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THE IMPACT OF RECIPROCATING COMPRESSOR PULSATIONS ON THE SURGE MARGIN OF CENTRIFUGAL COMPRESSORS

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KEYWORDS

Compressor, Surge, Pulsations, Natural Gas, Unsteady Flow.

ABSTRACT

Pressure and flow fluctuations can create an oscillating operating point for a centrifugal compressor. As the operating point varies in location on the compressor map, it may pass through regions of instability resulting in the compressor experiencing multiple transient surge events. This can create operational and safety concerns, and is a particular problem in mixed compressor stations that combine flow from reciprocating compressors with headers that provide flow or discharge flow from centrifugal compressors.

Mixed compression has been a topic of concern raised and discussed in theory by multiple authors. For example, Sparks [6], Kurz et al. [5], and Brun et al. [2], provided analysis and numerical predictions on the impact of discrete and periodic pressure pulsation on the behavior of a centrifugal compressor. This interaction came to be known as the "Compressor Dynamic Response (CDR) theory." CDR theory explains how pulsations are amplified or attenuated by a compression system's acoustic response characteristic superimposed on the compressor head-flow map. Although the CDR Theory describes the impact of the nearby piping system on the compressor surge and pulsation amplification, it provides only limited usefulness as a quantitative analysis tool, primarily due to the lack of numerical prediction tools and test data for comparison. Recently, Brun et al. [2], utilized an efficient 1-D transient Navier-Stokes flow solver to predict CDR in real life compression systems. Numerical results showed that acoustic resonances in the piping system can have a profound impact on a centrifugal compressor's surge margin. However, although interesting, the fundamental problem with both Spark's and Brun's approach was that no experimental data was available to validate the analytical and numerical predictions.

In 2014, laboratory testing of reciprocating and centrifugal compressor mixed operation was performed in an air loop at Southwest Research Institute's (SwRI) compressor laboratory. The specific goal was to quantify the impact of periodic pressure and flow pulsation originating from a reciprocating compressor on the surge margin and performance of a centrifugal compressor in a series arrangement. This data was to be utilized to validate predictions from Sparks' CDR theory and Brun's numerical approach. For this testing, a 50 hp single-stage, double-acting reciprocating *compressor provided inlet pulsations into a two-stage 700 hp centrifugal compressor operating inside a semi-open recycle loop which uses near atmospheric air as the process gas. Tests were performed over a range of pulsation excitation amplitudes, frequencies, and pipe geometry variations to determine the impact of piping impedance and resonance response. Detailed transient velocity and pressure measurements were taken by a hot wire anemometer and dynamic pressure transducers installed near the compressor's suction and discharge flanges. Steady-state flow, pressure, and temperature data were also recorded with ASME PTC-10 compliant instrumentation. This paper describes the test facility and procedure, reports the reduced test results, and discusses comparisons to predictions. Results provided clear evidence that suction pulsations can significantly reduce the surge margin of a centrifugal compressors and that the geometry of the piping system immediately upstream and downstream of a centrifugal compressor will have an impact on the surge margin reduction. In severe cases, surge margin reductions of over 30% were observed for high centrifugal compressor inlet suction pulsation. Pulsation impact results are presented as both flow versus surge margin and operating map ellipses. Some basic design rules were developed from the test results to relate predicted flow pulsation amplitudes to corresponding reductions in surge margin.*

BACKGROUND

Over the last 20 years it has become a common design practice to combine the operation of centrifugal and reciprocating compressors in a single compression plant for several reasons. This arrangement can provide benefits for highly cyclical process profiles. Also the market trend on older pipelines is to add new centrifugal compressors into existing reciprocating compressor stations for capacity addition or horsepower replacement. Thus, many compressor stations now operate a mix of reciprocating and centrifugal compressors either in parallel or in series arrangement. For example, on pipeline applications, the compressors are placed in parallel operation such that a large gas turbine driven centrifugal compressor provides the base-load compression while a smaller reciprocating compressor follows cyclical or peaking demand compression demands. Other applications such as gas gathering, gas reinjection, or gas storage often have reciprocating compressors placed in series with centrifugal compressors to achieve high-pressure ratios while taking advantage of the operational flexibility of centrifugal compressors. For example, a centrifugal compressor may be placed upstream of a reciprocating compressor in gas reinjection applications to handle the higher flow volumes at low pressure while the reciprocating compressors are better equipped for high pressures and pressure ratios.

Conventional thinking was that a centrifugal compressor may experience some pulsations from the reciprocating compressor when in series or from both the common suction and discharge headers, but good reciprocating compressor bottle and manifold designs would result in minimal impact on the operational stability of the centrifugal compressor. This assumption held true for small horsepower and low speed applications and with older, high-pressure drop and often significantly oversized pulsation control systems. However, recent experience with mixed compressor stations utilizing large power and high speed modern separable reciprocating compressors with modern efficient pulsation control systems shows that the centrifugal compressor can be moved into pulsation-induced operational instability for both parallel or series arrangements. This clearly presents a station design challenge, operational range limitation, and a basic safety concern.

Interfacing a centrifugal and a reciprocating compressor within the same piping system, introduces two transient fluid mechanisms that have the potential to create damaging inlet/outlet conditions on the centrifugal compressor. These mechanisms are: (i) pulsations generated by the reciprocating compressor and (ii) pulsations generated by the reciprocating compressor that are amplified by an acoustic resonance within the piping system. Both mechanisms can move the centrifugal compressor operating point into a surge or choke (stonewall) condition and, thus, should be avoided or at least controlled. Whereas the first pulsation mechanism can be mostly analyzed using a classic acoustic pulsation analysis, for, the second mechanism the model is more complex and must include the physical and thermodynamic functions of the compressors.

The surge and stall regions on a head vs. flow map of a compressor typically define its performance. This map is critical to assess the operating range of a compressor for both steady-state and transient system scenarios. However, the compressor map is generic to any piping arrangement and does not provide a complete picture of 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 is known as the "Compressor Dynamic Response (CDR) theory." CDR theory explains how pulsations are amplified or attenuated by a compression system's acoustic response characteristic superimposed on the compressor head-flow map. Although the CDR theory explains the impact of the nearby piping system on the compressor surge and pulsation amplification, it provides only limited usefulness as a quantitative analysis tool, mainly due to the lack of computational numerical tools and benchmark test data available at the time.

Detailed descriptions of centrifugal compressor surge can be found in numerous papers. A summary of the findings of these papers can be found in Kurz and Brun [5] and in Brun and Nored [3]. On the other hand, the impact of the piping system on surge from both the impedance and acoustic response has received very little attention. The principle reference paper on the topic is over thirty years old and was published by Sparks [6]. 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. Recently, Brun et al. [2], utilized an efficient 1-D transient Navier-Stokes flow solver to predict CDR in real life compression systems. Numerical results showed that acoustic resonances in the piping system can have a profound impact on a centrifugal compressor's surge margin. However, although interesting, the fundamental problem with both Spark's and Brun's approach was that no experimental data was available to validate the analytical and numerical predictions.

CENTRIFUGAL COMPRESSOR PERFORMANCE AND SURGE

Most researchers have agreed that the actual location of a surge line is a function of the compression system and not solely a function of the compressor itself. 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 is explained by CDR theory.

When CDR theory 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 operating design standpoint, it is imperative to be able to accurately predict the interaction phenomenon. For example, 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).

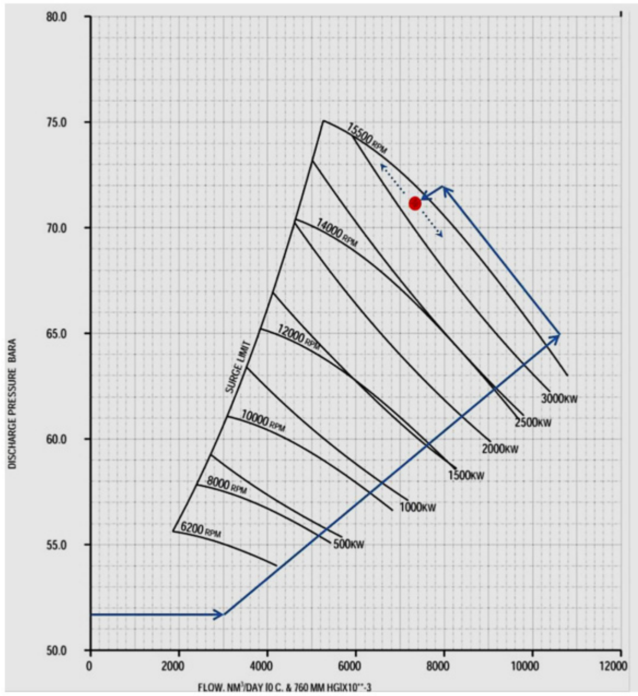

FIGURE 1. TYPICAL PIPELINE COMPRESSOR MAP AND STARTUP SEQUENCE

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. Surge margin is usually defined as:

$$SM(\%) = \frac{Q_A - Q_B}{Q_A} \cdot 100 \tag{1}$$

where Q_A is the actual volume flow at the operating point and Q_B is the flow at the surge line for the same speed line of the compressor. This should not be confused with turn-down, which is the flow difference between the operating point and surge line for the same head produced by the compressor. Throughout this paper, we will only use surge margin since all testing was done at a constant compressor speed. Most centrifugal compressor manufacturers design the machine to have at least 15% surge margin during normal operation and set a recycle valve control line at approximately 10% surge margin. That is, once surge margin falls below 10%, the recycle valve is opened to keep the compressor operating above the 10% surge margin line.

Thus, every compressor has a surge 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).

Surge is a global instability in a compressor's flow that results in a complete breakdown and flow-reversal through the compressor. 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.

This classic compressor performance 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. 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 [1]. 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. However, when the compressor experiences suction or discharge flow fluctuations are 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.

Therefore, it is clear that surge or choke can be induced in a centrifugal compressor by strong pulsations in pressure and flow. FIGURE 2 shows the velocity fluctuations in time at the inlet to a centrifugal compressor [1]. 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 corresponds to short duration surge cycles at a frequency equal to that of the inlet pulsations. As previously noted, the dynamic behavior of the compressor system near the surge line was outlined by Sparks [6], further discussed by Kurz et al. [5], and then implemented into a numerical analysis tool by Brun et al. [2]. They explained how pulsations are amplified or attenuated by a compression system's acoustic and piping response characteristic superimposed on the compressor head-flow map. Kurz et al. [5], 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.

