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IMPACT OF PIPING IMPEDANCE AND ACOUSTIC CHARACTERISTICS ON CENTRIFUGAL COMPRESSOR SURGE AND OPERATING RANGE

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ABSTRACT

The performance of a centrifugal compressor is usually defined by its head versus flow map, limited by the surge and stall regions. This map is critical to assess the operating range of a compressor for both steady state and transient system scenarios. However, the compressor map does not provide a complete picture on how the compressor will respond to rapid transient inputs and how its surge behavior is affected by these events. Specifically, the response of the compressor to rapid transient events, such as single or multiple (periodic) pressure pulses, is also a function of the compressor's upstream and downstream piping system's acoustic response and impedance characteristics.

This unique response phenomenon was first described in the 1970s and came to be known as the "Compressor Dynamic Response (CDR) Theory". CDR Theory explains how pulsations are amplified or reduced by a compression system's acoustic response characteristic superimposed on the compressor head-flow map. Although the CDR Theory explained the impact of the nearby piping system on the compressor surge and pulsation amplification, it provided only limited usefulness as a quantitative analysis tool, mainly due to the lack of computational numerical tools available at the time. To fully analyze pulsating flows in complex centrifugal compressor suction and discharge header piping systems, the principles of the CDR should be implemented in a dynamic flow model to quantify the magnitude of the amplifications of pressures pulses near the surge region.

When designing centrifugal compressor stations within a transmission piping system, it is critically important to have a full understanding of the impact of the station's piping system on compressor dynamic behavior. For example, if a compressor system's piping impedance amplifies the suction side pulsations, the compressor's operating range will be severely limited and will produce unacceptable discharge piping vibrations. Whereas it is usually desirable to limit the

downstream volume between the compressor discharge and the check valve to reduce the potential for transient surge events, a small discharge volume results in high piping impedance. This will amplify pressure pulsations passing through the compressor. The small downstream volume provides limited ability for any transient peak (such as a pressure pulse) passing through the compressor to be absorbed quickly, and an amplified discharge pressure spike will be the result. Also, if any periodic pressure excitation from upstream vortex shedding or any other continuously varying flow disturbance couples with a pipe resonance length, the result can be high fluctuations of the compressor operating point on its speed line, effectively resulting in a reduced operating range and higher than expected surge margin (surge line moves to the right).

Both acoustic resonance and system impedance are functions of pipe friction, pipe and header interface connections, valve/ elbow locations, pipe diameter, valve coefficients, i.e., the entire piping system connected to the compressor. Thus, a careful acoustic and impedance design review of a compressor station design should be performed to avoid impacting the operating range of the machine. This paper describes the methodology of such a design review using modern pulsation analysis software. Examples and parametric studies are presented that demonstrate the impact of system impedance and piping acoustics on the dynamic operating response of the compressor in a typical compressor station. Some recommendations to reduce the risk of pulsation amplification and unsteady operation are also provided.

INTRODUCTION

Surge in centrifugal compressors has been studied by a number of researchers over the last 120 years. It is beyond the scope of this paper to list all of them, but a good summary review of the centrifugal compressor surge phenomenon can be found in Kurz and Brun (2006). On the other hand, the impact of the piping system, both the impedance and acoustic response, has received very little attention. The principle reference paper on the topic is almost thirty years old and was published by Sparks (1983). Sparks discussed the theory of piping acoustics and resonances and how they can affect the surge line of a centrifugal compressor. At that time, numerical modeling capabilities did not exist to properly simulate this phenomenon.

Although only anecdotal field data is available, it has long been recognized that the location of a surge line for an individual compressor is not solely a function of its specific design but can also be a function of the compression system. The system is impacted by the impedance and acoustics of the piping system to which the compressor is connected. This interaction between the centrifugal machine and its surrounding piping and valves, CDR Theory, explains the impact of the nearby piping system on the compressor surge and pulsation amplification.

When CDR was first understood, engineering analysis tools were inadequate to properly predict pulsating flows in complex geometries although, from a surge control and safe compressor system design standpoint, it is imperative to be able to accurately predict the interaction phenomenon. If a compressor's discharge piping impedance design amplifies suction pulsations, the result could restrict the operating range and cause unacceptable discharge piping vibrations.

The typical centrifugal compressor performance map (head or pressure ratio versus flow rate) with the corresponding speed lines indicates there are two limits on the operating range of the compressor (see Figure 1). Global aerodynamic flow instability, known as surge, sets the limit for low-flow (or high-pressure ratio) operation while choke or "stonewall" sets the high flow limit. The exact location of the surge line on the map can vary depending on the operating condition and, as a result, a typical surge margin is established at 10% to 15% above the stated flow for the theoretical surge line. Thus, every compressor has a limit on its operating map where the mechanical input is insufficient to overcome the hydraulic resistance of the system, resulting in a breakdown and cyclical flow-reversal in the compressor. Surge occurs just below the minimum flow that the compressor can sustain against the existing suction to discharge pressure rise (head). This map is appropriate for the characterization of steady state and slowly changing operating conditions, but it is not fully applicable for rapidly transient or high frequency periodic compressor flow inputs.

