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INTEGRATING COMPRESSOR PERFORMANCE WITH THE EFFECTS OF PRESSURE PULSATION ACROSS A UNIT'S ENTIRE OPERATING MAP

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EXECUTIVE SUMMARY

The effects of pressure pulsation attenuation devices on reciprocating compressors can dramatically affect unit performance, especially load, flow, pressure drops and safe operating map. An acoustical (pulsation) study is ordered after the unit has been sized and appropriate unloading hardware selected. Recommendations from the acoustical study typically include pulsation bottle designs, pipe layout designs, orifice plate sizes and locations, and in some cases, changes to unloading sequences. Many acoustical studies are based on detailed analysis of a select set of design operating points reviewed across the compressor's specified speed range. The reality of compressor operations is that a compressor will operate at many points that are quite different from the selected design points. What really happens at these other operating points?

Fortunately, today's computers can model thousands of compressor operating points, including complete digital pulsation analysis, within hours. Thus, it is now possible to investigate a complete range of expected operating points prior to the unit actually being started. A complete acoustic review can identify areas in the operating map where high pulsations may still exist, to what extent flow and load are affected by the attenuation system, and how the effects of pulsations affect the safety of each load step.

When end-users more fully understand how their compressor units will really perform, they are in a better position to optimize unit performance and increase station safety. Furthermore, if significant issues are identified early, then the packager, the pulsation analysis company, and the end-user can review alternative designs (hardware arrangements) or operating schemes.

This paper covers:

- Modeling an in-the-field, high-speed, 2-stage compressor with added-clearance and end-deactivation types of unloading,
- Comparison of performance calculations with and without the compressor package acoustical analysis results,
- Importance of performing full acoustical reviews across the entire operating map,
- Incorporation of final results into end-user performance prediction software (and into PLC control panels), and
- Optimization considerations for new and upgraded compressors.

INTEGRATING COMPRESSOR PERFORMANCE WITH THE EFFECTS OF PRESSURE PULSATION ACROSS A UNIT'S ENTIRE OPERATING MAP

Introduction

Calculating accurate reciprocating compressor performance is required not only to keep units running safe, but also to optimize their performance. In general, three key factors determine unit performance: unit hardware and geometry (driver, frame, cylinders, valves, piping, load/flow control devices, etc.), gas thermodynamics (pressures, temperatures, gas constituents, etc.), and the physics of system flow (actual pressure pulsations, effects from added attenuation devices, and pressure drops).

Thanks to an array of published research from industry pioneers such as William "Bill" Hartwick (Cooper Industries), well-defined compressor performance models exist, based on unit geometries and gas thermodynamic properties. Therefore the performance prediction process can be both straightforward and quite accurate. However, the compressor performance calculations involve assumptions such as the system pressure drop and constant line pressure. These assumptions are made since there is usually not enough information early in the design stage. These assumptions can have a significant effect on the compressor performance. The compressor required power, flow, rod load magnitudes and reversals are all affected by these assumptions.

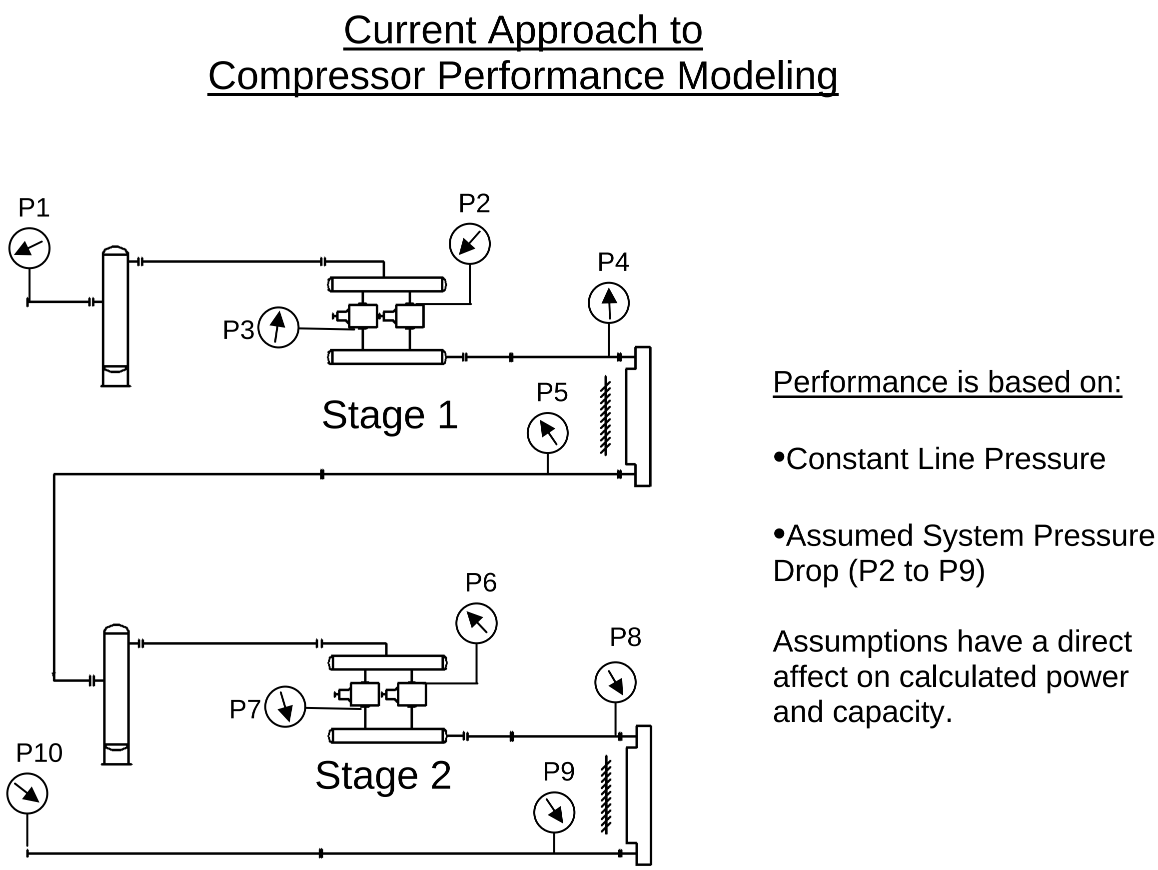


Figure 1

Compressor performance modeling is typically done for only a limited number of load steps, pressure ranges and temperature ranges. The intent of this approach is to capture the extremes of the compressor performance to ensure the equipment will meet the required design criteria. It has been shown in other work(1) that there is a benefit to calculating the compressor performance over the full operating map rather than just at points describing the operating envelop. There can be operating points within the performance map where operation would cause harm to the unit, such as insufficient rod load reversal. However, the limitation to calculating the full compressor performance map is that the performance calculations still include the pressure drop and constant line pressure assumptions in the model. Accurate calculation of the compressor performance requires that these parameters be included.

One consequence of operating any reciprocating compressor is the generation of acoustic waves from the reciprocating motion of the piston coupled with the opening and closing of the compressor cylinder valves. The energy from these acoustic waves (commonly called pressure pulsations) can result in high dynamic forces that can cause damage to the compressor, compressor package piping and vessels or even piping miles away from the station. The typical reciprocating compressor package design includes an acoustical simulation as per API 618 to minimize the effect of the pressure pulsations. Other results that are available from the acoustical simulation include the total system pressure drop as well as calculation of the effect of pressure pulsations on compressor performance. These are the key parameters required for more accurate calculation of the compressor performance map.

The objective of the work described in this paper was to determine a technique for improving the accuracy, efficiency and safety of the compressor operation as shown schematically in Figure 2. This is accomplished by improving the accuracy of the compressor performance predictions over the full operating map by including the effects calculated by the acoustical analysis. Data analysis and data reduction techniques were investigated to allow for integration of the acoustical results in the performance calculations as well as integration in the PLC/controller software.

