Introduction to Vibration & Pulsation in Reciprocating Compressors

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ABSTRACT

This paper is intended to provide a basic understanding of pulsation and vibration in reciprocating compressor installations. Common terminology used in acoustical and mechanical analysis will also be presented.

1.0 Vibration

Definition: The periodic movement of a body about an equilibrium position

1.1 Vibration Basics

Vibration amplitude is a function of an applied force and the stiffness (or flexibility) at a given frequency:

$$\text{VIBRATION} = \frac{\text{Dynamic Force}}{\text{Dynamic Stiffness}} = \text{Dynamic Force X Dynamic Flexibility}$$

In controlling vibration both aspects of the vibration equation must be considered.

To explain vibration, and the units it can be measured in, consider a spring-mass system. The following comments refer to Figure 1.1.

- A block hanging on a spring will stretch the spring until the upward force of the spring equals the weight of the block.
- If the block is displaced down a distance (e.g. 1.5”), and released, the spring will pull the block back to its original position. Momentum will cause the block to continue travelling upwards the same distance (i.e. 1.5”) from the original position, where it will stop and start moving back down.
- The time the system takes to complete one full oscillation is the **period** of oscillation. The number of periods per unit of time is the **frequency** of the vibration. Typical units of frequency are cycles per minute (CPM) and cycles per second (cps), which is also known as hertz (Hz).
- In terms of displacement, the **peak-to-peak** vibration amplitude (or range of motion) is the distance travelled by the block from the highest to lowest position. The **peak** amplitude of the vibration is the peak to peak distance divided by two. Displacement amplitudes are typically recorded in mils (1 mil = 1/1000 inch) and must also be defined as peak or peak-to-peak.
- Vibration velocity is the rate at which the block moves. The velocity of the block is maximum when the block passes through the equilibrium position. The velocity is zero when the block is at the highest or lowest displacement. Velocity amplitudes are typically recorded in inch/second (ips) peak.
- Acceleration is the rate of change of the velocity of the block. The acceleration of the block is maximum when the velocity is zero and the block is at the highest or lowest displacement. The block acceleration is zero when the velocity is maximum, and the block passes through the equilibrium position. Typical acceleration amplitudes are recorded in g’s peak (where 1 g = 386 inches per second squared).

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Vibration Basics

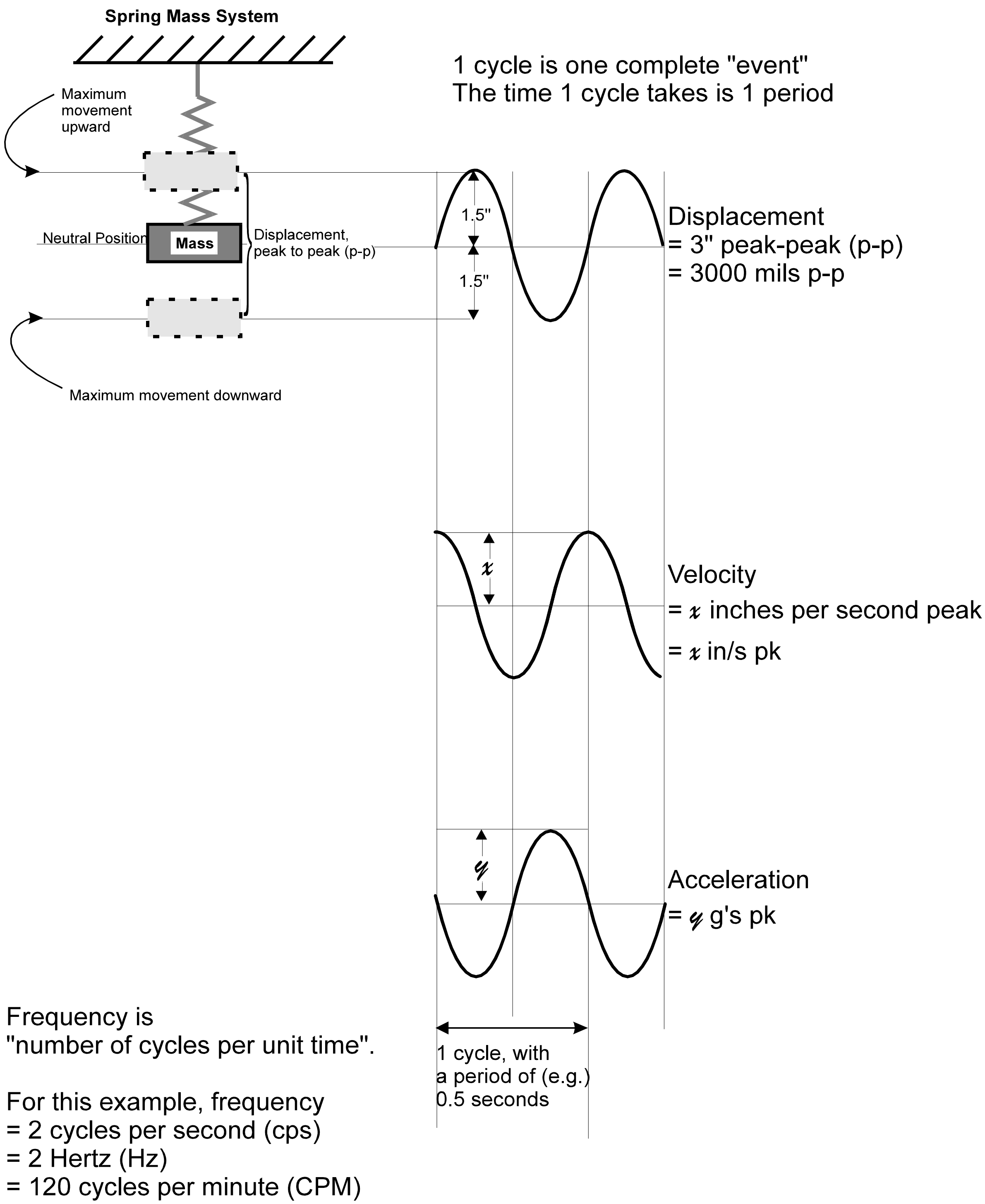

Figure 1.1 The motion of the block can be shown as displacement, velocity, or acceleration

1.2 Forcing Functions

Reciprocating machines always generate forces that cause vibration. Most of these forces are a function of the machine design, and cannot be avoided. Pulsation induced shaking forces are a function of operating conditions, and they can be minimized. Pulsation induced unbalanced forces are discussed in detail in Section 2.6.

The table below shows the forcing functions of concern in a reciprocating compressor, the frequency at which the forces are generally largest, and methods of minimizing the force, where possible.

Forcing Function	Dominant Frequency (Multiple of Run Speed)	How to Minimize Force
Mass Unbalance Mass unbalanced in opposing reciprocating components	***1X***, 2X	Minimize opposing mass unbalance (e.g. 0.5 to 1 lbs for 1000 RPM, 6" stroke unit).
Moment/Couple Created by the offset of opposed reciprocating components	***1X***, 2X	Inherent in design.
Alignment Angular and parallel alignment of driver and compressor	1X, 2X	Check angular and parallel alignment.
Pulsation *Pulsation induced shaking forces (see Section 2).	***1X, 2X***, 3X, 4X, . . .	Control pulsations using acoustical simulation techniques.
Cylinder Stretch *Elongation/shortening of cylinder assembly due to internal gas forces.	***1X***, 2X, 3X, 4X, . . .	Check that cylinder assembly bolts are properly torqued.

NOTE: * - On average these forcing functions decrease with increasing multiples of runspeed.
- The most significant forcing functions occur at 1X and 2X compressor run speed.

In some situations, torque fluctuations can cause vibration problems. These situations are rare and only occur when the skid is very flexible. Torque fluctuations will have input frequencies of .5X, 1X, 1.5X, etc. for units with four cycle engines, and 1X, 2X, 3X, etc. for units with two cycle engines or motors.

Crosshead forces, caused by gas pressures and reciprocating inertia, can excite foundation modes if the foundation is resonant. Crosshead forces act perpendicular through the crosshead guide primarily at 1X and 2X.

1.3 Mechanical Natural Frequencies (MNFs)

The mechanical natural frequency of a component is the frequency at which the component vibrates the most in response to a given force. For example, a spring-mass system will oscillate at its natural frequency if the weight is pulled down and then released.

All components, or groups of components, (piping, pulsation bottles, scrubbers, cylinders, relief valves, etc.) in a reciprocating compressor installation will have several mechanical natural frequencies. The mechanical natural frequencies of a pipe or piping system depend on lengths, schedules, diameters, elbows, supports, etc.

A typical mechanical natural frequency plot is shown in Figure 1.2. The peak shows the frequency at which maximum dynamic flexibility (or minimum dynamic stiffness) occurs in the system. Modifications to the system will shift that peak: increasing mass or decreasing stiffness shifts the peak to the left, and increasing stiffness or decreasing mass shifts it to the right.

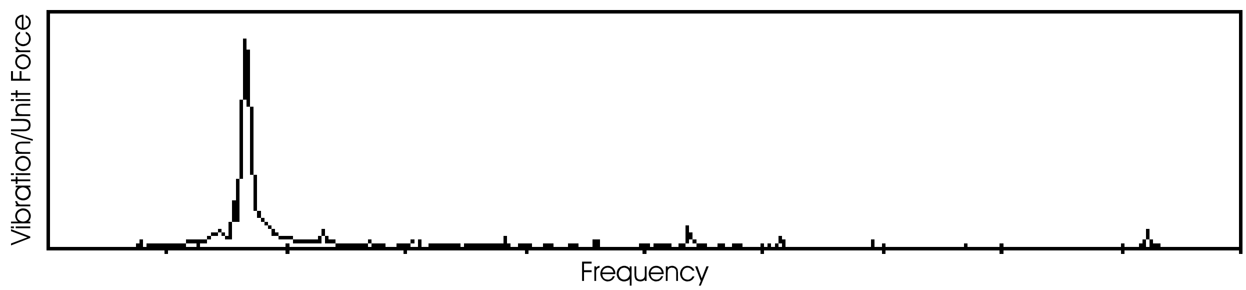

Figure 1.2 Typical Mechanical Natural Frequency Plot

1.4 Resonance

Mechanical resonance of a piping component occurs when a forcing function is applied at a frequency coincident with a mechanical natural frequency of the component. Figure 1.3 illustrates the concept. When a system, or part of a system, is mechanically resonant, normal (or even low) unbalanced force levels can couple with the system geometry to produce ***very high*** vibration levels.

