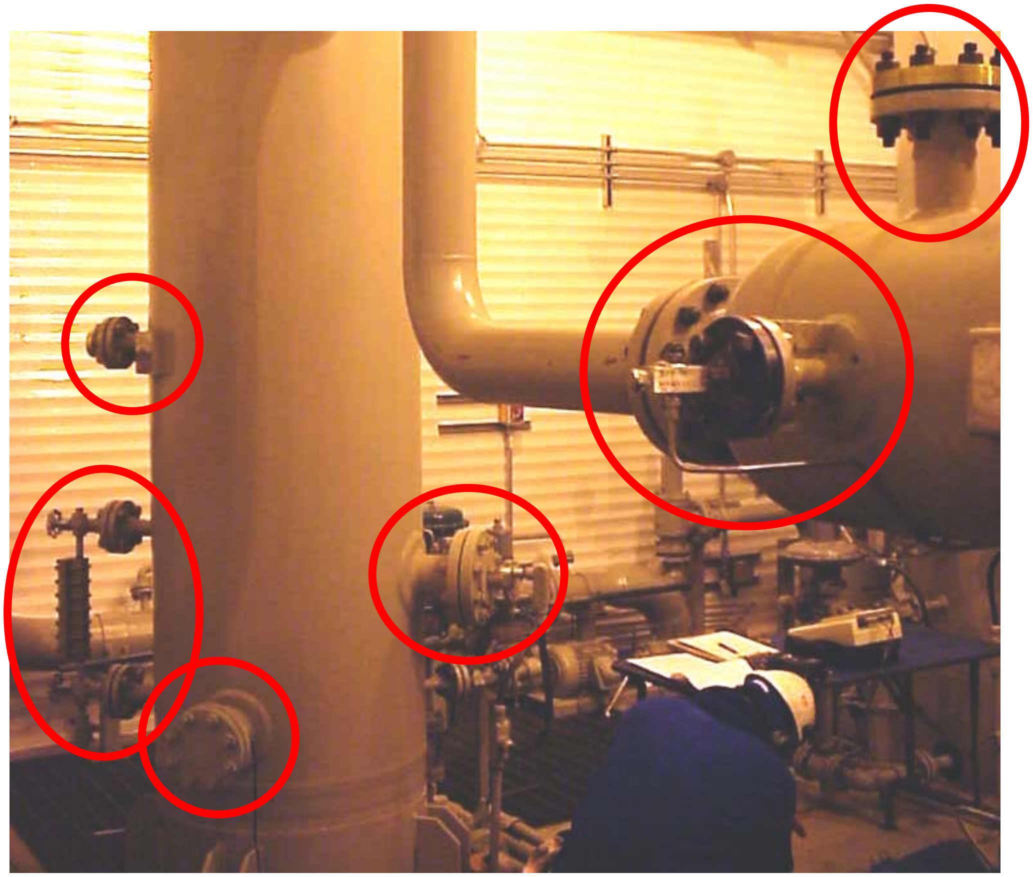

Figure 2. Suction Bottle / Scrubber Appendages and Tubing

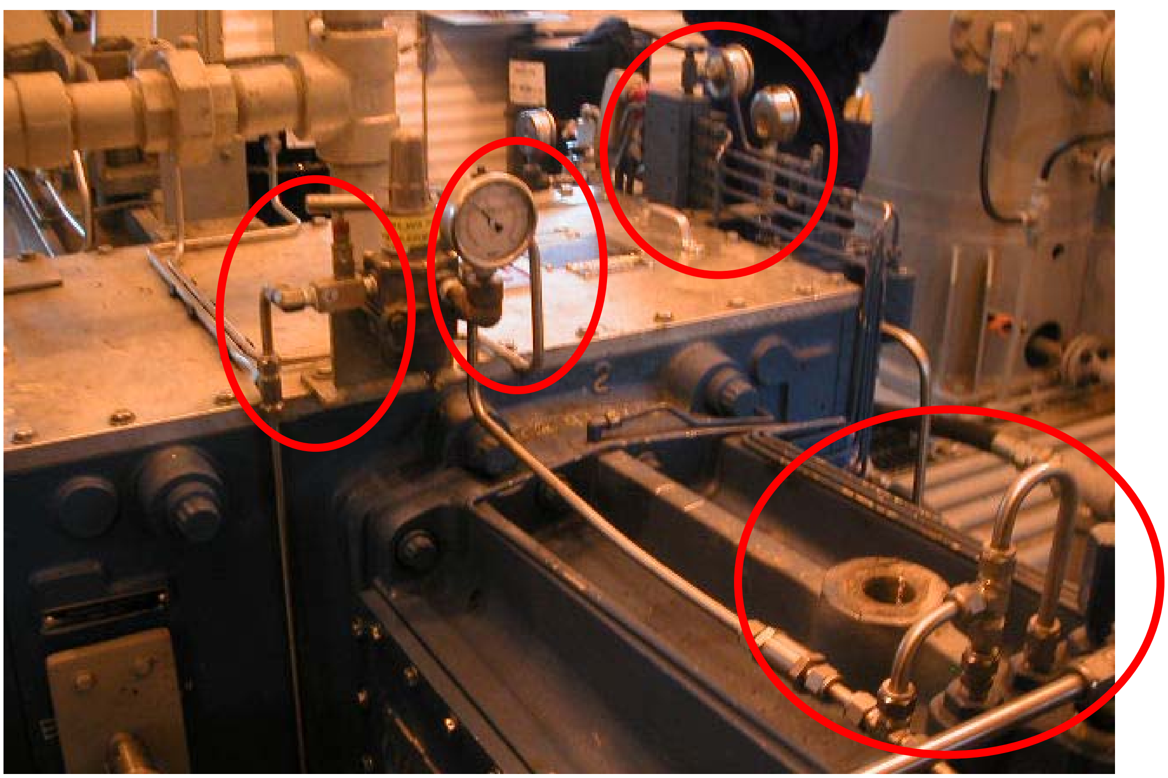

Figure 3. Frame Appendages and Tubing

Improving the Design

Good small-bore appendage design can be summarized with three R's:

- *Removing* appendages that are not needed;
- *Redesigning* appendages so they have less cantilevered and unsupported mass; and
- *Relocating* appendages to locations of less base motion.

Failures of small-bore appendages are generally due to alternating stresses caused by vibration of the appendage at or near the mechanical natural frequency (MNF) of the appendage (resonance). Designing the MNF of the appendage to be well away from the attachment point base motion frequencies will go a long way toward avoiding such failures. In addition, the MNF of the appendage should not correspond to the MNF of the pipe or vessel to which it is attached because the attachment may act as a vibration absorber.

The orientation of an appendage connected to a system has a significant impact on how much vibration an appendage will exhibit. Orienting an appendage in the direction of compressor cylinder motion is typically better for appendages close to a compressor cylinder. Unfortunately, cylinders occasionally vibrate in the vertical direction, so consideration must be given to vibration in that direction as well.

Where an appendage is mounted is also an important consideration. The vibration at the attachment point (base motion) is the driver for the vibration of the appendage. The appendage should be located at a point of low vibration. Some examples of better design are:

- Installing a pressure tap on a cylinder discharge nozzle instead of putting the same appendage on the discharge bottle.
