SCREW COMPRESSOR VIBRATION ANALYSIS

Portable real time vibration analyzers are being used more frequently today as a means of analyzing vibration and noise on all rotating machinery. These instruments are excellent diagnostic tools. A vibration meter in the hands of a trained technician is a powerful troubleshooting tool, and nearly essential in assuring rapid assessment of machinery problems, as well as giving confidence that a new machinery start-up is free of problems that could be unsafe.

Noise or vibration analysis on screw compressors is complex. Problems may be created for a myriad of reasons. This report is intended to stimulate ideas, to relate experience on problems that have been encountered, and to give a better understanding of what might be behind a particular problem. It is not intended to give detailed directions in vibration measurement.

A Few Basics

Most real time analyzers will give a plot of frequency on the X-axis versus amplitude on the Y-axis. Frequency can be measured in cycles per minute or more commonly Hertz (cycles/sec.). Amplitude can be in displacement, velocity, or acceleration, displayed in many different units, either on linear or log scales. The writer prefers a log scale of frequency in Hz with a linear scale of velocity in in/sec or in some cases acceleration in g's. The units of measure should be recorded on the plots because of the variety of unit systems in use. When vibration readings are sent from the field for comment 1 hour is spent trying to interpret the scales for every 1 minute spent interpreting the data.

Many instruments give options on narrow band readings, or various other types of scaling that average readings in frequency bands. Narrow band readings generally provide the most useful information for successful troubleshooting.

It is also important that the amplitude scale be calibrated. Some instruments use charge amplifiers to turn the signal from the accelerometer into a voltage readable by the RTA. Calibration must be performed with a device traceable to standards, or the measured amplitude values are less reliable. When calibrators are not available, use the manufacturer's calibration (g's/volt) to set the gain on the RTA. A note should be made on the readings that amplitude is not calibrated.

Normally screw compressor vibrations of interest are between 60 Hz and 10,000 Hz. Good quality, small accelerometers should be used to take readings. A magnetic base mount is normally a good choice up to 7,000 Hz. Hand held probes should not be used for any measurements over 1,000 Hz because the readings will not be accurate. There are however some special probes used for very high frequencies (ultrasonic) that will give repeatable data.

There are two basic reasons for taking vibration data which differ considerably in purpose.

1. New equipment analysis

2. Machinery condition monitoring

New Equipment Analysis

Vibration analysis is often performed on newly installed equipment to determine the cause of suspected high noise or vibration. If levels seem excessive to experienced listeners, vibration analysis can give clues for where to look, or help determine if vibration levels are high enough to warrant corrective action.

Hold something metal in your hand (to avoid burns) and touch all the components of the equipment to determine if there are places that seem to have higher levels than others. Once the highest vibration areas are found, use the accelerometer and instrument to determine frequency and amplitude.

If the levels are higher on piping or on the oil separator than on the compressor or driver, the vibration is probably the result of gas pulsations or resonance and not a mechanical problem in the rotating elements.

Avoid meters that only give peak or average vibration levels since most of the useful information comes from determining which frequencies are giving the highest velocity readings.

For this discussion we will assume a 3600 rpm motor driving a direct drive screw compressor with a 4 lobe male rotor. For each specific compressor size refer to Figure A to determine the characteristic frequencies of that compressor.

1. Peak levels at 60 Hz indicate imbalance on the male rotor, motor, or coupling. It can also indicate coupling misalignment or improper coupling installation.

COMPRESSOR LOBE PASSING FREQUENCIES

@ 60 HZ ELECTRIC DRIVE

FREQUENCY

MODEL	**LOBE PASSING (HZ)**	**FIRST HARMONIC (HZ)**	**GEAR TOOTH PASSING (HZ)**
RXB12	319	638	1980
RXB15	393	786	3300
RXB19	486	972	3600
RXB/RXF-24	319	638	1980
RXB/RXF-30	393	786	3300
RXB/RXF-39	486	972	3600
RXB/RXF-50	639	1277	3960
RWBII 60-676	240	480	N/A
RDB67	120	240	N/A
RDB 100-546	240	480	N/A

FIG. A

2. The keys on the compressor and motor half of the coupling should be 180 degrees apart (Figure B). The key supplied with the compressor half of the coupling should be a rectangular key, not curved to fit the milled keyway slot. A rectangular half height key is used in the balancing process, thus the rotor is dynamically balanced to offset the space at the back of the key (Figure C).

