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Channel Resonance in Reciprocating Compressor Cylinder Pressure Measurements

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Abstract

Incorrect installation of pressure transducers to monitor cylinder conditions on reciprocating compressors can give rise to a resonance phenomenon. Known as Helmholtz Resonance, this condition can lead to signal amplitudes that overwhelm the actual information in the pressure transducer signal, making diagnosis difficult or impossible and introducing unnecessary uncertainty into the thermodynamic calculations. Field-gathered data shows that these problems can be avoided when the installation is designed such that the calculated Helmholtz Resonance at suction conditions occurs at a frequency at least 100X higher than the running speed of the compressor. This article describes how Helmholtz Resonance occurs and shows how it can be calculated from actual installation geometries. Equipped with this information, the designer can help ensure an installation that will not suffer from data quality issues due to Helmholtz Resonance phenomenon.

Background

Cylinder pressure data acquisition requires the installation of a pressure transducer at the compressor cylinder. An ideal installation would have the pressure transducer installed such that the diaphragm of the transducer was exactly flush with the bore of the cylinder as shown in *Figure 1*.

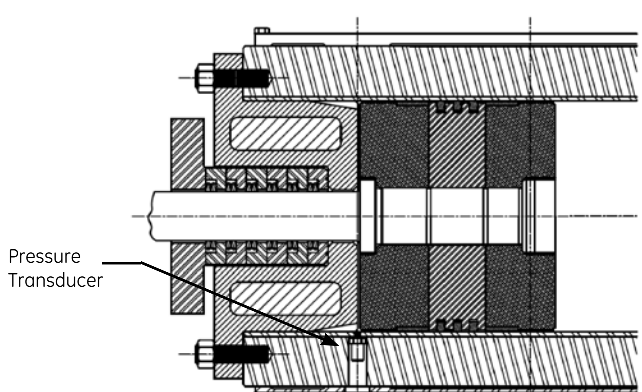

Figure 1. Cross-section of a cylinder with ideal pressure transducer installation.

Unfortunately, this is not possible on real compressor cylinders. As can be seen from *Figure 1*, the transducer would have to thread into the liner, the access hole would be quite large, and there would be no way to replace or isolate the transducer while the machine was operating. For these reasons, the transducer is usually installed at the end of a port that passes into the cylinder. Inclusion of isolation valves, adapters, etc., further complicates the situation by adding restrictions as well as changes in diameter and possibly direction. Under some operating conditions, the characteristics of the cylinder pressure port, isolation valving, and transducer can set up a standing wave phenomena known as *channel resonance*.

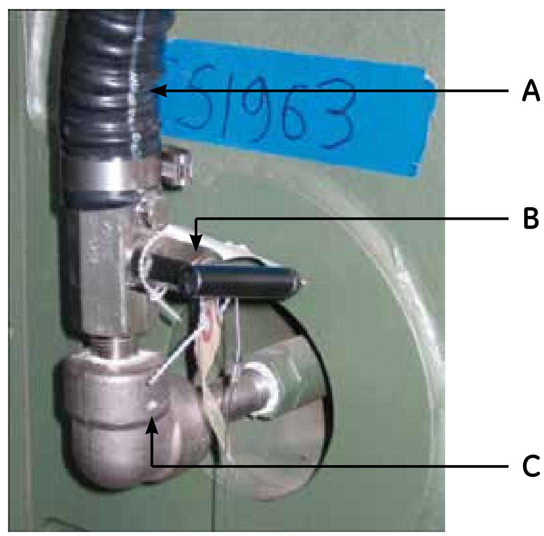

Figure 2. Cylinder pressure transducer installation showing protective flexible conduit (A) covering the transducer and its field wiring, isolation valve (B), and 90-degree elbow (C). The transducer is not visible, but is located directly above the valve.

Data was captured from a pressure transducer installation experiencing channel resonance (*Figure 2*) and presented as a pressure versus crank angle diagram (*Figure 3*). The particular cylinder in question is equipped with a Hoerbiger® HydroCOM™ stepless unloader. With this system, five distinct processes occur during a single revolution of the crankshaft. Referencing the labels and arrows in *Figure 3*, *Table 1* summarizes the activities that occur over one complete crank revolution.