- Installing a temperature sensor on the suction scrubber instead of the suction bottle cylinder nozzle.
- Installing a relief valve on a main piping run near a clamp instead of on an unsupported span or, worse yet, on the discharge bottle itself.

Studding outlets should be considered when providing connections for temperature and pressure measurements instead of a cantilevered pair of flanges. Studding outlets have the disadvantage of being somewhat more expensive to buy and having longer delivery than standard flanges. The saving comes in the elimination of potential problem vibration on the appendage and fewer failures.

Small-bore appendages can be gusseted down to a repad, but care must be taken not to introduce stress concentration factors near the attachment point. Reference 1 has several figures that show the preferred method of gusseting, where the gusset connects to the main piping (or vessel) several small-bore diameters away from the small-bore appendage attachment point.

Isolating small-bore appendages from the base motion is another option that should be considered when controlling base motion is difficult. Typically, this involves remote mounting the appendage (like the sight glass shown in Figure 4) and running tubing (or braided hose) to the appendage. The flexibility of the tubing or hose effectively isolates the appendage from the base motion. This will greatly reduce the chance of a failure.

Irrespective of location or design, the amount of dynamic stress that an appendage connection can tolerate will be a function of residual stress in the heat affected zone. If the weld is not stress relieved there could be large residual stresses that reduce the endurance limit for the attachment by an unpredictable amount. Use of correct weld procedures can greatly reduce these residual stresses.