If an oscillating force of constant amplitude is applied to a system over a range of frequencies, the resulting vibrations of the system will vary. Response will be high at frequencies where the system is flexible, and low elsewhere.

The shape and the magnitude of the response peak at resonance is a function of the structural damping, or resistance, in the system. The more damping in the system, the broader and lower the peak will be. Structural damping comes from flanged and bolted connections, clamping, material characteristics, etc.

The average reciprocating compressor package, however, has very little damping. Effective damping for most components is less than 3%. At resonance, the vibration is usually greatly amplified, sometimes by as much as a factor of 100. Therefore, the best way to avoid vibration problems due to resonance is to avoid high dynamic flexibility (low dynamic stiffness) at the frequencies where the input forces are the greatest (i.e. at 1X and 2X compressor speed).

> **Note:** Beta Machinery Analysis' field experience shows that over 90% of serious vibration problems encountered in the field, where failures are present, are related to mechanical resonance.

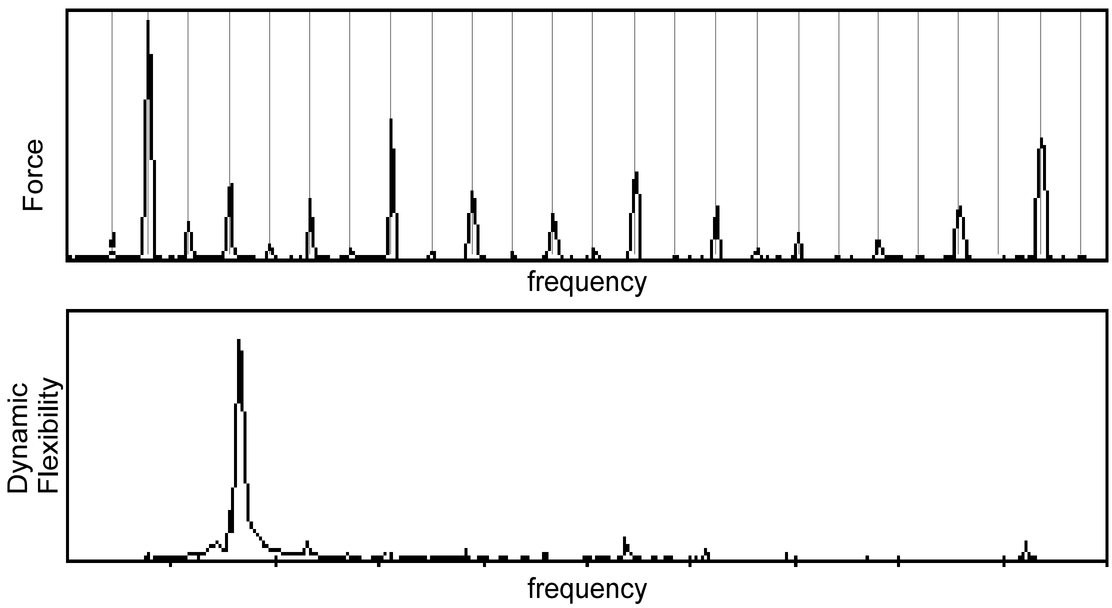

Figure 1.3 When force frequency and natural frequency coincide, you have resonance.

1.5 Acceptable Limits

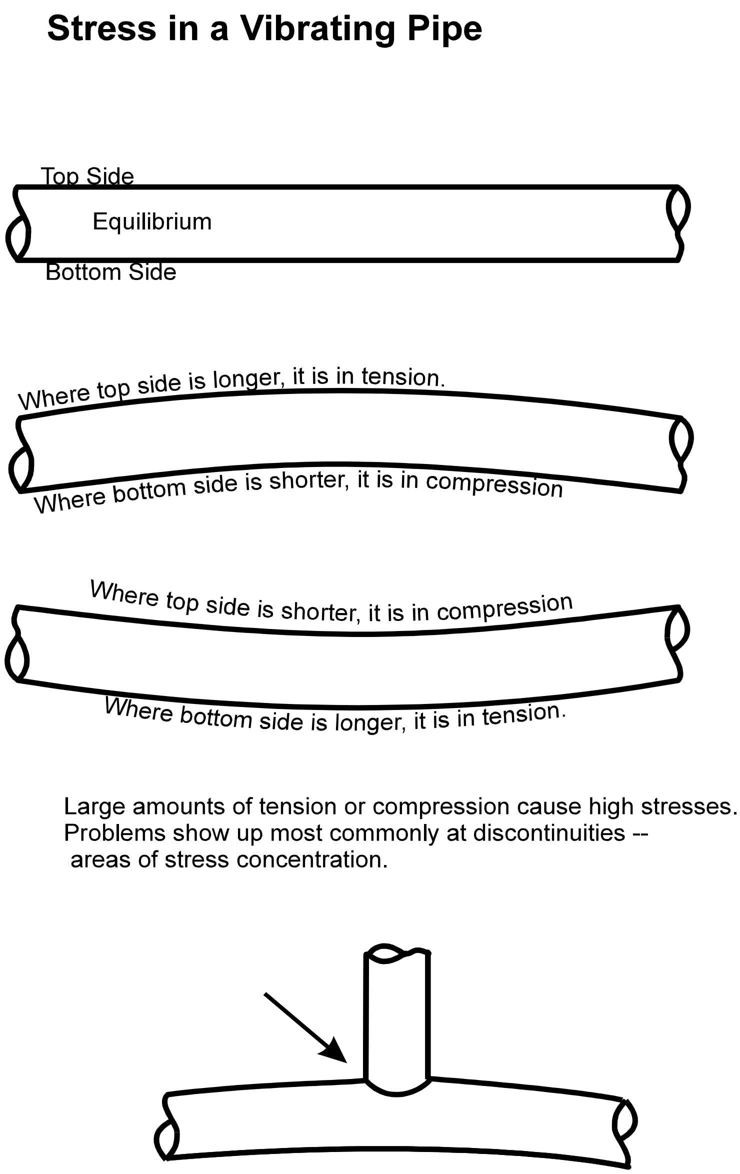

Figure 1.4 Stress in a Vibrating Pipe

Vibration is inherent in reciprocating machinery. Although it can be controlled, it cannot be completely eliminated.

Ideally, to determine what vibration levels are acceptable, stress levels resulting from the vibration should be considered.

When a mechanical system vibrates, it is moved from its normal or equilibrium position. Considering the top section of a piece of pipe, the vibration alternately puts the top of the pipe in tension and compression, causing stresses in the pipe.

Vibration causing large amounts of tension or compression can result in high stresses in piping, which can damage the pipe. Totally eliminating all the vibrations in a given system would not be necessary or practical from a design and/or economic standpoint. However, it is necessary to reduce vibration to a level where failure will not occur.

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It is much easier to measure vibration levels than stress; therefore, vibration limits are commonly used for defining problems. While generating vibration criteria for every possible situation would be next to impossible, vibration limits based on field experience have been established. Vibration limits generally considered industry standard are given in Figures 1.5 and 1.6. These charts give a guideline limit, as well as other limits, in displacement and velocity. The charts are presented as frequency versus vibration amplitude.

The vibration limit charts **do not** distinguish between very stiff areas, such as a well supported discharge bottle, and very flexible areas, such as piping rising to coolers. While following the limits on these charts will be satisfactory for most installations, there is no guarantee that they will be acceptable in **all** cases.

Acceptable stress and vibration levels depend on many factors, a few of which are:

- material (composition, strength, endurance, etc.)
- geometry (size, quality of manufacturing, stress concentrations such as tee intersections and cutouts, etc.)
- frequency of stress cycle, and
- amount of residual static stress.

1.6 Summary

Unacceptable vibration in a compressor installation at a particular frequency can be the result of high forces, low dynamic stiffness or a combination of the two. Some common causes of high vibration are:

- pressure pulsation-induced unbalanced forces,
- unbalanced forces and moments caused by reciprocating parts,
- crosshead forces caused by gas pressures and reciprocating inertia,
- cylinder stretch (ie. elongation and shortening of the cylinder assembly due to internal gas forces),
- misalignment between the driver and compressor,
- mechanical resonance,
- pipe strain (bolt-up),
- thermal strain, and
- low damping.