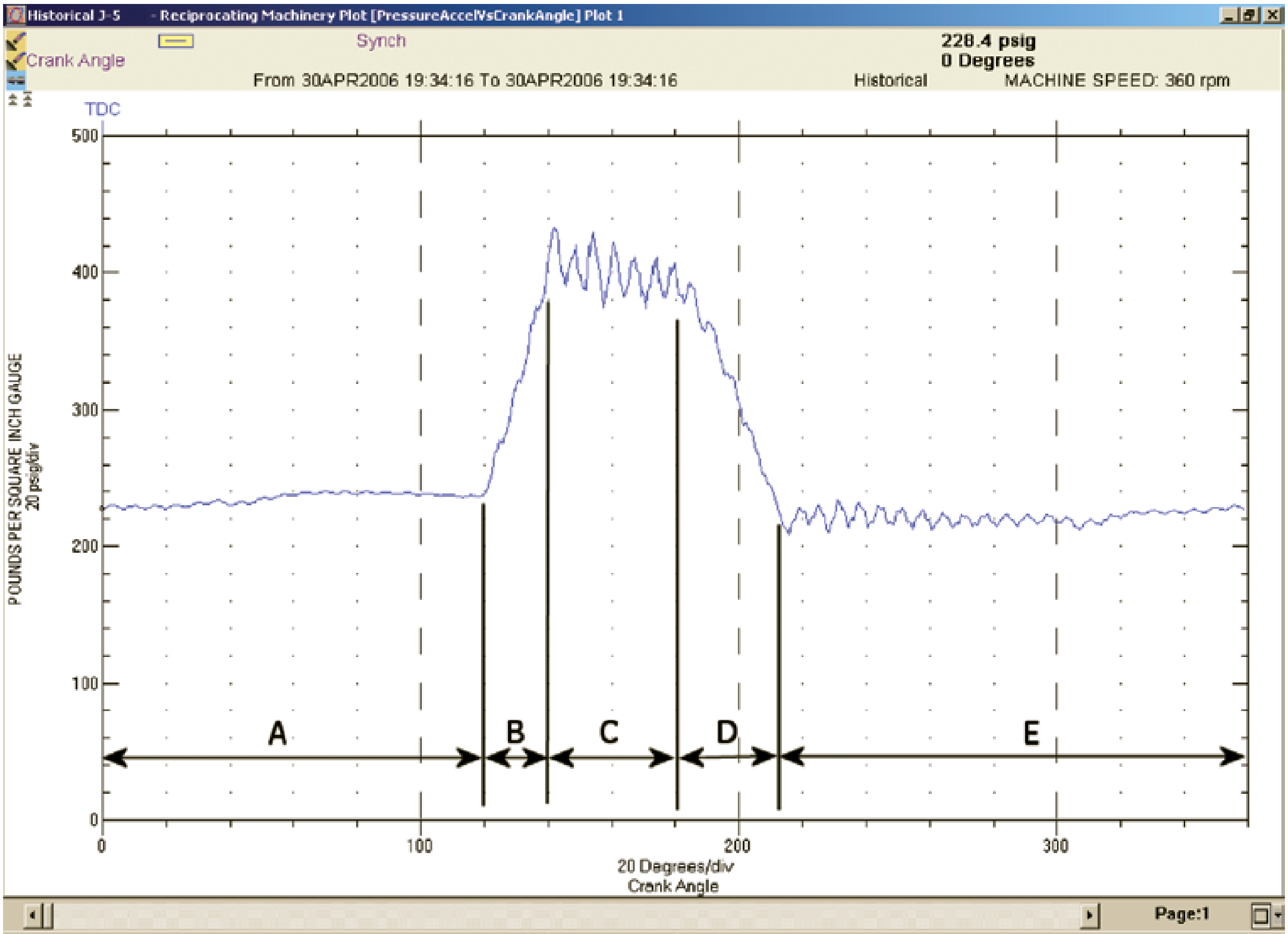

Figure 3. Crank-end cylinder pressure vs. crank angle. The waveform exhibits channel resonance resulting from pressure transducer installation downstream of an elbow.

Pressure pulsations can normally be observed when the valves are open. *Figure 7* shows a typical cylinder pressure curve exhibiting pressure pulsation during the discharge and intake processes. Note that the expansion and compression curves are smooth.

