ADVANCED RECIPROCATING COMPRESSION TECHNOLOGY (ARCT)

FINAL REPORT
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Reporting Period:
October 1, 2004 – September 30, 2005

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Prepared for

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December 2005

SOUTHWEST RESEARCH INSTITUTE®

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EXECUTIVE SUMMARY

The U.S. natural gas pipeline industry is facing the twin challenges of increased flexibility and capacity expansion. To meet these challenges, the industry requires improved choices in gas compression to address new construction and enhancement of the currently installed infrastructure. The current fleet of installed reciprocating compression is primarily slow-speed integral machines. Most new reciprocating compression is and will be large, high-speed separable units.

The major challenges with the fleet of slow-speed integral machines are: limited flexibility and a large range in performance. In an attempt to increase flexibility, many operators are choosing to single-act cylinders, which are causing reduced reliability and integrity. While the best performing units in the fleet exhibit thermal efficiencies between 90% and 92%, the low performers are running down to 50% with the mean at about 80%. The major cause for this large disparity is due to installation losses in the pulsation control system. In the better performers, the losses are about evenly split between installation losses and valve losses.

The major challenges for high-speed machines are: cylinder nozzle pulsations, mechanical vibrations due to cylinder stretch, short valve life, and low thermal performance. To shift nozzle pulsation to higher orders, nozzles are shortened, and to dampen the amplitudes, orifices are added. The shortened nozzles result in mechanical coupling with the cylinder, thereby, causing increased vibration due to the cylinder stretch mode. Valve life is even shorter than for slow speeds and can be on the order of a few months. The thermal efficiency is 10% to 15% lower than slow-speed equipment with the best performance in the 75% to 80% range.

The goal of this advanced reciprocating compression program is to develop the technology for both high speed and low speed compression that will expand unit flexibility, increase thermal efficiency, and increase reliability and integrity.

Retrofit technologies that address the challenges of slow-speed integral compression are: (1) optimum turndown using a combination of speed and clearance with single-acting operation as a last resort; (2) if single-acting is required, implement infinite length nozzles to address nozzle pulsation and tunable side branch absorbers for 1x lateral pulsations; and (3) advanced valves, either the semi-active plate valve or the passive rotary valve, to extend valve life to three years with half the pressure drop. This next generation of slow-speed compression should attain 95% efficiency, a three-year valve life, and expanded turndown.

New equipment technologies that address the challenges of large-horsepower, high-speed compression are: (1) optimum turndown with unit speed; (2) tapered nozzles to effectively reduce nozzle pulsation with half the pressure drop and minimization of mechanical cylinder stretch induced vibrations; (3) tunable side branch absorber or higher-order filter bottle to address lateral piping pulsations over the entire extended speed range with minimal pressure drop; and (4) semi-active plate valves or passive rotary valves to extend valve life with half the pressure drop. This next generation of large-horsepower, high-speed compression should attain 90% efficiency, a two-year valve life, 50% turndown, and less than 0.75 IPS vibration.

This program has generated proof-of-concept technologies with the potential to meet these ambitious goals. Full development of these identified technologies is underway. The GMRC has committed to pursue the most promising enabling technologies for their industry.

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1. INTRODUCTION

1.1 OBJECTIVES

The objective of the Advanced Reciprocating Compression Technology (ARCT) program is to create the next generation of reciprocating compressor technology to enhance the flexibility, efficiency, reliability, and integrity of pipeline operations. The suite of technologies developed by this program will not only provide pipeline operators with improved, affordable choices for new compression, but will also provide innovative solutions that can be retrofitted to existing machines to substantially improve current reciprocating compression.

The primary challenge is to provide added pipeline flexibility at reduced O&M cost. This challenge must be met without reducing equipment reliability or increasing capital cost. To address capital cost, the industry is moving to larger, higher-speed compression. Each machine is required to perform over a significantly larger operating range. The result is a strain on the synergistic technologies of capacity control, pulsation control, and compressor valves.

1.2 COMPRESSOR TECHNOLOGY STATUS ASSESSMENT

1.2.1 PIPELINE COMPRESSION EVOLUTION

Advances in compression technology helped the U.S. gas industry expand after World War II. The original first generation compression technology (Figure 1-1) consisted of many small slow-speed (180 RPM) compressors to move gas from producing regions to markets. To provide the necessary expansion, a developmental second generation of “larger, higher-speed” machines promised a significant reduction in installed cost. As industry installed the first of these machines, they experienced many reliability and operational problems. These problems involved flow pulsations and mechanical vibrations that resulted in piping failures. It was imperative for the industry to solve this vexing set of problems.

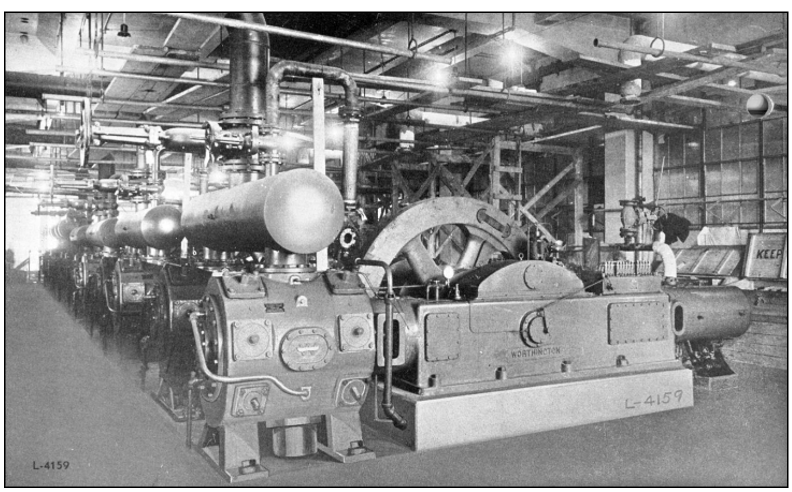

Figure 1-1. Early Slow-Speed Compressors Running at Less Than 180 RPM in the 500 to 750 HP Range Without Pulsation Control

To address this challenge, the pipeline industry formed what is now the Gas Machinery Research Council (GMRC), which contracted with Southwest Research Institute® (SwRI®). SwRI developed pulsation control systems that combined acoustic filters and dampers with effective mechanical restraints. The Analog Simulator, an ASME engineering landmark, was developed by SwRI to optimize the design of pulsation filter bottles and predict pulsation performance. SwRI has continuously operated the GMRC pulsation design service for the last 50 years, generating royalties that have funded GMRC research since 1955.

This second generation of compression technology (Figure 1-2) has now become known as "slow-speed integral" compression. A single crankshaft drives power and compression cylinders mounted in a single block. This equipment is nominally three times the horsepower running at twice the speed of the equipment it replaced. The slow-speed integral machines have been the U.S. pipeline industry's compression workhorse for the past 50 years. The current infrastructure of compressor stations consists of many individual machines, and the initial strategy of capacity control has been to activate a different number of machines. Prior GMRC research has documented a broad range in unit performance, as shown in Figure 1-3. With the increasing demand for pipeline flexibility, the industry is installing new compression and pushing the current fleet over an expanded operating envelope with a large turndown for each unit.

1.2.2 MODERN PIPELINE COMPRESSION

The promise of dramatic cost reductions has driven the industry towards even higher-speed, larger-horsepower reciprocating compression powered by efficient separate modern gas engines or large electric motors. Within the last few years, the first vintage of this new class of machines has been installed. This third generation of equipment (Figure 1-4) is four to five times the power of the prior generation and is now running at two to three times the speed. With this technology came new vibration and pulsation problems. The pipeline industry faces a technology transition similar to 50 years ago. As a few large machines replace many small machines, each machine must provide a wider capacity range and increased reliability. Varying speed more widely complicates pulsation control, and higher speeds have resulted in significant losses in compressor-efficiency, contributed in part by both pulsation control and conventional valve technology.

Figure 1-2. Slow-Speed Integral Compression Running at Less Than 300 RPM in the 1,500 to 2,500 HP Range With Pulsation Control Systems

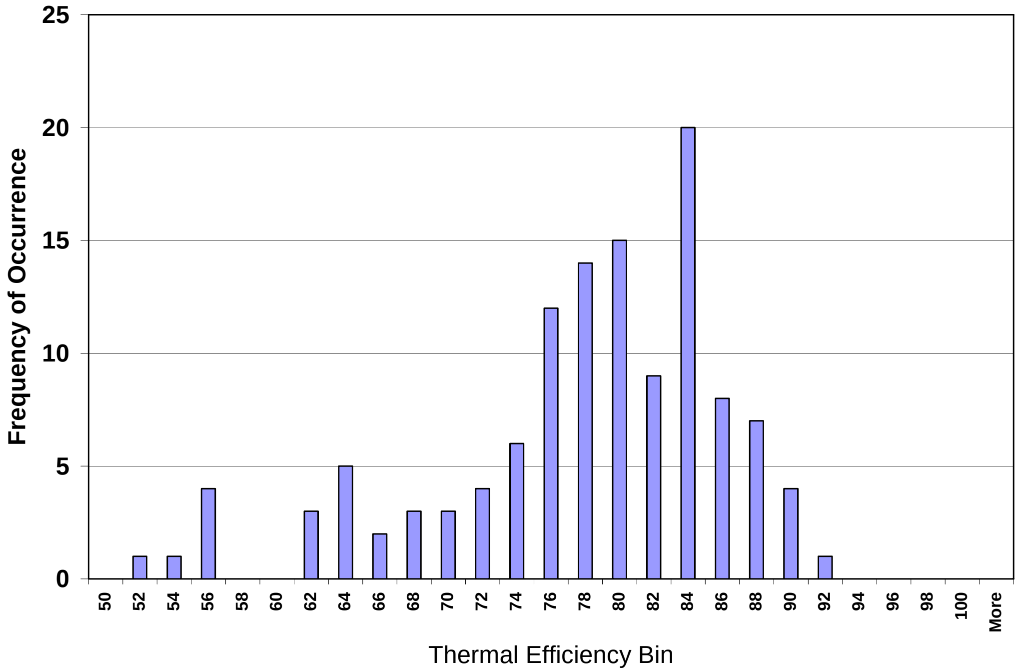

Figure 1-3. Compressor Thermal Efficiency Histogram Based on GMRC Survey

Figure 1-4. Modern Large High-Speed Separable Compressors Running in the 500 to 1,000 RPM Range in the 8,000 to 10,000 HP Range

The last generation of slow-speed integral machines is no longer commercially available because it was perceived as unaffordable. While affordable, the current generation of high-horsepower, high-speed compression requires advancements in technology to meet their full potential to address the pipeline industry's compression needs.

1.2.3 STATE-OF-THE-ART OF COMPRESSION TECHNOLOGY

1.2.3.1 Pipeline Operations

The U.S. natural gas transmission industry operates over 180,000 miles of interstate pipelines throughout key corridors of the country, with over 4,000 compressor stations at regular intervals (30 to 100 miles) to compensate for pressure drop and gas consumption. Within a pipeline, a central "gas control" facility monitors operation and establishes targets for flow or pressure, based on current contracts, or anticipated near-term needs. Much of the recent and anticipated growth is driven by the increased use of natural gas-fueled power plants and domestic consumption. Power plants are required to follow variable load profiles, thereby increasing pipeline flexibility demand. A recent INGAA Foundation study estimated over $60 billion in pipeline construction will be required over the next decade to meet the demand.

1.2.3.2 System Integration and Station Operation

At the individual compressor station level, flow flexibility requirements translate into an increasing need for automation (remote start-up, shutdowns), reliability, and broader capacity control. Short-term contracts, combined with large price swings, have led to less use of "line pack" to store gas in the pipeline. Volume flow requirements are up to meet the increased demand, while less "line pack" results in lower pressure ratio requirements. Pipelines earn revenue only by transporting "other people's" gas. Increased efficiency directly affects fuel consumption, operating cost, emissions, and capacity, but contractual arrangements do not always motivate the most efficient compression solution. As part of the pipeline system, a successful reciprocating compressor must manage these interacting factors: capacity control, pulsation control, and valves. System efficiency, smooth operation, and the resulting reliability are maximized at the system design point. Small departures from this design point have modest impacts on efficiency, smooth operations, and reliability. Major departures have significant adverse effects, but reciprocating compressors must increasingly operate over a wider range of conditions.

1.2.3.3 Pipeline Capacity Control

Capacity control is the method to vary the flow rate and engine load in response to end-user demand and pipeline required pressure ratio. Historically, pipelines installed many small compressors and adjusted flow rate by changing the number of machines activated. This capacity and load could be fine-tuned by speed or by a number of small adjustments (load steps) made in the cylinder clearance of a single unit. As compressors have grown, the burden for capacity control has shifted to the individual compressors. The following equation for volume flow rate (suction conditions) helps illustrate available options.

$$\textit{Flow} = \textit{Swept Area} \times \textit{Stroke} \times \textit{Speed} \times \textit{Volumetric Efficiency}$$
$$\textit{Volumetric Efficiency} = f\,(\textit{Pressure Ratio}, \%\ \textit{Clearance})$$

The common methods of changing flow rate are to change speed, change clearance, or deactivate a cylinder-end (hold the suction valve open). Another method is an "infinite-step" unloader, which delays suction valve closure to reduce volumetric efficiency. As Table 1-1 shows, each method has advantages and disadvantages.

Table 1-1. Capacity Control Methods

CAPACITY CONTROL METHOD	ADVANTAGES	DISADVANTAGES
RPM	Simple Control	Adverse Pulsation
Clearance Volume	Effective Control	Limited Range
Valve Unloader	Effective Control	Adverse Pulsation and Low Efficiency
Deactivate Cylinder	Effective Large Step	Adverse Pulsation and Low Efficiency

1.2.3.4 Pulsation Control

The critical development that allowed widespread use of slow-speed integral compressors was the pulsation control filter bottles and associated design tools. These bottles functioned as low-pass acoustic filters using a "volume-choke-volume" technique. Figure 1-5 shows idealized response of a low-pass filter bottle, superimposed on the pulsation spectrum for a 300-RPM double-acting compressor. For a fixed operating speed, the Helmholtz frequency and cut-off frequency of the filter are located between the fundamental operating speed and second order. The first pass band is located between the seventh and eighth order. This method is extremely effective for fixed speeds. However, even with small speed variations, the effectiveness of this method is complicated and certain speeds must be locked out to avoid response at the pass band frequency.

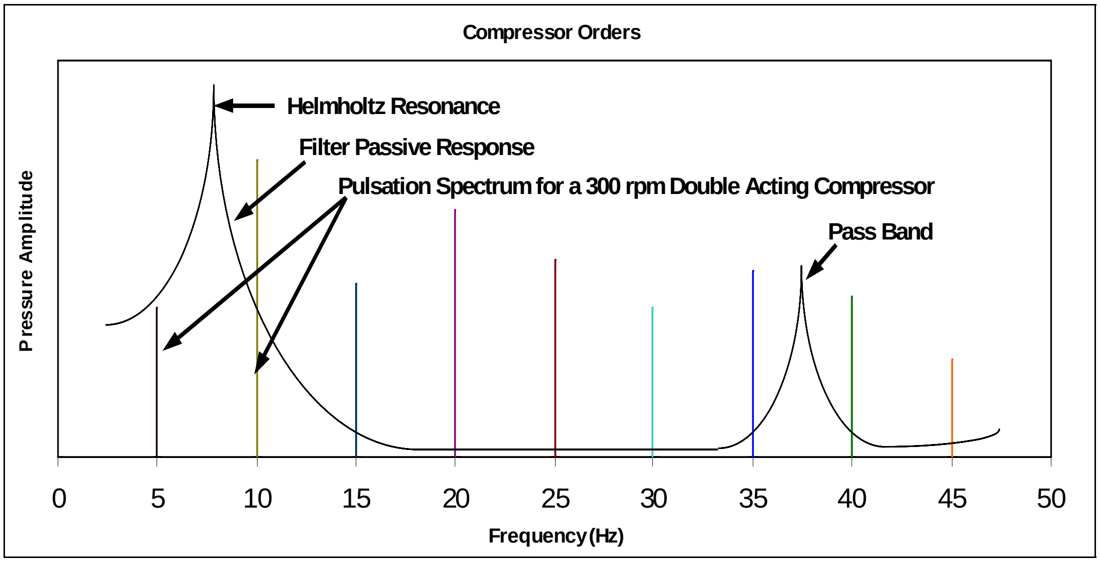

Figure 1-5. Filter Bottle Helmholtz Response

Acoustic filter bottles can be implemented as two-bottle or single-bottle designs that have advantages in terms of vibration control, but disadvantages in terms of space and weight. Nozzle orifices and side branch resonators are other pulsation control methods, and each method has its advantages and disadvantages as shown in Table 1-2.

Table 1-2. Pulsation Control Methods

PULSATION CONTROL METHOD	ADVANTAGES	DISADVANTAGES
Filter Bottle	Effective Pulsation Control at Fixed Speed	Cost, Size, and Weight
Orifices	Dampen Nozzle Pulsation	Pressure Drop
Side Branch Resonator	Fixed Frequency Control	Fixed Speed Only

1.2.3.5 Valves

Simple passive check valves have generally served pipeline slow-speed compression needs. The valve-sealing element opens when pressure overcomes the spring force. Ideally, the differential pressure due to flow creates a force, and the difference between pressure and spring forces accelerates the sealing element until it hits a guard, which limits travel ("lift"). In a simple model, sealing elements stay against the guard until the spring force exceeds the pressure force, then the spring closes the valve and holds it closed. Suction valves open when cylinder pressure drops below suction line pressure and closes near bottom-dead-center. Discharge valves open when cylinder pressure exceeds discharge line pressure and closes near top-dead-center (for head end cylinder). The sequence repeats in each cylinder, each revolution. Desirable attributes include perfect sealing when closed, rapid opening, sustained high flow area when open, rapid timely closing, tolerable impact with no bounce, tolerance to maximum temperature, and low-flow resistance of entire flow path. Proper parameter choices (material, mass, spring constant, spring preload, lift, flow area) maximize the chance of success. However, these simple, passive valves do not tolerate wide operating ranges well. Operational problems include "flutter," late/early closure, imperfect sealing, excessive impact force, and excessive compression temperature. Achieving long life with low losses while minimizing leak potential is very challenging. Valve manufacturers have made great advances (materials and configuration). Yet, design trade-offs are often mismanaged. The push for low-flow resistance leads to excessive impact forces, or the drive for long life leads to flow resistance so high the driver cannot provide needed capacity. The increased number of impacts and higher impact velocities with high-speed compression exacerbate the problems. In addition, large high-speed compression is required to operate over wider operating ranges, which results in more operation away from the design point. Valves are the single most significant maintenance item in pipeline compression. The advantages and disadvantages of each of the common passive valves are shown in Table 1-3.

Table 1-3. Compressor Valve Types

VALVE TYPE	ADVANTAGES	DISADVANTAGES
Plate	Low Cost/Simple	Relatively Short Life
Ring	Low Pressure Drop	Shortest Life
Poppet	Longer Life and Rugged	Highest Pressure Drop

1.2.3.6 Sensors and Automation

Slow-speed integral machines operate with minimal unit instrumentation. Pressure ratio and temperatures are measured at the station level, but unit flow rate is not typically measured.

Meter stations are located at pipeline custody transfer points, which in most cases are not collocated with the compressor station. Flow measured at the meter station is available to central gas control, but not compressor station operators. Current high-speed units have sophisticated engine instrumentation and controls but very little instrumentation on the compressor side. Load steps are programmed with various pocket and unloader settings to give prescribed capacity and load control without a closed-loop feedback control system. Remote operation from central gas control can start and stop units and can select speed and load steps through the unit control panel to provide the required capacity and loads. Since reciprocating compressors are positive displacement devices, prescribed settings give a reasonable approximation of the resulting capacity. However, sufficient margin is provided in recognition of this approximation.

1.3 APPROACH

The initial ARCT program was a five-year, three-phase program. The Phase I approach is to document the state-of-the-technology for each subsystem, develop new subsystem design concepts, provide proof-of-concept for each, and recommend the best technologies to be moved to the next phase of development. The program was divided to investigate five subsystems: pulsation control, capacity control, valves, sensors, and systems integration. The state-of-the-technology assessments of each subsystem were documented, and new design concepts were developed. These design concepts were qualitatively assessed, ranked, and analyzed. This resulted in a short list of practical ideas. Detailed engineering analyses were then completed to quantitatively rank each concept against the approved decision criteria. Proof-of-concept experiments were performed on breadboard hardware for the top concepts. Those top concepts that proved successful are recommended for the next phase of prototype development and roadmaps are provided to guide future development.

1.3.1 APPROACH TO IDEA CREATION

A series of brainstorming sessions were held for each of the major subsystems. The sessions were kicked off with a presentation of the state-of-the-technology reviews. These reviews were followed by idea creation sessions that resulted in a number of ideas to address the shortcomings of the current available technology. Each subsystem team then qualitatively assessed the ideas to create a short list of practical ideas. Detailed engineering analysis resulted in quantitative ranking against decision criteria approved by the industry advisors. A summary of the number of ideas and the resulting short lists are shown in Table 1-4.

Table 1-4. Summary of Brainstorming Ideas and Resulting Short List

SUBSYSTEM	NUMBER OF IDEAS	SHORT LIST OF IDEAS	TOP CONCEPT
Pulsation Control	44	10	5
Capacity Control	31	9	1
Capacity Measurement	13	8	2
Capacity – Variable Stroke	10	3	1
Valves	61	11	4
Sensors	10	10	3
Optimization	6	6	2
ARCT PROJECT	**175**	**57**	**18**

Each of the lists of ideas created, the method for reducing these ideas to a short list, a description of the concepts, the quantitative analysis, and the final ranking against the decision criteria are presented in a subsequent chapter of this report.

1.3.2 CONCEPT DEVELOPMENT APPROACH

The general decision criteria approved by the GMRC Project Supervisory Committee (PSC) for evaluating new technology concepts developed on ARCT are provided in Table 1-5. All criteria are in comparison to current technology available for sale to the pipeline industry. Each new technology concept is rated on a scale from 10 (excellent) to 1 (poor). For the first six criteria, the current state-of-the-art is considered as mid-scale. For the last four criteria, the current state-of-the-art is considered as maximum on the scale. Each subsystem is compared on a common set of criteria.

Table 1-5. General Concept Evaluation Criteria

CRITERIA	DEFINITION
Effectiveness	Effectiveness of the new technology concept at meeting the intended purpose.
Efficiency	Impact of the new technology concept on relative system efficiency.
Life/Reliability/Durability/Ruggedness	Life, reliability, durability and ruggedness of the new technology concept.
O&M Cost	Cost of operating and maintaining the new technology concept.
Installed Cost	Overall installed cost of the new technology concept after development.
Synergy	Impact on the balance of the compressor system of the new technology concept.
Technical Maturity	Level of technology maturity and associated schedule and cost to bring the new technology concept to market.
Retrofitable	Likelihood of the new technology concept being retrofitted to current compressor equipment.
Developmental Risk	Chance of successfully developing the new technology concept.
Economic Viability	Economic viability for vendor and owner that addresses critical market needs with high market acceptance.

Through a voting process, the project team developed weighting factors for the list of criteria. Each member assigned a number from 10 (most important) to 1 (least important) for each of the ten criteria. Each member had 55 votes to distribute between the ten criteria, but each criterion must receive a rating. The total score was normalized to give the criteria with the maximum score a value equal to 10.

The end result was the set of weight factors shown in Table 1-6. These weight factors were then used in design trade-off studies to rank each design concept against each ranking criteria. The result was decision tables presented in subsequent chapters of this report.

1.4 TECHNOLOGY DEVELOPMENT PROCESS

The process employed for assessing the state of a given technology concept is based on defined Technology Readiness Levels (TRL). A TRL is a systematic metric that supports an assessment of maturity of a particular technology. These metrics are used for the consistent comparison of maturity between different types of technology. The TRL scale ranges from 1 for basic principles observed through 9 for commercially available systems proven in a successful pipeline operation. The definition of each TRL is presented in Table 1-7. The ARCT technology development process is to advance the maturity of each subsystem technology to reduce the risk of system implementation for the next generation of natural gas compression. The ARCT program is divided into three phases. The Phase I program objective is to create new technology solutions and then advance these technologies to at least proof-of-concept (TRL 3). Subsequent phases would advance each successful technology through breadboards, prototypes, and eventually to proven commercial systems.

1.5 REPORT ORGANIZATION

The balance of the report is organized into chapters for each of the major subsystems. The chapter begins with a detailed assessment of the current state-of-technology. The results of the brainstorming sessions are then presented along with the qualitative analysis of the creative ideas to form a short list of design concepts. The design concepts are then described. Quantitative engineering analyses are presented, and the ranking against the above decision criteria. The top concept proof-of-concept experiments are then described. These chapters are followed by an evaluation of technologies and roadmaps for future development as required by the DOE guidelines.

Table 1-6. Weighting Factors

CRITERIA	OVERALL
Effectiveness	10
Efficiency	9.1
Life, Reliability	8.6
O&M Cost	7.7
Synergy	6.4
Retrofitable	6.4
Developmental Risk	6.4
Installed Cost	5.0
Technical Maturity	3.6
Economic Viability	3.6

Table 1-7. ARCT Development Process

TRL	DEFINITION
1	Basic principle observed and reported.
2	Technology concept and/or application formulated.
3	Analytical/experimental critical function proof-of-concept.
4	Component and/or breadboard validation in laboratory.
5	Breadboard validation in representative environment.
6	Prototype demonstration in representative environment.
7	System prototype demonstration in pipeline environment.
8	Commercial system completed and “qualified” in pipeline demo.
9	Commercial system “proven” in successful pipeline operation.

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2. PULSATION CONTROL RESULTS

2.1 PULSATION CONTROL TECHNOLOGY ASSESSMENT

2.1.1 Need for Pulsation Design

Reciprocating compressors inherently generate transient pulsating flows. Manifold system pulsation designs (filter bottles) are required to provide sufficient surge volume and acoustic filtering that will reduce the amplitudes of modulating flow and control the pulsation amplitudes throughout and exiting the manifold. An optimal design effectively isolates the piping from the compressor-generated pulsations and provides steady, relatively pulsation-free flow to the headers and adjacent units. Figure 2-1 illustrates the intricate interrelationship of the pulsation design (acoustic filter design) with the entire compressor design process. The pulsation design must take into account the effects it has on the compressor performance, mechanics, thermal stresses, and other issues incorporated into the design of the system. Shortening a segment of pipe to improve the pulsations can lead to increased coupling of mechanical vibrations to a less sturdy system of piping, and/or it could introduce thermal stress issues. An inadequate acoustic manifold design can significantly increase pulsations, degrading compressor performance. Cylinder stretch due to internal gas loads can result in the excitation of both skid mechanical natural frequencies (MNFs), and manifold MNFs, particularly with short, stiff nozzles required from acoustic design considerations.

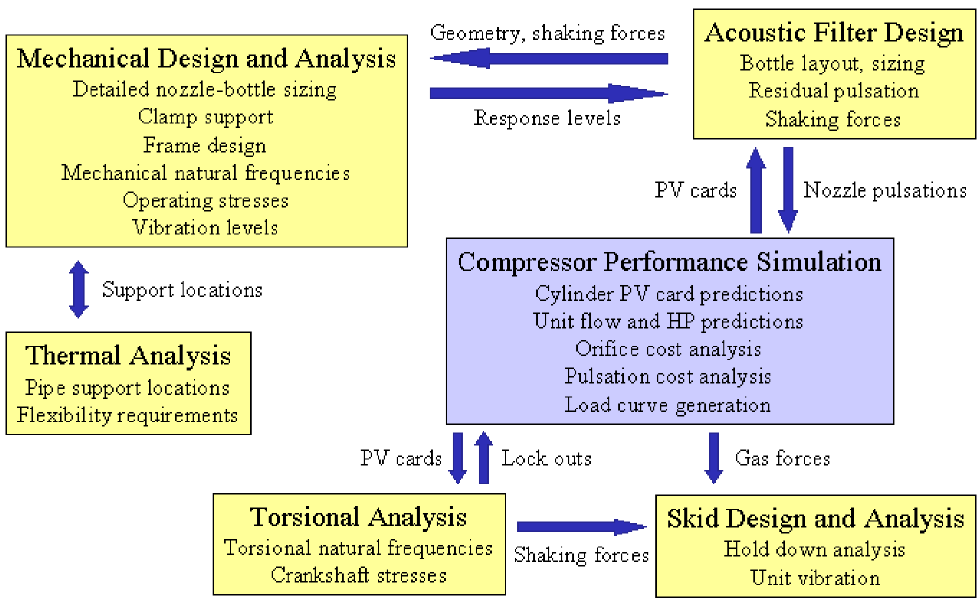

Figure 2-1. Overview of the Design Process for Reciprocating Compressors – Illustrating the Effects of Acoustic Filter Design on Many Other Parts of the Overall Compressor Design

The pulsations generated by reciprocating compressors can be particularly high if the manifold design is not done correctly, increasing downtime and maintenance costs. Poor manifold designs can lead to unnecessary performance loss and excessive vibration of adjacent units (reciprocating units and/or centrifugal units) and auxiliary equipment (including gas coolers and scrubbers).

A proper pulsation design reduces the system pulsations to acceptable levels without compromising compressor cylinder performance. Pulsation-altering devices, such as surge volumes, pressure drop elements, and acoustic filters, can be used to reduce the effects of inherent pulsations on the system. Surge volumes attenuate pulsations and smooth out the flow pulses introduced by the compressor. As shown in Figure 2-2, the pulsation filter is placed between the compressor and the station headers. A simple surge volume could be installed, such that the resulting flow to the headers would be steady. However, in a high flow environment, reducing pulsations to acceptable levels would require unrealistically large surge volumes. Acoustic filtering is the best design option, allowing the use of smaller surge volumes to achieve acceptable pulsation levels. Choke tubes play an important role in the design of an acoustic filter system; but because unsteady flow can result in excessive horsepower consumption through pressure drop elements, it is imperative that dynamic flow losses be properly accounted for in the design process. Orifice restrictions are often used to attenuate pulsations in the internal gas passage-nozzle region. Since this region is characterized by unsteady flow, it is also imperative that dynamic flow losses through nozzle orifices be accounted for properly and that the effects of these losses be reflected in a compressor performance summary.

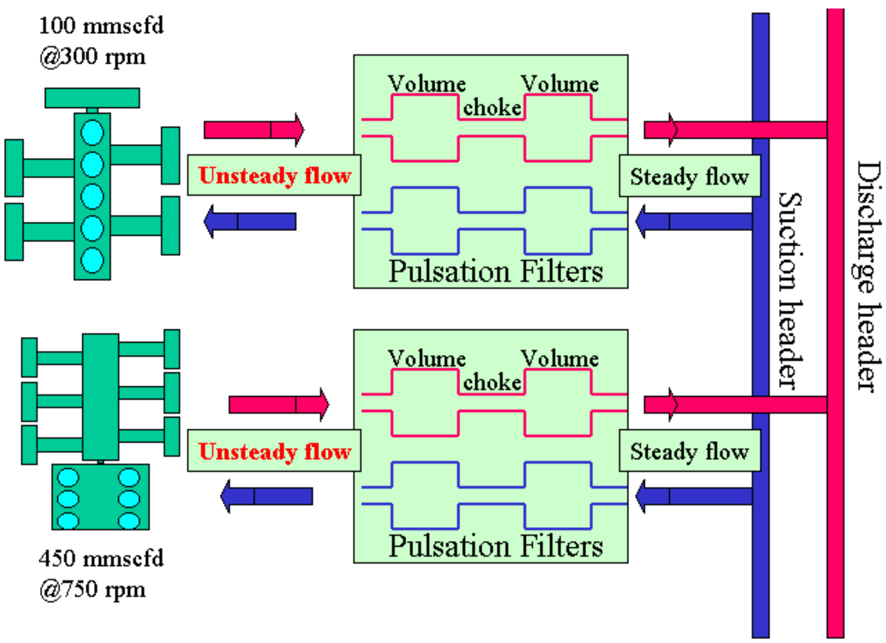

Figure 2-2. Schematic of Gas Flow in a Compressor Application Illustrating the Need for Pulsation Filters

Design of a pulsation filter typically begins with the placement of the Helmholtz response. As shown in Figure 2-3, a Helmholtz resonance is a sharp, high-amplitude response that is associated with the volume-choke-volume configuration of a pulsation filter. Placement of the Helmholtz frequency is based on the size of each volume element and the length and diameter of the choke tube. Preferred placement of a Helmholtz response is generally 20% to 30% below the lowest frequency to be attenuated. Pulsation filter designs attenuate pulsations at frequencies above the Helmholtz frequency, pass pulsations at frequencies below the Helmholtz frequency, and amplify pulsations at frequencies near the Helmholtz frequency.

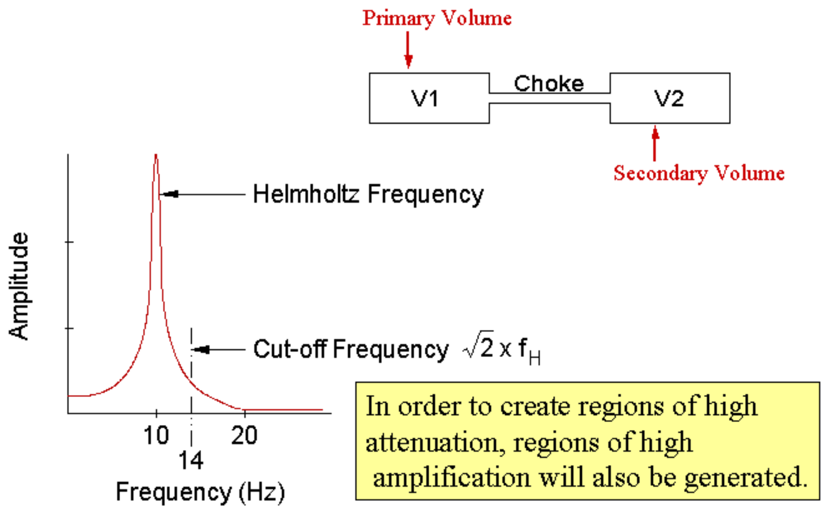

Figure 2-3. Acoustic Filtering

Figure 2-4-A shows a head end (HE) (blue line) and crank-end (CE) (red line) cylinder pressure trace for one revolution of a double-acting cylinder. The y-axis of the figure is pressure, and the x-axis is time. Starting from the left of the figure, the piston is at top-dead-center (at the HE of the cylinder), the suction and discharge cylinder valves are closed, and the piston begins to approach the CE as the two plots are followed to the right. The steep sections of the curves represent the block of time when the pressure in the CE of the cylinder begins to increase and the HE pressure begins to decrease. As each curve begins to level out, the HE suction valves and CE discharge valves open. At the fairly leveled-off sections of each curve, the pressure inside the HE is lower than the suction line pressure, and the pressure inside the CE is higher than the discharge line pressure. These small differential pressures are enough to hold the HE suction valves and CE discharge valves open. The valves close at the location where the CE curve quickly begins to decrease in pressure, and the HE curve quickly begins to increase in pressure. At this point, the piston is at bottom-dead-center (at the CE of the cylinder). Continuing to follow the curves to the right, the steep sections represent the block of time when the HE pressure begins to increase and the CE pressure begins to decrease. As the curves begin to level out the second time, the cylinder valves open (HE discharge valves open and CE suction valves open). The HE discharge valves and CE suction valves remain open until the end of the figure. Then the piston returns to top-dead-center, and the process is repeated 330, 420, 600, and 1,200 times per minute depending on the compressor speed.

Figure 2-4-B shows the suction and discharge flow pulses into and out of the cylinder that occur each revolution. Figure 2-4-C depicts the dynamic spectral flow pulse derived from the flow pulse shape and the unit speed. Compressor loading significantly affects the dynamic spectral flow pulse (U) in the equation shown in Figure 2-4. When the HE and CE are reasonably well balanced (i.e., when the cylinder is double-acting and there are similar flows on the HE and CE), flow pulsation is predominately at two times the compressor running speed (2x; i.e., for a 300 RPM machine, 2x refers to [2 x 300 RPM/(60 sec/min)] or 10 Hz) and even multiples of that frequency. When the HE and CE are unbalanced (i.e., volume unloaders are activated or the suction valve of one end of the cylinder is held open), the spectral content is significantly altered, such that the odd multiples of running speed are more prevalent, with 1x typically being the

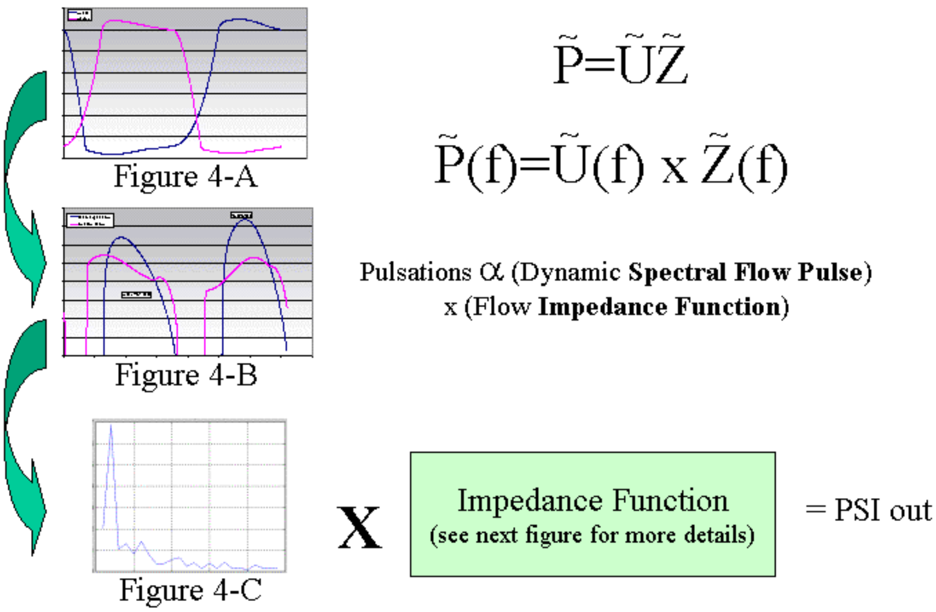

Figure 2-4. Components of a Filter Design

predominant frequency. Pulsations introduced into the system are directly proportional to the dynamic spectral flow pulse times the flow impedance function (P = U x Z). Unlike the dynamic spectral flow pulse (U), the impedance function (Z) can be altered with variations in the pulsation filter bottle volumes, piping lengths and diameters, and pressure drop.

Figure 2-5 depicts an example impedance function. This figure, for the sake of simplicity, shows only the three major responses that are left in the system once a pulsation filter is installed. These three major responses are the Helmholtz response, cylinder nozzle response, and a pass band frequency associated with the choke tube length. Altering the volumes and choke tube diameter and length associated with the pulsation filter sets the frequency placement of the Helmholtz response. Variation of the choke tube diameter and length, diameter being the dominant factor, also affects how much the Helmholtz response is damped. Altering the cylinder nozzle length is the main factor involved with setting the frequency placement of the cylinder nozzle response. The volume of the bottle cylinder chamber can also affect the placement of this response, but the bottle volume is predominantly determined based on attaining sufficient surge volume for the cylinder swept volume (based on the bore cross-sectional area and the stroke of the compressor) and placement of the Helmholtz response. Damping the cylinder nozzle response is typically done using an orifice. Altering the length of the choke tube associated with the pulsation filter sets the pass band frequency and changes the Helmholtz frequency. It is highly important to consider both the location of the Helmholtz response and the pass bands when determining the optimal choke tube length. It is equally important that dynamic pressure drop be accounted for properly. Damping the pass band response is typically accomplished with the choke tube pressure drop. It should be noted that damping or lowering the amplitude of the major responses raises the amplitude of the “attenuation zones.” Multiple changes can be made in the compressor manifold and attached piping system that affect the impedance function of the system.

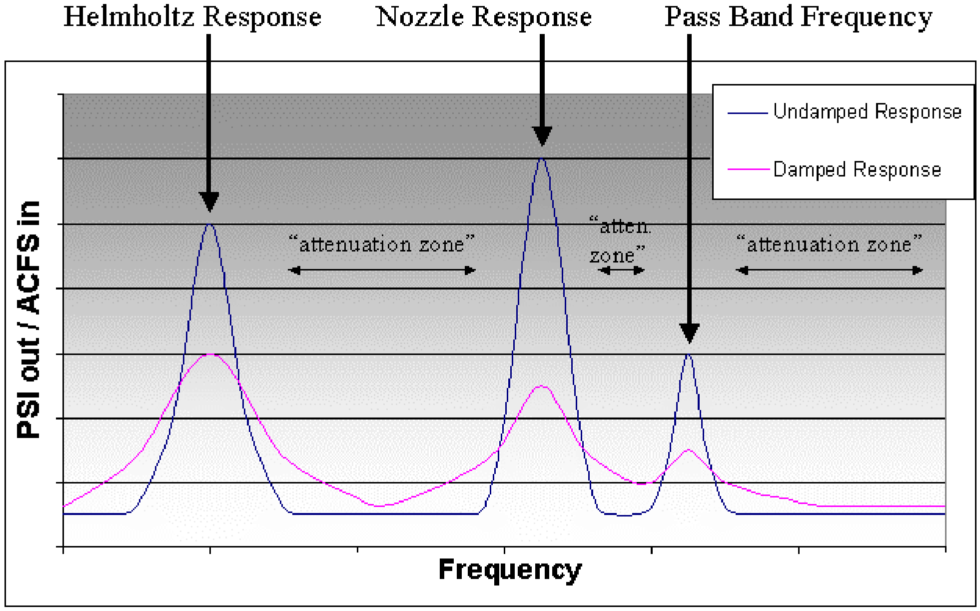

Figure 2-5. Impedance Function

The pulsation designer's challenge is to optimally locate regions of attenuation and avoid the regions of amplification. As shown in Figure 2-5, the designer must contend with the fact that regions of attenuation created from a filter design are not attainable without also creating regions of amplification. Regions of high amplification, such as the Helmholtz response and pass band response, are related to the volume-choke-volume filter system. The other region of high amplification is at the cylinder nozzle resonance frequency, which is associated with the internal dimensions of the compressor cylinder, the length of the cylinder nozzle, and the first surge volume. Because resonance avoidance is the most effective method of pulsation controls, the goal of the pulsation designer is to place the amplification regions (or responses) in frequency windows or gaps between compressor orders when possible. However, with current generation variable high-speed units, it is very difficult, and sometimes impossible, to avoid resonance. Figure 2-6 plots compressor orders for a low-speed machine operating from 270 RPM to 330 RPM and a high-speed machine operating from 750 RPM to 1,000 RPM. As shown in the figure, frequency windows are very small for variable, high-speed compressors. There are no windows above the third order (3x) for a variable, high-speed compressor operating from 750 RPM to 1,000 RPM. A variable, low-speed compressor operating from 270 RPM to 330 RPM has no windows above the fourth order. In cases such as these, resonance management is required. Resonance management involves placing responses at weaker compressor orders and/or damping the response when the coincidence of a response with a compressor order cannot be avoided.