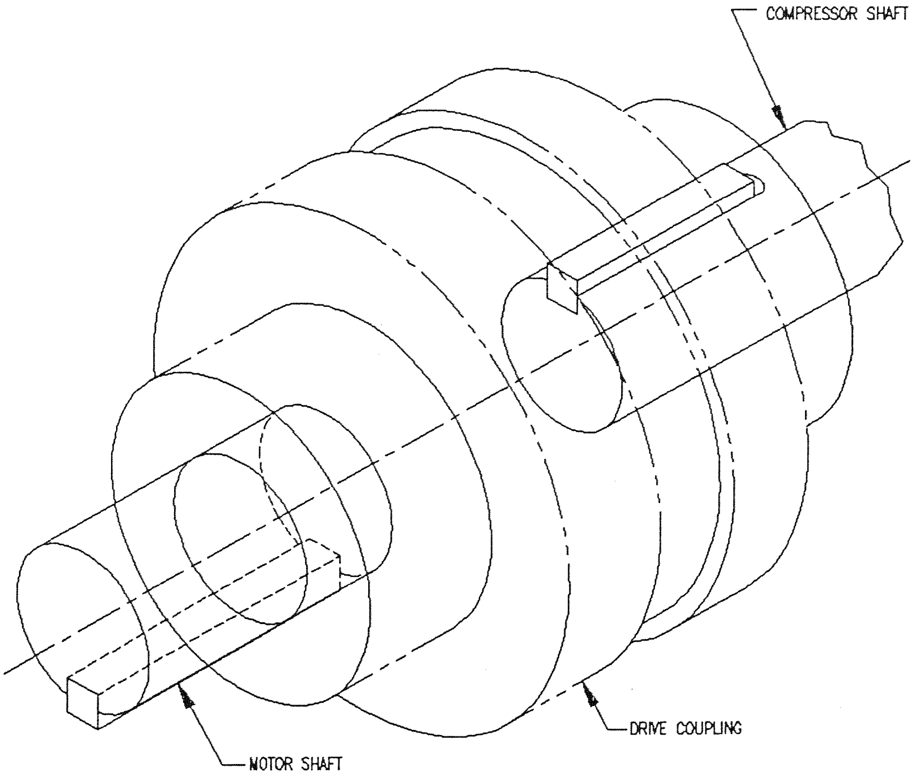
WITH COUPLING INSTALLED MOTOR KEY AND COMPRESSOR KEY SHOULD BE 180 DEGREES APART

Fig. B

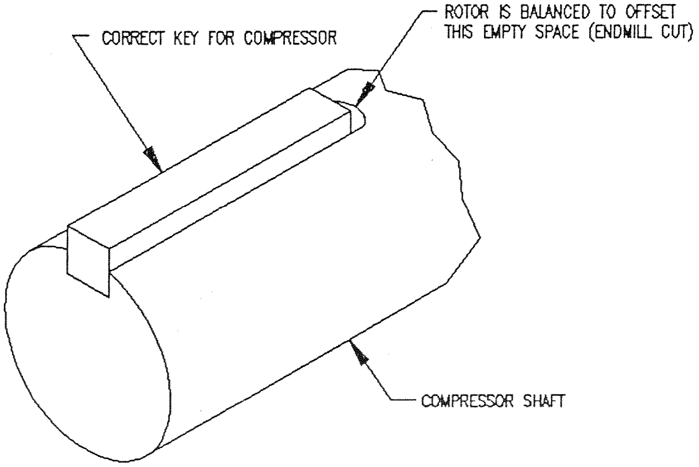

Fig. C

The coupling half is normally balanced to accept a half key filling the slot for its full length. If there is a significant difference between the coupling keyway length and the key length in the rotor, the key should be cut shorter to roughly be halfway between the two lengths. A closer approximation can be made by the following equation: Key Length = $[(x + y)/2] \times .95$ (Figure D). This will give a reasonable approximate balance. The best balance is obtained by balancing the coupling half with a half key the same length as the compressor key. While this may seem like a small point, the amount of imbalance that can be introduced to the rotor by improper keys can easily be two or three times the balance tolerance on the rotor.