However, during the compression and expansion process, all valves are closed and no significant pressure pulsations exist. Yet *Figure 3* shows a standing wave on the expansion and compression process as well as on the intake and discharge process. The standing wave has roughly the same period throughout the stroke. The standing wave makes diagnostic work very difficult and greatly increases the uncertainty of the thermodynamic calculations. For these reasons, the cylinder pressure port, isolation arrangement, and pressure transducer should be selected in such a way as to avoid conditions favorable to channel resonance.

Channel Resonance

In a practical pressure transducer installation there is always some volume, V, between the pressure transducer and the cylinder bore.

A proper selection of isolation valves and fittings will result in a long, straight column between the pressure transducer and the cylinder. An example of a good application would include a cylinder pressure port diameter of 0.19” and an isolation valve with a straight through

	Crank Positions	Activity
A	0 – 120°	**Suction:** The HydroCOM system holds the suction valve open and gas inside the crank end (CE) chamber exits through the suction valve.
B	120° – 140°	**Compression:** At approximately 120°, the HydroCOM system releases the suction valve. The pressure of the remaining gas trapped inside the cylinder begins to increase as volume decreases. This is the compression process during which all valves are closed.
C	140° – 180°	**Discharge:** At approximately 140°, the pressure inside the CE chamber rises above the pressure in the discharge manifold and the discharge valves open. For the remainder of this discharge process, gas exits the chamber.
D	180° – 215°	**Expansion:** At 180°, the piston reverses direction. Volume begins to increase and the discharge valve closes. For the remainder of this process, the remaining gas trapped between the piston and cylinder head expands. This is the expansion process, during which all valves are closed.
E	215° – 360°	**Intake:** At approximately 215°, the pressure inside the CE chamber falls below the suction manifold pressure and the suction valves open. For the remainder of this intake process, fresh gas is drawn into the cylinder.

Table 1. Activities occurring during each part of the compressor cylinder stroke of Figure 3.

bore of 0.19”, identical to the pressure port diameter. In addition, the installation would have no elbows or turns resulting in a long, straight column between the transducer and the cylinder bore. *Figure 4* shows the situation schematically.

For this arrangement, the gas moving past the cylinder pressure port creates an effect similar to that of blowing across the opening of a bottle. Vortexes shed as the fluid moves past the opening increase pressure at the hole entrance[1]. As the pressure increases across the opening of the cylinder pressure port, a “plug” of gas moves into the volume as shown in *Figure 5*. The stiffness of the gas in the volume under the pressure transducer reacts to the plug of gas and it is pushed out again. This condition sets up an acoustic resonance.

Several methods exist for estimating the acoustic resonance of this structure. The most accurate involve recursive calculation methods[2,3]. However, these methods can be quite time consuming. In order to present a method that can be used for estimating the acoustic resonance simply, the Helmholtz model will be used.

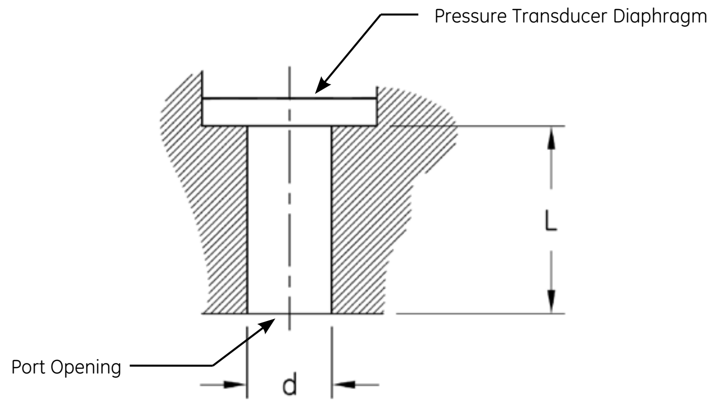

Figure 4. Schematic of straight bore arrangement.

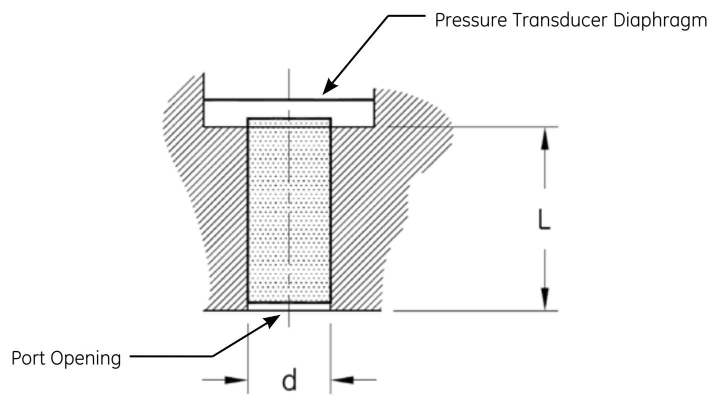

Figure 5. Increased pressure inside cylinder.