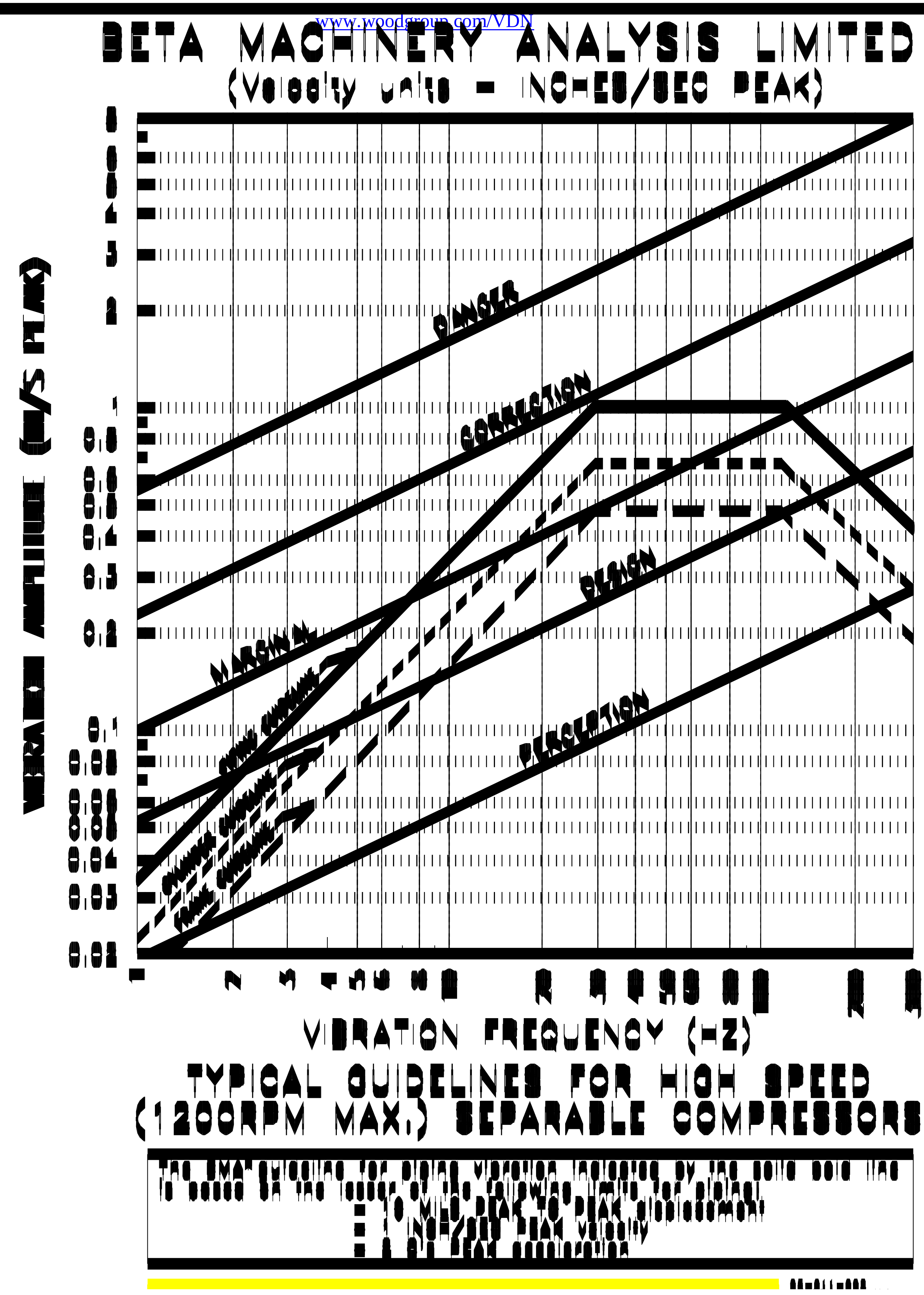


Figure 1.5 Vibration Standard Velocity Units

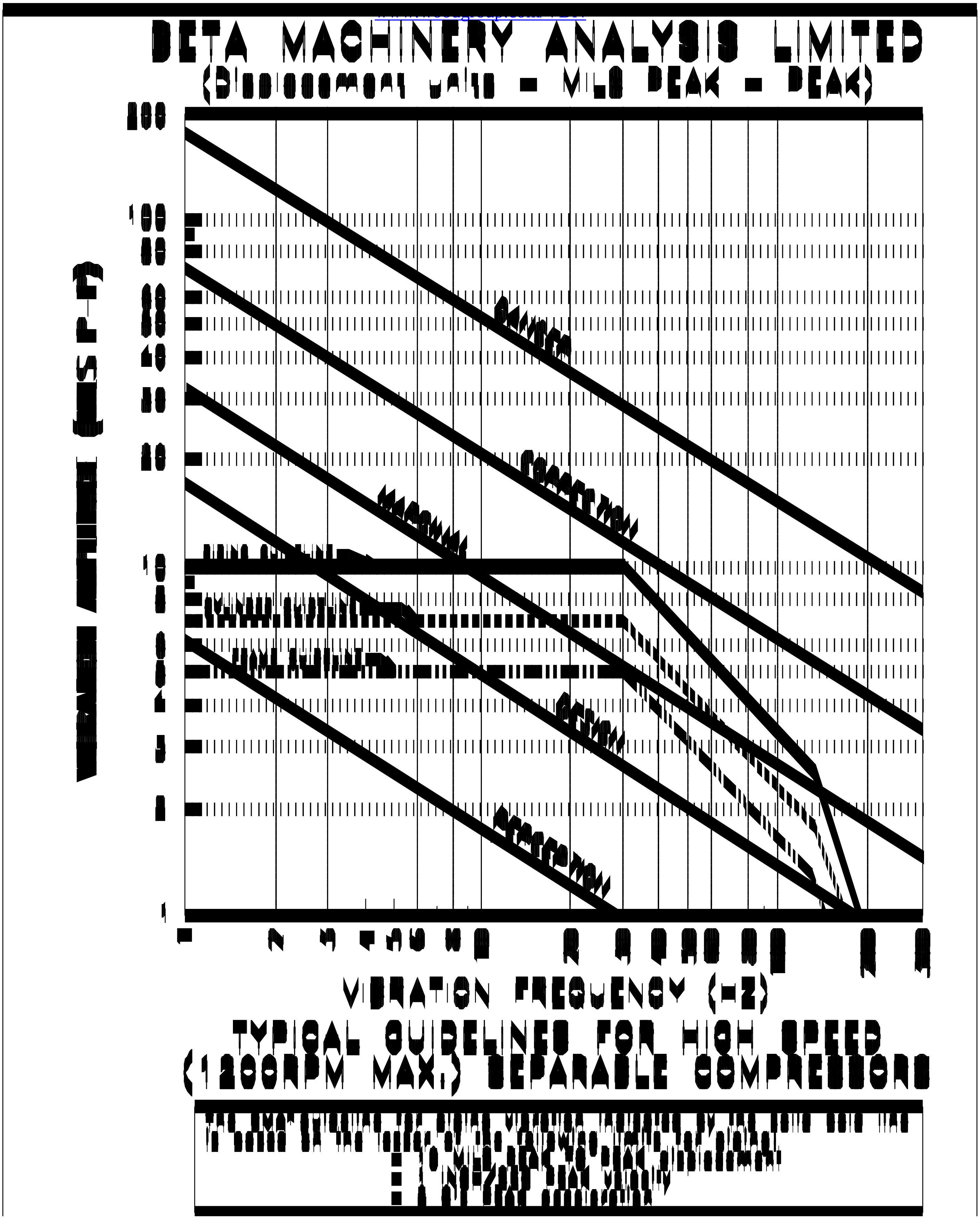


Figure 1.6 Vibration Standard Displacement Units

2.0 Pulsation

Definition: The rapid rise and fall of the instantaneous line pressure about a constant pressure level (i.e. gauge pressure). Pulsation is typically measured in psi (or kPa) **peak-to-peak**.

2.1 Development of A Pressure Pulsation

The reciprocating action of a piston causes pulses (regions of high pressure and low pressure relative to a constant pressure level) to travel through the piping system. Considering a simple compressor piston, with no valves, we can understand how pulsations are formed.

The piston moving back and forth forms regions of compression and rarefaction, which travel through the piping system at the speed of sound of the gas. The positive and negative pressure regions travelling away from the piston cause pressure fluctuations, or *pulsations*.

The wavelength, or the distance between regions of compressed gas (or regions of rarefaction) of the pulsation is a function of the speed of sound of the gas and the frequency of the pulsations:

$$\text{Speed of Sound} = \text{Frequency X Wavelength}$$

Or

$$C = f \text{ x } \lambda$$

There is basically a linear relationship between the velocity of the face of the piston and the resultant pressure. Since the piston motion is nearly sinusoidal the pressure wave will also be nearly sinusoidal.

