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Reciprocating Compressor System Performance Affected by Dynamic Pressure Drop

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Executive Summary

Static and dynamic pressure drop are important factors in calculating reciprocating compressor system performance. Calculation of static pressure drop has been well understood for many years. Calculation of dynamic pressure drop resulting from flow fluctuations requires specialized pressure pulsation analysis. Dynamic pressure drop can be as large, or larger, than static pressure drop in some cases, so accurate calculation of the dynamic pressure drop is critical.

Technical gaps in some pressure pulsation simulation programs do not allow for calculation of dynamic pressure drop. Also, industry practice for compressor performance modeling is to make generic assumptions about pressure drop through the piping system. These factors can lead to large errors in the calculated compressor performance.

New simulation tools have been developed to address this technical gap in calculating dynamic pressure drop as well as including the true total pressure drop in compressor performance calculations. Accurate calculation of dynamic pressure drop means the compressor performance can be verified in the design stage, thus avoiding costly redesign, or retrofit, after installation. Also, these new simulation tools result in accurate compressor pulsation studies that allow for optimization of the compressor package design and ensure safe, long term operation.

This paper will describe the analytical tools used in accurately calculating the system static and dynamic pressure drop. The effect of dynamic pressure drop on the compressor performance will be demonstrated through case studies. A design approach will be recommended to ensure the compressor is designed appropriately for the application.

1. Introduction

Performance simulation of reciprocating compressors typically includes calculation of parameters such as cylinder flow rate (capacity), power consumption, rod loads, pin load reversal, discharge temperature and many others. This paper focuses on the discussion of compressor capacity and power consumption.

The compressor capacity and power consumption are dependent on, and very sensitive to, the inlet and outlet pressures for a given compressor cylinder geometry. The effect of pressure drop (static and dynamic) on the cylinder inlet and outlet pressure will be discussed in detail in this paper.

1.1 Compressor Performance

1.1.1 Introduction to P-V Curve

Calculating the compressor cylinder capacity and power is done by simulating the thermodynamic process of compressing the gas inside the compressor cylinder from suction pressure to discharge pressure. The thermodynamic process inside the compressor cylinder involves four events:

- Suction Event: where gas flows into the cylinder through the open suction valves,
- Compression Event: where the gas is compressed to discharge pressure,
- Discharge Event: where gas flows out of the cylinder through the discharge valves, and
- Expansion Event: where the remaining gas inside the cylinder expands back to suction pressure.

A common method of illustrating this process is the pressure-volume, or P-V, curve. Figure 1 shows an idealized P-V curve. The curve represents the pressure inside the compressor cylinder clearance volume as the piston moves through each cycle. The section of the curve from B-C and D-A indirectly represent the time when the valves are open (suction and discharge respectively).

The dashed horizontal lines indicate the suction and discharge line pressures when the suction and discharge valves close. The curved lines between the suction and discharge pressure (A-B and C-D) are the expansion and compression lines that are defined by gas properties for a given cylinder geometry.

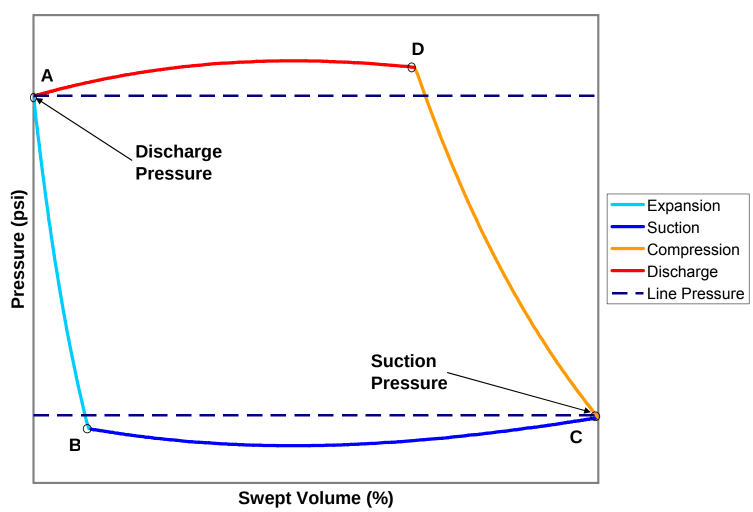

Figure 1: Typical Compressor Cylinder P-V Curve

1.1.2 Compressor Cylinder Capacity and Required Power

The compressor capacity is determined by the amount of gas that flows into the cylinder during the time when the suction valve is open. This capacity is graphically illustrated on the P-V curve as the length between points B and C. The distance from B-C divided by the total swept volume is referred to as the volumetric efficiency (VE). The VE is commonly used in analysis of compressor performance as it is a normalized quantity that gives a direct indication of the capacity (VE is proportional to the capacity).

Consider the example of a compressor cylinder that operates from 705 psia to 1030 psia. The VE is calculated to be 81.5% and the corresponding capacity is 28.2 MMscfd for a given compressor geometry and gas composition. If the suction pressure changes from 705 psia to 805 psia, the VE increases to 86.7% and the capacity increases to 34.7 MMscfd. Figure 2 shows the corresponding P-V curve for these two cases. This example shows that a pressure change can have a significant impact on compressor capacity. Consequently, the static pressure drop and dynamic pressure drop must be considered to determine the actual capacity.

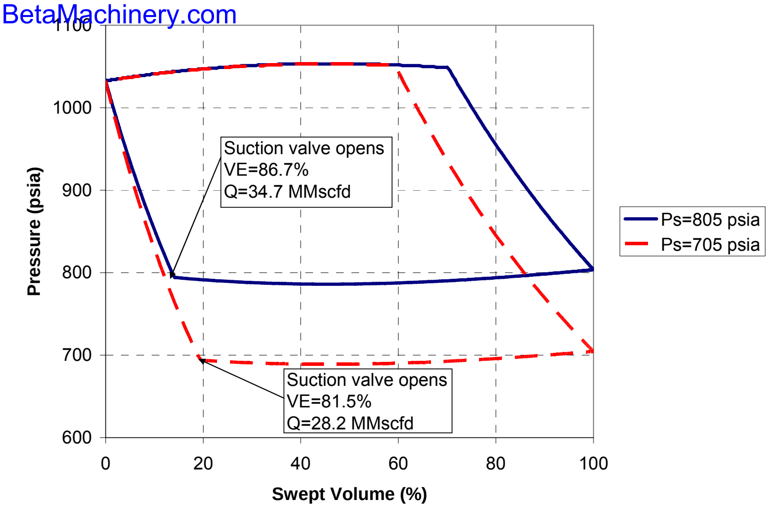

Figure 2: Capacity for Two Different Suction Pressures

The P-V curve is also a valuable tool to illustrate and calculate the power required to compress the gas from suction to discharge pressure. The area enclosed by the P-V curve is called the Indicated Power . Figure 3 shows the same P-V curves as the previous example with the changing suction pressure and all other parameters being constant. The area enclosed by the P-V curve is shown by the hatched regions. The area or Indicated Power for the lower suction pressure is shown to be about 25% greater than the case with the higher suction pressure. Therefore this example shows that changing the suction pressure has a significant impact on compressor power.