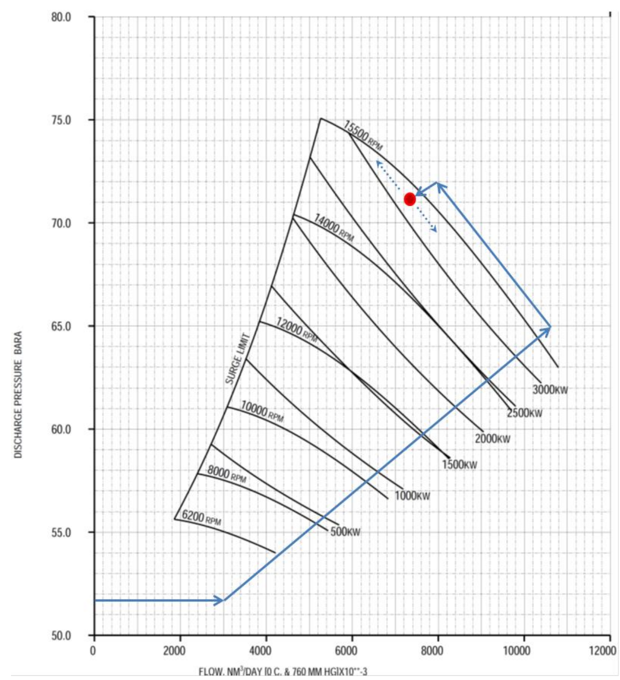

Figure 1. Typical Pipeline Compressor Map and Startup Sequence

Once surge occurs, the flow reversal reduces the discharge pressure or increases the suction pressure, thus allowing forward flow to resume until the pressure rise again reaches the surge point. This surge cycle continues at a low frequency until some change takes place in the process or the compressor conditions. The frequency and magnitude of the surge flow-reversing cycle depend on the design and operating condition of the machine, but in most cases, it is sufficient to cause damage to the seals and bearings and sometimes even the shaft and impellers of the machine. Thus, surge is a global instability in a compressor's flow that results in a complete breakdown and flow reversal through the compressor.

It is acceptable to assume that the relatively fast flow transients (above 1-2 Hz) experienced by the centrifugal compressor do not affect the compressor's operational speed (Brun and Kurz, 2010). The centrifugal compressor continues to operate at a constant speed as the rotational inertia of the compressor (and its driver) will torsionally dampen any fluid induced by the rapid blade loading changes. Thus, the compressor will operate on a fixed head-flow speed line.

When the compressor experiences suction or discharge flow fluctuations superimposed on the mean-flow, these fluctuations can often be enough to momentarily move the compressor operating point on the map's speed line across the surge limit and affect the forward flow stability of the compressor. Although this flow reversal event may be very short-lived (depending on the frequency of the flow fluctuation), it is usually sufficient to drive the compressor into a full surge cycle. Thus, even if a compressor is operating with an adequate surge margin based on the mean-flow, high inlet or discharge side pulsations have the potential to cause the compressor to operate in periodic unsteady surge cycles.

As previously noted, the dynamic behavior of the compressor system near the surge line was outlined by Sparks (1983) and further discussed by Kurz et al. (2006). They explain how pulsations are amplified or reduced by a compression system's acoustic and piping response characteristic superimposed on the compressor head-flow map. Kurz et al. (2006) also noted that centrifugal compressor stability is sensitive to highly pulsating flows, especially in cases where operating at piping acoustic resonance frequencies cannot be avoided. Choke should be avoided as it is an inefficient operating regime and may result in damage to the compressor. On the other hand, it is critical to avoid any kind of surge event, as these can cause bearing or seal damage, blade rubbing, or even catastrophic compressor failures.

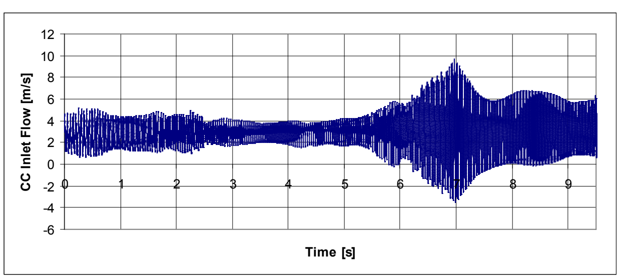

Figure 2. Centrifugal Compressor Inlet Velocity versus Time

Strong pulsations in pressure and flow have the capability to move a centrifugal compressor into surge or choke Figure 2 shows the velocity fluctuations in time at the inlet to a centrifugal compressor. Inlet pulsations (velocity fluctuations shown on the y-axis) became periodically negative given the acoustic and impedance system effects, in the 6-8 second period. This corresponded to short duration surge cycles at a frequency equal to that of the inlet pulsations.

A centrifugal compressor either attenuates or amplifies pulsations at its discharge or suction side, because it reacts to any fluctuation in flow with a fluctuation in head which can be easily seen from its performance characteristic in Figure 3 (Sparks, 1983).

