Part One

PREVENTING Refrigeration Line Vibration Problems

Structural and acoustical resonances as well as forced vibration can cause problems in discharge lines, which can impact compressor life

BY WES TAYLOR, CMS

Refrigerant leaks always have been a major concern in any air-conditioning or refrigeration system. Costs associated with refrigerant replacement and the reliability of the unit gives system designers and manufacturers the incentive to keep refrigerant losses to a minimum.

This article will assist technicians and equipment manufacturers in the design of refrigeration and air-conditioning systems to avoid refrigerant line vibration problems. Vibration problems can result in broken refrigerant lines.

Causes of vibration in discharge lines can be separated primarily into the following categories: one, structural resonances; two, forced vibration; and three, acoustical resonances. Of the causes of vibration, structural resonances are the most common, followed by forced vibration and acoustical resonances. Multiple combinations of the three also can cause vibration.

Structural resonances

Structural resonances occur when the natural frequency of the discharge piping matches the frequency of the discharge gas pulsations (exciting frequency). When the exciting frequency matches the frequency of the piping, the vibration of the piping becomes greatly amplified.

Imagine a tuning fork. It has a specific frequency at which it resonates and that is where the most vibration occurs. Think of a piping system as a very large tuning fork. Its design wouldn't have a structural resonance at or near the exciting frequency of the gas pulsations.

The resonant frequency is a function of the stiffness and mass of the line. A discharge line should be designed as stiff and light as possible to reduce the chance of vibration problems.

However, in systems where start-and-stop kicks or other system motion occurs, flexibility must be designed into the piping system to absorb the motion. A stiff and light discharge line will have a higher frequency to resonate than one that is less stiff and/or has more mass.

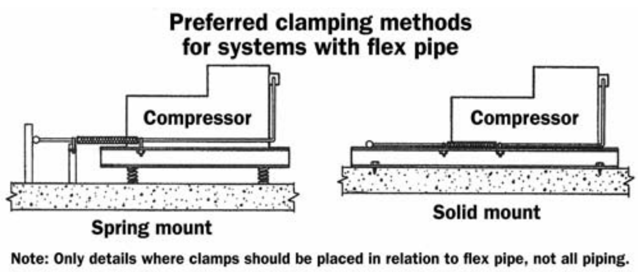
Troublesome discharge gas pulsations occur at relatively low frequencies; therefore, systems with higher natural frequencies will have less chance of being affected by structural resonances. Structural resonances are very difficult to predict. Extensively testing a piping design in a laboratory environment using the intended system refrigerant is the best method for avoiding structural resonances.

When a design is found to be free of structural resonances the exact design should be used consistently. Any changes to a structural resonance-free design, such as moving a fitting, would require retesting the design.

Forced vibration

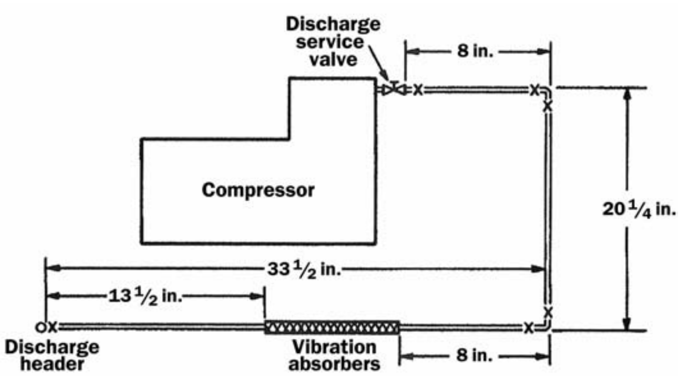
Forced vibration is the vibration caused by movement of the compressor (compressor-driven vibration) and/or discharge gas pulsations (pulsation-

FIGURE 1



Here are the preferred clamping methods for spring mounting when using flex pipe.

FIGURE 2



Proper piping and compressor design can prevent vibration.

driven vibration) emitting from the compressor. The term forced vibration excludes vibration due to any piping system resonances. The pulsations and effects will vary with the physical properties of different refrigerants.

