

# Acoustical Modeling of Reciprocating Compressors With Stepless Valve Unloaders

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## ABSTRACT

Capacity control of reciprocating compressors can be accomplished by a variety of techniques. Some common techniques include changing compressor speed, adjusting compressor cylinder head-end variable volume pockets, recycling flow, and changing compressor cylinder operation from double acting to single acting. These compressor loading techniques are well understood from an acoustical simulation perspective. One shortcoming with these compressor control techniques is they do not offer a continuous range of capacity control with minimal wasted energy. Stepless valve unloaders are one solution to address this limitation. Acoustical simulation of reciprocating compressor using stepless valve unloaders offers several challenges.

This paper discusses factors that must be considered to conduct acoustical simulations of systems with stepless valve unloaders. Results of an acoustic simulation are compared for a system with and without stepless valve unloaders.

## Introduction

There are many techniques used for capacity control on reciprocating compressors. One technique is a stepless valve unloader. A significant advantage of the stepless unloader over conventional recycle systems is that the energy wasted in recycling the gas is minimized. The flow is recycled locally in the cylinder and only the required gas capacity is compressed, thereby minimizing the compressor horsepower required. The paper will discuss issues involving the acoustical simulation of compressor systems that use stepless unloaders. First, some background about acoustical simulations.

## Background

Simulations of the gas flow in reciprocating compressor installations are conducted to determine the dynamic pressure and flow fluctuations that result from the periodic flow from the compressor cylinder. These simulations, first done in the 1950s using analog computers, are

now commonly done as part of many reciprocating compressor installations. These simulations are often called analog studies, but a more accurate description is an acoustical study. The simulation is called an acoustical simulation since the techniques used to calculate the system characteristics use linear acoustical theory. The pressure fluctuations in the system are assumed to be small compared to the mean pressure, so the governing equations can be simplified to a linear acoustical model.

A typical acoustical simulation requires definition of two fundamental characteristics of the system. First, the acoustical properties of the gas in the piping and vessels connected to the compressor cylinder must be determined. Second, the amplitude of the excitation, or energy that is put into the gas from the compressor cylinder, must be calculated. The combination of system acoustical properties and the source excitation allows for calculation of the dynamic pressure (or pressure pulsations) throughout the system.

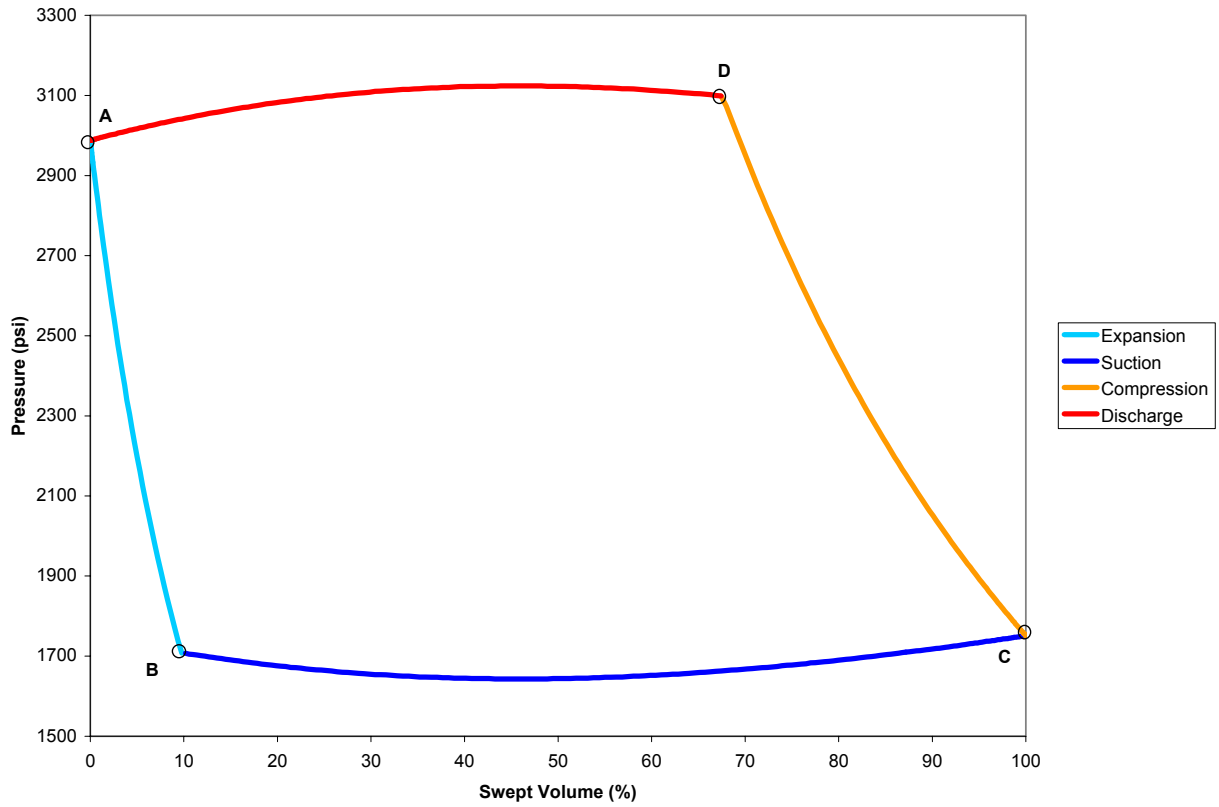
Calculation of the acoustical properties of the system involves collection of information such as the pipe and vessel dimensional information and their termination conditions, gas composition, and operating parameters such as pressures and temperatures. The acoustical resistance, capacitance and inductance of the components in the system are calculated based upon this information.