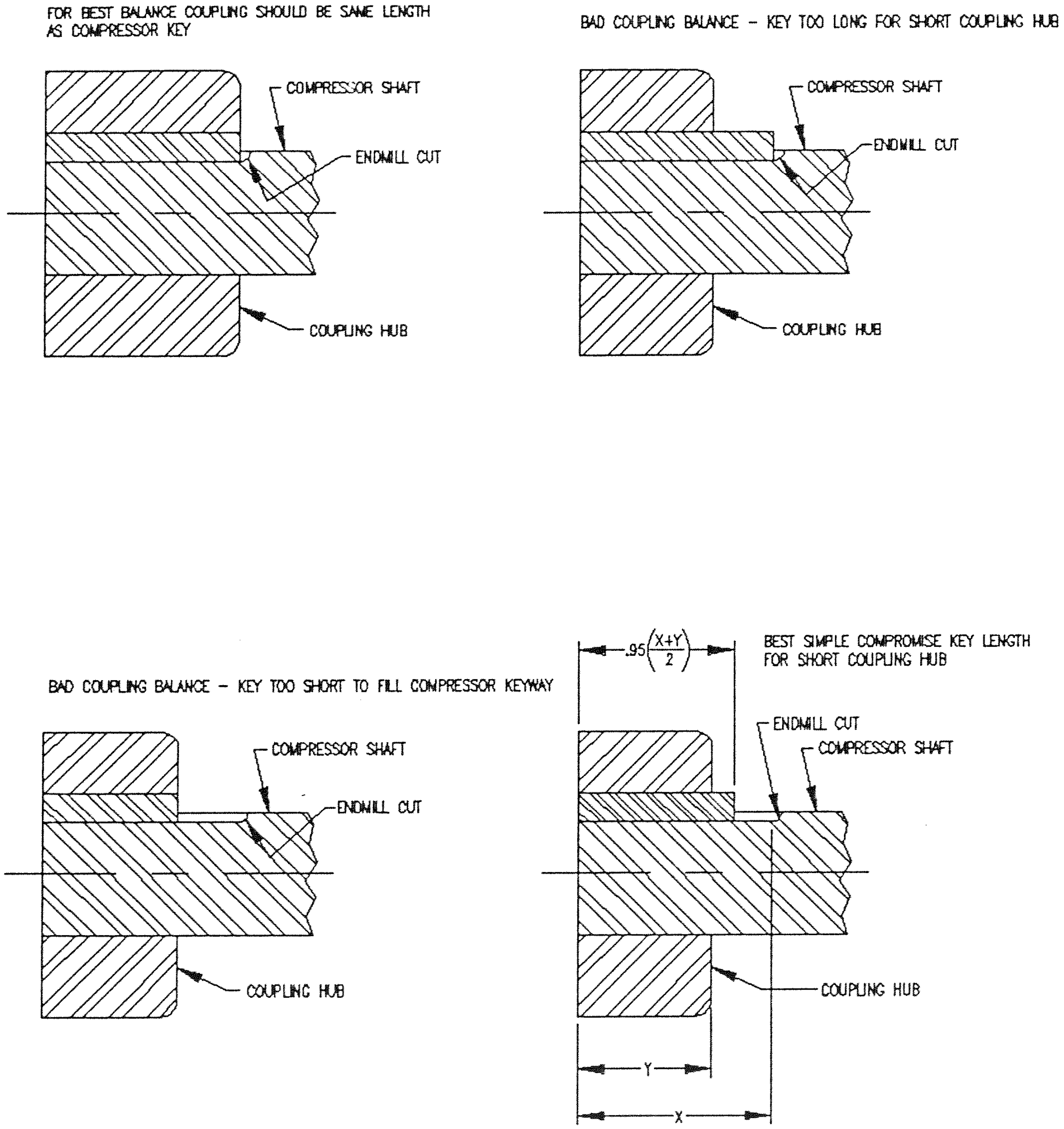

Fig. D

3. With the equipment off and electrically disconnected use a magnetic base dial indicator to check for runout of the coupling halves and both shafts while slowly turning by hand. More than a few thousandths of an inch runout is probably indicative of a bad component.