Within the last ten years, industry changed the hardware installation trend, such that low-speed integral units became less common and separable high-speed units became the norm. High-speed separable machines incorporate a smaller footprint and the flexibility of coupling to either an engine or electric motor drive. Changes in industry also influenced changes to pulsation filter design techniques. Figure 2-6 also shows approximate Helmholtz placements for low and

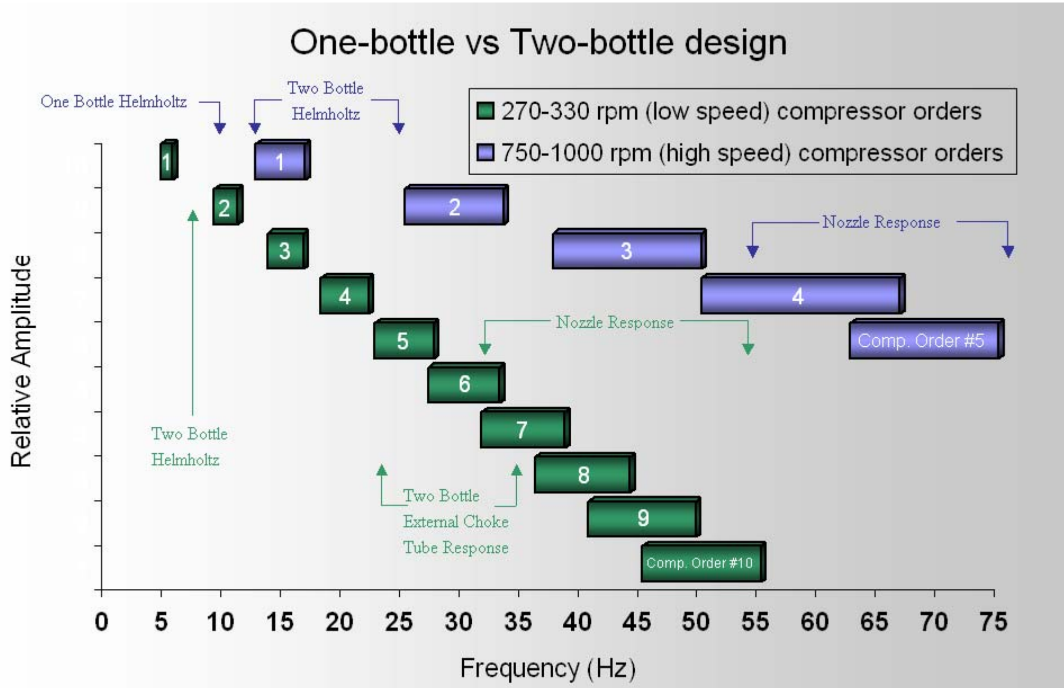

Figure 2-6. One-Bottle versus Two-Bottle Designs in High- and Low-Speed Applications

high-speed compressors using reasonable volumes and choke tubes. Generally, in the absence of resonance, relative amplitude will decrease as frequency increases. As shown in the plots of Figure 2-6, excitation amplitudes are lower at any given frequency for low-speed units than for high-speed units, making resonance management much easier for low-speed units. Increased pulsation amplitudes at the cylinder nozzle response frequency for high-speed machines force the designer to shift the response to higher orders that have lower relative amplitudes. This is typically accomplished by shortening the cylinder nozzle. As previously mentioned, shorter cylinder nozzles lead to increased mechanical coupling between the cylinder and all attached manifolds and piping.

2.1.2 STATE-OF-THE-ART IN PULSATION FILTER DESIGN

State-of-the-art in pulsation design for low-speed compressors involves a two-bottle pulsation filter system. Because the old low-speed units tended to operate in a double-acting mode, the major acoustic response (Helmholtz) frequency associated with the volume-choke-volume filter configuration was often placed between the first and second compressor orders. A two-bottle design on a low-speed unit allows placement of the Helmholtz response between the fundamental (1x) and second compressor orders (2x). Such placement of the Helmholtz response attenuates 2x pulsations and higher frequency pulsations from the piping attached to the secondary bottle. A one-bottle design would typically be unreasonably large in order to accomplish the same results that a two-bottle design can accomplish for low-speed units. Figure 2-7 presents two examples of two-bottle acoustic filter designs for low-speed compressors. A state-of-the-art two-bottle acoustic filter design will consider several key design issues for achieving acceptable performance, including:

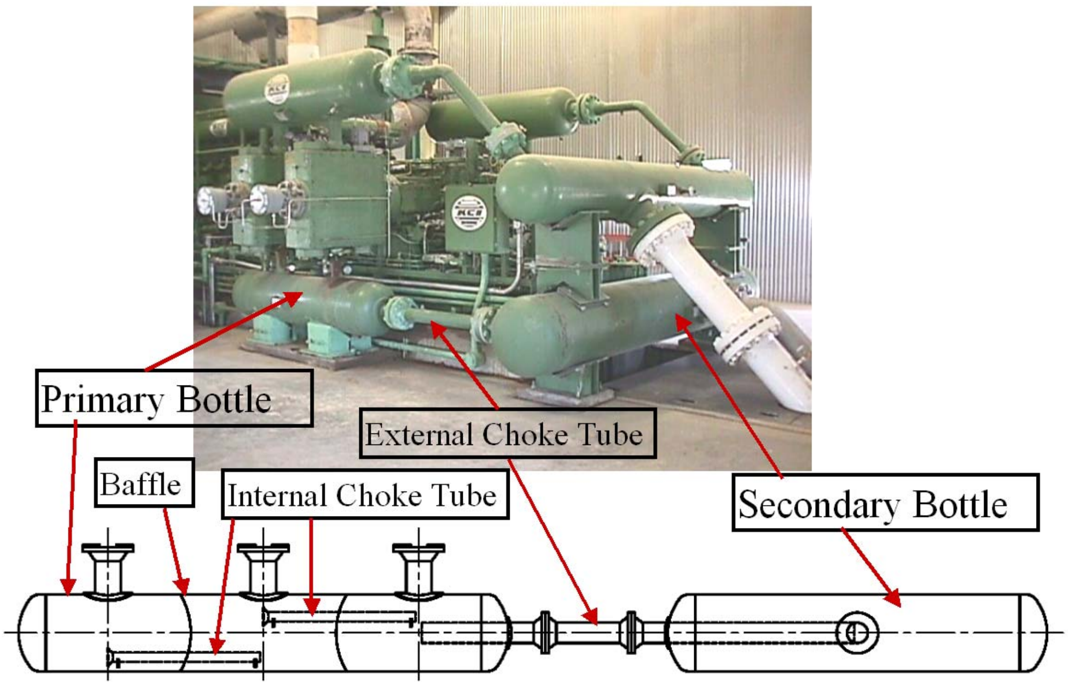

Figure 2-7. Low-Speed Compressor with a Two-Bottle Manifold System Installed and a Two-Bottle, Three-Cylinder Conceptual Bottle Design

- Symmetry for minimizing pass band responses. For example, the external choke tube and secondary bottle may be identical in length, thereby generating a single pass band frequency.
- Baffles and internal choke tubes for establishing separate cylinder chambers and minimizing bottle acoustic shaking forces.
- Center feeding pulsations into volumes such that the length responses of those volumes are minimized. For example, pulsations from the compressor are center fed into each cylinder chamber to minimize excitation of the chamber length response. The secondary bottle is also center fed.

State-of-the-art in pulsation design for high-speed compressors is a one-bottle pulsation filter system. Industry typically wants to operate the newer high-speed compressors with a large variation in capacity to optimize driver operation. This includes single-acting the compressor cylinders. As noted previously, single-acting the cylinders leads to a dramatic increase in 1x pulsation amplitudes. Increased 1x pulsation amplitudes created the need to attenuate the 1x pulsations generated by the compressor. A one-bottle design on a high-speed unit allows placement of the Helmholtz response well below 1x. Figure 2-8 presents examples of one-bottle acoustic filter designs for high-speed compressors. Baffles are located inside the bottle to establish separate chambers for each cylinder and a filter chamber. Separate choke tubes connect each cylinder chamber with the filter chamber. All cylinders are acoustically isolated from each other due to the use of internal baffles and choke tubes and placement of the Helmholtz frequency associated with each volume-choke-volume configuration below 1x. Such placement of the Helmholtz response also attenuates 1x pulsations and higher frequency pulsations from the

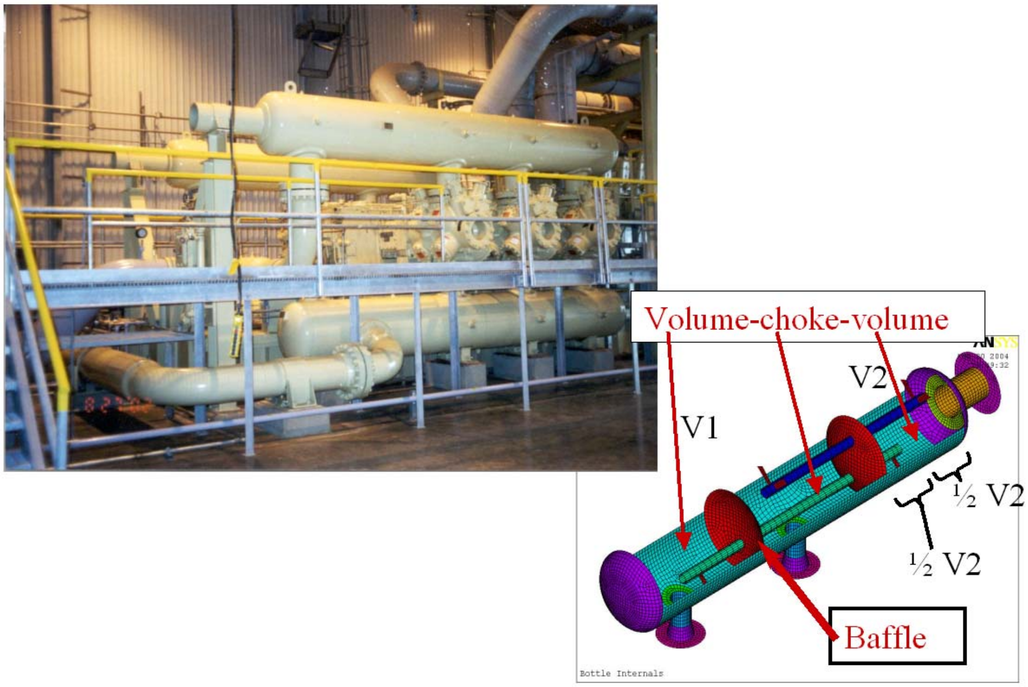

Figure 2-8. Modern High-Speed Compressor with a One-Bottle Acoustic Design Installed and a One-Bottle, Two-Cylinder Conceptual Bottle Design

piping attached to the primary bottle. In some cases, the two-bottle, variable high-speed filter design can be used to place the Helmholtz response below 1x, but that is typically done using a longer choke tube. In high-speed compressor design, it is very important to keep the choke tube lengths as short as possible. Keeping the choke tube lengths short will raise the acoustic length response of the choke tubes and reduce the amplitude of the response in the manifold area. Short choke tubes were not usually a critical design factor for low-speed units because, as previously discussed, the pulsation amplitudes dissipate more rapidly as frequency (and order) increases. For example, a choke tube response at 30 Hz would be at the second order of a compressor operating at 900 RPM, where the relative amplitude is high. That same 30 Hz choke tube response would be at the sixth order of a compressor operating at 300 RPM, where more dissipation would have occurred. Placing the response at a higher frequency does not avoid resonance, but it potentially reduces the response amplitude. A typical one-bottle acoustic filter design can include the following:

- Baffles and internal choke tubes to establish separate cylinder chambers, which minimize bottle acoustic shaking forces and create a secondary volume (V2) for filtering.
- Center feeding pulsations from the compressor into the cylinder chambers to minimize excitation of a chamber length response. Inlets and outlets of internal choke tubes are located at the midpoints of cylinder chambers and line chamber whenever possible to again incorporate center feeding of pulsations.
- A separate choke tube for each cylinder, which typically reduces pressure drop. A limiting factor associated with placement of the Helmholtz response is maintaining a reasonable pressure drop in the choke tubes. A smaller choke tube diameter leads

to a lower Helmholtz frequency. When multiple cylinders flow through a single choke tube (as is the case in a two-bottle filter design when the filter is connected to more than one cylinder), the choke tube diameter must be increased to reduce the pressure loss. The increase in the choke tube diameter raises the Helmholtz frequency. When each cylinder flows through its own individual choke tube (as is the case in the state-of-the-art one-bottle filter design), the choke tube diameter can be decreased without taking excessive pressure loss. Decreasing the choke tube diameter lowers the Helmholtz frequency.

Industry changes led to a need for changes in the pulsation filter design of the compressor manifold. These changes also led to modifications in the methods used for mechanical analyses of compressor manifolds. In the past, beam elements were sufficient for modeling compressor manifolds. With the increased horsepower of the newer high-speed units came increased mechanical shaking forces due, in part, to the acoustic need for shorter nozzles. The shorter nozzles created a stronger coupling between the compressor cylinder, the cylinder nozzle, and the manifold system. Vibration amplitudes due to cylinder stretch exceeded the guidelines published for low-speed compressors. New guidelines had to be established based on acceptable stresses. A shift to three-dimensional finite element shell modeling was necessary to accurately predict the mechanical natural frequencies and associated vibration and stress levels for the newer class of machines. Figure 2-9 shows a finite element shell model of a six-throw horizontally opposed high-speed separable compressor. Figure 2-10 shows the results of a poor manifold design. Multiple gussets and cylinder supports were used to control mechanical vibration that remained in the system. More accurate mechanical modeling of such a system would result in an engineering design with fewer, if any, modifications to the system after system start-up.

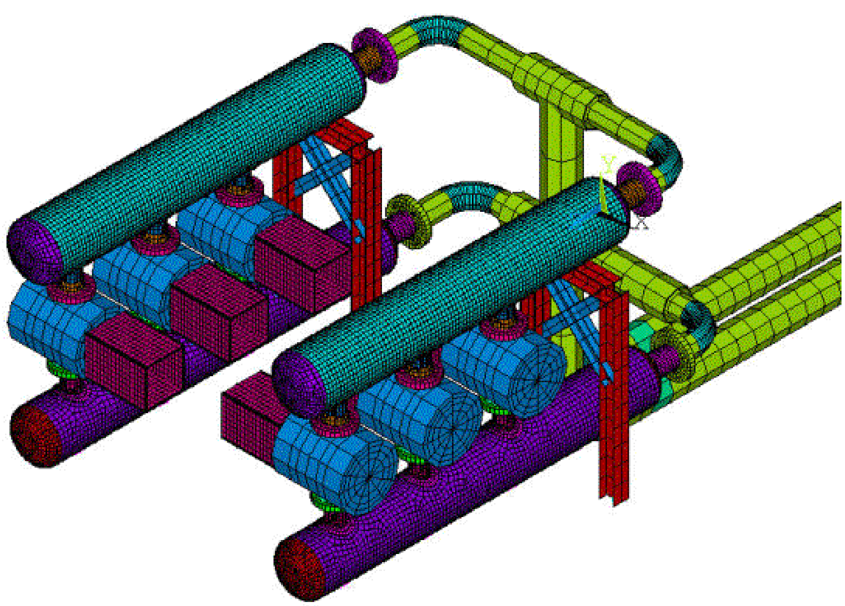

Figure 2-9. Mechanical Compressor Modeling Using FEA Design Tools

Figure 2-10. Modern High-Speed Unit (2,000 to 3,000 HP) – Unsuccessful Design Required Multiple Gussets to Fix the Design Problems

2.1.3 ROLE OF THE PULSATION FILTER BOTTLE DESIGN ON COMPRESSOR PERFORMANCE AND INTEGRITY

Locating the frequency of a nozzle resonance involves a significant design trade-off. The pulsation designer typically wants to shorten the nozzle length in order to place the nozzle response at an order with low excitation amplitude. However, shortening the nozzle length does not always shift the nozzle response high enough and it leads to increased mechanical problems. As previously stated, shorter and stiffer nozzles create a stronger coupling between the compressor cylinder, the cylinder nozzle, and the manifold system. Additional mechanical forces in the piping system can result in failures of small auxiliary lines, cooler vibration, etc. If the nozzles are lengthened in order to improve the mechanics of the system, the pulsations in the system will increase in amplitude, which could still cause piping failures. There is no clear solution to the nozzle resonance problem.

Figure 2-11 depicts the results of an industry survey regarding the efficiency of the existing low-speed compressors fleet. Efficiency of a low-speed compressor is expected to be 90% to 94%. This figure shows that many of the reciprocating compressors in use today are operating closer to 80% efficiency. Factors that are likely to be contributing to this efficiency loss are undersized orifices and, more importantly, poorly designed pulsation filter systems.

An orifice may be installed in a system to damp a particular response. An orifice that is sized too small will likely damp the pulsation response but may take more pressure drop than necessary. Unnecessary pressure drop is wasted horsepower and lost capacity.

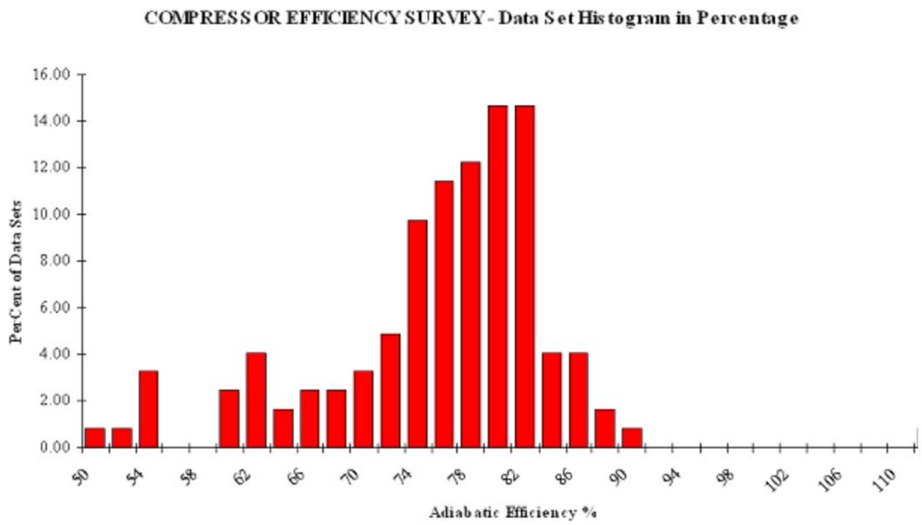

Figure 2-11. Need to Improve Overall Efficiency of Reciprocating Compressors

Poor manifold layouts can lead to extremely poor performance. Figure 2-12 presents two manifold designs and the associated measured cylinder pressure data from a low-speed integral. The original bottle design was limiting the compressor to 46% efficiency. It was determined that cylinder-to-cylinder interactions were responsible for high amplitude pulsations and performance degradation. Therefore, a new design focused on isolating the cylinders was developed. The new design placed the Helmholtz response associated with the two cylinder chambers and the external piping connecting the two chambers below 1x. Compressor efficiency of approximately 92%, nearly twice the original efficiency, was the result of an improved pulsation filter design. Figure 2-13 presents the results of a detailed performance prediction for a modern high-speed separable unit with various configurations of bottle designs. This study demonstrated that a bottle redesign would allow both a complete removal of the nozzle orifice and single-acting operation maintaining a high turndown capability. An estimated 9% increase in compressor efficiency is expected with the new bottle design.

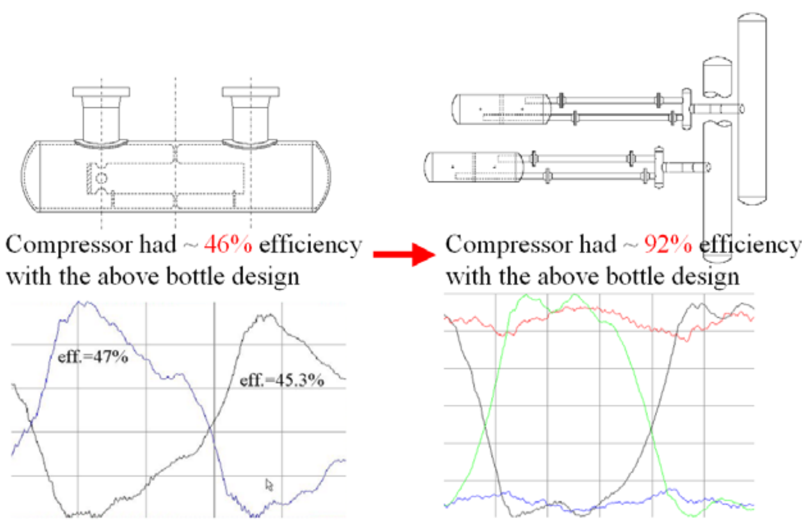

Figure 2-12. Bottle Design Affects Compressor Performance

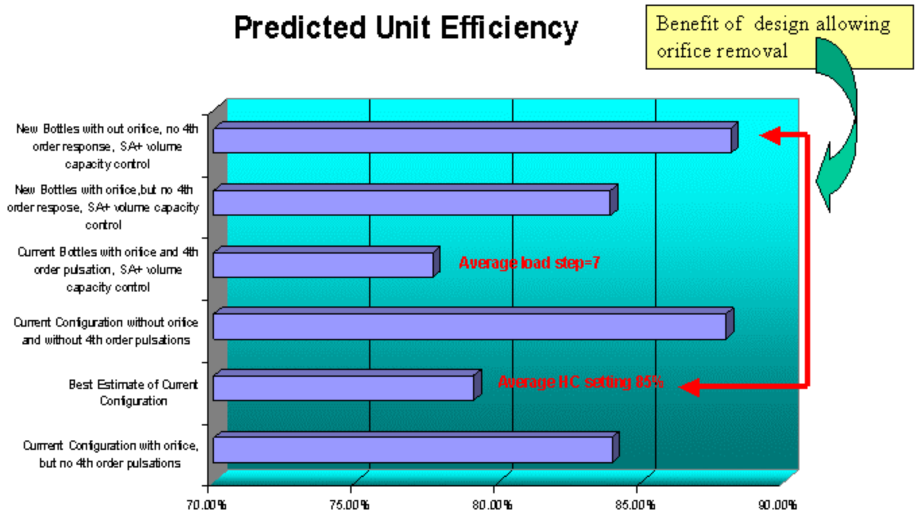

Figure 2-13. Pulsation Design Affects Compressor Performance

2.1.4 LIMITATIONS IN STATE-OF-THE-ART PULSATION DESIGNS AND IN PULSATION CONTROL TECHNOLOGY

2.1.4.1 Limitations of Acoustic Modeling Techniques

The Analog Simulator, developed in a cooperative effort between industry and SwRI, solved the linear acoustic wave equations by stimulating (with current pulses representing valve flow) circuits of resistors, capacitors, and inductors. The voltages developed in these networks represented pulsation levels in the manifold and attached piping. The Analog Simulator tool was continuously refined over a 40-year span to improve the cylinder models (generating more accurate current pulses more closely representing valve flow) and damping estimates. In the late 1980s and early 1990s, digital data acquisition of the voltage/pulsation levels was incorporated, significantly improving the reporting capability of the Analog Simulator design tool. The Analog Simulator was licensed to several companies in order to provide effective design services to the industry. In the mid-1980s through mid-1990s, digital design tools were developed. This software also solved the linear acoustic equations. Piston face velocities at the cylinder HE and CE valves were used to generate the fundamental pulses that traveled through the attached piping system, and calculations were made in the frequency domain. An improved digital code was then further developed by SwRI to allow more accurate modeling of the system. As shown in Figure 2-14, SwRI's advanced Interactive Pulsation Performance Simulation (IPPS) digital design code models the cylinder portion of the acoustic model in the time domain. After the driving point impedance matrix is calculated for the attached piping, the cylinder pressures and flow rates are generated using a fundamental thermodynamic model of the cylinder. The code incorporates a dynamic valve model with flow squared losses, and it can be enhanced to any level of sophistication, including valve stiction, non-linear spring forces, or other effects. This allows better representation of the compressor-generated flow pulses that propagate through the attached piping system. Since the valve flow is a function of the difference between the cylinder pressure and the pressure behind the valve, estimation of the valve mass flow rates is an iterative process. The valve flow is initially estimated assuming constant pressure behind the valves; the

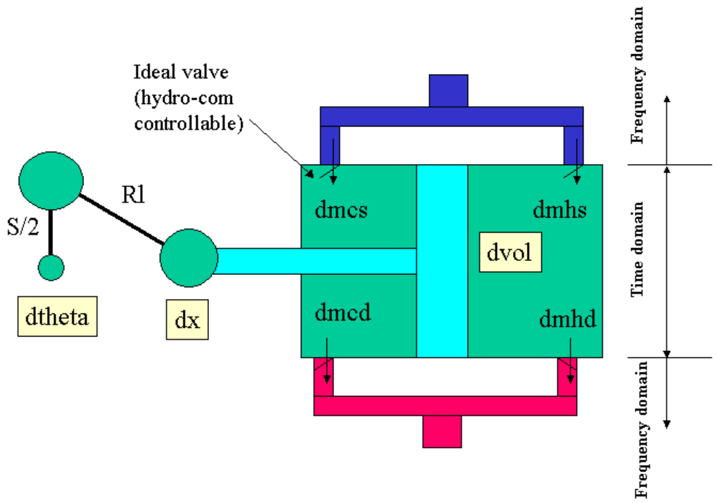

Figure 2-14. SwRI's Digital Code Includes Time Domain and Frequency Domain Modeling

dynamic pressure behind the valve is then computed. With the updated valve dynamic pressures, the cylinder pressures and flow rates are recomputed. This iterative process allows the frequency domain solution of the piping attached to the cylinder to be combined with the time domain solution of the cylinder pressures. When the process is complete, the full cylinder-to-cylinder and cylinder-to-piping interactive valve mass flows have been computed. The dynamic pressures throughout the piping system are then calculated in the frequency domain. Figure 2-15 shows a comparison between the IPPS model and the piston face flow model predictions for the predicted cylinder pressures. Figure 2-15-A shows that the predicted cylinder pressure based on piston face flow codes has a significant amount of separation from that of the measured pressure. As seen in Figure 2-15-B, the cylinder pressure trace calculated by the IPPS modeling tool follows the measured trace very well. Note that both cases result in the same cylinder flows, but the spectral components of the IPPS predicted flow rates are much more accurate. Improved predictions of compressor pulsations have improved the pulsation design capabilities over the years. Further advancements in design tools are required for higher speed, higher horsepower units.

The acoustic equations used in many modern digital acoustic design tools, including SwRI's IPPS digital acoustic design tool are accurate up to 140-150 dB (less than 1 PSI). For low-speed machines or fixed high-speed machines, where resonance avoidance is possible, these modeling tools have been sufficient. However, for variable high-speed machines, where resonance management is required, more sophisticated tools are needed to better quantify amplitude predictions. Frequency domain modeling codes assume that pressure waves at all frequencies travel at the same speed. However, non-linear effects can cause these pressure waves to travel at different speeds, resulting in phase shifts that may be significant. Figure 2-16 compares analog, IPPS, and field predictions in the cylinder nozzle region of a compressor. Model pulsation predictions were lower than the field measurements at 1x, at the nozzle response, and at frequencies above 110 Hz. Time domain code has been developed and is an option for acoustic modeling; however, further investigation into this type of modeling and comparison with laboratory grade data is needed to determine whether it will be beneficial to support technology development.

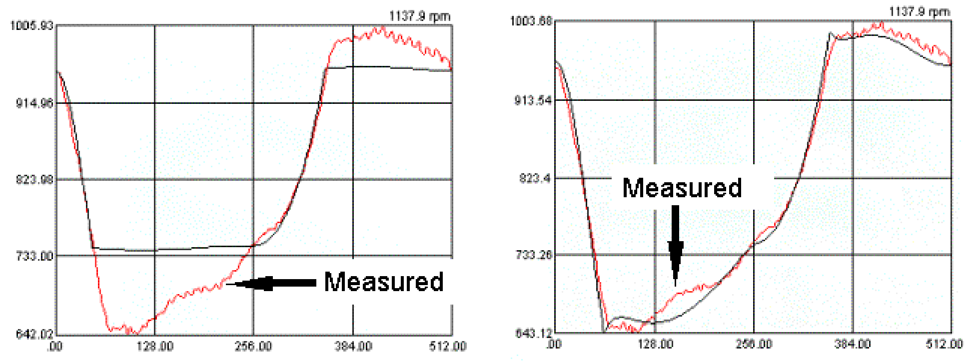

Figure 15-A
Cylinder Pressure using Piston Face Velocity Valve Flows

Figure 15-B
Cylinder Pressure using IPPS acoustic modeling tool

Figure 2-15. Comparison of Digital Acoustic Design Predictions – Cylinder Pressures

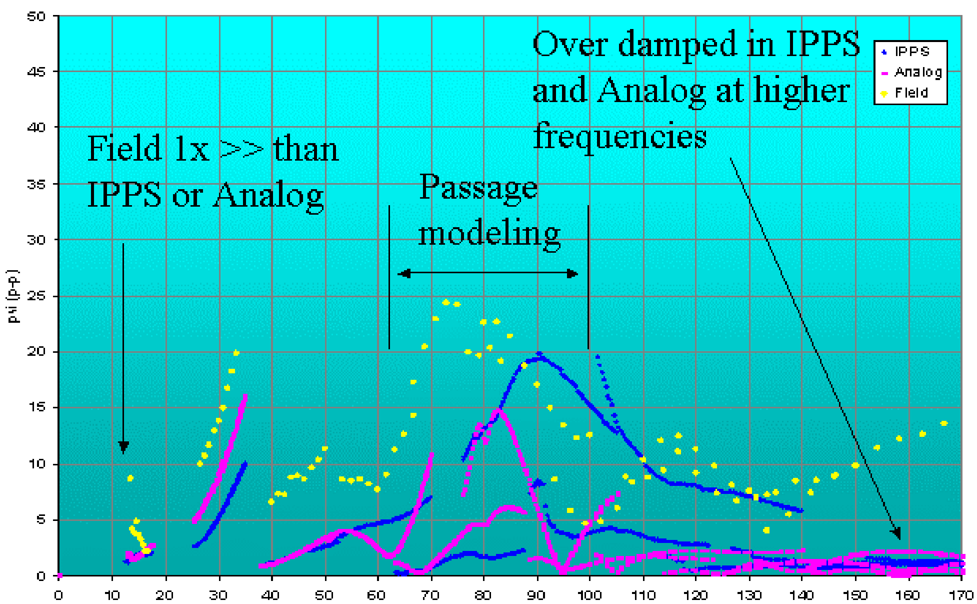

Figure 2-16. Comparison of Design Predictions

2.1.4.2 Limitations Involved with Nozzle Response Placement

The nozzle response is an extremely strong response in high-speed, high horsepower units. Changes can be made to the nozzle length, but typically, it can only be damped with the installation of an orifice in the cylinder nozzle. Some literature suggests that the use of multi-hole orifices (or perforated orifices) is better than a typical single bore orifice. Whether an orifice is a single bore or multi-hole orifice, it is still a high-pressure loss element that can be detrimental to compressor performance. There is a need for a minimal pressure loss method of controlling the nozzle response.

2.1.4.3 Limitations in the Reduction of 1x Pulsations

Residual 1x pulsations are inherent in high horsepower units due to the increased capacity controls that are used. The 1x pulsations are particularly high when a unit is operating with multiple cylinders in single-acting mode. There are times when filtering the fundamental frequency does not provide enough attenuation of 1x pulsation, sometimes leading to failures of small auxiliary lines, cooler vibration, etc.

Residual 1x pulsations are becoming more of a problem with low-speed units as well. A desire to increase the flexibility in capacity control resulted in more single-acting operation of low-speed machines. Single-acting a compressor increases 1x pulsations. First order pulsations are typically not filtered on a low-speed machine. Helmholtz responses for most existing low-speed machines have been placed between 1x and 2x for reasons previously described. This can lead to high vibrations in the attached piping system. Lateral lengths, for example, are often at or near 1x.

One possible solution to the residual 1x pulsations in compressor systems could be the use of a side branch Helmholtz resonator. A side branch Helmholtz absorber consists of a span of piping connecting the main piping system to a surge volume. The connecting piping is attached to the main piping system at a critical location, such that pulsations corresponding to the natural frequency of the side branch Helmholtz design are absorbed from the system. The diameter and length of the connecting piping and the size of the surge volume determine the natural frequency of the side branch Helmholtz resonator. As shown in Figure 2-17, the installation of a side branch Helmholtz resonator can significantly improve the levels of the 1x pulsations in the system. Pulsations at 1x were initially high as shown in the raster plot on the left. Installation of the side branch resonator lowered 1x pulsation as shown in the raster plot on the right. This is an effective solution for fixed speed machines; however, this is not an ideal solution for variable speed machines.

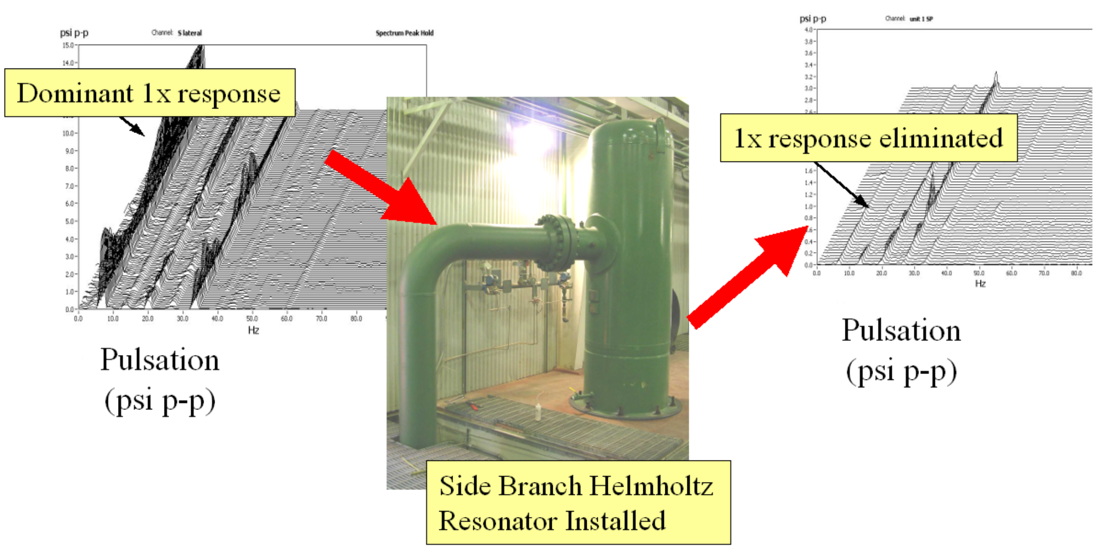

Figure 2-17. Effectiveness of a Side Branch Helmholtz Resonator

2.1.5 STATE-OF-THE-ART SUMMARY

The state-of-the-art in pulsation design and control technology has evolved as compressor technology installed by the industry changed. Designs for low-speed compressors are more mature, with fewer critical issues. Relatively recent high-speed, high horsepower compressor designs are placing significant challenges on the pulsation control designer. Critical technology needs exist to achieve reliable, efficient compressor options, including:

- *Design Tools* – Resonance management requires improved predictions of amplitudes to balance trade-offs between pulsation control and performance. The basic assumptions of the linearized acoustic equations allowing frequency domain solutions to be used may well be a limiting factor in the accuracy of developing new technology solutions. Efficient time domain solutions, at least in the near field of the compressor valves and cylinder nozzle regions, will likely improve the reliability of amplitude predictions. These are critical for the industry to continue with the current trend of technology enhancement.
- *Cylinder Nozzle Response* – The cylinder nozzle response represents the single most important challenge to high horsepower, high-speed variable speed units. Significant reductions in unit horsepower and capacity occur through use of pressure drop elements required to control pulsation amplitudes. Technology must be developed to allow control of the nozzle response, without significantly lowering cylinder performance. This is particularly important if compressor flexibility (in turndown ratio) is required, or if the trend toward higher speeds continues.
- *Residual 1x* – Control of residual low frequency pulsations must be improved. For high-speed, high horsepower units, damaging residual 1x pulsations propagated out to the lateral lines can occur with current designs. For low-speed designs, technology for residual 1x and 2x control will be required to ensure the industry demand for unit flexibility and load control.

Finally, future compressor designs, possibly involving advanced unloader technology or even radically different operating geometry, will require pulsation control technology. The state-of-the-art in design must be able to meet these emerging technologies and reduce the likelihood of dangerous start-up scenarios and industry wide problems that can occur with any significant technology change.

2.2 PULSATION CONTROL NEW CONCEPTS DISCUSSION

2.2.1 CONCEPT GENERATION

After identifying the needs in the area of pulsation control, a brainstorming session was held to generate ideas for addressing the needs. Then the ideas were categorized and short descriptions were given to each. Table 2-1 shows the list of ideas and descriptions.

Table 2-1. Pulsation Control Ideas and Descriptions

PULSATION CONTROL IDEAS	EXPLANATION OF THE IDEA
Why are there rigid connections from compressor to bottles? Why not use flexible hose?	A flexible hose could be installed in place of the cylinder nozzles.
Why not go to down-connected bottles?	Keep the discharge cylinder nozzles down connected, but also make the suction cylinder nozzles down connected.
Should we include vibration control?	Question about whether or not vibration control should be included in this part of the research.
Active dampers	Pulsation damping devices that actively change geometry, flow area, etc.
Add length by walking stick style	Vary the length/volume of the filter bottle(s) by sliding one end of the bottle in or out.
How good are multiple hole orifices?	Use a multiple hole orifice (perforated orifice) in place of a single hole orifice.
What about oblong nozzles bigger than old style?	Use an oblong cylinder nozzle (bigger than the older style) in place of the current round design.
Active variable nozzle (tapered cone)?	Use a cylinder nozzle that is cone shaped and can be varied in length.
Active pulsation control	Have variable aspects of the pulsation control system change such that it adapts to the changing acoustics of the system.
Active noise cancellation concept	Have variable aspects of the pulsation control system change such that pulsations are canceled.
Passive/active noise cancellation	Have aspects of the pulsation control system that vary (in an active or a passive manner) such that pulsations are canceled.
Active bladder control	Have the filter bottle actively change volume by way of a volume change similar to that of a balloon inflating/deflating.
Active control of damping device (multi-hole orifice gate valve)	In place of a single hole orifice, use a multi-hole orifice (perforated orifice) along with a gate that can cover some of the orifice.
Passive bladder control	Have volumes that change based on the conditions of the system.
Staged volume control	Allow a certain number of compression cycles to enter into a volume then release the gas from the volume at a lower frequency.
Circulator	Have a rotating device that interrupts the pulsations.
Bottle with pockets that can be opened and closed	Have pockets attached to the bottle volumes, such that opening or closing the pockets could change the volumes.
What are new geometries that would work?	Use a new geometry for the filter bottle and/or the attached piping.
Lined bottles (to change volume and shape)	Use a variable thickness liner inside the filter bottles.
Electro-rheological** fluids for liner	Line the filter bottles with an electro-rheological fluid.
Non-Newtonian fluids (impacted by vibrations)	Line the filter bottles with a non-Newtonian fluid.
Control shaking forces by changing bottle geometry (area) (i.e., not cylinder throughout)	Change the geometry (impacted areas) of the filter bottles such that the shaking forces would decrease.
Tapered nozzles	Use cone shaped nozzles.
Tuned vibration absorbers	Control the mechanical vibrations using vibration absorbers.
Surface damping treatment	Use a surface treatment on the piping such that the vibrations are damped.
Constrained layer damping	Use constrained layer damping on the exterior of the filter bottles and/or the attached piping.
Honeycomb in bottles or lines (lined piping, lined bottles)	Use multiple pipes inside of bottles and/or piping.

PULSATION CONTROL IDEAS	EXPLANATION OF THE IDEA
Multiple side branch absorbers (SBA)	Use multiple SBAs in a system.
Multi-passive (SBA)	Design a SBA such that it will naturally change the frequency at which it is tuned based on the current system conditions.
Honeycomb walls with damping fluid	Use multiple pipes inside of bottles and/or piping, and line the pipes with a damping fluid.
Polished choke tubes	Use polished choke tubes.
Choke tube enhancement	Change the choke tube such that it is just as effective, but takes less pressure drop and or takes up less space.
Tunable choke tube (use shutters for the choke tube entrance and exit)	Use a shutter type apparatus to be able to vary the flow area of an orifice.
Same as above, but with bladders (pinch valve)	Use a bladder to vary the amount that the flow is restricted.
Multi-frequency SBA (more than one volume in a single bottle)	Use a SBA within an SBA.
Multiple SBAs on the cylinder nozzle	Use multiple SBAs on the cylinder nozzle.
Variable Volume SBA	Use a SBA that can be tuned by changing its volume.
Infinite length nozzle	Gradually increase the number of holes (perforations) in the cylinder nozzle as it projects into the bottle.
Use phasing (varied manifold lengths) to cancel pulsations	Vary the length of the cylinder internals such that pulsations are canceled.
Variable volume filter bottle	Vary the volume(s) of the filter bottle.
Use shutter type design for variable orifice diameters	Use a shutter type design to vary the flow area of an orifice.
Damped SBA (with orifice)	Install an SBA with an orifice installed such that it damps the tuned SBA frequency response (if possible).
Higher Order Filters	Use more than 2 volumes and a choke tube per compressor cylinder to create a higher order filter.

2.2.2 Short List of Concepts

After generating, categorizing, and describing the ideas, the ideas were then narrowed down and rated in Table 2-2. The concepts that had the highest rankings were included in the following “short list” and considered reasonable concepts for further evaluation. Characteristic reasons for placing each concept on the short list are stated in the subsequent discussions in this section.

- Infinite length cylinder nozzle (perforated nozzle projection)
- Tapered nozzles
- Variable volume SBA (side branch absorber)
- Damped SBA (with orifice)
- Multi-frequency SBA (more than one volume in a single bottle)
- Pockets on each filter bottle volume (variable volume filter bottle)
- Higher order filters
- Multiple hole orifices

Table 2-2. "Long List" of Concepts

	Pulsation Control Ideas — Weighing Factors - ->	Cost 25%	Accept. 25%	Maint. 25%	Retrofit 15%	Time 10%	Total
Orifices / Pulsation Damping Devices	Multiple hole orifices	5	5	5	5	5	5.00
	Flexible hose cylinder nozzles [D]	2	4	4	2	2	3.00
	Honeycomb in bottles or line (lined piping, lined bottles)	3	4	5	2	2	3.50
	Honeycomb in bottles or lines that are lined with damping fluid	2	4	3	2	2	2.75
	Active pulsation damping devices	4	5	4	4	4	4.25
	Multi-hole orifice with a gate valve (active damping device)	4	5	4	4	4	4.25
	Shutter type orifice (variable diameter)	3	4	2	5	4	3.40
Nozzle Response	Infinite length nozzle	5	5	5	2	4	4.45
	Tapered nozzles	4	5	5	2	5	4.30
	Multiple SBAs on each cylinder nozzle [D]	3	3	5	2	3	3.35
	Active tapered cone nozzle	2	3	3	2	3	2.60
Filters (changing volume of chambers)	New filter bottle geometry (control shaking forces and pulsations)	3	4	5	1	2	3.35
	Line filter bottles (change volume and shape)	3	3	4	2	2	3.00
	Line filter bottles with an electro-rheological fluid	1	2	3	2	1	1.90
	Line filter bottles with a non-Newtonian fluid	3	3	4	2	1	2.90
	Variable volume filter bottles (general idea)	4	4	4	2	3	3.60
	Passive bladder on filter bottles	2	3	3	3	3	2.75
	Pockets on each filter bottle volume	4	5	5	3	3	4.25
	Add bottle volume using walking stick method	4	4	4	2	3	3.60
	Active pulsation control [D]	3	3	3	2	2	2.75
	Active bladder on filter bottles	2	3	2	2	1	2.15
Pulsations Freq. Content Altered	Circulator	3	4	4	4	3	3.65
	Cancel pulsations using phasing (varied manifold lengths)	4	4	5	2	3	3.85
	Active noise cancellation concept	3	4	4	2	3	3.35
	Passive/active noise cancellation	3	4	4	2	3	3.35
	Active pulsation control [D]	3	3	3	2	2	2.75
	Staged volume control	3	4	3	4	2	3.30
	Variable volume SBA (side branch absorber)	4	4	4	4	4	4.00
	Damped SBA (with orifice)	4	4	5	4	5	4.35

	Pulsation Control Ideas — Weighing Factors -->	Cost 25%	Accept. 25%	Maint. 25%	Retrofit 15%	Time 10%	Total
	Multiple SBAs	4	2	5	4	4	3.75
	SBAs that can each tune to multiple frequencies (multi-passive SBAs)	4	4	3	4	3	3.65
	Multi-frequency SBA (more than one volume in a single bottle)	4	4	4	4	4	4.00
	Higher order filters	4	4	5	4	2	4.05
	Multiple SBAs on each cylinder nozzle [D]	3	3	5	2	3	3.35
Choke Tube	Choke tube enhancement	4	4	5	2	3	3.85
	Polished choke tube	5	5	5	2	3	4.35
	Tunable choke tube (use shutters at entrance and exit)	3	4	3	2	2	3.00
	Tunable choke tube (use bladders at entrance and exit)	3	4	4	2	2	3.25
Mechanical	Oblong cylinder nozzles larger than old style	4	4	5	2	4	3.95
	Tuned vibration absorbers	4	4	4	4	3	3.90
	Surface damping treatment (piping)	1	3	5	1	2	2.60
	Constrained layer damping (piping)	1	3	5	1	2	2.60
	Flexible hose cylinder nozzles [D]	2	4	4	2	2	3.00
	Down-connected suction and discharge bottles	4	4	5	1	3	3.70

[D] = Duplicate (listed in more than one category)

Cost — relative cost of including the idea in the system
(scoring range: 1 = high cost, 5 = low cost)
Acceptance — measure of how acceptable the idea would be to the industry
(scoring range: 1 = not acceptable, 5 = easily accepted)
Maintenance — upkeep added to the system when the new idea is installed
(scoring range: 1 = high maintenance, 5 = minimal maintenance)
Retrofit — ease of installation in an existing system
(scoring range: 1 = easy installation, 5 = complex installation)
Time — time required to research and analyze the idea
(scoring range: 1 = very timely, 5 = short amount of time)

- Active pulsation damping devices (variable beta orifice)
- Multi-hole orifice with a gate valve (active damping device)

2.2.2.1 Infinite Length Nozzle

The infinite length cylinder nozzle (ILN) is focused on resolving the cylinder nozzle response issue. The physics behind the ILN is to create a non-reflective end of the cylinder nozzle using a perforated nozzle projection as shown in Figure 2-18. The preliminary results using the ILN's code showed a nearly 50% reduction in nozzle pulsations with minimal pressure drop and negligible increase in 2x pulsation (see Figure 2-19). The ILN will conceptually allow the cylinder nozzle to be lengthened to improve mechanical response.