Flowchart of Process for System Performance

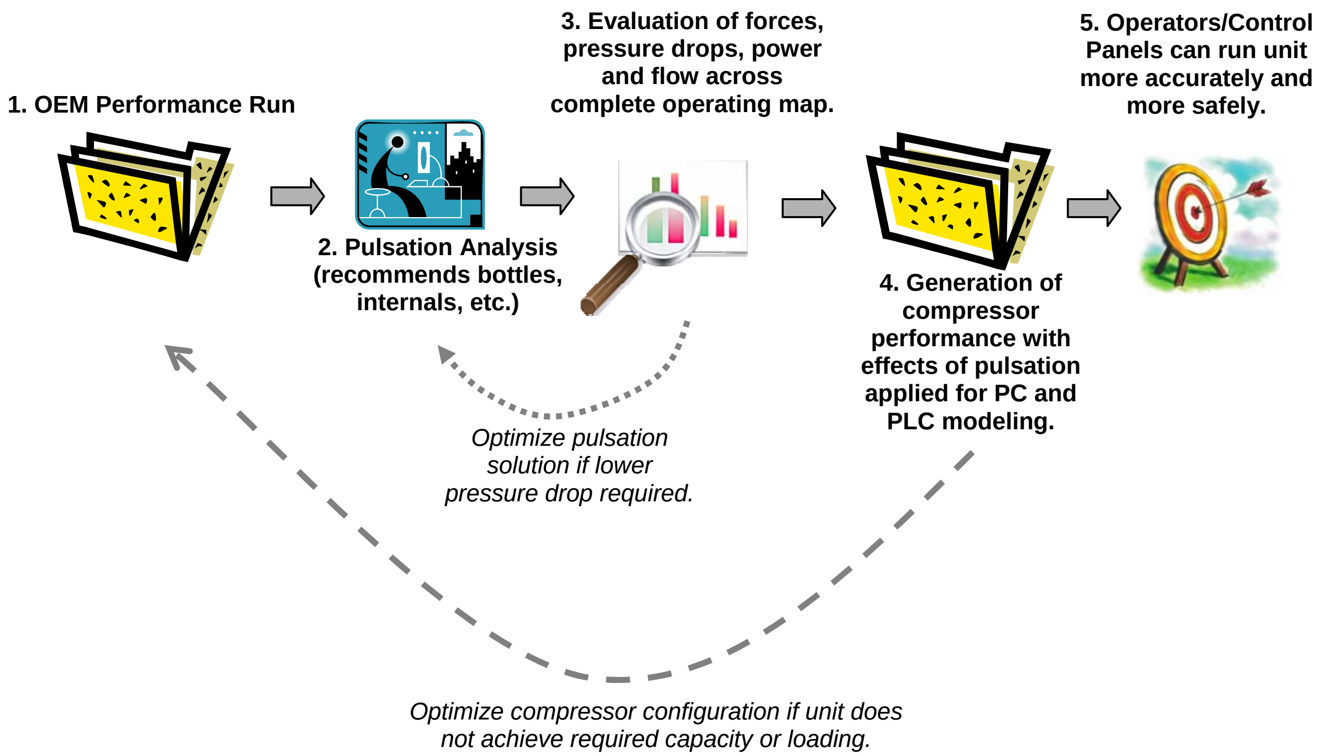


Figure 2

System Description

A 2-stage, 4-throw Ariel JGK compressor package was selected for this evaluation since this particular unit arrangement is quite popular (see Figure 3). The unit has manual variable volume pockets on all four (4) head ends, and also is set for additional unloading by removing the suction valves on one or both of the first stage head ends. This test unit provided a means for the simulation of various size volume pockets as well as double and single acting modes of operation.

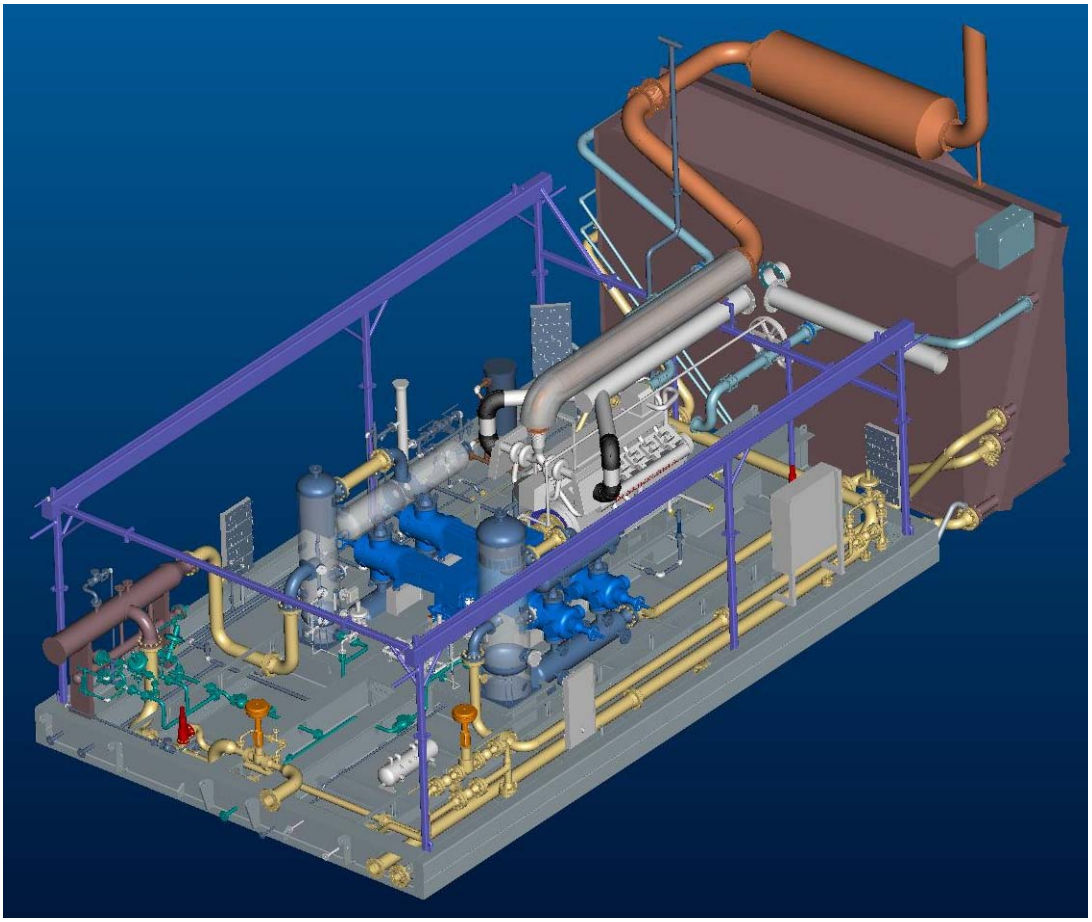


Figure 3: General Arrangement Drawing of the Compressor Package

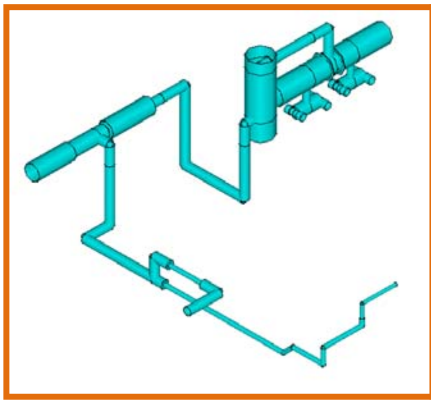
Typical Compressor Package Design Scenario

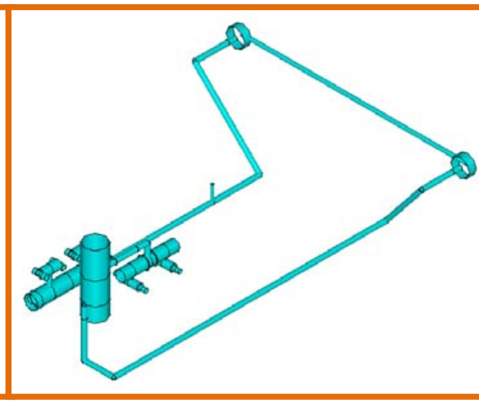
The compressor package was designed for an operating map with suction pressures ranging from 100 to 300 psig (689 to 2068 kPag), discharge pressures ranging from 1200 to 1400 psig (6895 to 9653 kPag), speed ranging from 900 to 1200 rpm, first stage suction temperatures ranging from 60 to 85 °F (15.6 to 29.4 °C).

The compressor performance was calculated by the compressor packager's application engineer for forty (40) design points. The design points included different combinations of pressure, temperature and loading that were intended to capture the maximum operating envelope.

Typically the acoustical analyst is supplied with a handful of operating points for the acoustical analysis. The standard approach is to review these points based on specific compressor performance parameters and the acoustical properties to determine a small set of design points. The detailed acoustical design is then done for this limited set of design points. The speed range is often divided into about fifty (50) speed increments to ensure that the peak acoustical response is calculated. The results from the acoustical analysis dictate the design for the pressure pulsation attenuation devices as well as possible piping and load step sequence changes. After the detail design is completed, the remaining design points are reviewed and the analysis of the acoustical simulations conducted.

For the test unit, forty (40) design points were supplied and fifty (50) speed increments were required to be run resulting in a total of 2000 operating points. The acoustical analysis is conducted using three separate models for a two-stage compressor: one model for the suction system, one model for the interstage and one model for the final discharge. Figure 4 through Figure 6 show plots of the acoustical models.





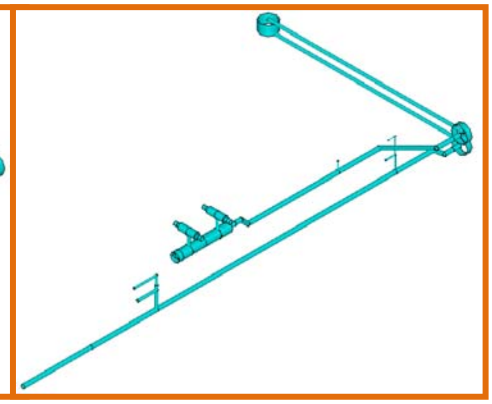


Figure 4: Suction Model **Figure 5: Interstage Model** **Figure 6: Final Discharge Model**

Finally, the unit being modeled included pulsation bottles and other attenuation devices installed, as initially determined by an acoustic review when the unit was ordered.

The results from the acoustical design study included the following pulsation attenuation devices:

- Orifice plates at the cylinder flanges and scrubber outlet flanges,
- Flow-thru baffles in the first and second stage suction bottles, and
- Choke tubes and solid baffle plates in the first and second stage discharge bottles.

Applying the Acoustical Analysis Results to the Full Compressor Performance Map

Integrating the acoustical analysis results for the complete compressor performance map required that additional acoustical simulations be run. The first step to running the acoustical simulations was generating a more detailed set of performance data for the test unit. For unloading purposes, the unit has nine (9) effective load steps:

1) All VVPs Closed
2) Open Stage-1 VVPs 25%
3) Open Stage-1 VVPs 50%
4) Open Stage-1 VVPs 75%
5) Open Stage-1 VVPs 100%
6) Open Stage-1 VVPs 100% & Stage-2 VVPs 50%
7) Open Stage-1 VVPs 100% & Stage-2 VVPs 100%
8) Pull Suction Valves on Head Ends #2/#4, & Close all VVPs
9) Pull Suction Valves on Head Ends #2/#4, & Open Stage-2 VVPs 100%

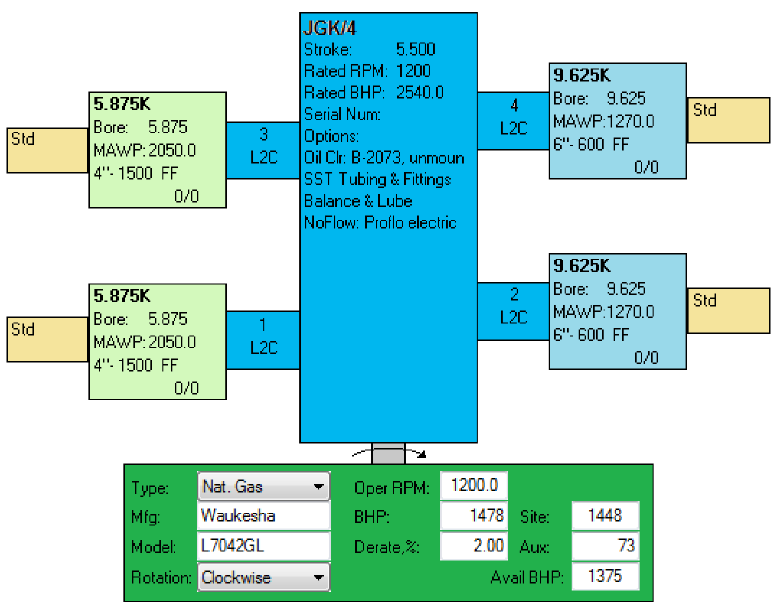


Figure 7

These load steps are graphically displayed in Figure 8. The horizontal blue line represents the maximum available power from the Waukesha engine of 1375 HP (1025 KW) at 1200 rpm.