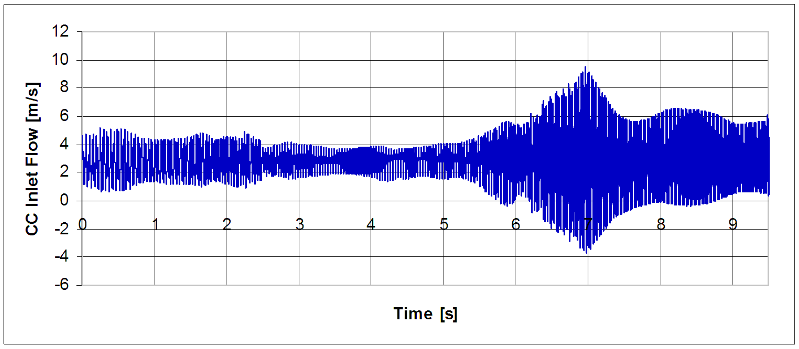

FIGURE 2. CENTRIFUGAL COMPRESSOR INLET VELOCITY VERSUS TIME [1]

Compressor Dynamic Response Theory

A piping system's acoustic impedance, Z, is given by (Equation 2):

$$Z = \frac{p}{U} = \frac{p}{uA} = \frac{z}{A} = \frac{\rho c}{A} \sim \frac{\rho u}{A} \qquad (2)$$

where Z is the acoustic impedance, A is the pipe cross section area, p is the sound pressure, U is acoustic volume flow, z is the specific acoustic impedance, ρ is the fluid density, u is the molecular particle bulk velocity, and c is the local speed of sound. This impedance relates pressure to acoustic volume flow for pressure waves traveling at the speed of sound. One should note that acoustic impedance Z, which is an extensive property, is affected by local pipe flow conditions, while the specific acoustic impedance z, which is an intensive property, is a fluid physical property for a given temperature and pressure in both incompressible and compressible flows. In pipe flow pressure pulse modulation and reflection occurs whenever there is a change of impedance Z.

A centrifugal compressor has a piping system that either attenuates or amplifies pulsations at its discharge or suction side as demonstrated by its reaction to any fluctuation in flow with a fluctuation in head. This can be easily seen from its performance characteristic in FIGURE 3 [6].

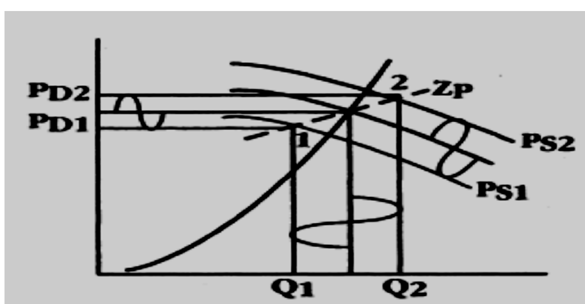

FIGURE 3. PULSATION TRANSMISSION IN CENTRIFUGAL COMPRESSORS [6]

Sparks [6] 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 differ widely from the steady-state characteristic for higher fluctuation frequencies. Sparks [6] also discusses practical piping system design approaches to reduce pulsation levels using acoustic elements such as bottles, nozzles, choke-tubes, and resonators.

Brun et al. [2], further explains that for fast transient pressure pulses in the acoustic range, the pipe flow resistance curve is not applicable, and the impedance curve must be utilized to analyze the compressor operating point on the compressor performance map. Based on this analysis, the surge line can be crossed when short pressure pulses result in flow variation due to low gradient (“flat”) impedance curves as shown in FIGURE 4. Here, a short suction pressure pulse interacts with the pipe’s flat impedance line to cause a large flow fluctuation that crosses the surge line momentarily and results in an onset of a surge event.

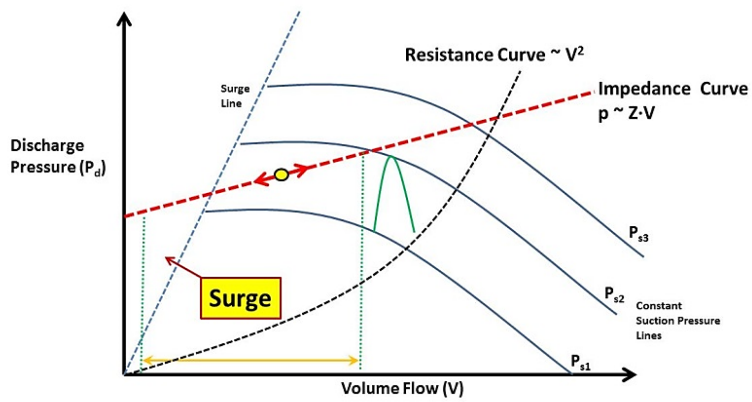

FIGURE 4. FLAT IMPEDANCE LINE RESULTS IN SUCTION PRESSURE PULSE CAUSING SURGE

Brun et al. [2], also validate CDR, numerically showing that pulse amplification or attenuation for centrifugal compressors is primarily a function of the piping upstream and downstream impedance. Piping impedance can be controlled by varying pipe diameter and pressure drop elements in the flow path as shown in FIGURE 5.

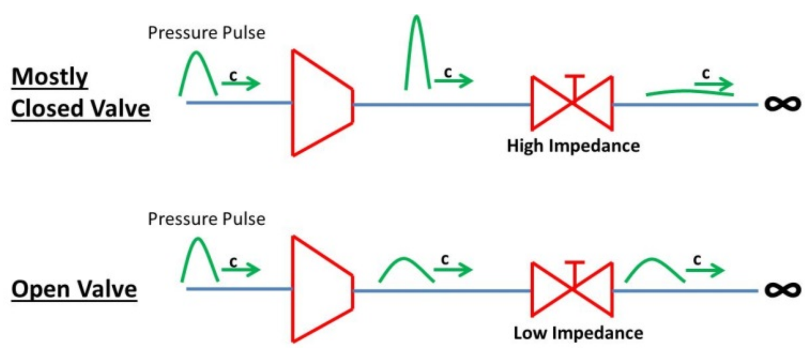

FIGURE 5. DOWNSTREAM PIPE IMPEDANCE DETERMINES PRESSURE PULSE AMPLIFICATION

A number of other studies are available in the public domain that discuss centrifugal compressor surge and the impact of flow pulsations on turbomachines. A detailed literature review on this topic was provided by Kurz, et al. [5], Brun and Nored [3], and an overview of the state-of-the-art of pulsation analysis technologies was included in Brun et al. [4].

TESTING OBJECTIVES

Due to the lack of numerical prediction tools and test data, CDR theory can only describe the impact of the nearby piping system on the compressor surge and pulsation amplification. Therefore, it provides only limited usefulness as a quantitative analysis tool. As discussed above, more recently Brun et al. [2], utilized an efficient 1-D transient Navier-Stokes flow solver to predict CDR in real life compression systems. However, although interesting, the fundamental problem with both Spark's and Brun's approach was that no experimental data was available to validate the analytical and numerical predictions.

Thus, while many compressor stations have both reciprocating and centrifugal compressors installed, there is still a limited understanding of how pulsations, piping resonance, and impedance impact centrifugal compressor performance and surge. Some analytical and computational predictions exist, but there is no test data available to validate them. Consequently, current compressor station design practices are limited because of lack of knowledge and data. The fundamental questions that must be answered are: 1. Can pulsations drive a centrifugal compressor into surge? 2. If so, what amplitudes and frequencies are required?

Laboratory testing of reciprocating and centrifugal compressor mixed operation was performed in a full scale laboratory setting to overcome these technology gaps and a deficiency of fundamental physical understanding. The specific goal was to develop benchmark data that quantifies the impact of periodic pressure and flow pulsation originating from a reciprocating compressor on the surge margin and performance of a centrifugal compressor. This data was then utilized to validate predictions from Sparks' CDR theory and Brun's numerical approach. For this testing, a single-stage reciprocating compressor provides inlet pulsations to a two-stage centrifugal compressor operating inside a semi-open recycle loop and utilizing atmospheric air as the process gas. Tests were performed over a range of pulsation excitation amplitudes, frequencies, and pipe geometry variations to determine the impact of piping impedance and resonance response. Although the testing was performed for a series arrangement, the results are equally applicable to mixed station parallel arrangements. Detailed transient velocity and pressure measurements were taken with a hot wire anemometer and dynamic pressure transducers installed near the compressor's suction and discharge flanges. Steady-state flow, pressure, and temperature data was also recorded with ASME PTC-10 compliant instrumentation. This paper describes the test facility and procedure, reports the reduced test results, and discusses the comparison to predictions. From the test results, some basic design rules were developed to relate surge margin reduction to flow pulsation amplitudes.

In summary, the primary objectives of this research were to:

- Determine whether pulsations from reciprocating compressors, vortex-shedding, or other sources can reduce the surge margin or even cause surge in centrifugal compressors.
- Develop an understanding of the physical process that causes pulsation-induced centrifugal compressor surge.
- Determine the amplitude and frequency of pulsations required to cause centrifugal compressor surge.
- Develop a simple physical relationship or engineering guideline for pulsation-induced surge avoidance.
- Determine the impact of pulsations on the compressor's performance.
- Evaluate the impact and interaction of acoustic pipe resonances and impedance on pulsation-induced centrifugal compressor surge.

- Validate CDR theory [6] and numerical modeling approach [2] predictions for pulsation amplification in centrifugal compressors.

MIXED COMPRESSION TEST LOOP

Laboratory testing of reciprocating and centrifugal compressor mixed operation was performed in an air loop at the SwRI compressor laboratory. FIGURE 6 shows a schematic of the test arrangement and photos of the compressors. The facility allows for open, semi-open, and closed loop operation with either or both reciprocating and centrifugal air compressors arranged in series or parallel operation. Semi-open loop operation was chosen for the subject testing to best control flow and pressures at low loop process pressures.