4. Check the motor and compressor to determine which has the higher amplitude of vibration. A higher reading on either component could indicate a bent or out of balance shaft.

5. Check coupling alignment since improper alignment is a very common cause of high vibration readings. Misalignment should show up at 1 x rpm (60 Hz), sometimes at 2 x rpm, and sometimes as fairly high axial vibration readings at 1 x or 2 x rpm. With spacer type couplings some signals may show up at the number of bolts through the shim pack, times rpm.

6. A high 2 x rpm (120 Hz) signal may indicate a soft foot on the compressor or motor. This could be visualized as the part rocking back and forth around some axis as the driveline rotates. To check for soft feet, loosen one mounting bolt at a time and see if the compressor foot rises more than a few thousandths of an inch. This can be checked with a magnetic base dial indicator mounted on the base with the indicator on the foot. If any foot moves, reshim and realign the coupling.

7. Generally all 60 Hz and 120 Hz amplitudes should be below 0.1 in/sec.

8. Stresses imposed on the compressor suction by piping can cause high vibrations, most likely at 60 Hz. This can drive the compressor out of alignment as pipes heat and cool. Excessive piping stress can distort the compressor casings and show up at 240 Hz if the casings are distorted enough to cause rotor contact. If piping stress is suspected, unbolt the suction line. If it moves when unbolted the pipe supports must be rearranged to eliminate flange loading on the compressor.

9. RXB and RXF compressors incorporate internal geared speed increasers. Some signal level will generally be detected at the gear tooth passing frequency. The gear tooth passing frequencies for each model are given in the attached Figure A. If these frequencies become excessive it could indicate a bearing failing on the gear jack shaft, misalignment of the housing, or excessive coupling misalignment. Lack of lubrication of the gear could also be a possible source of noise.

Lobe Passing Frequency Vibrations

Lobe passing frequency or its first harmonic will normally produce the highest vibration amplitude in a screw compressor. This will be at 240 Hz or 480 Hz for the sample given. Some level of vibration at this frequency is normal and should be expected. Each time the trapped gas pocket in the compressor opens to the discharge port a pulsation is produced. If the volume ratio is not calibrated correctly this frequency will produce higher amplitudes. Over-compression may produce very strong 240 Hz pulsations, under-compression may produce very strong 480 Hz pulsations. If the amplitude of the 240 Hz signal exceeds approximately 0.4 in/sec., or noise levels are unacceptable, further investigation may be required.

High vibration amplitudes at lobe passing frequency are the most common problem encountered in new installations. Vibration at this frequency is generally caused by the gas pulsation from the discharge ports, but these levels may become excessive for a variety of different reasons. To understand this we must first discuss resonance.

Resonance, natural frequency, or critical speed all refer to approximately the same phenomenon. Everything is a bell. If struck or excited by an external input every structure will tend to vibrate at some fixed frequency. A screw compressor package and its mounting base and piping comprise a complex network of many different structures, all of which have a certain resonant frequency. If any of these resonant frequencies happen to be at 240 Hz or 480 Hz excessive vibration can occur as the compressor discharge pulsations will excite the resonant structure at these frequencies.

Sometimes we will find an oil separator with high vibration levels at 240 Hz after hundreds of units of the same design do not display this problem. This can be caused by several situations.

1. Discharge pipe gas resonance can cause excessive vibration. The pulsations in the discharge line can produce a multiplying effect if the discharge pipe length happens to fall on an even half wavelength at the speed of sound at the temperature and pressure of the gas in the separator. This problem will generally be evident only at one slide valve position, and small changes in temperature should change the measured amplitude.

2. Separator gas volume resonance can also cause excessive vibration. On some applications the gas pulsations seem to be magnified in the separator volume. Slight changes in the oil level in the separator will generally reduce this problem. In some extreme cases the separator length has been changed so it does not fall exactly on a multiple of half wave lengths.