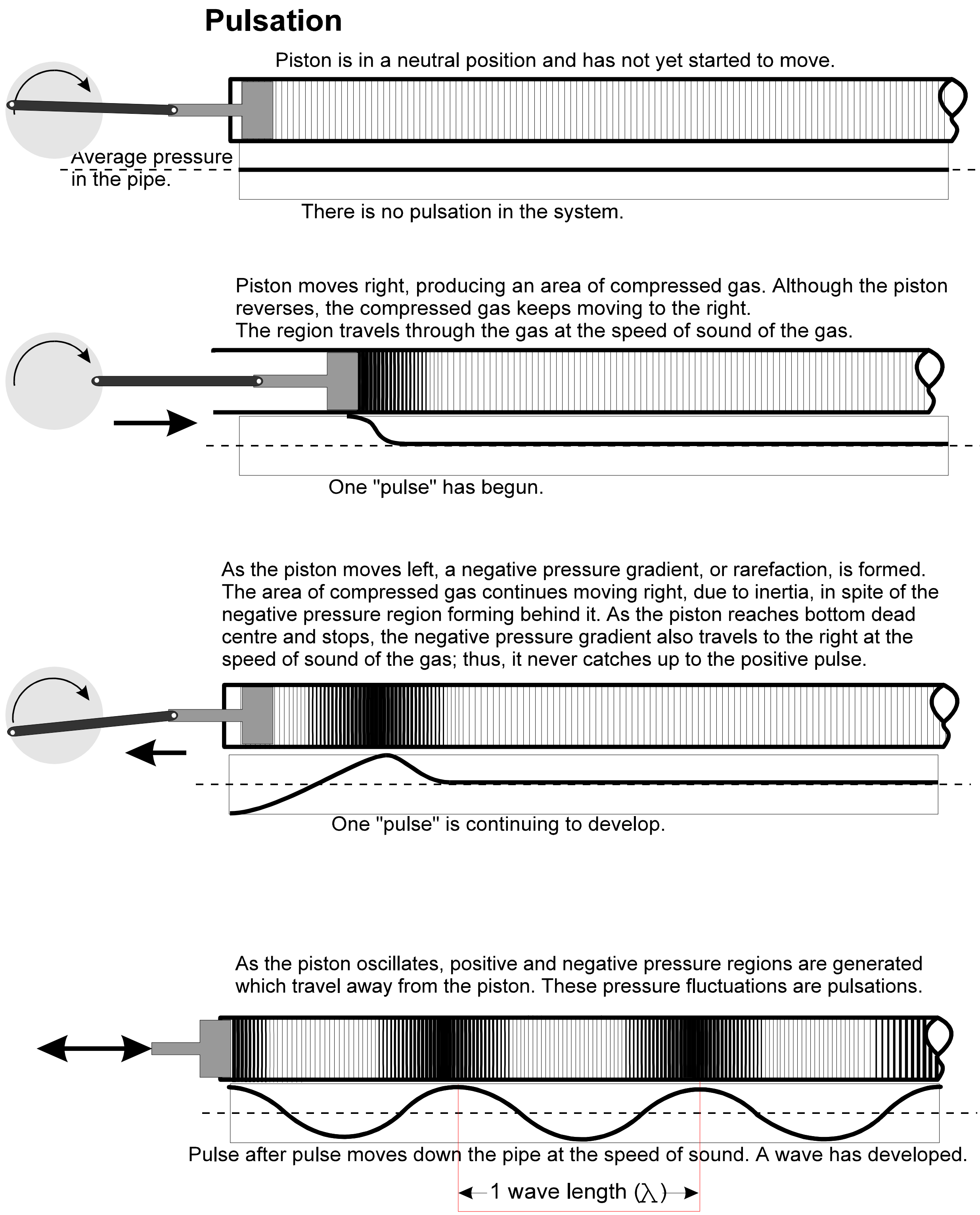


Figure 2.1 Piston Motion Generates Pulsations

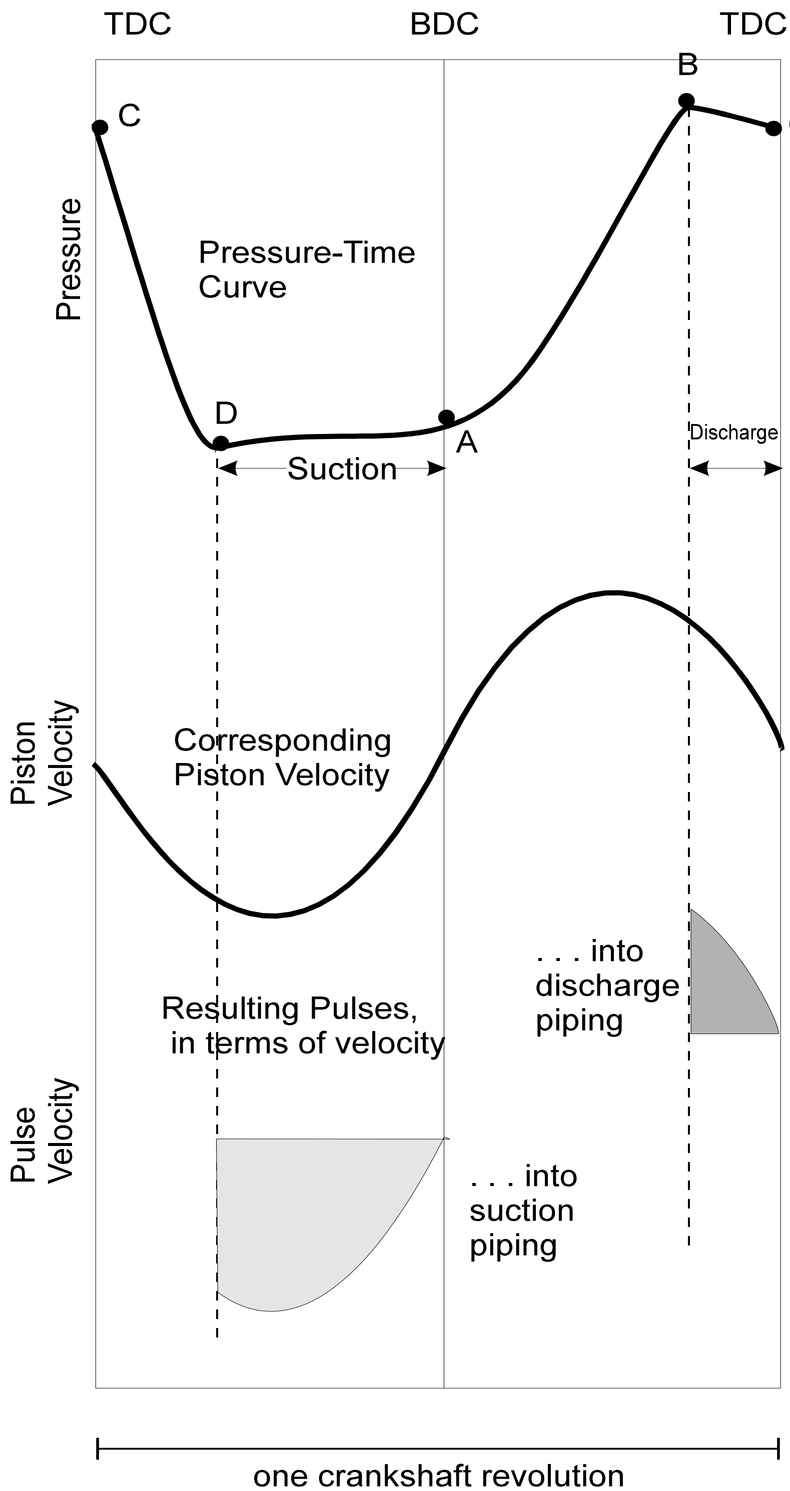


Figure 2.2 Pressure-Time and Velocity-Time Curves

Pressure-Time Curve

A pressure-time curve of a compressor cylinder will help explain how pulsations are formed in a reciprocating compressor system, i.e. with compressor valves.

At ‘A’ in Figure 2.2, BDC (bottom dead center), the head end cylinder cavity is filled with gas at suction pressure.

As the piston moves from BDC toward TDC (top dead center) the gas is compressed and the pressure rises to discharge pressure.

At ‘B’, when the cylinder pressure reaches a pressure slightly higher than discharge pressure, the pressure differential across the discharge valves causes the valves to open, allowing the fluid to be discharged into the system until the piston reaches TDC.

At ‘C’, at TDC, the discharge valves close.

As the piston moves from TDC toward BDC the fluid is allowed to expand until the pressure in the cylinder cavity drops to a pressure slightly less than suction pressure.

At 'D', when the cylinder pressure reaches a pressure slightly lower than suction pressure, the pressure differential across the suction valves causes the valves to open. Fluid is drawn into the compressor until the piston returns to BDC, at which time the suction valves close. The suction pressure is equalized inside and outside of the cylinder allowing the spring to close the valves.

Piston Velocity Curve
The velocity-time trace of the face of the piston shows the near sinusoidal motion of the piston. The motion is not purely sinusoidal because of the linkage between the piston rod and connecting rod.

Pulse Velocity Curves
Due to the continuous reciprocating motion, modulated by the valve action, periodic pulses are transmitted into the attached piping systems. The velocity of the piston when a valve is open largely determines the characteristic shape of the pulse of gas being emitted.

Changes in operating conditions such as temperature, gas composition, line pressure, compressor speed and compressor loading can change the characteristics of pulsations being emitted into a piping system. Pulsations produced by the reciprocating action of the piston, at a given operating condition, will have distinct amplitude and frequency characteristics.

2.2 Frequency Content (Harmonics)

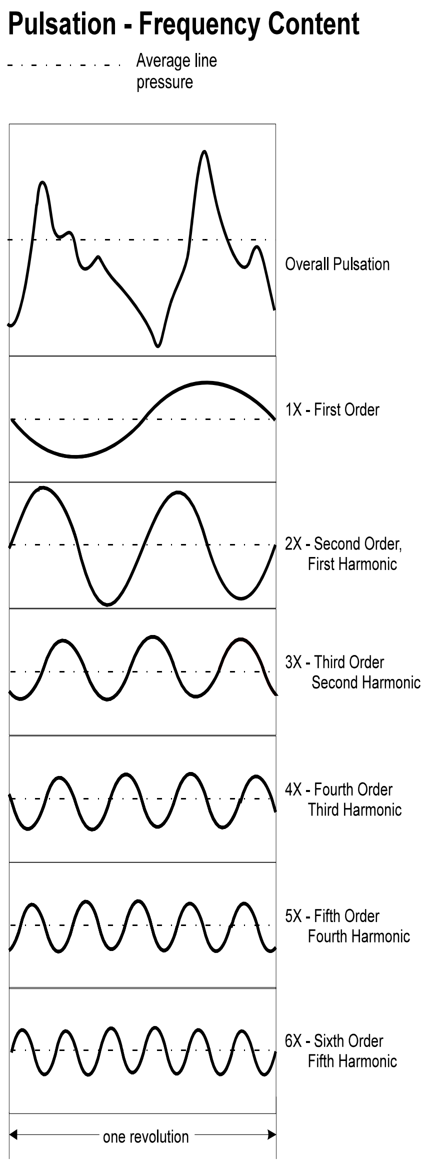


Figure 2.3 Pulsation – Frequency Content

Considering a theoretical cylinder model which does not require compressor valves, we can visualize how a single-acting cylinder (either the head end or the crank end of the piston is compressing gas) will emit one pulse of gas per revolution of the crank shaft. Thus the frequency of the pulsations being generated is one times the fundamental frequency (i.e. compressor speed). For a double-acting cylinder (where both the head end and crank end of the piston are compressing gas) two pulses of gas are emitted per revolution. The frequency of the pulsations produced by the double-acting cylinder is two times the fundamental frequency.

However, valves are necessary for the cylinder to compress the gas, and the pulses being generated are not pure sine waves. Thus the frequency content of the pulsations being generated is more complex than a simple sine wave. Given the basic characteristic shape of the pulse being emitted into the compressor system, the pulse can mathematically be broken down into frequency components.