Application of the Helmholtz model requires a few simplifying assumptions. The wavelength at the resonant frequency has to be much greater than the opening diameter (d) and the port length (L) so that a lumped parameter model may be used[4]. In addition, if the tube volume is half the volume of the chamber, or less, the Helmholtz model should not be used[5]. Finally, the gas composition should remain constant. Changes in gas composition resulting from combustion, for example, would likely cause significant deviation from the resonant frequencies predicted by the Helmholtz model and the actual conditions at the pressure transducer.

In *Figure 5*, the oscillating mass has been drawn as a cylinder. In practice, the shape of the gas involved in the oscillation differs from a cylinder. Empirical end condition corrections, also known as a shape factor, have been developed to account for this. For most applications, the shape factor is generally between 0.80 and 0.85[1,5].

A variety of sources present the Helmholtz model[1,4,5]. One general form of the equation is provided below as:

$$f_H = \frac{c}{4\pi}\sqrt{\frac{\pi d^2}{V(L + (m\ d))}} \qquad [1]$$

where

f_H = *Frequency of the acoustic resonance*

c = *Speed of sound*

d = *Diameter of port*

V = *Volume under pressure transducer diaphragm*

L = *Port length*

m = *Shape modifier*

From equation [1], it can be seen that desirable characteristics of a cylinder port include a large diameter (f_H~d), a small volume under the pressure transducer diaphragm (f_H~$1/V$), and a short length (f_H~$1/L$). Ideally, the acoustic resonance f_H predicted by the Helmholtz model should be higher than the response of the pressure transducer.

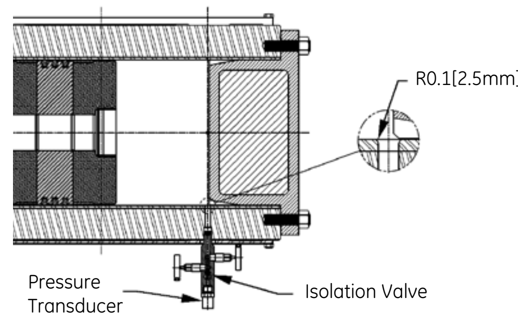

Figure 6. Channel entrance detail showing radiused entrance.

Practically, the closer f_H is to the running speed of the machine, the more likely channel resonance will show up on the pressure curves. Thus it is recommended that Installations be designed such that f_H is at least 100X higher than the machine's running speed.

In addition to ensuring that the resonance lies well above running speed, the channel entrance at the cylinder should be rounded off to reduce the forcing function caused by vortex shedding. Radius or chamfer this entrance generously, as shown in *Figure 6*.