The excitation from the compressor cylinder is the periodic forced flow of gas in or out of the system by the compressor cylinder piston when the compressor cylinder valve(s) are open. The operation of the compressor cylinder valves is dictated by the compressor geometry, valve properties (spring stiffness, valve plate mass), and the operating conditions (pressures, temperatures, and gas composition). A stepless unloader controls the gas flow into the system by controlling the closing of the suction valve. An understanding of compressor cylinder performance is key to accurate acoustical simulation of the system.

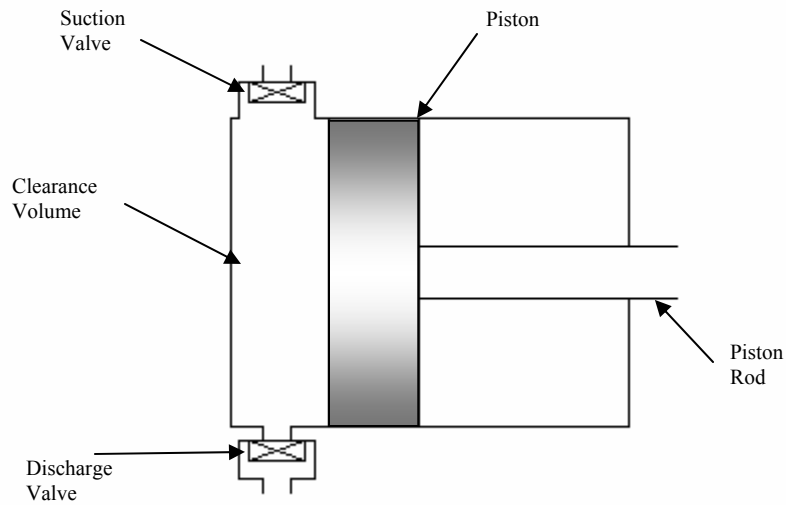
### Compressor Cylinder Performance

A common method of illustrating reciprocating compressor cylinder performance is the pressure in the cylinder clearance volume as a function of the piston swept volume. This format is called a pressure-volume curve, or P-V curve. A simplified pressure-volume curve for the head end of a compressor cylinder is shown in Figure 1. Figure 2 shows a schematic of a typical compressor cylinder.

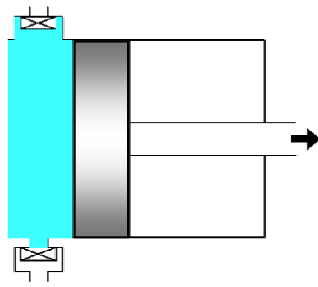
The piston starts at top dead center at point A and moves towards bottom dead center; an expansion process. The suction and discharge valves remain closed and the gas in the clearance volume expands until point B, where the suction valve opens. Gas then enters the clearance volume until the piston reaches bottom dead center and the suction valve closes point C; a suction process. The piston then moves back towards top dead center, compressing the gas in the clearance volume; a compression process. At point D, the discharge valve opens and gas in the clearance volume is forced into the discharge system; a discharge process. Figure 3a to 3d illustrates these 4 steps in the compressor cylinder operation.



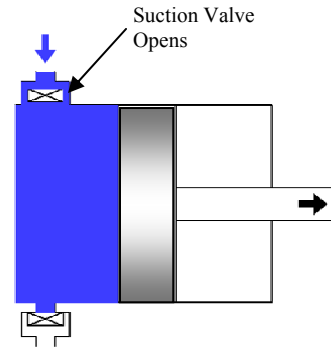
**Figure 1: Typical P-V Curve**



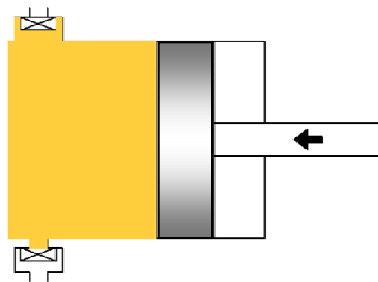
**Figure 2: Compressor Cylinder Schematic**