A compressor performance map is typically generated using the full range of load steps as described above and also a full range of pressures, temperatures and speeds. For the purposes of this investigation, the discharge pressure was increased from 1200-1400 psig that was used in the package design to a range of 1000-1400 psig. The increments for the operating parameters are usually set to a small value to capture the compressor performance as accurately as possible. The computation time for compressor performance calculations are relatively low so running a large set of compressor performance simulations in a relatively short period of time is quite feasible. However, the computation power required for acoustical simulations is far greater than that required for performance calculations. Therefore, the number of design points used for an acoustical simulation must be carefully considered, and reduced if necessary.

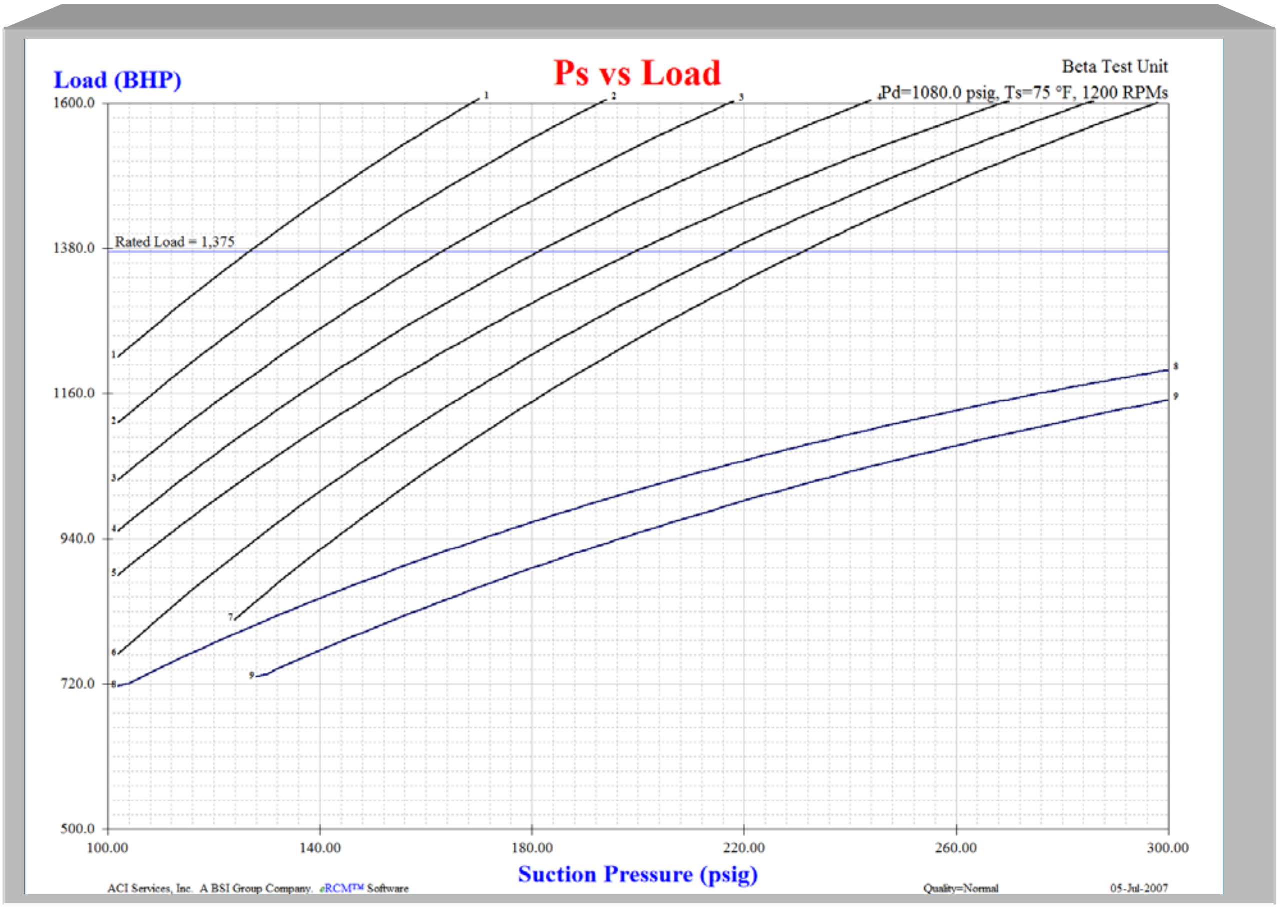


Figure 8

If pressures were checked every 2 psi, speed every 5 rpm, and temperatures every 5 degrees, then the acoustic review for the test unit would need to check:
= [(300-100)/2 + 1] * [(1400-1000)/2 + 1] * [(1200-900)/5 + 1] * [(85-60)/5 + 1] * [(120-90)/5 + 1] * 9 Load Steps
= 101 * 201 * 61 * 6 * 7 * 9 Load Steps
= 468,100,458 operating points – almost a half billion operating points.

If the acoustic software could review even just one (1) operating point per second, this would still take almost fifteen (15) years to complete. Thus, we need to create an effective compromise, one where the number of operating points reviewed is sufficient for adjusting performance models, and yet still be reasonable in terms of calculation efforts.

For the case of the test unit being modeled, increments of 20 psi were used for changes in suction pressure, increments of 40 psi were used for changes in discharge pressure, increments of 6.12 rpm were used for changes in speed, two different suction temperatures were selected for the first stage, and seven (7) of the nine (9) load steps were used. These criteria lead to a more realistic set of operating points to review:
= [(300-100) / 20 + 1] * [(1400-1000) / 40 + 1] * [(1200-900) / 6.12 + 1] * [2] * 6 Load Steps
= 11 * 11 * 50 * 2 * 6 Load Steps
= 72,600 operating points

Thus, the number of operating points to be reviewed is a modest 72,600 points. These particular increments were chosen for several reasons. Since the load step often has the most significant effect on the acoustical response, many loads steps were considered. The pressure pulsations generated from a double-acting compressor cylinder without extra head end clearance is dramatically different from the case where the compressor cylinder is single-acting. The effects on unit performance are typically most significant at the first and second order of compressor speed which are most dependent on load step. Finally, since the effect of added clearance per end can show significant differences at higher orders, different end clearances were considered.

The speed increment of 6.12 rpm was acceptable for the speed range of 900 to 1200 rpm. If the speed increment is too large, the peak response at the acoustical resonance could be missed. The acoustical analysis software used for these simulations also includes a routine to ensure the peak response is not missed. If the increments are too small, the extra effort reveals no real new information but consumes a lot of additional computations.

Gas temperature has a direct effect on the acoustic velocity. The acoustic velocity is also called the speed of sound or the speed at which the pressure waves travel in the gas. Therefore the temperature variation is an important parameter to consider in any acoustical analysis. The suction temperature is relatively constant for this application, as is true for many applications. Nevertheless, a small variation was considered. The discharge temperatures will have a much larger variation depending on the particular operating points reviewed.

The pressure increments were selected to provide a reasonable number of operating points and representative data for the compressor operating map. The pressures for this application will not have a significant effect on the acoustic velocity and acoustical properties.

Ariel generated the compressor performance for all 72,600 points with the results saved into an Excel spreadsheet. The performance calculations were completed in about one (1) day. The results were reviewed and almost 60% of the points had issues such as significantly overloaded the driver, exceeding rod loads, pin non-reversal issues, over cylinder MAWP, etc. While all 72,600 points were eventually acoustically analyzed, to minimize computational efforts, many of the points with safety-related issues could have been omitted from further analysis.

The performance data was sent to Beta for the acoustical analysis. The acoustical analysis first involved running additional performance calculations to be used as input to the acoustical simulations. The acoustical simulations were then run for all the specified operating points. Time domain acoustical simulations were completed over approximately six (6) days on typical PC workstations resulting in approximately 16 GB of data. It would be possible to reduce the acoustical simulation time to one to two days by optimizing the data management and running the simulations on more powerful workstations. The results from the acoustical simulations were reviewed and the following items extracted per operating point:

- system pressure drops,
- effect of pressure pulsations on compressor performance (load and flow), and
- pressure pulsations and resulting unbalanced forces for the complete performance map.

The extracted data from the acoustical simulations were then sent to ACI whose task was to determine the revised compressor performance maps. ACI then reviewed effective methods for incorporating the data in compressor performance software and in compressor control systems, such as PLC controllers. The integration of volumes of data into a reciprocating compressor model running on a typical PC does not present any real, insurmountable obstacles. However, implementing the same level of integration into the more restrictive and slower running PLC-based hardware can create considerable problems. Since a PLC controls the operations of the compressor at many stations, one of the main goals was to develop a method of integrating the results into PLC models.