3. Structural resonance of the oil separator and its supports can also create high vibration amplitudes. Sometimes the separator is found to be ringing at 240 Hz. This can be due to the separator design but it is greatly influenced by the quality of the mounting base and foundation. Proper grouting of the unit feet in order to tie the separator to the mass of the floor will generally reduce or shift a structural resonance problem on a separator. If proper foundations can't be provided or don't reduce the vibration to acceptable levels it may be necessary to stiffen the shell of the separator by welding rings or plates to the shell.

   If structural resonance of a steel supporting structure is suspected, try using a hydraulic jack or angle iron brace to stiffen the vibrating part. If the vibration improves or gets worse it indicates the part is near resonance. Sometimes slight stiffening may drive the resonant frequency of the structure closer to the driving frequency of the compressor and actually increase vibration. This verifies that resonance is the problem and permanent stiffening or possibly a reduction in stiffness should be the solution.

4. If the highest vibration levels on the package are on piping it is likely that the piping is at or near a resonant condition. Piping can usually be braced to shift its natural frequency away from the driving frequencies.

5. Sometimes high 240 Hz pulsations are not resonance. If temperature or slide valve position does not significantly affect the vibration it could be that the discharge pulsations are merely causing a forced vibration of the component. This is particularly true on high discharge pressure applications, or very dense refrigerants with high power per unit displacement. In these cases there is much more energy in the discharge pulsations, so they tend to excite other components to higher amplitudes.

Sometimes an orifice plate or a muffler can be fitted on the discharge pipe to reduce this problem. The RXB-50 is equipped with a muffler in all cases to absorb the discharge gas pulsations and reduce noise. This is possible on the RXB-50 because the lobe passing frequency is quite high (640 Hz) and can easily be reduced in an absorptive muffler. Absorptive mufflers are generally ineffective at frequencies below 500 Hz. Reflective mufflers must be used below 500 Hz but these must be designed to suit one operating condition (one wavelength at a particular speed of sound) and thus they are not suitable for wide application on refrigeration screw compressors.

Is Anything Wrong ?

Is the problem bad enough to require action? This is one of the more difficult questions to answer. In general, there are no specifications on allowable vibration on oil separators or piping. The primary concerns are to avoid breaking pipe nipples and tubing or other components attached to vibrating components, or to avoid unacceptable noise in occupied areas. If the measured amplitude of the vibrating part is below 0.4 in/sec. the chances of breakage are rather remote. Even if amplitudes are above this level basic fatigue theory teaches that if any component goes through 1×10^7 cycles and does not fail, it will not fail in fatigue if the amplitude does not increase. At 240 Hz this will occur in less than 12 hours. In other words, if a vibrating part has run without failure for more than 12 hours at a vibrational frequency of 240 Hz it is probably not going to break due to this vibration.

Many screw compressor units operate with vibration levels below 0.1 in/sec. However, when another unit shows higher levels it does not automatically mean something is wrong requiring immediate action. Screw compressors are very flexible in their ability to compress many gases at widely varying conditions. The peculiarities of each application and the components attached to the package can produce differences in noise and vibration level that are impossible to predict before the installation is running at design operating condition. The nature of custom field erected refrigeration systems give rise to differences in vibration and noise from one installation to another that must be expected.

Torsional Resonance

Torsional resonance can be understood by referring to a small drive shaft that has a flywheel on each end. If one flywheel is rotated with respect to the other, then released, the flywheel will oscillate at the natural frequency of the torsional spring/mass system, often called the torsional first critical. The first critical frequency of the screw compressor rotor is quite high, well above normal drive speeds. Electric motor driven screw compressors will also not normally approach critical frequencies. However, all systems using engine or turbine drives, with or without geared speed increasers, are at risk for torsional resonance. A torsional vibration analysis should always be performed before coupling selection on such a system so that the torsional natural frequencies of the system can be tuned away from running speed and its harmonics.

Field determination of a torsional resonance problem is supposedly possible with a rotating shaker device and accelerometer used on a stationary system. The writer has never seen this done so this subject will be left to more experienced writers.

Machinery Condition Monitoring

Machinery conditioning monitoring is the process of inspecting a successfully installed piece of equipment on a routine basis to look for change, and hopefully to detect the early stages of a problem before it leads to a complete machinery failure. Equipment inspection can either be on a continuous or a periodic basis.