The frequency content of a pressure pulsation can also be determined by sweeping a raw signal, the overall pulsation, through a filter over a frequency range. Pulsation components will appear at frequencies equal to integer multiples of the fundamental frequency (Similarly, a radio tuned to one station will filter out all but one frequency from the air waves.) Figure 2.3, Pulsation – Frequency Content, shows a theoretical pressure trace downstream of the discharge valves for a double-acting cylinder. The figure shows overall pressure versus time for one revolution (or cycle), as well as the breakdown of the frequency content.

- the raw signal is shown as the overall pressure trace, for one cycle.
- the subsequent pressure traces indicate the results of passing the raw signal through a filter at various frequencies.
- the signal, or overall pulsation, is made up of pulsations not only at the fundamental frequency, but also at 2 times, 3 times, 4 times, etc. the fundamental frequency. The multiples of the fundamental frequency are referred to as harmonics. (Abbreviated – 1X, 2X, 3X, etc.)

For a double-acting cylinder the pulsation components will be higher at the even multiples (2X, 4X, 6X,etc.). For a single-acting cylinder the pulsation components will be higher at the odd multiples (1X, 3X, 5X, etc.).

Figure 2.4, Pulsation – Spectra, shows a typical pulsation spectrum (amplitude versus frequency) for a double-acting operation.

Pulsations, and hence unbalanced forces, are generated not just at compressor run speed but at many harmonics Thus the potential for vibration problems occurs at all harmonics. By far the majority of pulsation related problems in reciprocating compressors occur in the 0 – 200 Hz (i.e. 1X to 10X runspeed for a 900 RPM compressor) range. Although some high pulsations exist at frequencies substantially above 200 Hz, such as in the internal gas passages of a cylinder, they remain localized and normally are not transmitted into the piping system.

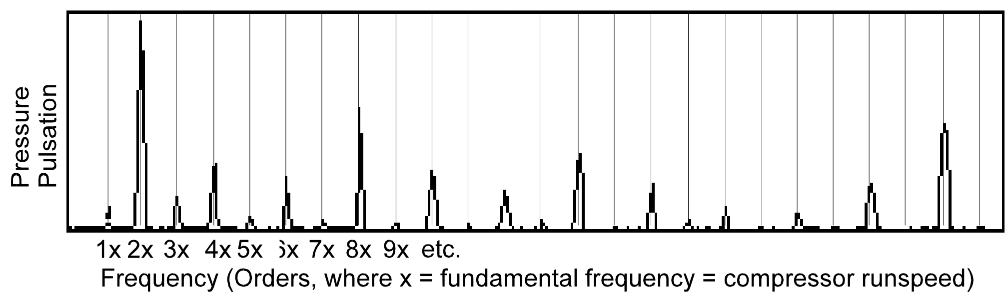


Figure 2.4 Typical Pulsation Spectrum for a Double-Acting Compressor

Figure 2.5 shows the range of frequencies swept by a particular harmonic, for a 600 to 1000 RPM compressor. For example, if a force is input into the system at 7X fundamental frequency, it would "influence" the system between 70 and 120 Hertz.

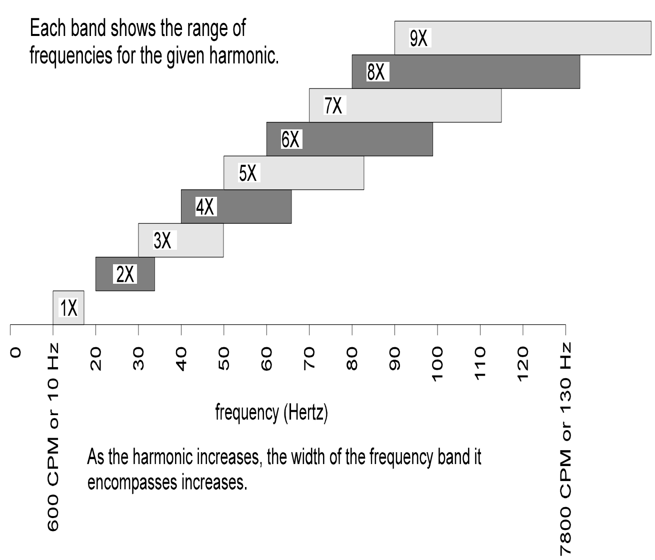


Figure 2.5 Harmonics versus Frequency range

2.3 System Definition: A set of components (bottles, pipe runs, scrubbers, side branches, coolers, etc.) which are affected by the same pulsation source or sources.

Figure 2.6 Interstage System

Interstage systems are the simplest to understand. They include all components from the cylinder passages of the discharge side of one stage to the suction cylinder passages of the next stage. Pulsations do not travel through the cylinders as there is no physical connection (the suction and discharge valves are not open on the same end at the same time).

Figure 2.7 First Stage Suction System

First stage suction systems usually include all components from the cylinder passages upstream to an inlet separator.

Figure 2.8 Final Discharge System

For final discharge the system usually includes all components from the discharge cylinder passages downstream to a storage vessel, dehydrator, pipeline, or metering station.

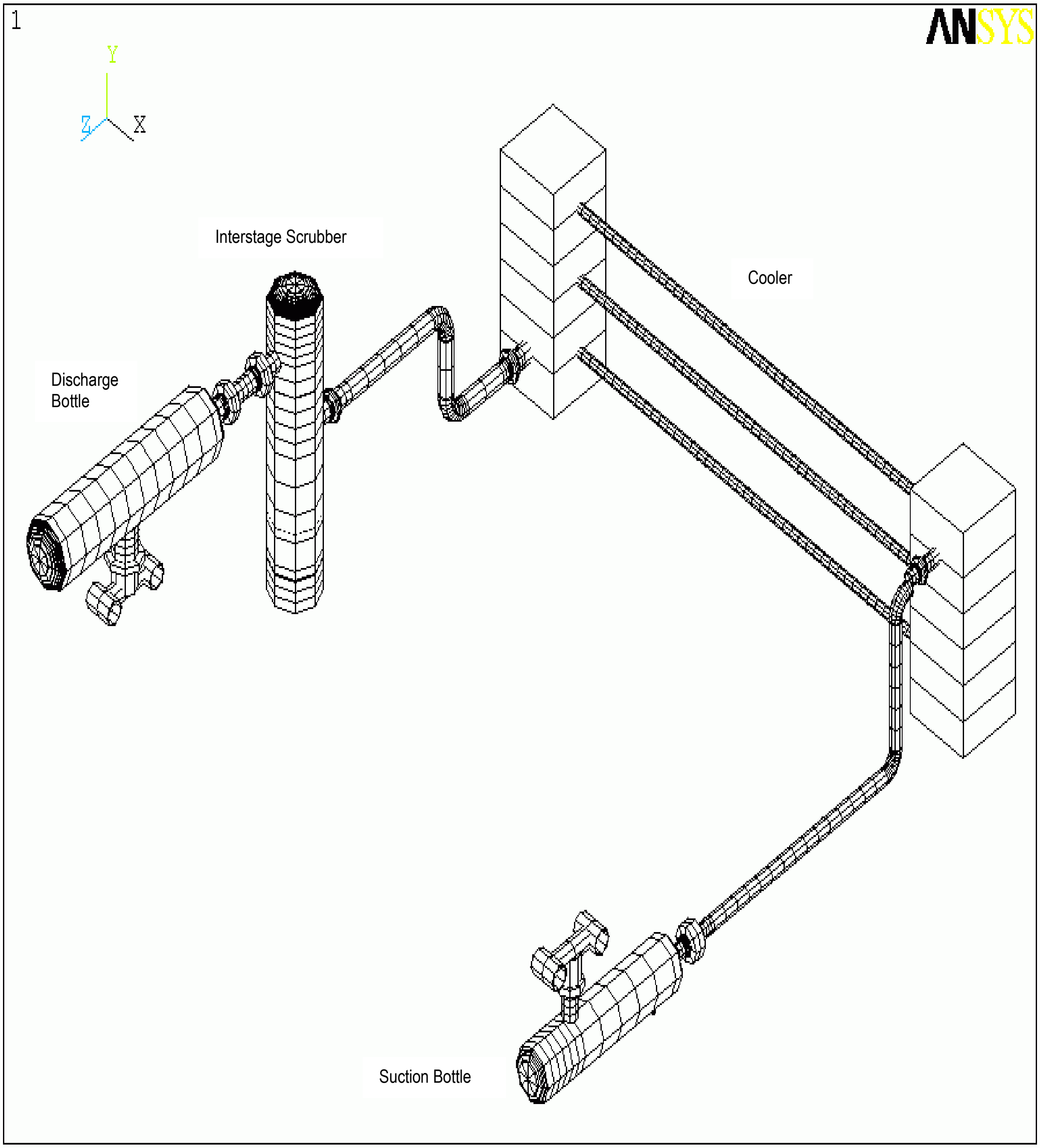
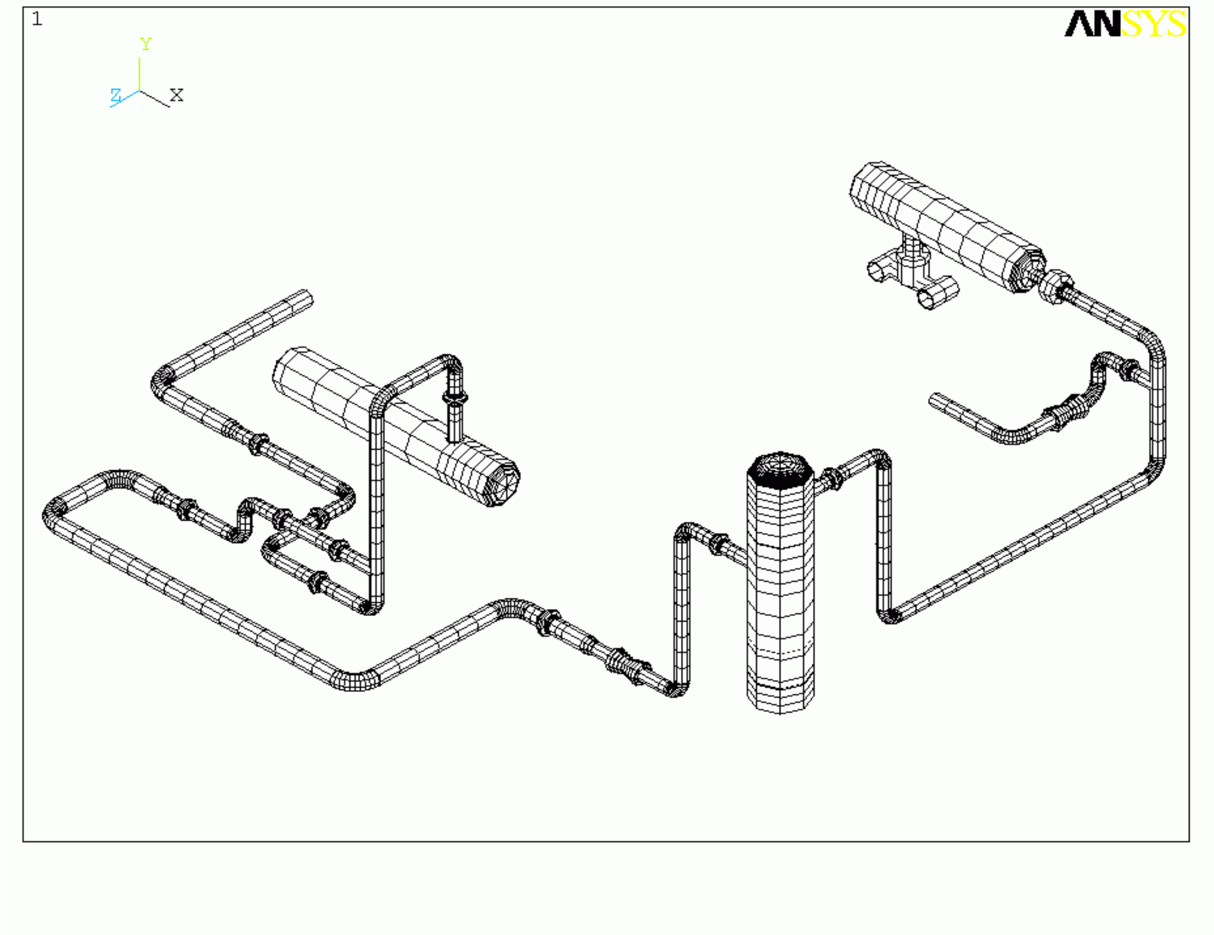