Concepts and Results - How do results from the acoustical analysis affect the compressor operation?

Pressure Drop and its Effect on Performance

Performance programs typically include an assumption of the pressure drop from skid edge to the cylinder flange as well as pressure drop between stages on a multi-stage compressor. Usual assumptions for the pressure drops are 2% to 5% of line pressure. These pressure drops have a direct effect on the calculated performance. Therefore, accurately calculating pressure drops is key to accurately calculating the performance.

The pressure drop can be calculated from the acoustical simulation since all the flow and loss information is part of the acoustical model. The pressure drop in a reciprocating compressor system includes both a static component and a dynamic component. The static pressure drop is from the mean flow in the piping and is easily calculated by classical fluid mechanics techniques. Generally the static pressure drop in the system increases with the addition of pulsation attenuation devices (orifice plates, choke tubes). The dynamic pressure drop is the result of the dynamic flow in the system. Just as there are pressure pulsations in the piping the vessels (or dynamic pressure fluctuations), there are corresponding flow fluctuations in the system. These flow fluctuations act throughout the piping system and cause additional pressure drop, referred to as dynamic pressure drop. The dynamic pressure drop can be significant compared to the static pressure drop; in fact, the dynamic pressure drop can be many times greater that the static pressure drop. The only means of determining the dynamic pressure drop at the design stage is by acoustical simulation [2].

Figure 9 shows a plot of the pressure drop in the interstage system starting from the first stage discharge cylinder through to the second stage suction cylinder. The total pressure drop is calculated to be approximately 4.1 psi compared to approximately 2.7 psi for the static pressure drop. The dynamic pressure drop is approximately 50% of the static pressure drop, a significant difference in the calculated pressure drop. The dynamic pressure drop can be much higher than the static pressure drop for some compressor packages. Therefore it is important to calculate the dynamic pressure drop as well as the static pressure drop and include these effects in the compressor performance calculations.

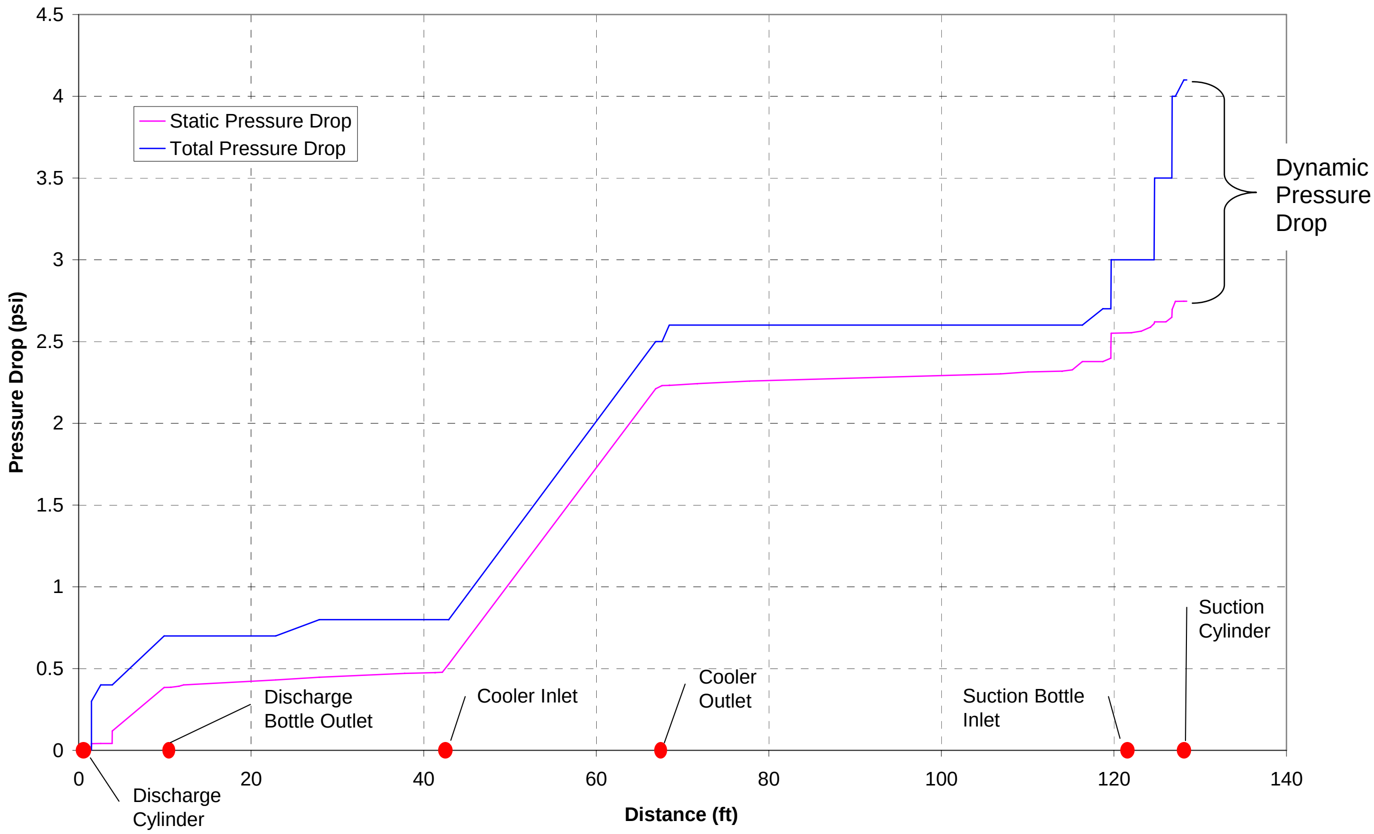


Figure 9: Calculated Pressure Drop for the Interstage System

As previously stated, the dynamic pressure drop is dependent on the acoustical properties of the system which vary with the operating condition and compressor speed. Therefore the acoustical analysis must be done for the full operating map to accurately determine the pressure drop over the full operating map. Figure 10 shows an example of the variation of the pressure drop versus speed. The static pressure drop changes with the square of the speed; however, the dynamic pressure drop does not have a fixed relationship with speed. In some cases the dynamic pressure drop can vary greatly with speed, particularly when pressure pulsations have not been adequately controlled by proper sizing of the pulsation dampeners.

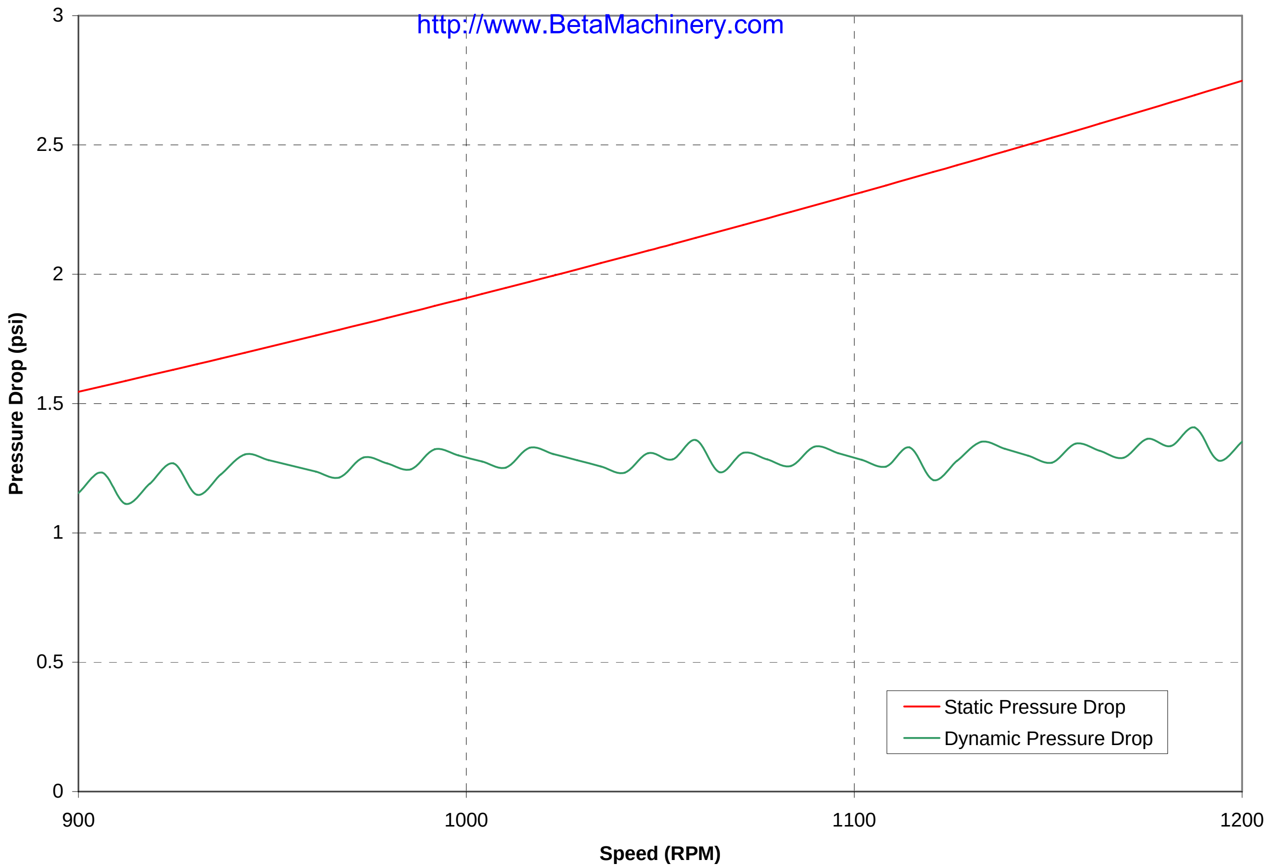


Figure 10: Static and Dynamic Pressure Drop versus Speed for One Operating Point

The compressor performance calculations are commonly done with an assumed pressure drop which is typically assumed to be constant for all conditions. The calculated pressure drop has a direct effect on the P-V and P-T cards, which means that the calculated performance parameters such as flow, power, rod load, etc. will be affected. For example, the calculated pressure drop effect changed the power requirement up to 5% for this test case as shown later in this paper.