Similarly, changes to the discharge pressure can have a significant impact on the compressor performance.

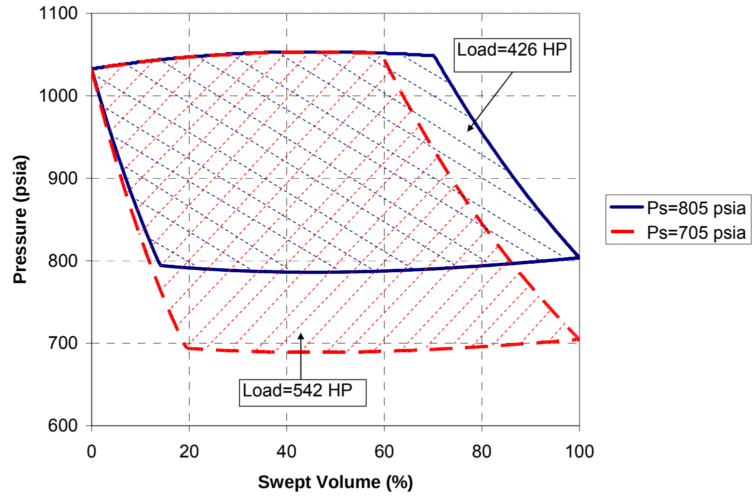

Figure 3: Power for Two Different Suction Pressures

1.1.3 Impact of Pressure Drop on Compressor Performance

The preceding discussion shows that an understanding of suction and discharge pressure is key to accurate calculation of the compressor performance. Typically the suction pressure and discharge pressure at the compressor skid edge is defined by the process or field requirements. There will be pressure loss or pressure drop between the compressor skid edge connection and the compressor cylinders as well as pressure drop between the stages for multi-stage compressors due to piping, vessels, and cooler (as discussed in Section 1.2 below). Understanding the pressure drop through each component is key to accurately calculating the compressor capacity and power requirements.

1.2 Static, Dynamic and Total Pressure Drop

Pressure drop is the pressure loss due to resistance to the flow. The resistance can come from a number of sources such as friction loss in the piping, change of pipe cross section, a tee, a cooler tube, an orifice plate or a choke tube. Pressure drop can be defined locally or on a system basis.

1.2.1 Local Pressure Drop

The local pressure drop is the pressure drop caused by a single component. The local pressure drop, ΔP at a location is defined by

$$\Delta P(t) = 1/2 \cdot K \cdot \rho \cdot V^2(t) \qquad (1.1)$$

Where

K is the resistance factor dependent on the local geometry,
ρ is the gas density,
V is the gas velocity,
t is the time.

When ΔP and V are periodic functions of t, as happens in a reciprocating compressor system, it follows from Eq (1.1) that

$$\Delta P_0 = 1/2 \cdot K \cdot \rho\, V_0^2 + 1/2 \cdot K \cdot \rho\, (1/2 \Sigma V_i^2) \qquad (1.2)$$

where ΔP_0 and V_0 are the mean pressure drop and mean velocity, respectively, and V_i (i=1,2,3,…) are the amplitudes of velocity harmonics representing dynamic flow. The first and second terms in the right hand side of Eq (1.2) are called the Static and Dynamic Pressure Drop respectively, while the sum of the two terms is called the total pressure drop. API 618 5th Edition uses the terminology "steady flow" and "total flow". "Steady flow" is the mean flow and "total flow" is the sum of the mean flow and periodic or dynamic flow.

1.2.2 System Pressure Drop

A "system" consists of a collection of piping, vessels and pulsation control devices. Static and dynamic pressure drop occur through all parts of a reciprocating compressor installation and not just the pulsation control devices. The system for packaged units such as that shown in Figure 4 is typically defined from the suction skid edge flange connection to the final discharge skid edge flange connection. There is static and dynamic pressure drop due to flow through a pipe, flow around elbows and tees, through vessels such as scrubbers and separators and other components like air coolers and shell-tube heat exchangers.

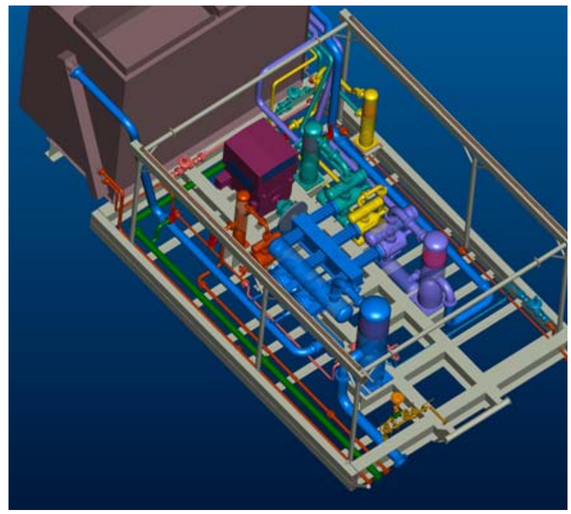

Figure 4: System pressure drop, includes losses through piping, vessels, cooler, compressor, pulsation control devices.

The system static pressure drop is defined as the difference of mean pressures between any two locations in the system when flow is steady, namely, $V_i = 0$. The total pressure drop is defined by the difference of mean pressures between any two locations in the system when the flow has both steady and dynamic components. The dynamic pressure drop is defined by the difference of the total and static pressure drops.

1.3. Example of Pressure Drop across an Orifice Plate

The pressure drop across an orifice plate in a flange set, Figure 5, will be used to illustrate these concepts more clearly. The pressure drop for an orifice plate can be easily calculated for the case where the gas velocity is steady or constant. However, in reciprocating compressor systems the gas velocity varies over the time period of one revolution of the crankshaft. The previous P-V curve shows that the suction valve is open for only a portion of the piston cycle pulling gas into the cylinder and similarly the discharge valve is open for a small portion of the cycle pushing gas out. This unsteady flow from the cylinder in combination with the acoustical characteristics of the piping and vessels results in a more complicated gas flow ($V_i \neq 0$)

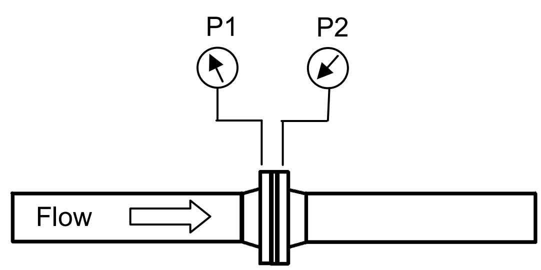

Figure 5: Orifice Plate in a Flange Set

The gas from the compressor flowing in the piping and through the orifice has a steady or mean component V_0 that results in a steady or static pressure drop as shown in Figure 6a and 6b.