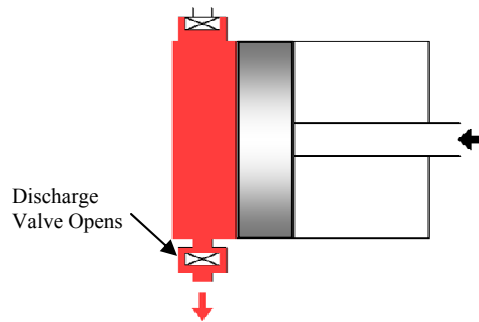
**Figure 3a: Expansion**



**Figure 3b: Suction**



**Figure 3c: Compression**



**Figure 3d: Discharge**

The compressor cylinder operation differs with a stepless unloader. Figure 4 shows the modified head end P-V curve with the stepless unloader. The operation differs, in that the suction valve is held open by the unloader past bottom dead center when the suction valve would normally close, point C to C'. Gas that flowed into the clearance volume during the suction part of the stroke flows back out into the suction system while the valve is held open. Figure 5 is a schematic of the compressor cylinder showing the back flow. The stepless unloader allows the suction valve to close at a prescribed position depending on the capacity requirements of the system. Compression and discharge steps of the cycles remain the same as described earlier.

### **Effects of a Stepless Unloader on the Acoustical Simulation**

The operation of the stepless unloader has a significant effect on the acoustical response of the system. Several factors must be accounted for if the acoustical simulation of the piping system is to be representative of the real system.

#### Effect of Capacity

The stepless unloader changes the capacity or mean gas flow velocity, in the system due to local recycling of gas at the suction valve. The acoustical properties of the system are dependent on the mean gas flow velocity. Therefore, accurate simulation of the compressor capacity is important. There are several modeling techniques whereby the compressor cylinder

characteristics could be adjusted so that the calculated capacity matches the capacity for the stepless unloader performance: for example, the compressor piston bore or cylinder clearance volume could be adjusted. However, adjusting the compressor cylinder characteristics in a model to match capacity introduces significant errors in the acoustical analysis, since these characteristics not only affect the compressor capacity, but also the source excitation that the piping system experiences.