Pressure Pulsation Effect on Compressor Performance

The pressure pulsations influence the pressure-volume (P-V) curve which is a direct indication of compressor performance. The API 618 overall pressure pulsation guideline is one criterion used in design studies to minimize the pressure pulsation effect on P-V curves. However pressure pulsations can still have a significant effect on the P-V curves even though the API 618 overall pressure pulsation guideline is met.

The ideal P-V curve for a double acting compressor cylinder is shown in Figure 11. The line pressure is assumed to be constant during the time when the compressor valve is open. The pressure gradually drops from the pressure at the valve opening event down to the valve closing event at dead center. The horsepower required to compress the gas is calculated as the area of the P-V curve. The flow from the cylinder can be calculated from the difference in the mass of gas trapped in the compressor cylinder clearance volume when the piston is at the suction valve closing and when it is at the discharge valve closing.

The primary effect of pressure pulsations on the P-V curve is changing the pressure during the time that the suction or discharge valves are open. The pressure pulsations cause the pressure inside the clearance volume and compressor cylinder gas passage to fluctuate resulting in a different pressure at the suction and discharge valve closings, which consequentially changes the compression ratio and therefore the capacity of the cylinder. This effect of pressure pulsations on the P-V curve is called the pulsation loading effect.

The changing pressure during the compressor valve opening distorts the P-V curve as stated above. The area of the P-V curve defines the power requirement so changing the shape of the P-V curve will also change the power requirements. Figure 12 overlays typical P-V curves with the distortion effects from pressure pulsations with the ideal P-V curves.

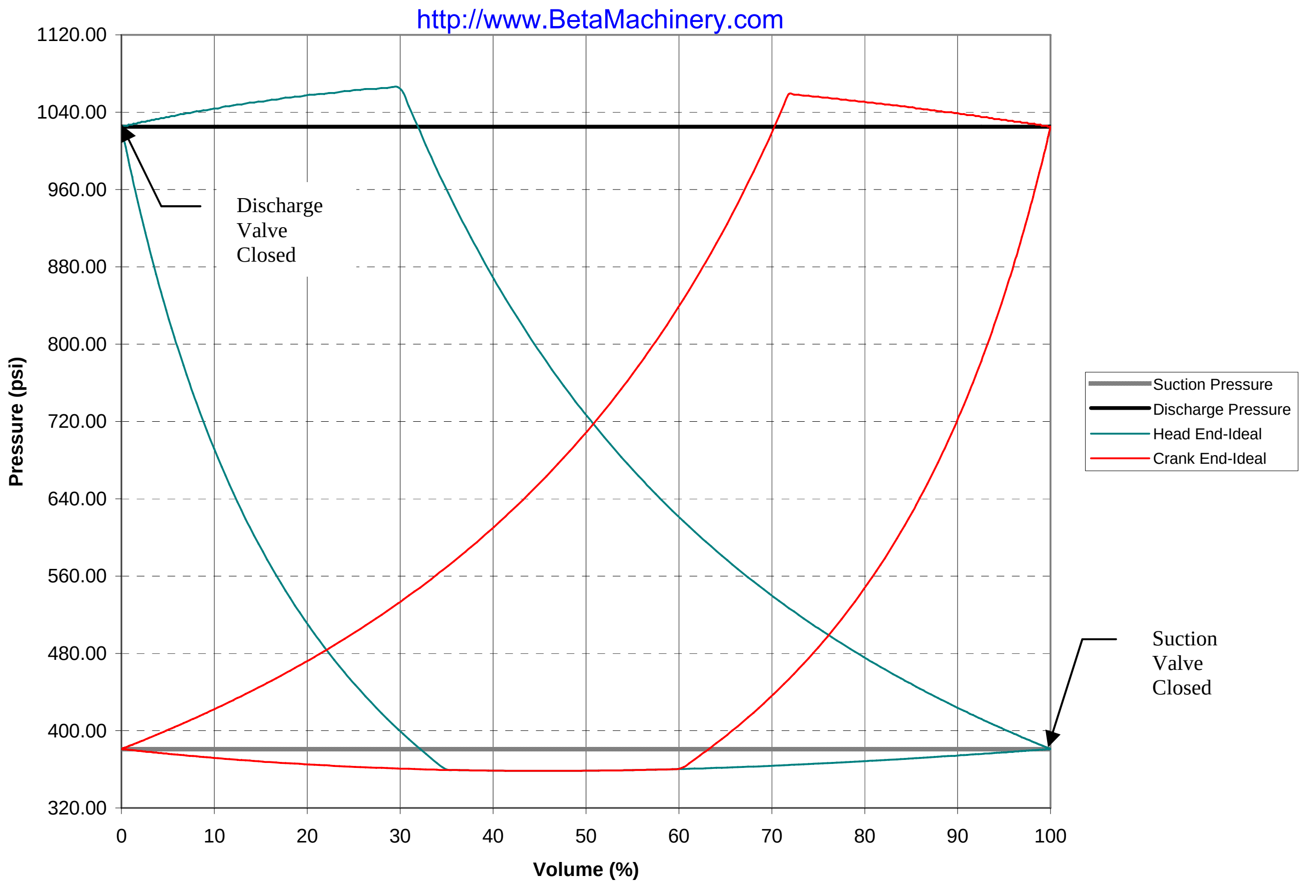


Figure 11: Ideal P-V Curve

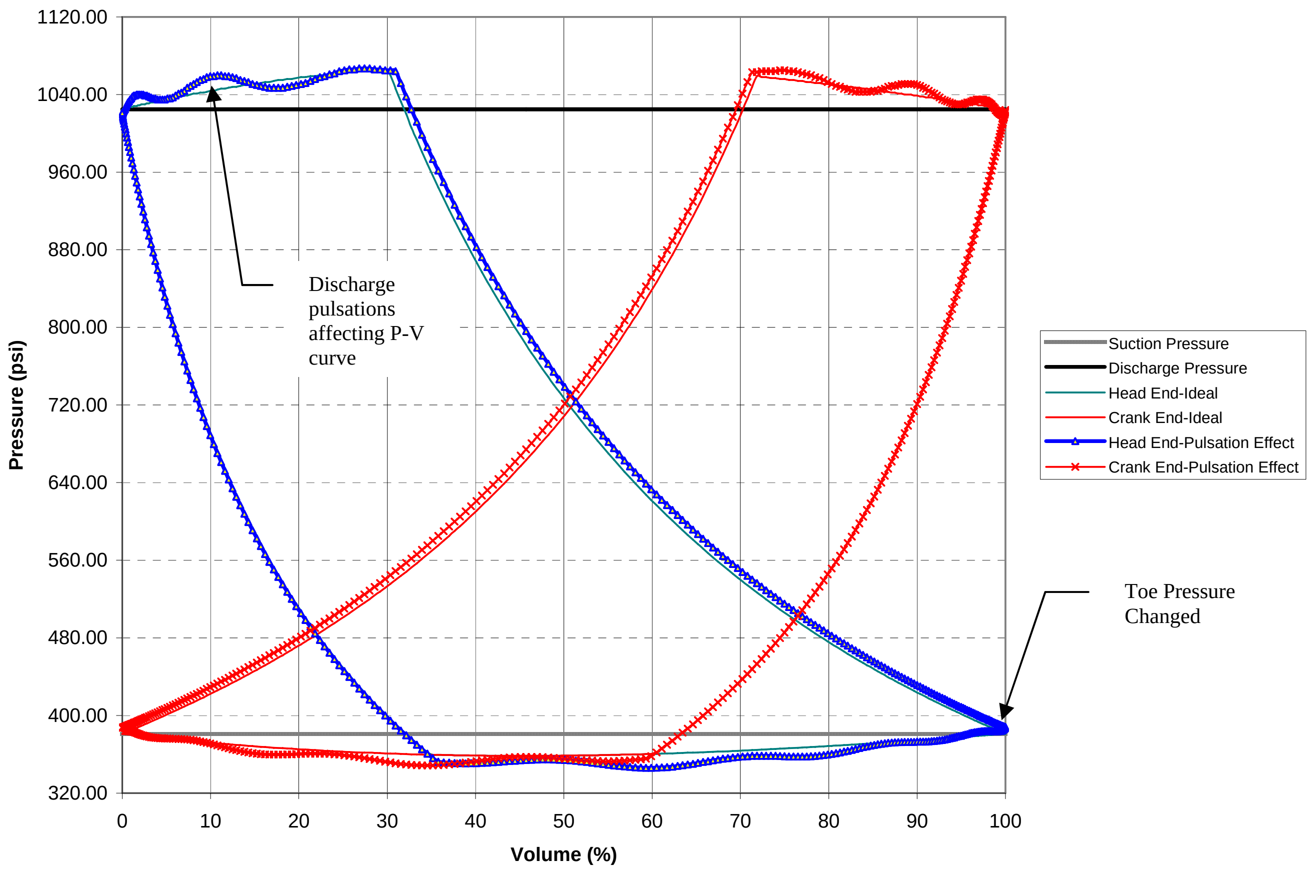


Figure 12: Ideal and Pulsation Affected P-V curves

The case shown in Figure 12 illustrates an increase in the flow of approximately 2.9%, and an increase in power of approximately 4.1% for a total change in HP/Q of 1.2% which is a typical number for a system with acceptable pressure pulsation control. The change in power and flow considering the full operating map for the test unit ranged from -8% to +9% as is shown later in this paper. The effect of pressure pulsations on the P-V curve can be higher for other units or for systems without proper control of pressure pulsations. Note that a negative change in HP/Q is in effect supercharging the compressor cylinder, that is, pressure pulsations are decreasing the HP/Q of the compressor.