Cylinder Pressure and Acceleration vs Crank Position

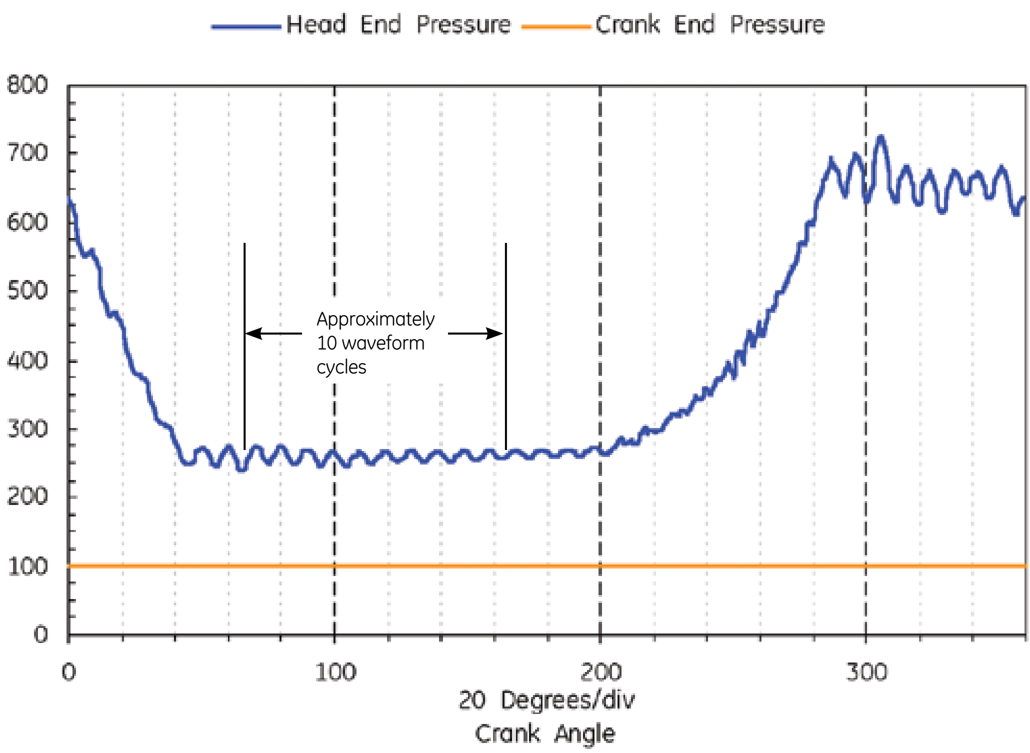

Figure 7. Pressure versus crank angle for an installation experiencing Helmholtz Resonance problems

Example 1 – Cylinder Configuration with Channel Resonance

Figure 7 shows a pressure versus crank angle diagram for an installation suffering from channel resonance. The data in *Figure 7* was collected on a cylinder with the following pressure port geometries:

Port length L = 12 inches

Port diameter d = 0.13 inches

Chamber volume V = 0.12 in^3

Equation [1] can be used to calculate the Helmholtz frequencies for this installation and these results can then be compared against the measured data of *Figure 6* to see how closely the expected and actual frequencies agree. Reference material for this situation recommends a shape factor between 0.80 and 0.85. A shape factor of 0.85 will be used in this example because it provides a slight more conservative estimate than 0.80 does.

The first step is to calculate the speed of sound at suction and discharge conditions. In this example, the working fluid is ethylene with process conditions as summarized in *Table 2*.

	Pressure *P* (PSIG)	Temperature *T* (R)	Compressibility *Z*	Isentropic Exponent *k*
Suction	266	553	0.89	1.212
Discharge	634	721	0.90	1.212

Table 2. Ethylene process conditions at suction and discharge for reciprocating compressor of Figure 7.

Assuming a reversible adiabatic process and an ideal gas, the speed of sound can be calculated from:

$$c = \sqrt{kRT} \qquad [2]$$

where

c = Speed of sound (ft/s)

k = Isentropic Exponent

R = Fluid Gas Constant {(ft-lbf)/(lbm-R)}

T = Temperature of the fluid (degrees Rankine)

Since the compressibility at both suction and discharge conditions is reasonably close to unity, the ideal gas approximation does not contribute significant error and will be used.

The fluid gas constant *R* for ethylene is:

$$55.08\frac{\text{lbf}}{\text{lbm}-{}^{\circ}\text{R}}$$

Using the data in Table 1 and equation [2], the speed of sound can be calculated at suction and discharge conditions:

$$c_S = \sqrt{(1.212)\ \frac{55.08\ \text{ft}-\text{lbf}}{\text{lbm}-{}^{\circ}\text{R}}\ (553°\text{R})\ \frac{\text{lbm}-32.2\ \text{ft}}{\text{s}^2}} = 1090.3\frac{\text{ft}}{\text{s}} = 13083.4\frac{\text{in}}{\text{s}}$$

$$c_D = \sqrt{(1.212)\ \frac{55.08\ \text{ft}-\text{lbf}}{\text{lbm}-{}^{\circ}\text{R}}\ (721°\text{R})\ \frac{\text{lbm}-32.2\ \text{ft}}{\text{s}^2}} = 1244.9\frac{\text{ft}}{\text{s}} = 14939.1\frac{\text{in}}{\text{s}}$$