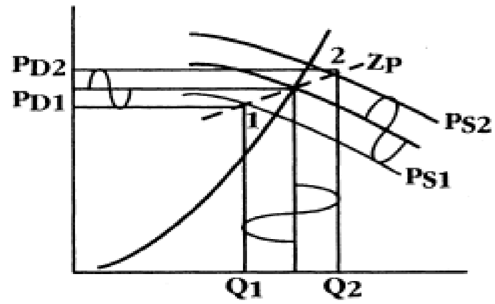

Figure 3. Pulsation Transmission in Centrifugal Compressors (Sparks, 1983)

Sparks (1983) explains the process as the interaction of a piping system with given acoustic impedance and a compressor that reacts to a change in flow with a change in head (or pressure ratio). The piping impedance is usually a combination of resistive impedance (i.e., due to frictional losses) and acoustic inertia (due to the mass of the gas in the pipe) and stiffness (due to the compressibility of the mass in the pipe).

The compressor head-flow characteristic will exceedingly differ from the steady-state characteristic for higher fluctuation frequencies. Sparks (1983) also discusses practical piping system design approaches to reduce pulsation levels using acoustic elements, such as bottles, nozzles, choke-tubes, and resonators.

A number of other studies are available in the public domain that evaluate the impact of flow pulsations on turbomachines. A detailed literature review on this topic was provided by Kurz, et al. (2006), and an overview of the state-of-the-art of pulsation analysis technologies was included in Brun et al. (2007).

BACKGROUND

Acoustic impedance, Z, is the ratio of acoustic pressure p to acoustic volume flow U, defined as:

$$Z = \frac{p}{U} \qquad \textit{Equation (1)}$$

Acoustic impedance can vary strongly with the excitation frequency. The acoustic impedance at a particular frequency indicates how much sound pressure is generated by a given gas molecule vibration at that frequency. It is the acoustic pressure divided by the particle velocity and the surface area through which an acoustic wave propagates.

Thus pipe flow impedance is defined as an increase or decrease of local static pressure for a given bulk volume flow. For a high impedance piping system, a small flow fluctuation will rise rapidly and create a strong, short pressure pulse whereas low impedance pipe will tend to dampen the same pressure pulse. This is somewhat counter-intuitive as one generally thinks in terms of pressure drop across piping system rather than pressure increase.

However, piping impedance specifically refers to the pressure increase as kinetic energy of a pressure wave is converted to pressure. This is similar to the dynamic pressure conversion to stagnation pressure but applies to individual pressure waves or transient pressure fluctuations. An increased impedance of a pipe system results in a more rapid conversion of volume flow to pressure than a low impedance system. An easy way to understand this is to realize that with increasing impedance, the flow capacitance is reduced (the discharge flow does not provide for a quick relief of the mass out of the system, and the pressure must rise). For example, if a pipe is highly flow restricted (because of an orifice, pipe friction, or nearly closed valve), even a small increase of flow into it will result in a quick build-up of pressure. Compare this to a wide open, unrestricted pipe which will allow the flow to pass, and pressure will remain nearly unchanged.

Good design practices for centrifugal compressor piping systems typically dictate that the compressor downstream volume should be minimized to reduce the likelihood of transient surge events (Brun and Nored, 2008). Small distances and total volume in the recycle line allow for faster recycle system responses and are generally preferred (such as in an emergency shutdown scenario). However, while it is usually desirable to have small volumes between the compressor discharge, from an impedance amplification perspective, large downstream volumes with low resistance actually avoid pulsation amplification and improve the acoustic response of the system. Even if the system operates on or near an acoustic resonance, the low impedance of a high volume system may help avoid the build-up of strong standing wave resonance pulsations. Thus, from an acoustic impedance perspective, a large volume in the discharge piping system and a low resistance check valve are preferred. This design requirement is clearly opposite to that of conventional station design criteria for ESD and surge system responsiveness, and it must be carefully balanced.

Acoustic impedance in compressor systems is only a concern when some form of periodic pressure excitations is present. Although one would expect that a centrifugal compressor system has relatively steady flow, there are several sources of excitation that can result in significant pulsations. These sources of excitation in centrifugal compressors are:

- Strouhal excitation or vortex shedding
- Blade passing and blade vane interactions
- Turbulence induced radial or other 3-D acoustic responses in large vessels
- External pulsation from reciprocating compressors or other positive displacement machines
- Unstable flow control valve control cycling (process valve dynamics)
- Check valve or relief valve chatter
- Surge cycles
- Diffuser rotating stall and other stalls
- Mismatch between the operating points of the compressors resulting in periodic “hunting” for a stable operating point.
- Other process and aero flow instabilities

All of these mechanisms, when amplified by piping resonance or piping impedance changes, can move the centrifugal compressor operating point into a surge or choke (stonewall) condition. This phenomenon has been described by several compressor operators when they experienced cyclic surge events at operating conditions where sufficient surge margin should have been present.

It is important to realize that low frequency pulsations require long distances to dampen out. Pulsations below 100 Hz can travel several miles (Kurz, et al., 2006) with relatively low attenuation. Figure 4 shows an example of how a pulse of 15 psi dampened out in a pipeline and that even after 10 miles, the pulse amplitude is above 1 psi. If this pulse were periodic in nature, it could easily excite a resonance frequency in a compressor station which would result in amplification and very high pulsation levels. Thus, an upstream or downstream external source, such as a reciprocating compressor station or piping vortex shedding, can still excite pulsation and acoustic responses inside the centrifugal compressor station piping.