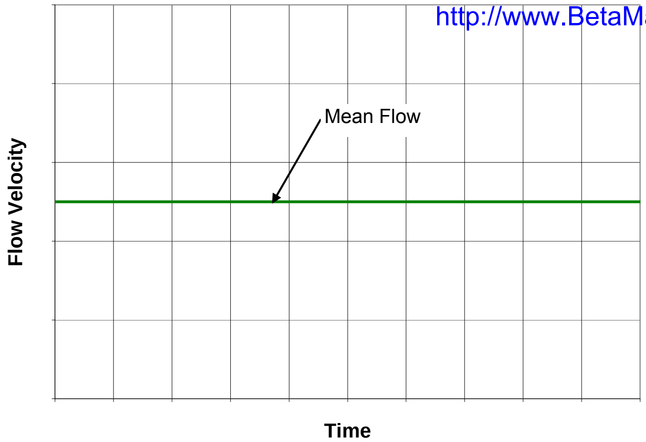

Figure 6a: Flow vs. Time with no Dynamic Flow

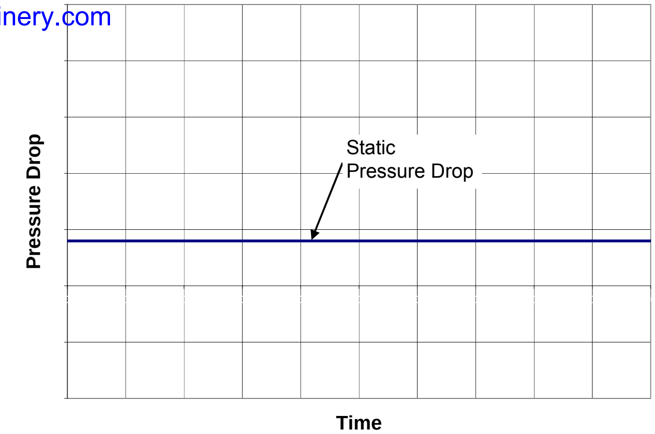

Figure 6b: Pressure Drop vs. Time with no Dynamic Flow

The gas from the compressor also has a dynamic component V_i . The dynamic flow is shown in simple terms as a sinusoidal shape in Figure 7a. Equation 1.2 says the pressure drop is a square function of the flow velocity. This relationship causes the instantaneous sinusoidal flow velocity to be distorted with the top of the curve getting sharper and the bottom of the curve getting flatter as shown in Figure 7b. The instantaneous pressure drop between 2 points results in a mean or total pressure drop.

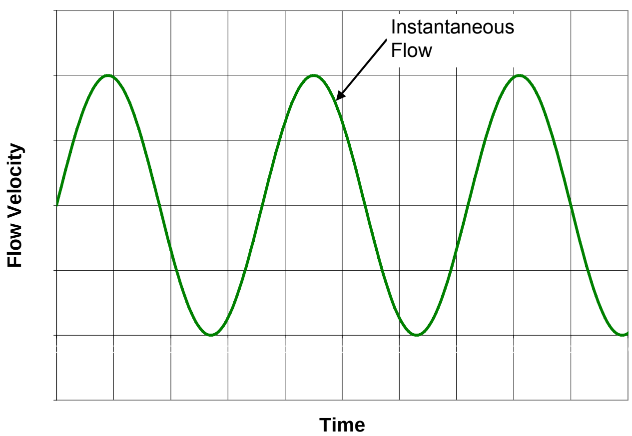

Figure 7a: Flow vs. Time with Dynamic Flow

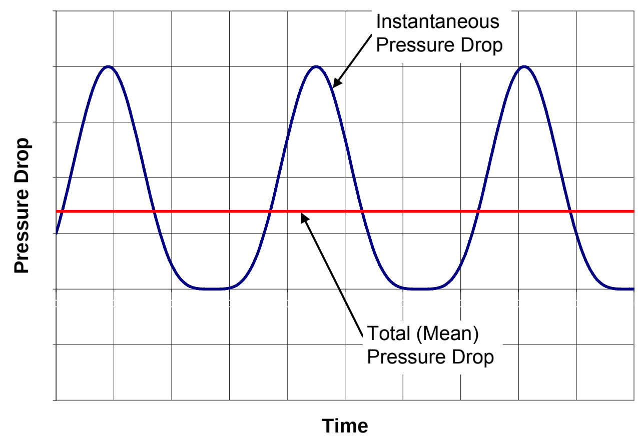

Figure 7b: Pressure Drop vs. Time with Dynamic Flow

The total pressure drop is higher than the static pressure drop due to dynamic flow components. The difference between the static pressure drop and total pressure drop is referred to as the dynamic pressure drop as illustrated in Figure 7c.

The magnitude of the dynamic pressure drop will depend upon the magnitude of the flow oscillations in the system. Dynamic pressure drop from dynamic flow has been shown to be equal to or greater than static pressure drop in many cases. Calculation of the compressor performance typically includes consideration for the static pressure drop but if the calculated performance does not include the effects of dynamic pressure drop, there can be significant differences between the calculated and actual performance.

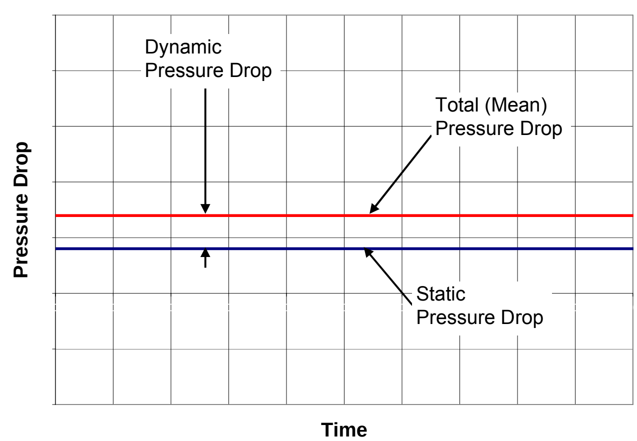

Figure 7c: Mean Static and Total Pressure Drop

Section 3 illustrates the system pressure drops and their effect to compressor capacity and power consumption through two case studies.