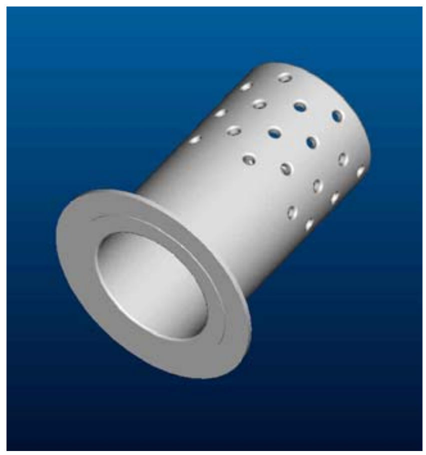

Figure 2-18. Infinite Length Nozzle Concept

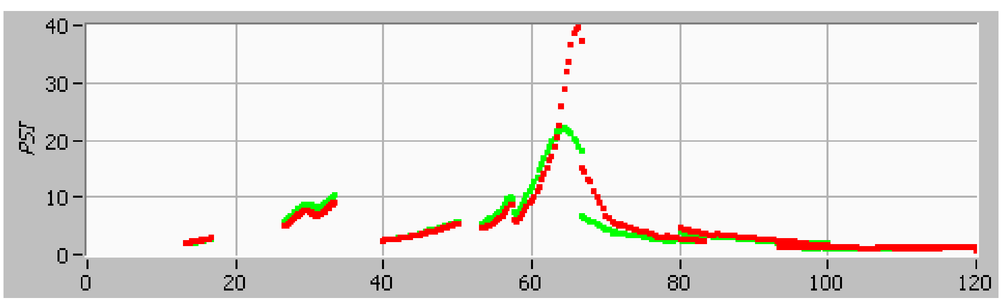

Figure 2-19. ILN Pulsation Response (Red Curve Without ILN, Green Curve With ILN)

2.2.2.2 Tapered Cylinder Nozzle

The tapered cylinder nozzle also focuses on resolving the cylinder nozzle response issue. The idea behind this concept is to create a non-reflective cylinder nozzle end using a gradual expansion of the cylinder nozzle diameter up to the bottle, as shown in Figure 2-20. The preliminary analysis results show a nearly 50% reduction in nozzle pulsations with some increase in higher order response, which has lower amplitudes. The 2x pulsations are reduced, and the pressure drop is minimal, see Figure 2-21. This concept also will allow a longer nozzle for improving mechanical response.

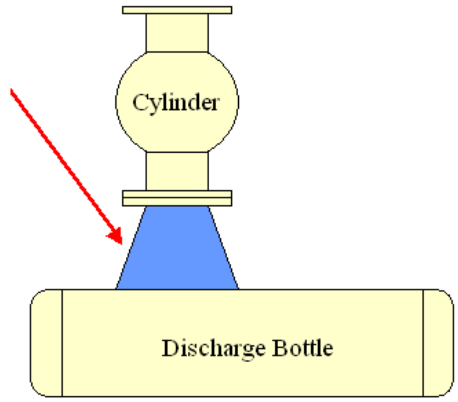

Figure 2-20. Tapered Cylinder Nozzle Concept

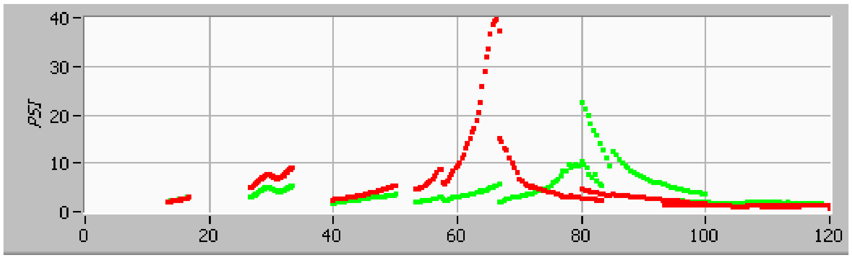

Figure 2-21. Tapered Nozzle Response (Red Curve Without Tapered Nozzle, Green Curve With Tapered Nozzle)

2.2.2.3 Variable Volume Side Branch Absorber

The variable volume or tunable side branch absorber (SBA) focuses on resolving the pulsation issue of residual low frequency pulsations by altering the frequency of the responses found in the piping. A concept drawing for a variable volume SBA on a lateral line is shown in Figure 2-22. The preliminary results indicate first order and/or second order pulsation levels can be significantly reduced. A variable volume may not be necessary for the removal of the 1x or 2x component; however, removal of both components may require a variable volume. Further analysis is needed to correlate data for damping effects, and frequency ranges need to be understood. An example of field data showing the effectiveness of a fixed SBA on a fixed speed compressor was shown in the state-of-the-art technology review (Figure 2-17).

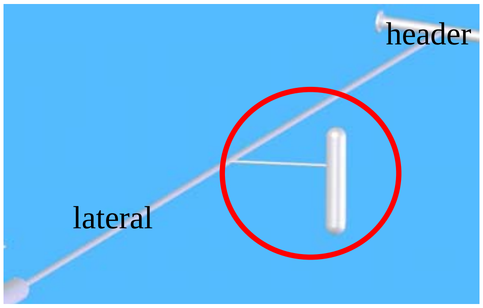

Figure 2-22. Variable SBA Concept Applied to the Lateral Piping

2.2.2.4 Side Branch Absorber on a Cylinder Nozzle

The variable volume SBA on a cylinder nozzle focuses on resolving the cylinder nozzle response. A conceptual drawing is shown in Figure 2-23. The preliminary results indicate a 40% to 50% reduction in pulsation with a fixed volume and possibly better performance with a variable volume, see Figure 2-24. The pressure drop is minimal.

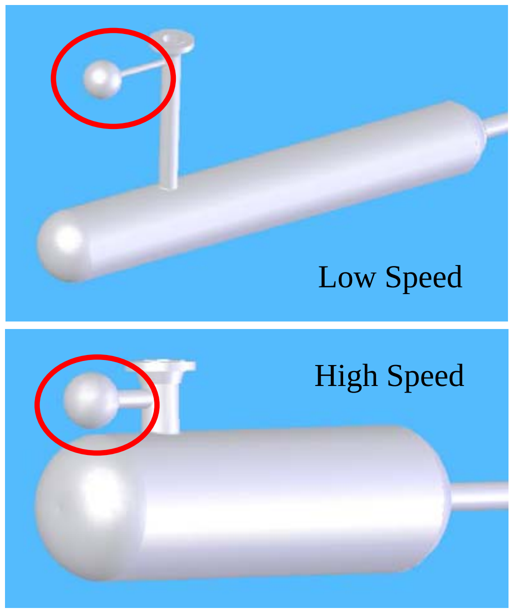

Figure 2-23. SBA Concept Applied to the Cylinder Nozzle

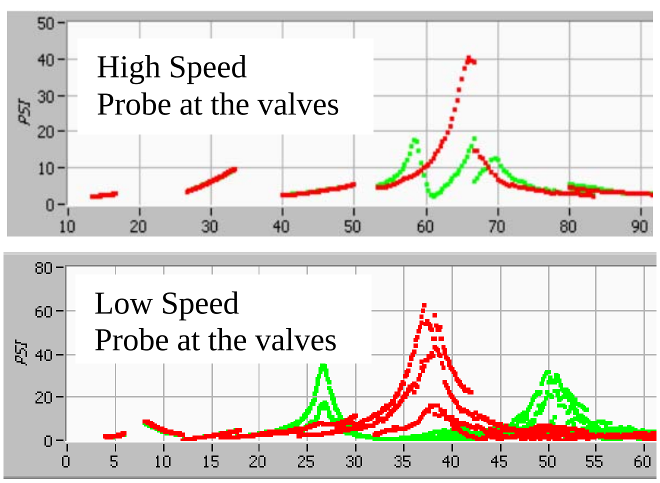

Figure 2-24. Cylinder Nozzle SBA Response (Red Curve Without SBA, Green Curve With SBA)

2.2.2.5 Variable Geometry Filter Bottle

The variable geometry or higher order filter bottle focuses on resolving the issue of residual low frequency pulsation by altering the filter system response frequencies. Higher order filtering can be accomplished by tuning the geometry of the bottle as the speed of the machine is varied or more than two volumes and more than a single choke tube per cylinder. This concept addresses a broader range of frequency responses.

2.2.2.6 Orifices/Pulsation Damping Devices

Multiple hole orifices (also known as perforated orifices), active pulsation damping devices (variable beta orifice), and multi-hole orifice with a gate valve (active damping device) focus on resolving the residual low frequency pulsations with the use of various damping mechanisms. Perforated orifices currently exist; however, the effectiveness of these orifices needs to be further investigated/researched. Variable beta orifices and active damping devices could be effective; however, they are last resort solutions due to their dependence on pressure drop as a means for controlling the pulsations. Model simulation results show that decreasing the beta ratio (smaller orifice bore) when single acting (less flow) can be significantly beneficial for the acoustic of the system.

2.3 PULSATION CONTROL CONCEPT RANKING AND SELECTION

After the short list was developed based on the original rating criteria and some preliminary analyses were performed (see the previous section of this report), more thorough ranking criteria were applied to the short list of ideas as noted in Table 2-3. In order of each criterion's weighting factor, the criteria used were effectiveness, efficiency, life and reliability, operation and maintenance cost, retrofitable, development risk, synergy, installed cost, economic viability, and maturity. The updated ranking criteria were used to help further reduce the short list of concepts down to a more manageable list of six concepts that could be researched and/or developed during the anticipated time frame of this research project. The top concepts based on the updated criteria are as follows:

- Infinite Length Cylinder Nozzle (ILN)
- Tapered Cylinder Nozzle (TCN)
- Multi-frequency Side Branch Absorber (SBA)
- Damped SBA (damped with an orifice)
- Higher Order Filters
- Variable Geometry Filter Bottles

2.4 PULSATION CONTROL PROOF-OF-CONCEPT

2.4.1 RECIPROCATING COMPRESSION TEST FACILITY MODIFICATION

In order to perform proof-of-concept testing, modifications were made to the Reciprocating Compression Test Facility (RCTF). The original RCTF bottle design (empty bottles) did not have all of the acoustic characteristics that were needed to allow sufficient evaluation of the top pulsation control concepts. An acoustic filter design was developed for the RCTF such that the necessary characteristics would be present in the system, and the design had acoustic characteristics that were more representative of a typical filter design of high-speed units.

Table 2-3. Updated Ranking Criteria Applied to the Short List of Concepts

	Effectiveness	Efficiency	Life & Reliability	O&M Cost	Retrofitable	Development Risk	Synergy	Installed Cost	Economic Viability	Maturity	Summary
Infinite Length Nozzle	Excellent	Excellent	Excellent	Excellent	Excellent	Excellent	Good	Excellent	Fair	Good	Excellent
Tapered Nozzle	Excellent	Excellent	Excellent	Excellent	Poor	Excellent	Excellent	Fair	Fair	Good	Excellent
Multi-frequency SBA	Good	Good	Good	Fair	Fair	Good	Excellent	Good	Good	Fair	Good
Damped SBA	Fair	Fair	Excellent	Good	Fair	Fair	Fair	Good	Good	Fair	Good
Higher Order Multiple Filter Bottle	Excellent	Fair	Excellent	Good	Poor	Fair	Good	Fair	Good	Poor	Good
Variable Geometry Filter Bottle	Excellent	Good	Good	Fair	Poor	Fair	Good	Fair	Good	Poor	Good
Multiple Hole Orifice	Poor	Poor	Excellent	Good	Excellent	Fair	Fair	Excellent	Fair	Good	Fair
Variable Beta Active Pulsation Damp	Poor	Poor	Fair	Fair	Excellent	Good	Fair	Good	Fair	Poor	Fair
Multi-hole Orifice w/ Gate Valve	Poor	Poor	Fair	Fair	Good	Fair	Fair	Good	Fair	Fair	Fair

The reciprocating compressor in the RCTF is a single cylinder 40 horsepower compressor that has an 8.5-inch bore, a 3-inch stroke, and a speed range of approximately 300 to 1,000 RPM. Originally, pulsation control for the system used a single bottle design on both suction and discharge as illustrated in Figure 2-25. This original design had some acoustic characteristics that were not desirable when installing and evaluating the ARCT pulsation control concepts, specifically:

Figure 2-25. Original Reciprocating Compression Test Facility (RCTF) Bottle Design

- There was no acoustic filtering (empty bottles); therefore, there was no Helmholtz resonance.
- Residual 1x pulsations were very low; therefore, the pulsations were not suitable for the evaluation of the tunable (variable frequency) SBA.
- The cylinder nozzle responses were at relatively high compressor orders, which resulted in low response amplitudes (experimental and simulation data). The suction cylinder nozzle response was at 98 Hz (sixth order at 978 RPM), and the discharge cylinder nozzle response was at 80 Hz (sixth order at 798 RPM). Figure 2-26 shows test data that was taken in the RCTF to confirm the predicted (simulated) placement and amplitude of the suction and discharge cylinder nozzle responses.

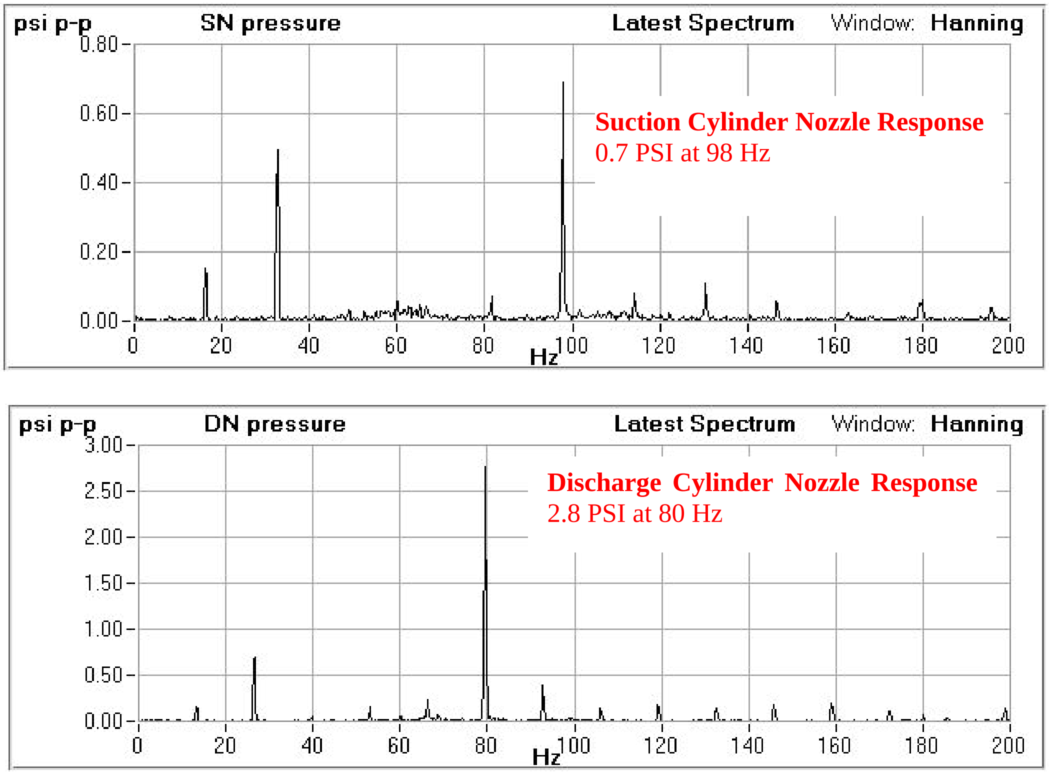

Figure 2-26. Original RCTF Test Data Used for Correlation with Simulation Data

The predicted (simulation) amplitude and frequency of the discharge cylinder nozzle resonance were 5.1 PSI and 79 Hz, respectively. Test data showed that the resonance was 2.8 PSI at 80 Hz. The predicted amplitude and frequency of the suction cylinder nozzle resonance were 0.6 PSI and 96 Hz. The data also showed test resonance was 0.7 PSI and 98 Hz. Both sets of data are listed in Table 2-4. These results show good frequency and amplitude correlation.

Table 2-4. Correlation Data – Simulation Data and Experimental Data

	Simulation Data		Experimental Data	
	Frequency (Hz)	Amplitude pk-pk (psi)	Frequency (Hz)	Amplitude pk-pk (psi)
Suction Cylinder Nozzle Response	96	0.6	98	0.7
Discharge Cylinder Nozzle Response	79	5.1	80	2.8

After showing the ability to accurately simulate the acoustics of the existing system, the following design features were implemented into the modified RCTF filter bottle design:

- Suction and discharge cylinder nozzle responses were placed on the fourth order (approximately 720 RPM) to raise the pulsation levels and place the response near a frequency that is typical for high-speed compressors.

- A volume-choke-volume design was implemented such that the system is a more typical design. The Helmholtz resonance was placed on the first order at approximately 540 RPM
- Placement of the cylinder nozzle resonance and Helmholtz resonance are at different running speeds so each resonance could be evaluated separately.
- The design took into consideration the future installation of a Multi-Frequency Tunable Side Branch Absorber (SBA).
- Cylinder nozzles were separated into sections to allow the installation of an ILN and a TCN.

The above design features were implemented into the modified system to allow the evaluation (proof-of-concept) of the ARCT pulsation control concepts. Figure 2-27 illustrates the constructed RCTF modifications. Notice that there are now two bottles with a choke tube (relatively small piping) between them. That two-bottle system creates a volume-choke-volume configuration for the suction and discharge piping systems. The larger piping that is vertically between the two bottles on both suction and discharge is a non-flow piece of pipe that is for support purposes only. Lengthened suction and discharge cylinder nozzles placed the resonance associated with these nozzle lengths at the desired frequencies.

Figure 2-27. Modified RCTF – Two-Bottle Acoustic Filter Design Installed

Characterization test data was taken to verify the designed acoustic characteristics. Simulation data showed the discharge cylinder nozzle resonance would be 9.1 PSI at 49 Hz as depicted in Figure 2-28. Test data showed similar results with an 11.7 PSI response at 49 Hz (see Figure 2-29). Helmholtz resonance predictions showed 5.7 PSI at 9 Hz (see Figure 2-30). Test data showed 5.3 PSI at 10 Hz when the Helmholtz resonance was observed (see Figure 2-31).

These results and other acoustic characteristic predictions and test data are listed in Table 2-5. The data shows good frequency correlation and reasonable amplitude correlation.

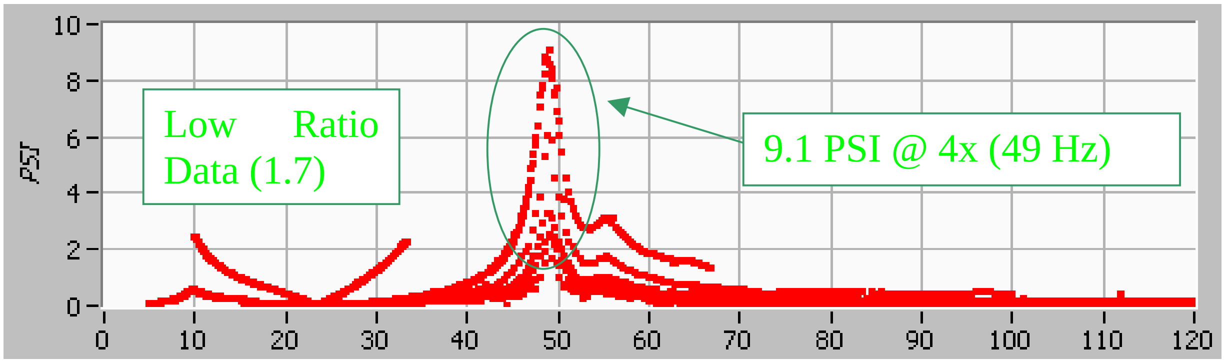

Figure 2-28. Simulation Data of the Discharge Cylinder Nozzle Response

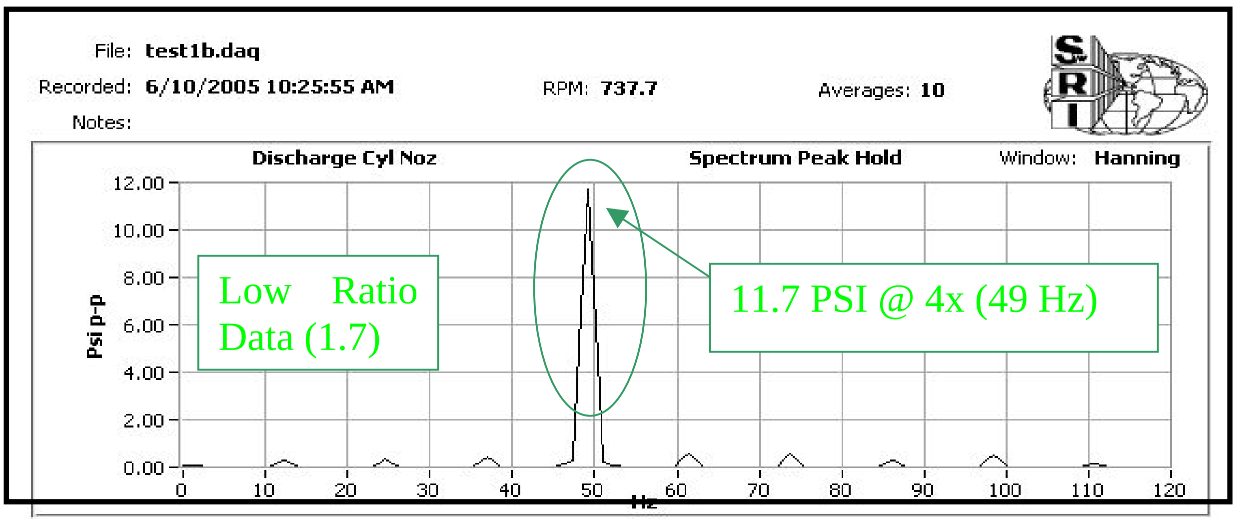

Figure 2-29. Test Data of the Discharge Cylinder Nozzle Response

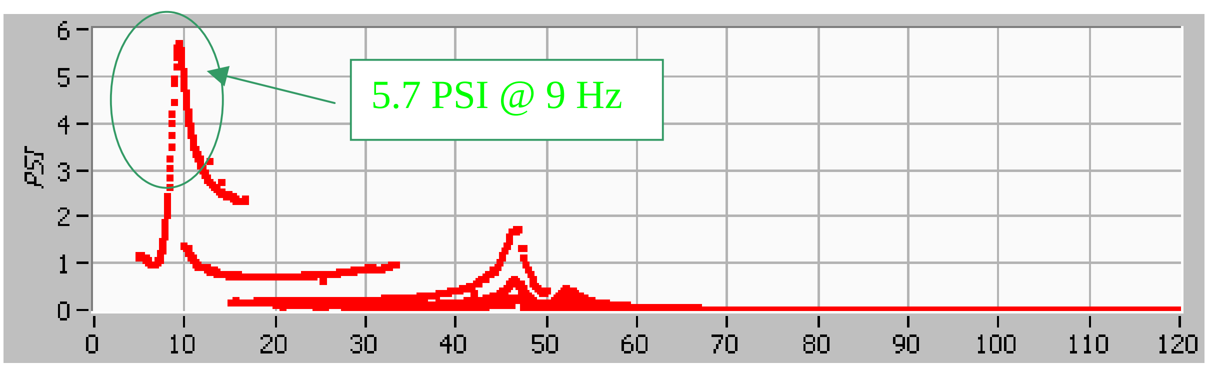

Figure 2-30. Simulation Data (Single Acting) of the Discharge Helmholtz Response

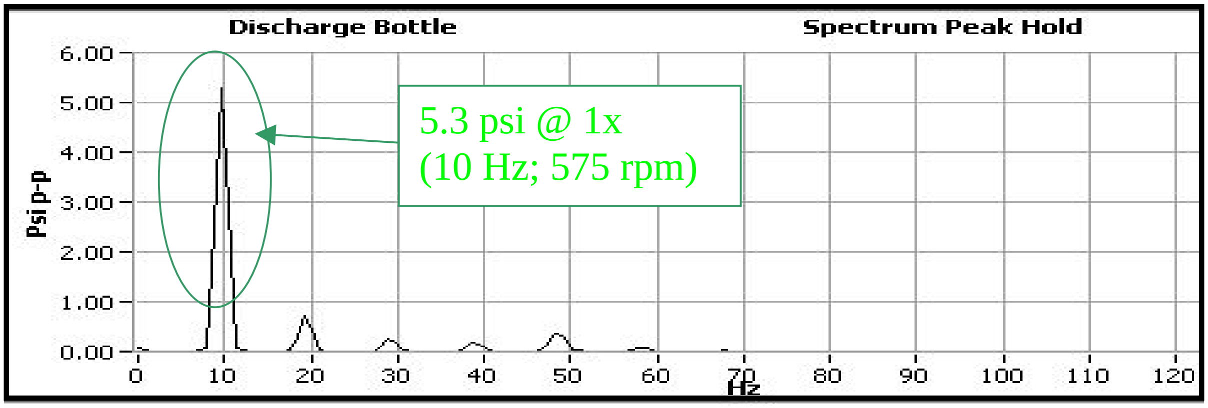

Figure 2-31. Test Data (Single Acting) of the Discharge Helmholtz Response

Table 2-5. Correlation Data – Simulation Data and Experimental Data for the Modified RCTF

	Simulation Data				Experimental Data			
	Frequency (Hz)		Amplitude pk-pk (psi)		Frequency (Hz)		Amplitude pk-pk (psi)	
	Low Ratio	High Ratio	Low Ratio	High Ratio	Low Ratio	High Ratio	Low Ratio	High Ratio
Suction Cylinder Nozzle Response	46	47	6.2	4.5	46	47	6	3.4
Discharge Cylinder Nozzle Response	49	49	9.1	20.5	49	50	11.7	12.2
Suction Helmholtz Response	9		2.3		9		3.0	
Discharge Helmholtz Response	9		5.7		10		5.3	

2.4.2 TAPER CYLINDER NOZZLE PROOF-OF-CONCEPT

For high-speed compressors, there is a need to lengthen the cylinder nozzles (to reduce mechanical coupling), raise the resonance frequency, and reduce pulsations at the cylinder nozzle resonance frequency. The tapered cylinder nozzle (TCN) concept shows the potential to meet these needs. Benefits that are associated with this concept include:

- Significant increase in the cylinder nozzle resonant frequency;
- Lower amplitudes of excitation (pulsations are shifted to higher/weaker harmonics);
- And, it could possibly be used as a means to lengthen the cylinder nozzle such that mechanical coupling between the cylinder and the rest of the piping is reduced.

A breadboard design (used for proof-of-concept) included an 8” x 4” reducing elbow followed by a 12” x 8” reducer as depicted in Figure 2-32. Simulation data of the TCN predicted that it would significantly shift the cylinder nozzle resonant frequency and reduce the cylinder nozzle pulsations by 21% to 64% depending on the operating pressure ratio (see Figure 2-33 and Figure 2-34). After modeling and designing the TCN, it was constructed and installed as shown in Figure 2-35. Test data was taken and compared with the simulation data as listed in Table 2-6. For the low ratio operating conditions, simulation data predicted a discharge cylinder nozzle resonance of 7.2 PSI at 72 Hz, and test data showed a 7.7 PSI response at 75 Hz. The table also lists the cylinder nozzle resonance pulsations and frequencies <u>without</u> the TCN installed for direct comparison with that same data <u>with</u> the TCN installed. Results showed useful correlation between the test data and simulation data when the TCN is installed.

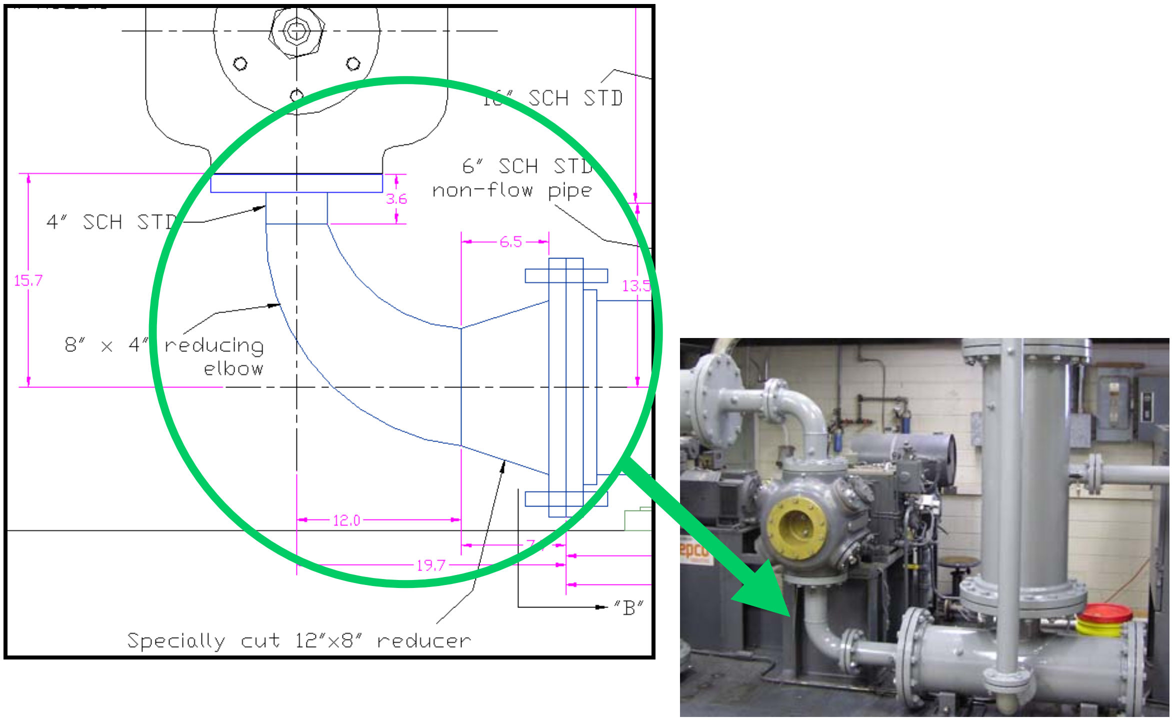

Figure 2-32. TCN Concept for the RCTF Discharge Cylinder Nozzle

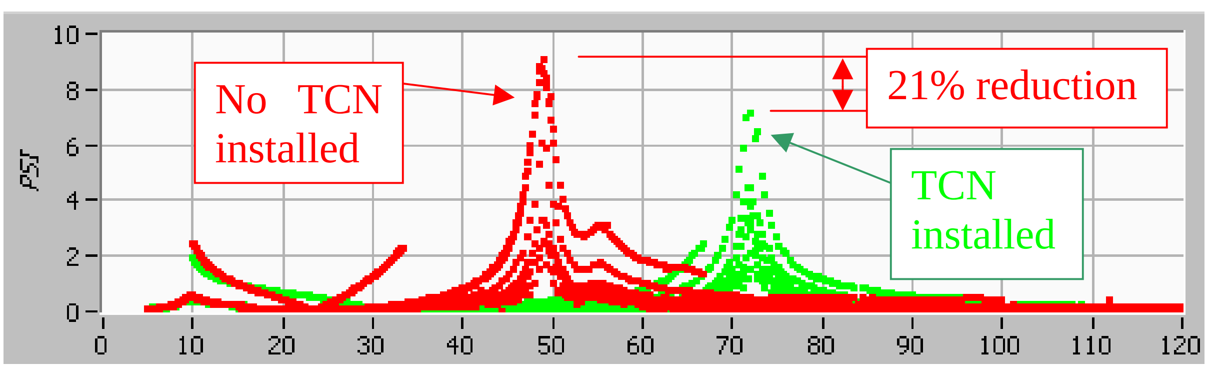

Figure 2-33. Simulation Data of the RCTF at Low Ratio (with and without TCN installed)

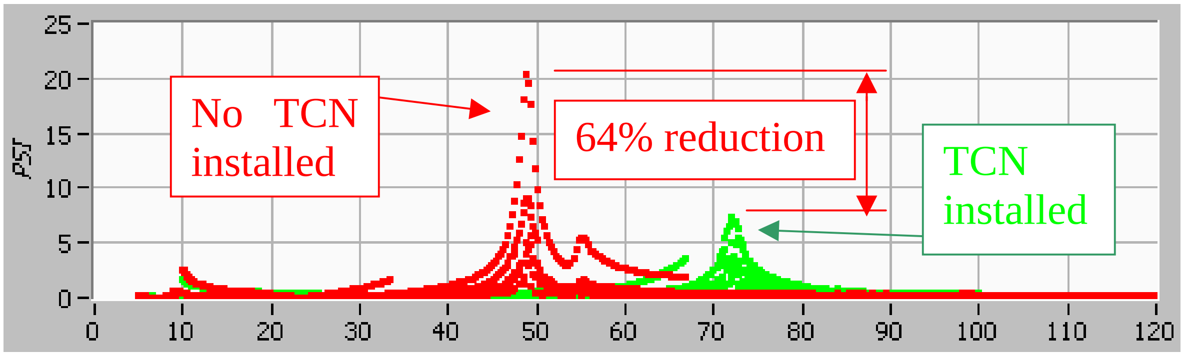

Figure 2-34. Simulation Data of the RCTF at High Ratio (with and without TCN installed)

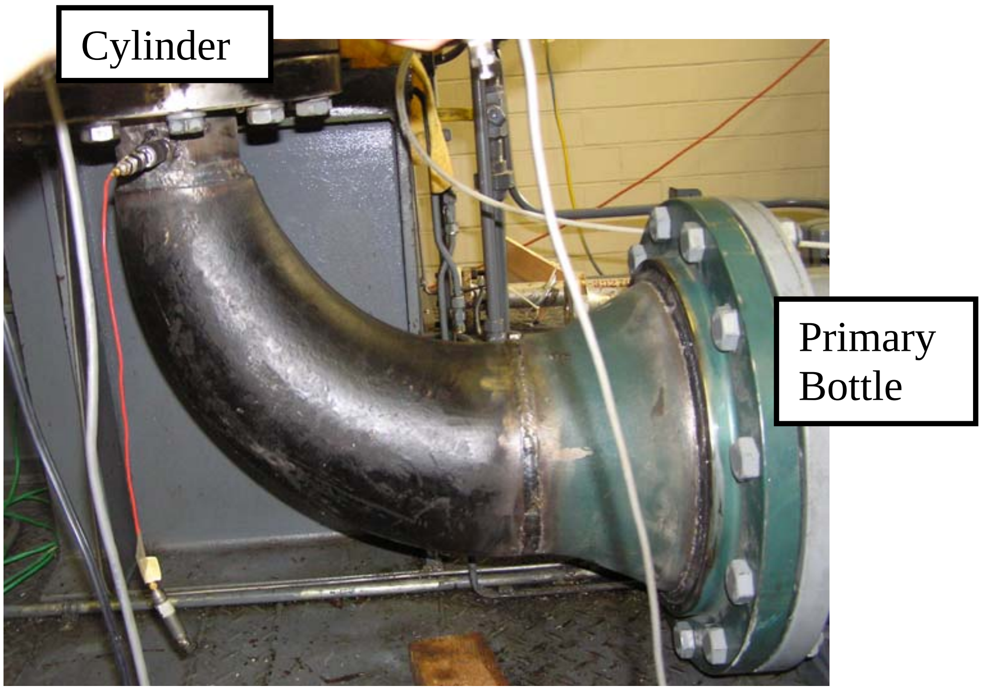

Figure 2-35. TCN Installed in Place of the Original RCTF Discharge Cylinder Nozzle

Table 2-6. Correlation Data With and Without the TCN Installed in the RCTF

	No Tapered Disch. Cyl. Noz.				Tapered Nozzle			
	Frequency (Hz)		Amplitude pk-pk (psi)		Frequency (Hz)		Amplitude pk-pk (psi)	
	Low Ratio	High Ratio	Low Ratio	High Ratio	Low Ratio	High Ratio	Low Ratio	High Ratio
Simulation Data	49	49	9.1	20.5	72	72	7.2	7.3
Experimental Data	49	50	11.7	12.2	75	74	7.7	7.9

High Ratio – 2.7
Low Ratio – 1.7

When the TCN was installed, the cylinder nozzle resonance frequency shifted up significantly as predicted with the acoustic simulation tool. The observed frequency shift of 25 Hz (50% increase) is noted in Figure 2-36. The “Baseline System” noted in the figure refers to the system without the TCN installed. A significant frequency shift, such as the observed shift, placed the cylinder nozzle resonance on a higher weaker compressor harmonic.

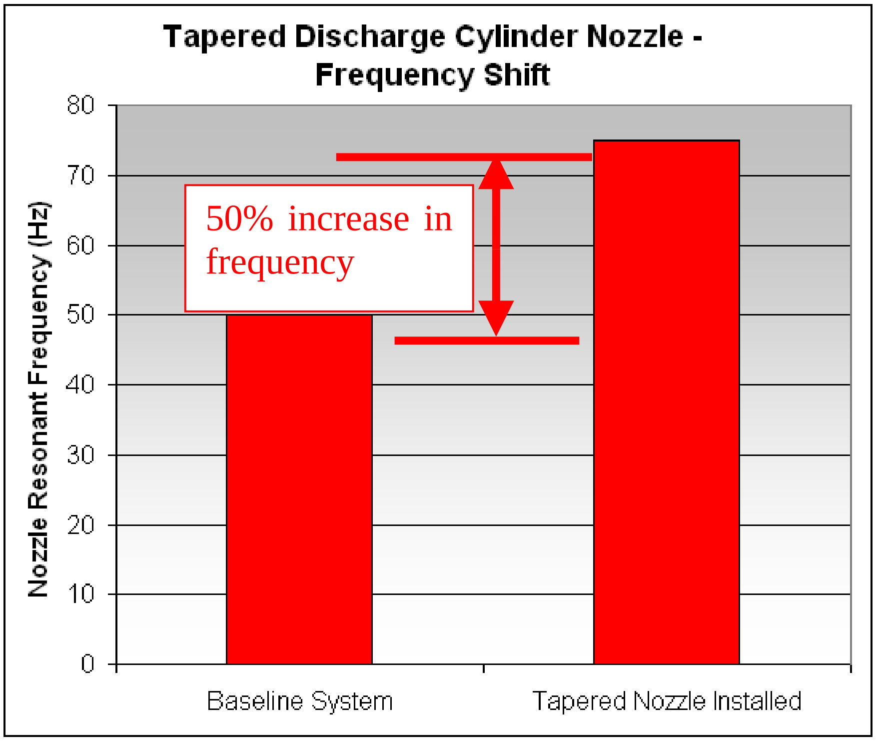

Figure 2-36. Large Frequency Shift Observed When the TCN was Installed

To evaluate the relative pressure drop of the systems (with and without the TCN installed) with and without orifices installed, pressure differential data was taken across the cylinder nozzle, primary bottle, choke, and secondary bottle as noted with the “Delta P” box and arrows in Figure 2-37. The box labeled “Orifices Installed” notes the locations in the system where the orifices were installed. Most of the various orifices were installed at the location noted by the upper arrow (the cylinder flange), but one orifice was installed at the cylinder nozzle to bottle connecting flanges when the TCN was installed. Pulsation data was also taken during these various setups. Resulting data provides additional insight into the advantages of the TCN as compared to a typical cylinder nozzle.