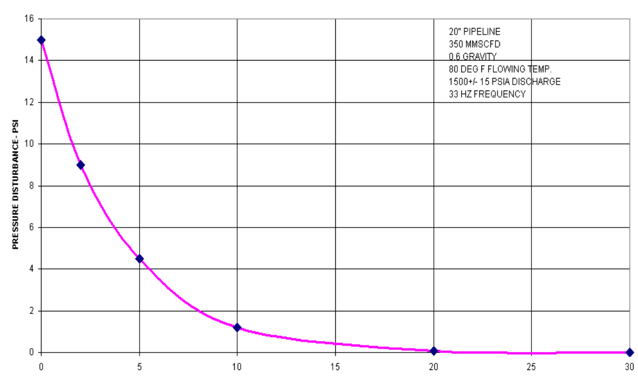

Figure 4. Pulsation Dampening in a Typical Pipeline (Kurz, et al., 2006)

A complete understanding of the system effects on the compressor dynamic behavior is required during the design process and for troubleshooting. Compressors can amplify suction/discharge pulses to where the result could limit the compressor’s surge-to-choke range and cause severe piping vibrations. There are two related factors that determine whether a periodic flow fluctuation is amplified or damped within a piping system:

1. The pipe system impedance and the gradient of the impedance: This determines whether a velocity fluctuation converts to a rapidly rising pressure pulse or a declining pressure pulse. Generally, a high impedance or a positive gradient of impedance results in increasing pressure pulse, whereas low impedance or a negative gradient results in a declining pressure pulse.

2. The acoustic resonance response of the piping system to a given excitation pulse (frequency and amplitude): If a periodic flow fluctuation encounters a piping resonance length corresponding to one of its primary acoustic mode shapes, an amplification of the pressure pulse (i.e., a resonance response) can result. The amplification rate of the resonant response is a complex function of the wave superposition at its end conditions and the wave's viscous energy dissipation. It is important to note that not just a closed or open end wall can act as an acoustic end condition, but a rapid impedance change along the flow path is also an acoustic end condition.

The analysis of these two factors is particularly important in the piping near the compressor for the determination of the pulsation amplitudes at the compressor suction and discharge and to assess their impact on the compressor's operating stability. Both impedance and acoustic response analysis of a piping system is highly complex and requires the determination of the transient flow field in the entire piping system. This can be accomplished by using a 1-D transient pipe flow solver for a multi-element interconnected pipe system, as discussed below in this paper.

It is imperative to understand that both acoustic resonance and system impedance are functions of pipe friction, pipe and header interface connections, valve/ elbow locations, pipe diameter, valve coefficients, etc., i.e., the entire piping system connected to the compressor. Thus, a careful acoustic and impedance design review of a compressor station design should be performed to avoid impacting the operating range of the machine.

SOLVER METHODOLOGY

Systems of Equations

For the analysis of highly transient flow in complex piping systems any "linearized" solutions of the transient wave equation or even transient perturbation transport solutions, such as those employing the method of characteristics or finite wave methods, are inherently not suitable, as they do not fully model the fluid flow and compressor physics. Thus, a full solution of the Navier-Stokes equation coupled with physical compressor models is the most appropriate solver to model the transient fluid flow and interaction of centrifugal compressors and their piping systems. This model was described in detail by Brun, et al. (2006) and is only briefly outlined below.

One-dimensional unsteady compressible inviscid flow is governed by a system of three equations: continuity, x-momentum and energy. After a suitable non-dimensionalization, the system of equations can be written as

$$\frac{\partial \bar{\boldsymbol{q}}}{dt} + \frac{\partial \bar{\boldsymbol{F}}}{\partial x} = 0, \qquad \textit{Equation (2)}$$

where

$$\bar{\boldsymbol{q}} = \begin{pmatrix} \varrho \\ \varrho u \\ \frac{p}{\gamma(\gamma-1)} + 0.5\varrho u^2 \end{pmatrix} \textit{and } \bar{\boldsymbol{F}} = \begin{pmatrix} \varrho u \\ \varrho u^2 + \frac{p}{\gamma} \\ \left(\frac{p}{\gamma-1} + 0.5\varrho u^2\right) u \end{pmatrix}$$

Equation (3)

where ϱ is the density, u is the velocity, p is the pressure and γ is the specific heats ratio.

A full one-dimensional time-domain flow solver applicable to any complex interconnected manifold and piping system was developed to determine the highly transient fluid pulsations in mixed reciprocating and centrifugal compressor stations. This three-equation transient flow solver includes all terms of the governing equations, including fluid inertia, diffusion, viscosity, and energy dissipation. Physical models for both centrifugal and reciprocating compressors were also derived and implemented into the solver.

In a complex piping system, the above set of equations must be individually solved for all pipe segments with the appropriate inlet and outlet conditions updated at each time-step. Within each pipe segment, a simple central difference discretization and time-space forward marching solution was utilized. Pressure losses inside the pipe and at the interfaces are determined from basic pipe friction loss models and viscosity losses are directly calculated from the discretized viscosity term of the momentum equation (along the flow direction). These two terms are applied at every time-step at every applicable node. Thus, the viscous terms are not simply treated as a pipe friction loss but also include a discretization of the x viscous term (in the flow direction). The x discretized term and the pipe friction viscous term are calculated for each node after determining the half step and are then included to determine the full step. This allows for coupling of the viscous terms with the inertial and pressure terms of the momentum equations.