1.4 Pressure Drop Guidelines

It is commonly acknowledged that pressure oscillations (pulsations) are a critical factor that must be considered in reciprocating compressor installations. API 618 has included many specifications for the control of pressure pulsations as well as the effects of pressure pulsations.

API 618 5^{th} Edition also includes a guideline for the static and dynamic pressure drop. Section 7.9.4.2.5.3.1 states the maximum allowable pressure drop is to be less than 0.25% of the average absolute line pressure, or the value calculated by the following equation.

$$\Delta P = 1.67\ (R - 1)/R\ \% \qquad (1.3)$$

Where

ΔP is the maximum pressure drop based on steady flow through all pulsation suppression devices expressed as a percentage of mean absolute line pressure at the inlet of the device,
R is the stage compression ratio.

This pressure drop limit is to be based on the "steady flow," so this is the static pressure drop as discussed previously in this paper. The API 618 5^{th} Edition goes on to state the allowable pressure drop "shall be increased by a factor of two when the pressure drop is calculated using the total flow, where total flow is the sum of the steady flow plus dynamic flow components." This API limit shows that dynamic pressure drop has been recognized as an important factor.

The dynamic pressure drop must be calculated and compared to the API 618 limit when reciprocating compressor packages are designed, just as pressure pulsations and dynamic forces have been calculated and evaluated against previous versions of API 618. The dynamic pressure drop can only be calculated by acoustical simulation of the reciprocating compressor system. There are many different digital simulation tools available for calculation of the pressure pulsations and dynamic forces. However, accurately calculating dynamic pressure drop requires a specialized acoustical model that employs a proven time-domain solver (discussed further in section 2).

1.5 Compressor Performance Map

The concept of compressor performance mapping has been discussed in the past [1]. The basic concept is to simulate the compressor performance over the full range of operating pressure, temperatures, speeds, and cylinder loading to clearly understand the compressor operation and avoid operating points that may cause problems.

Compressor Performance Mapping must also include the effects of system pressure drop to properly quantify the compressor operation. The system pressure drop is dependent upon the static and dynamic pressure drop which in term is dependent on the mean and dynamic flow in the system as discussed previously. The change in the mean flow and static pressure drop is relatively easy to quantify for the different operating points. However the change in the dynamic flow and dynamic pressure drop requires that acoustical simulations be done for a large range of operating points. The effect of changing the compressor loading for single acting operation to double acting operation can have a significant impact on dynamic flow, dynamic pressure drop and the compressor performance.

2. Acoustical Simulation of Reciprocating Compressors

There have been many methods used for acoustical simulation of reciprocating compressor systems. Analog computers were used starting in the 1950's and found to be very effective for many applications. With the rise of digital computers, software was developed to model reciprocating compressor systems starting in the 1970's. The core of any simulation software is the mathematical model of the physical phenomena and its accuracy in representing the real world process. While there are several different mathematical models which can be used to simulate the acoustics of reciprocating compressors, each model has its own strengths and weaknesses.

2.1 Three-dimensional theory of fluid mechanics

The most complete theory for a model of acoustics can be traced back to the general theory of three dimensional fluid mechanics.

Let *p* be a point in any material body such as gas, liquid or solid. For simplicity of discussion, the direct notation will be used in what follows. Use $\boldsymbol{x}$ to denote a place occupied by *p* in a three dimensional vector space at time *t*. Thus, $\boldsymbol{x} = \boldsymbol{x}(p,t)$ represents a motion of point *p*. The function $\mathbf{v} := \partial \boldsymbol{x}(p,t)/\partial t = \mathbf{v}(p,t)$ and $\mathbf{a} := d\mathbf{v}/dt := \partial^2 \boldsymbol{x}(p,t)/\partial^2 t = \mathbf{a}(p,t)$ are called the velocity and acceleration of *p*, respectively. To describe the theory of fluid mechanics, it is convenient to use spatial description. The spatial description of velocity and acceleration can be written as

$$\mathbf{v} = \mathbf{v}(x,t) \qquad \mathbf{a} = \mathbf{v}' + (\text{grad}\mathbf{v})\mathbf{v} \tag{2.1}$$

where $(\ )' := \partial(\ )/\partial t$ is the partial derivative with respect to time and grad denotes the gradient.

The mass conservation, the balance equations of linear momentum, and the balance equation of energy of *p* can be expressed by [2]

$$\rho' + \text{div}(\rho\mathbf{v}) = 0 \tag{2.2}$$
$$\text{div}\,\mathbf{T} + \mathbf{b} = \rho\mathbf{v}' + \rho(\text{grad}\mathbf{v})\mathbf{v} \tag{2.3}$$
$$\rho\,(e' + \mathbf{v}\cdot\text{grad}e) = \mathbf{T}\cdot\mathbf{D} - \text{div}\,\mathbf{q} + \rho h \tag{2.4}$$

where ρ is the density, e the internal energy, **b** the body force, **q** the heat flux and *h* the heating generation of point *p*, **T** is the Cauchy stress tensor acting on *p*, **D** the stretching tensor defined by $(\text{grad}^T\mathbf{v} + \text{grad}\mathbf{v})/2$, and div denotes the divergence,

For an isotropic fluid, the relationship of **T** and **D** satisfies [2]

$$\mathbf{T} = \alpha\mathbf{I} + \mu\mathbf{D} + \beta\mathbf{D}^2 \tag{2.5}$$

where α, μ and β are scalar functions of invariants of **D**, and **I** is the unit tensor. As the special instances of isotropic fluids, a linearly viscous (Stokes) fluid can be represented by [3]

$$\mathbf{T} = (-p + \gamma\,\text{Tr}\,\mathbf{D})\,\mathbf{I} + \mu\,\mathbf{D} \tag{2.6}$$

where Tr denotes the trace of a tensor, γ and μ are called viscosities. For a fluid described by Eq (2.6), if $\gamma = \mu = 0$, it is called the elastic, inviscid, or perfect fluid. A fluid is said to be incompressible if $d\rho/dt = 0$, or equivalently Tr $\mathbf{D} = \text{div}(\mathbf{v}) = 0$ during motion . For an incompressible fluid with μ independent of **D**, it is called the Newtonian fluid.

For a Stokes fluid with $\gamma = 0$, the balance equations of linear momentum (2.3) is reduced to

$$\mu\,\text{div}(\text{grad}\mathbf{v}) - \text{grad}\,p + \mathbf{b} = \rho\mathbf{v}' + \rho(\text{grad}\mathbf{v})\mathbf{v} \tag{2.7}$$

which, together with Eq (2.2), are known as the Navier-Stokes equations.