The testing was performed using a 50 hp single-stage, double-acting reciprocating compressor mounted upstream of a two-stage 700 hp centrifugal compressor. The reciprocating compressor suction was open to atmosphere with an operating range of 300 to 1,000 rpm (5-17 Hz) using a variable frequency driver. Similarly, the centrifugal compressor was operating in a semi-open recycle loop with the loop's discharge throttled back to atmospheric pressure, a speed range of 2,000 to 14,000 rpm, and maximum pressure ratio of 3:1. However, for safety reasons, the centrifugal compressor's speed and discharge pressure were limited to 7,000 rpm and 2 bar (30 psi), respectively, for the subject test series. The centrifugal compressor's normalized performance map is shown in FIGURE 7.

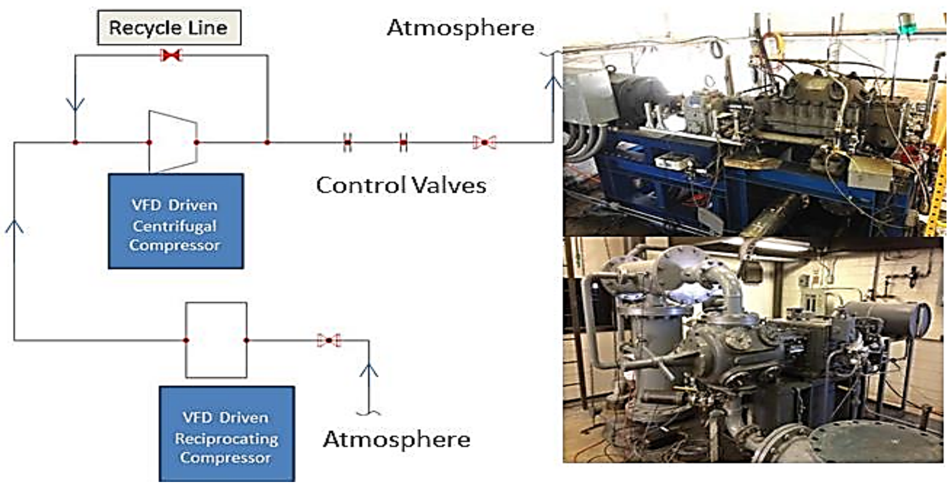

FIGURE 6. SCHEMATIC OF THE TEST LOOP ARRANGEMENT

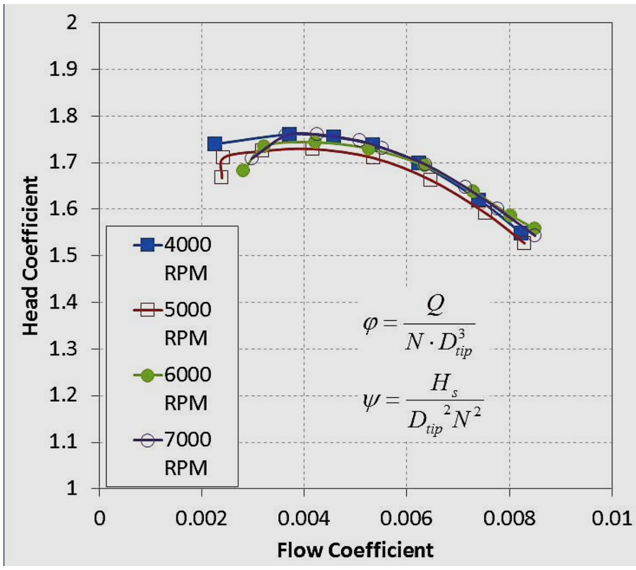

FIGURE 7. CENTRIFUGAL COMPRESSOR NORMALIZED PERFORMANCE MAP

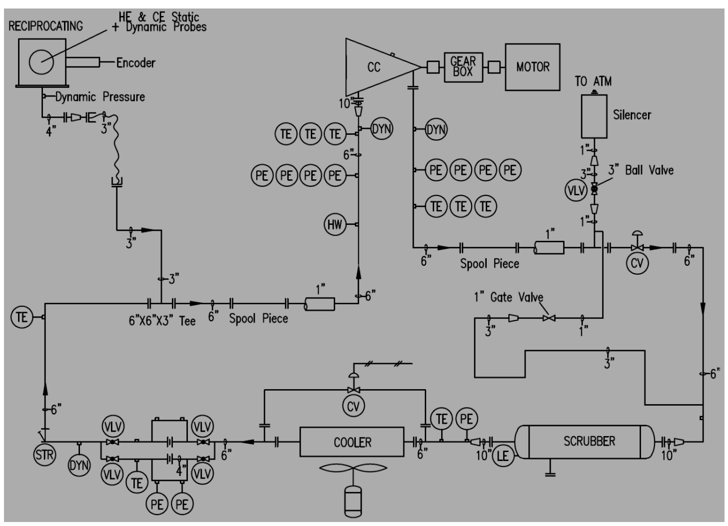

FIGURE 8. PROCESS AND INSTRUMENT DIAGRAM OF TEST LOOP

Since the raw test data and reduced results produced herein are intended to be open to the industry for benchmarking, code validation, and design comparison, a detailed test loop process and instrumentation diagram (P&ID) is included in FIGURE 8. The piping distance between the reciprocating and centrifugal compressor loop was 17 m (55 ft), the recycle line length was 25 m (80 ft), and the atmospheric discharge line length was 3 m (10 ft). The loop also included an air-to-air cooler and a discharge scrubber. Pipe diameters and critical dimensions are indicated on FIGURE 8.

To reduce overall measurement uncertainty (as can be seen in the P&ID in FIGURE 8), flow, pressure, and temperature measurements were installed in the test loop per ASME PTC-10 requirements. Instrumentation included four (4) static pressure and temperature measurements on each of the compressor's suction and discharge sides, six (6) dynamic pressure transducers mounted throughout the loop, steady-state flow measurement from an orifice plate meter in the main and recycle loop, reciprocating compressor flow and power from a PV card measurement, centrifugal compressor power from a shaft torque meter and enthalpy rise analysis, and transient inlet flow measurement using a high speed hot wire anemometer near the compressor's suction flange. All individual instruments were calibrated prior to the test to manufacturer's accuracy requirements. End-to-end calibrations with all data acquisition system included were also performed. Total measurement and data acquisition uncertainties were predicted using the perturbation method and validated versus data scatter. Typical uncertainties for raw and reduced parameters are shown in TABLE 1 as percent deviation from the operating point.

TABLE 1. PERCENT MEASUREMENT UNCERTAINTIES FOR RAW AND REDUCED PARAMETERS

	Pressure	Temperature	Flow	Torque	Power	Efficiency	Surge Margin
Steady	0.3	0.2	0.5	0.1	0.5	0.4	0.6
Transient	0.5	-	1.2	0.2	1.6	0.9	1.4

RESULTS

Since the primary objective of the project was to determine the impact of pulsation on the surge margin of the compressor, accurately identifying the onset of surge is imperative. On most centrifugal compressors, surge symptoms include high axial rotor vibrations and suction/discharge cyclic pressure fluctuations at low frequencies. Previous experiments on the subject compressor had demonstrated that the compressor surges at a frequency of approximately 4 Hz. Thus, a series of tests were performed to validate the surge line of the compressor and to establish clear instrument measurement output criteria indicating that the machine is operating in surge.

COMPRESSOR MAP VALIDATION

All surge tests were performed with the centrifugal compressor operating at 7,000 rpm and near ambient suction pressure to minimize the risk of damage to the compressor during the anticipated surge events. For example, FIGURE 9 shows 4 Hz axial rotor vibration measured by proximity probes for no-surge versus surge operating conditions of the compressor. Clearly, a significant rise in 4 Hz vibration amplitudes is observed. Similar results showing significantly elevated surge pressure pulsations at the characteristic 4 Hz were seen from the suction and discharge dynamic pressure transducers. Thus, for all test series, the onset of surge criteria was established to be a rapid increase in 4 Hz compressor axial vibrations and suction/discharge flange dynamic pressure pulsations.

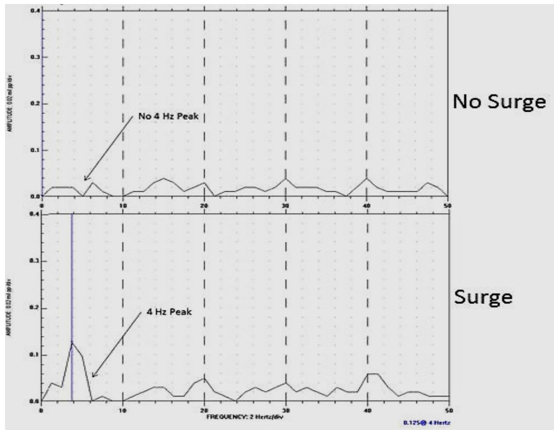

FIGURE 9. AXIAL COMPRESSOR VIBRATIONS BEFORE AND IN SURGE CONDITIONS

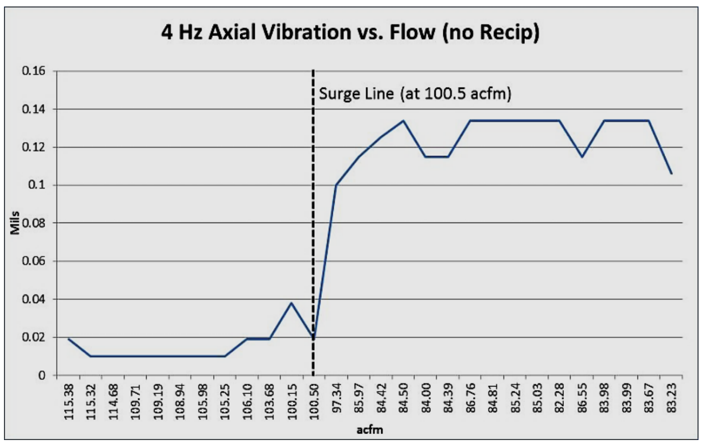

FIGURE 10. AXIAL VIBRATIONS VERSUS FLOW AT THE 7,000 RPM SPEED LINE

FIGURE 10 shows axial vibrations as a function of flow measured by the hot-wire anemometer upstream of the compressor. The centrifugal compressor is seen to surge when the flow reaches approximately 100.5 acfm. Based on the surge vibration and pulsation criteria, the compressor map and surge line were thus established as shown in FIGURE 11. However, one should note that based on changing ambient conditions, the steady-state surge line varied throughout the testing and had to be re-measured for every single test series.