Pulsation data was taken for the systems with and without the orifices while the compressor was operating at a low ratio and at a high ratio. Pulsations for the system without the TCN were similar at low and high ratio. With the installation of a relatively small orifice, the pulsations during the high ratio (low flow) operating conditions were three times the pulsations during the low ratio (high flow) conditions (see Figure 2-38; note that the pulsations are in percent of line pressure). This data illustrates that the acoustic effectiveness of an orifice depends on the flow velocity through the orifice. The following notable points are also captured in this test data:

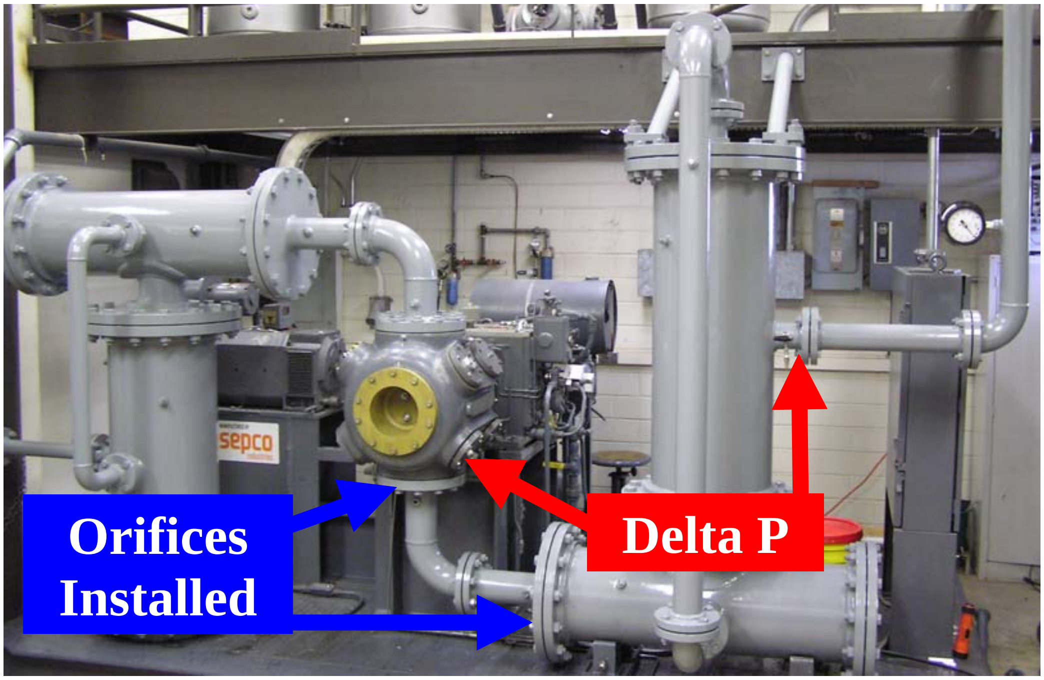

Figure 2-37. Orifice Installation Locations and Pressure Drop (Delta P) Probe Locations

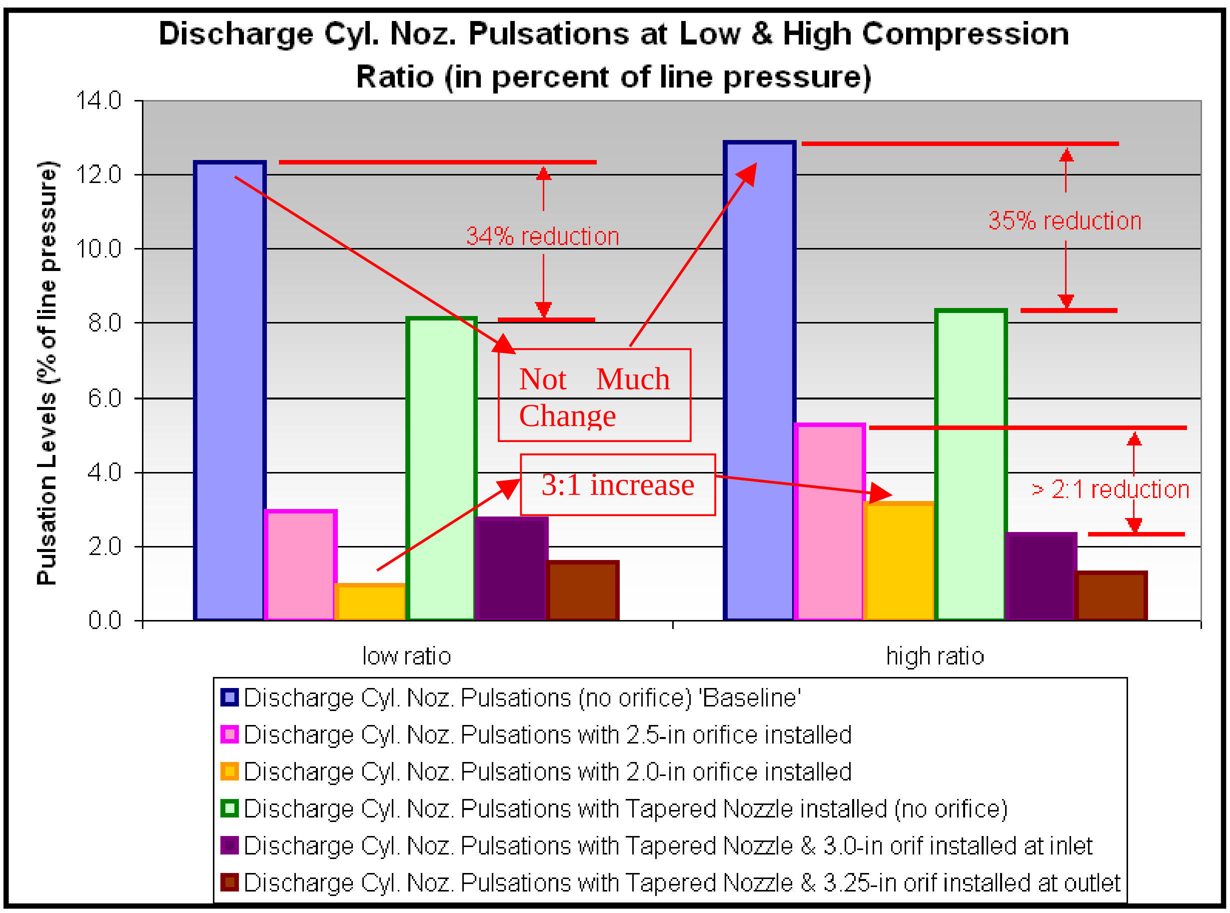

Figure 2-38. Discharge Cylinder Nozzle Pulsations at Low and High Compression Ratios

- There was a 34% to 35% reduction in cylinder nozzle resonant pulsation amplitudes when the TCN was installed (no orifices installed).
- There was a 2:1 reduction in cylinder nozzle resonant pulsations when a relatively large bore orifice (low pressure drop) was installed in conjunction with the TCN as compared to that of a small bore orifice (high pressure drop) installed in conjunction with a typical cylinder nozzle.

Pressure drop data was taken for the systems with and without the orifices while the compressor was operating on and off the discharge cylinder nozzle resonance. Pressure drop data for the system without the TCN installed were very different on and off resonance (3:1 difference) as depicted in Figure 2-39. This difference is due to the dynamic effects that pulsations can have on pressure drop. The following notable points are also captured in this test data:

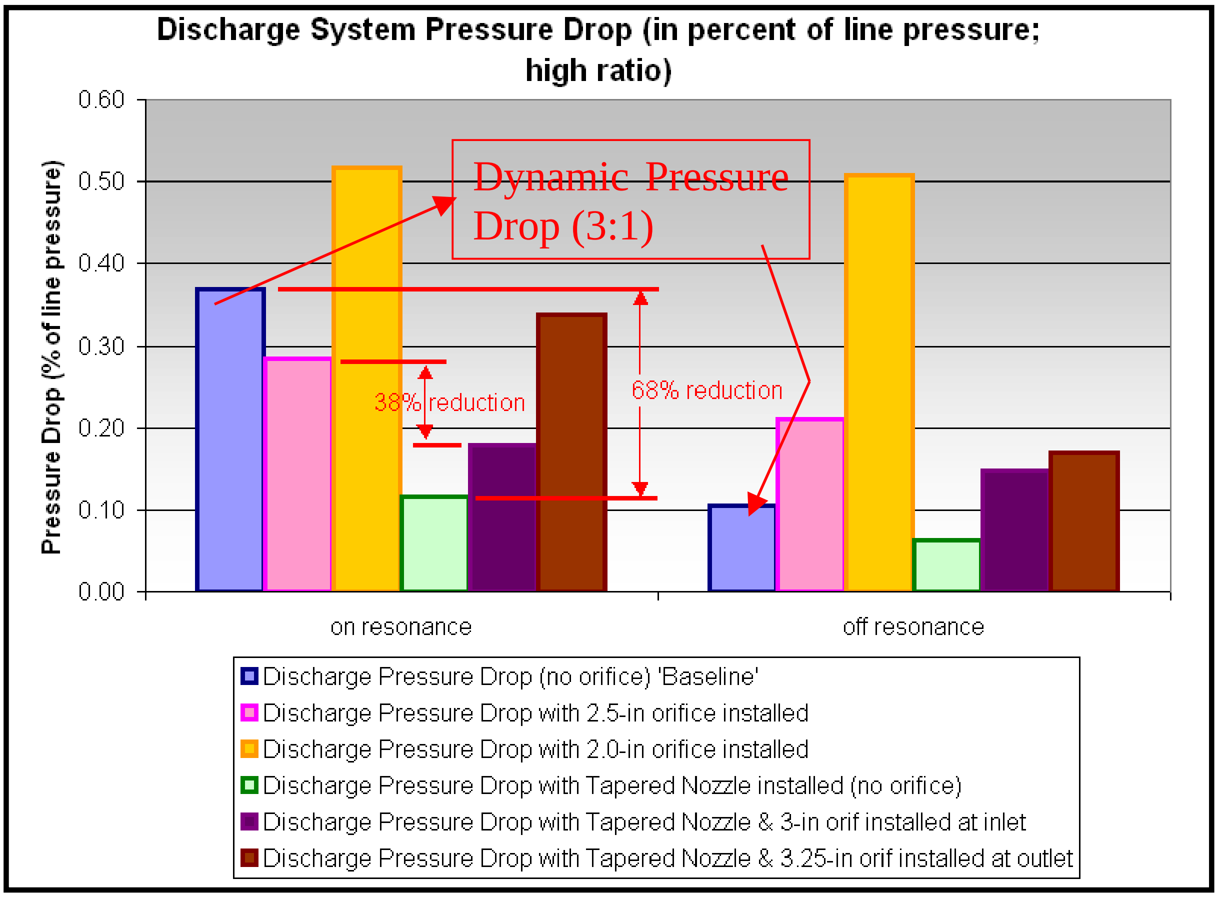

Figure 2-39. Discharge System Pressure Drop at High Compression Ratios

- There was a 68% reduction in discharge system pressure drop when the TCN was installed and the compressor operated on the cylinder nozzle resonance (no orifices installed).
- There was a 38% reduction in pressure drop when a relatively large bore orifice (<u>low pressure drop</u>) was installed in conjunction with the TCN as compared to that

of a small bore orifice (high pressure drop) installed in conjunction with a typical cylinder nozzle.

The high ratio data that was illustrated in the previous two figures is summarized in Figure 2-40. The x-axis is the discharge cylinder nozzle pulsations in percent of line pressure, and the y-axis is the system pressure drop in percent of line pressure. <u>It is desirable to be near the origin of the graph such that the system has low pulsations and low-pressure drop</u>. The red plot with diamonds is the system without a TCN installed, and the orange plot with circles is the system with a TCN installed. Starting with the "no orifice" test point for the red plot and following the plot to the 0.625β orifice, the pulsations decrease significantly, but the pressure drop decreased slightly. Continuing along the red plot to the 0.5 β orifice, the pulsations decreased even more; however, the pressure drop increased. When comparing the red plot with the orange plot, it is important to note that the 0.75 β orifice data point on the orange plot is significantly closer to the origin than the 0.5 β orifice data point on the red plot. Therefore, the pressure drop and pulsations were lower for a system with TCN and a 0.75 β orifice installed as compared to a system with a typical cylinder nozzle installed and a 0.625 β orifice installed.

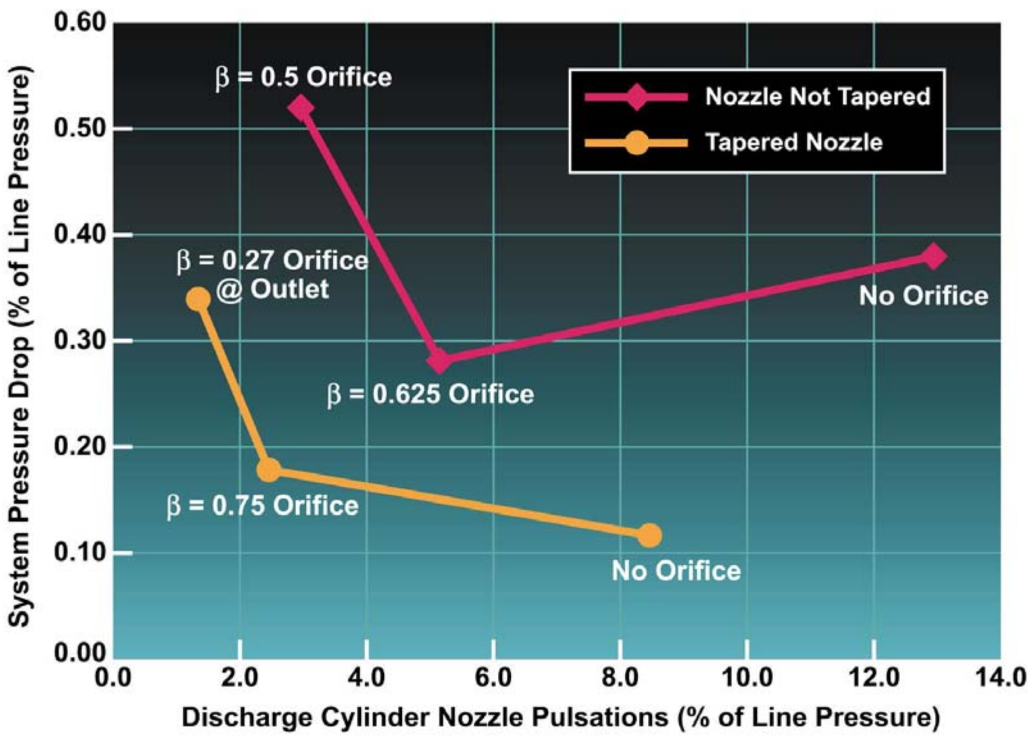

Figure 2-40. Discharge System Cylinder Nozzle Pulsations and Pressure Drop

The following major points summarize the results that were observed when the TCN was installed:

- The cylinder nozzle resonant frequency shifted above the fourth order when the TCN was installed (24-26 Hz frequency increase, 50% increase).

- There was a 34% to 35% reduction in maximum cylinder nozzle pulsation amplitudes when the TCN was installed (no orifice).
- A 3.1:1 pressure drop reduction was observed when the TCN was installed (no orifice).
- The TCN results showed *less pulsations* (below 3% of line pressure) with *less pressure drop* than would be required in a traditional system that had no TCN installed. (The API pulsations guideline is 7% of line pressure for these operating conditions.)

The conclusion is that the TCN is a viable concept that warrants further development. This concept has demonstrated the potential to resolve the critical cylinder nozzle problem experienced in modern high-speed compression. This very well may be the needed enabling technology for next generation compression.

2.4.3 INFINITE LENGTH NOZZLE PROOF-OF-CONCEPT

For high-speed compressors, there is a need to lengthen the cylinder nozzles (to reduce mechanical coupling), raise the nozzle resonant frequency, and reduce pulsations at the cylinder nozzle resonant frequency. The Infinite Length Nozzle (ILN) concept should be able to meet those needs. The theory behind the ILN concept is that multiple reflections (at each perforation) reduce the amplitude of the cylinder nozzle resonance. Benefits that are associated with this concept include:

- Using it as an insert into existing systems;
- Lower amplitudes of excitation;
- And, it could possibly be used to lengthen the cylinder nozzle such that mechanical coupling between the cylinder and the rest of the piping is reduced.

Different perforation distributions along the projection portion of the ILN were simulated to parametrically develop a well-designed ILN breadboard to be evaluated in the RCTF. Figure 2-41 presents some sample simulation data for two different perforation distributions. The plots in the figure depict the pulsations for a typical system with and without an ILN installed. The ILN simulated in the upper plot had six perforations evenly spaced along the ILN projection, and the perforations increased from 5% to 29% of the ILN bore area (the ILN flow area). The ILN simulated in the lower plot had four perforations evenly spaced along the ILN projection, and the perforations increased from 0.6% to 7% of the ILN bore area. The simulation data in the upper plot depicts a 12% pulsation reduction at resonance and the lower plot depicts a 58% reduction. Notice that an undesirable effect is also illustrated by the simulation data. The resonant frequency decreases when the ILN is installed in the model simulation. These simulation results, along with other perforation distributions, are summarized in Figure 2-42. The pink plot with the square data points presents the perforation distribution that was used for the ILN that was tested in the RCTF. Notice that the predicted reduction of the resonant pulsations is 62% to 73%. This range of pulsation reduction is based on high and low ratio operating conditions.

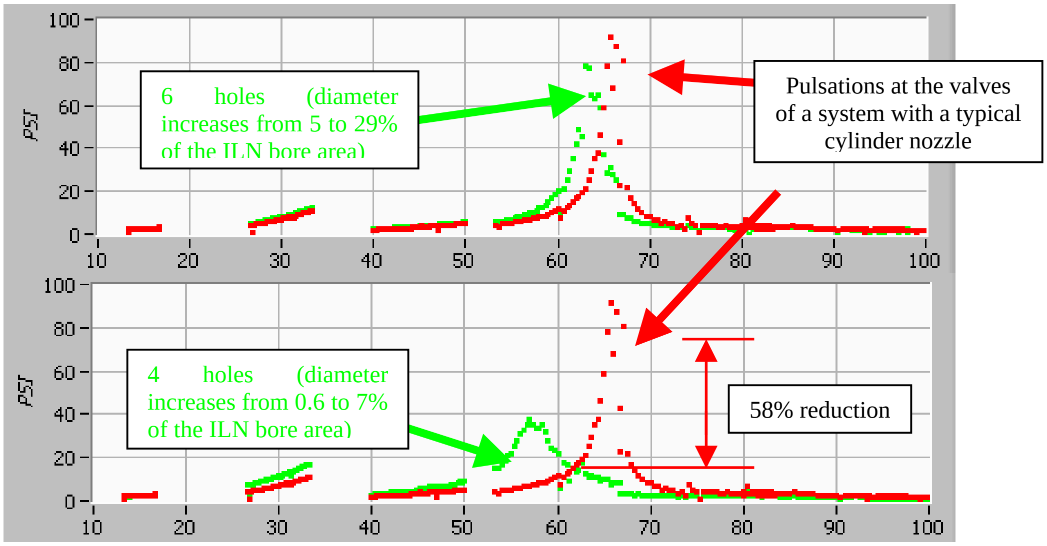

Figure 2-41. Simulation Data With and Without the ILN Installed in the System

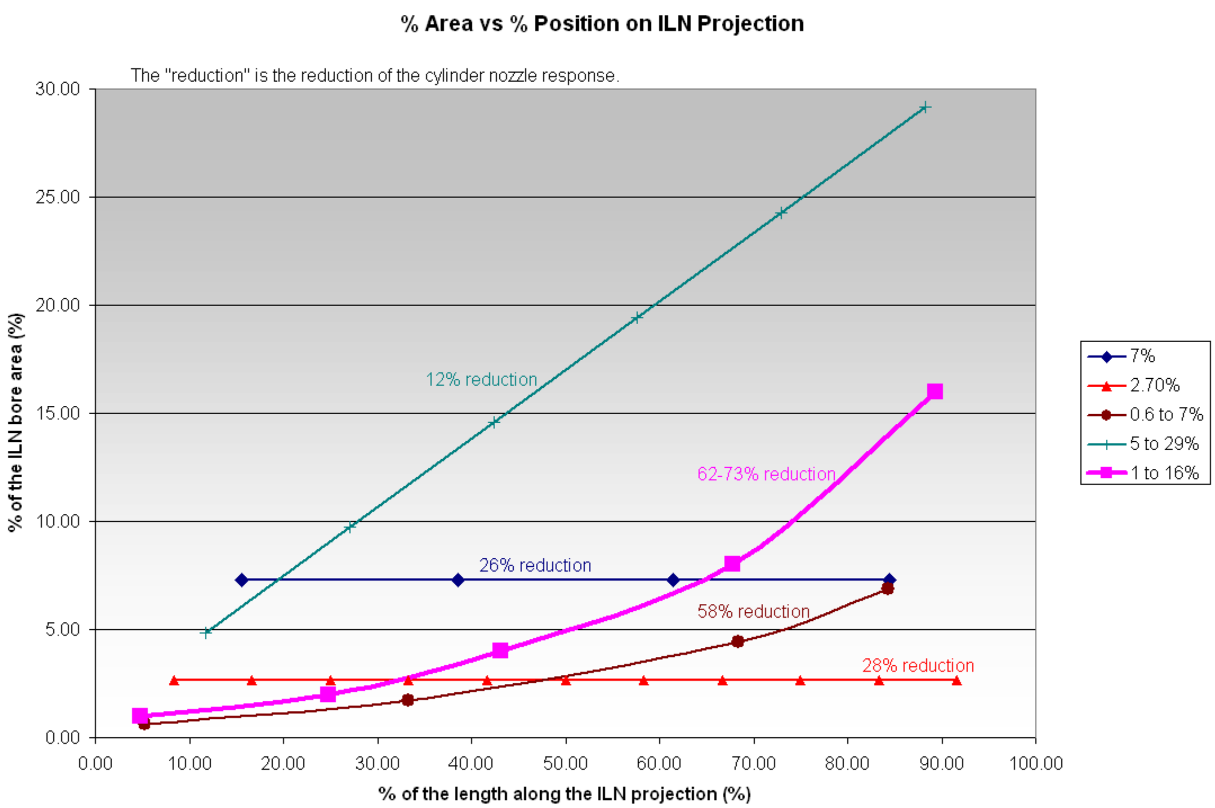

Figure 2-42. Simulation Results Based on Various ILN Configurations

The ILN breadboard design (used for proof-of-concept) installed in the RCTF had a perforation distribution along the ILN projection that ranged from 1% to 16% of the ILN bore area. The ILN conceptual model is depicted in Figure 2-43. Figure 2-44 depicts what the ILN looked like once it is installed in the system. The length of the ILN projection was determined based on the simulations of ILN installations in a typical size cylinder and bottle. After modeling and designing the ILN, it was constructed and installed as shown in Figure 2-45.

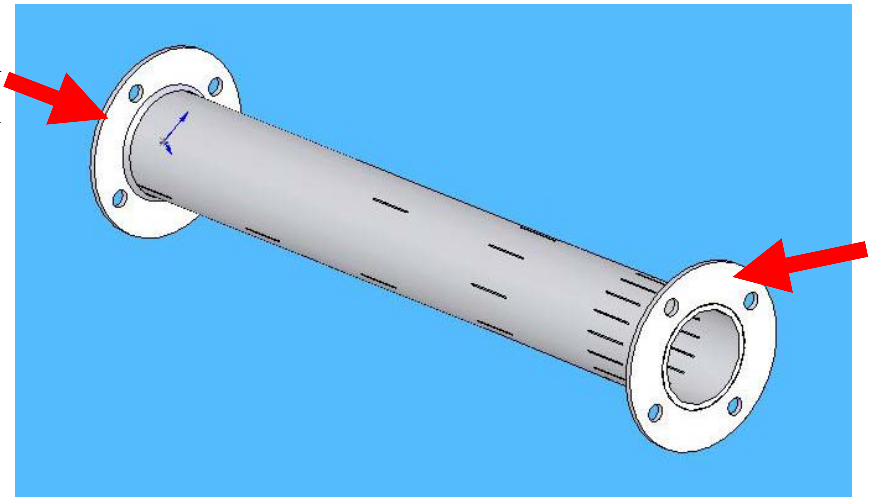

Figure 2-43. ILN Concept for Installation in the RCTF

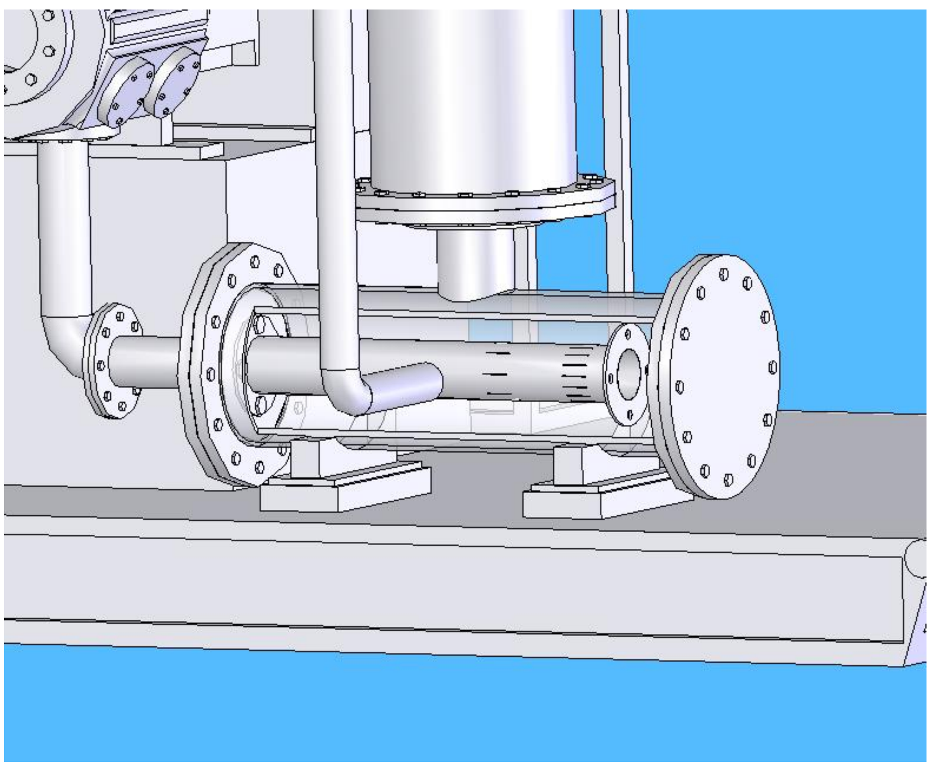

Figure 2-44. Conceptual Installation of the ILN in the RCTF Bottle System

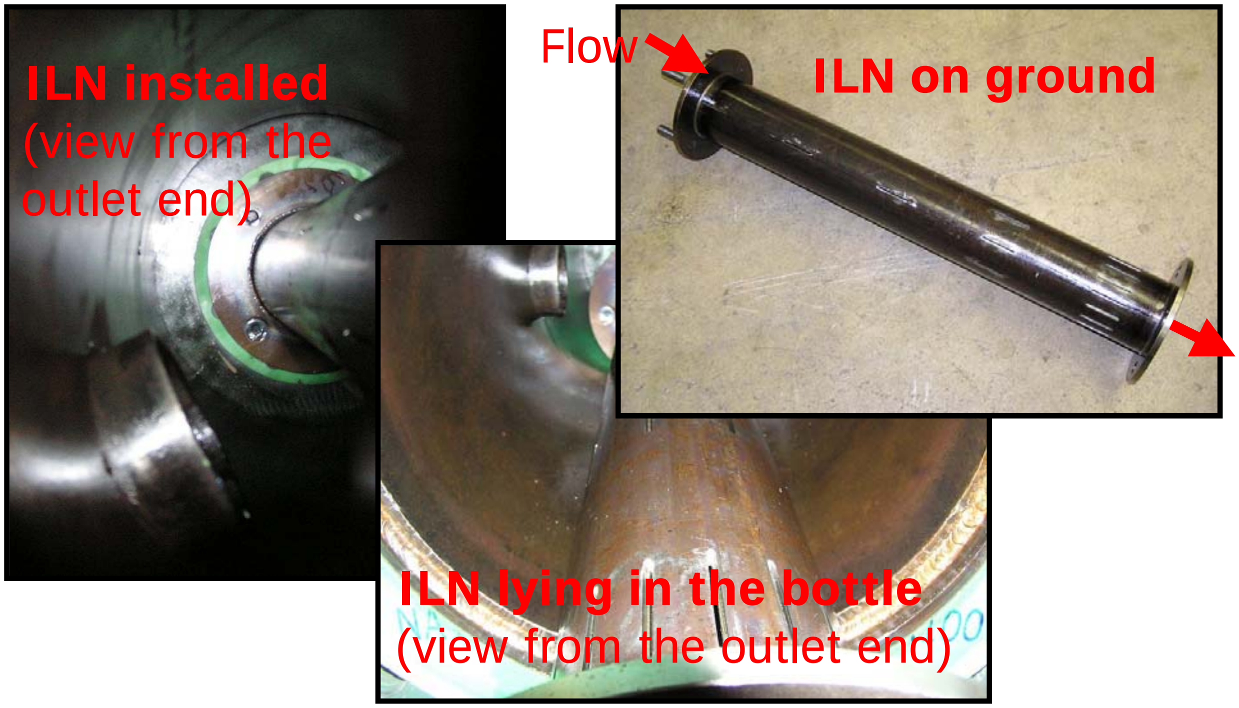

Figure 2-45. Different Views of the ILN That was Installed in the RCTF for Testing

Test data was taken with the ILN installed in the system. Data including a fourth order speed sweep over the cylinder nozzle resonance was recorded after the ILN was installed. That data is depicted in Figure 2-46. The shape of the resonance (wide response) suggests high damping. Also, note that the amplitude of the resonance is much lower than that of the cylinder nozzle resonance before the ILN was installed. This data and simulation data are summarized in Table 2-7. For the low ratio operating conditions with the ILN installed, simulation data predicted a discharge cylinder nozzle resonance of 3.5 PSI at 42 Hz, and test data showed a 3.5 PSI resonance at 39 Hz. The table also lists the cylinder nozzle pulsations and frequencies <u>without</u> the ILN installed for direct comparison with that same data <u>with</u> the ILN installed. Results showed good correlation between the test data and the simulation data when the ILN is installed.

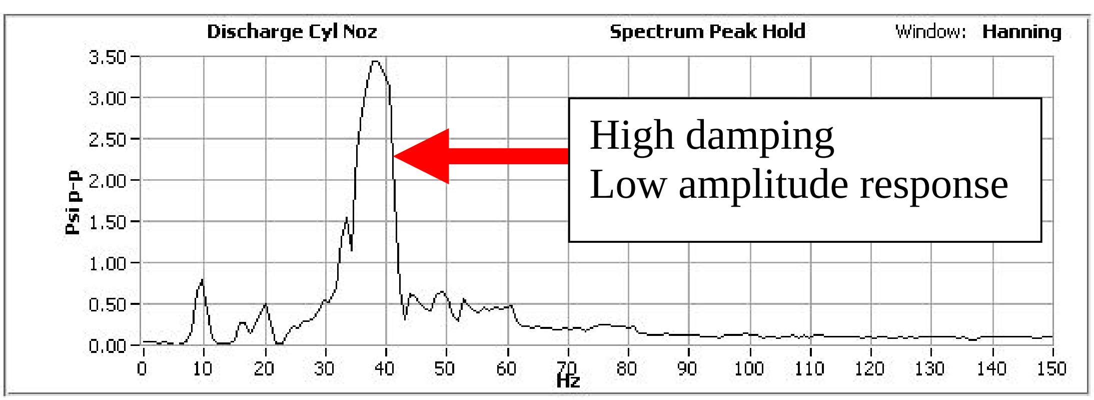

Figure 2-46. Test Data – Fourth Order Speed Sweep Over the Cylinder Nozzle Resonance

Table 2-7. Correlation Data With and Without the ILN Installed

Data Correlation	No ILN				ILN installed			
	Frequency (Hz)		Amplitude pk-pk (psi)		Frequency (Hz)		Amplitude pk-pk (psi)	
	Low Ratio	High Ratio	Low Ratio	High Ratio	Low Ratio	High Ratio	Low Rati	High Ratio
Simulation Data	49	49	9.1	20.5	42	44	3.5	5.5
Experimental Data	49	50	11.7	12.2	39	39	3.5	4.1

High Ratio – 2.7
Low Ratio – 1.7

The following major points summarize the results that were observed when the ILN was installed:

- There was a 70% reduction in the maximum cylinder nozzle pulsation amplitudes (no orifice installed).
- There was a reduction of approximately 3.4:1 in pressure drop (fourth order on resonance).
- A weakness that was observed was the 10 Hz frequency decrease of the cylinder nozzle resonant frequency.
- A second weakness observed was an increase in the second order pulsations due to the distribution of the cylinder nozzle resonance over a large frequency range that results from the installation of the ILN.

More testing of the ILN should provide information that will help determine the correlation between performance and geometry.

2.4.4 RECOMMENDATION FOR FUTURE WORK

Both the tapered nozzle and infinite nozzle concepts have been proven by breadboard validation in the laboratory. While the tapered nozzle appears extremely promising, only a single geometry was tested at the two pressure ratios. The combination of a tapered nozzle and various orifice sizes allows for trading off pulsation reduction with increases in pressure drop. These results warrant taking the technology to the next level. A range of prototype geometries (inlet-to-outlet ratio and length) must be evaluated over a range of orifice beta-ratios and a wider more realistic operating range of pressure, pressure ratio, realistic gas composition, and flow rates. By varying length, we can investigate the potential to decouple pulsation control and mechanical vibration due to cylinder stretch for short nozzles.

The infinite length nozzle also appears promising. For the single geometry tested under limited operating conditions, it demonstrated a pulsation amplitude reduction and reduced pressure drop. However, for this geometry, the cylinder nozzle resonant frequency decreased instead of increasing. This effect must be further investigated. However, this technology also warrants being taken to the next level. A series of prototype geometry must be investigated in a range of more realistic operating conditions. It is premature to select which of these new nozzle concepts will be successful or if both are valuable for different applications. There is significant synergy by testing these concepts together and improving the predictive capability for each. To be useful tools to the pulsation control designer, accurate prediction tools are required.

In addition to these two nozzle concepts, the tunable SBA has been proven analytically. The next step is to fabricate and test a breadboard in the laboratory. It is important that concepts to address the 1x problem in the lateral piping be moved forward.

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3. CAPACITY CONTROL RESULTS

3.1 CAPACITY CONTROL TECHNOLOGY ASSESSMENT

The reciprocating compressor market has changed over the last ten years to meet the needs of a changing industry. One of the new needs in the gas compression industry is better load and capacity control methods that allow compressors to meet varied operating demands, flexibility requirements, and efficiency goals of the new more competitive pipeline industry. These varied demands cause rapid flow changes as well as changes in suction and discharge pressures requiring a compressor to operate over a wide range of flows and pressure ratios. The need for better load and capacity control methods in reciprocating compressors is driven by the following characteristics of the current gas industry: (1) a deregulated industry with fluctuating supply and demand patterns; (2) competition in a growing market causing a push for reduced capital costs for new or upgraded installations by using a few large reciprocating compressors to replace several small integral engine/compressors; and (3) pipeline companies striving to reduce operating costs to improve profitability by making gas compression more efficient. This section will describe the current load and capacity control methods used in reciprocating compressors and highlight the associated advantages and disadvantages of each method.

The deregulated gas utility market has perhaps been one of the biggest causes of the need to identify better load and capacity control methods. Deregulation and the resulting increase in required flexibility and the competition among pipeline companies motivated a Southwest Research Institute survey of natural gas pipeline companies. The survey examined the common situations requiring flexible compression and the present options available [McKee and Smalley, 2001][1]. Some of the situations that require compressors to operate over a broader range are low-flow conditions (30% to 70% of pipeline capacity), new pipelines and fewer compressors, increased pressure ratios at existing stations, highly variable power station supply requirements, and storage fields with varied injection and withdrawal requirements.

Improved load and capacity control methods developed in the ARCT program will provide solutions to meet the requirements described by pipeline companies in the flexible compression survey. One reciprocating compressor manufacturer, GE Nuovo Pignone, estimates that process compressors are required to operate at 60% to 100% of the nominal flow rating, but reductions further than 60% of nominal flow require advanced capacity control methods [Chellini, 2003][2]. In the SwRI compression survey, pipeline companies identified many disadvantages to the existing capacity control methods for reciprocating compressors. The main disadvantages to the current methods are the high cost of additional units, electric drives, or capacity control hardware, the ineffectiveness of several methods at reducing compressor capacity significantly, the resulting losses in efficiency, and the limited turndown of engine drivers.

One aspect of what is referred to as capacity that needs to be understood is that the primary purpose for small changes in capacity is to control the load (torque) within compressors. Only when large changes in pipeline throughput and operating conditions (pressures) are required are capacity control methods, such as deactivation, unloaders, or starting and stopping entire units, used to control flow rate. Normally, as operating conditions change and pressure ratio increases, capacity control methods, such as pockets or infinite step unloaders, are used to control (limit) the load in a reciprocating compressor so that the unit operates within its load limits and its driver operates smoothly, efficiently, and safely. It is important to keep in mind the primary reason that reciprocating compressors have capacity control devices is for load (torque, and HP) control. Capacity control for capacity's sake becomes more important to a station's operation when the station needs to make large changes in the compressor and pipeline throughput. This section addresses the techniques that achieve either type of control.

The existing control methods for varying a single unit's load and capacity fall into one of two categories—methods that reduce or add to the compression volume or methods that vary the amount of gas compressed by changing the valve motion or the gas characteristics on the suction side (gas pressure and gas volume) of the compressor. The various current technology methods are discussed in the following pages. Changing the compressor speed changes the rate at which compression volume is swept and, hence, the compressor's load and capacity. In addition to these methods, another technique for station capacity control (used more commonly with smaller integral machines) is to vary the number of compressors in operation. This method, based on multiple small compressors, has a large capital cost implication and is more challenging to automate due to the difficulty and time required for start-up. The six categories of capacity and load control methods for a single compressor and the related specific technologies available will be discussed individually.

3.1.1 CLEARANCE POCKETS

One existing method of changing the compressor load and capacity is to vary the amount of clearance at the end of the compressor cylinder. Multiple volume pockets can be installed in order to add volume to the cylinder end. Alternatively, valve pockets may be used to change the clearance volume in each cylinder end. Both of these approaches change the total clearance volume in the compression cylinder. The early versions of this method have been used in gas compressors for over twenty years. The original design used a manually operated threaded actuator and hand wheel. The volume in the head end or pocket can be adjusted by manually turning the wheel causing the screw to adjust the location of the head or pocket end. The head end version of this design is shown in Figure 3-1.

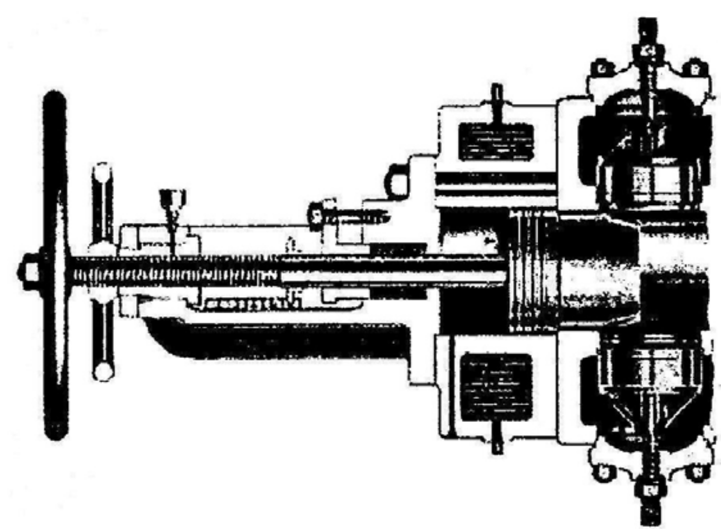

Figure 3-1. Manual Pocket Adjustment with Hand Wheel and Screw Thread [Woollatt, 2002][3]

The manually operated head or clearance pocket requires on-site labor to change the clearance volume. An obvious improvement to the design is to be able to automatically adjust the clearance during compressor operation. The existing automated designs for variable clearance pockets use either hydraulic or pneumatic actuation to automatically vary the volume in the pocket. Various designs have been patented in this area. One design is described in a patent issued to Cooper Industries and is now being manufactured by ACI Services. Described as the Automatic Variable Volume Clearance Pocket (AVVCP) system, this design moves an unloader piston to adjust the clearance volume by hydraulic action. The assembly is shown in Figure 3-2. The system works by pressurizing the oil in cavities B and C using the accumulators shown on the right. A linear displacement transducer is used to determine the location of the hydraulic and unloader piston for the system controls. This system is used in many existing compressors as a replacement for manual volume control.

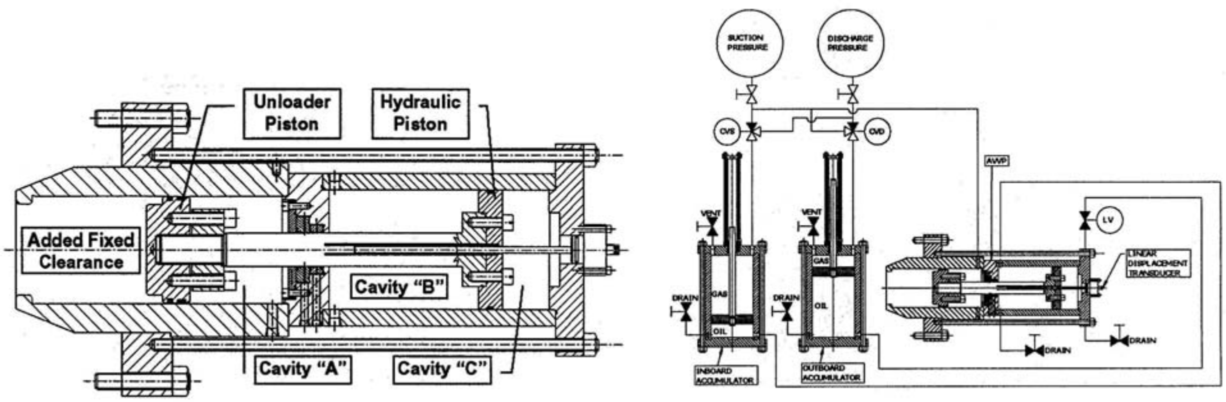

Figure 3-2. Automatic Variable Volume Clearance Pocket (AVVCP) System Patented by Cooper Industries and Licensed to ACI Services, Inc. [Phillippi, 2002][4]

Another version of a hydraulically actuated pocket is shown in Figure 3-3. This design, termed the Hydraulic Variable Volume Clearance Pocket (HVVCP) is used by Dresser-Rand. The HVVCP can be configured with manual needle valves, automatic solenoid, or pneumatic valves. Infinite step adjustments are possible, but <u>only</u> within the limits of the clearance pocket volume. The system can be used to control the position of the clearance piston during compressor operation, providing a significant cost savings advantage over the older hand wheel system [Woollatt, 2002][3].

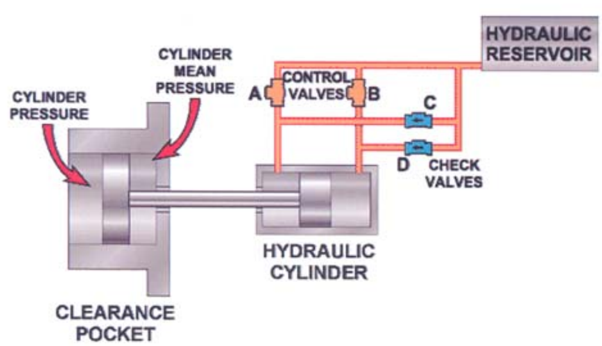

Figure 3-3. Dresser-Rand Hydraulic Variable Volume Clearance Pocket (HVVCP) [Woollatt, 2002][3]

The pocket clearance volume may also be controlled with a pneumatically actuated control system, such as the valve shown in Figure 3-4, developed by Gas & Air Specialty Products. The suction valve clearance is controlled by a pressure regulator system. This technique results in the same advantages as the hydraulic version, where stepless control of the compressor horsepower and flow is possible. The system is noted to be relatively simple to install and results in improved compressor operation [Compressor Tech, 1999][5]. A successful test of the product on a Worthington compressor showed that the pneumatic controls could be used to adjust the horsepower and flow through the crank end of the compressor.

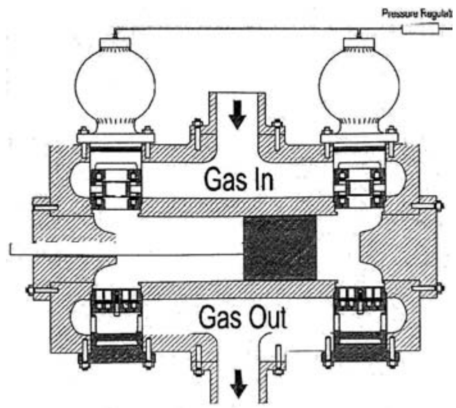

Figure 3-4. Pneumatic Variable Clearance System Developed by Gas & Air Specialty Products [Compressor Tech, 1999][5]

Another method of varying the volume or clearance in the cylinder is to control multiple fixed volume clearance pockets. Two examples of pneumatically operated fixed volume clearance pockets are shown in Figure 3-5. In both designs, a plug is pneumatically actuated to fully open or close a cylinder pocket. Both of these designs are easily adaptable to any existing compressor installation. The reliability and effectiveness of these methods is determined by the repeated use of the pneumatic actuation system, since some designs are more reliable than others [Wirz, 2003[6]; Ariel Corporation Application Handbook, 2001[7]].