The Navier-Stokes equations draw the attention to both scientists who are interested in the mathematical behaviors of the equations, and engineers who are interested in solving practical fluid dynamics problems occurring in three-dimensional space. Together with boundary and initial conditions, the Navier-Stokes equations are solvable. Many commercial computational fluid mechanics software packages use a solver that is based on the Navier-Stokes equations.

A comprehensive 3-dimensional fluid dynamic analysis of a typical reciprocating compressor piping systems is usually computationally intensive for the application of the complete set of Navier-Stokes equations. Simplifying assumptions are typically made to reduce the simulation time for one-dimensional flow. This approach can result in a technically accurate analysis tool for engineering purposes as well as being a practical tool for day-to-day design.

2.2 One-dimensional theory of fluid mechanics

One-dimensional theory of fluid mechanics applies to the flow in ducts or pipes. In this theory, it is assumed that velocities are uniform in the cross section of the duct or pipe, and velocities normal to the axis of the duct or pipe are zero. For one-dimensional compressible flow, the equation of mass conservation and balance equation of linear momentum may be written as [4]

$$\partial\rho/\partial t + u\partial\rho/\partial x + \rho\partial u/\partial x = 0 \quad (2.8)$$

$$\rho(\partial u/\partial t + u\partial u/\partial x) + (\partial p/\partial x) + F = 0 \quad (2.9)$$

where u is the flow velocity, and $F = \rho fu|u|/(2D)$ is the friction force between fluid and duct with f as friction factor and D as the inner diameter of the duct or pipe. For gases, the balance equation of energy may be written as [4]

$$\partial p/\partial t + u\partial p/\partial x - c^2[\partial\rho/\partial t + u(\partial\rho/\partial x)] - (k-1)uF = 0 \quad (2.10)$$

where c is the sonic speed of gas defined by $c^2 := \partial p/\partial\rho$.

There are two typical methods to solve Eqs (2.8) to (2.10) numerically, namely, **frequency-domain** method and **time-domain** method.

It is assumed in the **frequency-domain** method that the variations in flow are a periodic functions of time, the oscillation of pressure and velocity are small compared to their mean values, and density at a given location is constant. With these assumptions, the equations (2.8) and (2.9) can be linearized and Eq (2.10) can be omitted. In reciprocating compressors, pressure and velocity are periodic so this seems a reasonable approximation to make. A benefit of the frequency-domain method is the computer processing time is less than other techniques. As a result, a frequency-domain solver has been attractive to acoustic analysis of pulsation in reciprocating compressors. Beta Machinery Analysis pioneered the development of acoustic simulation software for high speed reciprocating compressors based on the frequency-domain method in 1973.

The limiting assumptions adopted by the frequency-domain method are not required for a **time-domain** method. The time-domain method can be applied to the analysis of transient phenomena as well as periodic flow. It takes non-linearity of flow into account. Furthermore, the time-domain method can model the opening and closing events of cylinder valves of a compressor precisely. The mean pressure, which is of interest to pressure drop of the system, can be calculated anywhere in a system. Therefore, the time-domain method is more accurate and has more applications than the frequency-domain method.

As computer processing power increased, time-domain solvers for acoustical analysis of low speed and high speed reciprocating compressors were developed (circa 1998). Using parallel processing techniques, the time-domain solver became a practical tool for day-to-day design of reciprocating compressor packages.

As noted earlier, there are several mathematical models that can be used to simulate a physical phenomenon. After one as has decided upon the mathematical model, a numerical method of performing the calculations to solve the model must be selected. There are several methods of solving the time-domain method of simulating flow in a pipe. The most common numerical solvers for Eqs (2-8)-(2.10) are the finite difference method, finite element method and the method of characteristics.

It was found that for a set of partial differential equations, when each equation is a linear function of partial derivatives, then the set of partial differential equations can be precisely transformed to a set of ordinary differential equations on particular curves in a space without introducing any assumptions [5]. These particular curves are called the characteristic curves and the resulting ordinary differential equations are called the characteristic equations. The numerical method used to solve the characteristic equations is called the method of characteristics.

Since Eqs (2.8)-(2.10) are linear with respect to their partial derivatives; they can be transformed to ordinary differential equations on the characteristic curves [6] and can be accurately solved by the method of characteristics. It has been shown that acoustical simulation with the application of fluid mechanics and using the method of characteristics solved in time-domain allows for accurate calculation of the total pressure drop (static plus dynamic). Nevertheless, regardless of what mathematical model that is used or what numerical solver is used to simulate the acoustics in a reciprocating compressor package, the tool must be proven with real world examples.

3. Case Studies Illustrating Static and Dynamic Pressure Drop Effects

Case Study 1

This case study includes a variable speed compressor package in a sweet natural gas booster type application. Some specifics on this unit are given below and illustrated in Figure 8.

- Sweet natural gas, 0.63 SG
- Single stage
- Six Throws
- 9.25-inch bore
- 750 -1000 rpm
- 3800 HP driver
- Ps=700 to 900 psia
- Pd=1000 to1400 psia
- Cylinder loading – all double acting with Head End Variable Volume Pockets

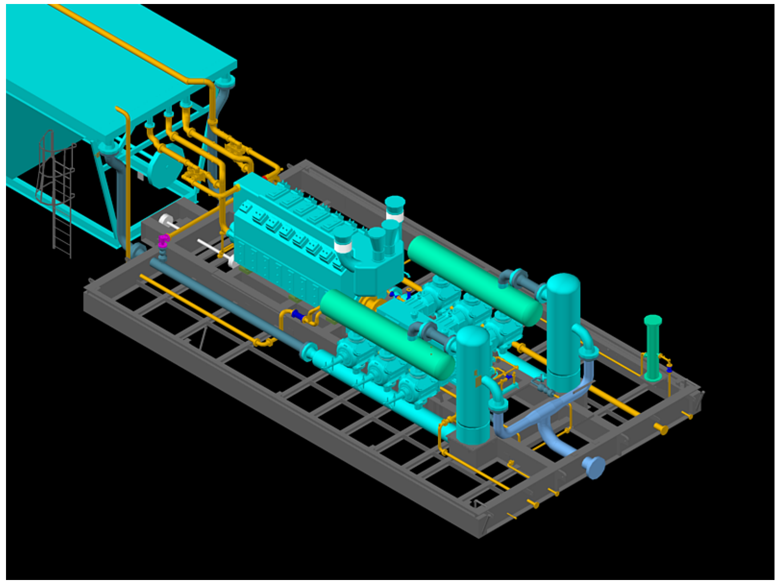

Figure 8: Case Study 1 Compressor Package Model

The compressor performance supplied from the packager included fifteen operating points (combinations of cylinder loading and suction and discharge pressures). The compressor performance assumed a pressure drop of 1% from the skid edge to the compressor cylinder on the suction and 2% from the compressor cylinder to the skid edge connection on the discharge.