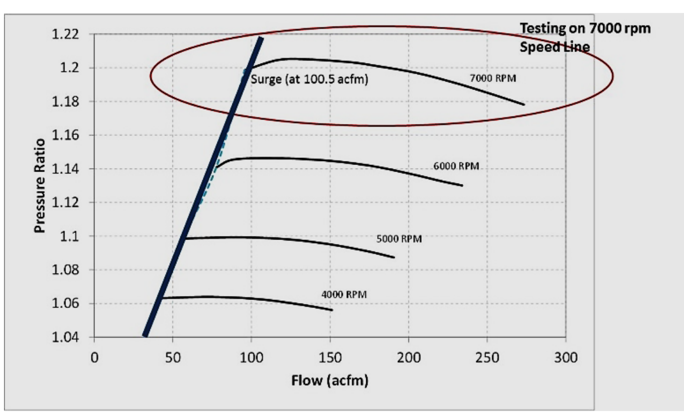

FIGURE 11. COMPRESSOR PERFORMANCE MAP WITH ACTUAL SURGE LINE

EFFECT OF PULSATING FLOW ON CENTRIFUGAL COMPRESSOR PERFORMANCE

To determine the impact of externally induced pulsations on the surge margin of the centrifugal compressor, the reciprocating compressor is operated upstream in series with the centrifugal compressor. The pulsations from the reciprocating compressor thus travels through 55 ft of varying diameter pipe as shown on the P&ID before they enter the centrifugal compressor suction flange. Some pulsation attenuation or acoustic amplification will occur in this pipe segment and must be quantified for the analysis. Time domain pulsations were measured at the reciprocating compressor discharge, the centrifugal compressor suction, and the centrifugal compressor discharge.

For the testing, the centrifugal compressor was initially operated at a stable point with approximately 15% surge margin to the right of the surge line. The reciprocating compressor was then started, brought to a fixed speed, and its pressure and flow pulsation amplitudes were recorded at the centrifugal compressor suction flange using dynamic pressure transducers and the hot wire anemometer, respectively. The centrifugal compressor loop recycle valve was then slowly closed until surge was clearly indicated by axial vibrations and suction/discharge pulsations. Only when the previously established surge criteria were fully met, was the pulsation-induced surge point recorded. The difference between the previously recorded steady-state surge point (from the performance map) and the pulsation-induced surge point was used to calculate the difference in surge margin.

FIGURE 12 shows a typical test run with the reciprocating compressor running. Pressure fluctuations and compressor inlet flow are plotted versus test recording time. In this case, the 4 Hz discharge pulsations are used as an indicator for the onset of surge. The test started at 1:55:44 pm with the recycle valve being slowly closed, decreasing the flow into the centrifugal compressor (red line). At the time of 2:00:53 pm, the 4 Hz discharge volute pulsations are seen to increase which corresponds to a flow of 106.5 acfm, about 6 acfm away from the steady-state surge line. As the recycle valve is further closed, and the flow is further decreased, the pulsations are seen to increase, approximately linearly with flow reduction. A drastic 4 Hz pulsation increase can then be observed when the flow reaches the steady-state surge line at 100.5 acfm. In this particular case, a gradual onset of surge can be observed well before the machine is operating near the surge line. This provides evidence that periodically unsteady flow can induce surge in a centrifugal compressor when the compressor is seemingly still operating to the right of the surge line on the performance map.

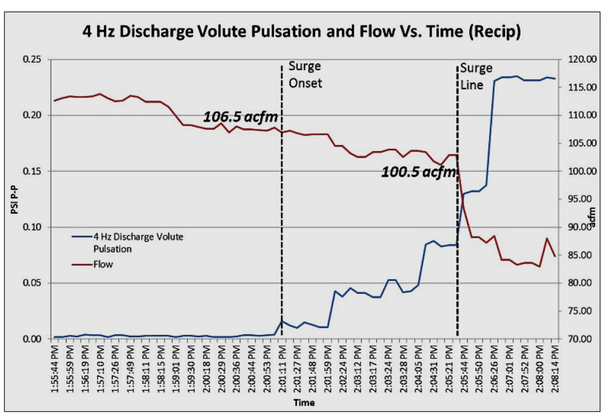

FIGURE 12. TEST RESULTS OF 4 HZ PULSATIONS AND FLOW VERSUS TEST TIME

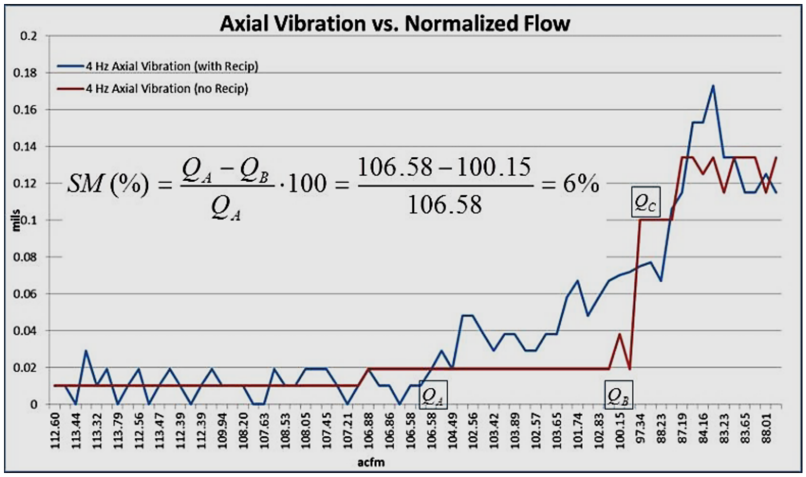

FIGURE 13. AXIAL VIBRATION VERSUS FLOW SHOWING DIFFERENCE IN SURGE ONSET FOR CASES WITH AND WITHOUT SUCTION PULSATIONS

A direct comparison of the onset of surge and surge margin for cases with and without the reciprocating compressor running can thus be made. For example, FIGURE 13 shows the axial vibrations versus flow for cases with and without the reciprocating compressor running. The axial vibrations in the case with the reciprocating compressor running start rising at a much higher flow (106.6 acfm) than the case without the reciprocating compressor running (100.2 acfm). The difference in surge margin is thus approximately 6% between the two cases. This difference between steady-state and pulsating flow surge margin is the pulsation-induced surge margin differential. The surge margin differential corresponds to an actual reduction of compressor operating range and can severely impact a mixed compressor stations operability and safety.

OPERATING MAP ELLIPSE

For the 106 acfm surge onset case (with the reciprocating compressor running), the pulsations entering the centrifugal compressor just before the onset of surge were measured to be 44 acfm peak-to-peak in flow and 0.058 psi peak-to-peak in pressure. One can thus plot an ellipse of the actual operating point of the centrifugal compressor based on its fluctuating suction flow and pressure conditions as shown in FIGURE 14. This operating map ellipse defines the cyclically unsteady operating point of the compressor on its performance map. Specifically, the ellipse closely represents the true operating range of the compressor on the compressor map, since with periodically transient cyclic suction/discharge operating pressures, the compressor operating point will never be on its time-averaged or steady-state operating point. One should note that pulsations from a reciprocating compressor are not perfectly sinusoidal. The actual shape of the periodic operating cycle will typically not be a perfect ellipse but more irregular, depending on the higher order frequency content of the pulsations. Nonetheless, the assumption of an elliptical shape significantly simplifies further analysis and was found to not introduce significant error when used as a design tool.

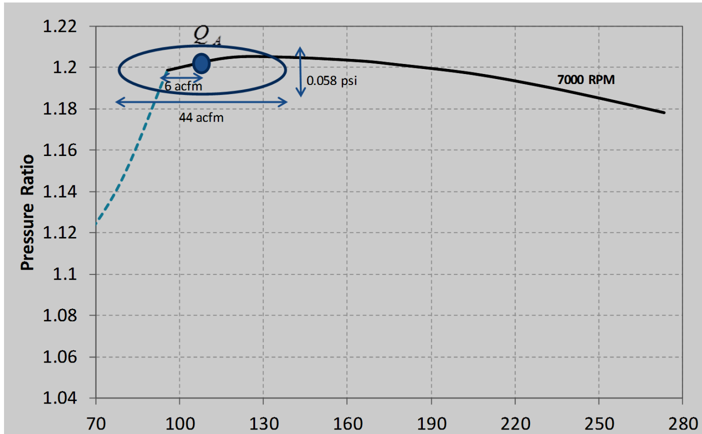

FIGURE 14. PULSATION-INDUCED OPERATING CYCLE ELLIPSE VERSUS ACTUAL SURGE MARGIN REDUCTION

From FIGURE 14, one can clearly see that, although pulsation amplitudes in flow (44 acfm) indicates that the compressor should have surged at the much higher flow of approximately 122 acfm, truly measurable surge occurred only around 106 acfm. Specifically, the surge margin reduction caused by the flow pulsation is significantly lower than the flow fluctuation magnitude. Similar trends were found for all cases measured. The cause for this is still under investigation but may be attributable to higher order frequency induced pulsations not having sufficient transient time to affect the low frequency surge cycle. Further discussion of this phenomenon is provided below.

The operating map ellipse can be further refined by showing the different pulsation frequency orders as shown in FIGURE 15. In this example, the reciprocating compressor running speed was 615 rpm, the excitation frequency was 10.25 Hz, and a surge margin differential of 41.2% was found. Clearly, the compressor surged before it had reached its peak pressure ratio on the 7000 rpm speed line. The 1st order ellipse shows the induced pulsation only from the pulsations at the 10.25 Hz operating frequency (running speed), the 1st and 2nd order ellipse show energy from both 1st and 2nd orders (up to 20.5 Hz), and the 1st, 2nd, and 3rd order ellipse show energy up to 30.75 Hz for all three orders. Pulsations from reciprocating compressors have significantly higher order frequency content when the cylinders are double-acting since the pulse wave form is not sinusoidal. Thus, in this case, most pulsation energy was in the 2nd order as can be seen by the much larger 1st and 2nd order operating map ellipse. The energy from the 3rd order does not significantly increase the radius of the operating map ellipse (1st, 2nd, and 3rd) as it does not contain significant pulsation energy. Fourth and higher order frequency energy content was also found to be negligible. In most cases, for reciprocating compressors operating below 1,500 rpm, almost all surge-relevant pulsation energy is contained in the frequency orders below 90 Hz.