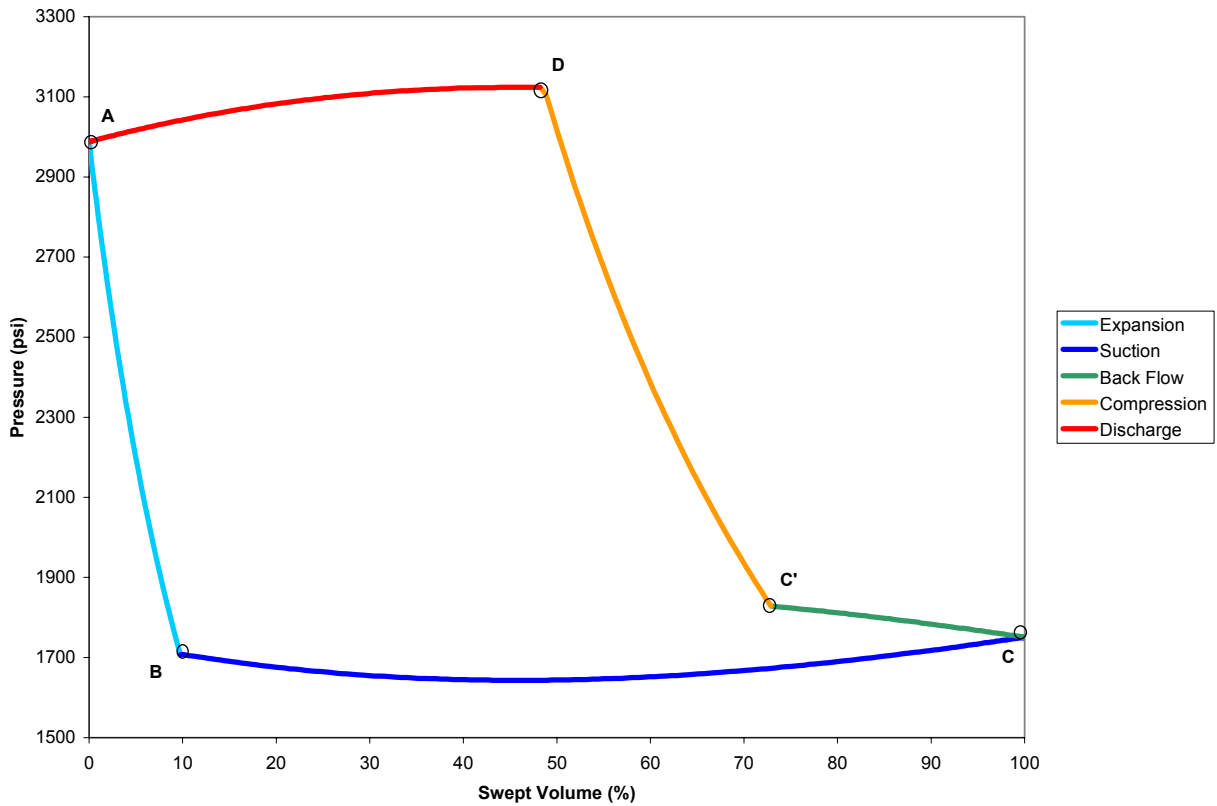


Figure 4: Stepless Unloader P-V Curve

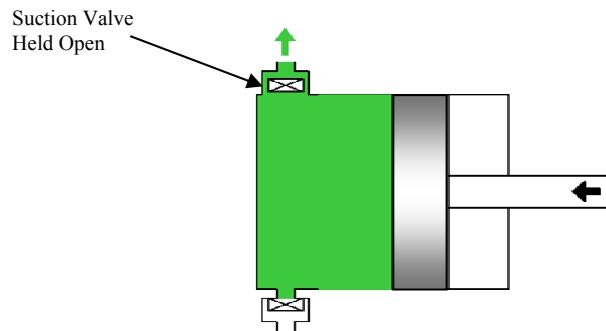
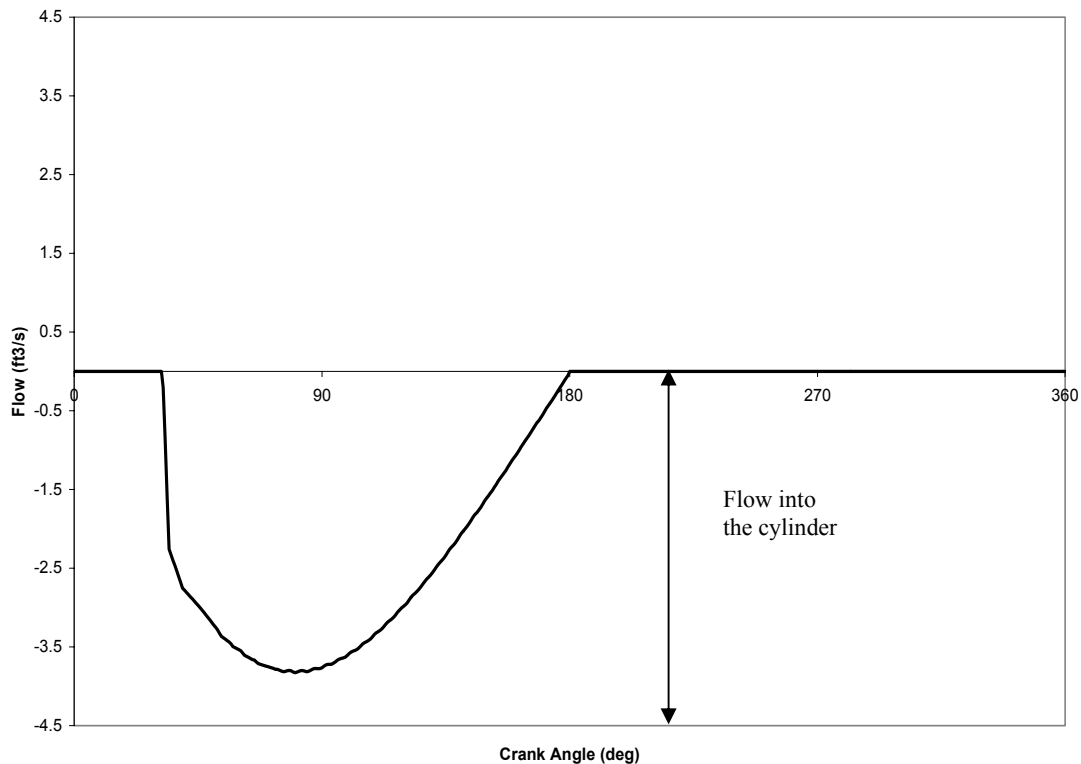


Figure 5: Back Flow

## Effect of Back (or Reverse) Flow

The source excitation for the acoustical analysis is the pulse of gas that moves in or out of the compressor cylinder clearance volume when the compressor cylinder valves open and close. Figure 6 is a plot of the ideal head end flow versus crank angle for the P-V curve shown in Figure 1. The suction valve opens some time after top dead center (crank angle =  $0^\circ$ ), reaches a maximum instantaneous flow, which then drops to zero at bottom dead center when the valve closes (crank angle =  $180^\circ$ ). Figure 7 illustrates the flow versus crank angle characteristics when a stepless unloader is used. The first half of the cycle is similar to Figure 6. The second half of the cycle shows the back flow from the clearance volume out to the suction system. It is obvious from inspection that the two curves are different and therefore the source excitation in the acoustical model will be different.

Presentation of the source excitation in a spectrum format, rather than the time domain, is a simpler technique to see the differences in the source excitation. Figure 8 is a plot of the spectrum for the flow versus crank angle for a single acting compressor cylinder with a stepless unloader active (cross-hatch bars). Figure 8 also shows the source excitation spectrum for a compressor cylinder where the clearance has been adjusted to make the capacity similar to the capacity when the stepless unloader is active (solid bars). There is a significant difference between the source excitation at the fundamental frequency and some of the higher orders.