Figure 2.6 Interstage system extends from discharge of one stage to suction of the other stage.



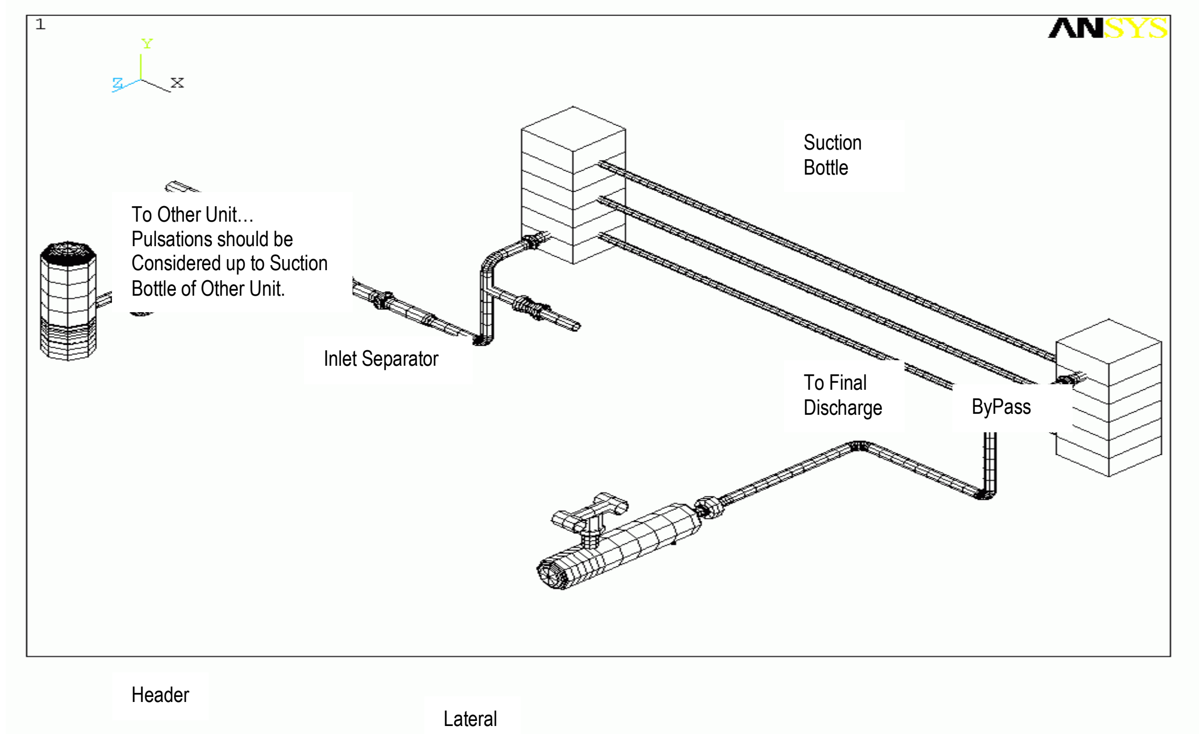


Figure 2.7 First stage suction system

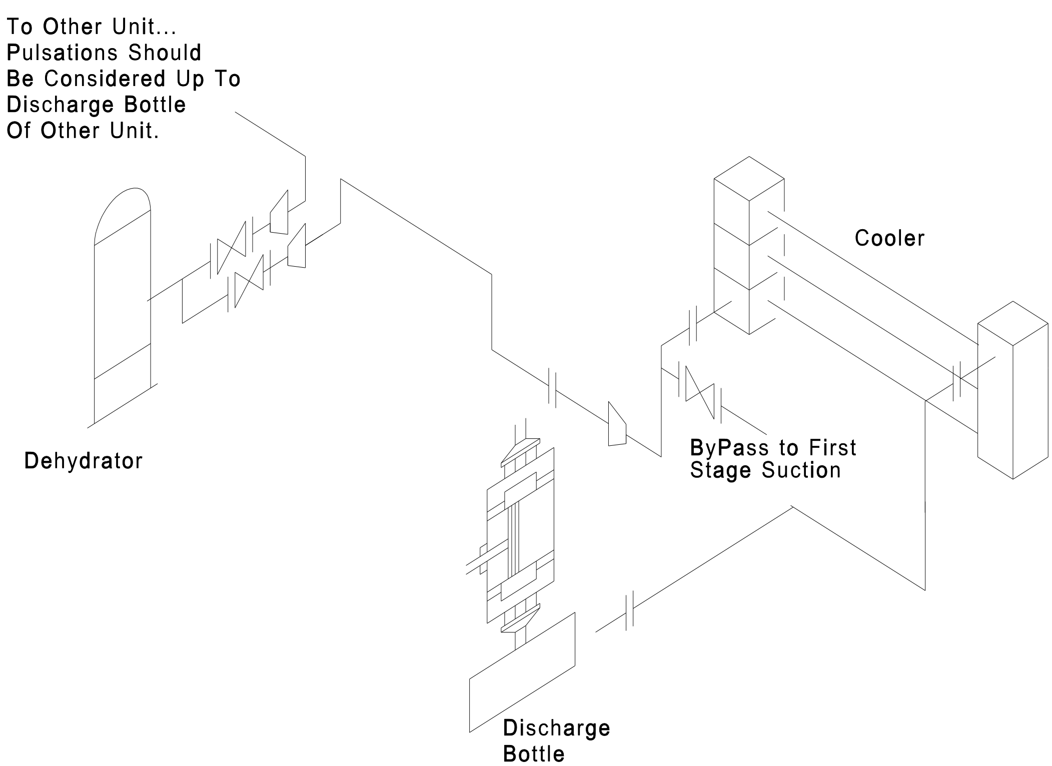
2.3 Extent of Pulsations

Pulsations always travel away from their source (i.e. the compressor) irrespective of the direction of gas flow. This occurs because the speed of sound at which the pulsations travel (typically in the range of 1200 ft/sec) is much greater than the flow velocities in the piping (typically in the range of 20 to 40 ft/sec).