One should also note that the local impedance line can be directly determined from the operating map ellipse as shown in FIGURE 15 (red line). The fundamental definition of acoustic impedance, as shown in Equation 2, relates bulk acoustic flow to pressure rise for pulsations at acoustic frequencies (i.e., at the speed of sound). Thus, the slope of the impedance line must be equal to the aspect ratio of the operating map ellipse.

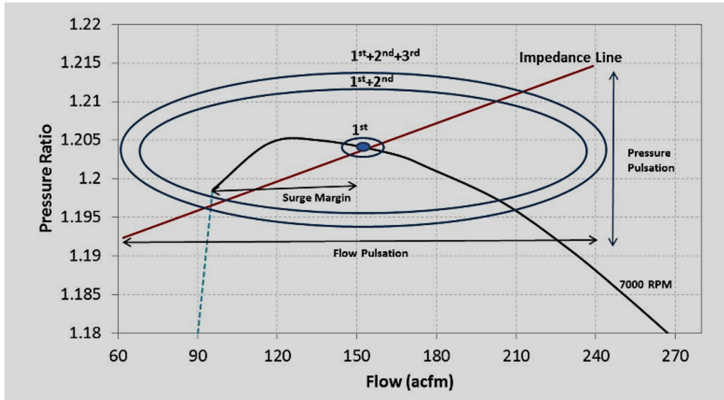

FIGURE 15. OPERATING MAP ELLIPSES FOR DIFFERENT PULSATION FREQUENCY ORDERS

RESULTS AND ANALYSIS

The above described tests were performed to determine the impact of pulsation on the surge margin (surge margin differential) of the centrifugal compressor for a number of operating conditions, pulsation amplitudes, pulsation frequencies, and piping suction/discharge geometry changes. For most of the tests, the compressor suction conditions were held at near ambient conditions with inlet temperatures varying between 70 and 90 degrees Fahrenheit. Because of the changing inlet temperatures and associated air density variations, the surge line had to be retested and validated for every test series.

To test varying pulsation amplitudes on the centrifugal compressor and to determine the impact of piping acoustic resonances, measurements for surge margin and operating map ellipses were performed at four (4) different reciprocating compressor speeds. Also, to increase and vary the pulsation amplitudes into the centrifugal compressor, three (3) of these operating points corresponded to piping acoustic resonance frequencies. Suction piping resonances were found at the 405 rpm (6.75 Hz), 480 rpm (8 Hz), and 615 rpm (10.25 Hz) reciprocating compressor operating speeds. An additional test point at 310 rpm (5.17 Hz), corresponding to a non-resonant operating condition, was also tested as a low amplitude pulsation test case. For example, FIGURE 16 shows the suction piping pressure spectrum and acoustic resonances measured at the centrifugal compressor inlet for the 615 rpm (10.75 Hz) running case. Since the compressor was double-acting, most of the pulsation energy is seen on the second running order at 21.5 Hz. Only limited energy is seen in the 1st, 3rd, and 4th order frequencies. The other two resonance cases are also shown.

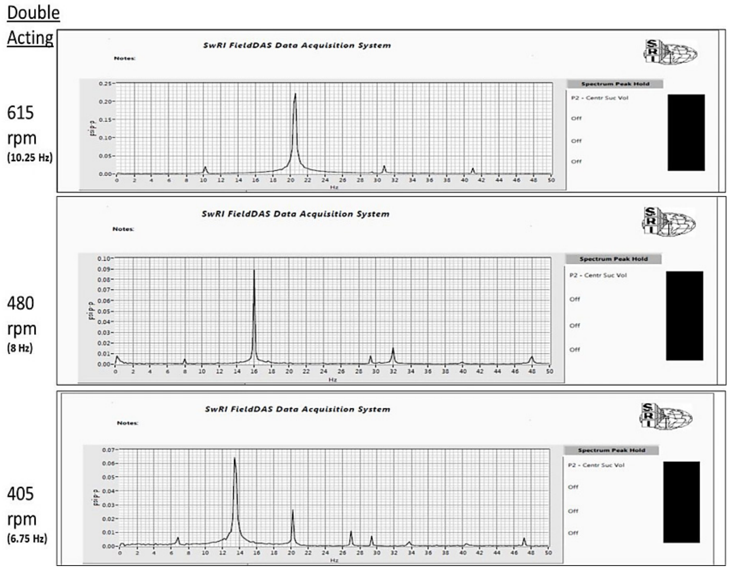

FIGURE 16. PULSATION FREQUENCY SPECTRUM AT THE CENTRIFUGAL COMPRESSOR INLET

TABLE 2 and TABLE 3 summarize the results from the four (4) different cases that were tested and include the flow and pressure pulsation levels at both the reciprocating compressor exit and centrifugal compressor inlet. For these cases the centrifugal compressor was running at 7,000 rpm and the reciprocating compressor speeds are indicated in the tables. Normalized results are also shown (see definition of head and flow coefficient in FIGURE 7).

TABLE 2. IMPACT OF ACOUSTIC RESONANCE: PULSATIONS AND FLOW FLUCTUATIONS

	Condition	1	2	3	4
	Recip Speed (RPM)	310 (5.17 Hz)	405 (6.75 Hz)	480 (8 Hz)	615 (10.25 Hz)
At Recip Discharge	Excitation pulsations (psi pk-pk)	0.37	0.65	0.55	0.88
At Centrifugal Suction	Excitation Pulsations (psi pk-pk) [normalized]	0.15 [0.0298]	0.11 [0.0158]	0.13 [0.02252]	0.38 [0.10699]
	Flow fluctuations (p-p acfm) [normalized]	44.9 [0.0014]	90 [0.0028]	110 [0.0034]	250 [0.0078]

In TABLE 3 the pressure and flow results are converted into surge margin differential and percent area across the surge line.

TABLE 3. EFFECT OF PRESSURE AND FLOW FLUCTUATIONS ON SURGE

Recip Speed (RPM)	405 (6.75 Hz)	480 (8 Hz)	615 (10.25 Hz)
Excitation Pulsations (psi pk-pk) [Pressure ratio fluctuations]	0.11 [0.0091]	0.13 [0.0108]	0.38 [0.0315]
Flow fluctuations (p-p acfm) [normalized]	90 [0.0028]	110 [0.0034]	250 [0.0078]
Flow at Surge (acfm)	102.5	121.0	150.8
Pressure Ratio at Surge	1.201	1.204	1.204
Surge Margin Differential %	7.9	24.3	41.2
% Area Across Surge Line	31	29	31

To evaluate the impact of suction/discharge piping impedance changes, for each of the four (4) running speed cases (5.17 Hz, 6.75 Hz, 8 Hz, and 10.25 Hz), six (6) different piping system geometries were tested. The piping geometry changes involved replacing 1.2 m (4 ft) segments of suction or discharge pipe with different diameter pipes of 7.6 cm, 10.2 cm, 15.2 cm (3 inch, 4 inch, and 6 inch) spool pieces. For the 5.17 Hz case, only the 6 inch suction and discharge piping case was tested. Thus a total of 16 operating conditions were tested. TABLE 4 shows the matrix of test cases and their test number.

TABLE 4. MATRIX OF CONDITIONS AND PIPING GEOMETRIES TESTED

		SPEED			
		5.17 Hz	6.75 Hz	8.00 HZ	10.25 Hz
Suction/ Discharge Piping	6 inch / 6 inch	1	2	3	4
	6 inch / 4 inch	-	5	6	7
	6 inch / 3 inch	-	8	9	10
	4 inch / 6 inch	-	11	12	13
	3 inch / 6 inch	-	14	15	16

EXPERIMENTAL TEST DATA

The raw data from tests 1 through 16 was reduced to determine pulsating flow, pulsating flow coefficient, pulsating pressure, pulsating pressure ratio, pulsating head coefficient, operating map ellipse, surge margin differential, area of operating map crossing the surge line, pulsation amplification/attenuation, steady head and flow coefficient, compressor power, compressor efficiency, impedance slope, etc. It is beyond the scope of this paper to present all this data. However, since the raw test data and reduced results produced herein are intended to be open to the industry for benchmarking, code validation, and design comparison, TABLE 5 shows test results for the most critical parameters: Pulsation pressure amplitude, pulsation flow amplitude, surge margin differential, and pulsation amplification/attenuation.

As shown by the data in the table, for very high pulsation amplitudes, surge margin reductions above 40% were seen. However, for more moderate pulsation levels, similar in range to pulsation to mean line pressure ratios found in industrial compressor station applications, surge margin reductions were closer to 15-25%. The flow and pressure pulsation data in TABLE 5 can be utilized to determine the operating map ellipse and the area of the ellipse that crosses the surge line. For all test cases, this area was found to be approximately 30-35% of the total operating map ellipse's area.