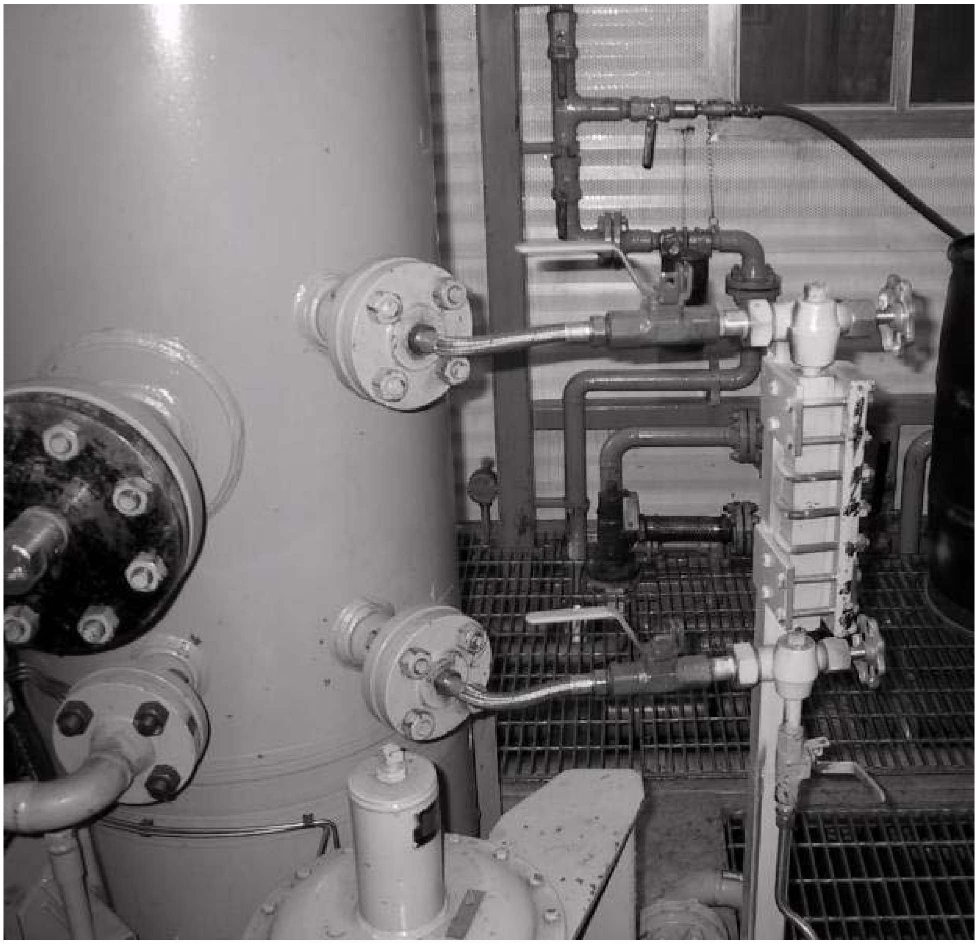

Figure 4. Example of Remote Mounted Sight Glass Connected to Scrubber with Tubing

BASE MOTION

Sources of Excitation Forces

There are several sources of high frequency forces that can excite the mechanical natural frequencies (MNFs) of small-bore appendages. Generally, these MNFs are above the fourth order of run speed, and often as high as 150 to 200 Hz.

Gas forces within a cylinder contain harmonics of run speed and act in the direction of piston motion (stretch direction). Generally, these stretch forces decrease in amplitude as the frequency increases. Any mechanical natural frequencies of the cylinder assembly (typically between 150 to 250 Hz) amplify the effect of these harmonic forces, causing more base motion of the cylinder (stretch vibrations).

Pulsation in cylinder nozzles can create significant unbalanced forces in the vertical direction that can excite mechanically resonant components in the vertical direction. The pulsation-induced forces in the bottles and piping can be significant at high frequencies as well.

Unbalanced forces due to pulsation resonances in the cylinder gas passages (between the head end and crank end) have been implicated on occasion as the cause of high vibration and failures of appendages. This so called gas passage resonance force will often be in the range of 250 to 350 Hz on high speed compressors. A coincidence of the gas passage resonance frequency and the MNF of the cylinder in the stretch direction can result in high vibration near the compressor.

Control or Elimination of Forces

Bottle and nozzle forces can usually be controlled by design (e.g., symmetrical chambers) or by added damping (cylinder nozzle orifice plates).

It is much more difficult to change the gas forces inside a cylinder as they are a function of operating conditions and compressor geometry. Care must be taken to avoid coincidence between stretch vibration frequencies and small-bore appendage MNFs

The acoustical resonance of the gas passage can be avoided only with changes inside the cylinder casting. This is a difficult area to modify, even during manufacture. Cleverly designed orifice plates inside the gas passage can limit resonant pulsations. Chambers can be added to the valve covers to move the acoustical natural frequency (ANF), but this should be a last resort because of the potential for creating MNF problems. Careful consideration of the expected gas properties during the life of the compressor can allow selection of the appropriate cylinder model to avoid coincidence between the cylinder MNF in the stretch direction and the gas passage ANF. This is the preferred approach. The MNF of the cylinder can theoretically be changed to avoid coincidence between the gas passage ANF and the cylinder stretch direction MNF.

Other Considerations

“Upset” conditions are difficult to anticipate in the design stage but can also be a source of small-bore appendage failure. Two examples are given below.

- Failure of small-bore appendage occurred for no apparent reason. Analysis after the failure was repaired could find no reason for the failure. Vibrations and pipe strain were found to be reasonable. It turned out that the support under the appendage had come loose, allowing the vibrations to increase significantly. These higher vibrations initiated the crack.
- If a compressor valve fails, gas loading on the cylinder will change significantly because it is made single-acting instead of double-acting. The cylinder stretch forces and

vibration will be different, which sometimes leads to significantly different vibrations on appendages.

The range of possible operating conditions should be considered. Many compressor cylinders are operated over a wide range of compression ratios and load steps (head end and crank end percent clearances). The cylinder stretch forces and vibrations will vary over a wide range as a result. This is especially important when measuring base motion of the cylinders in the field. The operating condition at which measurements are taken may not be the worst case.