Pipe inlet and outlet conditions, which are also enforced at every time-step, were either active inlet forced (sinusoid, square wave) or active unforced functions (compressor), pipe intersections (branching nodes), or passive-end conditions (infinite pipe, open- or closed-end). Formulations must also be provided to determine boundary conditions at multiple pipe interfaces, such as pipe tees or joints. The continuity equation may be applied to determine the resulting velocity in each pipe inlet and outlet. The average area of the intersection point is used to solve for the inlet velocity in each reach at the intersection point. Pressure is assumed to be equal at the intersection point at each time-step. As the interface pressures and mass flows are determined from a balance of the surrounding nodes, transient terms of the governing equations are implicitly included.

Compressor Model

The state of the gas in any compressor manifold and attached piping system is determined by two factors: (1) the kinematics of the compressor drive, which provides a forcing inlet boundary condition to the piping system, and (2) the fluid dynamic behavior (response) of the piping system and all associated outlet boundary conditions. For a centrifugal compressor, boundary conditions can be derived directly from a specific compressor performance map as shown in Figure 1. This map must be obtained from the centrifugal compressor manufacturer or from test data. For numerical simplicity, this map can often be further simplified by non-dimensionalizing the flow and head into characteristic psi-phi curves. However, given the very short period of flow transients (in the order of 0.5 to 100 Hz) and the significant rotational inertia of a gas turbine driven centrifugal compressor, it is accurate to model the centrifugal compressor as a constant speed machine. This assumption can be validated for a given application by calculating a non-dimensional parameter that ratios pulsating aerodynamic torque on the impeller with the rotor's total angular inertia.

The authors derived the following non-dimensional number, C, for this analysis:

$$C = \frac{\omega r^2 \Delta \dot{m}}{(2\pi) J f^2} \qquad \text{Equation (4)}$$

where $\Delta \dot{m}$ is the inlet mass flow pulsation peak-to-peak magnitude, ω is the angular speed of the centrifugal compressor (in rad/s), r is the tip radius of the centrifugal compressor impeller, J is the moment of inertia of the rotor, and f is the pulsation frequency (in Hz). For small values of this number ($C < 0.1$) the inertial forces dominate the pulsating aerodynamically induced torque, and one can assume that the impeller speed remains constant and unaffected by pulsations. This simplifies the centrifugal compressor boundary condition to a simple second order polynomial:

$$H = \Gamma_1 + \Gamma_2 Q + \Gamma_3 Q^2 \qquad \text{Equation (5)}$$

where the coefficients are determined from the head-flow compressor map. A parabola was used herein but for some applications, a higher order polynomial can provide a better representation of the compressor map. The limits of this polynomial are surge on the low flow side and choke on the high flow side. Thus, for a given compressor speed, Equation (5) and the fan law are utilized to obtain flow for a given pressure suction and discharge condition. Similarly, transient forcing boundary functions for vortex shedding or other unsteady flow excitations are assumed to be sinusoidal and their magnitudes were simply obtained as a percentage of total flow pressures. All boundary conditions were applied at pipe ends.

Code Implementation

The unsteady one-dimensional Navier-Stokes code was written for a complex system of pipes with multiple interfaces and has all standard boundary conditions, such as open ends, closed ends, a compressor cylinder, sine wave, and square waves, available. Interfaces between pipe segments can be one-on-one, two-on-one, one-on-two, one-on-three, or three-on-one and include discrete pressure drops at the segment interfaces. Other boundary conditions are either passive (closed wall, open wall, infinite pipe) or active (compressor, sine wave, square wave). The pipe areas can change either gradually (transition piece) or abruptly (open-end or bottles) within the pipes or at the interfaces of pipe segments. Models with up to three hundred interconnected pipe segments were successfully tested. A Windows file pre-processor, graphical user interface, and graphical post-processor (for frequency and time domain data) were also implemented. Figure 5 shows the user interface for a typical compression station model. Full frequency sweeps can be performed with all boundary conditions. The post-processor interface includes a Hanning window function, a time domain, and an FFT output option. A detailed validation of the subject code was presented by Brun, et al. (2007).

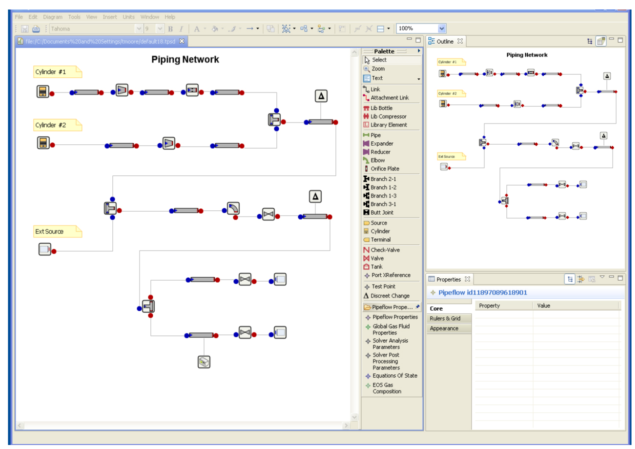

Figure 5. Graphical User Interface for Solver

CASE STUDIES

The above-described transient flow solver was utilized to determine the transient pipe flow and compressor operation of a single unit compressor system. To study the impact of piping impedance and acoustic resonance frequencies on periodic excitations in compression systems, a single compressor with a simple recycle loop system was analyzed. Figure 6 shows the loop that was simulated in this case study. Although similar studies have been performed on other compressor station arrangements and operating conditions, there is no field data currently available to compare with these prediction results. The phenomenon described and modeled herein has been described by many compressor station operators; i.e., significant anecdotal evidence exists that compressors experienced cyclic surge events when operating well to the right of the surge line.