**Figure 6: Gas flowing into the clearance volume during the suction event (stepless unloader inactive)**

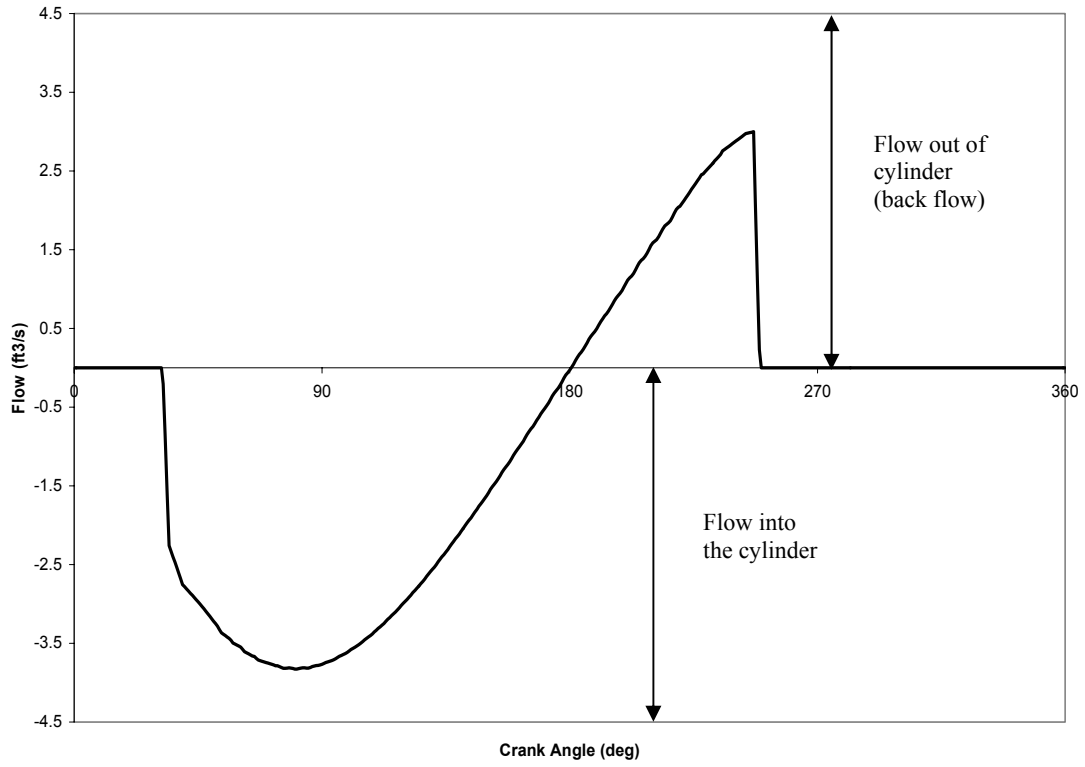


Figure 7: Gas flow for the suction and back flow with the stepless unloader active

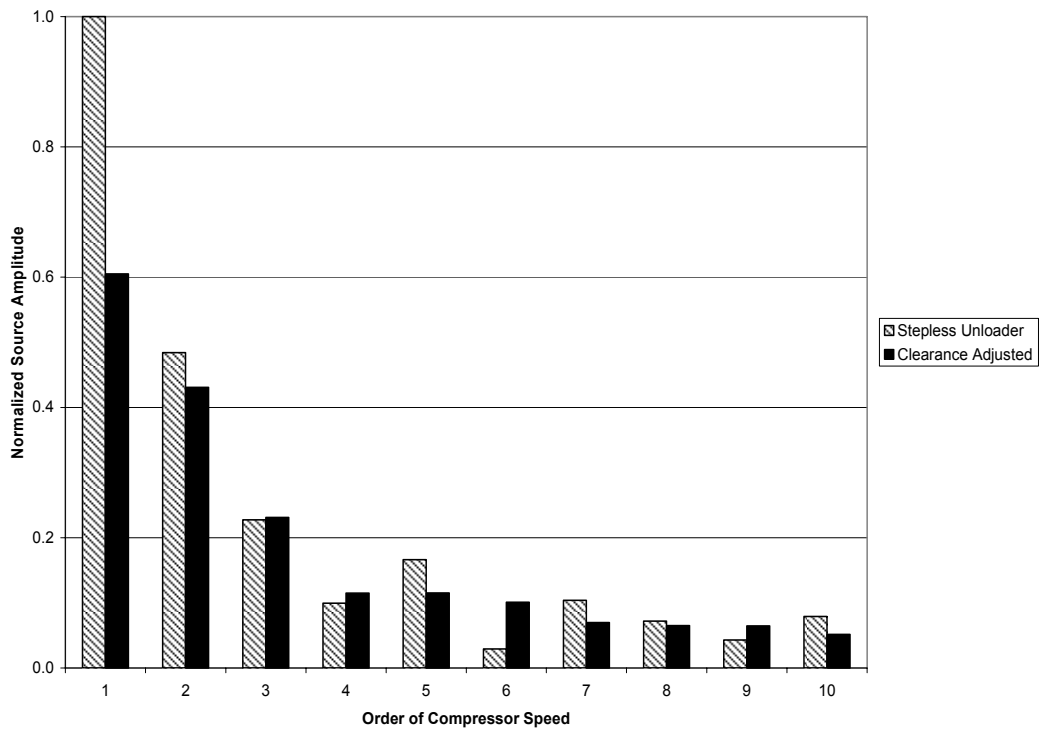


Figure 8: Spectrum of Suction Flow

## Effect of Valve Position and Clearance Volume

All computer simulations require definition of boundary conditions, that is, the properties that define the end points of the model. The acoustical model of a reciprocating compressor system requires a boundary condition at the compressor valves. The change from a closed valve to an open valve at the model boundary must be included in the model, since the acoustical properties of a closed valve are significantly different from an open valve. Also, the effect of the clearance volume and pressure drop through the compressor valve when the valve is open must be included. The effect of the changing boundary condition at the valves is of increasing significance as cylinder clearance increases and cylinder valve pressure drop decreases.