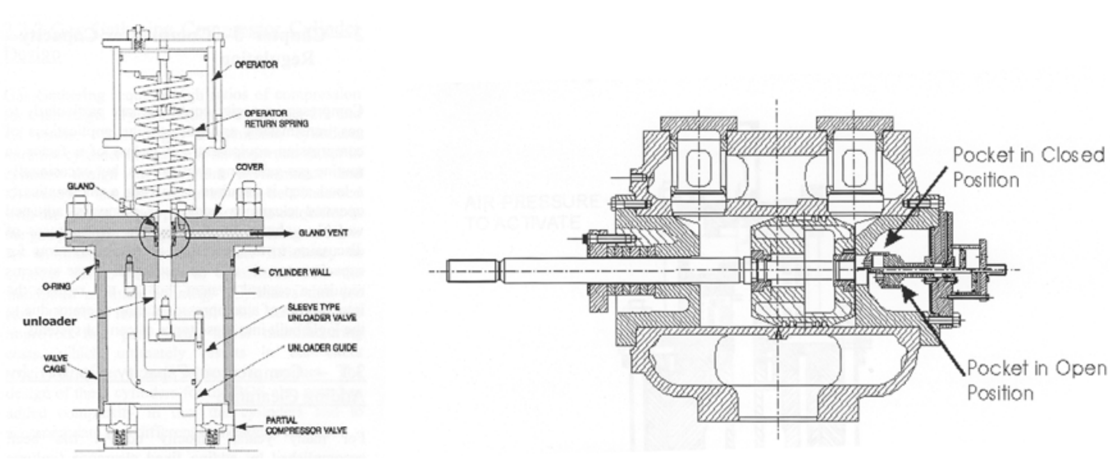

Figure 3-5. Pneumatically Actuated Fixed Volume Pockets, Dresser-Rand Version on Left [Wirz, 2003][6], Ariel Corporation Example on Right [Ariel Corporation Application Manual, 2001][7]

A final example of varying the volume in a compressor pocket is found in the Gas-Controlled Stepless Pocket (GSP). This design (shown in Figure 3-6) was patented by Gas & Air Specialty Products and is licensed to Dresser-Rand. The cylinder capacity is varied over a limited range determined by the pocket volume. A valve on the pocket controls the pocket opening with control gas pressure. The control gas (typically process gas) pressure may be varied to control when the pocket valve opens, which may be during part of the normal cylinder cycle. This is a relatively simple mechanical system, which does not require complicated hydraulics or expensive seals. The system may be installed on either the head end or crank end of the compressor cylinder.

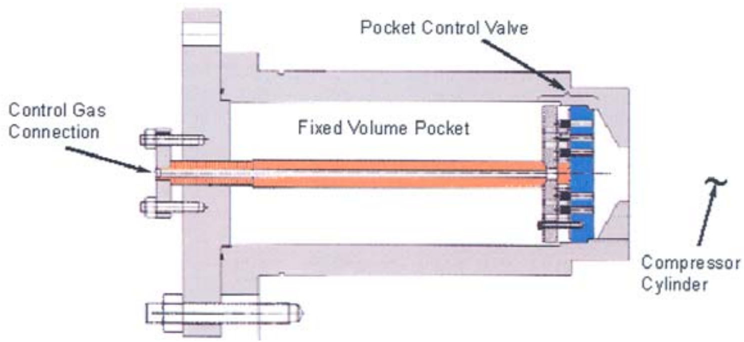

Figure 3-6. Gas-Controlled Stepless Pocket (GSP) Made by Dresser-Rand for Variable Load Operation [Dresser-Rand Technology, 2004][8]

These methods of controlling compressor capacity with variable clearance work fairly well on existing high- and low-speed compressors except at low-pressure ratios. The systems presented here are known to be reliable and to improve compressor load control and reduce operating costs. One of the primary disadvantages with the concept of controlling pocket volume is that the improvement or gain in compressor range (capacity, load) is fairly small particularly for low ratio applications [Phillippi, 2002][4]. To achieve a reduction in capacity at low compression ratios, the pocket volume may not be large enough to be effective [Woollatt, 2002][3]. In addition, these methods can be fairly costly to install, with additional installation costs incurred due to the required check valves. Low compression ratios may also be a challenge for the hydraulic versions of the variable volume pocket systems, because they rely on a minimum force on the suction side pressure in order to work.

3.1.2 UNLOADERS AND ACTIVE VALVE CONTROL

As an alternative to varying clearance area with pockets, end unloader technology may be used on a double-acting cylinder. Instead of adding and removing small volumes in the form of pockets, an unloader effectively removes one end of the compressor cylinder. Using an unloader on one end of a double-acting cylinder causes the cylinder to operate in a single-acting mode. In a typical suction valve unloader, the valve element (plate or ring) is held against the valve guard, which allows gas to pass equally into and out of the end of the cylinder. Other devices for deactivation open a separate cylinder port to the suction volume. Ariel Corporation recommends always deactivating the head end of the cylinder [Ariel Corporation Application Manual, 2001][7]. An example of a suction valve unloader is shown in Figure 3-7.

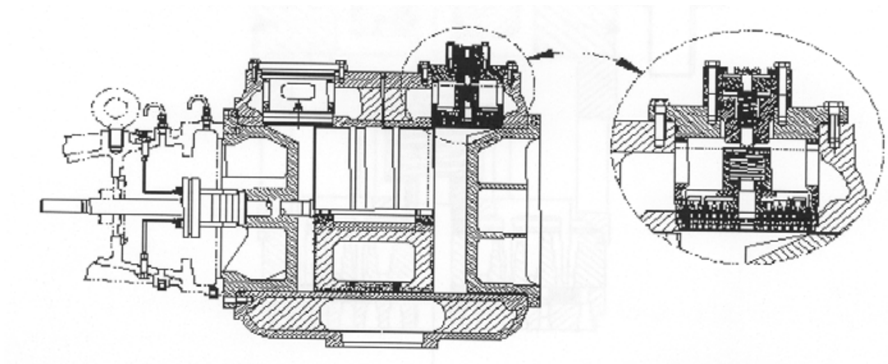

Figure 3-7. Ariel Corporation – Example of Suction Valve Unloader

This unloading technique is possibly more effective in reducing the compressor capacity, because the entire end volume of the cylinder is removed instead of just a portion of it. Though similar in its physical design to the variable pocket design, the unloader method causes more losses in horsepower because of the pumping losses for moving gases in and out of the head end. Suction valve unloaders can cause increases in temperature due to the re-circulation of undelivered gas on the deactivated end. Additionally, these unloader designs can have serious consequences for torsional and acoustic pulsation response.

Another currently used technique for varying compressor capacity is to vary the amount of gas compressed. One means of varying the amount of gas compressed is to actively control the suction valve motion, which effectively controls the amount of gas allowed into the cylinder for compression. If the suction valve is held open longer than the natural passive response of the valve would demand, then some of the gas flows back out of the cylinder, reducing capacity. Two existing products that achieve this objective are available in the current compressor market—the Infinite Step Controller (ISC) by Dresser-Rand and the Hydrocom valve by Hoerbiger.

The ISC, the Dresser-Rand version of a fully active valve control, uses finger unloaders that are actuated by high-pressure hydraulic pulses. The hydraulic signal is received from the hydraulic distributor in the control system. The control system is tied to the compressor operating torque. The finger unloaders hold the suction valve in its open position for a portion of the stroke, which reduces the amount of gas and capacity of the compressor. The system is shown in Figure 3-8. The ISC has been implemented on Ingersoll-Rand compressors at stations with highly fluctuating suction or discharge pressures, such as stations operating off underground storage fields or near a demanding market. The ISC has been installed and runs successfully at multiple stations worldwide.

The Hoerbiger Hydrocom valve has also been used in many applications worldwide, such as chemical plants, refineries, gas transport and storage applications, and stations serving power plants. The Hydrocom valve also uses a hydraulic control system where oil pressure is used to depress the suction valve, according to the amount of gas volume required. A fast-acting solenoid valve is used to control the hydraulic piston motion. The Hydrocom valve is shown in Figure 3-9. This system is also used to monitor temperature conditions at the suction valve to identify leaking valves or leaking piston rings.

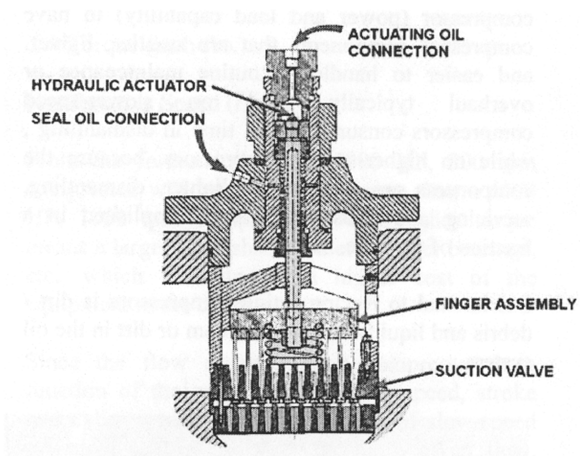

Figure 3-8. Dresser-Rand Infinite Step Controller (ISC), an Active Valve That Uses a Hydraulically Actuated Finger Unloader System

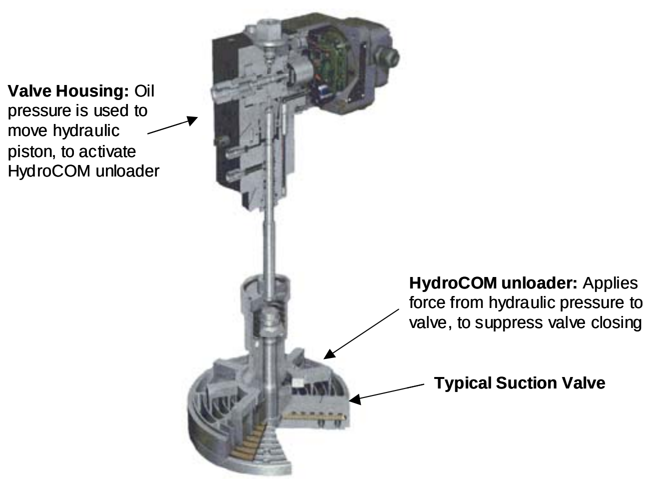

Figure 3-9. Hoerbiger Hydrocom, a Semi-Active Valve that Uses Hydraulic Force to Control Suction Valve Motion

The major disadvantages in the two designs described above, which control the suction valve motion, are their cost, pressure loss (and corresponding loss in efficiency), effect on pulsation, and complexity. The loss in efficiency and the increase in pulsation level can be particularly high for high-speed compressors. In general, these designs can also induce pulsations into the flow or cause pulsation control to be disrupted. The various electronic and mechanical subsystems can cause the systems to be unreliable. The associated cost of each system makes it difficult to justify the use of these designs on compressor stations with less fluctuating demand patterns.

3.1.3 ADDITIONAL METHODS

The other methods of controlling the capacity of a compressor are to vary the speed, as discussed previously, to change the condition of the suction gas, or to recycle flow around the compressor. A reduction in compressor speed will result in a reduction in the rate at which gas is compressed. The speed range of most engine drivers is limited to 20% or a little more, while the capacity range needed for compressors is 50% to 60%. Using a variable speed drive with an electrical motor or a mechanical variable speed transmission will achieve sufficient variations in speed for most reciprocating compressors. This method does effectively enhance the operational range of the compressor and the compressor efficiency, since lower-speed operation is often more efficient than high-speed operation [McKee and Smalley, 2001][1].

Although varying compressor speed achieves higher efficiency and increases the range of a compressor, the disadvantages of this method have caused it to only be used as an option when other capacity control methods do not work. One of the major disadvantages is that variable speed electric motors are considerably more expensive than constant speed motors. The gas engines normally used as compressor drivers are typically not rated over a wide speed range. In addition, reciprocating compressors can develop high torsional stress levels when operated below their design point and pulsation control is more difficult for a wide speed range compressor. Finally, electrical power for an electric motor is frequently more expensive than gas fuel, making variable speed electric motors even more costly [McKee and Smalley, 2001][1].

The remaining two methods for capacity control currently used in the industry essentially rely on varied operation schemes for the compressor, i.e., throttling suction pressure and recycling or bypassing gas. Throttling the suction pressure is one technique that reduces the amount of gas compressed by pinching a valve in order to lower the suction pressure and, hence, gas density, which reduces throughput. Booster compressors can be used to increase suction pressures and, hence, increase throughput. Recycle loops can also be used in operations where the gas from the discharge side is fed back to the suction side in order to reduce the total amount of gas delivered. It should be noted that recycling gas has little effect on compressor load. Pinching the suction valve to lower suction pressure actually increases the load (torque) for low ratio applications. Though less costly and less complicated to implement, these two methods are extremely wasteful, in that the net horsepower used for compression is not reduced. Energy is wasted, both by throttling and by recycling.

3.1.4 SUMMARY

In summary, the current load and capacity control techniques fall short of achieving effective control at a reduced installation cost. Many of the current technologies involve complicated control systems that have a high initial cost and require a high level of maintenance. Several of these methods are less effective at controlling load and capacity at certain conditions, such as low-pressure ratio, than they are at other conditions. Excluding the more wasteful techniques of throttling suction pressure and recycling gas, the current state-of-the-art methods include varying clearance volume, deactivating a cylinder end, actively controlling the suction valve motion to adjust the amount of compressed gas, or varying compressor speed.

3.2 CAPACITY CONTROL NEW CONCEPTS DISCUSSION

To be advantageous in the future, reciprocating compressors must be capable of meeting a larger capacity range, safely and efficiently. In the reciprocating compressor industry, improved methods of capacity control and load control are needed because the existing methods do not offer cost-effective means of reducing capacity sufficiently without a loss of efficiency. The “next generation” of capacity control methods must achieve reductions in flow rate of at least 50% despite changes in pressure ratio, including low-pressure ratios, and without a loss in efficiency or reliability. These three factors (effectiveness, efficiency, and reliability) will be combined with other criteria in determining the best options for improved capacity control techniques.

3.2.1 CONCEPTS GENERATION

In the initial stages of investigating improved capacity control methods, a brainstorming session was held with project team members. The session resulted in many ideas for capacity control, capacity measurement, and control methods for station optimization. Some of the ideas required an entirely new compressor design, while other concepts could be retrofitted to an existing compressor with the addition of new parts. The entire list of ideas is presented in Table 3-1 and Table 3-2. The ideas were separated after the brainstorming session by topic. The capacity control methods that pertained to a new compressor redesign in the area of variable stroke (rather than an external addition or single part redesign) were categorized separately. The different categories allowed the project team to analyze variable stroke concepts and capacity measurement concepts in more detail, apart from capacity control concepts. This present section of the report is limited to direct capacity control concepts. The remaining categories (variable stroke and capacity measurement) will be discussed separately in another section of this report.

Table 3-1. Capacity Control Concepts from Brainstorming Session

Capacity Control Concepts	Description
Concepts that could be retrofit:	
Head-end / crank-end bypass	Bypass gas to either end, through piston or around
Voricome gear box	Constant drive input, variable speed output
Discharge valve control	Actively control operation of discharge valve to allow gas in/out
Piston deactivation	Deactivate piston motion in cylinder to reduce capacity
Multiple drive planetary gear box	Drive gear box through ring or sun gear control
Hydraulic drive	Vary piston speed with hydraulic drive control
Cylinder with external end bypass	Bypass gas through external end of cylinder to reduce flow
Automatic transmission	Control speed through automatic transmission
Manual transmission	Control speed through manual transmission
Improved plug valve unloading mechanism	Unload cylinder end with insertion plug
Variable cylinder geometry	Vary geometry of cylinder to affect comp.volume
Use RF communication to vary piston motion	External non-intrusive method of varying piston
Zero-loss rings	Use valve with less frictional ring surface to actively control
Improved rod materials	Consider materials to enforce rod strength, aid in unloading
Include pocket in piston	Open/close pocket in piston to allow for more volume
Torque coupling	Use torque coupling analysis to decide how to unload cylinders
Piston in head, crank end adjustable doughnut	Control volume in crank end by moving donut
Crank or rod adjustable screw method	Make crankshaft or rod length adjustable
Control bypass line flow, external or in piston	Control bypass flow in piston or externally
Power turbine with recycle gas to drive gear box	Use power turbine on exhaust gas line, fed to gear box
Temp control (turbocharge) suction gas	Cool suction side gas to change capcity & reduce load
Pulsation optimization for suction gas control	Use increased P from high pulsation to aid suction

Table 3-2. Capacity Control Concepts (cont.)
Station Optimization Concepts from Brainstorming Session

Capacity Control Concepts	Description
Concepts requiring entirely new compressor design:	
Variable stroke geometry concepts	Use variable geometry to somehow vary piston stroke
Linear motor for variable stroke	Use linear motor to change length of piston stroke
Variable stroke piston	Vary length or time of piston stroke
Swash plate compressor	Use swash plate to control length of piston stroke
Variable length piston	Change length of piston or cylinder housing
Hydraulic variable stroke	Use hydraulic power to change stroke
Sanderson mechanism	Use patented Sanderson mechanism to vary stroke
Variable power on head-end, compress crank	Eliminate crankshaft and rod by powering piston
Free floating piston with variable pressure source	Control piston with pressurized hydraulic, no crank
Station optimization techniques:	
Supercharge or turbocharge suction	Change upstream gas to boost compression
Binary-weighted compressors	Use multiple comp. with binary-weighted flow amounts
Binary-weighted cylinders	Use one comp. w/ multiple cylinders w/ binary weight
Lead screw compressor	Change compressor to lead screw mechanism
Use high speed engine with low speed compress	Gain in efficiency with low speed comp, high speed engine
Rod load optimization	Optimize load on rods by controlling rod motion / work

To analyze the promising concepts in more detail, a shorter list of the more advantageous capacity control methods was created. In order to reach a shorter list of concepts, a rating criterion was applied to each of the brainstorming session ideas. The criteria included five characteristics of capacity control methods and an associated weighting factor for each characteristic. The criteria were efficiency, effectiveness, developmental risk, installed cost, and maintenance cost. Although other factors will be considered in the more detailed analysis, this initial set of five criteria provided a simplified method of ranking the larger list of concepts against one another. The description of each criteria and its associated weighting factor is provided in Table 3-3. The concepts' effectiveness and developmental risk were viewed as the most important factors, followed closely by efficiency.

Table 3-3. Initial Ranking Criteria Applied to Concepts from Brainstorming Session

Installation Cost	**15%**	Installation cost: Cost of installing device / concept and retrofitting existing compressor, ranked from 1 to 3, where (1) is mostly costly.
Effective Control	**25%**	Effective control: Effectiveness of control technique at reducing or adding capacity, ranked from 1 to 3, where (1) is least effective.
Develop. Risk	**25%**	Develop Risk: Developmental risk (likelihood of success) accounts for idea maturity, precedent designs with similiar concept, acceptance by industry and intellectual property concerns (others patent rights), ranked from 1 to 3, where (1) is most developmentally risky.
Maintenance	**15%**	Maintenance: Reliability component, susceptibility to operational maintenance expenditures, ranked from 1 to 3, where (1) is most costly.
Efficiency	**20%**	Efficiency: Gain in compressor efficiency compared to current technology, ranked from 1 to 3, where (1) is least efficient.

Each of the concepts was rated from a score of 1 to 3 in each criterion. Weighting developmental risk and effectiveness more heavily allowed the more practical and promising ideas to be rated highest. Developmental risk included many factors, such as idea maturity, current technologies of a similar concept, intellectual property that might make development more costly, and the acceptance by the industry (in terms of the cost to a manufacturer and marketing potential). Effectiveness was viewed as the range of capacity that the method offered, compared to the current technologies. Efficiency of the method took into account the operating efficiency of the compressor over the capacity range for which the method could be applied. Installation cost examined the cost of the hardware and downtime associated with installing the new concept. In addition, the cost to retrofit the new technology on an existing station was taken into account. Maintenance was also considered important in evaluating the concepts. Each concept's maintenance was evaluated, in terms of the product's reliability and cost to maintain or repair.

3.2.2 Short List of Concepts

The ranking criteria were applied to the entire list of brainstorming ideas. From the larger list, nine basic concepts were selected for the short list. In some cases, the concepts were renamed to allow the general concept to be considered in its most general form. For example, the idea of a greatly expanded clearance volume was approached through different ideas for pockets and variable geometry in the brainstorming session. However, in essence, the general idea of adding greater volume to control capacity is the same. Therefore, the concept was renamed "Larger Clearance Volumes" when it was added to the short list. The short list of concepts is shown in Table 3-4, along with the concept's primary objective and essential advantages.

Table 3-4. Short List of Concepts and Primary Design Objective

CAPACITY CONTROL CONCEPT	PRIMARY OBJECTIVE	CONCEPT ADVANTAGES
Variable speed (single drive)	Large range covered with speed variation	Very effective, retrofitable, similar effect on low and high speed machines
Variable speed (multiple drive)	Large range covered with speed variation	Very effective, retrofitable, similar effect on low and high speed machines
Temp control – suction gas	Easy to achieve temp change	No major installation required, waste heat is readily available
Improved deactivation	Larger flow through area will improve deactivation	Larger flow area will minimize losses, remote deactivation
Pocket in piston	Space conserved by using portion of piston volume for pocket	Reduction in swept volume will reduce capacity, simultaneously
Larger clearance volumes	Greatly expanded clear volume compared to current	Proven, efficient method with improved volume (expanded capacity range)
Bypass or bleed gas	Use bypass line to control amount of gas compressed	Reduction of horsepower required, can vary amount in bypass
Active valve control	Control valve actively for less losses than current valves	Full range of control available through improved flow area valve
Piston deactivation	Deactivate piston in order to bring down flow through machine	Possibly effective when used on a station basis with other machines.

The short list includes the following capacity control methods: variable speed with single or multiple drives, temperature control of suction gas, improved deactivation, pocket in piston, larger clearance volumes, bypass or bleed gas, active valve control, and piston deactivation. The preliminary evaluation of each of these nine concepts included a *qualitative* analysis of each concept. This analysis looked at the potential benefit from the concept meeting its primary objective and the advantages and disadvantages, from an engineering standpoint. In completing the preliminary evaluation, the concepts became more concrete and certain advantages/disadvantages more pronounced.

The variable speed methods can achieve a wide range of capacity, as shown in Figure 3-10. The figure shows the results of a compressor simulation performed over varied operating speeds. As indicated, a capacity range from 45% to 115% can be achieved for a high-speed reciprocating compressor machine by operating from 500 RPM to 1,200 RPM. The preliminary analysis showed variable speed methods to be promising concepts for both high and low speed machines that should be analyzed further.

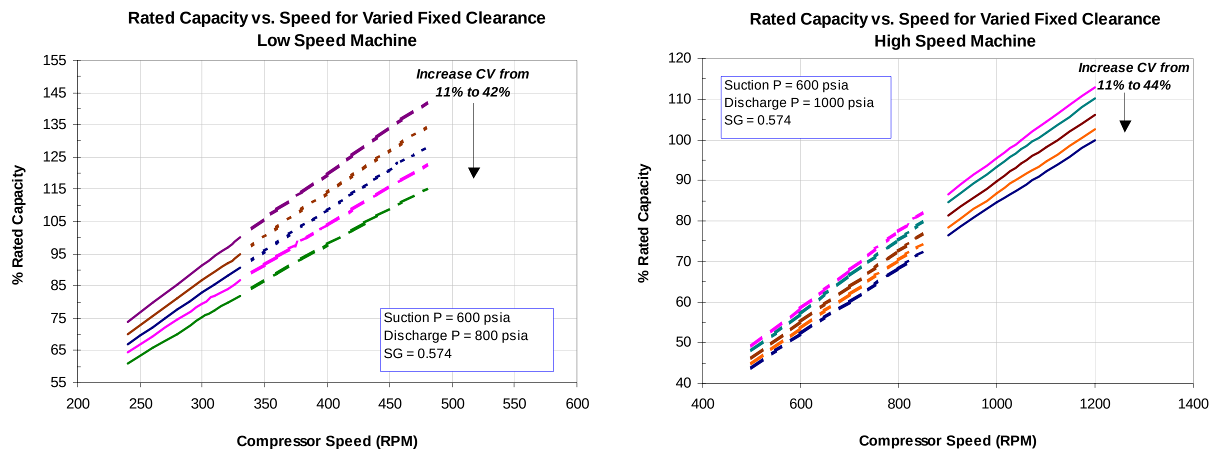

Figure 3-10. Variable Speed Method – Evaluation of Compressor Speed Range Required to Achieve 45-115% Change in Rated Capacity at a Pressure Ratio of 1.667

Larger clearance volumes are a well-proven method of capacity control. If the available space can be used with a well-designed pocket, this method would prove to be very useful. If the volume of the pocket can be made large enough, this method is very effective at reducing capacity without a penalty in efficiency. Figure 3-11 shows how this method's effectiveness is improved at higher-pressure ratios. By increasing clearance volume by 60% at a pressure ratio of 2.0, the reduction in capacity is only 10%. At a pressure ratio of 3.0, an increase of 40% in clearance volume reduces the capacity of a single cylinder head end from 80% to 20%.

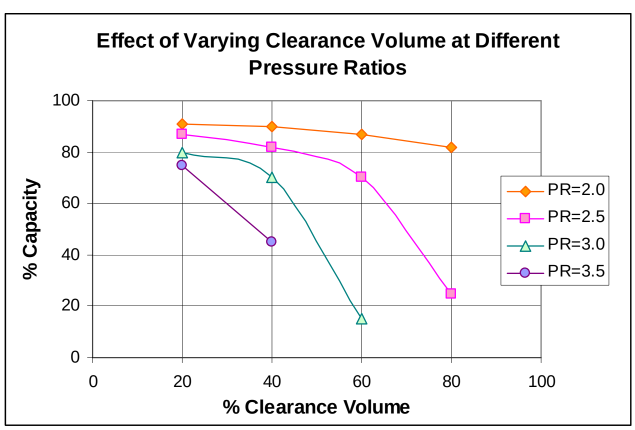

Figure 3-11. Larger Clearance Volumes – Effect on Rated Capacity at Varied Pressure Ratios for Cylinder Head End

Other concepts that received further analyses included improved deactivation, the pocket in piston design, bypass and bleed gas methods, and active valve control. An improved method of deactivating the head end (or possibly the crank end) of a cylinder ranked high among the list of ideas because the current losses associated with the process can be decreased with better flow-through area. However, the method only offers essentially one reduced capacity point (at 50% load) per cylinder. The pocket in piston idea concept is essentially a pressure-relief pocket placed in the unused space within the piston. Unfortunately, this idea of using the piston space will make the compressor operation less reliable if the pocket springs were to fail. Bypass or bleed gas methods were considered because they also can achieve a fairly large reduction in capacity with less of a cost in efficiency. The concept requires bypassing gas from the cylinder to the suction valves. Finally, active valve control is similar to current technologies that allow a portion of the available gas volume to be withheld from compression. This reduces the capacity of the machine and allows for a very effective range of capacity. One of its primary disadvantages to this method is the resulting loss in efficiency in reducing the gas volume in this manner.

3.2.3 Quantitative Analysis of Capacity Control Methods

In the quantitative portion of the analysis, the capacity control methods were ranked against the criteria provided by the industry Project Supervisory Committee (PSC). These criteria consisted of efficiency, effectiveness, reliability, installed cost, O&M cost, developmental risk, synergy, technical maturity, retrofitability, and economic viability. Each criterion was defined specifically in its relation to capacity control methods. Analyses of each method were conducted to determine how the methods ranked against each of the criterion.

Greater effectiveness and increased efficiency were viewed as the primary design objectives for improved capacity control. In order to quantitatively evaluate the methods' effectiveness and efficiency, a compressor performance simulation model was used. The model can accommodate changes in the compressor design or operation, in order to simulate the various control methods. For example, a typical compressor can be modeled as a baseline run. The baseline values can then be altered to expand the amount of clearance volume or deactivate the head end of a cylinder. The model output results provide the indicated horsepower, flow rate (in MMSCFD), valve losses, and pressure traces for each cylinder head end and crank end. The indicated horsepower per MMSCFD (million of standard cubic feet per day) can be plotted as a measure of efficiency for each method. In addition, the flow through the compressor can be compared to a typical baseline run in order to estimate the percentage reduction in capacity.

Four of the methods in the short list of concepts were modeled using the compressor simulation tool to understand the concept effectiveness and efficiency—variable speed, larger clearance volumes, active valve control, and improved deactivation. The capacity control methods were simulated for a typical high-speed and low-speed machine. The simulations were compared to a baseline run. Both types of compressors showed similar results. For the high-speed machine, the baseline run was performed at 900 RPM with a clearance volume of 65%. Simulations were run to determine efficiency of the compressor over a variable speed range (300 to 1,200 RPM), a variable clearance volume (20% to 300% of swept volume), using deactivation of the head end, and using active valve control. The results for each of the four methods for a typical high-speed unit, considering the compressor efficiency only, are compared in Figure 3-12.

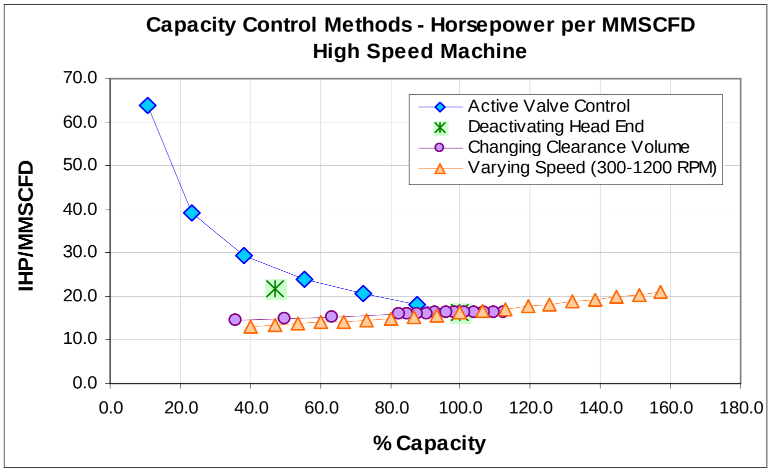

Figure 3-12. Summary of Four Capacity Control Methods' Efficiencies As Capacity is Reduced

As the figure shows, variable speed and larger clearance volumes showed an equal (or slight decline) in the horsepower required per MMSCFD, as the capacity was reduced from 100%. This improvement in compressor efficiency contrasts with the increase in horsepower required for active valve control and deactivation of a cylinder. Though active valve control achieves a wide range of capacity, which makes it an effective method, it shows a marked decline in efficiency. Deactivation shows a slight decrease in efficiency at 50% capacity, requiring 22 IHP/MMSCFD instead of 17 IHP/MMSCFD at 100% capacity. In addition, deactivation does not offer a continuous range of lower capacity points, making it less effective than some of the other methods. The difference between clearance and speed can be even greater if driven performance is considered. For constant speed variable clearance volume, the driven efficiency decreases as torque is reduced.

The remaining two concepts, bypass gas and pocket in piston designs, were also evaluated in terms of efficiency and effectiveness. Data supplied by Ariel Corporation at the 2005 Gas Electric Partnership Workshop and subsequent private communication [Phillippi, Ariel, 2005][9] was used to evaluate the bypass method. In the study of an Ariel compressor, a bypass line connecting the suction side gas to the cylinder volume was gradually opened, which caused the compressor flow to fall. The horsepower per MMSCFD was measured in the process. Figure 3-13 shows the constant speed curves for the bypass method. The curves show that the bypass method also requires more horsepower per MMSCFD as capacity is reduced, similar to active valve control or deactivation.

The analysis of the pocket in piston method revealed that it would not be as effective as the other capacity control methods because the internal piston volume does not offer a significant volume. The capacity range offered by this method would not compare to the larger range available for the variable speed method or larger clearance volume. The efficiency of the method was somewhat harder to model using the existing compressor software. However, an initial model used the added pocket volume as an additional clearance volume in the compressor cylinder. The method showed no penalty in efficiency as the capacity was reduced.

In addition to efficiency and effectiveness, the concepts were also rated in terms of the other criteria. Some of these criteria were more difficult to quantitatively define, such as

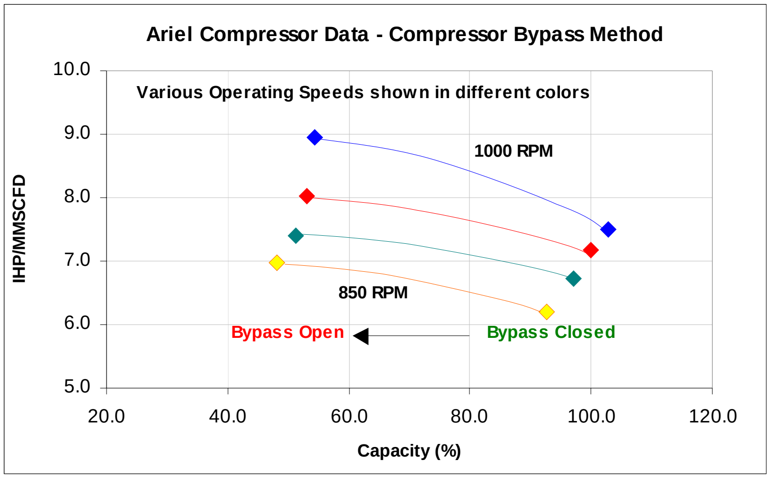

Figure 3-13. Efficiency versus Capacity for Bypass Method at Various Speeds

developmental risk, technical maturity, and retrofitability. However, as a starting point, a market comparison of the cost for existing technologies was completed. This survey allowed the concept's installed cost and O&M cost to be compared, relatively. In addition, synergy of each concept was evaluated by comparing the likely effects of the capacity control method on the pulsation characteristics of the compressor, the operational flexibility of the unit, and the maintenance costs. The results of the quantitative analysis were used to generate a decision matrix, in order to determine the capacity control concepts with the highest potential.

3.3 CAPACITY CONTROL CONCEPT SELECTION

After developing simulations of the various capacity control methods, the performance of each method was compared to current technologies for capacity control. It was necessary to determine which methods showed promise for improving current technology for reciprocating compressors by adding more flexibility, improving efficiency, or lowering cost. The most promising methods were to be developed further in a proof-of-concept test. The methods were given quantitative scores, in terms of effectiveness, efficiency, life, reliability, O&M cost, retrofitability, developmental risk, synergy, installed cost, economic viability, and technical maturity. The scores were then grouped into general categories, which could be scored in a decision matrix.

3.3.1 Ranking of Concepts

The seven capacity control methods from the short list were ranked in a decision matrix using the quantitative evaluation of each method. The scores for each method in each of the criteria categories were grouped on a general scale of "excellent," "good," "fair," and "poor." The resulting decision matrix is shown in Figure 3-14. For methods that offered improvements over the current technology in each category, the scores assigned were either "excellent" or "good." Methods that did not show great value or improvement in a particular area (such as effectiveness or efficiency) received a score of "fair." In some cases, the new capacity control methods actually performed worse than the existing technology, such as in the categories of O&M cost, economic viability, and installed cost. These methods were given a "poor" score in the respective categories.

	Effectiveness	Efficiency	Life & Reliability	O&M Cost	Retrofitable	Development Risk	Synergy	Installed Cost	Economic Viability	Maturity	Summary
Large Clearance Volume	Fair	Good	Excellen	Excellen	Good	Good	Excellen	Good	Poor	Fair	Good
Variable Speed, Single D	Good	Excellent	Fair	Good	Excellen	Excellent	Poor	Fair	Good	Excellent	Good
Improved Deactivation	Poor	Fair	Excellen	Good	Fair	Excellent	Fair	Excellen	Fair	Fair	Good
Bypass or Bleed Gas	Fair	Poor	Good	Fair	Good	Excellent	Good	Good	Poor	Fair	Fair
Active Valve Control	Excellen	Fair	Poor	Fair	Fair	Good	Poor	Fair	Excellent	Excellent	Fair
Pressure Relief Pocket	Poor	Good	Good	Poor	Excellen	Fair	Good	Fair	Good	Excellent	Fair
Variable Speed, Multiple	Good	Excellent	Fair	Poor	Poor	Fair	Poor	Poor	Poor	Poor	Fair

Figure 3-14. Final Decision Matrix Used to Rate the Capacity Control Methods Against Design Criteria

The decision matrix evaluated the methods' scores by applying a weighting factor to each criteria, as determined by the industry committee. The two methods that received the highest score were larger clearance volumes and variable speed with a single drive. These two concepts were selected for further development in a proof-of-concept test. (The other concept receiving a "good" score, as shown in the decision matrix, was improved deactivation. Based on the industry committee's input, this concept was not advanced to a proof-of-concept test because it was viewed as ineffective at offering a *range* of flow points for the compressor capacity, which was one of the essential design objectives in capacity control.)

3.3.2 Description of Selected Top Concepts

One of the top capacity control concepts was the method of using larger clearance volumes for capacity control. This method examined the type of capacity reduction that could be gained from using a sizeable increase in clearance volume, added either to the head end of a cylinder or through a pocket with some type of restriction. Existing clearance volumes use smaller increments of volume through the addition of pockets or "unloader" ends with relatively small additional volumes. The preliminary analysis indicated that if the clearance volume were increased from 200% to 300% of swept volume (a larger increase than existing technology), the corresponding change in the flow through the compressor would be significant. The initial compressor simulations did not indicate that the increased clearance volume would cause the efficiency to be reduced, although temperature rise in the compressed discharge gas was expected. Another factor affecting the amount of capacity reduction is the pressure ratio. The amount of capacity reduction per percent of clearance volume is greatly enhanced with larger pressure ratios.

The other selected capacity control concept was the method of varying the speed of the compressor with a single drive. This method would typically require that a variable speed drive be installed on an electrical motor that drives the compressor. An electric motor operated at single speed can be converted to variable speed with an inverter to vary frequency supplied to the motor. The net effect is that the compressor and driver are allowed to operate at a lower speed, which is more efficient for the compressor. Many compressors are offered as variable speed units because of the flexibility offered by this method. The preliminary modeling indicated that lowering the speed would allow the compressor to become more efficient for a reduced throughput.

3.4 CAPACITY CONTROL PROOF-OF-CONCEPT

The proof-of-concept testing was performed at the SwRI Metering Research Facility using an existing 250 HP Ariel compressor. The compressor operates at low pressures (45 to 210 PSIA) in a natural gas flow loop. It is a two-cylinder unit, with four suction and four discharge valves per cylinder. In its original configuration, the compressor operated at a single speed of 900 RPM with a small adjustable clearance volume (unloader) on the head end. The existing unloader allowed the clearance volume to be varied from 16% to 63% through a manually adjustable piston. Modifications to the compressor were carried out to convert the compressor to a variable speed unit and add a significantly larger clearance volume, in order to test the two selected capacity control methods. A variable frequency drive (VFD) was added to the existing electric motor. The VFD allowed the compressor to run over a continuous range of speeds between 400 and 900 RPM. Testing of the variable speed method was conducted in conjunction with the larger clearance volume test.

3.4.1 LARGE CLEARANCE VOLUME PROOF-OF-CONCEPT

The objective of the larger clearance volume proof-of-concept test is to establish that the model predictions correctly estimate the efficiency of the compressor as the clearance volume is increased, thereby dropping capacity. The testing of a significantly larger clearance volume demonstrates that clearance volumes can be used effectively to change capacity on a larger scale than typical compressor designs currently permit. The proof-of-concept test was designed to continuously measure the compressor cylinder gas temperature when the compressor was run with a larger clearance volume, in order to identify adverse heating effects, if any, due to larger clearance volumes. In addition, a restriction was to be added at the entrance of the clearance volume to model the effect of a restricted entrance on a typical volume pocket.

The design for a larger clearance volume essentially required adding a large cavity to the existing compressor cylinder with an adapter piece with the same bolt pattern as that of the existing unloader on the unit. The internal diameter of the cylinder bore and the corresponding unloader diameter was 9.562 inches. The new unloader was designed to bolt to the cylinder flange with the adapter piece. The design drawing for the larger clearance volume fixture is shown in Figure 3-15. The larger clearance volume test fixture had a small passage drilled through the end flange for temperature measurement during the proof-of-concept test. In addition, plugs were designed to fit inside the new volume, to allow various amounts of clearance volume to be tested. The total volume added to the cylinder through the test fixture was approximately 230% of swept volume.

In order to test different percentage amounts of clearance volume, several test fixtures were used to gain a more continuous spectrum of clearance volume points before testing the largest clearance volume (described above). The test fixtures used to generate the different clearance volumes are shown in Figure 3-16. The existing Ariel unloader allowed clearance volumes of 16% to 63% to be tested. After testing these two points, a "blind flange unloader" was installed in place of the existing Ariel unloader. This tested a clearance volume of 112%. Finally, the new larger clearance volume was added. Removable plugs were used to vary the clearance volume from 150% to 225%. A restriction was also added at the entrance to the large clearance volume as shown in Figure 3-17 in order to simulate the effect of a pocket restriction.

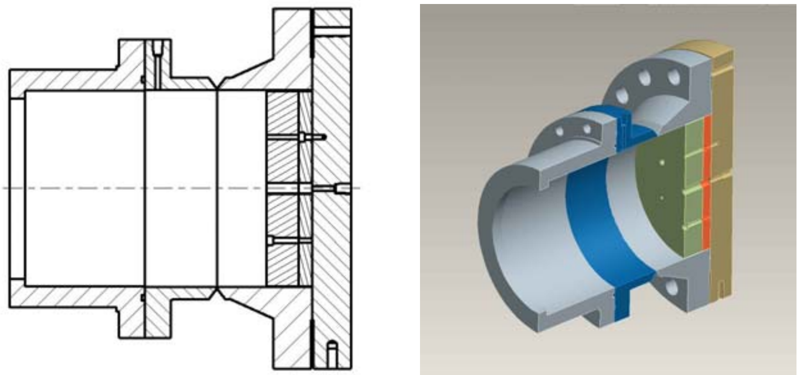

Figure 3-15. Conceptual Drawings of the Larger Clearance Volume Test Fixture

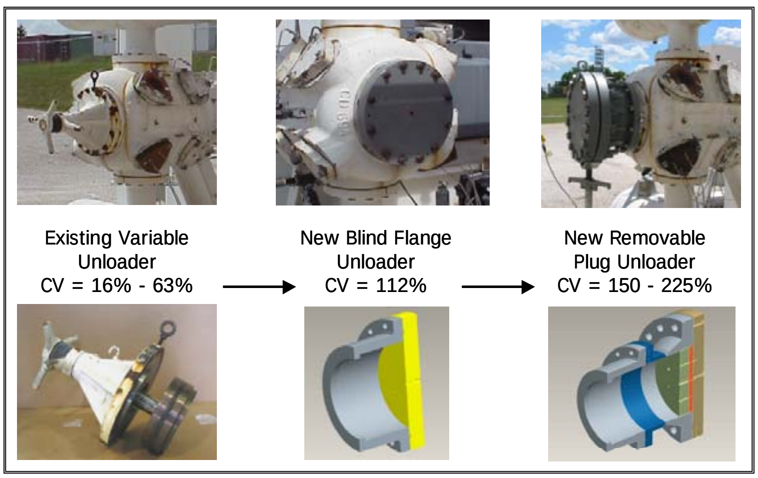

Figure 3-16. Stages of Tested Clearance Volumes on Ariel 250 HP Unit

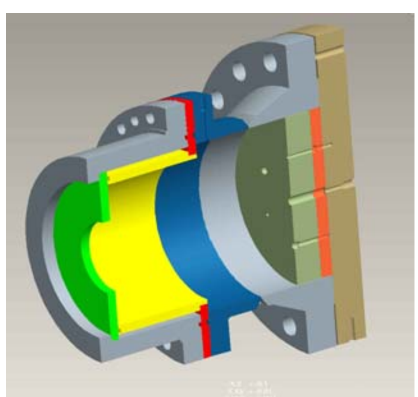

Figure 3-17. Large Clearance Volume Test Fixture with Inlet Restriction

The test plan for the proof-of-concept testing consisted of testing six fixed clearance volumes at 16%, 63%, 112%, 150%, 175%, and 200%. Each clearance volume was tested at three speeds of 500, 700, and 900 RPM. Two pressure ratios of 1.33 and 1.67 were used to compare performance of the compressor at different ratios. During each test, pressure-volume cards of the compressor were recorded in order to measure the required horsepower and the flow through the cylinder. The required horsepower was compared to the calculated adiabatic power in order to calculate compressor efficiency. This test allowed the variable speed method and the larger clearance volume method to be compared directly, in terms of effectiveness and efficiency.