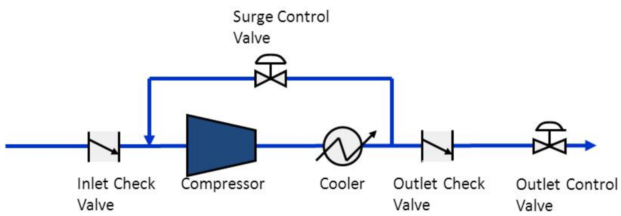

Figure 6. Arrangement Drawing of Simulated Compressor Pipe System

The loop dimensions and specifications were as follows:

- All pipe diameters are 0.5 m, and all pipe segments are 5 m long, except for the inlet and outlet segments which are 500 m long.
- Inlet process gas is natural gas at 25°C and 52.4 bara.
- The compressor was modeled after typical pipeline compressor with a single stage (compressor map is shown in Figure 1)
- The cooler provides a constant discharge temperature of 25°C and has a pressure drop coefficient of 0.05.
- All valves had a nominal pressure drop coefficient of 0.02 when fully open and were assumed to have a linear pressure drop versus time when closing.

The simulated startup sequence of operation of the compressor station model was identical to an actual pipeline compressor startup utilizing an electric motor driver. The simulated operating sequence was as follows:

1. The recycle valve and the outlet control valve start wide open.
2. The compressor speed ramps up from 0 to 15,600 rpm (0-3.0 seconds), stays at 15,600 rpm (3.0-4.0 seconds), and then ramps down to 14,400 rpm (4.0-5.0 seconds), where it stays for the remainder of the simulation.
3. The recycle valve is wide open until 1.5 seconds and then linearly closes so it is fully closed by 5 seconds, where it stays for the remainder of the simulation.
4. The outlet control valve starts closing after 0.5 seconds and stops at 90% closed at 1 second, where it remains.
5. After about 10 seconds, the compressor has reached a stable operating point.

Once the system reached a steady state operating point, as observed on the compressor maps (with no significant flow fluctuations), a 5% or 10% mean flow sinusoidal excitation is artificially induced into the flow at the T-pipe intersection between the main loop and the recycle loop upstream of the compressor. (A 10% variation in mean flow is within the range of field test results measuring the inlet velocity at a centrifugal compressor station and is a good guide for the high end expected dynamic velocity.)

This flow fluctuation simulates the type of flow unsteadiness observed from vortex shedding or other periodic flow excitation. The resulting transient pressure ratios, volume flows, and mass flows were measured at test points placed on the suction and discharge of the centrifugal compressor in the simulation model.

To change the system impedance, the pipe diameter between the downstream compressor T-intersection and the downstream check valve was increased from 0.13 to 0.30 meters and 0.5 meters corresponding to high, medium, and low impedance, respectively. After an initial steady state test run to determine the surge line, all transient cases were run with excitation frequency sweeps from 0 to 128 Hz for both the 5% and the 10% pulsation amplitude cases.

CASE STUDY RESULTS

For the above described compressor and piping geometry as well as operating conditions, the following cases were analyzed (Table 1):

Table 1: Simulation Cases for Typical Simple Compressor Station

Excitation of Mean Flow	Low Impedance (D=0.5m)	Medium Impedance (D=0.3 m)	High Impedance (D=0.13 m)
5% (≈ 2 m/s)	Case 1	Case 2	Case 3
10% (≈ 4 m/s)	Case 4	Case 5	Case 6

Although excitation sweeps from 0 to 128 Hz were performed for all cases, for the sake of clarity, only results from 5 to 30 Hz are presented. This is the frequency range most commonly associated with vortex shedding and externally induced excitation frequencies in compression stations, and it also is the range in which the most significant impact of acoustic response and impedance on the flow field was observed.`

Results from these simulations were post-processed for mass-flow, pressure ratio, and volume flow across the compressor versus excitation frequency. For example, Figure 7 shows compressor discharge mass-flow versus excitation frequency for Case 6 (10% excitation and high impedance). Here the response of compressor discharge mass-flow is clearly seen to be a function of the piping system's acoustic response, as there are distinct changes and peaks in pulsation amplitude.

In this case, a strong system response at 19 Hz and some other less significant peaks are observed. The 19-Hz response corresponds to a system acoustic closed-closed end half wave resonance between the compressor discharge and the discharge valve. In this case, the inlet excitation of 10% total flow (i.e., 38 to 42 kg/s input excitation) is seen to be amplified by a factor of approximately 3 (i.e., 34 to 46 kg/s system response). As a comparison, for the same excitation but the low impedance case (Case 4), the compressor discharge flow fluctuation peak-to-peak was seen to be only approximately 1 kg/s at 19Hz. This means that for the high impedance case, the excitation pulsations were amplified while for the low impedance case, they were damped.