## Effect on Discharge System Acoustical Response

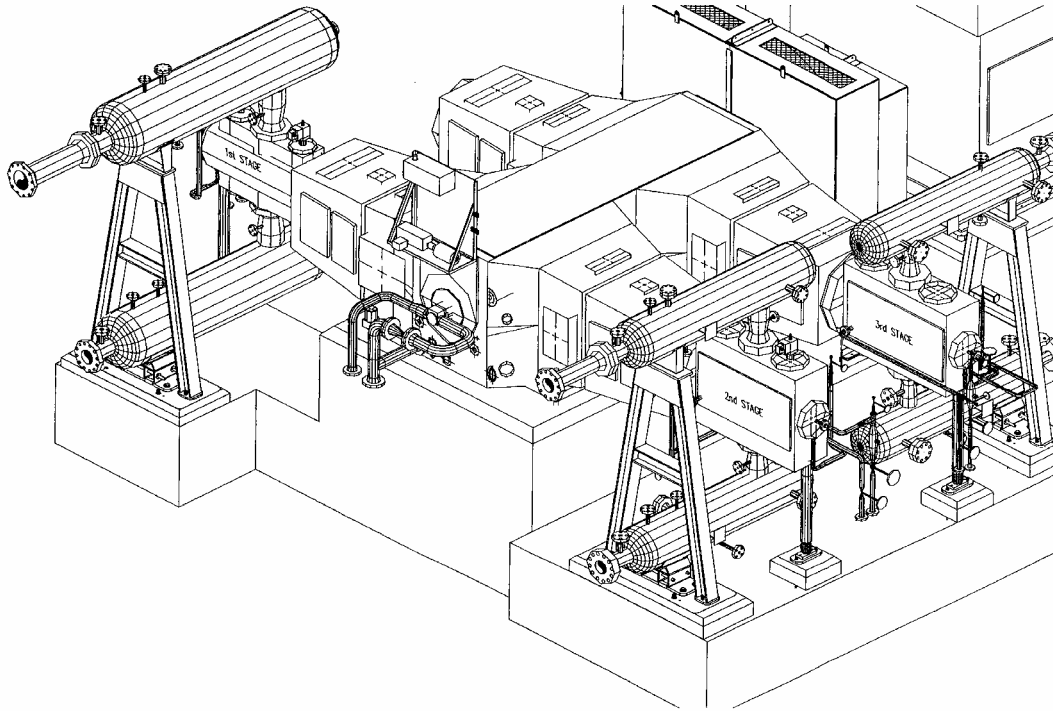
The normal operation of the stepless unloader functions by delaying the closing of the suction valve. The delayed closing of the suction valve prevents the discharge valve from opening at the position it would normally open if the stepless unloader was not installed. Therefore, the source excitation into the discharge system is affected by the suction stepless unloader and accurate simulation of the discharge system is dependent on accurate calculation of the P-V curve. The key difference relative to the suction system is that no reverse flow occurs on the discharge.