Pressure pulsations can affect the compressor performance in many different ways. The method used in this analysis is a “first order” estimate of the pressure pulsation effects on compressor performance and has been shown to be accurate for engineering and operation purposes. Testing has been done to re-evaluate the acoustical response based on the P-V curve modified by pressure pulsations. The results showed very little change in the acoustical performance of the system. Field verification of the acoustical models have shown that the model agrees very well with measurements. Therefore the approach used is acceptable.

Acoustical Analysis

The typical API 618 M2 and M3 (acoustical) study includes analysis of one (1) to fifty (50) design points or operating conditions. An operating condition is defined as a specific set of operating parameters such as compression ratio, temperature, and cylinder loading. The design for this particular compressor package was done for forty (40) operating conditions. These conditions were selected by the compressor packager as being representative of the envelope of all the expected operating conditions for this particular application.

The purpose of this analysis is to evaluate the acoustical response over the complete operating map which can includes thousands of operating points, to determine if in fact analysis of the limited set of operating conditions is representative of the full performance map, or if there will be specific points within the map that exceed the acoustical design guidelines.

Note that the compressor discharge pressure range was expanded for this evaluation to include a discharge pressure range of 1000 to 1400 psig compared to the discharge pressure range of 1200 to 1400 psig that was considered in the original design study for this unit. The reason for increasing the discharge pressure range was to evaluate the compressor performance at off-design conditions as may be experienced in some applications.

Running acoustical simulations for the complete operating map generates thousands of operating points which results in gigabytes of report files. Analysis of this quantity of data is not possible using traditional methods. A new analysis tool developed by Beta called *Pulsation Analysis DataMiner™*, was developed to “mine” the results and evaluate the design.

Datamining of the acoustical results showed that in all cases the pressure pulsations and forces were within the design guidelines for the suction, interstage and final discharge systems. When the discharge pressure range was expanded to include the off-design conditions, some bottle forces were determined to be over the design guideline.

Samples of the typical results for the full operating map are shown in Figures 13a to 13d. These plots show the maximum pulsation and force in a particular part of the system for a particular order of run speed. Recall that fifty (50) speed increments are run for each order so the plots illustrate a considerable compression in the amount of data into a format that can be easily analyzed. Also, more than 220 nodes and sixty (60) forces were calculated in these models for twenty (20) orders of compressor speed with only a small sample of the results shown here.

These particular plots have the result amplitudes on the vertical-axis normalized with respect to the design guideline. Any value exceeding 1.0 indicates a value over the design guideline. The front-axis represents the operating condition numbers that were used for this analysis (726 operating conditions at fifty (50) speed increments). The results for some operating conditions are not shown as these points represent conditions where the compressor cannot operate over the complete speed range and therefore were removed from this particular analysis and presentation.

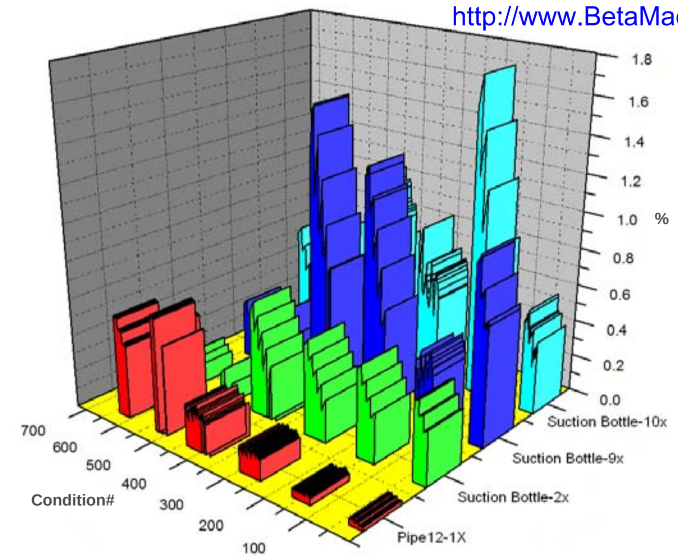


Figure 13a: Suction Force Summary

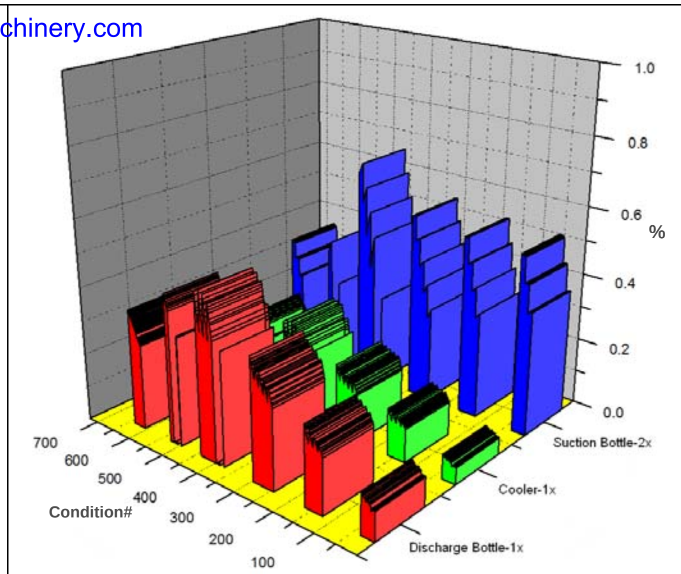


Figure 13b: Interstage Force Summary

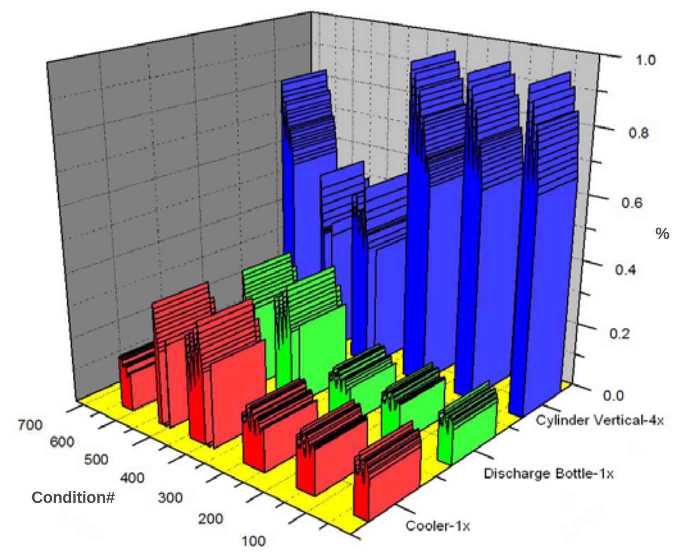


Figure 13c: Final Discharge Force Summary

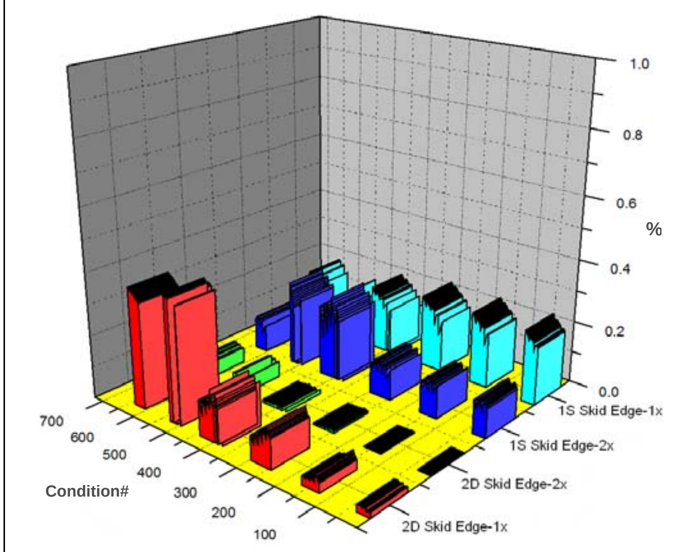


Figure 13d: Skid Edge Pulsation Summary

The conclusion from the acoustical analysis for the compressor performance map is that the acoustical design (pulsation bottles, choke tubes, orifice plates) were effective in meeting the design guidelines for the operating map defined by the original compressor package design requirements. If the compressor is to be operated at pressures outside the specified design points (e.g. lower discharge pressure range), changes to the acoustical design may be required.

Applying Results to PC-based and PLC-based Models

Operators and automation panels need to be able to predict reasonably well the expected power requirements, and resulting flow rates, of all units compressing gas. Furthermore, compressor monitoring services, pipeline flow models and in-house engineering software packages can all possibly benefit from having access to more accurate performance models.

An initial goal for performing a full acoustic review was that of effectively implementing into PC-based and PLC-based models the effects that pressure pulsations can have on load requirements and on flow rates for a reciprocating compressor. However, an acoustic analysis across a unit's entire operating map generates volumes of data. Thus, if this data could be efficiently reduced while maintaining relevancy, then OEM models, third-party software models, and PLC-based programs may be able to successfully utilize the data. Theoretical performance predictions could thus be augmented so as to include the effects of pulsations, and hence give better estimates for load requirements, flow rates generated, and provide another basis for optimizing compressor performance. In some cases, better estimates of pressures drops (both static and dynamic) can lead to safer unit operations as both rod loads and pin non-reversals are dependent upon internal cylinder pressure predictions which are themselves affected by inlet and outlet pressure drops.

Performance-related Goals of Acoustic Review of Entire Operating Map

- Find areas, if any, where pulsation effects on load exceed a certain percent (e.g. over 10%).
- Find areas, if any, where dynamic pressure drops significantly affect flange pressures, and/or load corrections.
- Implement acoustic results in OEM and third-party compressor performance software.
- Implement acoustic results in PLC/controllers.

Identifying Areas of Interest/Concern

After scanning all data for **Load Step #1** of the test unit, the maximum change in load is shown in the adjacent plot. First stage suction pressures are plotted against their corresponding first stage discharge pressures. Here, changes in prediction of the first stage's power requirements vary from 1% to almost 5%. Nonetheless, the unit's total power adjustment is based on individual adjustments from both stages.