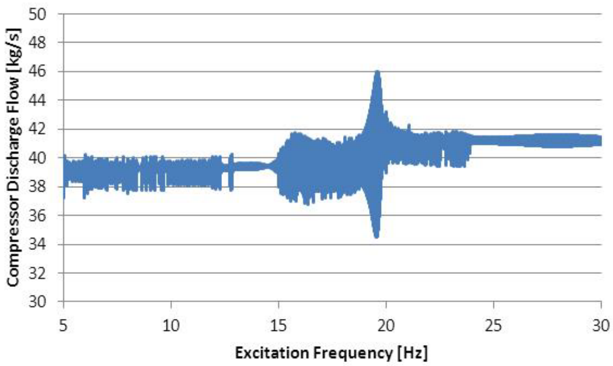

Figure 7. Compressor Discharge Mass-Flow for Case 6 (10% excitation high impedance case)

A good illustration of how piping impedance can affect the system response and subsequently the compressor operating point can be seen in Figure 8. Here, the compressor pressure ratio is plotted versus excitation frequency for the high and low impedance case at 10% excitation. Clearly in both cases, a system acoustic response around 19-20 Hz is still identifiable, but the total magnitude of the pressure ratio variations across the compressor increase drastically when the discharge piping impedance is changed from the low to the high case. The pressure ratio magnitude peak-to-peak difference is a factor of 4 between these two cases. Obviously, the low impedance response would be the desirable case, as it does reduce the piping pulsation and keeps the compressor operating pressure ratio much more stable. However, as previously noted, the larger pipe diameter of the low impedance case also results in a much larger discharge volume which is detrimental from a transient shut-down surge and system control perspective.

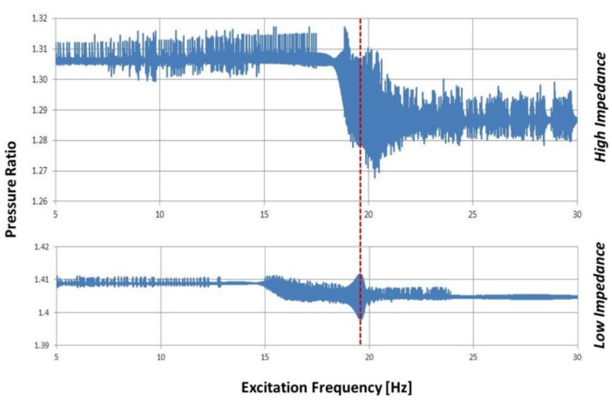

Figure 8. Compressor Pressure Ratio Comparison for High versus Low Impedance (Cases 4 and 6)

To further illustrate this point, Figure 9 shows a comparison of the peak-to-peak compressor pressure ratio variations versus discharge piping diameter (i.e., impedance).

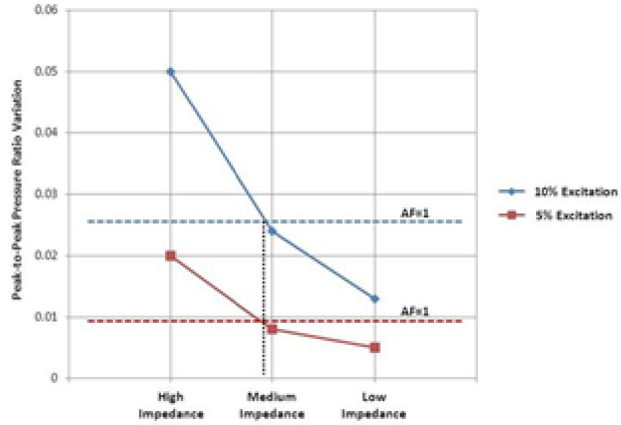

Figure 9. Compressor Peak-to-Peak Pressure Ratio versus Discharge Piping Impedance

As expected, the piping pulsation response and thus the peak-to-peak pressure ratio variations across the compressor decline with decreasing impedance. This chart also includes lines that indicate at which level there is no pulsation amplification (i.e., where the pulsation amplification across the compressor would equal unity). These lines are a function of the input pulsation magnitude, frequency, and the overall system dynamic behavior, but in this case, these lines nearly coincide with the medium impedance case. From a piping design point, it would be desirable to have a piping system whose amplification factor is always well below unity to avoid pulsation amplification and compressor instability or periodically transient surge.

Many other system acoustic responses are possible in complex piping systems, and it is critically important to avoid these when designing the piping system for a centrifugal compressor, especially in cases where some periodic excitation source near the compressor is anticipated (vortex shedding, reciprocating compressors, etc.).

The above figures clearly demonstrate that the system response is not only a function of the piping acoustic characteristics, but also of the piping's impedance. In this simple case, the impedance was changed by reducing the pipe diameter from 0.5 m to 0.13 m. This effectively reduced the volume between compressor discharge and the first downstream critical flow orifice (discharge valve), resulting in an increase in piping impedance. Here the system response at 19 Hz is seen to significantly increase with increasing piping impedance. In more complex piping systems, there will be multiple responses and varying pulsation amplifications depending on operating condition and excitation frequency. The easiest method to analyze this and to validate the compressor stability is to review the compressor's surge over its entire operating range.