The acoustical analysis was conducted for the design as proposed by the compressor packager. Pressure pulsations and dynamic forces were calculated to be well over the API design guidelines. The pulsation bottles were redesigned to provide acoustical filters as well as orifice plates recommended at key locations to reduce acoustical resonant responses. The resulting modifications reduced the pressure pulsations and dynamic forces to guideline levels.

The static pressure drop and total pressure drop plots due to the acoustical devices (choke tubes, baffles, orifice plates) in the suction system are shown in Figures 9 and 10. The static pressure drop exceeds the API 618 guideline for some operating conditions. However, the conditions with the static pressure drop above API 618 are off-design operating conditions where there is excess driver power available. The total pressure drop is approximately 20% to 55% greater than the static pressure drop for the acoustical devices.

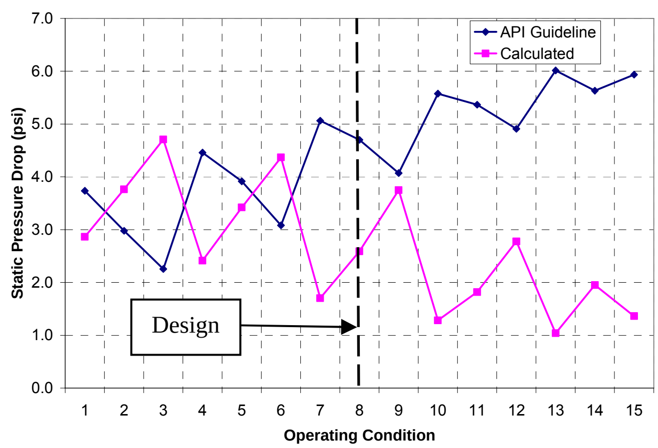

Figure 9: Static Pressure Drop due to Suction Pulsation Control Devices

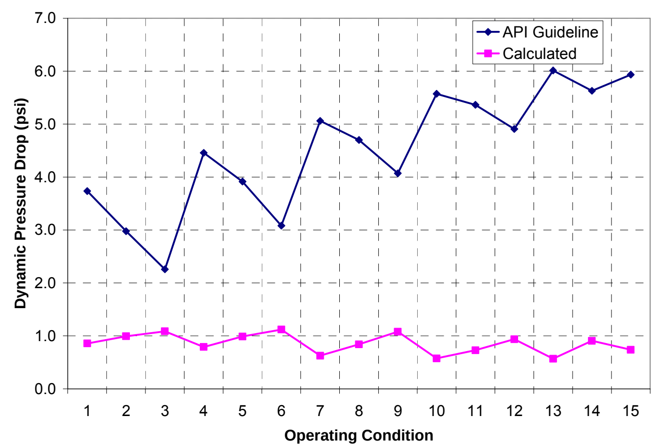

Figure 10: Dynamic Pressure Drop due to Suction Pulsation Control Devices

Figure 11 is a plot of the static and total pressure drop for the suction system acoustical devices compared to the total pressure drop for the suction system from the skid edge to the compressor cylinder. Also shown on this plot is the assumed pressure drop from skid edge to the compressor cylinder that was used in the compressor packager performance runs. This plot is interesting in that it shows if the static pressure drop from the acoustic devices is only considered in the pressure drop evaluation, the pressure drop for the suction system could be underestimated by a factor of two compared to the total system pressure drop.

Also note the difference in the pressure drop assumed by the packager performance runs and total (static plus dynamic) system pressure drop calculated by the acoustical simulation. The assumed pressure drop is up to 3 times higher than the actual pressure drop for some conditions. This difference in the pressure drop can have a significant effect on the calculated compressor performance (the compressor capacity could be lower than calculated or load higher than calculated.)

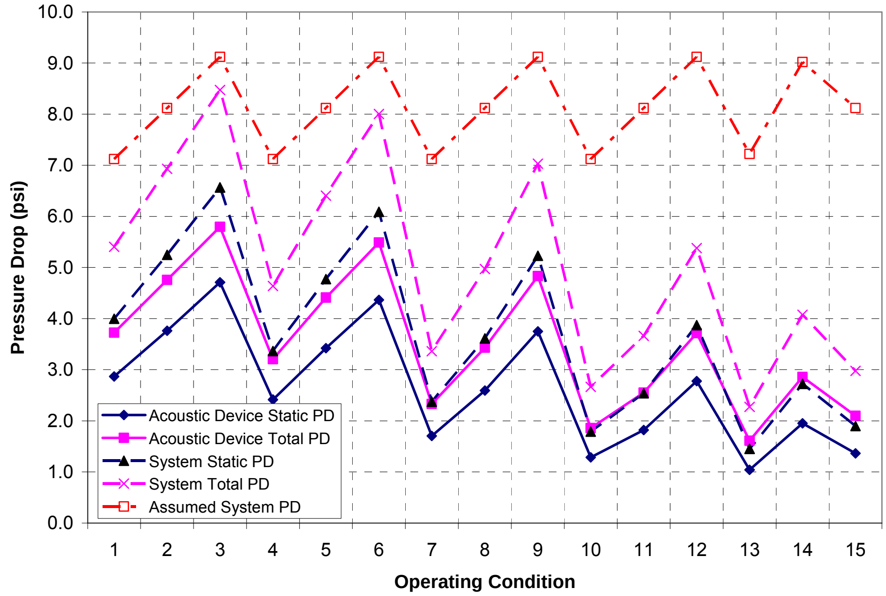

Figure 11: Suction Acoustic Device Pressure Drop Compared to the Suction System Pressure Drop

Figure 12 is a similar plot for of the pressure drop calculated for the discharge system. The results for the discharge system are different than the suction system, in that the total pressure drop is similar to the static pressure drop. Also, the system pressure drop is much higher than the pressure drop due to the acoustical devices, approximately 2.5 times higher. The high system pressure drop is due to the pressure drop through the cooler. The cooler pressure drop is typically a significant pressure drop on discharge systems.

Also note in Figure 12 the difference between the assumed pressure drop in the compressor performance runs compared to the calculated total system pressure drop. The calculated pressure drop ranges from 28 psi above to 21psi below the assumed pressure drop.