The entire speed range must be considered as well. Small-bore appendage vibrations that are acceptable at design speed may be much worse at a lower speed. Even if the unit will rarely see that speed, high amplitude high frequency vibrations can cause fatigue failures very quickly (e.g., a small-bore appendage vibrating at 70 Hz will reach a million cycles in four hours). Remaining at a speed that corresponds to an MNF of a small-bore appendage (i.e., resonance) for even a short time can cause a fatigue failure, if the vibrations are severe enough.

SCREENING GUIDELINES

After installation and startup of a compressor system, field testing should be performed to ensure that vibrations and stresses are acceptable and to document the baseline conditions for the system. Small-bore appendage failures can happen quickly so field assessment should be given high priority.

Ideally, guidelines for screening should be derived from a finite element model of the actual geometry of the appendage. This can be done economically by use of parametric models. The goal is to calculate the stress per deflection (psi/mil) at the reference test point (usually the top of the appendage or a fixed dimension away from the connection point). Divide the allowable stress (endurance limit) by the stress per deflection to get the allowable reference test point deflection. This can be done for every mode shape, but typically the first-order mode is most important.

For complicated geometry, vibrations as a screening tool will be less accurate. Ultimately, stress estimates from strain gauge readings may be necessary for some locations.

STRAIN MEASUREMENTS

The most obvious way to determine if stresses due to vibrations are acceptable is to measure strains with strain gauges. There are some drawbacks to this approach.

Attaching a strain gauge is time consuming. The use of weldable strain gauges is probably faster than glue-on gauges, but still much more difficult than attaching a vibration transducer. Reference 1 mentions a “press-on gauge” that allows stress measurements to be done efficiently. Unfortunately, we have been unable to locate information on “press-on gauges”.

Another problem with strain measurement is where to locate the strain gauge. Reference 1 provides guidance regarding location and gauge size (from British Standards BS7608 and BS5500).

In view of these factors, strain measurements are often not practical for screening vibrations on appendages. However, they can be useful in evaluating appendages that exhibit high vibration.

VIBRATION MEASUREMENTS

Both References 1 and 2 focus on vibration measurements as a screening tool for small-bore appendages. Both recommend measuring vibration velocity at individual frequencies on the appendages. Geometry of the attachment must be taken into account in determining the allowable vibration level. Charts are provided for typical geometries.

Individual Frequency Components

The use of individual frequency vibration components to assess the likelihood of failure for an attachment is only valid if:

- there is only one vibration component of significance; and
- if the vibration of the attachment point (base motion) is insignificant compared to the vibration of the appendage.

The case study below illustrates some problems when these two assumptions are not valid.

Case Study #1 - Discharge Bottle Inspection Nozzle

This case study is taken from an inspection flange on a discharge bottle (labeled "Inspection Nozzle" in Figure 1 on Page 2). The unit is running at 1200 RPM and all vibration measurements are taken at the top of the inspection nozzle flange in the horizontal (cylinder stretch) direction.

The spectrum plot (Figure 5) shows high velocity vibration at four sequential orders of runspeed. The base motion (the horizontal motion of the discharge bottle) is primarily 2x compressor runspeed with little motion at 9x-12x.

The small-bore appendage vibration at each of these four sequential orders are above or at the guideline recommended in Reference 1 for this configuration, which is approximately 24 mm/s or 1 ips true peak. However, a finite element (FE) model of the appendage (Figure 6) showed the allowable reference point deflection was 10 mils peak-to-peak (pp). Compared to this displacement guideline, the individual frequency vibrations were below guideline. However, the overall vibration gave reason for concern.