## **Case Study**

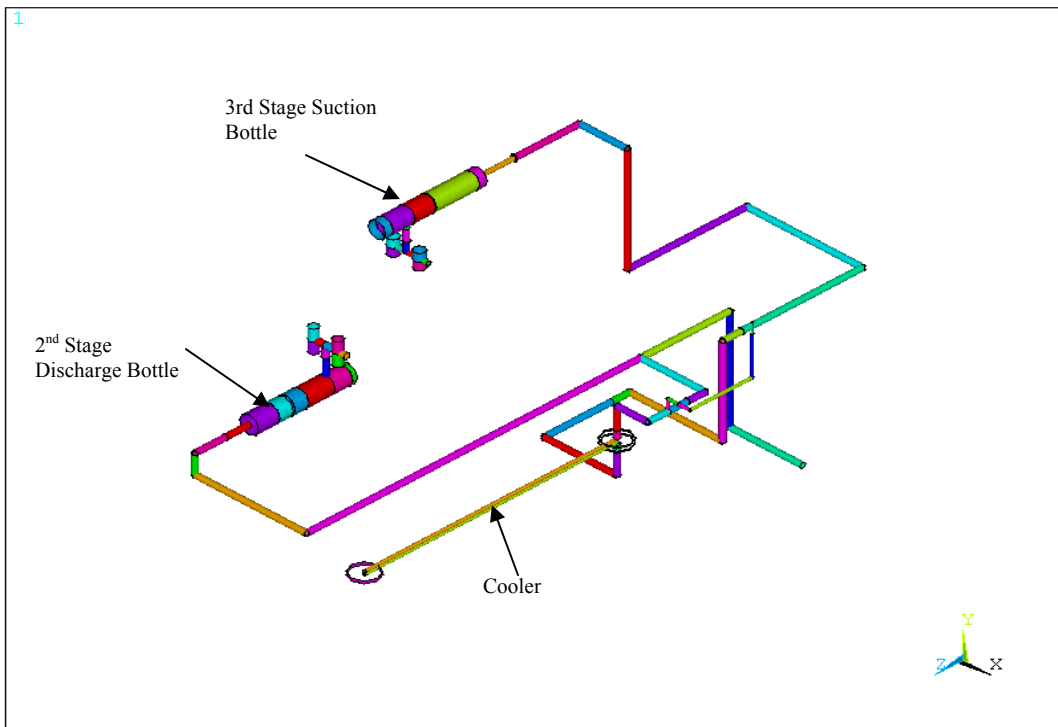
This case study includes an evaluation of a system that includes a 3 stage, 3 throw separable compressor rated at 3200 HP and 327 rpm. The compressor operates in an ethylene service with a suction pressure of 450 psig and discharge pressure of 3000 psig. The compressor includes stepless unloaders on all 3 stages. An isometric general arrangement drawing of the compressor and pulsation bottles is shown in Figure 9.

An acoustical simulation of the second interstage system is presented here. The second interstage system includes all piping and vessels from the discharge of the second stage cylinder to the suction of the third stage cylinder, including the interstage cooler. A plot of the acoustical model of the second interstage system is shown in Figure 10. The third stage cylinder includes a stepless unloader on the head end only.

Pressure pulsations were measured in the third stage suction bottle outlet nozzle with the stepless unloader active and inactive. The compressor capacity is reduced from approximately 25 mmscfd to 20 mmscfd when the stepless unloader was fully activated. Plots of calculated and measured pressure pulsations with the stepless unloader inactive and active are shown in Figures 11 and 12. The pressure pulsations are calculated by the acoustical simulation for a speed range  $\pm 10\%$  of the actual speed range, a typical technique used in acoustical analysis. This speed range is shown on the plots by the band around each order of compressor speed. The compressor is driven by a fixed speed motor. Therefore, the measured pressure pulsation is a single value at each order of compressor speed which is displayed as a sharp spike on the plots.



**Figure 9: Compressor General Arrangement**



**Figure 10: Second Interstage Acoustical Model**

The calculated pressure pulsation amplitudes agree very well with the measured pressure pulsations. It is also evident that there is a significant difference between the pressure pulsations' characteristics when the stepless unloader is active and inactive. The pressure pulsations at the compressor fundamental speed are much higher when the stepless unloader is active. Pressure pulsations at the fundamental speed are typically more difficult for the acoustical designer to control. Larger pulsation bottles and more complex bottle internals are often required to control pressure pulsations at the fundamental speed.

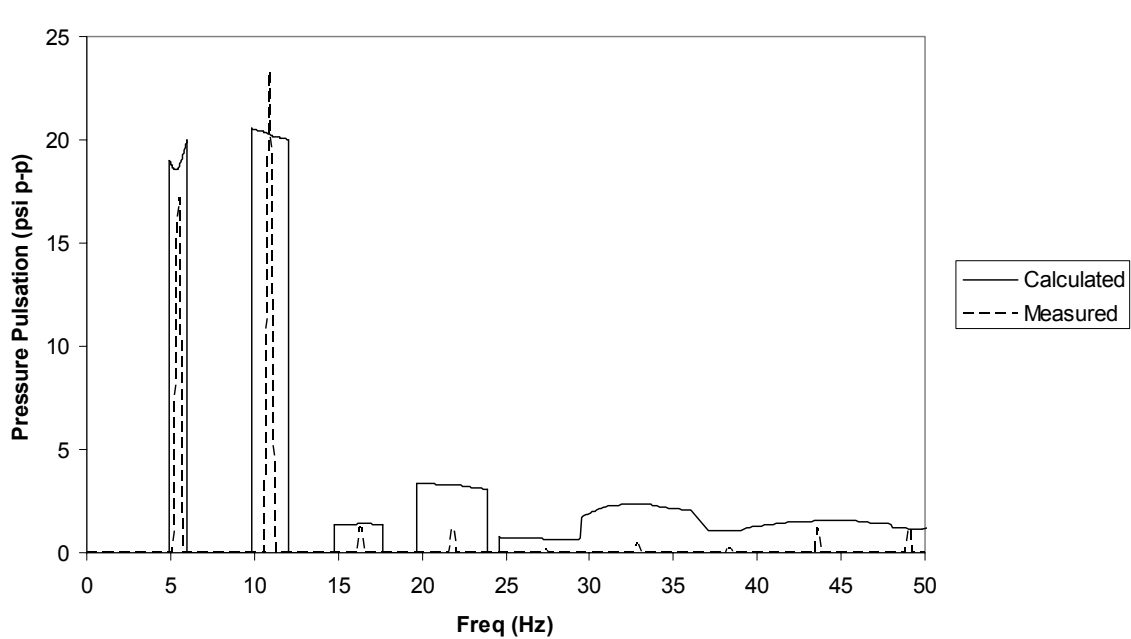


Figure 11: Measured and Calculated Pressure Pulsation with the Stepless Unloader Inactive