Compressor-driven vibration is not a typical problem when the compressor is solid mounted. But when the compressor is spring mounted the chances of vibration problems are greater. Spring or rubber mounting kits are typically used in environments where vibration transmitted from the compressor to the floor may cause excessive noise or damage to a sensitive area.

When spring or rubber mounting kits are used the piping system (which is where most vibration problems occur) must be designed to absorb start-and-stop kicks, and handle the additional motion of the compressor during operation. That is usually accomplished by using flexible piping (vibration absorbers) and spring piping hangers.

Although these items may reduce the transmission of movement of the compressor to the system, they may greatly enhance the effects of piping system vibration. The chances of pulsation-driven vibration and structural resonances increase with the introduction of springs into the system. Spring or rubber mounting of compressors is sometimes necessary depending on the environment and/or rack or unit design.

Spring mounting generally makes piping geometry more complex and increases the possibility of vibration problems. As a rule, if there is no reason for spring mounting compressors or racks, do not use springs. Note, however, that rooftop packages may require spring mounting.

On solid-mount applications utilizing flexible piping (vibration absorbers), clamp the inlet and outlet of the vibration absorber. Allowing one or both ends of the vibration absorber to flex can increase the chance and/or magnitude of vibration problems.

Most manufacturers don't recommend the use of vibration absorbers in solid-mounted compressor systems, unless they are securely clamped at both ends. Figure 1 shows preferred clamping methods.

On spring-mounted applications, vibration absorbers should be used only in accordance with the manufacturer's requirements and recommendations. Typically, vibration absorbers should be parallel to the crankshaft with the inlet clamped to the compressor or spring-mounted base and the outlet clamped to the solid mounted frame.

When piping spring-mounted compressors, try to arrange the compressor and piping to the straightest possible piping geometry, using as few bends as possible. The previous statement holds true on solid-mount systems as well, but spring-mounted systems usually become more of a problem because the discharge line has to be parallel to the crankshaft. Careful planning can eliminate excessive piping and bends.

Common cause of vibration

Gas pulsation-driven vibration is the most common cause of forced vibration. Pulsation-driven vibration does not mean that the compressor is emitting such high pulsations that it forces the line to vibrate regardless of the piping geometry. All reciprocating compressors emit discharge gas pulsations (a reciprocating compressor generates a constant stream of pulsating flow).

When discharge gas pulsations react with the piping system geometry in such a way as to set up an oscillating force, discharge pipe vibration may occur. Condensations and refractions are clusters of various lengths of sound waves that catch up with each other within the piping system. This causes the amplitude of the sound level to increase, causing loud disturbing pulsating harmonic sounds.

An example of this is when the discharge line comes off the compressor service valve and enters one, two, three or more elbows. Picture the pulsating discharge gas flowing from the compressor through the first straight section of discharge pipe. The discharge gas then hits the first elbow and bounces into the next section of straight pipe.

An oscillation in the gas already has started and each elbow may increase the oscillation, creating a significant amount of line vibration. Designing the discharge piping as straight as possible will reduce the chances of pulsation-driven vibration occurring.

Another cause of forced vibration is the Bourdon Tube Effect. When discharge piping forms a U shape, the high-pressure discharge gas tries to straighten the piping, resulting in vibration. Gas pulsations have little if any affect on discharge lines composed of a single straight pipe. Figure 2 shows a diagram of the compressor and piping.

Acoustical resonances

Acoustic resonances result from the specific discharge gas properties and the piping system geometry (not the structural dynamics). The effect of acoustical resonances is to amplify the gas pulsations at specific locations in such a magnitude as to cause significant vibration.

There are many types of acoustic resonances and most are difficult to predict. The most common type results from longitudinal standing wave patterns.

The following paragraphs detail a sample critical-length calculation for a standing wave resonance. Although critical-length calculations are straightforward, if the terminology is not familiar, it can become confusing. Becoming familiar with the following definitions will make the critical length discussion more understandable.