If vibration data is taken on a periodic basis there are certain procedures required to produce good data.

1. Establish a baseline vibration spectrum once the equipment is running at design condition and all alignments and calibration have been performed. Readings should be taken radially over each bearing position on the compressor and the motor. Axial readings may also be desirable. It is of utmost importance to take readings with the compressor operating at a stable and repeatable condition. 100% slide valve position (full load) and optimum volume ratio is recommended. All operating pressures and full information from the compressor control panel should be recorded so that comparisons can be made as closely as possible to the same operating conditions at a future date. Clearly label the position of the accelerometer on the compressor housings with a marker. Also, clearly label the recorded data so the accelerometer position will be known later; for example, drive end male radial, drive end female axial, motor bearing outboard radial, etc. Make sure that the mounting locations on the compressor are clean and flat,

giving no flexibility to the accelerometer.

2. If data is recorded later as part of a routine maintenance inspection try to control the pressures to the same levels. If this can't be done record the differences in conditions so this can be considered if variations in recorded readings are found. If possible take readings at the same locations on the housings, same slide valve positions, and same volume ratio setting.

3. Compare the readings taken against the baseline and look for changes. If significant differences are found try to determine what component or condition might produce an increase at the noted frequency. Refer back to the section above to track frequencies to possible sources.

If significant changes are noticed, gather as much other information as possible to help isolate the problem. Did anyone notice a change? Were any components or new piping added or changed recently? Was the coupling changed and perhaps not realigned properly or a new motor installed? Have any of the compressor feet or unit feet anchor bolts come loose?

Bearing Fault Detection

One of the primary purpose of machinery condition monitoring is to predict failure of rolling element bearings in the early stages of damage. Frick Publication, CES 3805, pages 1 - 5 give the characteristic frequencies of each element in every bearing in each different size Frick compressor. When taking readings on a compressor, be aware of the frequencies that might indicate a particular type of bearing defect.

This data is offered because it is frequently requested by vibration "experts" who are recording spectrums on screw compressors. Supposedly a bearing defect of any element should show up on a spectrum at its characteristic frequency, indicating the bearing should be changed. There are many reports of this method being used effectively on some types of machinery. The success of this approach has not always been good on screw compressors.

Apparently the early stages of bearing fatigue generate very low signal levels of perhaps 0.01 - 0.05 in/sec. at the characteristic frequency of the bearing element with the flaw. Viewing these low levels on a screw compressor spectrum which will almost always have 0.1 in/sec. at lobe passing frequency along with normal structural and gas resonances and harmonics can be difficult. There is always a fair amount of activity on screw compressor vibration spectrums, with different frequencies showing up at different load and vi positions. Recognizing a fairly low signal level from a bearing is not easy, if it shows up at all.

We have conducted some tests where we purposely failed one element of one bearing in a lab compressor and had some vibration "experts" try to find it with vibration analyzers. In some cases the bearing characteristic frequency did not appear on the spectrum even though a trained ear could easily tell not only that a bearing was bad but what position in the compressor it was located! So, throw away those analyzers and buy your best mechanic a long screwdriver.

What we found to be the best method of identifying the early stages of bearing failure was not an increase in vibration at the characteristic frequency but at much higher frequencies.

We have found that signal levels from 2500 Hz up to even 40,000 Hz will increase significantly due to the friction involved in fatigue of bearing elements. The spectrum at very high frequencies is generally a fairly level field. There are not as many structural resonances at these high frequencies to generate any significant signal level until something like a bearing failure occurs. Acceleration may show bearing fatigue somewhat better than velocity but an absolute alarm level is not clear. Looking for significant change compared to a baseline is the best method. The impacts and friction of deteriorating bearing elements create a wide band vibration in the higher frequency ranges that is more reliable than trying to find a change at the characteristic frequencies, which are in a fairly active frequency range.

4. If a change is noted in signal level compared to the baseline, do not immediately assume the machine is about to fail. Check the spectrum every day for a few days to see if the signal level continues to rise, or if it may go down. If it does not seem to be rising over time, let the time interval increase to once per week. Signal levels and noise will generally increase dramatically when a bearing is fatiguing, with little doubt that action is needed.