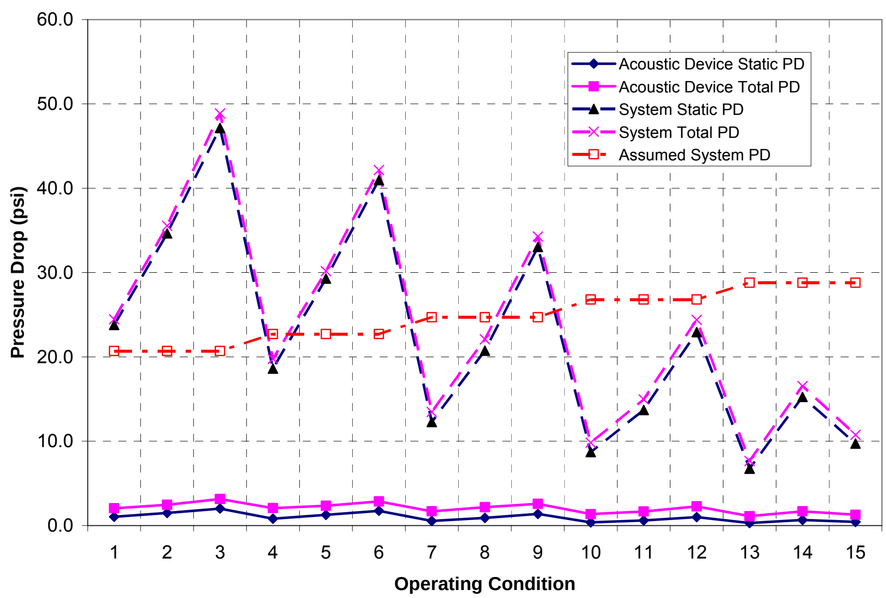

Figure 12: Discharge Acoustic Device Pressure Drop Compared to the Discharge System Pressure Drop

The effect of the system pressure drop on compressor performance was then evaluated. Figure 13 shows a plot of the percent change in the flow and load from the assumed pressure drop in the original compressor performance compared to the performance with the calculated total system pressure drop. This figure shows that the load would be approximately 14% higher, or 0.5% lower, than was originally calculated, which can be significant in terms of the additional fuel costs. The capacity difference ranges from -1% to +4.5%. This corresponds to -2 MMscfd to 4.6 MMscfd difference in the assumed versus calculated capacity in absolute terms, a significant impact on the compressor throughput.

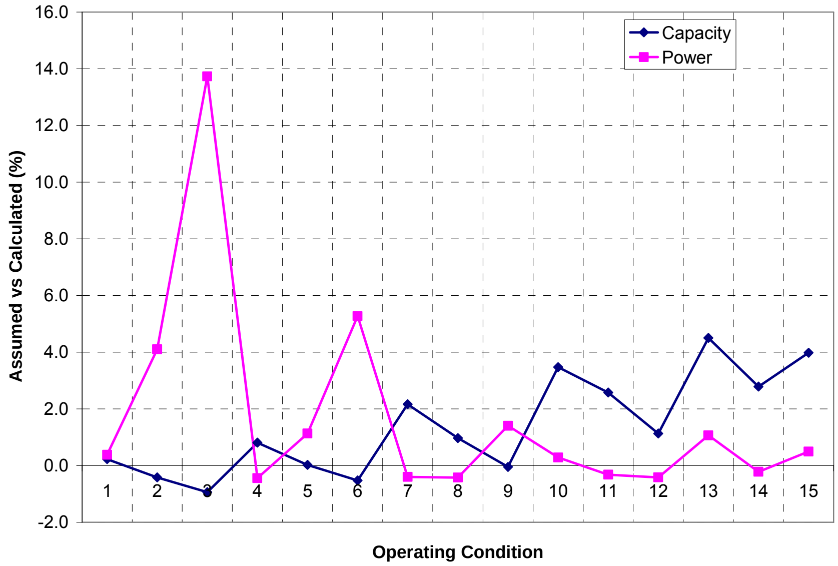

Figure 13: Percent Change in Capacity and Power for Assumed Pressure Drop vs. Calculated Pressure Drop

Case Study 2

This case study involves a four throw, 3 stage compressor in a refrigeration (propane) service as shown in Figure 14.

- 700-1000 rpm
- 1200 HP driver
- 98% propane, sg=1.5
- Ps=12 to 16 psia, Pd=190 to 267 psia
- All cylinders double acting with Head End Variable Volume Pockets on the first stage cylinders

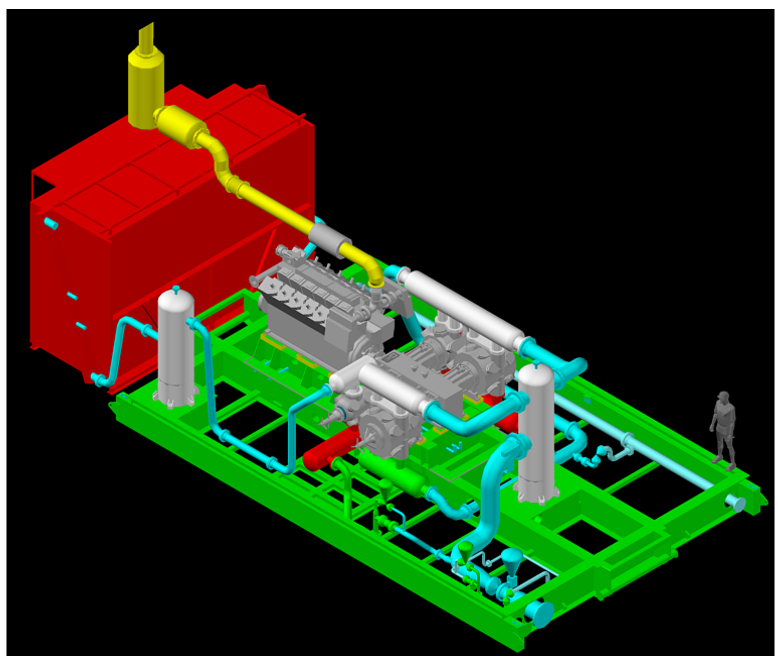

Figure 14: Case Study 2 Compressor Package Model

This was a new compressor package that had recently been put into service. The package design included orifice plates in several locations as well as choke tubes and baffles in some pulsation bottles. The owner was experiencing problems operating this unit as the engine was being overloaded and could not operate at the design conditions.

Field tests were done and it was determined that there was high pressure drop in the second interstage system. The pressure drop was measured to be approximately 17 psi. The owner asked for our evaluation of the system to determine if the orifice plates could be removed to reduce the pressure drop.

An acoustical analysis of the compressor package was conducted. Figures 15a and 15b show plots of the acoustical model. The static pressure drop for the orifice plates and baffle in the second interstage system was calculated to be 2.7 psi, much less than the observed 17 psi, and within the API 618 static pressure drop guideline of 2.8 psi. The total pressure drop (static and dynamic) from the acoustical devices was calculated to be 3.4 psi, again much less than the observed measurement and within the API 618 guideline. Removing orifice plates would reduce the pressure drop, however not significantly. The dynamic pressure drop was not excessive, although significant, about 50% of the static pressure drop.