Figure 10 shows the remaining surge margin of the pipeline compressor for Case 6 (High Impedance and 10% excitation). The steady state surge margin for this case is slightly above 20%, but as can be seen from the figure, when the excitation frequencies are between 18-25 Hz, the surge margin can drop as low as 3%. If this compressor had operated at a point on the map with less than steady 20% surge margin, the compressor would have likely entered into periodic surge cycles due to the induced pulsation response. Because of the high pulsation frequency, a standard surge control system would be unable to respond properly to these events and the compressor would experience real surge cycles.

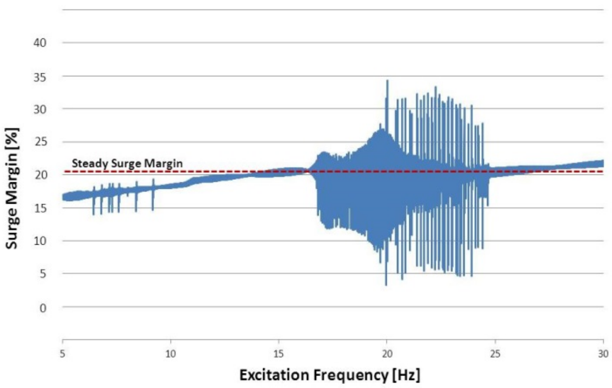

Figure 10. Surge Margin versus Excitation Frequency for Case 6 (10% Excitation and High Impedance)

SUMMARY AND CONCLUSIONS

A steady-state head versus flow centrifugal compressor performance map does not predict how the compressor reacts to transient and periodic inputs and how its surge margin could be reduced by these events. The compressor's response to rapid transient events, such as single or multiple (periodic) pressure pulses, is also a function of the compressor's upstream and downstream piping system's acoustic and impedance characteristics. This paper described the methodology of a CDR design review using modern pulsation analysis software. Six case studies for a simple single compressor station layout were analyzed for two different excitation levels and three impedance scenarios. These studies demonstrate the impact of system impedance and piping acoustics on the dynamic operating response of the compressor in a typical pipeline compressor station. Although there is no field data available to compare with these prediction results, the phenomenon modeled herein has been described by several compressor operators when they experienced cyclic surge events at operating conditions where sufficient surge margin should have been present.

Recommendations and key findings from this study are summarized as:

- Any flow unsteadiness or periodic excitation in a centrifugal compressor station piping system can be amplified by either piping resonance or impedance and can significantly decrease the compressor's surge margin.
- Periodic flow excitation (pulsations) can originate from vortex shedding, blade passing, and blade vane interactions, turbulence induced radial, 3-D acoustic responses, external pulsation, unstable flow control, check valve or relief valve chatter, surge cycles, diffuser rotating stall and other stalls, compressor "hunting" for a stable operating point, and other process and aerodynamic flow instabilities.
- Low frequency pulsations require long distances to dampen out. In pipelines that utilize both centrifugal and reciprocating compressors, the pulsation impact of upstream and downstream stations must be considered as an excitation source.
- Large discharge piping volumes with low impedances usually result in damping of pulsations whereas high impedance systems (small piping volumes) can result in pulsation amplification. Whereas it is usually desirable to limit the downstream volume between the compressor discharge and the check valve to reduce the potential for transient surge events during shutdowns, a small discharge volume results in a high discharge piping impedance which will usually amplify pressure pulsations passing through the compressor.
- The combination of high compressor discharge impedance with acoustic resonance can easily result in pulsation amplification factors exceeding 4x-5x. In this scenario, even at surge margins that are normally considered to be safe (>20%), moderate flow excitations of or 10% mean-flow can result in a high pulsation response with the compressor operating in periodic surge cycles.
- A transient dynamic flow analysis can be used to predict the impact of flow excitation in combination with acoustic and impedance pulsation amplification on the surge margin of a compressor. This type of analysis should always be performed during the compressor station design process.

When designing compressor stations, it is critically important to have a full understanding of the impact of the station's piping system on the compressor dynamic behavior. Both acoustic resonance and system impedance are functions of the entire piping system connected to the compressor, including pipe friction, interface connections, valve/ elbow locations, pipe diameter, valve coefficients, etc. Thus, a careful acoustic and impedance design review of a compressor station design should be performed to avoid impacting the operating range of the machine and to properly balance these needs against the surge control system design requirements.

NOMENCLATURE

f	=	frequency
m	=	inlet mass flow pulsation peak-to-peak magnitude
p	=	pressure
q	=	transient terms of governing equations
r	=	tip radius of the centrifugal compressor impeller
t	=	time
u	=	velocity
x	=	direction along the pipe
AF	=	amplification factor
F	=	inertial terms of governing equations
H	=	head
J	=	moment of inertia of the rotor
Q	=	bulk volume flow
U	=	acoustic volume flow
Z	=	acoustic impedance
γ	=	specific heats ratio

ϱ = density
ω = angular speed
Γ_n = head-flow curve polynomial coefficients

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