By scanning all data, areas of high deviation from steady-state theoretical models can be determined and reviewed if necessary. This provides an opportunity to reconfigure the attenuation system and/or unloading sequence if operations in these areas are to be more effective.

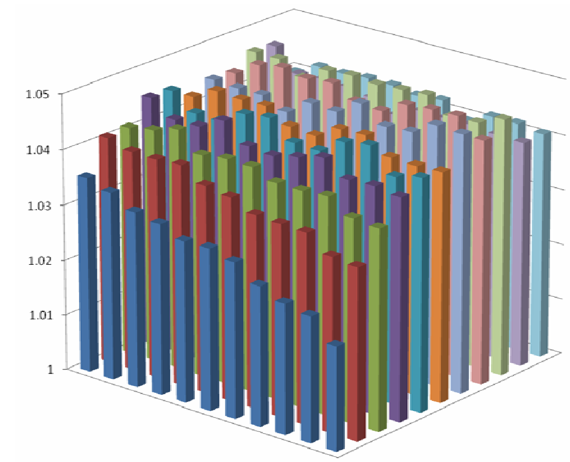


Figure 14

A review of the data from the full map acoustic analysis of the test unit found the following items of interest:

- Adjustments to load varied from -9% to 9% (i.e. final load is from 91% to 109% of initially predicted load),
- Average load correction was about 2% (i.e. on average, pulsations raised required loading by 2%), and
- Pressure drops (static plus dynamic) varied from -5% to +2% for suction pressures, and -1% to +3% for discharge pressures.

Figure 15a identifies potential areas (red areas) where higher levels (greater than 4%) of pulsation effect loading occur during the second stage of compression for Load Step #1. Figure 15b combines results from ***all load steps***, and thus shows the highest levels (red indicates areas that exceed 7.5%) of pulsation effects predicted during the second stage of compression, across the entire operating map – all pressures, all speeds, all load steps.

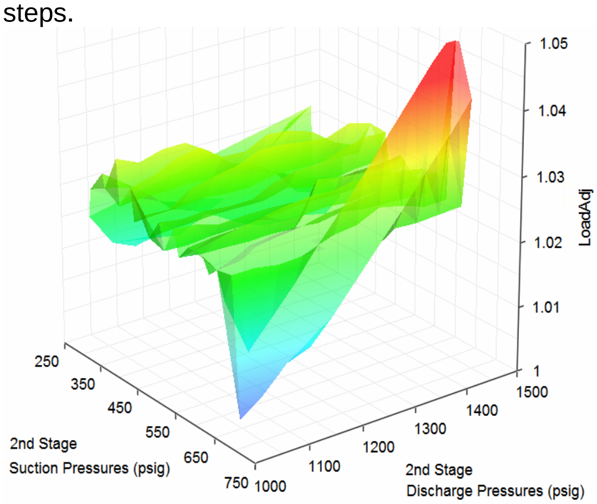


Figure 15a

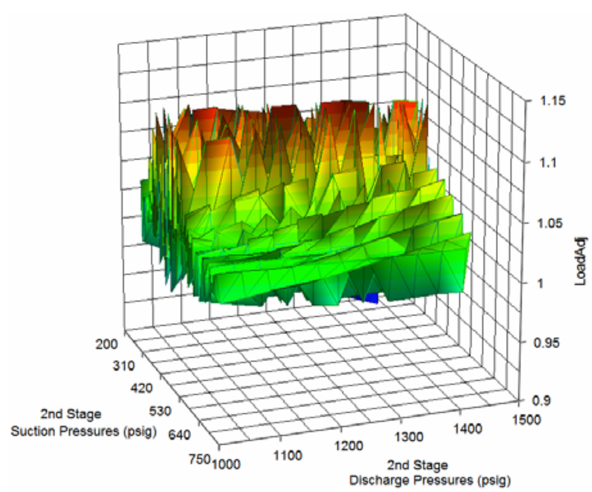


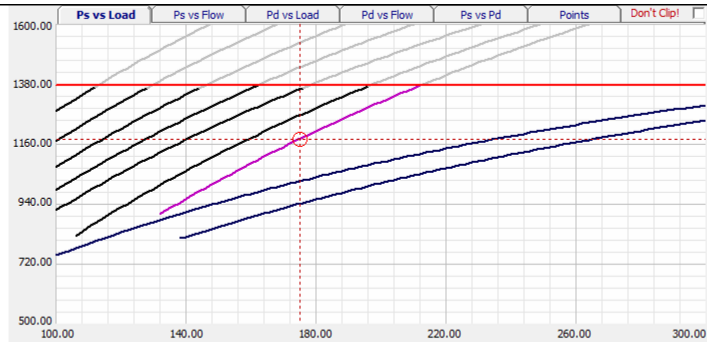
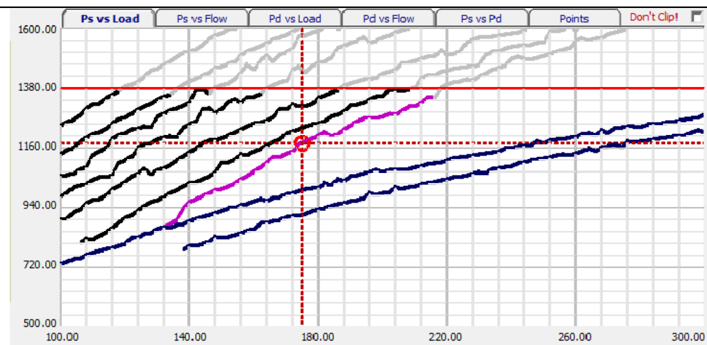
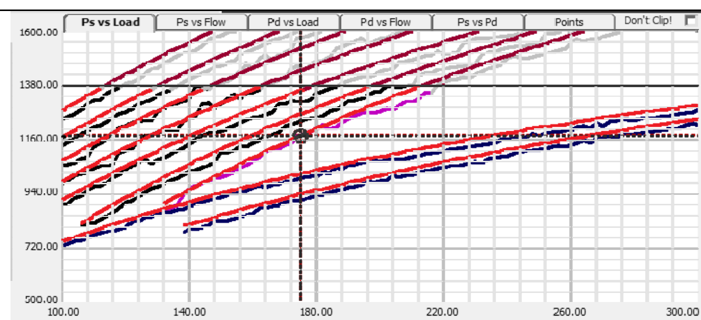
Figure 15b

Since the unit's effective load is the direct consequence of individual stage loading, proper combining of the stage adjustments is required. This combined effect may be a better indicator of need to change attenuation devices, or unloading sequence, than just the individual stage effects. For example, stage 1 may have a 10% increasing effect while stage 2 has a 7% decrease effect, yet the effective unit change may be something around a modest 2% change.

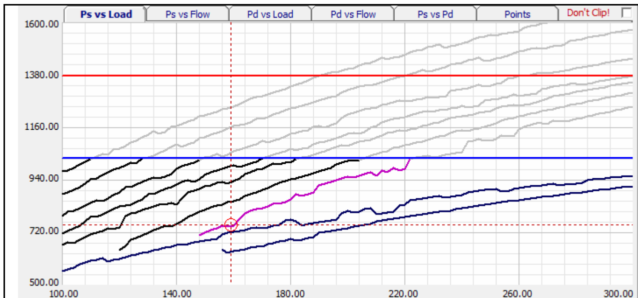
Performance Curves: With and Without the Effects from Pulsation Loading

Figure 16 shows a plot of the unit's performance curves based on normal steady-state flows. Figure 17 shows the same performance curves, but now with the benefit of adjustments for load based on the full acoustic study. Figure 18 showing the two curves overlaid provides a quick conclusion that the curves vary more than by just a few squiggles. In this operating case, the curves prior to the acoustic study are almost always over-predicting load.

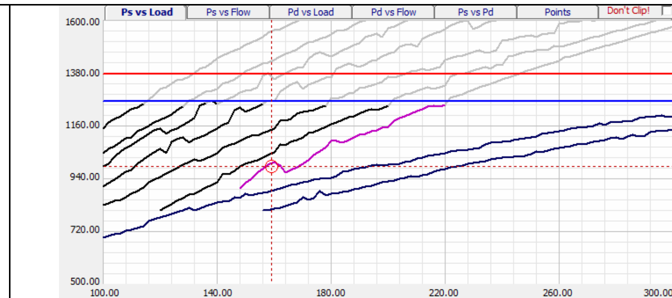
Ideally, most units would show only minor variations in load between loads predicted before and after the effects of pulsation loading are incorporated into the model (±6%). However, some units which may not have optimal design of their pulsation attenuation devices, such as small diameter orifice plates or choke tubes, may result in differences of 10% to 30% changes from classic steady-state predictions – typically the consequences of high dynamic pressure drops.

Figure 16: Original Compressor Performance	
Figure 17: Compressor Performance with dynamic effects included	
Figure 18: Original curves overlaid in red.	

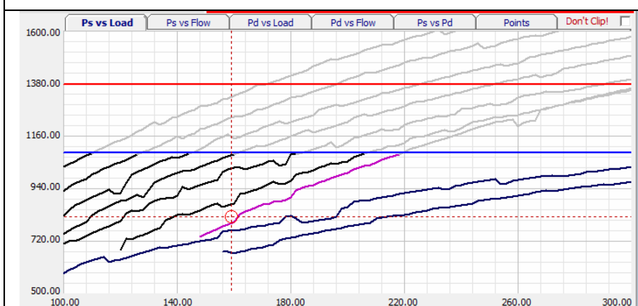
The following sets of images in Figure 19 detail one particular area of the operating map where predicted load noticeably changes as speed is changed, in this case from 900 rpm to 1200 rpm. The predict power at the crosshair, as speed is incremented, changes about 5%.