$$f_{H,S} = \frac{c}{4\pi}\sqrt{\frac{\pi d^2}{V(L+(0.85\ d))}} = \frac{13083.4\ \text{in}}{\text{s}}\ \frac{1}{4\pi}\ \sqrt{\frac{\pi(0.125^2\ \text{in}^2)}{0.12\ \text{in}^3(12\ \text{in}+(0.106\ \text{in}))}} = 191.4\ \text{Hz}$$

$$f_{H,D} = \frac{c}{4\pi}\sqrt{\frac{\pi d^2}{V(L+(0.85\ d))}} = \frac{14939.1\ \text{in}}{\text{s}}\ \frac{1}{4\pi}\ \sqrt{\frac{\pi(0.125^2\ \text{in}^2)}{0.12\ \text{in}^3(12\ \text{in}+(0.106\ \text{in}))}} = 218\ 5\ \text{Hz}$$

Shape factor m=0.85

Wavelength Check

Earlier, we stated that application of the Helmholtz model of equation [1] requires the wavelength at the resonant frequency to be much greater than the dimensions of the pressure port. Specifically,

$$\lambda_H >> d \text{ and } \lambda_H >> L \qquad [3]$$

where

λ_H = *wavelength of the Helmholtz Resonance*

f_H = *frequency of Helmholtz Resonance.*

We can check these assumptions using the relationship

$$\lambda = \frac{c}{f} \qquad [4]$$

where

c = speed of sound (in/s)

f = frequency (Hz)

λ = wavelength (in)

Substituting c_D and $f_{H,D}$ into [4] we obtain

$$\lambda_{H,D} = \frac{c_D}{f_{H,D}} = \frac{14939.1\ \text{in/s}}{219.5\ \text{Hz}} = 68.1\ \text{in}$$

The wavelength is considerably larger than the port diameter or length. A similar calculation can be performed for suction conditions. We thus conclude that equation [3] is well satisfied; the assumptions required for the use of equation [1] are met.

Agreement with Measured Data

From *Figure 7* it can be seen that for the 100-degree span of crank rotation between 60 degrees and 160 degrees where suction occurs, there are approximately 10 complete cycles of the resonance waveform. This corresponds to period T of 10 degrees per waveform cycle. Since the machine runs at 328 rpm (0.1829 sec/rev), the time for the crank to traverse 10 degrees of rotation is approximately equal to the period T of the waveform

$$T \approx (0.1829 \text{ sec}) \frac{10 \text{ deg}}{360 \text{ deg}} = 0.00508 \text{ sec}$$

The frequency is found as

$$f_{H,S} = \frac{1}{T} = \frac{1}{0.00508 \text{ sec}} = 197 \text{ Hz}$$

This is in reasonably good agreement with our calculations of 191.4 Hz at suction conditions. Similar calculations can be made for the discharge part of the stroke. Approximately three cycles of the waveform occur between 300 and 329 degrees. Working through the numbers as before gives a frequency of 204 Hz, again in reasonably good agreement with our predicted value of 218.5 Hz.

As noted earlier, a good design will have a Helmholtz resonance frequency that is at least 100X higher than the running speed of the machine. For this machine, the running speed is 328 rpm (5.47 Hz). The Helmholtz resonance frequency should therefore be at least 547 Hz to avoid interference with our diagnostic waveforms. In this example, the measured Helmholtz resonance frequencies of approximately 197 Hz and 204 Hz correspond to 36X and 37X running speed respectively. Not surprisingly, the pressure versus crank angle waveform shows appreciable resonance, indicative of a poorly designed installation.