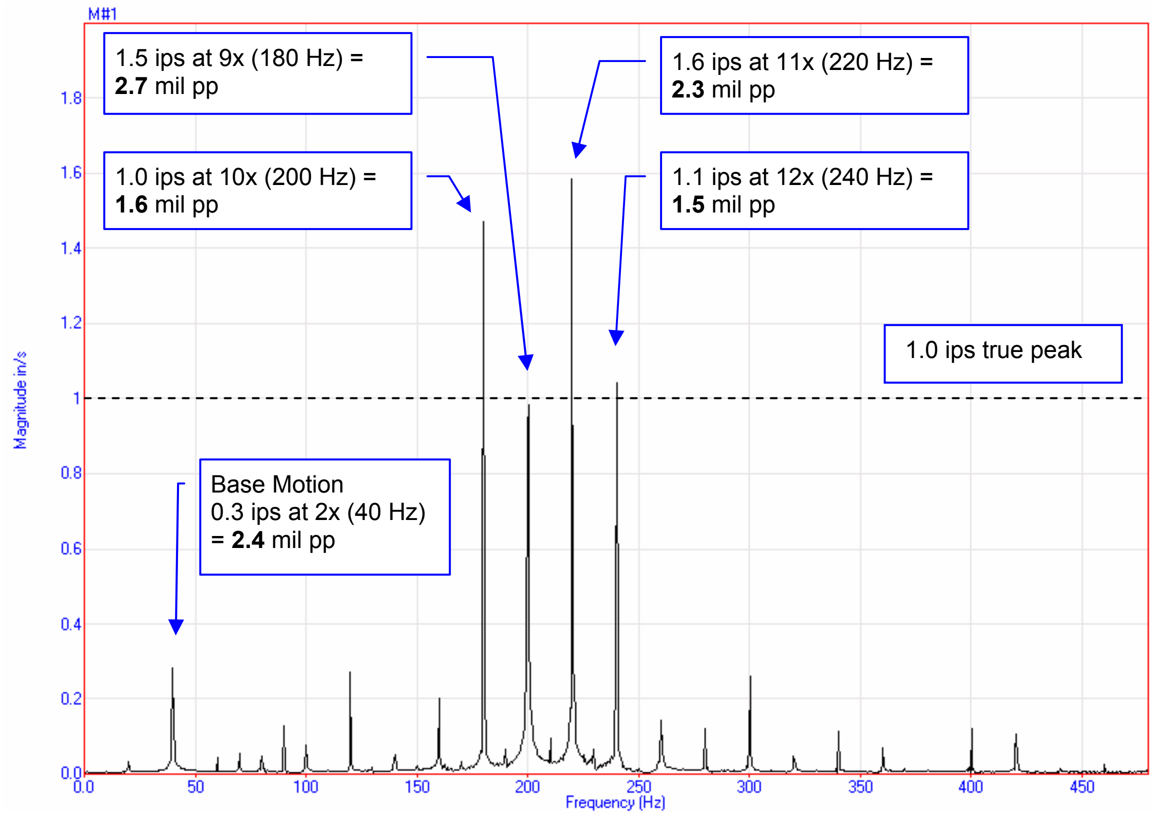

Figure 5. Inspection Nozzle - Spectrum of Velocity Time Waveform

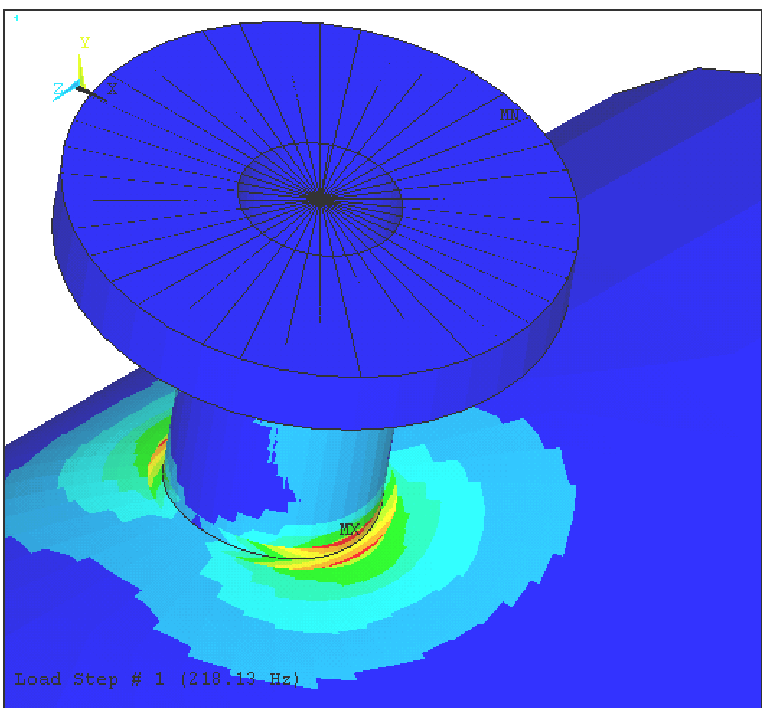

Figure 6. FE Model of Inspection Nozzle - First Order Mode Stress Distribution

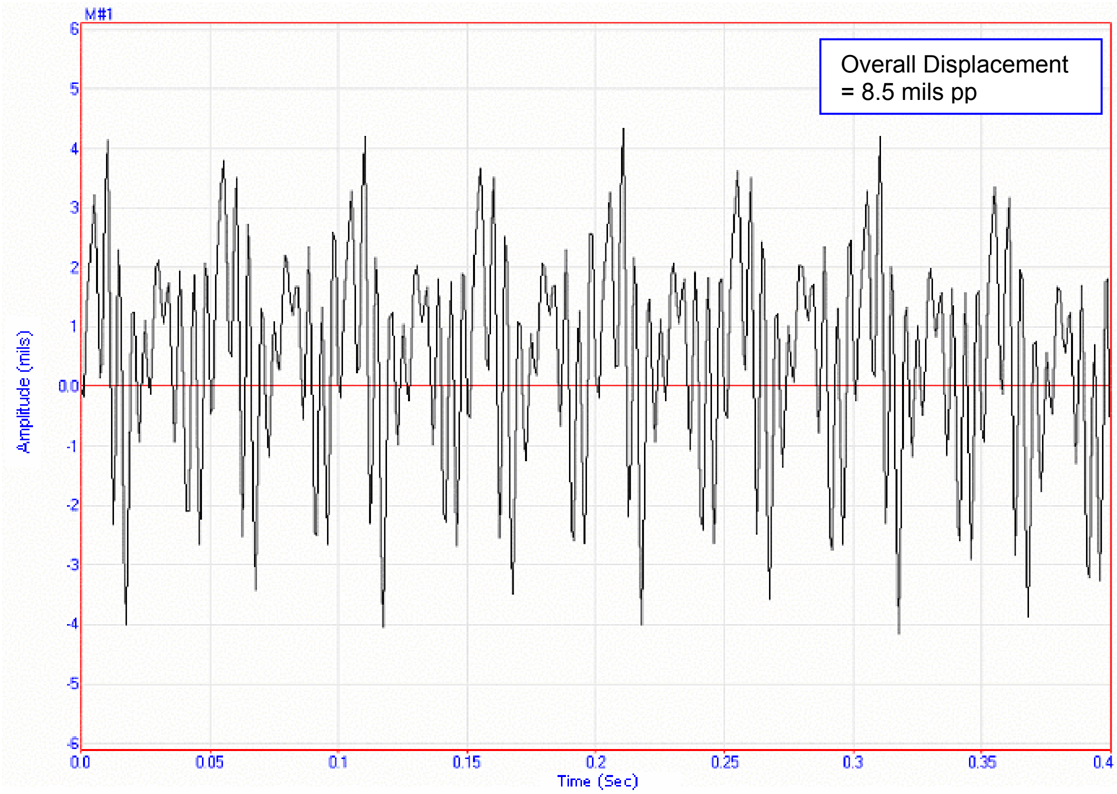

Figure 7. Velocity Time Waveform Integrated to Displacement

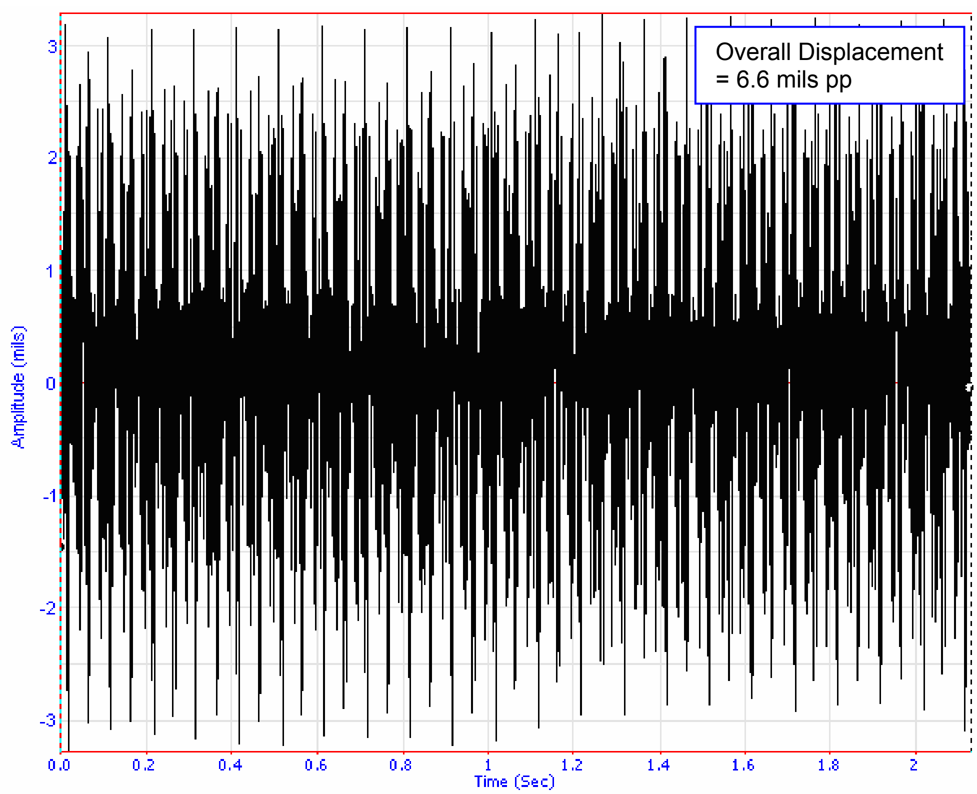

Figure 8. Velocity Time Waveform Integrated to Displacement with 1x and 2x Filtered Out

When analyzing the overall vibration, two techniques were used. The first technique was to analyze the overall displacement of all of the frequency components of the entire spectrum. This is typically how vibration data is captured and is relevant when the base motion is not significant when compared to appendage motion.

The second technique was done by analyzing the overall displacement of the frequency components above 40 Hz (i.e., with 1x and 2x filtered) thus removing the component of base motion from appendage vibration.