The test results from the proof-of-concept testing are shown in terms of capacity versus operating speed in Figure 3-18. Constant clearance volume lines are shown to illustrate how the speed was reduced for each of the clearance volume steps. The capacity reduction for different clearance volumes is shown, from the minimum clearance volume of 16% to the highest clearance volume of 200%. The increase in clearance volume causes a reduction in capacity to approximately 60%. Larger clearance volumes are a more advantageous method of reducing capacity at higher-pressure ratios, as shown in Figure 3-19. This graph compares the model predictions for compressor capacity at various clearance volumes for two pressure ratios. At a clearance volume of 150%, the capacity is reduced to 50% at the lower pressure ratio of 1.33. With a higher-pressure ratio of 1.67, the same clearance volume provides a reduction of less than 20% capacity for the cylinder head end. The actual test points from the proof-of-concept test are shown to illustrate the good correspondence between the model predictions and the experimental data.

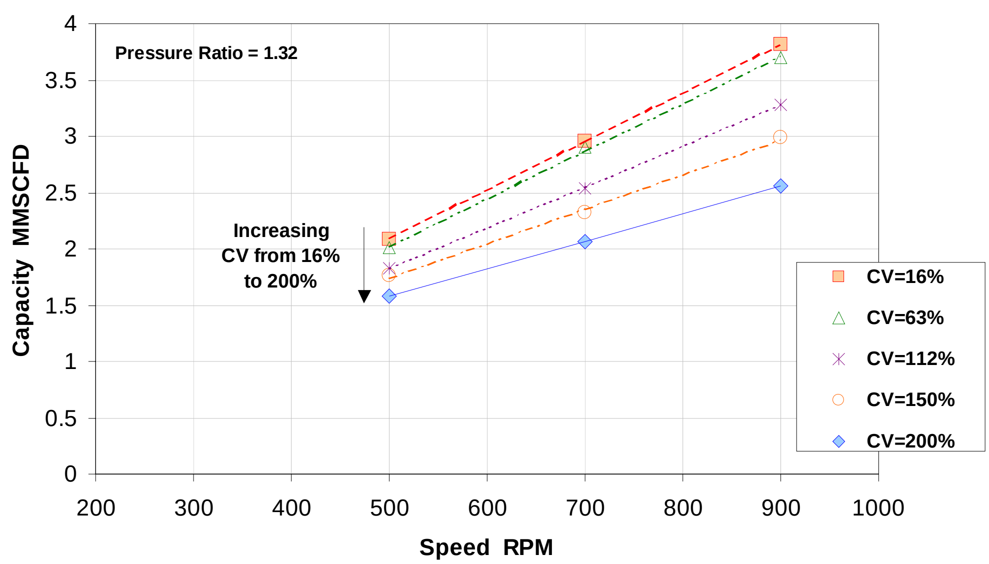

Figure 3-18. Capacity Points for Proof-of-Concept Test of Larger Clearance Volumes and Variable Speed Method

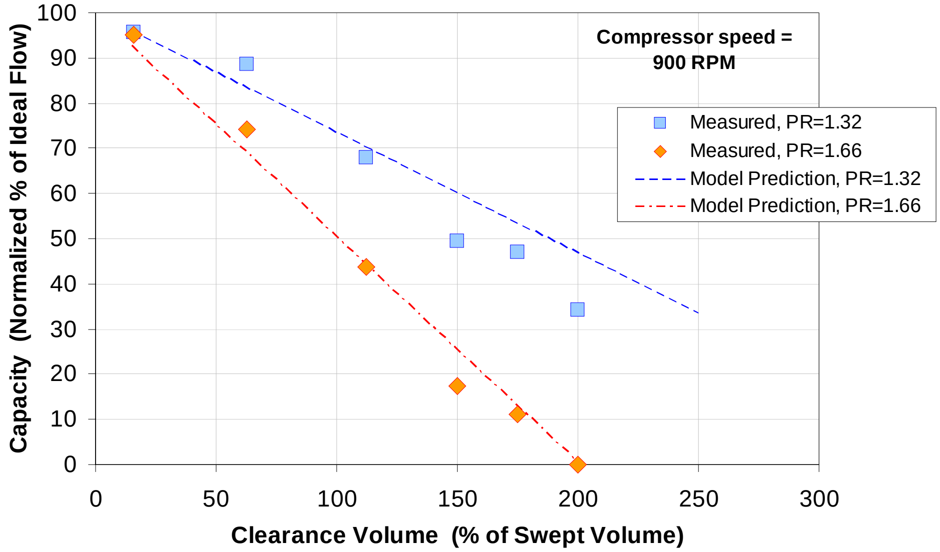

Figure 3-19. Comparison of Model Predictions with Experimental Data at Two Pressure Ratios

Testing was also performed with a restricted entrance in the clearance volume to simulate the effect of a restricted opening to a pocket volume on a cylinder. The testing did not show a significant change in the horsepower used, thus the adiabatic efficiency of the compressor was not affected by the restriction. However, a noticeable increase in the cylinder gas temperature did occur. For a restriction of 4.5 inches in a 9.5-inch bore, the cylinder temperature rise was 15 to 20°F depending on pressure ratio, in the cylinder with the restriction compared to the same compression without a restriction. This data is shown in Figure 3-20. Although the temperature increase within the cylinder was noticeable, the restriction did not cause a significant temperature increase in the discharge gas or a corresponding decline in adiabatic efficiency.

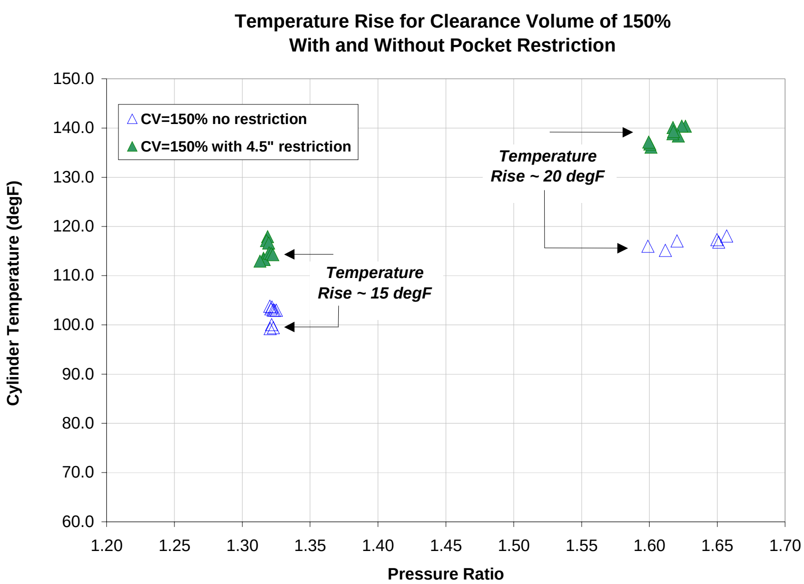

Figure 3-20. Cylinder Temperature for Clearance Volume of 150%, With and Without Pocket Restriction

The proof-of-concept test demonstrated that a variable clearance volume (16% to 200% of swept volume) could be used reliably to vary compressor capacity over a wide range (60% to 100%). In addition to the calculated flow through the compressor, the pressure-volume measurements of the cylinder provided an experimental value of the indicated horsepower. The expended horsepower could be compared to the adiabatic horsepower as a measure of compressor efficiency at each stage of clearance volume. The test results showed that the efficiency stayed nearly constant as clearance volume was increased and capacity was thereby reduced. An example of the measured efficiency for clearance volumes between 16% and 200% is shown in Figure 3-21 as the solid blue line. Though this example is for the case of 900 RPM at a pressure ratio of 1.32, results for all speeds and pressure ratios were similar. Figure 3-22 shows the efficiency as a function of capacity as reduced using clearance volumes at the higher-pressure ratio of 1.66 and compressor speeds of both 500 and 900 RPM. The fact that capacity can be controlled without a loss in efficiency by using large clearance volumes is demonstrated.

Figure 3-21 further shows the efficiency trend over a variable speed range of 500 to 900 RPM at a constant clearance volume of 16% as a red dashed line. As indicated, the compressor efficiency actually increases, from 68% to 82% as the speed is reduced. This trend is consistent with the known improvement in efficiency for lower-speed compression. As a capacity control method, varying speed appears to be a better option for reducing capacity in terms of efficiency of the compressor but is very dependent on performance of the driver. While increasing clearance volume causes the compressor capacity to be reduced, the efficiency of the compressor is maintained at a constant level. Variable speed, however, reduces capacity with a corresponding increase in efficiency of compression.

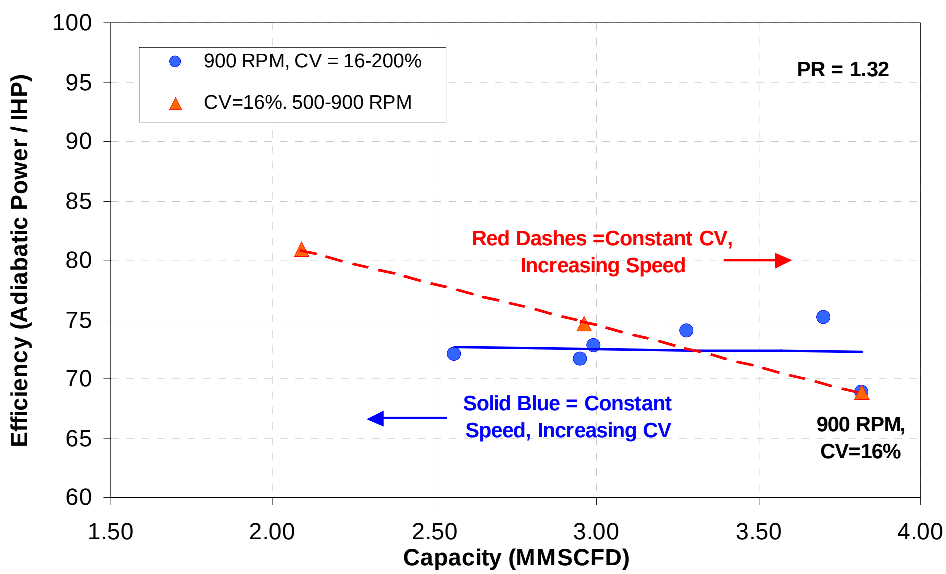

Figure 3-21. Measured Efficiency During Proof-of-Concept Test for Constant Clearance Volume and Constant Speed Cases

3.4.2 RECOMMENDATIONS FOR FUTURE WORK

Future testing of the larger clearance volume should be performed in order to analyze the compressor efficiency through measurements of enthalpy rise. These measurements should be

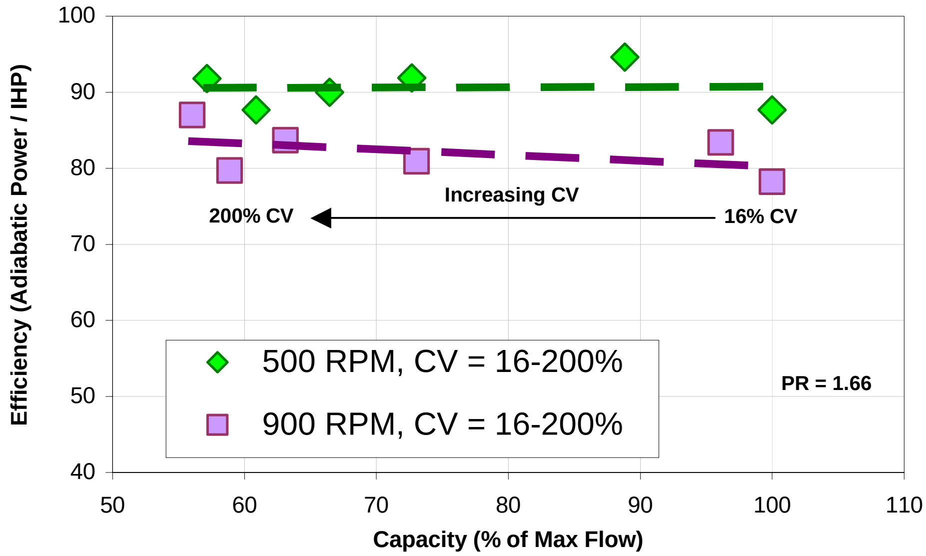

Figure 3-22. Efficiency as a Function of Capacity Achieved with Large Clearance Volumes at a Pressure Ratio of 1.66 at Speeds of 500 and 900 RPM

compared to the initial data presented, in order to conclusively determine the method's effect on accurately determined thermodynamic efficiency. In addition, more testing with the restricted flow area would provide insight into the effects of high cylinder gas temperatures on the compression process and the effect of currently used small clearance pocket openings. This information would help in designing pocket clearance volumes for compressors and predicting their effect on compressor performance. Additional work should also be aimed at designing methods of increasing clearance volume with the least penalty to space requirements.

The proof-of-concept test provided clear indications of the effectiveness and efficiency of using variable speed and larger clearance volumes for capacity control. Both methods prove to be effective at offering a continuous, wide range of capacities for the compressor. In analyzing the most efficient method for providing a reduced capacity, the efficiency of the compressor at 100% flow must be compared to the efficiency at capacities less than 100%. In this regard, the speed reduction causes the efficiency of the compressor but not necessarily the driver to increase, while the increase in clearance volume neither "helps nor hurts" the efficiency of the compressor, that is the efficiency stays constant. However, the use of variable clearance volume and fixed speed could result in lower torque and lower driven efficiency. The pulsation control is simplified when operating at a fixed speed and the driver requirements are not as complicated or costly. In summary, the two methods have distinct advantages that should be considered based upon the application, but both methods offer efficient ways to reduce capacity over a wide range.

3.5 CAPACITY VARIABLE STROKE NEW CONCEPT DISCUSSION

Varying the piston stroke length in a reciprocating compressor allows for capacity control without changing the drive speed of the compressor. Calculations and analyses performed demonstrate that changing the length of travel for the compression piston can have a significant impact on overall capacity. Figure 3-23 shows a representative efficiency curve for a high-speed reciprocating compressor. The plot shows that efficiency increases with a reduction in stroke length. In order for a variable stroke concept to be developed for commercial use, several technical and economic issues must be addressed. The following few sections present an overview of variable stroke concepts evaluated for this project and their feasibility.

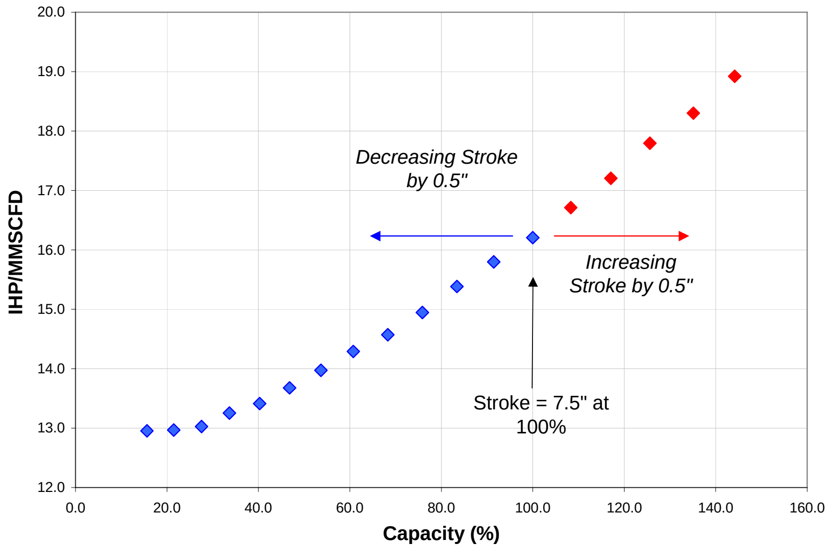

Figure 3-23. Efficiency Curve for Fixed Speed Reciprocating Compressor with Variable Stroke (100% Capacity at 7.5-inch Stroke)

3.5.1 Concept Generation

Multiple means of obtaining variable stroke were investigated. These mechanisms were consolidated into the nine general concepts listed in Table 3-5. Variable stroke approaches were selected to achieve variable capacity without requiring the drive speed of the compressor to be altered. The table gives a brief overview of the operational theory of each concept.

Table 3-5. Summary of Variable Stroke Concepts

CONCEPT	DESCRIPTION
Phased Piston	This concept utilizes two opposing pistons moving along the same axis. Capacity control is achieved by changing the phase angle between the two pistons. Electric motors could be used to alter the phase without stopping operation of the compressor. With the pistons in phase (zero degrees), the pistons move together and no compression takes place. With the pistons out of phase (180º), maximum capacity for the configuration would be attained. A full range of capacity is attainable at phase angles between these two extremes.
Linear Electric Motor	Two linear electric motor concepts were generated. The first is an external motor element that would use a linear drive in place of a crankshaft. A drive piston would power the piston rod. The second concept utilizes a free-floating piston to be driven without an external moving motor element. Thus, only a cylinder unit would be required for compression. This approach uses a linear induction motor similar to ones found on amusement park rides and magnetic-levitation trains.
Hydraulic Drive	Hydraulic fluid would be used to linearly actuate a drive piston. A traditional engine would be replaced by a series of hydraulic pumps that would be used to control hydraulic pressure in the drive section. Applying hydraulic pressure to the other side of the drive mechanism would allow for easy double-acting motion without the need for springs. Multiple configurations are possible for hydraulic drive compressors.
Expanding Piston	The compression piston would be constructed as two halves with a set of piston arms in between the two pieces. The arms could be extended or contracted by use of hydraulic fluid or a mechanical device. Such a method essentially allows for continuous variation of clearance volume. While not a

CONCEPT	DESCRIPTION
	variable stroke concept in the strictest sense, the expanding piston approach was analyzed as a part of the variable stroke investigation as it is a variable capacity concept that operates at a fixed compressor speed.
Adjustable Rod	The axial length of the connection rod could be extended or contracted by use of a linear actuator. Changing the length of the rod would alter the total stroke of the compression piston. The change in length could be achieved mechanically, hydraulically, or electrically.
Movable Pin	This method would encompass changing the offset between the center of the crankshaft and the connection point of the rod. Such a controllable connection would allow the stroke to be varied. Different mechanisms for changing this offset are possible.
Variable Cam	The crankshaft of a traditional engine would be replaced with a cam system. The cam would drive the connection rod. Springs in the compression cylinder would be required to assist with the return of the piston in most practical cam configurations.
Sanderson Mechanism	This concept utilizes a swiveling union joint to allow for piston actuation along multiple axes. A traditional rotary engine could be used as the drive mechanism. This concept would require technology similar to the movable pin concept to be developed in order for variable stroke to be achieved.
Crankless Engine	This concept essentially removes the "middle man" of the power system by directly using linear motion in an internal combustion engine to drive the piston. The cylinder section on one side of the piston would be used for ignition and the other side for compression. The drive section would use the combustion of compressed fuel to move the piston along the cylinder axis. A small amount of discharge gas could be used to assist with piston return.

3.5.2 Quantitative Analysis

An initial assessment of viability for each of the concepts was performed. The evaluation criteria for the proposed variable stroke approaches were grouped into four categories based on expected industry requirements: reliability, efficiency, cost, and other. Descriptions of each evaluation category, as well as scoring guidelines, are given in Table 3-6. Within a particular category, each concept was given a score of one (1–below nominal), two (2–nominal), or three (3–above nominal). The product of weight and score gave the point total for a particular concept in that evaluation category. This weighting system was used to provide an in-depth decision-making tool for determining a short list of concepts to analyze with more detail.

Table 3-6. Category Descriptions and Guidelines Used for Concept Evaluation

	CRITERIA	WEIGHT	DESCRIPTION
Reliability	*Failure Modes*	3	Component and/or subsystem failures. A score of one (1) was assigned to failure modes that are considered catastrophic to compressor operation. Temporary compressor shutdown can result from failures with a score of two (2). Minor failure modes were given a score of three (3).
	Failure Frequency	3	Both the likelihood and frequency of failure modes being achieved. The following scale was used to assign scores: frequent failure (1), occasional failure (2), rare failure (3).
	Required Control System	2	The complexity of the control system, both in overhead and in material resources. A complex control system with multiple failure possibilities and/or significant repair/maintenance requirements received a score of one (1). A self-sustaining system with only minor chance of failure was given a score of three (3).
Efficiency	*Maintenance*	2	Amount of labor and material resources required maintaining system. Desired maintenance program is infrequent shutdown with minimal required resources. This situation resulted in a score of three (3). Frequent and/or costly maintenance and repairs resulted in a score of one (1).
	Gas Losses	1	Amount of compressed gas not used for direct transmission. A potential cause could be a requirement for some discharge gas to be used to aid in piston actuation. No loss of gas resulted in a score of three (3), some gas loss a score of two (2), and significant loss a score of one (1).

	CRITERIA	WEIGHT	DESCRIPTION
	Mechanical Losses	3	Friction, heat, etc. diminishes amount of horsepower that can be transferred from engine to actual compression. Marginal loss resulted in a score of three (3), acceptable loss in a score of two (2), and significant loss a score of one (1).
	Engine Loading	2	System should cause minimal excess loading of input source. If only small loading is present, or if concept does not utilize a traditional engine, the score was assigned as three (3).
	Amount of Variable Stroke	3	Concept should provide significant amount of stroke variation as well as good resolution and control of this stroke. The score is a relative ranking of the various concepts. The nominal score was two (2). Only methods that have significantly small (1) or large (3) displacements received scores other than two (2).
	Maximum Flow Rate	3	Takes into account required cylinder power, piston actuation speed, etc. A concept that acts too "slow" or requires too many cylinders for operation received a score of one (1).
Cost	*Implementation Cost*	3	Disregarding existing infrastructure, this is the cost it would take to construct the compressor with the proposed approach. It is a relative measure of the total cost of each concept. The assigned scores were nominally a value of two (2). Excessively expensive or inexpensive concepts, relative to the other approaches, were given scores of one (1) or three (3), respectively.
	Retrofit Cost	3	Cost required modifying an existing compressor to allow new concept to work. Same criteria apply as given in Implementation Cost. If concept requires a new compressor altogether, the score assigned was one (1).
	Extra Equipment	1	Cost required for extra components, such as hydraulic pumps, electric motors, heat exchangers, etc. A concept requiring little or no such components was given a score of three (3).
	Power Cost	2	Costs incurred for power required to drive compressor. This is relative metric between the various concepts.
	Maintenance Cost	2	Costs incurred for routine maintenance and repairs on the compressor. This is relative metric between the various concepts.
	Developmental Cost	1	Costs incurred for the development of concept from initiation to working full-scale prototype. This is relative metric between the various concepts.
Other	*Environmental Concerns*	3	Implementation of concept impacts environment in means, such as excess power consumption, air and/or water pollution, noise, emissions, etc.
	Safety	3	Concept may jeopardize safety and health of workers, maintenance personnel, nearby residents, etc.
	Innovation	3	Novel approaches and technologies make concept more attractive to investors, government subsidies, etc.

The nine concepts were scored for each of the categories previously detailed. The scores were normalized to the maximum score (for linear electric motor) to provide a relative ranking of the concepts. The rankings for each concept are provided in Table 3-7. Three concepts had significantly better scores than the other concepts: linear electric motor, phased piston, and hydraulic drive. These three concepts were chosen as the "short list" for further evaluation and are described in more detail in Section 3.5.3. A demarcation in scores is apparent between the top three concepts and the remaining six variable stroke ideas. This split is largely due to significant design or implementation issues associated with each of the six lowest-related concepts. It should be noted that the scores were based upon predicted behavior of each concept from initial quantitative analysis.

Table 3-7. Ranking of Variable Stroke Concepts

	Weight	Linear Electric Motor	Phased Piston	Hydraulic Drive	Expanding Piston	Variable Cam	Movable Pin	Adjustable Rod	Sanderson Mechanism	Crankless Engine
Failure Modes	***3***	3	2	2	2	2	2	1	2	2
Failure Frequency	***3***	3	3	3	2	2	1	1	2	1
Required Control System	***2***	1	2	2	2	2	2	2	2	2
Maintenance	***2***	3	3	2	2	1	2	2	2	2
Gas Losses	***1***	3	2	3	2	3	3	3	3	1
Mechanical Losses	***3***	2	2	2	2	1	2	2	1	1
Engine Loading	***2***	3	2	2	2	2	2	2	2	3
Amount of Variable Stroke	***3***	3	3	3	2	2	3	3	2	1
Maximum Flow Rate	***3***	3	3	2	3	3	3	3	2	2
Implementation Cost	***3***	1	2	2	2	2	2	2	1	1
Retrofit Cost	***3***	1	2	2	2	2	2	2	1	1
Extra Equipment	***1***	2	2	1	2	2	2	2	1	2
Power Cost	***2***	3	2	2	2	2	2	2	2	2
Maintenance Cost	***2***	3	2	2	2	1	1	1	1	2
Developmental Cost	***1***	1	2	2	2	2	2	2	1	2
Environmental Concerns	***3***	3	2	3	2	2	2	2	2	1
Safety	***3***	3	3	3	2	2	1	1	2	2
Innovation	***3***	3	3	3	1	2	1	1	3	2
Relative Score		**1.000**	**0.963**	**0.944**	**0.804**	**0.776**	**0.766**	**0.738**	**0.720**	**0.645**

3.5.3 SHORT LIST OF CONCEPTS

Three ideas were chosen for further evaluation based on their high scores in the initial concept evaluation. The three approaches are linear electric motor, phased piston, and hydraulic drive. This section provides a brief description of each concept.

3.5.3.1 Linear Electric Motor

Two means of integrating a linear electric motor into a reciprocating compressor have been identified. The first is an external motor element that would drive the piston rod. This arrangement allows existing cylinders to be used. A rotary engine would be replaced with a linear motor. The second method only requires the cylinder and not any external moving parts. The piston in the cylinder is free-floating and is reciprocated by the use of a linear induction motor. Such a motor is used in amusement park rides to quickly accelerate roller-coaster trains. Similar technology is also in wide use for magnetic-levitation high-speed trains. This concept would involve lining sections of the cylinder with electromagnets.

3.5.3.2 Hydraulic Drive

A hydraulic drive compressor design would use pumps in place of a traditional engine to drive the compression piston. A series of hydraulic pumps would provide pressurized fluid to the drive section of the compressor. A simplified configuration of a hydraulic drive is shown in Figure 3-24. The blue tubes on the left are hydraulic supply lines that drive the power piston in a reciprocating manner. This power unit is essentially a double-acting piston. Pressure would be applied to each side of the piston in an alternating fashion. One possible constraint is that high speeds may not be attainable due to limits on how much fluid could be pumped. However, longer stokes may be possible and leveraging differential areas in the pistons would reduce the hydraulic pressure requirements. A manifold arrangement is shown in Figure 3-25 without the cylinders. This configuration would allow for single-acting compression in multiple cylinders.

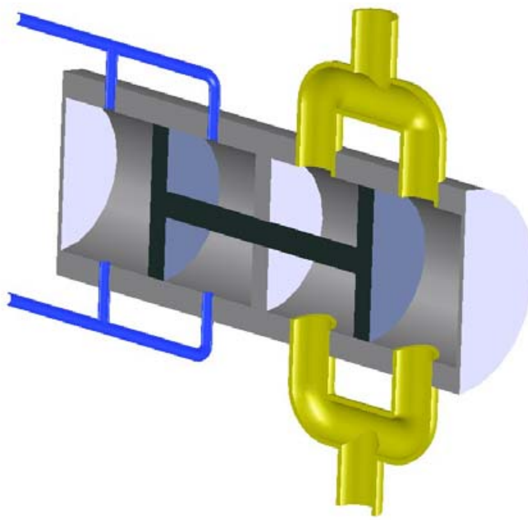

Figure 3-24. Simple Configuration of Hydraulic Drive Compressor

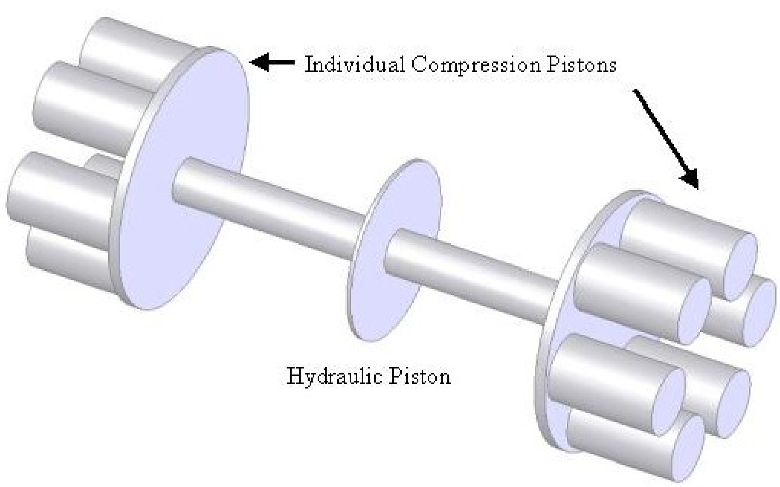

Figure 3-25. Manifold Arrangement Without Cylinder Shown

3.5.3.3 Phased Piston

The phased piston concept utilizes two traditional engines to drive opposable pistons. A cylinder would be constructed to house both piston assemblies. The arrangement could allow for either single- or double-acting compression. The two figures shown in Figure 3-26 depict a single-acting configuration with suction and discharge valves housed between the two pistons. In a double-acting compressor, additional valves would be placed on the crank end clearance of each piston. The first image shows the two pistons out-of-phase with each other. This arrangement allows for maximum compression. The second image shows the pistons moving in phase. In this configuration, no compression is taking place between the pistons. Varying the phase angle, therefore, provides virtually infinite variation in capacity between zero compression and maximum capacity. Straightforward analysis would provide exact capacity outputs for any desired phase angle. If a double-acting cylinder were in use, compression would also occur on the return path of each piston. It should be noted that the speed of each engine would remain constant, so once a phase angle is selected, the phasing would hold constant.

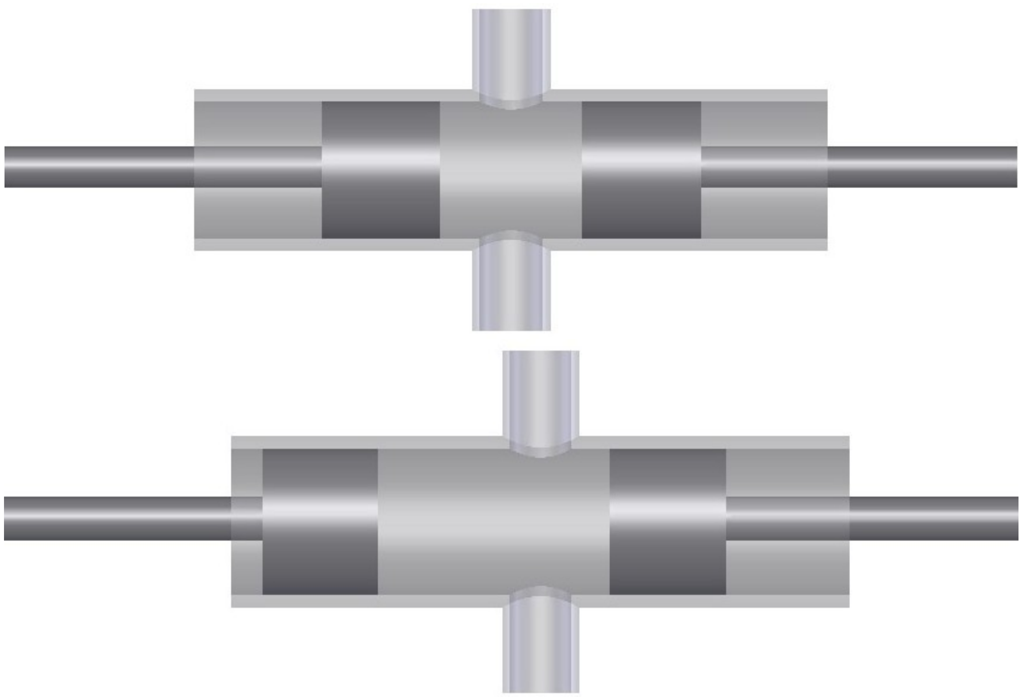

Figure 3-26. Single-Acting Configuration With Suction and Discharge Valves Housed Between Two Pistons

3.6 CAPACITY VARIABLE STROKE CONCEPT RANKING AND SELECTION

Further analysis of each concept was performed with emphasis on the three concepts on the short list: linear electric motor, phased piston, and hydraulic drive. These concepts, along with the other six original variable stroke ideas, were evaluated based on the criteria for the ten decision categories outlined in Table 3-8. A summary of the results is presented in Figure 3-27. Two concepts, linear electric motor and phased piston, were considered “good” options and were studied more completely. The driving mechanism for the hydraulic drive concept being removed from the short list is the significant flow rate and pressure required for implementation for real-world use. The cycling of the hydraulic section would also introduce significant control valve life issues and repetitive hydraulic pumping would create heat removal problems. The hydraulic drive was rated as “poor” in the area of effectiveness.

Table 3-8. Capacity Control Concept Evaluation Criteria

CRITERIA	DEFINITION	HIGH RANKING	LOW RANKING
Effectiveness	Ability to provide a 50% turndown for required ratio	Significantly more effective than current technology	Less effective than current technology
Efficiency	Highly efficient with low parasitic pressure drop	Significantly higher efficiency with lower pressure drop than current technology	Lower efficiency with higher pressure drop than current technology
Life/Reliability/Durability/ Ruggedness	Long-life with high reliability, durable and ruggedness	Significantly better life than current technology	Shorter life than current technology
O&M Cost	Cost of O&M	Significantly lower O&M cost than current technology	Higher O&M cost than current technology
Installed Cost	Overall cost to engineer, purchase and install.	Significantly lower cost than current technology	Higher cost than current technology
Synergy	Interaction with balance of system	Minimal adverse effect on valves and pulsation control	High adverse effect on valves and pulsation control
Technical Maturity	Highly developed concept maturity	Minor deviation from current technology or in use in other applications	Basic principle not yet reduced to practice
Retrofitable	Interchangeable within the current compressor system	Direct replacement for current technology	Not easily installed within the current compressor system
Developmental Risk	Risk of successful development	High probability of success within project schedule and budget	Significant schedule and budget to develop with low probability of success
Economic Viability	Economic viability for the vendor and owner that addresses critical market needs with high market acceptance	High viability with high market needs and high market acceptance	Low viability for the vendor and owner with low market acceptance

	Effectiveness	Efficiency	Life & Reliability	O&M Cost	Retrofitable	Development Risk	Synergy	Installed Cost	Economic Viability	Maturity	Summary
Linear Motor	Excellent	Excellent	Excellent	Excellent	Fair	Fair	Good	Fair	Good	Good	Good
Phased Piston	Excellent	Good	Good	Good	Good	Good	Fair	Good	Good	Good	Good
Hydraulic Drive	Poor	Fair	Good	Good	Good	Fair	Good	Fair	Good	Good	Fair
Expanding Piston	Fair	Good	Good	Fair	Good	Good	Fair	Good	Fair	Fair	Fair
Adjustable Connection Rod	Good	Good	Fair	Fair	Fair	Fair	Fair	Good	Good	Fair	Fair
Connection Rod Offset	Good	Good	Fair	Fair	Fair	Good	Fair	Good	Fair	Fair	Fair
Variable Cam Drive	Good	Good	Fair	Fair	Poor	Fair	Fair	Fair	Good	Poor	Fair
Crankless Engine	Fair	Fair	Fair	Good	Poor	Poor	Good	Fair	Fair	Poor	Fair
Sanderson Mechanism	Fair	Fair	Poor	Fair	Poor	Fair	Good	Poor	Fair	Fair	Fair

Figure 3-27. Evaluation Table for Variable Stroke Concept

Some initial calculations were performed on the phased piston concept. While the concept has significant capacity control, it is relatively inefficient when compared with other compressor configurations. Additionally, unbalance and pulsation effects could be significant. Thus, the linear electric motor was selected as the variable stroke idea to evaluate for proof-of-concept.

3.7 LINEAR ELECTRIC MOTOR COMPRESSOR PROOF-OF-CONCEPT

The objective of this analysis was to determine the feasibility of driving a reciprocating compressor with linear electric motors. Such a compressor would have the ability to vary the length of the piston stroke to provide a wide range of flow capacity control while operating at a fixed drive speed. A fixed speed compressor minimizes pulsation control problems and allows for relatively simple control system complexity. A linear motor compressor would have a high resolution of stroke control and would produce relatively low noise and onsite emissions.

A key design issue is the thrust force required to act against rod loads induced by inertial loading and gas compression. A mathematical model was developed to calculate the cylinder rod loads for a linear motor driven compressor cylinder. The model was then used to investigate design trade-offs such as stroke length and bore size on the resulting linear motor drive requirements. Once the linear motor performance requirements were defined, the performance capabilities of linear motor were investigated by gathering information from literature and by contacting motor manufacturers.

3.7.1 LINEAR ELECTRIC MOTOR BACKGROUND

Linear motors operate on the same general principles as rotary motors. A linear motor can be thought of as a rotary motor cut down the center of the shaft and rolled out flat. The coil windings are called the primary element of the linear motor. The rotor element in a linear motor is called the secondary element. There are two common types of linear motors. Permanent magnet (synchronous) motors contain permanent magnets in the secondary element. Linear induction (asynchronous) motors use a conductive material, such as aluminum or copper, for the rotor. Both types of motors have been explored for their benefit to the work in this project.

Permanent magnet motors require that the power to the coils be correctly timed to match the magnet position in the secondary element. These motors require a position feedback sensor and a motor controller with a variable frequency power supply. Permanent magnet motors are commonly used for applications requiring precise velocity or position control such as motion control for machine tools. Positioning accuracy of less than 0.001 inch is possible with some permanent magnet linear motors. These motor-controller systems provide position and speed control capabilities well beyond the requirements for driving reciprocating compressor pistons.

Linear induction motors have rotor elements that are commonly made from aluminum or copper reaction plates. These motors use a three-phase AC current in the coils to produce a moving magnetic field that, in turn, induces current in the reaction plate. The induced current in the reaction plate also produces a magnetic field that interacts with the coils to produce the motor thrust force. Because the current in the reaction plate is induced, there is no requirement to carefully time the power to the coils as in a permanent magnet motor. Control systems for linear induction motors, therefore, can be simpler than permanent magnet motors.

3.7.2 MATHEMATICAL MODEL

The model computes the gas pressure forces on the piston and the force needed to accelerate the reciprocating mass. The losses associated with friction and valve pressure losses were neglected and ideal gas behavior was assumed. The output from the model included the force, velocity, and acceleration that the linear motor must provide to drive the cylinder. Feasibility was determined by defining the linear motor requirements for driving a compressor cylinder (force, displacement, and velocity) and determining if current linear motor technology meets these requirements.

The simulation model computed the piston displacement, velocity, and acceleration as a function of time throughout the piston stroke. The force needed to accelerate the combined piston and motor masses during the stroke were also calculated. The motor force that is needed to drive the piston is the sum of the gas pressure forces on the piston and the inertial loads needed to accelerate the piston, piston rod, and moving motor element masses. Following the conventions shown in Figure 3-28, the force the motor must produce (F_{motor}) is given by:

$$F_{motor} = (m_{piston} + m_{motor}) \cdot a - F_{HE} - F_{CE} \qquad \text{(Eq. 3.1)}$$

where the masses of the piston and motor are given by m_{piston} and m_{motor}, "a" is the acceleration of the piston, and F_{HE} and F_{CE} are the gas pressure forces on the piston head and crank ends, respectively.

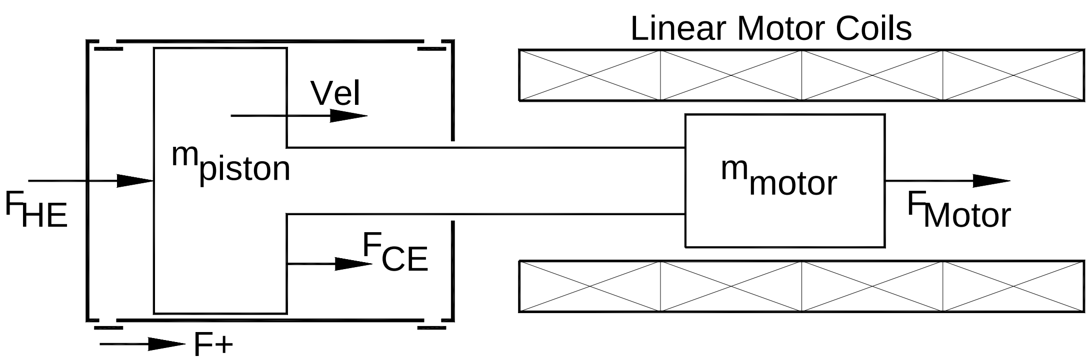

Figure 3-28. Diagram Showing the Forces Acting on the Piston and Accelerated Masses

The input parameters used in the calculations to analyze the drive motor performance included:

Compressor Design:	Cylinder Bore Diameter
	Cylinder Rod Diameter
	Stroke Length
	Clearance Volume
	Piston and Rod Mass
Pipeline Conditions:	Compressor Suction Pressure
	Compressor Suction Temperature
	Compression Ratio
	Gas Properties: Gas Constant and Specific Heats

Motor Parameters: Reciprocating Motor Mass
Stoke Rate (Piston Strokes per Unit Time)
Piston Acceleration/Deceleration Times

A trapezoidal velocity profile, shown in Figure 3-29, was used as a basis for model calculations. The piston undergoes constant acceleration from rest beginning at top dead center (TDC). After a given time, the piston moves at a constant velocity until the piston is slowed at a constant deceleration to bottom dead center (BDC). The piston returns to the top of the cylinder in a profile mirroring this first leg.

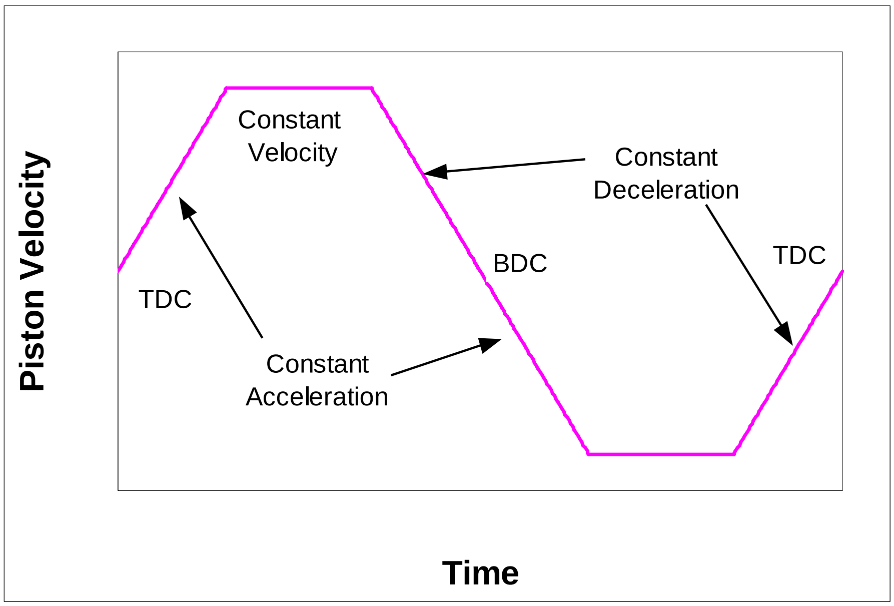

Figure 3-29. Trapezoidal Piston Velocity Profile

The drive motor force, velocity, and power requirements vary throughout the cylinder stroke and depend upon the cylinder geometry and the operating conditions. The model was used to determine the operating conditions where linear motors would best be able to drive compressor cylinders.