Example 2 – An Installation Without Channel Resonance

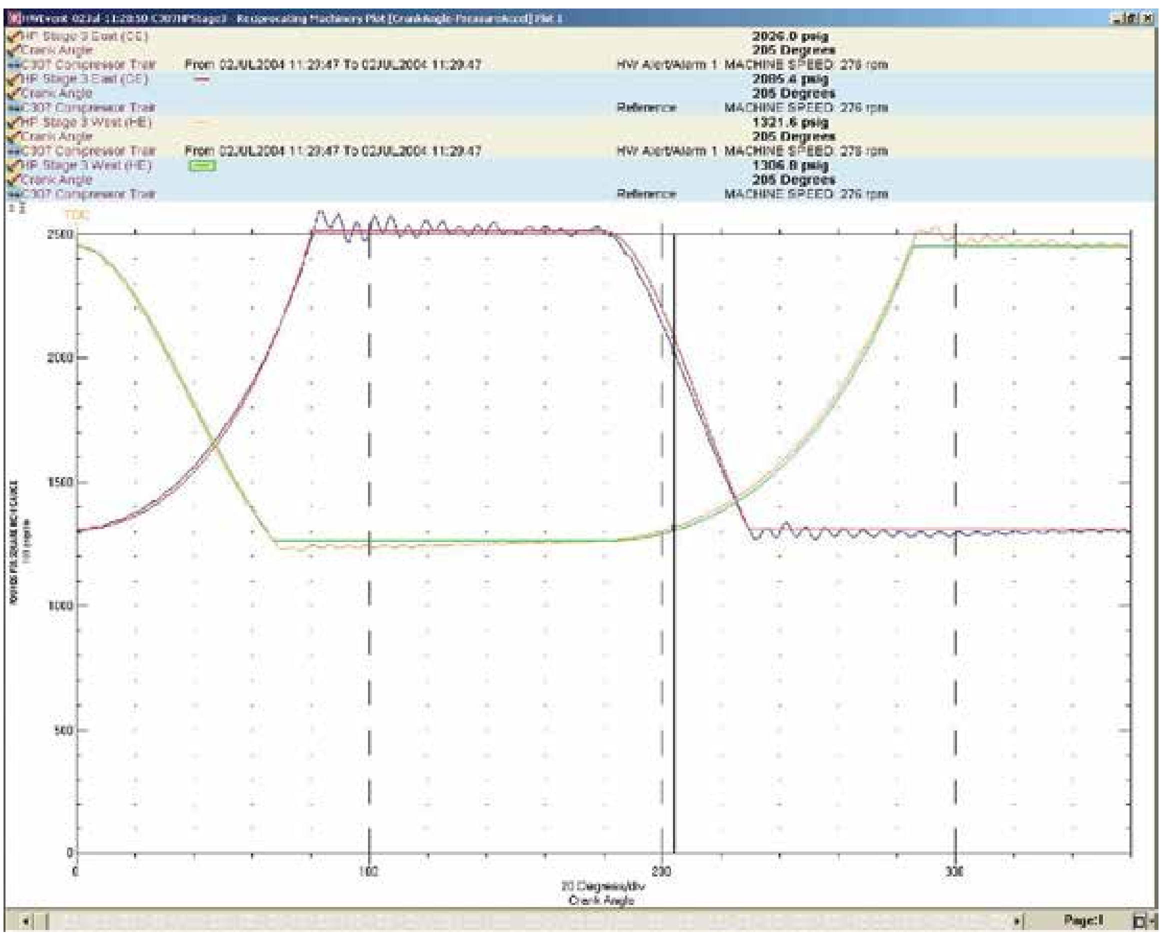

Figure 8. Pressure versus crank angle for both crank and head end cylinders in an installation free of Helmholtz Resonance problems.

Figure 8 shows a pressure versus crank angle curve for an installation that does not suffer from channel resonance. The pressure port has the following geometries:

Port length L = 18 inches

Port diameter d = 0.19 inches

Chamber volume = 0.15 in^2

Figure 9 summarizes the gas composition for this cylinder while *Table 3* summarizes the process conditions at suction and discharge.

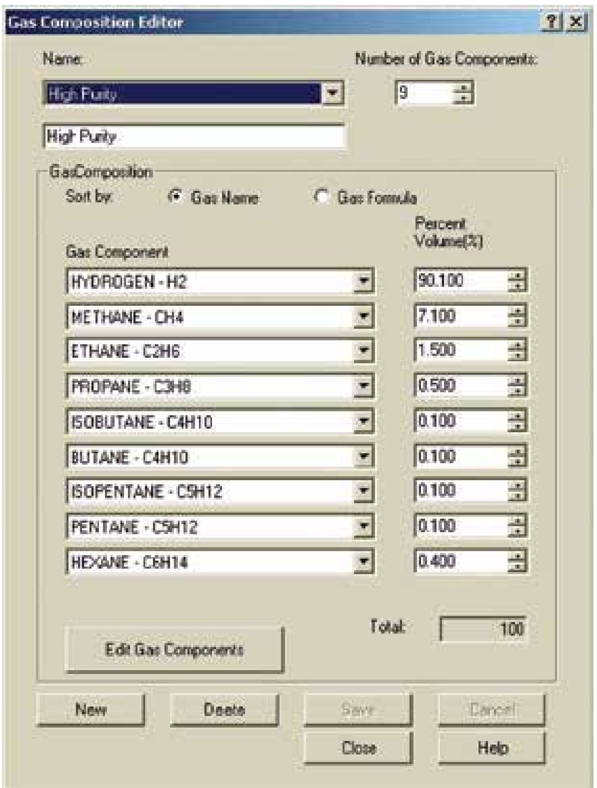

Figure 9. Gas composition for reciprocating compressor of Figure 8.