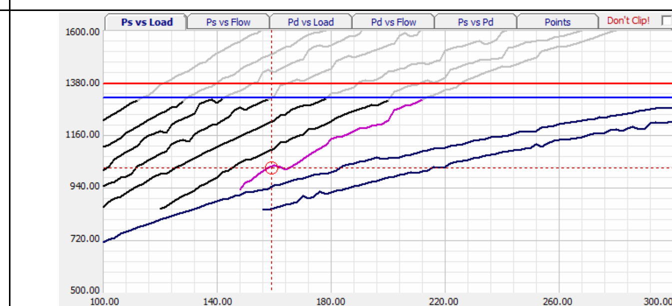
900 rpm



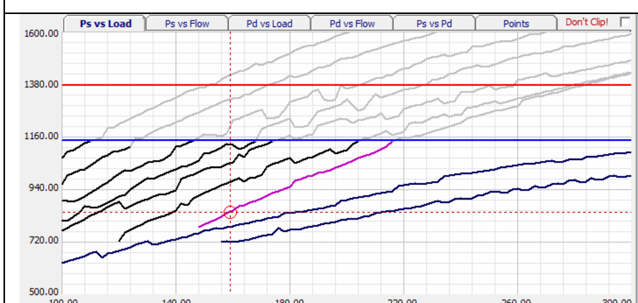
1100 rpm



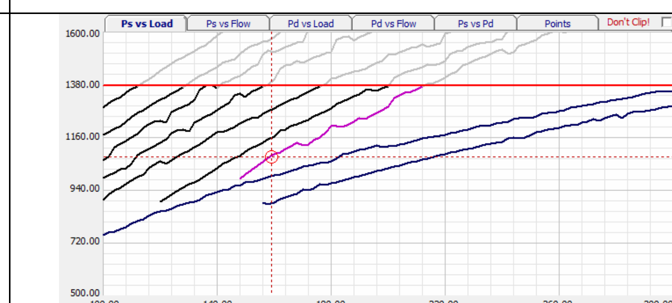
950 rpm



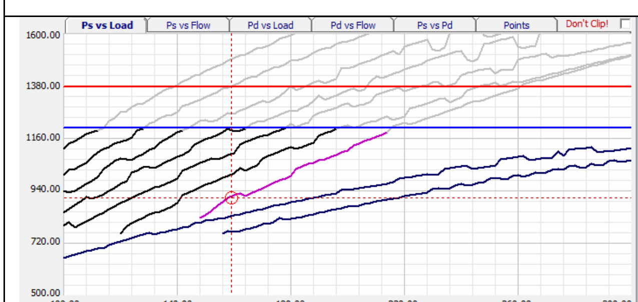
1150 rpm



1000 rpm



1200 rpm



1050 rpm

Figure 19: Predicted Compressor Performance Varying with Speed

Thus, just the act of changing speed can create a 5% change in theoretical performance predictions due to acoustic effects on power requirements for just one section of a load step curve. This helps to explain why in-the-field tuning of compressors can be such a tedious and often challenging task.

Primary Data Reduction Efforts

An important endeavor was that of reducing the volumes of generated data down to more concise and useable formats. The original data submitted from Ariel to Beta was over 25 Megabytes (Mb) of data for the 72,600 points – that's a spreadsheet with about 32,000 rows by 155 columns.

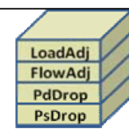
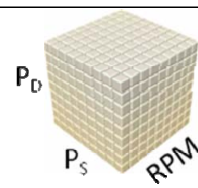
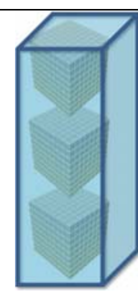
The acoustical simulations run by Beta generated more than 16 Gb of results. Key data was extracted and for the performance mapping, the files totaled about 25 Mb. Removing redundant information reduced these data files to about 7 Mb total. Finally, after merging the individual reports into a single database, the final database was just a bit over 624 Kb.

Thus, as a database of pressure drops and load and flow adjustments for allowing PC-based software to better predict flows, loads, and pressure drops (and hence better rod loads and pin non-reversals predictions), a 624 Kb file is not really that large. Since 50% of the data is for each stage, single stage databases would be in the 312 Kb range while 3-stage databases would be in the 936 Kb range. None of these file sizes represent problems for today's PCs. However, they present serious problems for incorporating this massive volume of data into older PLCs/controllers. *Today's newer PLCs often come standard with a couple Megabytes of RAM; therefore, end-users implementing these type of PLCs do not need to be concerned with data reduction issues.*

Secondary Data Reduction Efforts

One way to understand data requirements is by conceptualizing the basic information as a small record of data that contains the actual information of interest: both pressure drops and the acoustic effects on flow and load.

Table 1: Raw Performance Map Data

Data Record: Load Adjustment, Flow Adjustment, Suction Pressure Drop, Discharge Pressure Drop. Four (4) individual items.	 A single record of data.
Data Cube: For each combination of suction pressure (P_S), discharge pressure (P_D), and speed (RPM), store one Data Record. For 50 speeds, 12 suction pressures, and 12 discharge pressures, this becomes 7,200 records, or 28,800 items.	 Group of records.
Temperature Collection: For each suction gas temperature combination, store one Data Cube. If three different first stage suction temperatures and one second stage suction temperature were reviewed (three combinations), then the data requirement balloons to 21,600 records, or 86,400 items.	 Collection of groups of records.
Load Step Collection: For each load step, store one Temperature Collection. If twelve different load steps were reviewed, then the data requirement again balloons, this time to 259,600 records, or 1,036,800 items. *Of course, all of this is for a single stage. Thus, the data requirement is proportionately larger for multi-stage units. 2-stage models would require about 2.1 million items, and 3-stage would require about 3.1 million items.*	 Database of collections.

For an effective PLC implementation, the problem is that there are just too many dimensions. For each load step (1), you may have multiple datasets based on stage number (2) and various suction temperatures (3). These datasets are themselves based on various combinations of suction pressures (4), discharge pressures (5), and speeds (6). Thus, with six dimensions of freedom, the amount of data becomes unwieldy. When appropriate, these can be reduced to fewer dimensions:

- Performance curves are often created at an average suction temperature. Therefore, generation of results at average suction temperatures may be acceptable, or just averaging the data from various suction temperature datasets generated may be acceptable. This option can likely be applied to most models.

- While pressure pulsations vary with suction and discharge pressures, are they more related to compression ratios instead of individual pressures combinations? If so, the two dimensions (suction and discharge pressure) can be combined into one dimension (ratio).

- For fixed-speed motor driven units, the number of dimensions can be reduced as speed does not vary.

- Units with just limited changes in load step via added fixed clearance may also have similar acoustic consequences across all load steps or groups of load steps, and thus fewer datasets may be needed to create modeling that is useful for all load steps. This option can likely be applied to most models.

- Another reduction may be accomplished by doing the following: reiterating the acoustic study with the acoustic-study's determined pressure drops and thus, in essence, combining the change in load and flow due to pressure drops into the LoadAdj and FlowAdj data. When applicable, this cuts the database in half. Another cut in half can be accomplished by not storing the FlowAdj data. Why? In general, individual compressors are controlled based on load since very few units have their own flow meters. Thus, while better estimates of flow are nice, the station PLC still controls the individual unit PLCs based on throughput determined by the station meter. This option can likely be applied to most models.

- Finally, if the LoadAdj data can be fitted to a trend curve (curvefit), then fifty (50) items of LoadAdj may be reduced to just three or four coefficients of a polynomial.

Therefore, some potential reductions in information storage can lead to both less raw data and less complex data structures for implementations in older PLCs. Nevertheless, proper care needs to be taken to insure that these reductions still provide useful corrections to theoretical performance predictions. These issues are summarized in Table 2.

Table 2: Data Reduction Scenarios

Action	Reduction	Consequences?
Use just one average suction temperature and only store the LoadAdj items per rpm.	This would result in reducing the original database from 624 Kb to about 78 Kb.	May not be appropriate for systems that cover wide suction temperature ranges.
Use just one average suction temperature, combine pressures into ratios and only store the LoadAdj items per rpm.	This would result in reducing the original database from 624 Kb to about 19 Kb.	Same as above, plus: May underestimate or overestimate effects for certain pressure combinations.
Use just one average suction temperature, combine pressures into ratios, and curvefit the LoadAdj items.	This would result in reducing the original database from 624 Kb to about 6 Kb.	Same as above, plus: Curvefits are useful for nice trends, but often over-smooth and/or over compensate certain areas.

Conclusions

The worked conducted for this paper determined that:

- Conducting acoustical simulations over the complete operating map of a reciprocating compressor is possible and can be done in conjunction with an API 618 acoustical study,
- Extracting key results from the acoustical simulations and applying the data to the compressor performance map can be done efficiently,
- The results calculated by the acoustical analysis (pressure drop, effect of pressure pulsations on performance) have a significant effect on the compressor performance map, and
- Results from a full acoustic model can be effectively implemented into PC-based and PLC-based software.

Benefits of the enhanced performance modeling are:

- The end user can review alternate designs or operating schemes and identify the best possible solution for efficient and safe operation, and
- Ensuring that the unit will meet the intended load, flow and performance requirements.

Issues to Consider

The analysis techniques employed in this endeavor were based on test points determined via a fixed-increment approach. That is, each item, such as suction pressure, were incremented by a fixed amount for each new dataset. However, as is often the case for natural phenomena that involve acoustics and harmonics, a variable-increment approach may be a better choice. A fixed-increment approach can possibly miss certain harmonics, or it may tend to highlight higher peaks or lower valleys, and thus may skew results.

Since this project was very time consuming, only one unit was modeled. While important information was generated, care needs to be taken not to infer too much from one study based on one particular unit.

References

1. Hickman, Dwayne A., “Reciprocating Compressor Performance – Seeing The Big Picture”, Short Course, Gas Machinery Conference, 2006
2. Howes, Brian C. “The Cost of Dynamic Pressure Drop”, Seminar on Acoustic Vibrations and Pressure Pulsations in Rotating Machinery, Oakville, ON, Canada, March 4 & 5, 1993

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