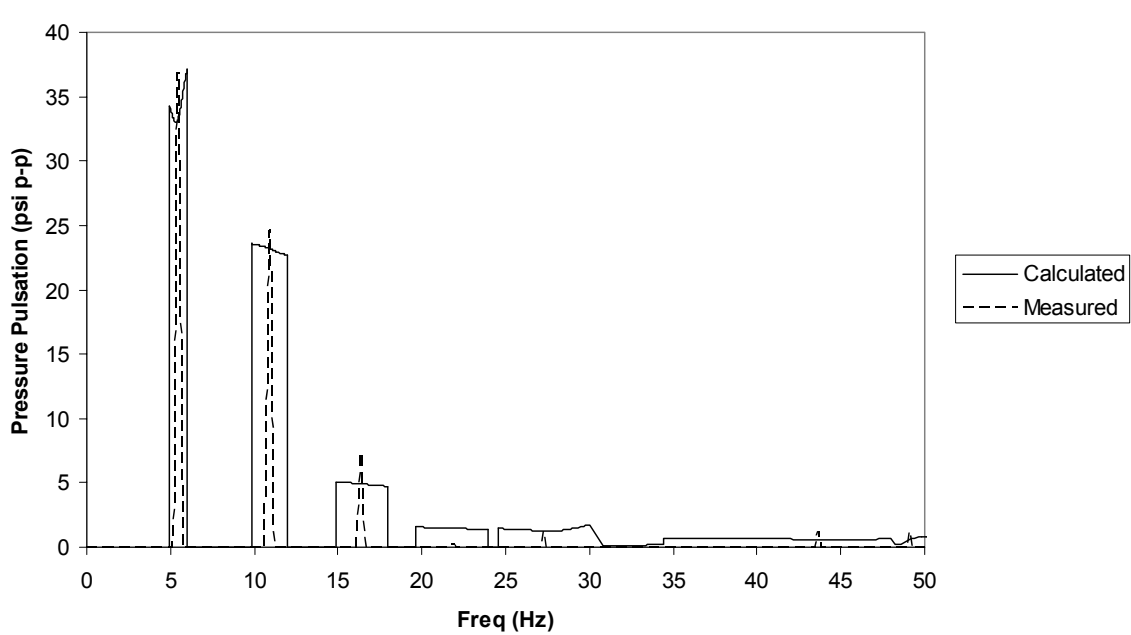


Figure 12: Measured and Calculated Pressure Pulsation with the Stepless Unloader Active

The comparison between the calculated and measured pressure pulsations shows some differences at orders 4 to 8. The pressure pulsations at these orders are much lower in amplitude than those at the fundamental and second order of compressor speed. If the system is designed to control the pressure pulsations at the fundamental and second order of compressor speed, the pressures pulsations at the higher orders are typically controlled to acceptable levels as well. Further investigation into the acoustical simulation and evaluation of the field testing procedure are required to determine the reason for the differences.

## **Conclusion**

Operation of stepless unloaders has a significant impact on the acoustical performance of the system. The acoustical response of the system can be calculated with accuracy suitable for engineering purposes when the simulation of compressor valve operation and the flow in and out of the clearance volume throughout the piston cycle is included.

## **Acknowledgements**

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## **Biographies**

### **Kelly Eberle, P.Eng. Principal Engineer**

Kelly graduated from the University of Saskatchewan with a Bachelor of Science in Mechanical Engineering in 1986 and joined Beta Machinery Analysis in 1988.

Kelly has performed design studies as well as field troubleshooting of reciprocating compressor and pump facilities. The scope of the design studies range from acoustical simulations, flexibility studies, structural analysis (static and dynamic), and foundation analysis.

Co-authored paper “Cylinder Stretch as a Source of Vibration in Reciprocating Compressor.”

Presently a Principal Engineer with Beta Machinery Analysis, Kelly works in their Calgary office.

### **Brian C. Howes, M.Sc., P.Eng. Chief Engineer**

Brian graduated from the University of Calgary with a Master of Science in Solid Mechanics. His thesis was entitled *Acoustical Pulsations in Reciprocating Compressor Systems*.

Brian has worked with Beta Machinery Analysis since 1972. In his present position as Chief Engineer for Beta, he has performed troubleshooting services all over the world.

Brian has many technical papers to his credit. The range of machinery problems they cover includes all manner of reciprocating and rotating machinery and piping systems, balancing and alignment of machines, finite element analysis, modelling of pressure pulsation, torsional vibration testing and modelling, flow-induced pulsation troubleshooting and design, pulp and paper equipment such as pulp refiners, etc. He has also worked on hundreds of reciprocating compressor installations.