TABLE 5. RESULTS FROM TESTS 1 THROUGH 16

	Test No	Recip Speed (RPM)	Excitation Pulsations (psi pk-pk) at Recip Discharge	Excitation Pulsations (psi pk-pk) at Centrifugal Suction (0-90 Hz)	Flow fluctuations at Centrifugal Suction (p-p acfm)	Surge Margin Differential %	Compressor Pulsation Amplification or Attenuation Factor
6/6 Suction/ Discharge	1	**310 (5.17 Hz)**	**0.37**	**0.154**	**44.9**	**6.1**	**0.56644**
	2	**405 (6.75 Hz)**	**0.65**	**0.167**	**209**	**7.9**	**0.93785**
	3	**480 (8 Hz)**	**0.55**	**0.186**	**153**	**24.3**	**1.40769**
	4	**615 (10.25 Hz)**	**0.88**	**0.326**	**376**	**41.2**	**3.69318**
6/4 Suction/ Discharge	5	**405 (6.75 Hz)**	**1**	**0.171**	**213**	**6.8**	**0.69388**
	6	**480 (8 Hz)**	**0.66**	**0.176**	**174**	**17.8**	**1.75000**
	7	**615 (10.25 Hz)**	**2.04**	**0.417**	**407**	**39.8**	**5.12346**
6/3 Suction/ Discharge	8	**405 (6.75 Hz)**	**1.1**	**0.196**	**203**	**8.2**	**1.30000**
	9	**480 (8 Hz)**	**0.55**	**0.196**	**182**	**8.2**	**1.95000**
	10	**615 (10.25 Hz)**	**2.25**	**0.407**	**391**	**27.3**	**2.48466**
4/6 Suction Discharge	11	**405 (6.75 Hz)**	**1.03**	**0.152**	**297**	**19.6**	**0.61633**
	12	**480 (8 Hz)**	**0.575**	**0.21**	**294**	**26.2**	**1.11702**
	13	**615 (10.25 Hz)**	**2.2**	**0.72**	**394**	**40.1**	**1.07692**
3/6 Suction/ Discharge	14	**405 (6.75 Hz)**	**1.13**	**0.159**	**336**	**35.7**	**0.74528**
	15	**480 (8 Hz)**	**1.03**	**0.22**	**371**	**42.9**	**0.80769**
	16	**615 (10.25 Hz)**	**2.35**	**0.34**	**396**	**42.9**	**2.55814**

Impact of Pulsation Amplitude on Surge Margin

The impact of pulsation amplitudes on compressor stability can be illustrated by comparing the pulsation flow coefficient, $\Delta\varphi$, as defined in Equation 3, to the reduction of surge margin (surge margin differential) of the centrifugal compressor.

$$\Delta\varphi = \frac{\Delta Q}{N \cdot D_{tip}^{3}} \tag{3}$$

Here, ΔQ is the pulsation flow amplitude, N is the running speed of the compressor and D_{tip} is the tip diameter of the compressor impeller. Similarly, a pulsation head coefficient, $\Delta\psi$, can be defined as:

$$\Delta\psi = \frac{\Delta H_s}{D_{tip}{}^{2} N^{2}} \tag{4}$$

where ΔH_s is the pulsation head amplitude. Results recording the pulsation flow coefficient versus surge margin differential for tests 1 through 4 are shown in FIGURE 17. As anticipated, the surge margin reduction appears to be a near linear function versus pulsation amplitude. Increasing pulsations will increase the radius of the operating map ellipse and thus, decrease the distance between transient operating points and the surge line. Tests 5 through 16 showed very similar results and trends.

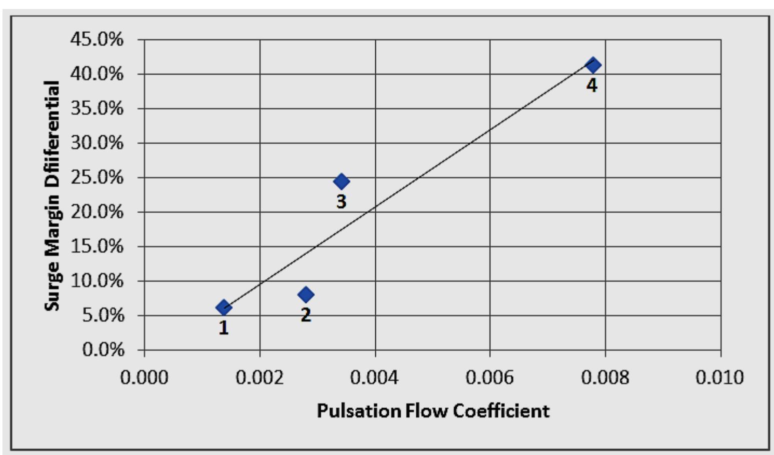

FIGURE 17. PULSATION FLOW COEFFICIENT VERSUS SURGE MARGIN REDUCTION

One should note that pulsation pressure amplitude or pulsation head coefficient could also be plotted versus surge margin reduction and the trends would look identical. The reason for this is that pulsation pressure and flow are directly related by impedance as shown in Equation 2. The piping impedance in this test series (1 through 4) was not changed, so the relationship between pulsation pressure and volume flow stays constant.

FIGURE 17 also demonstrates that when the reciprocating compressor is running on or near acoustic piping resonance frequencies, the pulsation amplitudes into the centrifugal compressor will be changed and thus, the surge margin differential will be affected. Specifically, test cases 1 through 4 correspond to reciprocating compressor running speeds on three (3) piping acoustic resonance frequency (405 rpm/6.75 Hz, 480 rpm/8 Hz, 615 rpm/10.25 Hz) and one non-resonant frequency (615 rpm/10.75 Hz). Clearly, each one produces a different amplitude pulsation and different surge margin reduction. When designing a mixed compressor station, it is important to keep this in mind, as the worst case surge margin reduction scenarios will likely occur for cases where the reciprocating compressor is running on an amplifying acoustic piping resonance.

Finally, the pulsation flow coefficients versus surge margin curve as shown in FIGURE 17 was compared to numerical results presented by Brun et al. [2]. They modeled the impact of various levels of periodic pulsations on the surge margin of an industrial centrifugal compressor operation at 14,400 rpm and 52.4 bara suction pressure inside a 20 m recycle loop. Although speed, pressures, temperatures, and gas compositions of the compressor modeled by Brun et al. [2], were significantly different than the cases tested herein, the predicted impact on surge margin differential versus pulsation flow coefficient are similar. Specifically, for their modeled case, Brun et al. [2], predicted a surge margin differential of 17% for an equivalent pulsation flow coefficient of 0.0039. This is within 15% of the curve shown in FIGURE 17. However, one should not assume that this one good correlation between test data and numerical predictions (for these very different operating cases) allows for a broad generalization of the test results from FIGURE 17. They are specific for the geometry and operating conditions described herein only.

Impact of Piping Impedance on Surge Margin and Pulsation Amplification

CDR theory [6] and numerical results from Brun et al. [2], predict that changing the acoustic suction or discharge impedance of the pipe will affect the level of modulation of a pulse as it travels through the centrifugal compressor. That is, based on CDR, one can predict whether a compressor and piping system will either amplify or attenuate an externally induced pulse. Tests 5 through 16 were primarily intended to validate or disprove this theory.

An easy method to alter the piping impedance is to change its internal flow area. Tests 5 through 10 changed the discharge piping diameter area of the centrifugal compressor and its impedance by replacing the original 6 inch pipe with smaller 4 and 3 inch spool pieces. Similarly, tests 11 through 16 changed the suction piping impedance by replacing the 6 inch suction piping with 4 and 3 inch spool pieces. Since pipe acoustic impedance increases linearly with decreasing through flow area, it has a square functional relationship with decreasing pipe diameter. As explained above, the piping impedance line slope can be calculated from the aspect ratio of the operating map ellipse. TABLE 6 shows the impedance line slope for the different suction and discharge pipe diameter test cases. Here the impedance line is based on the dimensional ratio of pressure (psi) divided by volume flow (acfm).

TABLE 6. IMPEDANCE, Z (PSI/ACFM), LINE SLOPE

	Pipe Diameter		
	6/6 inch	**6/4 inch**	**6/3 inch**
Suction	0.0001	0.0003	0.0006
Discharge	0.0001	0.0002	0.0005

The primary effect of a compressor's associated piping system impedance is on the amplification or attenuation of periodic pulses through a centrifugal compressor. This indirectly impacts the surge margin, as larger pulses tend to decrease the surge margin differential. FIGURE 18 shows pulsation amplification through the centrifugal compressor versus discharge piping impedance slope for the 405 rpm (6.75 Hz) running case. For the 0.0001 and 0.0002 impedance slope cases, the pulsations are attenuated by factors of 0.7 and 0.9, respectively. For the highest impedance slope case (0.0005) the pulsations are amplified by a factor of 1.2. Thus, consistent with CDR and numerical predictions, an increase in the discharge pipe impedance line slope will result in higher discharge pulsations out of the centrifugal compressor. Similar trends were observed for the other running speed cases.

The above finding can provide some compressor station design guidance: To minimize the risk of pulsation amplification through a centrifugal compressor, the downstream piping should be designed with the flattest possible impedance slope. However, a flat impedance slope can also result in the highest conversion of pressure to flow fluctuations and thus the greatest risk for pulsation-induced surge. Furthermore, low impedance piping often corresponds to large downstream compressor volumes which are undesirable from a transient ESD surge perspective. A careful compressor station piping design must balance these opposing requirements for mixed flow applications.

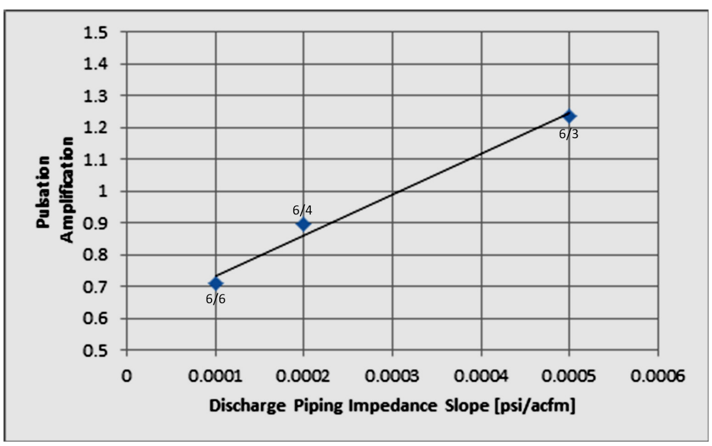

FIGURE 18. PULSATION AMPLIFICATION VERSUS DISCHARGE IMPEDANCE SLOPE

The impact of suction piping impedance changes on the surge margin differential is shown in FIGURE 19 for the 480 rpm (8.00 Hz) running case. Here the surge margin differential is decreasing with increasing impedance slope. The cause for this is that with a flat impedance slope, more pulse pressure is converted to pulse flow fluctuations which decreases the distance between the operating point and the surge line. Thus, the steeper the impedance slope on the compressor suction side, the lower the surge margin differential. This trend was consistent for all running cases with suction piping changes.