(Note that the overall vibration required is in true peak units, as opposed to RMS. Stress is proportional to amplitude, not to average motion.)

The result of each technique yielded significantly different results, as shown in Figure 7 and Figure 8 above. While the overall displacement without filtering (8.5 mils pp) was close to the allowable reference point deflection (10 mils pp), the filtered overall displacement was better (6.6 mils pp). The appendage has been in services for at least six months and has not had a failure.

This example shows that in certain cases (i.e. when base motion is significant and there are multiple vibration peaks), overall displacement vibration is better as a screening tool than vibration velocities at individual frequencies.

Profiling Overall Differential Displacements

As a result of this and other similar cases, it was decided that a better vibration screening tool for appendages should be based on true peak overall displacement. Further study showed that the results of such measurement at a single point (on the top of the appendage) could give excessively conservative results due to the combined rigid body motion of the appendage and the pipe to which it was attached. The differential motion of the appendage relative to the pipe is required. It is also possible that the differential vibration could be greater than the absolute vibration of the top of the appendage. Either situation is important to know.

Figure 9 shows a profile of measurements made in this fashion on 16 similar appendages on each of four units in a compressor station. The profile displays true peak-to-peak differential displacement between the base of the appendage and the lumped mass at the top of the appendage (red bars). For comparison, the absolute vibrations measured at the top of the appendage are also shown (cyan bars). In many cases, the absolute vibration differed significantly from the differential motion.

Measuring the differential motion on small-bore appendages requires a special setup (Figure 10). The input from the base transducer must be subtracted from the input from the reference point transducer. This can be done either during data collection with a subtraction box or from two time waveforms using software.