➤ **Standing wave:** Specific type of acoustical resonance.

➤ **Exciting frequency:** The frequency of the discharge gas pulsations in hertz, denoted by the symbol f. Some dominant frequencies for reciprocating compressors are as follows:

- 2 cylinder: 58 hertz. (X)
- 4 cylinder: 117 hertz. (2X)
- 6 cylinder: 58 hertz and 117 hertz. (X+2X)

"X" is a measured length of the trial system.

➤ **Element:** Segment of discharge pipe between two terminations. A termination is defined as a compressor, muffler, elbow or header.

➤ **C:** Speed of sound of the discharge gas, in feet per second.

➤ **L:** Wavelength for an exciting frequency.

➤ **L/2:** One-half the wave length for an exciting frequency.

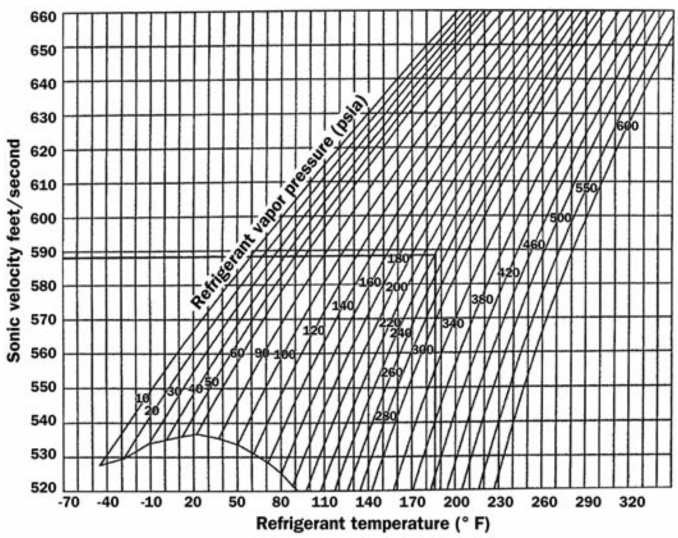
➤ **Critical length:** An integer multiple (1, 2, 3 ... times) of one-half the wavelength for an exciting frequency (L/2).

Simple "standing wave" resonances occur when an "element" matches a "critical length." Critical lengths can be easily calculated using the following formula: L/2 = Cf2f

Critical-length calculation

For this example, we will assume a four-cylinder compressor is in use. Therefore, the significant exciting frequencies will occur at two and four times rotating speed (58 hertz and 117 hertz). We also will assume the refrigerant to be R-22 at 110° F and 185° F discharge gas temperature. With a sonic velocity of 588 feet per second, the

FIGURE 3



This chart helps determine sonic velocity in a critical-length calculation.

sonic velocity was determined using the information from Figure 3:

$$\frac{L}{2} = \frac{588}{(2)\,58} = 5.06 \text{ feet} = 60.82 \text{ inches.}$$

Allowing a ±10 percent margin for variation in the system conditions, the critical lengths to avoid would be:

- At 58 hertz, 55 to 67 inches.
- At 117 hertz, 27 to 34 inches (four times operating speed will be exactly half of two times operating speed).

Evaluating Figure 2, there are no lengths between discontinuities that fall into the critical length ranges calculated. An example of a potential acoustic resonance would be if the first elbow is located in the 27-inch to 34-inch range. If a potential acoustic resonance occurs, the fitting would have to be moved out of the critical length range to avoid it.

One-half of the critical length (L/4) also should be avoided between the service valve and the muffler. Also, in general, all field piping should be of prime value, not divisible equally by 2, 4, 6 or more.

In part two on this topic in next month's issue of *RSES Journal* I will address design and retrofit considerations for refrigerant piping problems.

Editor's note: This two-part report is part of an ongoing series on compressors by Wes Taylor, CMS. The first article — "A Baffling Issue" — appeared in the December 2005 issue of *RSES Journal.*◆

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