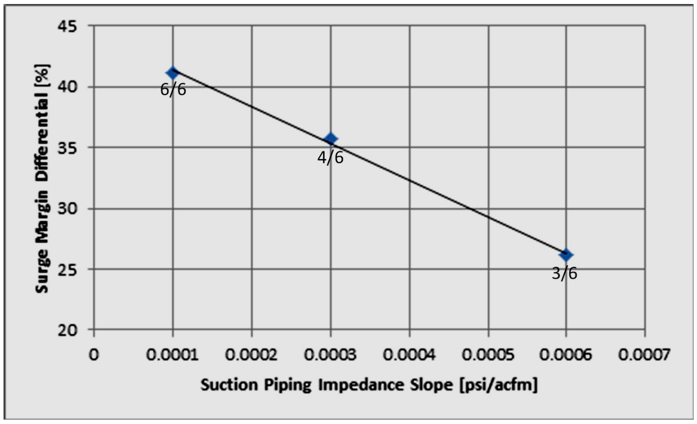

FIGURE 19. SURGE MARGIN DIFFERENTIAL VERSUS SUCTION IMPEDANCE SLOPE

The above test findings can be summarized into trends as shown in TABLE 7. Here ↑ is for increasing, ↓ is for decreasing. The number of arrows corresponds to weak, medium, and strong effects.

TABLE 7. IMPACT OF SUCTION/DISCHARGE PIPING IMPEDANCE ON PULSE AMPLIFICATION AND SURGE MARGIN

	Increasing ("Steeper") Impedance Slope	Decreasing ("Flatter") Impedance Slope
Suction Piping	**↓↓↓ Surge Margin Differential**	**↑↑↑Surge Margin Differential**
	↑ Pulse Amplification	**↓ Pulse Amplification**
Discharge Piping	**↓↓ Surge Margin Differential**	**↑↑ Surge Margin Differential**
	↑↑↑ Pulse Amplification	**↓↓↓ Pulse Amplification**

Although the information in the above table can be used to guide the design process, it does not replace the need to perform a proper pulsation analysis for mixed compressor stations.

Comparison to 1-D Transient Analysis

To validate current engineering tools and to improve the compressor station design process, the test data was benchmarked to SwRI's 1-D time domain transient pipe flow analysis code (TAPS). This code and its capabilities to predict the impact of pulsations on surge was previously described in detail by Brun et al. [2].

From a compressor station design perspective, the critical question is whether the 1-D transient piping codes that are currently utilized by the industry can adequately predict the suction/discharge pulsations on a centrifugal compressor such that the induced pressure and flow fluctuation accurately represent the characteristic ellipse around its steady-state operating point on the compressor's performance map. Thus, a number of numerical studies were performed using the model of the complex piping system of the previously described test facility and compressor operating characteristics for comparison with test data. FIGURE 20 shows the system models as analyzed by the TAPS software. The model is a geometrically accurate 1-D representation of the test facility and includes performance maps for the compressors, heat exchangers, scrubbers, and valves as installed in the loop. TAPS provides a pulsation analysis by solving the 1-D time-domain

Navier-Stokes equations of fluid flow through complex piping networks for highly transient flows and is commonly utilized to analyze piping acoustic problems in compressor stations. This code has been widely validated for piping pulsation analysis [4].

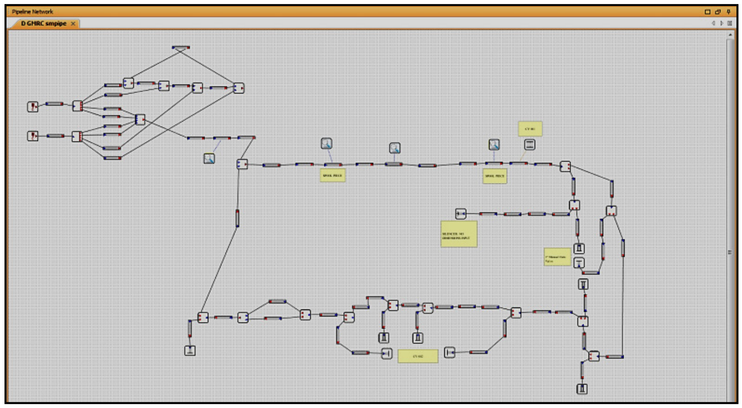

FIGURE 20. TAPS MODEL OF TEST LOOP FOR TRANSIENT PULSATION ANALYSIS

FIGURE 21 shows a frequency domain comparison between test data for the 480 rpm speed reciprocating compressor case versus pulsation amplitude and frequency predictions from TAPS. The predictions are seen to match test data to within 6%. This demonstrates that 1-D transient pulsation analysis, when properly applying model and boundary conditions, can accurately predict suction/discharge pulsation into a centrifugal compressor and thus define the operating map ellipse of the compressor. The operating map ellipse can then be utilized to determine the surge margin differential of the centrifugal compressor and its potential for operating in an area of instability. Having the capability to numerically analyze pulsating flow in the complex piping and equipment network of a mixed compressor station is imperative for design work and safe operation.

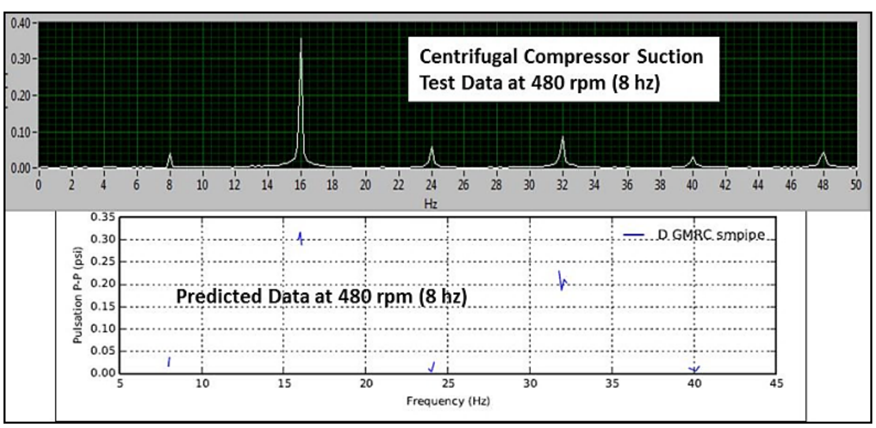

FIGURE 21. COMPARISON OF CENTRIFUGAL COMPRESSOR INLET PULSATION FROM TAPS PREDICTIONS VERSUS TEST DATA

Design Recommendations

To avoid centrifugal compressor surge or operating range reduction due to pulsations from a reciprocating compressor in a mixed compressor station, it is worthwhile to establish some basic engineering guidelines for station design. The most conservative design rule is simply to require that the centrifugal compressor's operating map ellipse does not cross the surge line for all operating conditions of the compressor station. However, this rule may be excessively conservative and could result in a significantly reduced operating range of the centrifugal compressor. Specifically, for high pulsation amplitudes and low piping impedance cases, an operating map ellipse with a large radius may force the compressor manufacturer to set the recycle valve control line well beyond the industry typical standard of 10%.

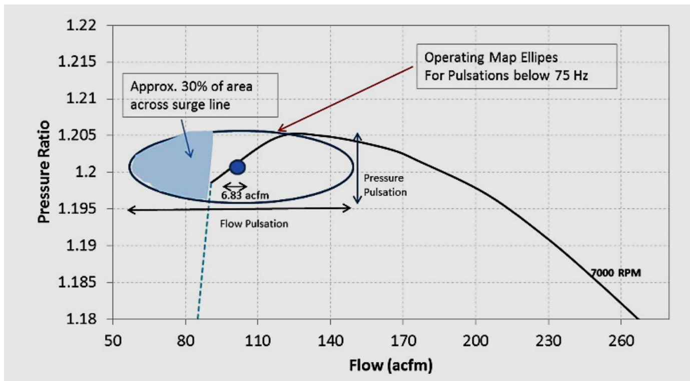

FIGURE 22. OPERATING MAP ELLIPSE WITH 30% OF THE ELLIPSE'S AREA LEFT OF THE SURGE LINE

A more pragmatic approach is to set a threshold interference level between the operating map ellipse and the surge that results in measurable surge. From the results of the above testing, it was found that surge was consistently identified when approximately 30% of the area of the operating map ellipse had crossed the surge lines for all suction/discharge pulsation frequency orders under 90 Hz. This basic design guideline is graphically illustrated in FIGURE 22 and FIGURE 23. FIGURE 22 shows the operating map with 30% of the ellipse's area crossing surge. FIGURE 23 is a plot of operating point surge margin versus time. While FIGURE 22 better conveys the design guideline concept, FIGURE 23 is more useful for analyzing the intersection area and thus, as a practical engineering tool aid.

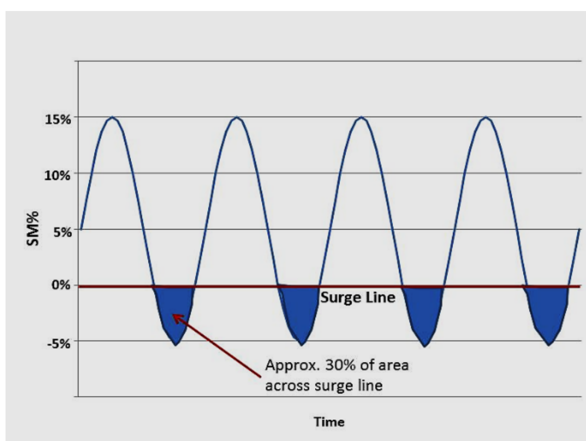

FIGURE 23. OPERATING POINT SURGE MARGIN VERSUS TIME SHOWING 30% OF AREA CROSSING SURGE LINE

It is important to emphasize that for the limited testing performed, it was found that when the operating map ellipse for pulsation orders below 90 Hz and the surge line overlap area was less than

30%, surge was not measurable and assumed to not have occurred. The reason for this consistent finding requires further investigation, but it is likely a result of the higher order frequencies not having sufficient transient time and amplitude across the surge line to initiate the onset of a surge cycle. Specifically, the wavelengths of higher order pulsations are likely to be too short to cause flow-reversal throughout the internal gas passage of a centrifugal compressor. For example, the half-wavelength of 100 Hz pulsations is less than 1.6 m (5 ft), which is shorter than the internal through-flow passage length of most industrial centrifugal compressors. Intuitively, one would not expect a surge cycle to be initiated unless the induced pulse half-wavelength can alter the pressure and flow of the full length of a compressor's meridional internal flow path. Thus, the above design guidance could be expanded beyond the proposed 90 Hz frequency limit and include all orders whose half-wavelength is greater than the centrifugal compressor's internal flow-path. However, to determine whether this assumption can be generalized to all operating conditions of a mixed compressor station or is simply limited to the test conditions and compressor geometry tested, requires further testing and analysis.