A typical medium speed compressor operating case will be presented here to demonstrate the simulation model calculations. The cylinder geometry and operating conditions are summarized in Table 3-9. For this case, the piston velocity versus time is shown in Figure 3-30. The piston peak velocity for this case is 20.8 ft/sec and a complete stroke requires 0.08 seconds (750 strokes per minute). The gas pressures on the piston head and crank end for a single stroke are shown in Figure 3-31 along with the force needed to accelerate the reciprocating masses of the piston, rod, and motor. The variations in the gas pressure forces during the stroke coincide with the cylinder pressure going from the suction to the discharge pressure during the stroke. The force needed to accelerate the reciprocating mass varies from zero during the constant velocity portion of the stroke to over 70,000 lbf when the piston is accelerating or decelerating. The inertial forces are significantly high because the reciprocating mass must be accelerated and decelerated very rapidly to complete an entire stroke in 0.08 seconds.

Table 3-9. Operating Parameters for the Example Calculations of a Medium Speed Compressor Cylinder

PARAMETER	VALUE
Compressor Design	
Cylinder Bore Diameter (inch)	8
Cylinder Rod Diameter (inch)	2
Stroke Length (inch)	7
Clearance Volume	0.1
Piston and Rod Mass (lb)	300
Pipeline Conditions	
Compressor Suction Pressure (PSIG)	600
Compressor Suction Temperature (F)	80
Compression Ratio	1.4
Gas	Methane
Motor Parameters	
Reciprocating Motor Mass (lb)	1000
Stroke Rate (Strokes per Minute)	750
Piston Acceleration/Deceleration Time (Fraction of ½ Stroke Time)	0.3

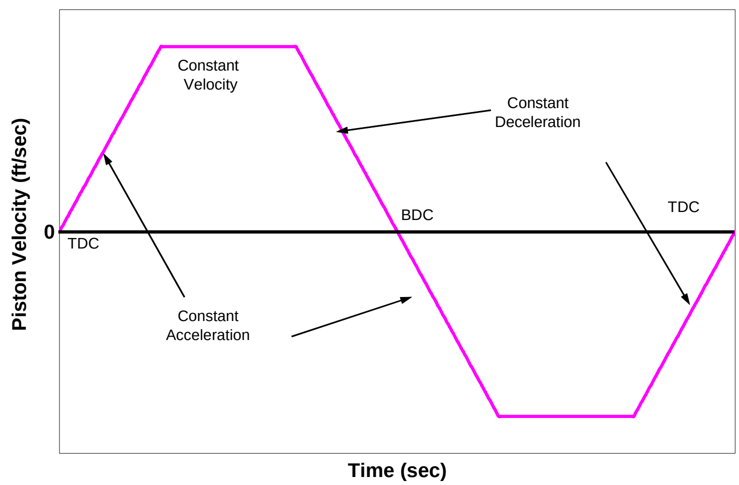

Figure 3-30. Piston Velocity Through One Stroke

The force required by the motor to drive the piston is obtained by summing the forces in Figure 3-31 according to Equation 3.1. The required cylinder drive force time history is shown in Figure 3-32. The required motor force varies from 77,000 lbf to –81,000 lbf over the course of one cycle. The large cylinder drive forces are primarily due to the inertial forces needed to accelerate the reciprocating mass. If the mass acceleration force could be eliminated, the maximum required drive force would be approximately 14,000 lbf, which is the force acting against the gas at discharge pressure. This example calculation highlights the fact that a linear motor driven compressor does not have a crank system to transfer the piston momentum to other pistons or a flywheel. For a linear motor compressor, the motor must supply the acceleration and deceleration forces. A longer piston stroke can reduce the motor force requirements by allowing slower piston acceleration while maintaining the same piston displacement rate. With a longer deceleration distance, the piston momentum can be used to compress gas in the cylinder and reduce the load on the motor.

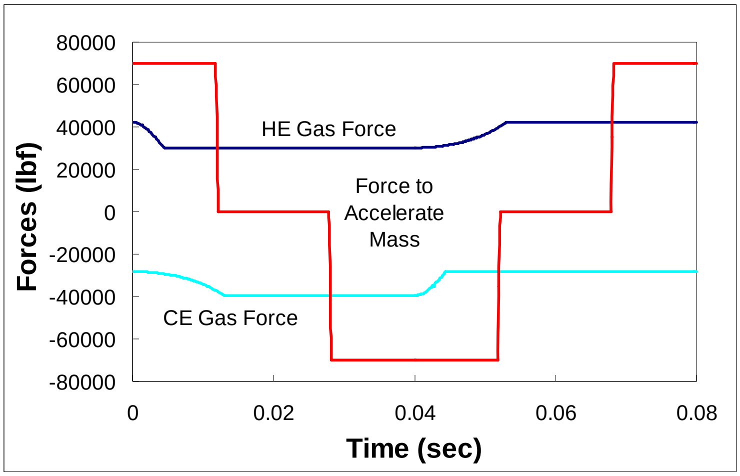

Figure 3-31. Piston Gas Pressure Forces and the Force Needed to Accelerate the Reciprocating Mass

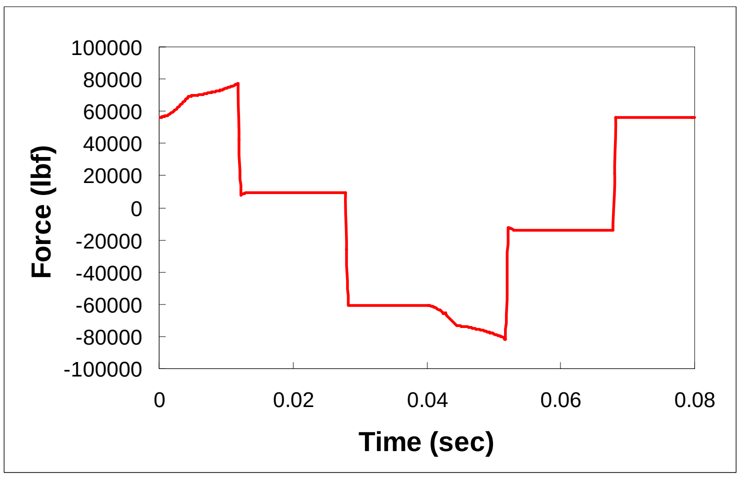

Figure 3-32. Motor Force Required to Move the Piston Through One Complete Stroke

3.7.3 MODEL RESULTS AND PARAMETRIC STUDY

Permutations of model input parameters were used to study the effect of different model values on key compressor design metrics. A parametric study was performed to determine the optimal conditions at which to run a linear motor compressor. Both low-speed and medium-speed compressor cases were selected to use as a basis for linear motor compressor performance. The next few paragraphs present representative data for a low-speed compressor design. It should be noted that all of the data presented in this section assumes the compressor design must achieve the same capacity as the base case.

A key issue in the design of a linear motor compressor is the maximum thrust load the motor must supply. This thrust load must be at least equal to the maximum required rod load of the compressor. Figure 3-33 shows the rod load requirements as a function of stroke length. The rod load was defined as the maximum absolute value of load in either compression or tension.

The plot shows curves for compressor designs with different moving masses of the drive motor. The figure suggests that the required loads are significantly higher for short strokes. Since all of the cases have the same flow capacity, designs with smaller strokes must have higher speed to match this capacity. The high speeds require quick acceleration steps that significantly increase inertial loads at the peak of acceleration. For these small strokes, the inertial loads are the largest contributor to the required rod loads. For longer strokes in which the acceleration is not as steep, the maximum rod load is dominated by gas forces. The gas loads are highest when the piston is acting against the discharge pressure of the cylinder. One way of mitigating the high inertial loads is to change the acceleration profile of the piston. The base case utilized a trapezoidal acceleration profile with 30% of the time from TDC to BDC used for acceleration and 30% for deceleration. The remaining 40% of stroke time is dedicated to constant velocity stroking.

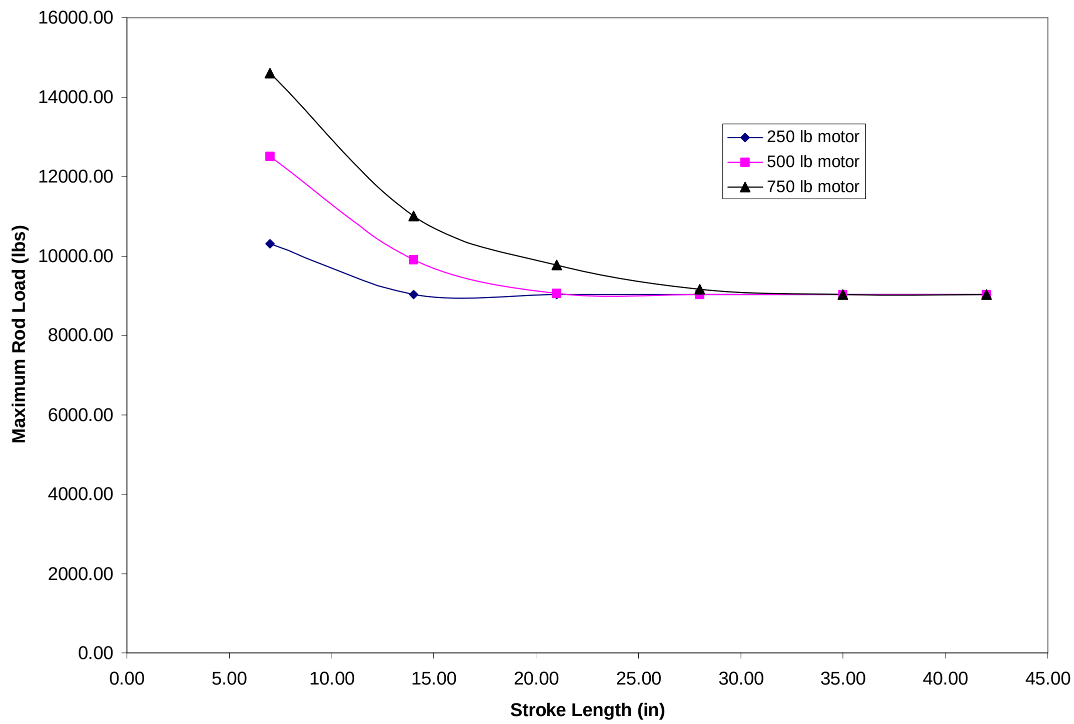

Figure 3-33. Maximum Rod Load as a Function of Stroke Length

The bore diameter also plays a significant role in the performance of a linear motor compressor. Figure 3-34 plots the maximum rod load and peak velocity for a low-speed compressor as a function of bore size for a given set of compressor specifications. The velocity trend given in the plot is expected, as a higher velocity is needed for smaller bore sizes to match capacity. The rod load is high for small bore sizes due to the significant piston acceleration required. Increasing the bore size reduces these inertial loads to the point where the maximum loads are dominated by gas forces. However, the rod load eventually increases with size as a larger bore has more mass to drive.

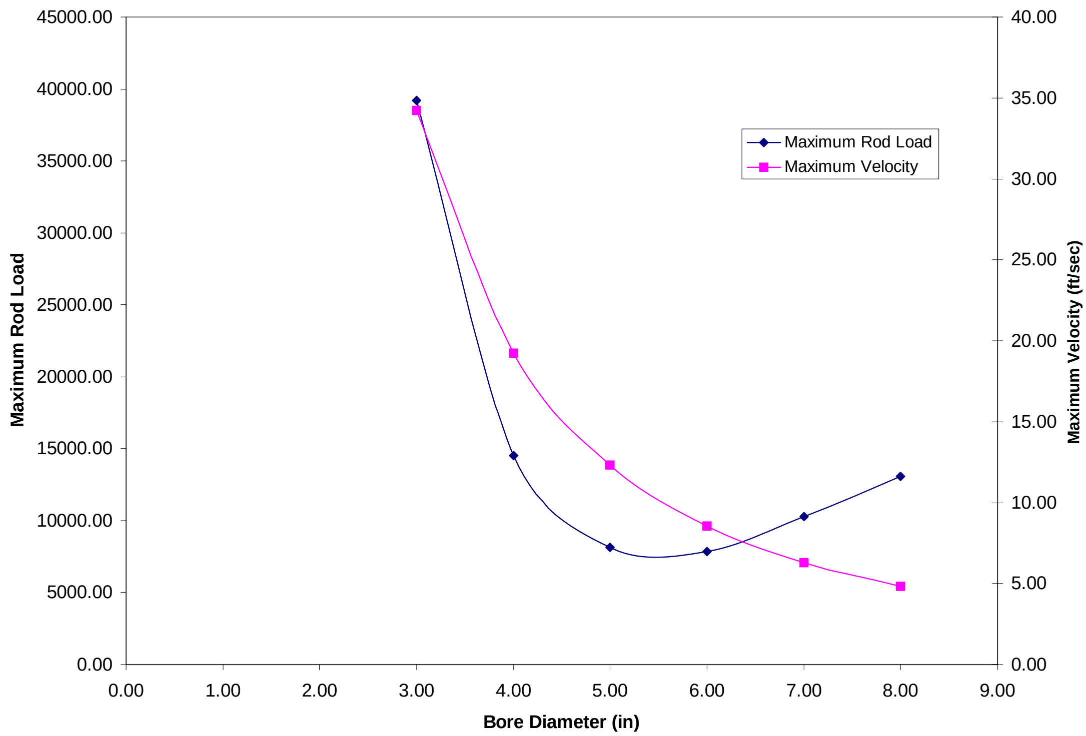

Figure 3-34. Model Parameters as a Function of Bore Diameter

Multiple permutations of model parameters were used to find a best-case design for both a low-speed and a medium-speed compressor. Some model parameters are given in Table 3-10. Plots of the absolute value of rod loads and piston acceleration are given in Figure 3-35. The figure shows the load implications of altering the stroke. A key point in the feasibility of a linear motor compressor is whether or not commercially available electric motors meet the motor requirements. Table 3-11 gives representative data on some commercially available linear electric motors. The table serves as a mechanism to put the compressor requirements in perspective. A low-speed compressor that requires 9,000 lbf of thrust force could likely be constructed using two off-the-shelf motors acting together. Such multiple motor configurations are very common in industrial applications. However, a medium-speed compressor with a load requirement closer to 30,000 lbf seems impractical using existing technology. Such a compressor would likely require a custom motor.

Table 3-10. Selected Compressor Cases

	LOW SPEED	MEDIUM SPEED
Stroke (in)	28	28
Bore (in)	6.5	8
Speed (RPM)	75	188
Peak Force (lbf)	9,500	28,900
Peak Velocity (ft/sec)	8.3	20.8
Average Power (HP)	71	329

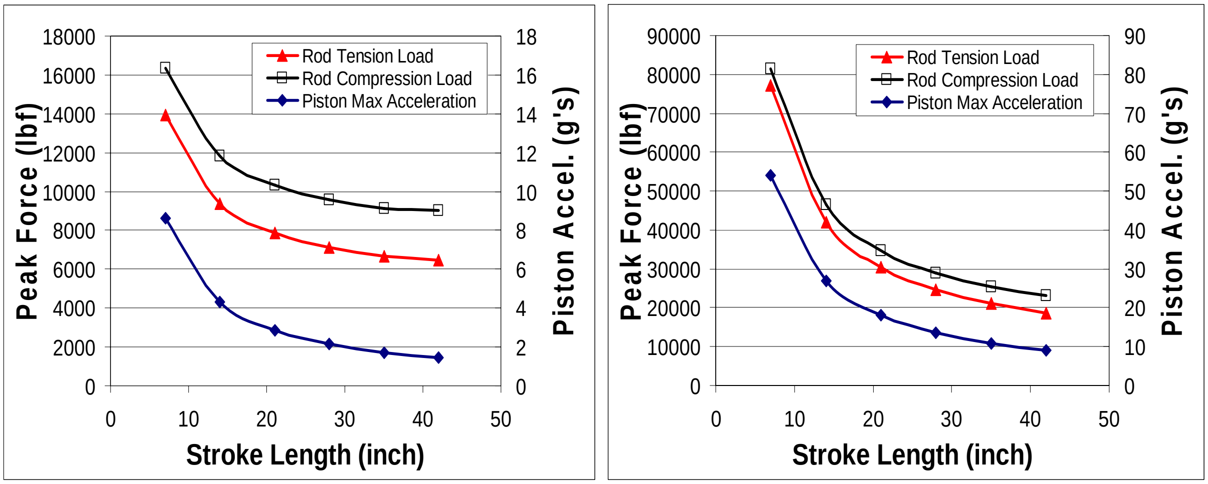

Figure 3-35. Representative Low-Speed (left) and Medium-Speed (right) Compressor Cases

Table 3-11. Commercially Available Motors

COMPANY	MOTOR TYPE	PEAK FORCE (LBF)	MAX VELOCITY AT PEAK (FT/SEC)	MAX CONT. FORCE (LBF)	MAX VELOCITY AT CONT. FORCE (FT/SEC)
Siemens	PM	4,653	6.3	1,821	13.8
Parker	PM	877	-	193	-
Bosh Rexroth	PM	4,541	5.5	1,933	12
California Linear	PM	1,100	1.3	710	2.1
H2W Tech.	LIM	214	-	43	-
“	PM	600	-	200	36.1
Baldor	PM	3,102	-	1,164	26.2
“	LIM	500	-	100	22.5
Anorad	PM	2,094	-	1,272	-
Power Superconductor	LIM	4,721	-	-	150.9
Copley Controls	PM	418	-	66	8.5
Aerotech	PM	1,066	-	267	-

3.7.4 RECOMMENDATION FOR FUTURE WORK

The work conducted in this task demonstrated that a linear motor could be used to drive a reciprocating compressor. Small diameter, low-speed cylinders could be driven with currently available linear motor technology. However, linear motor drives for larger-scale compressors would likely require a custom motor and control system to be designed. Linear motor driven compressors would have the inherit capability to vary the piston stroke over a wide range and, therefore, provide a wide range in the compressor flow capacity. This wide flow range could be accomplished with a fixed stroke frequency so that pulsation problems associated with variable frequency drives are eliminated. Current linear motor technology limits compressor applications to smaller diameter cylinders operating at lower speeds with a relatively long stroke length. Further development work should be conducted to develop linear motor driven compressors for their capability to provide a wide flow capacity range and because they could provide reduced valve pressure losses, lower frictional loss, and increased valve life by reducing the cycle frequency with a longer stroke length. Some specific technical issues to be addressed in future work on linear motor compressors include:

- Low-cost motor control is needed to make linear motor driven compressors attractive. An effort is needed that focuses on control system development that allows for economically feasible designs. For example, it might be possible to control the piston movement by simply turning on and off multiple smaller motors driven by AC power.
- Substantial motor power is required to accelerate and decelerate the piston and motor masses for every stroke. To minimize the linear motor force and power requirements, an equivalent of the flywheel on a rotary compressor should be developed. A mechanical or gas spring at the ends of the piston stroke could be used to reverse the piston direction without requiring additional motor work. The springs could conserve the kinetic energy and would allow a constant piston velocity until it hit the spring. This could increase the flow from the cylinder since the piston acceleration and deceleration times could be reduced, and at the same time reduce the motor force requirement since the motor would not have to accelerate the piston mass.
- Development work is required on the long stroke cylinders that are needed for an efficient linear motor driven compressor. Some of the areas requiring development work include: the design of the long cylinder bore/piston/rod/lubrication system, new valve designs might be considered for the lower reciprocating frequency, guides needed to keep the reciprocating portion of the motor aligned, and the effectiveness of providing some in-cylinder gas cooling should be investigated.
- Pressure pulsations control systems for linear motor driven compressors with long stroke should be analyzed to determine if conventional pulsation control methods would be effective. Non-traditional pulsation control methods for the long stroke compressors should also be investigated. Operating scenarios that could minimize or eliminate pressure pulsation problems should be investigated. The low cycle frequency and the long steady outflow and inflow cycles may allow pulsation levels to be minimized by timing multiple cylinder cycles. Variations in the piston acceleration and deceleration rates at the ends of the strokes should also be studied to determine how they could be varied to limit pressure pulsations and flow variations.

4. COMPRESSOR VALVE TECHNOLOGY RESULTS

4.1 COMPRESSOR VALVE TECHNOLOGY ASSESSMENT

Reciprocating gas compressors have traditionally used passive valves to control the suction and discharge processes in the compressor. With the immergence of larger machines operating over wider speed ranges and the continuing need to compress gas more efficiently at a lower cost, the gas industry has to consider improvements in valve technology. In the current state-of-the-art review, passive valve designs will be reviewed as well as some of the recent advancements from compressor valve manufacturers.

Pipeline companies are concerned with compressor valve reliability because studies have shown that valve repairs and associated downtime are a primary maintenance and operational expenditure [Foreman, 2002][10]. Valve failures are cited as the most common cause of unscheduled compressor shutdowns. The efficiency of a reciprocating compressor also strongly depends on the performance of its suction and discharge valves. As Figure 4-1 shows, a typical PV diagram for a reciprocating compressor varies from the ideal behavior based on valve losses.

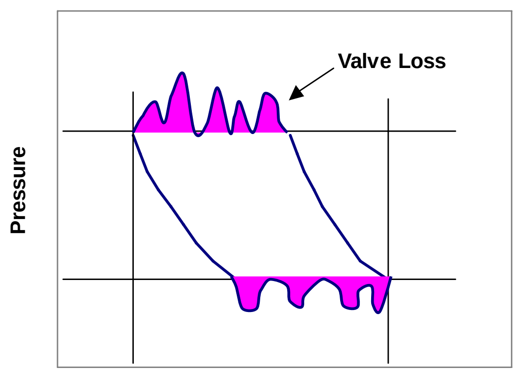

Figure 4-1. Typical Compressor PV Diagram with Valve Losses Shown as Deviations from Ideal [Foreman, 2004][1]

Since valve performance is critical to compressor efficiency and reliability, pipeline companies have found it advantageous to invest in studies to determine which of the existing passive valve models perform most reliably. In a 2003 study, engineers at El Paso Corporation stated that a reciprocating gas compressor's efficiency is linked to the performance of the intake and discharge valves. The study investigated six different compressor valves from five manufacturers in order to determine valve efficiency, reliability, and cost of operation [Noall and Couch, 2003][11]. The six-month investigation found that valve efficiencies were directly related to compressor ratios, where a decline in efficiency was attributed to valve leakage at low compression ratios. This finding shows the critical problem in operating the traditional passive valve at non-ideal (off-design) operating conditions. In addition, this study found that the losses through suction valves were approximately twice as great as the discharge valve losses [Noall and Couch, 2003][11].

An extended study of valve performance (beyond the six-month period) would expand the data on valve reliability and compliment the existing operational valve data from the El Paso study. The ARCT program offers the opportunity to investigate the traditional passive valve design and improve upon the traditional reciprocating compressor valve, based on the weaknesses in the design. The ARCT program is motivated by the industry's need to make gas reciprocating compressor operation more reliable and efficient, recognizing the significant influence of valve performance on compressor operation. By improving valve performance, the ARCT program will enable valve operation to be fine-tuned and operational expenditures to be more foreseeable, resulting in less downtime.

4.1.1 CONVENTIONAL PASSIVE VALVE DESIGN

In a typical passive valve design, the control mechanics of the valve does not vary, regardless of the compressor operation point as valves utilize internal springs with fixed stiffness and dampening to control the motion of the valve element. The valve is held in its closed position by the internal spring force. When the gas pressure force exceeds the spring force on the valve, the valve opens as a passive response to the force distribution acting on the valve. The suction valve will open when the suction pressure exceeds the internal cylinder pressure. The discharge valve will open when the cylinder pressure exceeds the discharge pressure on the outlet side of the valve. In both cases, the valve element movement is controlled solely by the gas pressure overcoming the constant spring force.

The motion of the valve element is illustrated in Figure 4-2 in a standard poppet process valve. The valve spring moves the poppet element against the seat. The gas pressure on the opposite side of the valve poppet compresses the valve springs, and the process gas is allowed to pass through the valve in this process. Other passive valve designs work under the same mechanical principle, where the spring force is used to control the mechanical motion of the valve element. Though the design is relatively simple, the valve timing and lift cannot be varied; these are essential constants based on the particular passive valve design.

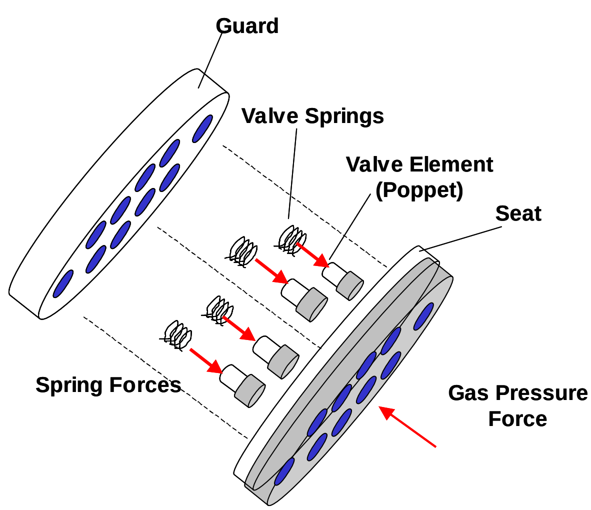

Figure 4-2. Typical Poppet Passive Valve Design

The passive valve design and materials can be altered in order to make the compressor more efficient through reducing leakage and allowing for smoother gas passage through the valve. These design variables also affect the valve reliability and susceptibility to dirt and particle accumulation. Desirable characteristics in the passive valve design are good sealing, rapid opening, sustained flow area, rapid closing, reduced impact force, and low flow resistance.

However, the design of these passive valves inherently relies upon a constant valve behavior (constant valve actuation, movement, and compression volume), which is usually not the case in the valve operation, given the wide operating range and profile requirement of gas compressors. Valve losses associated with flutter, late or early closure, and imperfect sealing can lead to increased inefficiency of the compressor. Clearly, adaptive valve behavior through active valve control would be more beneficial at preventing valve failure and improving compressor efficiency.

4.1.2 SOURCES OF VALVE FAILURE

Of the four essential valve components, the valve guard and valve seat are not viewed as major considerations for improving valve reliability [Woollatt, 2003][12]. Valve seats may sometimes fail in the event a liquid slug enters the compressor. The moving element is more of a consideration for valve reliability, where the typical failure modes result from repeated impacts, varied differential pressure, and corrosion. The valve springs are the most common cause of failure in modern valves according to one major compressor manufacturer, Dresser-Rand [Woollatt, 2003][12]. The spring material must be resistant to corrosives. In addition, the spring motion is controlled by the differential pressure force on the valve, which results in a non-ideal pattern of rapid opening and closing of the valve because of the passive valve response. Because springs fail most often due to the valve forces, a new valve design that actively or semi-actively controls the valve element motion will improve valve reliability considerably.

The majority of the environmental/operational effects of valve failure are due to the components in the compressed natural gas. Some of these effects can be controlled by filtering and gas separators placed upstream of the compressor. However, the inherent corrosiveness of the gas is not easily prevented. A valve's material (especially the valve element and springs) must be able to withstand the corrosive elements of the natural gas stream, to a certain extent.

Another environmental/operational factor to consider in valve life is oil sticktion. Improper lubrication of the valves can cause the valve to stick, which delays the natural passive valve response, for which the valve was designed. Oil sticktion was seen as one of the primary causes in a case study of a reciprocating compressor valve failure in a Belgian chemical plant [Chaykosky, 2002][13]. In the case study, an adhesive agent was added to the lubricating oil in the third stage cylinder to ensure better operation. As a result, the additive caused the oil to stick to the valve. Improper lubrication of the valves most often results because the compressor environment is not well known and the wrong oil/additive is chosen. Sticktion inhibits the valve from opening properly and prevents the valve from closing properly. When the valve finally opens (or closes) and overcomes the sticktion, the valve responds by slamming into its final position; the excessive force on the valve results in a high impact force on the valve element against the guard or seat [Hoerbiger Bulletin][14].

The mechanical causes of valve failure can be controlled through a good passive valve design, but they are not completely preventable for all applications. A good design to a passive valve will ensure good opening and closing motion of the valve. Valve lift and valve spring design are two of the primary design considerations affecting mechanical failure. The proper lift must be chosen to ensure good flow area through the valve. However, there is an upper limit beyond which no more flow area can be obtained for a particular valve design. High valve lift also corresponds with high impact forces, which can accelerate valve failure. Low valve lift ensures longer life, but at the price of lower compressor efficiency. The gas molecular weight is also a consideration in designing the valve lift. Figure 4-3 shows a correlation between valve lift and suction valve life based on a Dresser-Rand survey of 36 cylinders used for hydrogen service [Woollatt, 2003][12].

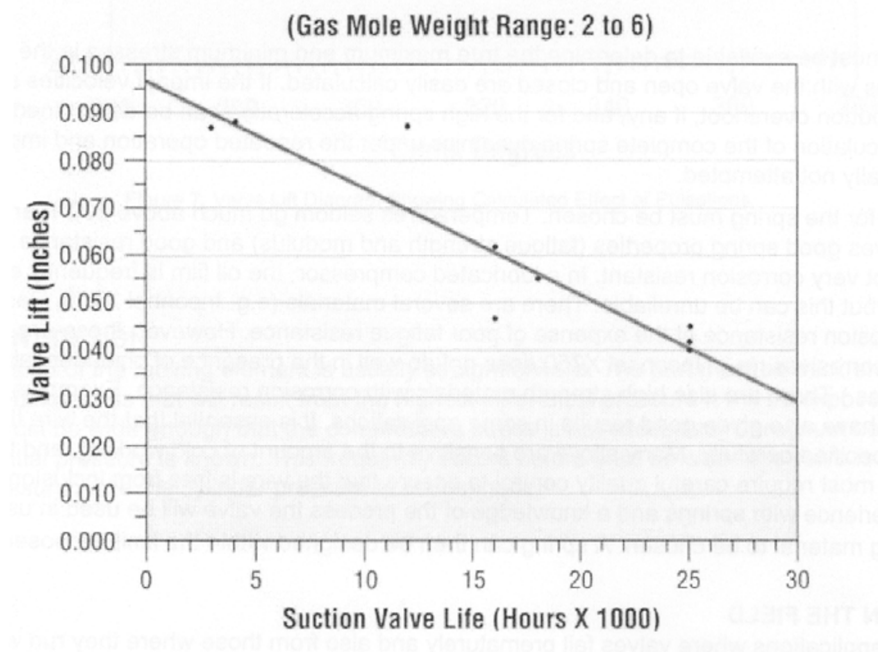

Figure 4-3. Dresser-Rand Survey of Suction Valve Life for Hydrogen Service

Valve springs should be chosen carefully in a valve design, since these can also lead to a mechanical failure. Valve flutter can be reduced by lowering spring stiffness, but this should be weighed against the valve spring's ability to close the valve fully [Chaykosky, 2002][13]. Dresser-Rand recommends using a stiffer spring to avoid the possibility of sticktion causing late valve opening [Woollatt, 2003][12].

Another mechanical cause of valve failure is the lateral motion of the valve element. Ideally, the valve element should move perpendicular to the guard and seat. In reality, the valve plate or ring moves non-uniformly, which causes certain parts to impact the seat first. This often causes the plate or ring to fracture prematurely. A good design will take into account the non-uniformity of the initial impact force on the valve [Chaykosky, 2002][13]. However, one of the weaknesses in the current passive valve design is that the valve element motion cannot be controlled completely, making the impact force and its contact area a constant source of valve susceptibility.

Finally, in designing a passive valve to work reliably without failure, a range of operating conditions is assumed. The design, however, is optimized for a single design point (a single set of operating pressures, temperatures, speed, cylinder clearance, gas molecular weight, lubrication rate, and pulsation level). When the operating conditions significantly deviate from the ideal case, the valve reliability is not necessarily ensured [Chaykosky, 2002][13]. With the recent increase in speed range of compressors in the reciprocating compressor market, the off-design operation of valves is more evident, causing more failures in passive valves. The inability of the current passive valve designs to adapt to a range of operating conditions and the cost of valve failures illustrates the need for an improved compressor valve design.

4.1.3 CURRENT VALVE DESIGNS USED IN RECIPROCATING COMPRESSORS

The current valve designs used in today's reciprocating compressors can be classified as one of three standard types—plate valves, ring valves, or poppet valves. Each of these valve types utilizes a different valve element geometry that is acted on by internal springs. The spring force and its interaction with the valve element drives the passive response of the valve element, as well as the reliability and efficiency of the valve. Table 4-1 shows a comparison of the different valve designs offered by the major compressor valve manufacturers.

Table 4-1. Comparison of Current Reciprocating Compressor Valve Designs

PLATE VALVE DESIGN	POPPET VALVE DESIGN	RING VALVE DESIGN
1. Dresser-Rand – PF Valve	1. Cook Manley – Moppet® Valve	1. Cook Manley – Manley® Valve
2. Hoerbiger – CS Valve, CT Valve, HDS Valve, R valve	2. Hoerbiger – Standard and Mini Poppet Valve	2. Hoerbiger – CX, CE Valve
	3. Dresser-Rand – Poppet Valve	3. Compressor Products International (CPI) – CPI Valve
	4. Dresser-Rand – Magnum Valve *(Similar to poppet valve design, but "bullets" are used instead of poppets.)*	
	5. Cooper Compression – Texcentric High Efficiency and Damped Plate Poppet	

Since compressor valve repair costs and replacement of valves are the largest contributors to compressor maintenance costs, many of the end users and valve manufacturers have examined ways to make valves more reliable. Changes in the design geometry and flow passage through the valve, as well as new materials, have allowed the standard types of passive valves to become more reliable. In addition, many of the valve manufacturers have found ways to make valve parts interchangeable through fixed geometries or interchangeable cartridges, regardless of compressor and valve size.

The standard plate valve uses a movable circular plate as the valve element. The movable plate seals against the upper or lower portion of the valve when the spring force is acting to close the valve. When the gas pressure force overcomes the spring force of the valve, the plate element moves, allowing gas to pass through the grooves in the plate. In this passive valve design, the particular pattern inscribed on the plate is viewed as a design characteristic.

Dresser-Rand and Hoerbiger have developed valves that can be categorized as a plate valve design. Dresser-Rand manufactures the PF valve that uses a patented scalloped plate with PEEK material. The plate used in the PF valve is not truly circular, which is attributed to plate strength and helps the plate withstand high impact forces. The PF valve is shown with a cross-sectional view in Figure 4-4. Hoerbiger manufactures the CS valve, the CT valve, the HDS valve and the R valve, all of which fall in the plate valve design. Two of the Hoerbiger valve designs are shown in Figure 4-5.

Both the Dresser-Rand and the Hoerbiger plate valve designs use a series of small springs to control the plate element motion. The springs lie near the external diameter of the plate. The plate and spring materials are viewed as design variables that can lead to higher valve reliability. Figure 4-5 shows the breakout view of a typical Hoerbiger CT valve, with its small internal springs and series of plates.

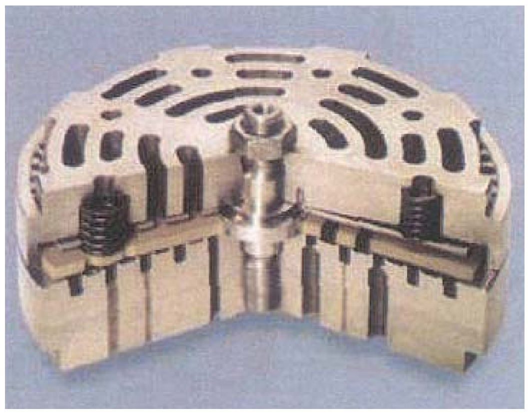

Figure 4-4. PF Plate Valve Made by Dresser-Rand [Dresser-Rand Valve Information, 2004][15]

Figure 4-5. Hoerbiger CT Valve – Assembled View (on left) and Expanded View (on right) [Hoerbiger Valve Information, 2004][16]

The poppet valve design uses individual spring-loaded cartridges held between two ported plates. The hole pattern on the plates varies with the design, as well as the design of the individual poppets. Different manufacturers use different types of material to create the poppets. When the gas pressure force acts on the plate, the individual poppet springs are compressed and the gas is allowed to flow through the poppet valve. Dresser-Rand has designed a standard poppet valve as well as the Poppet GT valve, which is used in gas transmission applications. These two valves are shown in Figure 4-6.

The Moppet® compressor valve made by Cook Manley is an example of the advantages of interchangeable parts. This valve acts similar to a standard poppet valve, except that each disc hole contains a radiused disc valve. The disc valves are made of thermoplastic discs containing stainless-steel-alloy, modular cartridges. The gas is allowed to flow through a disc in the center of each cartridge. Figure 4-7 shows an example of the Moppet compressor valve and the interchangeable disc [Cook Manley Valve Information, 2004][17].

Figure 4-6. Dresser-Rand Poppet Valve Designs – Standard Poppet Design Shown on Left, Poppet GT Valve Design Shown on Right [Dresser-Rand Valve Information, 2004][15]

Figure 4-7. Cook Manley Moppet® Valve Design [Cook Manley Valve Information, 2004][17]

A ring valve design is similar to the plate valve concept in which the valve element fits into the grooves in the bottom seating plate. The common ring valve designs in today's market are marketed as radiused disc valves. Each radiused disc or ring moves under the control of the valve springs. The valve consists of concentric discs mounted on a plate. Similar to a standard ring valve design, the discs (or rings) are designed to rise up from the bottom plate to allow the gas to flow under and around each ring, through the holes on the upper plate. The discs are often contoured in order to reduce the impact force on the disc hitting the seating plate, which can lead to valve failure. One example of a ring valve design is shown in Figure 4-8. The CPI ring valves vary the thermoplastic material of the radiused discs and the spring type to match the requirements of the compressor.

Cook Manley manufactures a version of a ring type valve. The Manley® compressor valve was first introduced into the compressor market in 1971. The Manley valve is different from a typical ring valve because the discs are contoured to fit within contoured grooves in the bottom plate, like the CPI valve. The valve is shown in Figure 4-9. The Manley valve design prevents the accumulation of debris and allows for more aerodynamic gas flow. Figure 4-10 shows a depiction of the gas flow pattern through the Manley valve and the contoured grooves. The contoured grooves in both the CPI and the Manley valve help to prevent dirt accumulation in the seating pockets in the valve plate. In addition to the geometrical design change through the contoured discs, the manufacturer uses thermoplastic for the disc material to provide better valve sealing [Cook Manley Valve Information, 2004][17].

Figure 4-8. CPI Valve Design [Compressor Products Valve Information, 2004][18]

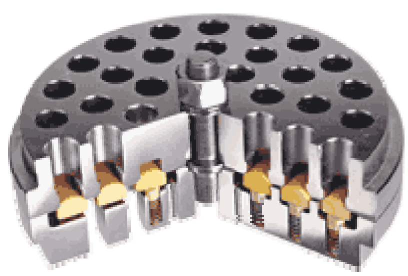

Figure 4-9. Manley® Valve Design [Cook Manley Valve Information, 2004][17]

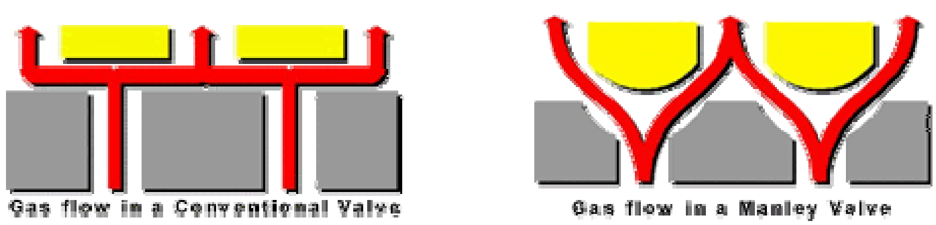

Figure 4-10. Gas Flow Pattern in a Conventional Ring Valve Compared to Flow Around Contoured Discs in Manley® Valve [Cook Manley Valve Information, 2004][17]

The last example of the passive valve designs in the current compressor valve market is the Dresser-Rand Magnum valve, shown in Figure 4-11. This valve is analogous to a poppet valve, where individual elements act against a perforated plate. Instead of poppets, the Magnum valve uses individual bullet shaped elements. Dresser-Rand claims the bullet design offers an advantage over the poppet element, by improving the impact stresses on the valve elements.

Figure 4-11. Dresser-Rand Magnum Valve Design [Dresser-Rand Valve Information, 2004][15]

4.1.4 CURRENT ACTIVE VALVE DESIGNS FOR RECIPROCATING COMPRESSORS

The lack of durability and low efficiency of the passive valve design demonstrates the need to control the valve motion actively—to adjust the valve motion in order to fine tune the valve performance and element motion to control the suction or discharge process in reciprocating compressors. Several active valve concepts have also been investigated, specific to the compressor valve arena. These concepts will be reviewed in this section, in order to understand the advantages of the various active valve concepts presently available.

The Hoerbiger Hydrocom is a fully active valve currently in use in reciprocating compressors. The Hydrocom valve picture from the Hoerbiger website is shown in Figure 4-12 [Hoerbiger Valve Information, 2004][16]. The Hydrcocom uses an electrically controlled hydraulic actuation system. The hydraulic actuator is actuated by a quick response solenoid. The actuator is used to control a standard ring valve or plate valve. The Hydrocom design allows the suction valve to remain open during the compression stroke, which broadens the operating range of the process by allowing only the required amount of gas to be compressed.

Figure 4-12. Hydrocom Valve Design Made by Hoerbiger (Electrohydraulic Actuation) [Hoerbiger Valve Information, 2004][16]

The second example of active valve control currently available for reciprocating compressor valves is the Dresser-Rand Infinite Step Control (ISC) system. This system controls a standard D-R suction valve motion through an electronically actuated hydraulic subsystem. In principle, the system is very similar to the Hoerbiger Hydrocom. The ISC uses an electronic controller to vary the pressurization of the hydraulic valve actuators according to the compressor capacity set point. The high-pressure hydraulic fluid drives the valve element motion, using a piston and series of plungers, as shown in the depiction of the valve design in Figure 4-13 [Dresser-Rand Valve Information, 2004][15]. This active variation of the valve motion allows accurate variation in the timing of the hydraulic signal, which is linked to the compressor capacity set point.

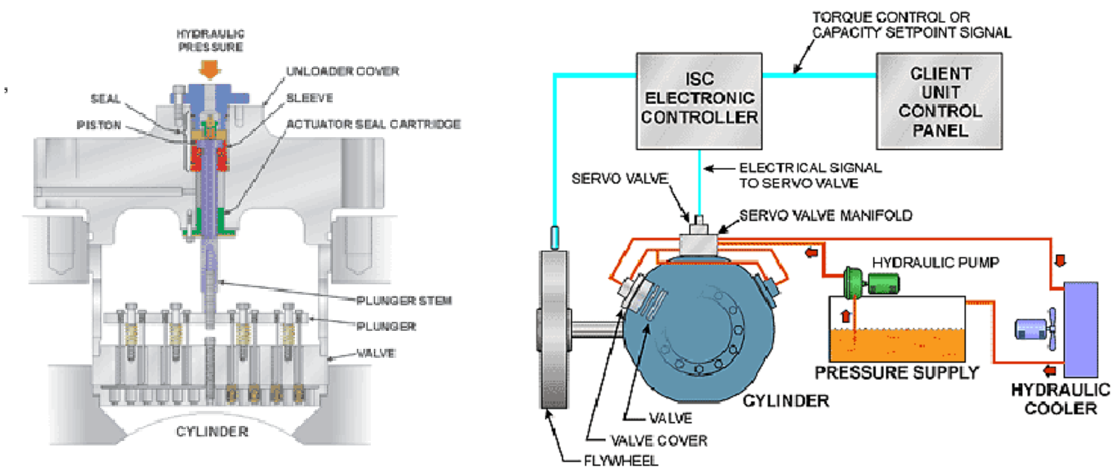

Figure 4-13. Dresser-Rand Infinite Step Control (ISC) System for D-R Suction Valves [Dresser-Rand Valve Information, 2004][15]

A concept for an actively controlled rotary valve design was presented by the Cozzani Company in 2001 at the European Forum for Reciprocating Compressors. The Cozzani rotary valve is driven by an electronic driver, which rotates the valve shutters. The rotational movement eliminates the impact forces that traditional passive valve designs encounter, when the valve element moves between the seat and the guard [Bianchi and Schiavone, 2001][19]. The rotary valve is pictured in Figure 4-14. The valve also maintains a simplified geometry, which allows for good efficiency in the gas flow through the valve.