	Pressure *P* (PSIG)	Temperature *T* (R)	Compressibility *Z*	Isentropic Exponent *k*
Suction	1310	561	1.020	1.400
Discharge	2460	686	1.076	1.412

Table 3. Process conditions at suction and discharge for reciprocating compressor of Figure 6.

The fluid gas constant R for this mixture is:

$$365.2\frac{\text{lbf}}{\text{lbm}-°\text{R}}$$

From Table 3, the speed of sound can be calculated at suction and discharge conditions:

$$c_S = \sqrt{(1.400)\,\frac{365.2\ \text{ft}-\text{lbf}}{\text{lbm}-^{\circ}\text{R}}\,(561°\text{R})\,\frac{\text{lbm}-32.2\ \text{ft}}{\text{s}^2}} = 3039\frac{\text{ft}}{\text{s}} = 36470\frac{\text{in}}{\text{s}}$$

$$c_D = \sqrt{(1.412)\,\frac{365.2\ \text{ft}-\text{lbf}}{\text{lbm}-^{\circ}\text{R}}\,(686°\text{R})\,\frac{\text{lbm}-32.2\ \text{ft}}{\text{s}^2}} = 3375\frac{\text{ft}}{\text{s}} = 40500\frac{\text{in}}{\text{s}}$$

Shape factor m=0.85

$$f_{H,S} = \frac{c}{4\pi}\sqrt{\frac{\pi d^2}{V(L+(0.85\ d))}} = \frac{36470\ \text{in}}{\text{s}}\ \frac{1}{4\pi}\sqrt{\frac{\pi(0.19^2\ \text{in}^2)}{0.15\ \text{in}^3(18\ \text{in}+(0.1615\ \text{in}))}} = 592.2\ \text{Hz}$$

$$f_{H,D} = \frac{c}{4\pi}\sqrt{\frac{\pi d^2}{V(L+(0.85\ d))}} = \frac{40500\ \text{in}}{\text{s}}\ \frac{1}{4\pi}\sqrt{\frac{\pi(0.19^2\ \text{in}^2)}{0.15\ \text{in}^3(18\ \text{in}+(0.1615\ \text{in}))}} = 657.6\ \text{Hz}$$

Wavelength Check

As in Example 1, we check to ensure that the wavelength of the Helmholtz Resonance is indeed considerably larger than the port length and diameter. For the suction conditions we obtain:

$$\lambda_{H,S} = \frac{c_S}{f_{H,S}} = \frac{36470\ \text{in/s}}{594.\ 8\text{Hz}} = 61.3\ \text{in}$$

A similar calculation can be performed for discharge conditions. As before, the wavelength is well above the pressure port dimensions and the assumptions of [3] required for the use of [1] are again well satisfied.

Agreement with Measured Data

The compressor operates at 277 rpm (4.6 Hz) while the lowest calculated resonance on the suction side is 592 Hz or approximately 128X running speed. On the discharge side, the lowest calculated resonance is 658 Hz or approximately 142X running speed. Not surprisingly, *Figure 8* shows no evidence of resonance anywhere in the waveform, representative of a well-designed installation.

Summary

Provided the gas composition and process conditions are known, and the pressure port geometries are considerably smaller than the acoustic resonance wavelength, equation [1] as outlined herein can be used to determine the Helmholtz Resonance frequency fH. for a pressure port contemplated for use with a cylinder pressure transducer. To ensure the pressure transducer does not suffer from appreciable signal interference due to this Resonance, a resonant frequency that is at least 100X the compressor's running speed should be chosen. When these guidelines are adhered to, the installation will yield good data integrity that is more easily interpreted, allows for more accurate thermodynamic calculations, and permits easier assessment of overall cylinder health.

References

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