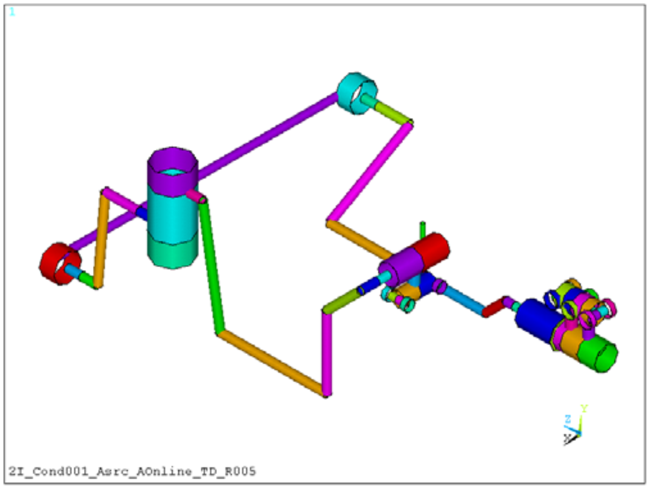

Figure 15a: Volume Plot of Case 2 Acoustical Model

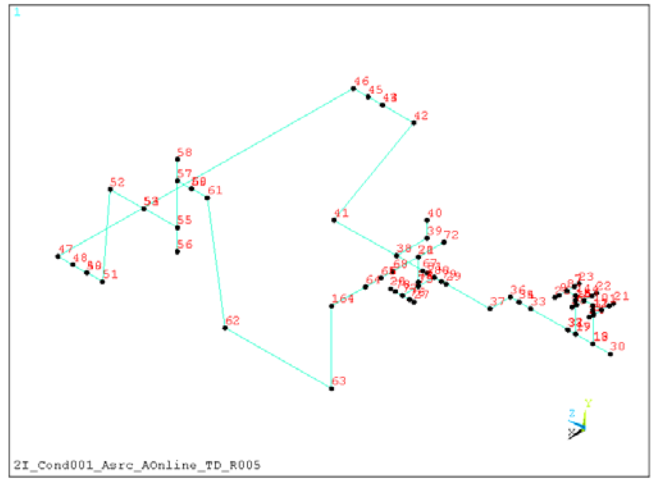

Figure 15b: Node Locations for the Case 2 Acoustical Model

The system pressure drop was then evaluated. The system pressure drop refers to the pressure drop from all components from the second stage discharge cylinder to the third stage suction cylinder. The static and total pressure drop was found to be approximately 10.5 psi and 15 psi respectively. Figure 16 shows a plot of the calculated mean pressure from the second stage discharge to the third stage suction. The highest pressure droo is shown to occur between nodes 46 and 47, which corresponds to the cooler. Also note the relatively large dynamic pressure drop in this case. The system dynamic pressure drop is approximately 4 psi which is more than the 2.7 psi static pressure drop from the orifice plates and choke tubes that were installed.

The calculated total pressure drop of 15 psi is close to the measured pressure drop of 17 psi and the associated error with these measurements. The measured pressure drop was based on interpretation of the P-V curves measured on the cylinders.

This case study shows that dynamic pressure drop and system pressure drop can be more significant than the obvious static pressure drop from orifice plates and chokes tube. All pressure drop factors must be included in the evaluating the compressor performance.

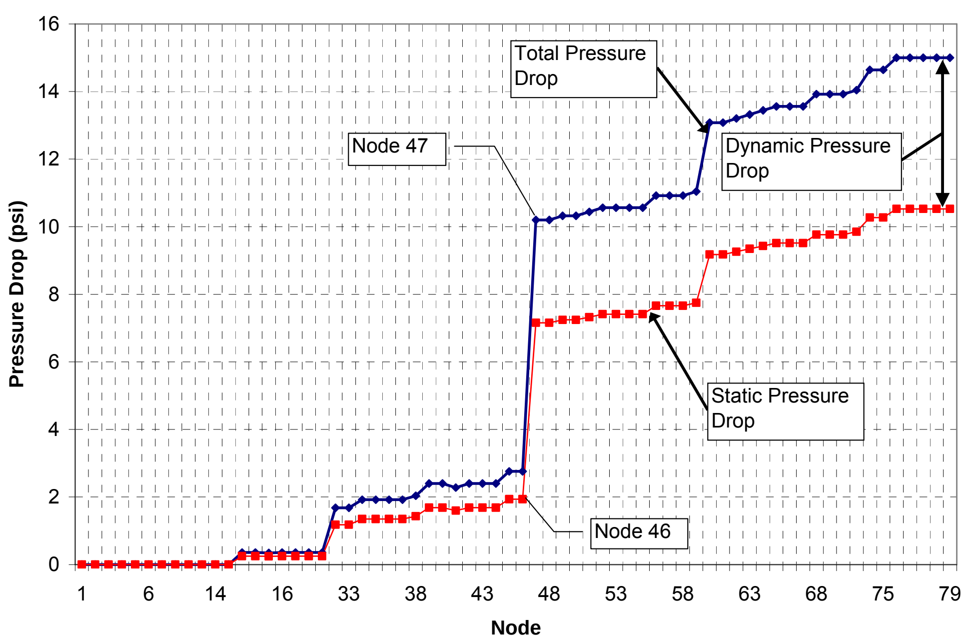

Figure 16: Calculated Pressure Drop For Case 2 Acoustical Model

4. Conclusion and Recommendations

Static and dynamic pressures are important factors to consider when calculating the compressor capacity and power requirements for low speed and high speed reciprocating compressor.

Static and dynamic pressure drop have been shown to vary with the compressor operating parameters like suction and discharge pressures, and cylinder loading. The pressure drop must be calculated for each operating condition to accurately determine the compressor performance.

Total pressure drop should be assessed over the complete system to ensure the compressor package meets the required capacity, power and operating costs. Case studies show the actual performance can vary significantly from the initial assumptions. A system analysis is relatively easy to perform, once the pulsation analysis is complete.

Static pressure drop can be calculated easily, however, dynamic pressure drop can only be calculated by the use of specialized acoustical analysis software. The acoustical analysis software must use an accurate mathematical model of the physical system to achieve an accurate result. It has been shown that acoustical simulation with the application of fluid mechanics and use of the time-domain method achieves this goal of accurately calculating the total system pressure drop. Not all acoustical analysis software has this capability.

API 618 5th Edition includes a requirement for the static and dynamic pressure drop design limit. These pressure drops must be calculated and reported for reciprocating compressor packages designed to meet API 618 5th Edition. It is recommended that the pressure drop be calculated for the full range for compressor operating conditions to ensure safe, reliable operation of the compressor.

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