Continuous Monitoring

The most desirable type of machinery condition monitoring is with permanently mounted accelerometers that tie into a dedicated monitoring system. An onboard system can normally be calibrated for the baseline machine condition and be set to look for an established amount of increase in amplitude in order to sound an alarm. Probably the most effective system for a screw compressor should look in the higher frequency ranges where bearing failures are easier to detect. One side benefit of continuous monitoring is that it will often save a compressor from catastrophic failure, for example from liquid slugging or stray check valve buttons that find their way to the rotors. Instant shutdown under catastrophic conditions can often prevent extensive compressor damage, and can only be done by a mounted, dedicated system.

Noise

Noise is just vibration of surface areas that are pushing air to create sound waves. In cases where vibration is deemed to be within normal levels and component failure is not occurring it is still common to get noise complaints in some installations.

Some action can be taken before the equipment is ordered to avoid unrealistic expectations of installed noise level.

Expected noise levels on Frick screw compressors are given in Frick publication, CES 8101-8109. These are based on measurements taken on units installed and running at one set of conditions. While they are called "typical" there is really no such thing as typical noise levels for this type of equipment. Just as operating conditions, percent load, vi, gas density, and motor power effect vibration, all of these factors, plus several more, effect noise.

Reverberant Noise Correction

A reverberant noise correction should be added to the typical noise levels from CES 8101-8109 to reflect the reality of customer machinery rooms. When a wall is nearby, all the sound at screw compressor frequencies (200-1000 Hz), is reflected back into the space, doubling the noise (3 db increase). From an airborne sound standpoint it is just as if two units were present. As the number of surfaces increase the correction increases. The dimension used for "nearby" is 2 wavelengths of sound at pumping frequency (10 ft.). If no surfaces are within 10 ft. but are within 20 ft. then 1/2 the given corrections are added for these surfaces.

By giving each customer the ability to make this reverberant correction the effects of walls and small equipment rooms will be apparent.

For large surfaces within 10 feet of the screw compressor add the following dB corrections to the 250, 500, and 1000 Hz octave bands and the dBA level of the spectrum.

For 1 surface add 3 dB
For 2 surfaces add 5 dB
For 3 surfaces add 6 dB
For 4 surfaces add 7 dB
For 5 surfaces add 8 dB

Note that ARI-575 is the log average of 16 A-scale levels measured at locations around the unit. The minimum position will be 3-5 dB lower than the average, and the maximum will be 3-5 dB above the average level.

The tests conducted to produce the "typical" sound levels utilized fairly quiet motors. It is possible to have a motor noise level in excess of the compressor noise if quiet motors are not specified.

If the expected noise levels are above the customers desired levels some provisions should be made for sound absorbing material on walls, sound enclosures, or curtains around the compressors, or possibly for noise dampening blankets on the compressor or oil separators.

Probably the most effective method of reducing noise is to run screw compressors at lower speeds. A reduction in speed from 3600 rpm to 1800 rpm should normally reduce noise by 6 dba.

If the problem is only recognized after the equipment is installed, all of the above options may still be possible but cost will certainly be more of an issue.

In extreme cases, an expert in noise reduction is probably a good investment. By systematic measurement of vibration on all major components it is possible to identify the primary sources of noise and recommend specific treatment of the biggest offenders in order to reduce their contribution to the overall noise levels. For example, isolation of components from the vibration source will reduce the areas acting like speakers. Isolation of coupling guards, electrical panels, or sometimes separators may reduce overall levels. Rockwool or other absorptive materials applied to the oil separator may reduce its transmission to the air.

Conclusion

In general, screw compressor noise and vibration problems are some of the more difficult problems to solve because they can be influenced by so many different factors. You should have noticed throughout this paper an abundance of words like may, might, could, probably etc. This is not an accident or attempt to hide something. There are no hard and fast rules in vibration and noise problems. There are no clear cut levels at which we can clearly say a particular problem exists. If someone gives you simple absolute rules they probably haven't tried using them in practice. We hope some of these comments and suggestions will help spread a broader understanding of what can cause compressor noise or vibration. As with many problems in life the more accurately a problem can be defined the easier it is for anyone to find a solution.