When considering the presence and effects of pulsations in a reciprocating compressor installation, and to properly approach and solve pulsation related problems, it is important to realize how far the pulsations can travel. Consider the following example for a 900 RPM compressor:

- Fundamental Frequency: f = (900 CPM)/(60 sec/min) = 15 Hz
- Typical Speed of Sound of Gas: C = 1200 ft/sec
- As defined in Section 2.1: C = f x λ or λ = C/f = Wave Length
  λ = (1200 ft/sec)/(15 Hz) = 80 ft
- For 2 X Fundamental Frequency: f = (900 CPM)/(60 sec/min)X 2 = 30 Hz
  λ = (1200 ft/sec)/(30 Hz) = 40 ft

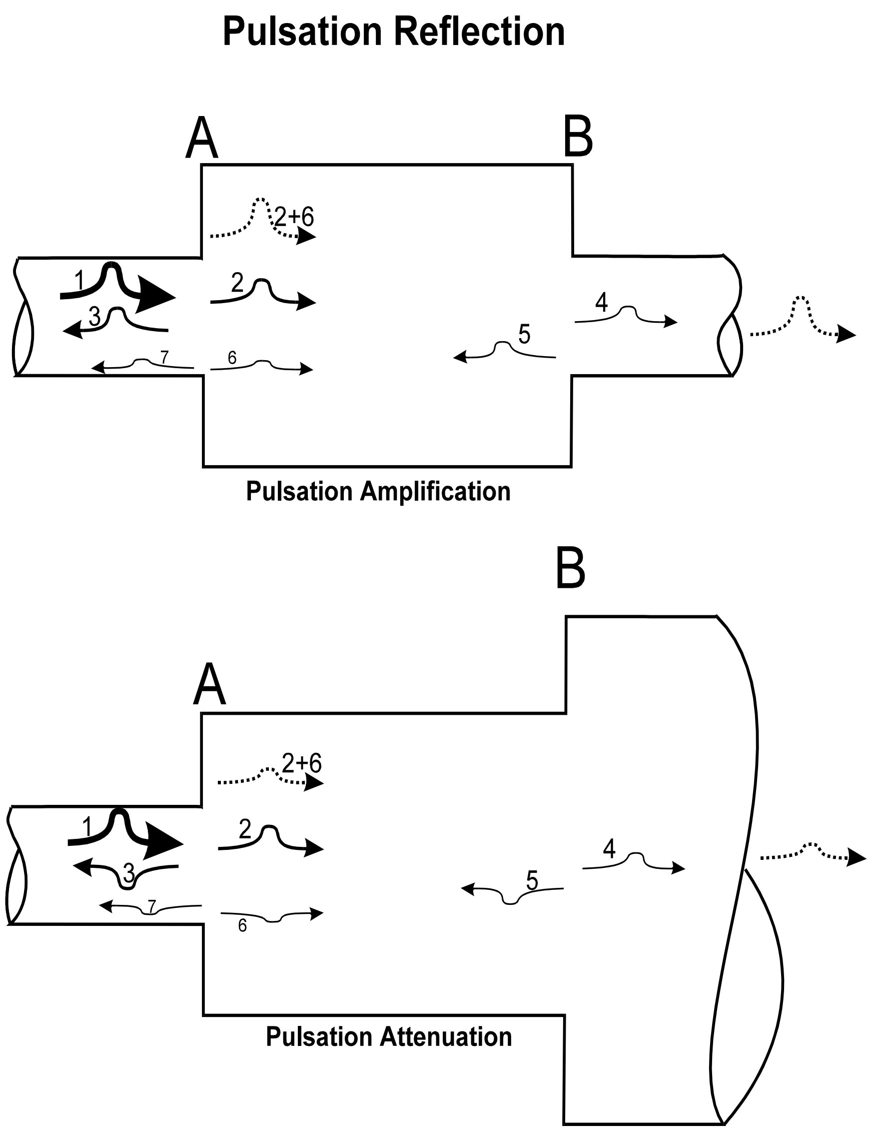
FINAL DISCHARGE SYSTEM



A pulsation travelling uninterrupted down a pipe will take several wave lengths before the amplitude shows a significant reduction due to the resistance of the piping. For this reason it is important to consider the system far enough upstream of the compressor, especially for single-acting compressors where the wave lengths of the pulsations are longer and carry further.

One problem encountered at locations considerably removed from the compressor is RMS (root mean square) metering errors at orifice meters.

2.4 Acoustical Resonance



Pulsations travel through a system at the speed of sound of the gas and at frequencies equal to compressor run speed and multiples thereof. The pulsation will travel away from the compressor uninterrupted until it comes across a change in pipe continuity (eg. change in diameter, tee, closed valve, etc.). At a change in continuity the pulsation will be reflected. The amount of reflection is a function of the severity of the discontinuity. The larger the change in continuity, the larger the reflection will be. NOTE: Pulsations will travel around corners without reflecting the pulse (at least within the 0 – 200 Hz frequency range)

Depending on the lengths and diameters of the attached system, the reflected pulsations will either amplify or attenuate. The greater amplification occurs at acoustical natural frequencies of the system. (The pressure fluctuations are referred to as *acoustical*, since they travel at the speed of sound in the gas). Acoustical resonance occurs when the frequency of the pulsations generated by the compressor coincides with an acoustical natural frequency (ie. when compressor run speed or multiples thereof coincides with the acoustical natural frequency).

Consider a pulse travelling in a pipe segment, Figure 2.9, Pulsation Reflection:

- Pulse "1" travels from left to right. At "A", when the pulse encounters the change in pipe size, the pulse is partially transmitted as pulse "2" and partially reflected as pulse "3".

- Pulse "2" continues to section "B", where another discontinuity is encountered. Pulse "2" is partially transmitted as pulse "4" and partially reflected as pulse "5".

- Pulse "5" travels back toward section "A", where it is partially reflected as pulse "6".

- If pulse "2" and pulse "6" start out from "A" at the same time they will either add or subtract, depending on the acoustical characteristics of the piping system.

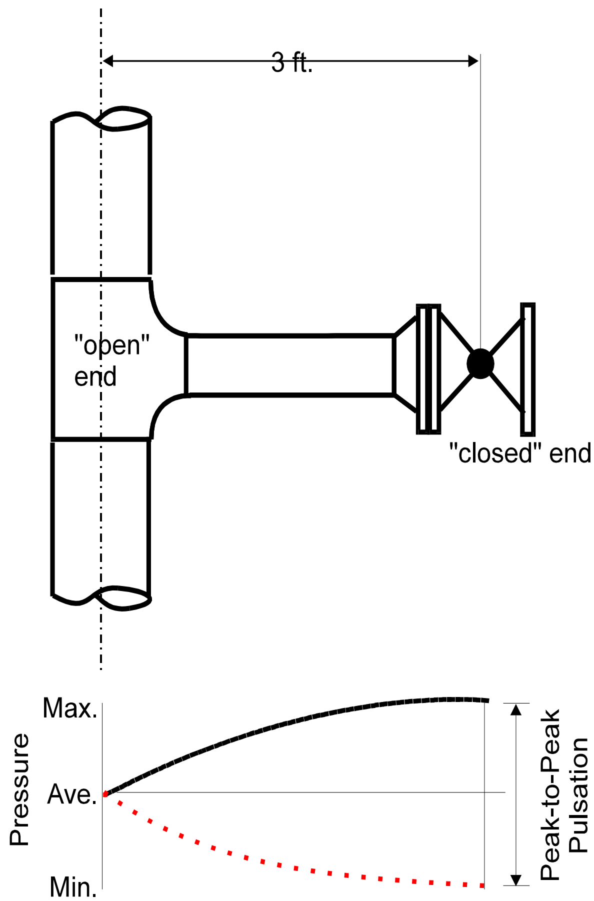
- The addition of reflected pulses to travelling pulses occurs at the acoustical natural frequency of the system. (Case 1)

Two waves travelling at the same frequency in opposite directions will form a standing wave.

Acoustical natural frequencies of a given piping section will occur at fixed frequencies. Acoustical natural frequencies will be excited (or become resonant) when the compressor is run at a speed which corresponds to a natural frequency. Figure 2.10 is an example of acoustical resonance in the bypass line:

An acoustical natural frequency, or resonance, is created when a dead leg is present in the piping. A good example of a dead leg is a closed bypass line. This 'system' has one open end and one closed end. At resonance, a pulsation minimum will develop at the open end and a pulsation maximum will develop at the closed end. This produces a quarter wave resonance. For the example given, the quarter wave acoustical natural frequency is 100 Hz.

Acoustical Resonance in a Bypass Line



Assume speed of sound (C) = 1200 ft/sec.

Since the bypass line is 3 ft., we look for a wave length of 12 ft., so that the quarter wave length is 3 ft.

From C = f x λ (Speed of sound = frequency X wave length), a 12 ft. wave length occurs when f = 100 Hz.

$$f = \frac{C}{\lambda} = \frac{1200 \text{ ft/sec}}{12 \text{ ft}}$$

$$= \frac{100 \text{ cycles}}{\text{sec}} = 100 \text{ Hz}$$

Pulsation maximums at the closed valve will occur whenever a multiple of run speed = 100 Hz. For a compressor operating over a speed range of 600 to 1000 RPM, this quarter wave resonance will be excited when the compressor is running at:

- 1000 RPM by a 6X input
- 857 RPM by a 7X input
- 750 RPM by an 8X input
- 666 RPM by a 9X input
- 600 RPM by a 10X input.

Figure 2.10 Acoustical Resonance in a Bypass Line

2.6 Unbalanced Forces

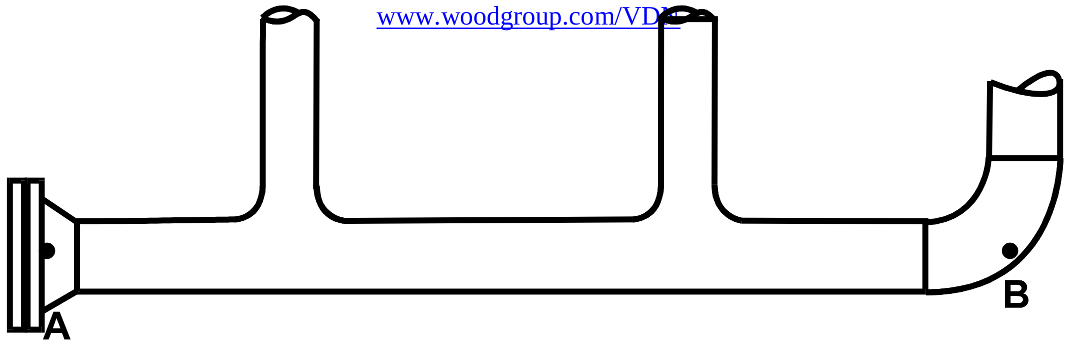
It is important to realize that pulsations as such do not cause vibrations. Rather, the potential for vibration is directly related to how the pulsations couple with the geometry of the compressor and its attached piping.