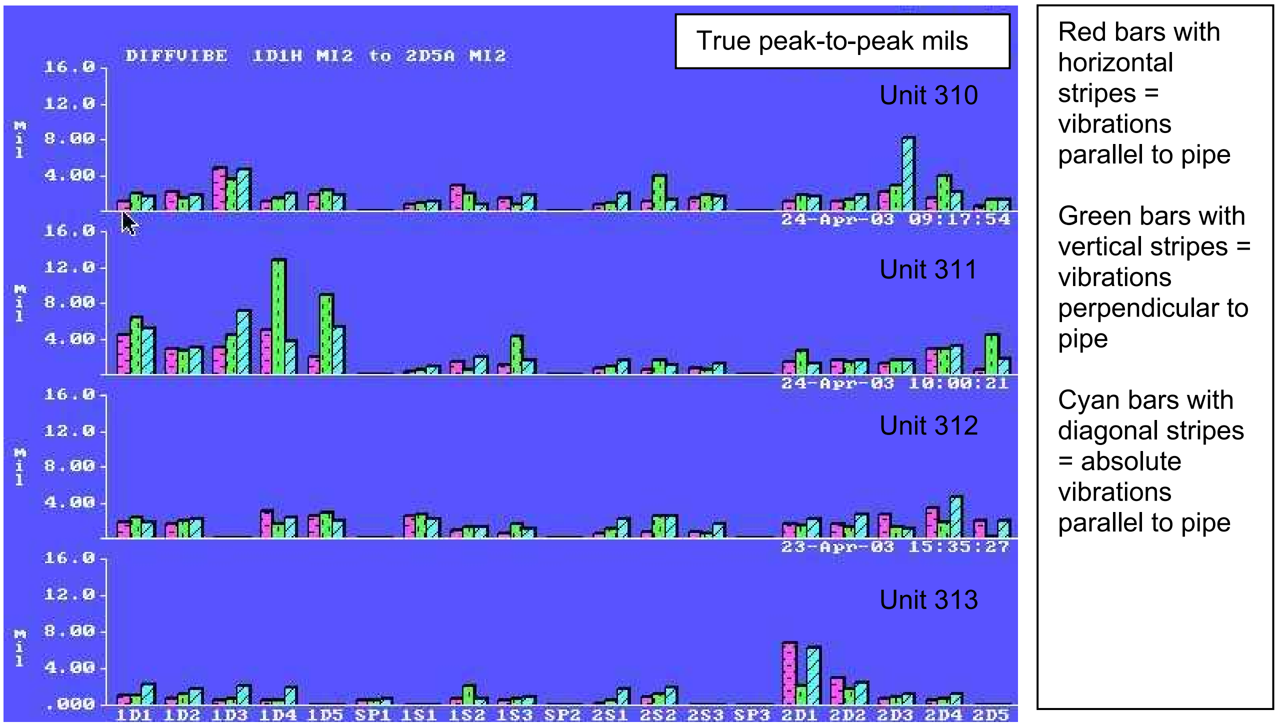

Figure 9. Profile of Vibrations on Small-bore Appendages at Multiple Locations

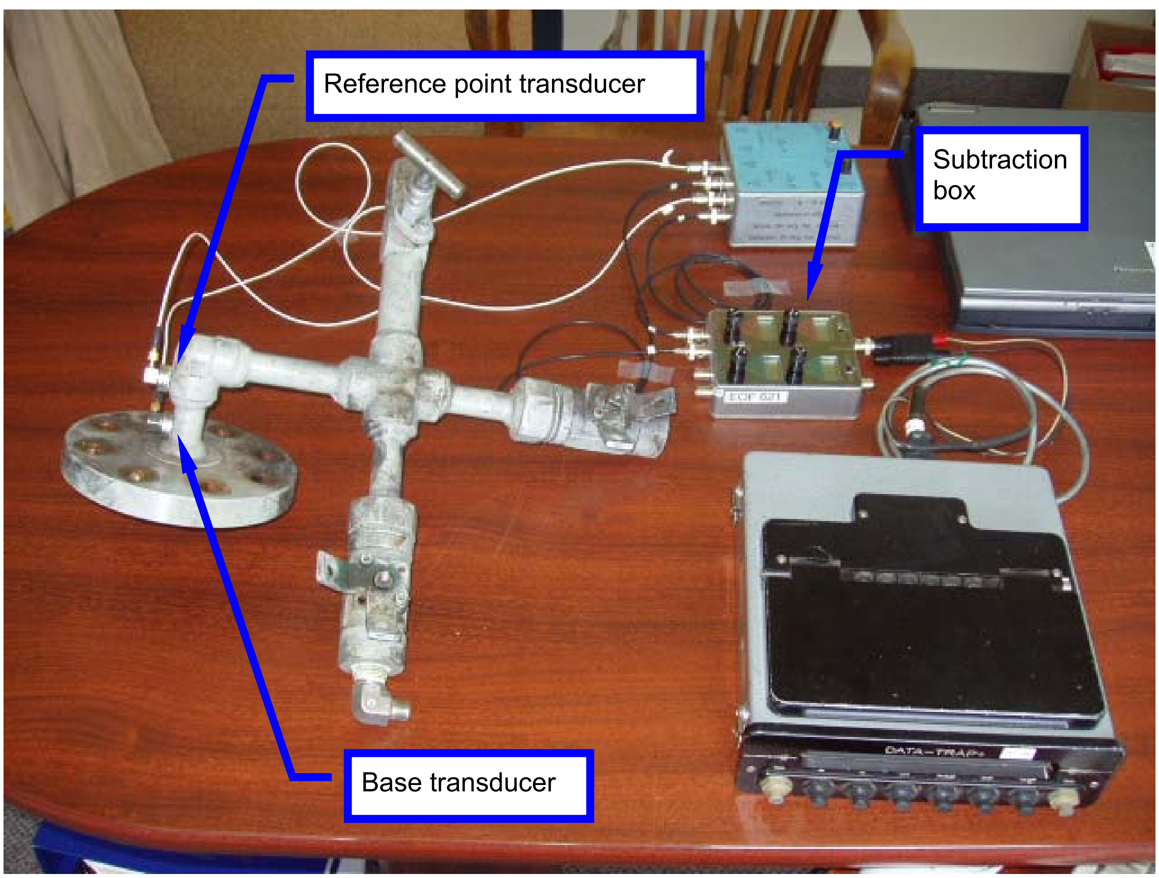

Figure 10. Analysis Setup to Measure Relative Vibration between Base and Reference Point

CONCLUSIONS

Small-bore attachments are a significant source of failures in reciprocating compressor installations. Many failures, mostly due to vibrations, occur on these seemingly unimportant parts of large piping systems. More attention is warranted for small-bore attachments both in the design stage and after start-up.

We recommend removing small-bore attachments where possible, redesigning them so they are less susceptible to vibrations, and relocating them to areas of less base motion. Consider the mechanical natural frequency of the appendage relative to the frequencies of vibration expected at the attachment point.

Residual weld stresses are a major source of uncertainty in the prediction of the risk of failure for appendages as the endurance limit is reduced by the presence of residual stresses. Stress relieving welds is recommended in critical areas.

The influence of the small-bore appendage base motion and the effect of high vibrations at multiple frequencies must be considered when interpreting measured vibrations. Differential overall vibrations can be a useful tool in screening small-bore attachments. Guidelines should be based on geometry, preferably determined through a finite element model.

REFERENCES

1) The Marine Technology Directorate Limited (MTD), "Guidelines for the Avoidance of Vibration Induced Fatigue in Process Pipework," Publication 99/100, 1999.
2) Wachel, J. C., SGA-PCRC Seminar on Controlling the Effects of Pulsations and Fluid Transients in Piping Systems, Chapter VI, Piping Vibrations, November 1981.
3) Wachel, J. C., "Field Investigations of Piping Systems for Vibration Induced Stresses and Failures," Pressure Vessel and Piping Conference, ASME Bound Volume #H00219, June 27-July 2, 1982.
4) Babcock Energy Limited, "Assessment of Design and Operating Experience of Pipework Fatigue in the Offshore and Petrochemical Industries", OTI 95 631, 1997.