FINDINGS

External pulsations applied to the suction or discharge flange of a centrifugal compressor can reduce its surge margin significantly, going so far as to move the compressor into surge conditions. In mixed pipeline compressor stations where centrifugal compressors operate in series or parallel with reciprocating compressors, this is of special concern for operational range and safety reasons and must be considered during the design process.

CDR theory, first developed in 1983, describes how a centrifugal compressor and its associated piping system interact with pulsations from external sources such as a reciprocating compressor. Unfortunately, CDR provides only limited usefulness as a quantitative analysis tool, primarily due to the lack of validated numerical prediction tools and benchmark test data for comparison. The lack of experimental data for the validation of engineering analysis tools posed a fundamental problem.

To further the development of this theory into a useful analysis tool, testing of reciprocating and centrifugal compressor mixed operation was performed in an air loop at the SwRI compressor laboratory. The specific goal was to quantify the impact of periodic pressure and flow pulsation originating from a reciprocating compressor on the surge margin and performance of a centrifugal compressor in a series arrangement and to utilize this data to validate predictions from CDR and numerical approaches. For this testing, a 50 hp single-stage double-acting reciprocating compressor provides inlet pulsations into a two-stage 700 hp centrifugal compressor operating inside a semi-open recycle loop and utilizing atmospheric air as the process gas. Although the testing was performed for a series arrangement, the results are equally applicable to mixed station parallel arrangements were performed over a range of pulsation excitation amplitudes, frequencies, and pipe geometry variations to determine the impact of piping impedance and resonance response. This paper described the test facility, the test procedure, reports the reduced test results, and discusses the comparison to predictions. Basic design rules for pulsation-induced surge avoidance are also included.

From the test and analysis results, the following conclusions can be made as summarized below:

- Surge was consistently identified when approximately 30% of the area of the operating map ellipse had crossed the surge lines for all suction/discharge pulsation frequencies orders under 90 Hz.

- Test result and trends were consistent with predictions from CDR [6] and numerical predictions [2] for pulsation amplification and attenuation across a centrifugal compressor.
- Utilizing the transient operating map ellipse of the centrifugal compressor to identify whether induced pulsations can result in the operating point temporarily crossing the surge line is a useful tool to identify the potential onset of surge. From the operating map ellipse surge margin differential can be calculated for various orders of pulsations.
- If the upstream piping system impedance curve is flat, pressure pulses are converted to high volume flow pulses which increase the centrifugal compressor pulsation-induced surge margin differential. On the other hand, steep piping impedance curves of the downstream piping reduce the surge margin differentials.
- The geometry of the piping system immediately upstream and downstream of a centrifugal compressor can have significant impact on the surge margin reduction (surge margin differential).
- The reduction of surge margin due to external pulsations is a function of the pulsation's amplitudes and frequencies at the compressor suction and discharge flange. High suction flange amplitudes at low frequencies significantly increase the risk of surge. Surge margin reductions (differentials) over 40% were observed during testing.
- A transient time domain 1-D Navier-Stokes pipe network analysis model was able to accurately predict suction/discharge pulsations into a centrifugal compressor and thus, its operating map ellipse. Using the above described basic design rule (30% of the operating map area across the surge line for all pulsations below 90 Hz), these pressure/flow pulsation amplitude predictions can be related to surge margin differential.

A critically important step in designing a compressor station is to evaluate 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.

Implication to field installations

The impact of external pulsations on surge margins in field compressor stations is challenging to identify and characterize since (i) in most installations a reduced surge margin is not apparently obvious because of the initiation of recycle by the surge control system once the surge control line is crossed and (ii) few compressor stations have dynamic instrumentation at the proper compressor suction/discharge locations and of sufficient fidelity to capture pulsations entering the compressor. Thus it is quite possible that a centrifugal compressor can operate for extended periods in a pulsation induced surge without the operator being aware of it until damage to the machine's close-clearance components has occurred. The only effective method to identity this potential problem in mixed compressor installations is to either perform a field study with appropriate dynamic pressure/flow transducers, to measure pulsation amplitudes into the centrifugal compressor, or to perform a pulsation analysis of the entire compressor station including both reciprocating and centrifugal compressors.

DEFINITION OF SYMBOLS

Symbols and Units:

c	=	local speed of sound
p	=	is the sound pressure
u	=	molecular particle bulk velocity
z	=	specific impedance
A	=	pipe cross section area
D	=	impeller tip diameter
H	=	head
N	=	impeller speed
Q	=	volume flow
U	=	acoustic volume flow
SM	=	surge margin
Z	=	acoustic impedance
Δ	=	quantity difference
ρ	=	fluid density
φ	=	flow coefficient
ψ	=	head coefficient

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REFERENCES

[1] Brun, K. and Kurz, R., 2010, “Analysis of the Effects of Pulsations on the Operational Stability of Centrifugal Compressors in Mixed Reciprocating and Centrifugal Compressor Stations”, Journal of Engineering for Gas Turbines and Power, vol. 132, n 7.

[2] Brun, K., Nored, M., and Kurz, R., 2014, “Impact of the Piping Impedance and Acoustic Characteristics on Centrifugal Compressor Surge and Operating Range,” ASME Paper No. 2014-25054.

[3] Brun, K. and Nored, M., 2008, “Application Guidelines for Surge Control System,” Gas Machinery Research Council Publication.

[4] Brun, K., Deffenbaugh, D. M., and Bowles, E. B., Jr., 2007, “Development of a Transient Fluid Dynamics Solver for Compression System Pulsation Analysis,” Gas Machinery Conference, Dallas, Texas.

[5] Kurz, R., McKee, R., and Brun, K., 2006, “Pulsations in Centrifugal Compressor Installations,” ASME Paper No. GT2006-90700.

[6] Sparks, C. R., 1983, “On the Transient Interaction of Centrifugal Compressors and Their Piping Systems,” ASME 83-GT-236.

[7] Abdel-Hamid, A. N., 1985, “Dynamic Response of a Centrifugal Blower to Periodic Flow Fluctuations,” ASME 85-GT-195.

[8] Aust, N., 1988, “Ein Verfahren zur digitalen Simulation instationaerer Vorgaenge in Verdichteranlagen,” Diss U Bw, Hamburg.

[9] Baldwin, R. M., Simmons, H. R., 1986, “Flow-Induced Vibration in Safety Relief Valves: Design and Troubleshooting Methods,” The American Society of Mechanical Engineers, 84-PVP-8.

[10] Bar, L. C., 1979, “The Unsteady Response of an Axial Flow Turbomachinery Rotor to Inlet Flow Distortions,” M.S. Thesis, Department of Aerospace Engineering, The Pennsylvania State University.

[11] Blodgett, L. E., 1992, “Theoretical and Practical Design of Pulsation Damping Systems,” Flow Measurement and Instrumentation, Vol. 3, No. 3.

[12] Durke, R. G., McKee, R. J., 1986, “Identification of Pulsation Induced Orifice Metering Errors Including Gage Line Shift,” The American Society of Mechanical Engineers.

[13] Fletcher, C. A. J., 1991, “Computational Techniques for Fluid Dynamics, Volume I,” Springer-Verlag, Germany.

[14] Henderson, R. E., 1972, “The Unsteady Attenuation of an Axial Flow Turbomachine to an Upstream Disturbance,” Ph.D. Thesis, Engineering Department, University of Cambridge, England.

[15] Ingard, U., Singhla, V. K., 1974, “Sound in Turbulent Pipe Flow,” J. Acoust. Soc. Am., Vol. 55, No. 3, pp. 535-538.

[16] Iwasaki, M., Ikeya, N., Marutani, Y., and Kitazawa, T., 1994, “Comparison of Turbocharger Performance between Steady Flow and Pulsating Flow on Engines,” Proc. SAE International Congress and Exposition, Detroit, Michigan.

[17] Kinsler, L. E., Frey, A. R., Coppens, A. B., and Sanders, J. V., 2000, “Fundamentals of Acoustics,” John Wiley & Sons, New York.

[18] Meyer, W., 1988, “Untersuchungen zum Einfluss von Einlaufdrallstoerungen auf das stationaere Betriebsverhalten von Turbostrahltriebwerken,” Diss U Bw, Muenchen.

[19] Morini, M., Pinelli, M., Venturini, M., 2006, “Development of a One-Dimensional Modular Dynamic Model for the Simulation of Surge in Compression Systems,” ASME Transactions, Journal of Turbomachinery, 129 (3), pp. 437-447

[20] Shapiro, L. 1996, “Performance Formulas for Centrifugal Compressors,” Solar Turbines Publication.

[21] Smalley, A. J., Jungbauer, D. E., and Harris, R. E., 1995, “Reciprocating Compressor Reliability Issues,” Proc. 4th Process Plant Reliability Conference, Houston, Texas.

[22] Szymko, S., Martinez-Botas, R. F., and Pullen, K. R., 2005, “Experimental Evaluation of Turbocharger Turbine Performance under Pulsating Flow Conditions,” ASME Paper GT2005-68878.

[23] Wachter, J., Loehle, M., 1985, "Identifikation des dynamischen Uebertragungsverhaltens eines dreistufigen Radialverdichters bei saug- und druckseitiger Durchsatzvariation," VDI Bericht 572.2, pp. 365-379.

[24] Yocum, A. M. and Henderson, R. E., 1980, "The Effects of Some Design Parameters of an Isolated Rotor on Inlet Flow Distortions," ASME Journal, Journal of Engineering for Power, Vol. 102, pp. 178-186.