Once a compressor has been operating at a given set of operating conditions for a short period of time the interaction of the travelling pulses and reflected pulses causes standing wave patterns. The standing wave pulsation patterns couple with piping geometry to create unbalanced forces. Unbalanced forces are best explained with an example; see Figure 2.11.

Case 1: The header pressure at "A" is at the highest value and at "B" the pressure is at the lowest value. At the instant in time when the pressure at "A" is highest and "B" is lowest, the header is being pushed left at "A" with a greater force than it is being pushed right at "B". At the next instant in time the highest pressure will be at "B" and the lowest at "A". The net result is that the header has an unbalanced force pushing to the left, then to the right. In this case the force is equal to the peak-to-peak pulsation times the header area.

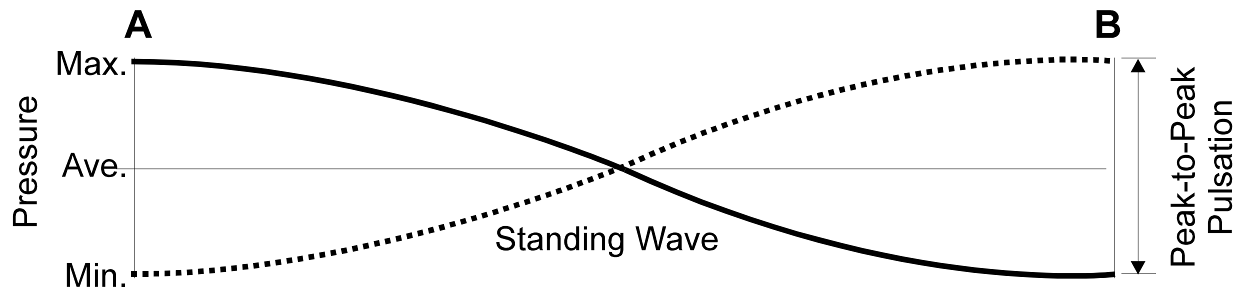
Case 2: At one instant in time the header pressures at "A" and "B" are both at the highest value. At the next instant in time the pressures at "A" and "B" are at the lowest level. The net result is that at any instant in time the header has equal forces acting at "A" and "B", therefore the unbalanced forces in the header are zero.

Unbalanced forces can occur between ends of a header, between bottle ends, between the cylinder and bottle, or between any two discontinuities in a piping system. See Figure 2.12.

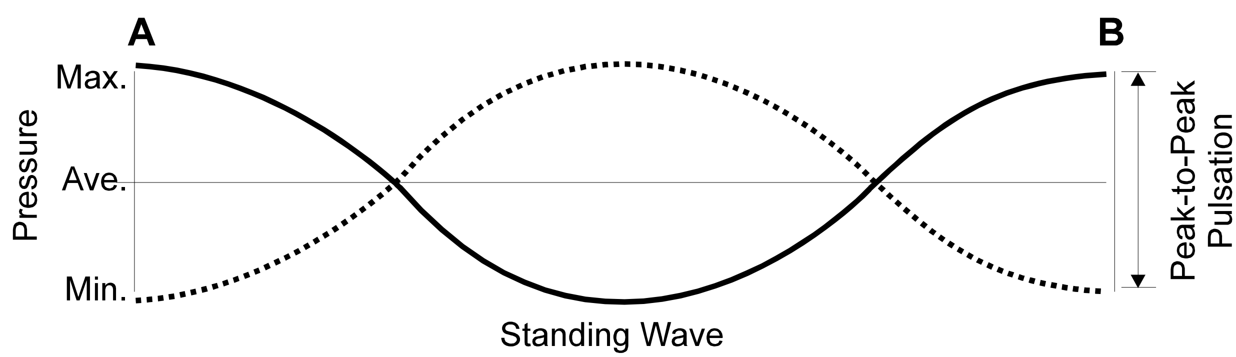


Force at end = Pressure at that instant at that end times Header Area

Net force on header = Force at A minus Force at B



Case 1
$\text{Force} = (P_{\text{at A}}\ \text{X}\ \text{area}_{\text{at A}}) - (P_{\text{at B}}\ \text{X}\ \text{area}_{\text{at B}})$



Case 2
$\text{Force} = (P_{\text{at A}}\ \text{X}\ \text{area}_{\text{at A}}) - (P_{\text{at B}}\ \text{X}\ \text{area}_{\text{at B}}) = 0$

2.11 Header Unbalanced Forces

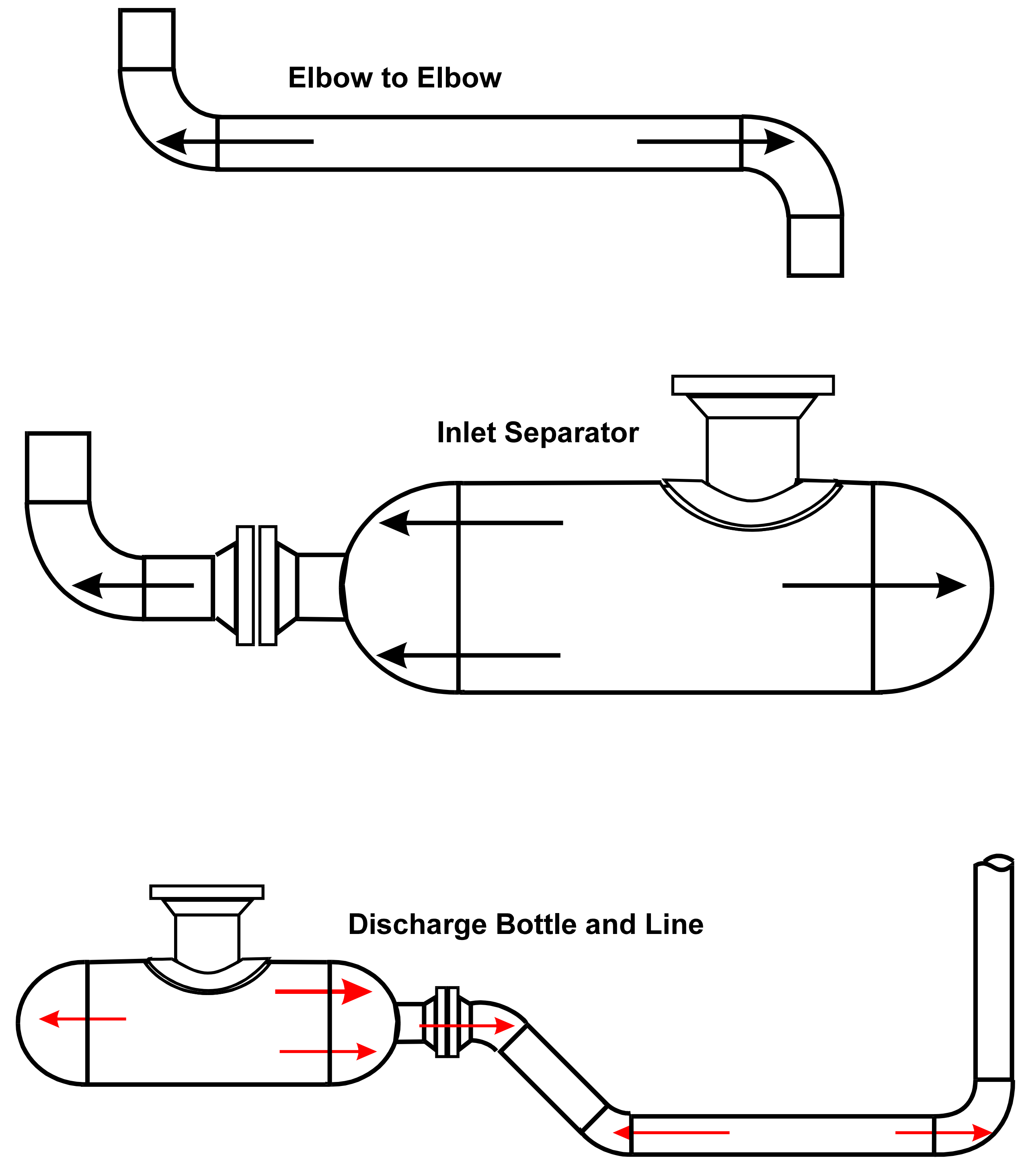


Figure 2.12 Unbalanced Force Examples

2.7 Pulsation Effects on Pressure Drop

Pressure drop occurs whenever gas flows through piping. The steady state flow pressure drop is proportional to the square of the speed of the gas in the pipe. Gas flowing at 60 ft/sec will have 4 times as much steady state pressure drop as gas that is flowing at 30 ft/sec.

When pulsations are present in the gas, the flow of gas is no longer "steady". The actual pressure drop will be a combination of the steady or "static" pressure drop and the unsteady or "dynamic" pressure drop.

Dynamic pressure drop is the pressure drop caused by the pulsating, or dynamic, flow component. As the pressure fluctuates, or pulsates, the velocity of the gas molecules also fluctuates. The gas molecules moving faster and slower than the mean flow of the gas is referred to as the dynamic flow component.

Since pressure drop is proportional to the square of the velocity, the increase in pressure drop at a velocity maximum will be significantly greater than the decrease in pressure drop at a velocity minimum. For this reason, both the static and dynamic pressure drops must be considered at areas where the resistance, or loss factor, is high (eg. choke tubes, orifice plates, nozzle entrances and exits).

2.8 Additional Effects of Pulsation

Excessive pulsations can directly affect compressor performance and indirectly have an adverse effect on performance. Uncontrolled pulsation will, in the long run, cost money in the repair of worn or failed parts, shutdown time, and inefficient operation. Some of the effects are:

- lower suction volumetric efficiency leading to decreased capacity,
- higher losses resulting in increased brake horsepower/mmscfd,
- increased dynamic pressure drop,
- pulsation induced valve vibration leading to possible failure,
- increased rod loading possibly causing rod failures or torsional vibration,
- acoustic resonance coincident with the torsional mechanical natural frequency possibly causing rod failure or crankshaft failure.