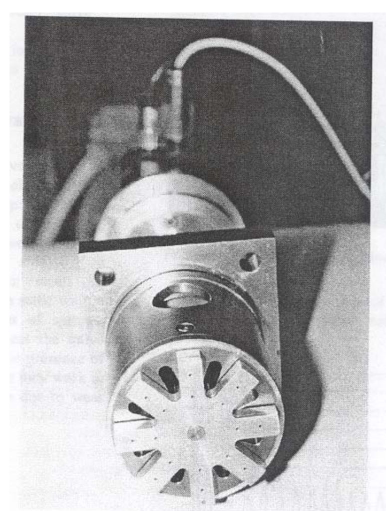

Figure 4-14. Cozzani Company Rotary Valve Design

4.1.5 Background Summary

The traditional passive valves used in reciprocating compressors have a significant effect on compressor efficiency. Valve life is often limited to less than twelve months because of the continual impact forces in the intake and discharge process.

Furthermore, replacement of valves is the most significant maintenance cost for many reciprocating compressors. The following summary conclusions concerning the traditional passive valve design can be made from the state-of-the-art review:

- Valve failures can primarily be attributed to high cycle fatigue, sticktion, and the accumulation of dirt and debris, although improper lubrication and liquid slugs can also be a cause of failure.
- Of the three passive valve types, poppet valves tend to be the most reliable but least efficient. Ring valves are more reliable than plate valves but have a lower efficiency.
- Passive valves are designed for an optimal operation point; hence, valve operation is impaired when the operating conditions deviate significantly from the design point.
- In the traditional passive valve design, an increase in valve life (reliability) directly relates to a decrease in valve efficiency. This converse relationship is due to an increase in valve lift (and flow-through area) being limited by the corresponding increase in the critical valve impact force.
- Above a certain impact velocity, valve plate failure is attributable to plastic deformation of the valve springs, which fail to provide adequate damping for the plate. The design of the valve springs is a major weakness in the traditional passive valves currently used in industry.
- The lack of durability and low efficiency of the passive valve design demonstrates the need to control the valve motion actively.

4.2 CONVENTIONAL VALVE LIFE IMPROVEMENT PROGRAM

4.2.1 BACKGROUND

In designing a reciprocating compressor valve, desirable functional attributes include good sealing, rapid opening and closing, sustained high flow area (when open), minimum bouncing upon impact, tolerance of impact forces and maximum temperatures, and low flow resistance. Proper design choices, such as material, mass, spring constant, lift and flow area, will maximize the successfulness of the design. However, simple, passive valves do not tolerate wide operating ranges well. The challenge to the current research program is to achieve longer valve life with low loss and acceptable valve operation (including leak potential). Design improvements often come at a price because low flow resistance, which adds to longer life, conversely leads to excessive impact forces. Valve manufacturers have made many advances in materials and configuration. Yet, design trade-offs, parameter selection, and operation of valves are often mismanaged because applications engineering tools are not readily available.

Valve failures can be divided into two major categories: Environmental and Mechanical[14]. Environmental causes are those elements in the valve environment that can lead to valve failure, such as corrosive contaminants, foreign material, liquid slugs, or improper lubrication. Environmental failures can sometimes be prevented by the proper choice of valve material and conditioning of the gas stream (filtration, separation, etc.). Mechanical causes are valve failures that result from high cycle fatigue and abnormal mechanical motion of the valve, caused by high valve lift, valve operation at off-design conditions, valve flutter, pulsations, or spring failure. As impact velocities increase due to higher valve lift or valve operation at off-design conditions, the velocities cause excessive impact stresses and an accelerated damage rate to the valve. Some mechanical causes can be controlled with a good design of the valve components (guard, seat, moving element, and springs), although a valve's design is usually optimized for a single design point based on the fixed mass, stiffness, and damping of the valve plate and spring.

Perhaps the most common cause of plate or spring failure is valve operation at off-design conditions. In designing a valve to work reliably without failure, a range of operating conditions is assumed. The design, however, is optimized for a single set of operating pressures, temperatures, speed, cylinder clearance, gas molecular weight, lubrication rate, and pulsation level. When the operating conditions significantly deviate from the ideal case, the valve design reliability is not necessarily ensured[4]. With the recent increased need for variable speed compressors in the reciprocating compressor market, the off-design operation of valves is more evident, seen by more failures in valves.

The efficiency of a reciprocating compressor also strongly depends on the performance of its suction and discharge valves. In high-speed units (600 RPM or more), valve performance is aggravated further by the increased frequency of valve impacts, higher impact velocities, and reduced flow areas. In general, the ideal compression cycle of a reciprocating compressor is affected directly by valve losses, which lower the adiabatic efficiency of the machine[20]. Valve performance is critical to both compressor efficiency and reliability. In turn, compressor efficiency has a direct impact on capacity when the driver is operating at its maximum power.

Several past studies have been conducted to understand valve failures and prolong valve life in order to improve compressor performance. The El Paso study, mentioned in the previous section, found that the losses through suction valves were approximately twice as great as the discharge valve losses[11]. Figure 4-15 shows an example of a PV trace from a typical compressor with a pressure ratio of 1.4, tested in the El Paso study. The shaded gray area between the cylinder pressure trace and the suction and the discharge nozzle traces can be interpreted as the lost work required to pump gas into and out of the cylinder against the valve resistance.

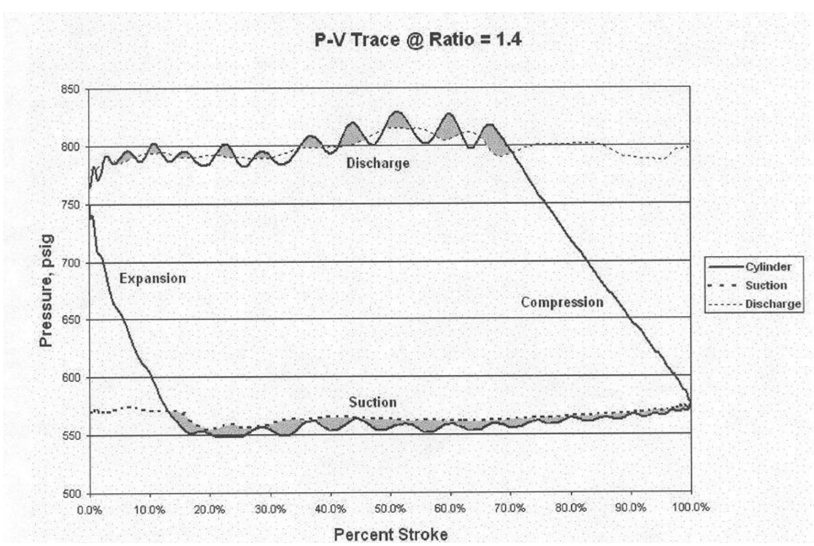

Figure 4-15. Typical PV Chart Illustrating Valve Horsepower Losses

In 2001-2002, a GMRC program examined valve stresses in order to predict valve life. Southwest Research Institute® engineers tested a series of valves at the GMRC Reciprocating Compressor Test Facility (RCTF) and the Hoerbiger valve slapper facility. This program was initiated because of the need for improving analysis tools in valve application engineering[21]. Figure 4-16 shows the test setup at the Hoerbiger valve slapper facility. The program provided a limited set of data and yielded insight as to the effort required for future research. The research showed the need to validate a dynamic model of valve motion against actual test data. This previous GMRC research program provided a starting point for determining valve stress and modeling the valve's physical behavior.

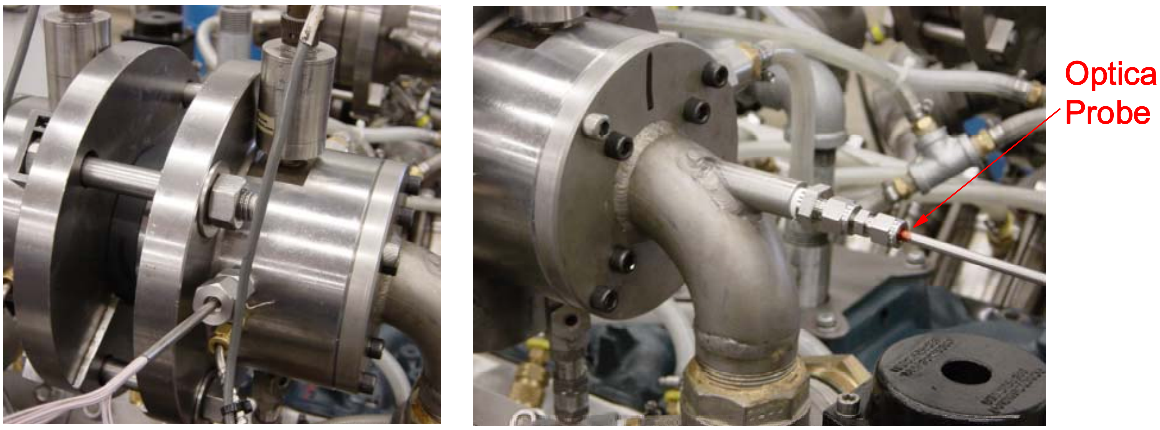

Figure 4-16. Previous GMRC Valve Testing at Hoerbiger Valve Slapper Facility

Beyond the current valve designs, a better tool is needed to predict valve life based on compressor operating profile, geometry, material, and valve type. This discussion describes the ARCT GMRC valve research program, which aims to develop such a tool for valve behavior and performance. This program will enhance the understanding of valve motion and the consequent stresses created by the valve's operating behavior. Namely, through the integration of experimentally validated FE analysis of transient plate stresses, a probabilistic valve plate motion model based on real measured field data, and accurate plate material properties, a predictive tool for reciprocating compressor valve plate life can be developed. In combination with performance predictions, the predictive tool will support enhanced applications engineering for reciprocating compressor valves.

4.2.2 RECIPROCATING COMPRESSOR VALVE PROGRAM

The objective of this research is to better understand the factors affecting reciprocating compressor valve performance. In meeting this objective, the research aims to develop a tool for predicting reciprocating compressor performance together with valve plate life. Valve life is considered a function of the plate cyclic 3-D kinematics and material properties. The program investigates these two fundamental aspects of reciprocating compressor valves in developing a method to predict valve life. The benefit of the research is primarily to increase compressor efficiency and reduce downtime by allowing the user to optimize machine performance and valve life for a desired operating profile and valve material.

The components of the reciprocating valve program are outlined in Figure 4-17. In order to model the motion of a valve plate and the compressor performance, the compressor geometry, operating conditions, and valve parameters must be known. Although these parameters dominate the requirements for predicting valve motion and compressor performance, the flow resistance and pulsations in the piping are contributory factors. The influence of these conditions should be evaluated by measuring the motion of the plate under varied operating conditions (*Program Element 1)*. The results of the first round of *Program Element 1* testing will be discussed briefly for one set of operating conditions. The results provided an average impact velocity, angle and location for the opening and closing events. Future testing will examine other valve designs (varied plate designs, spring stiffness, etc.) and operating conditions (pressure ratio, speed, etc.).

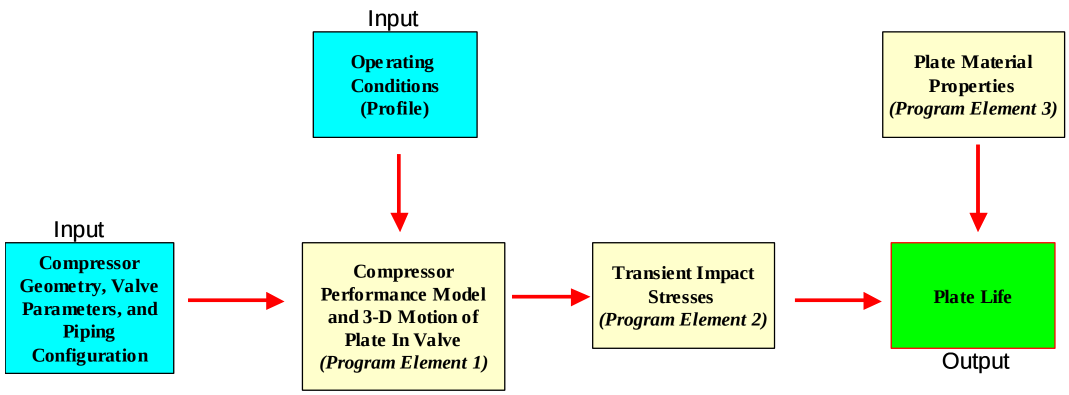

Figure 4-17. Roadmap for Reciprocating Compressor Valve Research

Characterizing the plate motion in three dimensions significantly differs from one-dimensional models of reciprocating compressor valve motion that the state of the art currently offers. Most of these one-dimensional models assume the plate to lift off the seat and move towards the guard without any angularity to its movement. In reality, the plate moves toward the guard non-uniformly, with a significant level of angularity. The results of this research clearly show that the assumption of purely translatory motion of the plate is rarely valid. The precise motion of the plate is not yet sufficiently understood in order to use angular motion in deterministic valve motion predictions. This research aims to provide a more detailed understanding of the plate movement, in translation from the seat to the guard, with realistic angularity of the plate.

The 3-D motion of the plate can be used as an input to a finite element (FE) model that predicts plate transient impact stress as a function of impact velocity, angle of impact and location (*Program Element 2)*. However, the FE model must be validated against real data in order to calibrate and verify the model. To validate the model, controlled single impact tests were completed using a burst-membrane shock-tube. These tests provided useful characterizations of the kinematic behavior of the plate. The single impact tests were recorded to determine plate impact velocity, angle and location. In addition, strain gauges provided strain measurements on the plate that correlated with the plate position recorded by displacement probes. Using the results from the single impact testing, the FE model was validated. The results of this program element will be discussed in detail.

Finally, a materials analysis of the plate material can be used in combination with a predictive model of plate stress to predict valve plate life (*Program Element 3*). The materials analysis is currently being conducted on PEEK material at various temperatures to characterize the fatigue behavior of PEEK. The results show conservative agreement with the manufacturer data. The fatigue testing will generate stress versus life data (SN curves) that can be tied to model estimations of plate impact stress to predict plate life.

4.2.3 DEVELOPMENT OF MODEL TO PREDICT VALVE IMPACT STRESSES

One critical aspect of predicting valve plate life is linking the realistic physical behavior of the valve plate to a validated stress model. Obtaining useable results from compressor test data to validate the model can be challenging because of the high frequency of impacts and the complex non-uniform motion of the plate. To understand the plate kinematics and validate the model, single impact testing was performed using a burst-membrane shock-tube. This testing allowed a realistic single impact event to be analyzed in detail without additional complicating valve motion characteristic of a typical reciprocating compressor.

A 12-inch diameter burst-membrane shock-tube was constructed, as shown in Figure 4-18. A typical reciprocating compressor valve was mounted 4.0 meters downstream of the shock-tube. The valve was instrumented with optical position probes (in order to measure displacement of the valve plate) and strain gauges (in order to measure strain on the plate). The instrumented valve is shown on the right in Figure 4-18. Three precision strain gauges (manufactured by Micro-Measurements with 120-ohm resistance) were mounted on the valve plate, which was affixed with reflective coating to optimize the available light for the optical probe measurement. The valve plate is shown in Figure 4-19. The high-speed optical probes were located on the outer diameter of the valve. This test setup provided detailed three-dimensional motion data and plate strain results.

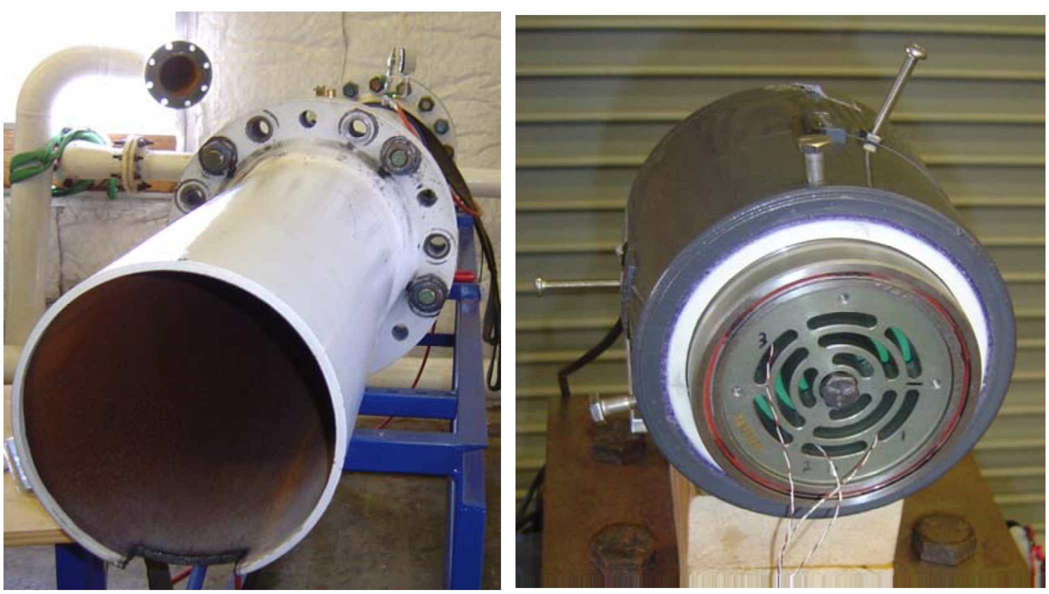

Figure 4-18. Burst-Membrane Shock-Tube (Left) and Instrumented Valve Fixture (Right)

Figure 4-19. Valve Plate with Reflective Coating and Mounted Strain Gauges

The burst-membrane shock-tube testing was triggered with a plastic membrane and nichrome heat wire mounted to the membrane surface. Once the membrane burst, a normal shock traveled downstream and hit the valve assembly. The valve test fixture measured the movement of the valve plate as it impacted the guard and bounced multiple times after the shock. No springs were used in the valve in order to understand the plate motion completely without the influence of the spring elements. For each test, the data acquisition period began when the membrane burst. The data acquisition rate was approximately 10 kHz on six separate channels, which allowed sufficient sampling of the three high-speed optical position probes and the three strain gauges.

Four single impact tests were performed to capture a range of impact locations, angles, and velocities. The single impact profile matched previous plate profiles obtained in the Hoerbiger valve slapper. Figure 4-20 compares the two motion profiles. As the figure shows, the motion profile from single impact testing correlated with the multiple impacts recorded in the

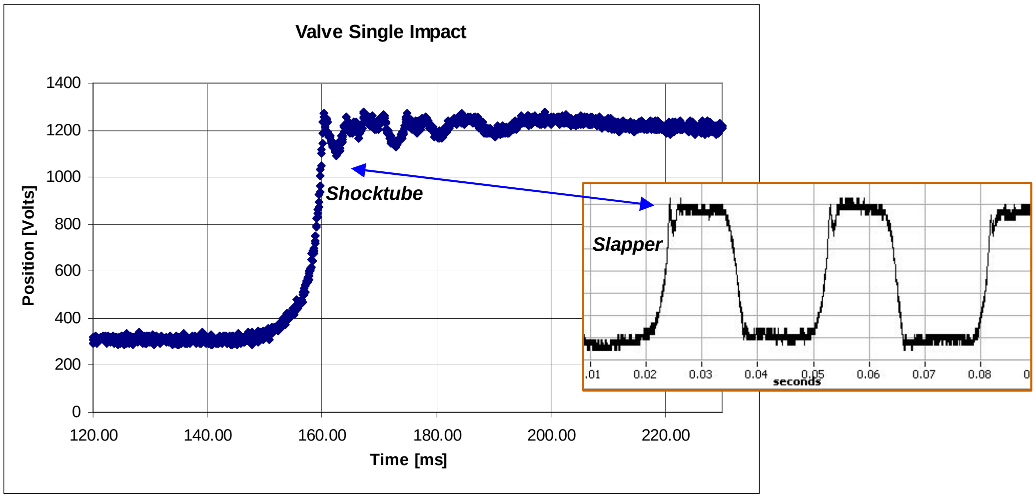

Figure 4-20. Comparison of Plate Motion Profile in Single Impact Shock-Tube Test and Hoerbiger Valve Slapper

valve slapper. The three optical probe sensor measurements were used to determine how the plate impacted the guard, its subsequent motion and the impact velocity. Figure 4-21 shows the plate motion plot generated from the three optical position probes. The motion of the plate is not uniform and causes the plate to impact at an angle rather than flat. (However, valve springs tend to reduce the angularity of the impact.) The plate motion data also indicates that the plate bounces after the initial impact. In the first impact, the plate hits at the 9 o'clock position, followed by a more flat impact at the 12 o'clock and 3 o'clock positions, and then lastly at the 6 o'clock position. In the repeated hits after the initial impact and the subsequent "ringing" effect, different areas of the plate show varied amounts of movement, which confirms the plate's angular movement.

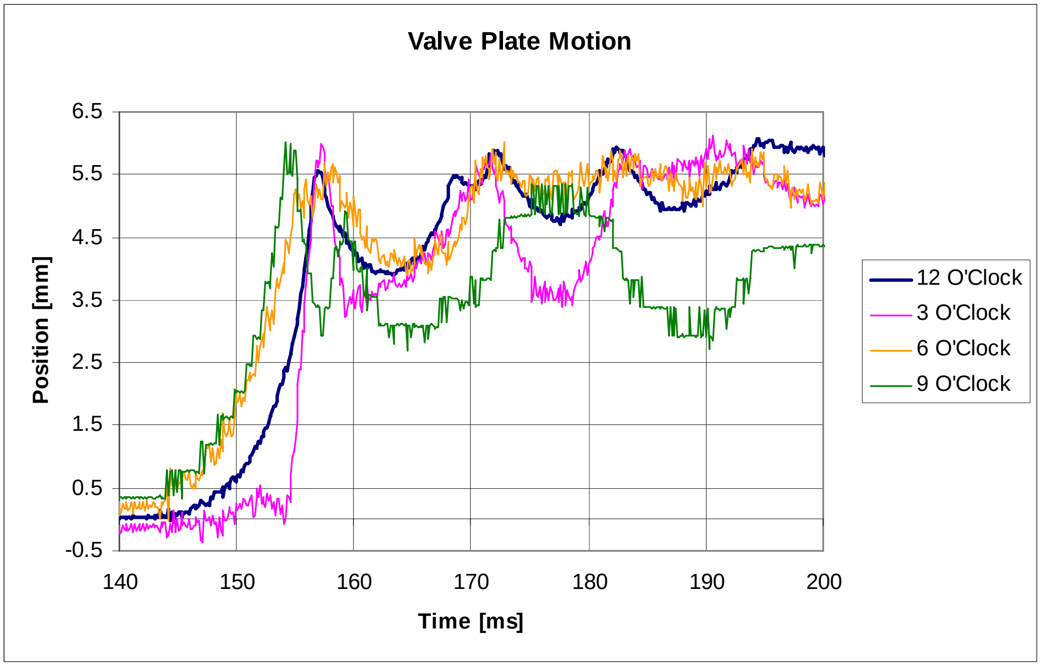

Figure 4-21. Valve Motion Recorded by Optical Position Probes, with Probe Coordinate Transformation

Figure 4-22 shows that the time traces from the strain gauges were consistent with the plate motion profile. The plate strain is at a maximum at the first impact at approximately 1,150 μΣ peak-to-peak. The second and third bounce of the valve plate show significantly less strain and tend to follow a logarithmic decay in strain over time. This result is significant because it means that the primary impact causes the greatest strain on the valve plate and the resulting impacts are considerably weaker. Figure 4-23 compares the plate position to the valve strain to reveal the good correlation between the measured strain values and the plate displacement. The plot also shows the ringing phenomenon after the plate's discrete impact points. The results from the shock-tube testing demonstrated a maximum plate impact velocity of 3.6 m/s, an impact location at 245 degrees, and an impact angle of approximately 5.6 degrees. Measurement uncertainties for velocity, impact angle and stress were around 20%, while the uncertainty for the stress measurement was slightly less at approximately 15%.

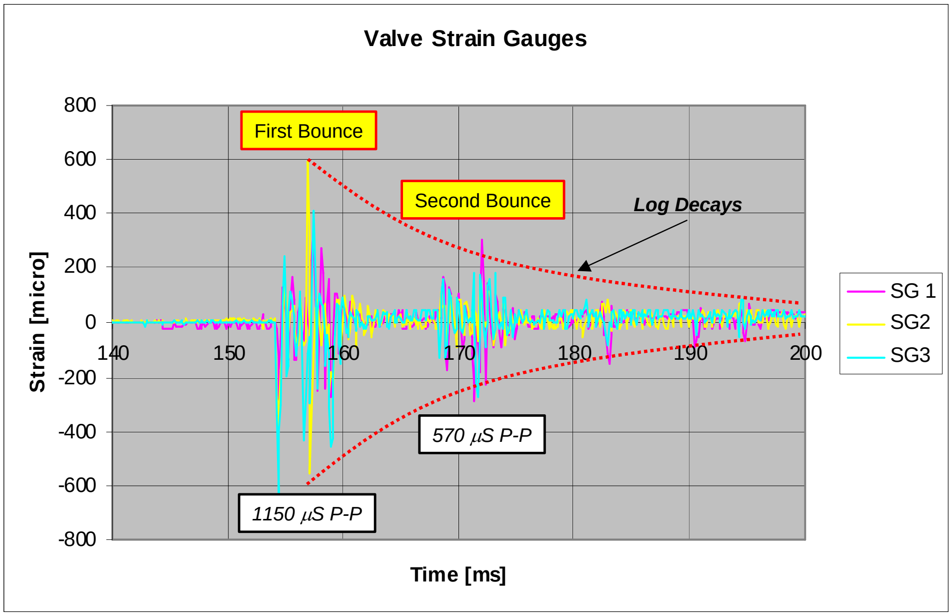

Figure 4-22. Corresponding Valve Strain Measured by Strain Gauges During Simple Impact Testing

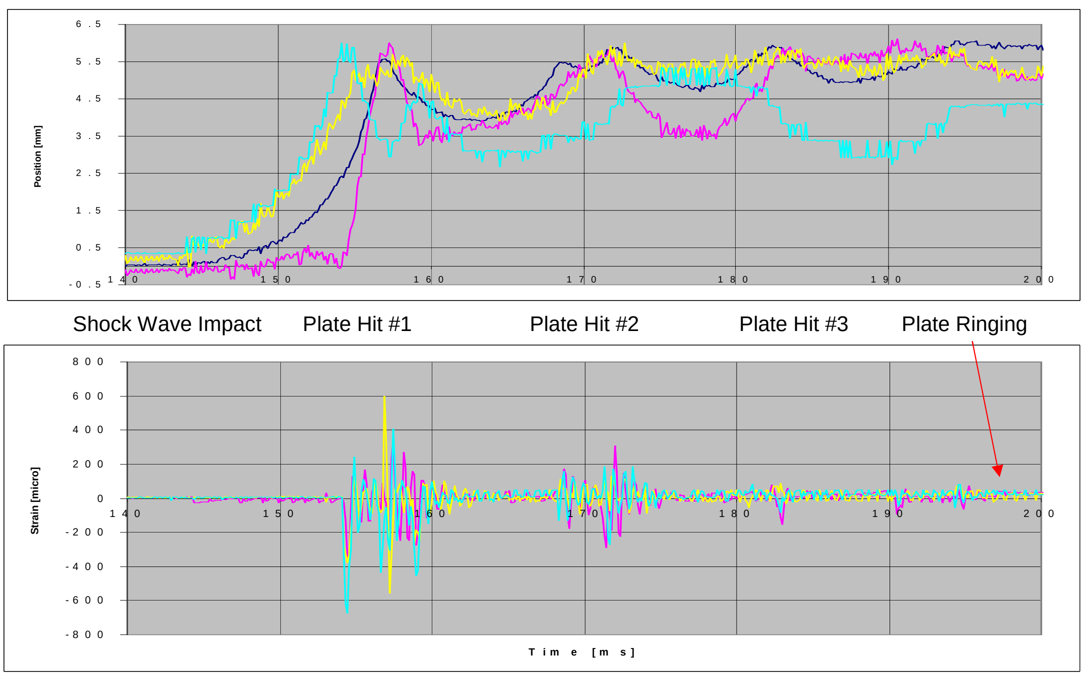

Figure 4-23. Comparison of Measured Valve Position and Strain During Single Impact Test

An enhanced view of the initial impact (i.e., the first bounce) of the valve plate provides a detailed explanation of the strain generated by a single impact event. Figure 4-24 shows the strain of the valve plate at four locations during the first hit. From this strain plot, it is evident that the plate hits on its edge first at an angle of 5.6 degrees, generating a peak-to-peak strain of approximately 950 μΣ. The plate then hits almost flat with a maximum peak-to-peak strain of

1,140 μΣ. Last, the opposing edge strikes the guard at a less significant level (570 μΣ). These values of strain show the initial edge impact, and the latter full plate hit generate the maximum strain in the valve plate. After the first impact events, the valve plate continues to bounce against the guard, but with a considerably reduced level of strain (and stress).

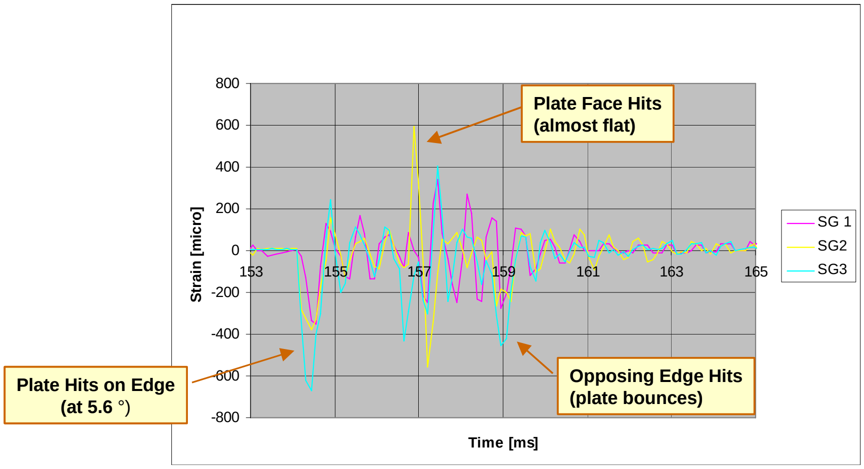

Figure 4-24. Expanded Valve Strain Measurements for Initial Plate "Bounce"

Thus, the single impact testing yielded a characterization of the valve plate kinematic behavior versus strain. The test results can be used to validate an FE model developed to simulate a single impact event for the valve plate. The FE model inputs were the impact velocity, angle, and location of the valve plate. To model the dynamic plate behavior, ANSYS was used to perform a transient stress calculation (typical valve plate geometry using 2,438 elements). Model predictions for transient stress could be compared directly to the calculated stress based on measured strain values. Figure 4-25 shows an example of the finite element model transient stress calculation. Model predictions of the plate movement could also be compared to the plate position plot. Table 4-2 highlights the good agreement between the FE model and the experimental data. The model predicted stress was slightly less than the measured stress in all cases, but within the uncertainty of the shock-tube test.

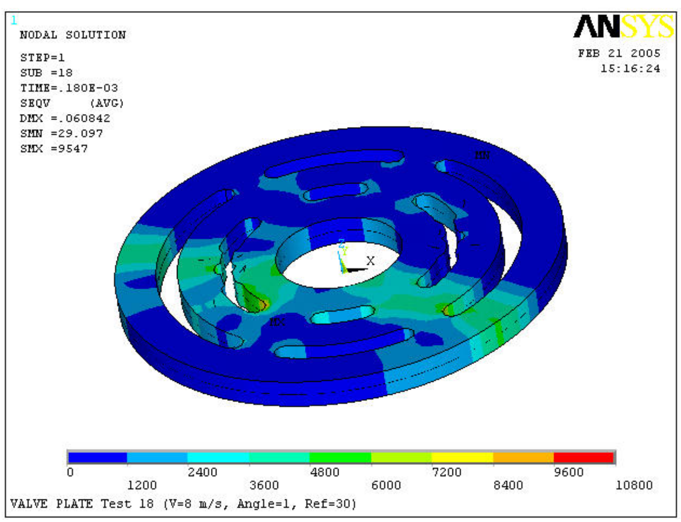

Figure 4-25. Finite Element Model of Valve Plate After Plate Single Impact and Resulting Stress Valve Propagation

Table 4-2. Summary of Results Comparing FE Model Performance to Measured Stress Data

TEST NO.	TEST #25	TEST #31	TEST #45	TEST #49
Velocity (m/s)	3.2	3.6	2.5	2.3
Impact Angle	3.8º	5.6º	5.3º	5.5º
Location	74º	245º	251º	247º
Measured Stress (MPa) (Closest SG, E = 4.3E4 kPa)	38	41	26	27
FE Model Stress (MPa) (Closest SG, E = 4.3E4 kPa)	33	36	24	23
FE Model Deviation	**15%**	**14%**	**8%**	**17%**
FE Model Deviation is within stress measurement uncertainty (U_{st}=20%)				

To predict the stress levels at other impact angles and velocities, the FE model was used in a parametric study to determine the relationship between the plate kinematic behavior and transient stress levels. For a range of impact angles, velocities, and locations, the parametric study identified the highest stress location for a particular set of input values. The quarter symmetry about the plate allowed a detailed parametric evaluation of the plate to be performed on a single quarter section. Maximum and minimum stress levels expected at particular locations were identified for specific impact angles and velocities. An example of the results obtained at an impact location of 30 degrees is shown in Figure 4-26. The figure shows the increase in peak-to-peak stress as velocity increases. At an increased impact angle, the stress level increases more rapidly as velocity increases.

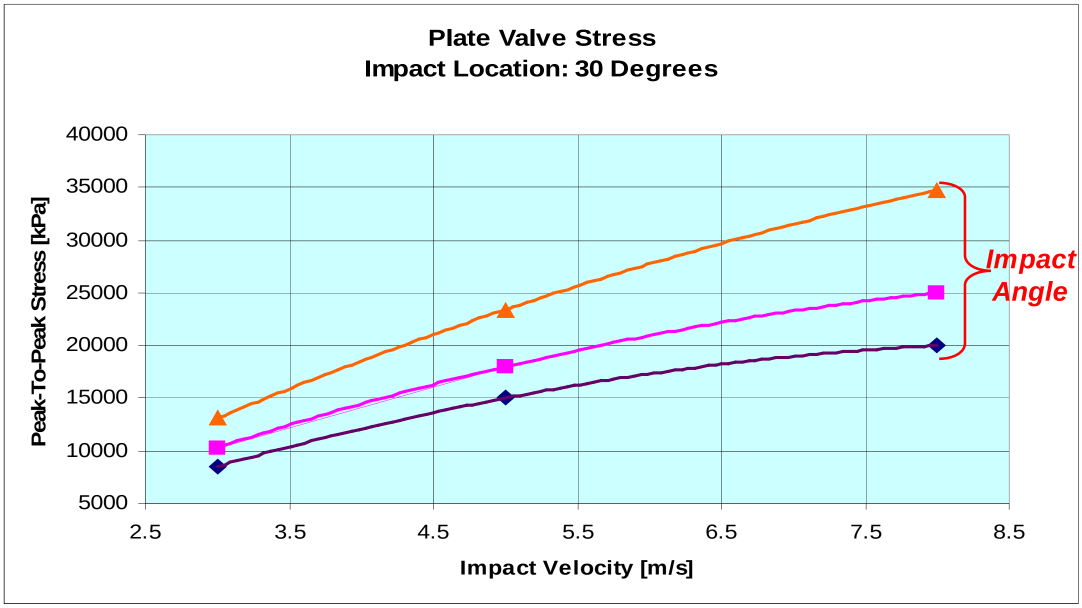

Figure 4-26. Parametric Study Results Using FE Model for Impact Plate Location of 30 Degrees

Thus, the FE model will be used to predict valve life based on material stress-life properties and a characterization of the valve motion over a range of operating conditions. The testing proved to be worthwhile in understanding the kinematic behavior of the valve plate and the stress created by the differential pressure force across the plate. The single impact testing also provided the following conclusions about the kinematic motion of the plate:

- The valve plates were seen to *always* hit at an angle.
- The plate usually “bounces” at least one more time after the initial impact, but the subsequent impacts are softer.
- The highest stress events within each impact occur when the initial edge of the plate hits and the full plate hits, which is followed by the opposing edge hit.
- The initial edge and full plate hit generate the highest stresses (evident in Figure 4-24), and both of these hits create similar levels of stress (within 20% of magnitude).
- The highest stress levels decay quickly, but the plate continues to “ring” after impact, creating ongoing dynamic low stress.

4.2.4 PLATE MOTION ANALYSIS

In order to effectively use the transient stress predictions of the FE model, the 3-D plate motion must be understood for a range of operating conditions and valve/piston designs. Thus, another critical element of the reciprocating compressor valve program is the study of the plate motion in reciprocating compressor machines (*Program Element 1*, as shown in Figure 4-17). This element of the program will provide the data needed to characterize the three-dimensional motion of the valve plate. Once the characterization is complete for a range of operating conditions and design parameters, a probabilistic method can be developed to provide the necessary inputs (impact velocity, angle, and location) to the transient stress model.

To study the valve plate motion, a series of tests were conducted at the SwRI Metering Research Facility using an Ariel 250 HP reciprocating compressor, shown in Figure 4-27 (left). Three optical position probes were used to monitor the position of the valve plate. The optical position probes were mounted on the discharge valve (see Figure 4-27 at

Figure 4-27. Ariel Reciprocating Compressor Used in Measuring Valve Motion (Left) with Optical Position Probes Mounted on Valve (Right)

right). The displacement data from the probes is then used to determine the plate velocity, angle of impact and impact location. An example of the calculated impact velocity determined from the optical position probe data is shown in Figure 4-28. The first round of testing measured the plate valve response for one particular set of springs at a speed of 900 RPM and a range of compression ratios. In the second round of testing, the valve spring stiffness was varied to determine the effect of spring stiffness on valve motion.

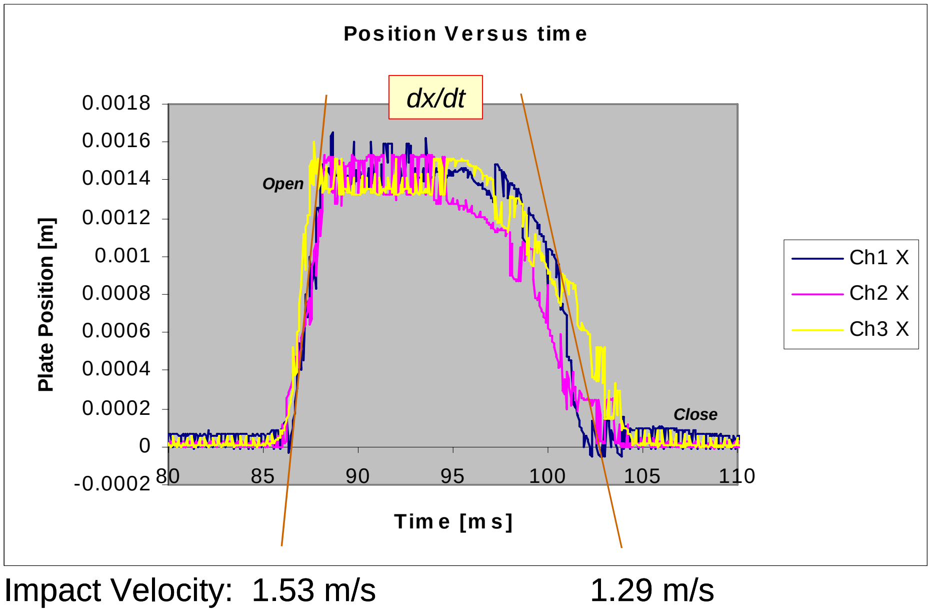

Figure 4-28. Typical Position Probe Data Used to Calculate Impact Velocity on SwRI Ariel Compressor

The testing indicated that the most consistent empirical relationship is that between impact angle and plate velocity, as shown in Figures 4-29a and 4-29b. These figures show that opening and closing impact velocities depend on the impact angle of the plate. In both instances, the plate velocity increases for decreasing impact angle. In the second round of testing, the spring stiffness was increased in the discharge plate valve. Initial results of the second round of testing are shown in Figure 4-30, which compares the two values of spring stiffness to the impact velocity of the plate. The opening impact velocity is not affected by the change in spring stiffness, while the closing impact velocity increases significantly when the stiffness is increased. These results suggest that, in the case of the discharge valve, the pressure forces acting on the valve from the cylinder dominate the opening impact velocity. The closing impact velocity is controlled by the spring forces, as shown by the significant increase in closing impact velocity when stiffer springs were used. If the springs have relatively low spring stiffness, the valve closes more softly.

After the second round of testing on the 250 HP reciprocating compressor, a final set of data was taken with a variable frequency drive installed on the compressor. The variable drive allowed the compressor speed to be varied. Valve motion data for compressor speeds of 500, 600, 700, and 800 RPM, and pressure ratios of 1.3 to 1.6 was added to the current database. The effect of operating speed on impact velocity and impact angle is determined based on this data.

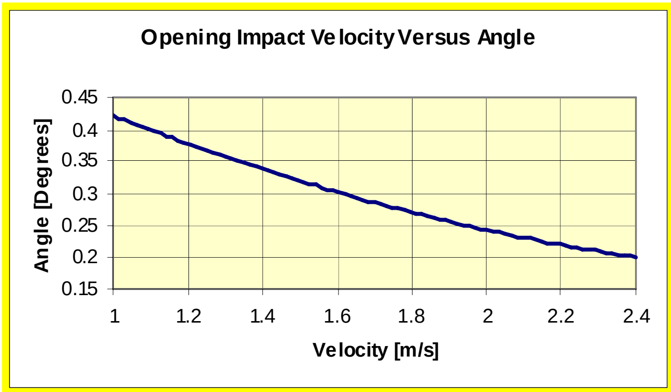

Figure 4-29a. Correlation Between Impact Angle and Opening Velocity for Valve Plate, Based on Displacement Probe Data

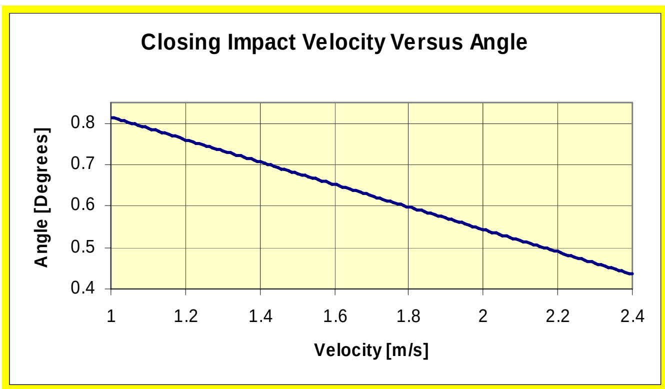

Figure 4-29b. Correlation Between Impact Angle and Closing Velocity for Valve Plate, Based on Displacement Probe Data

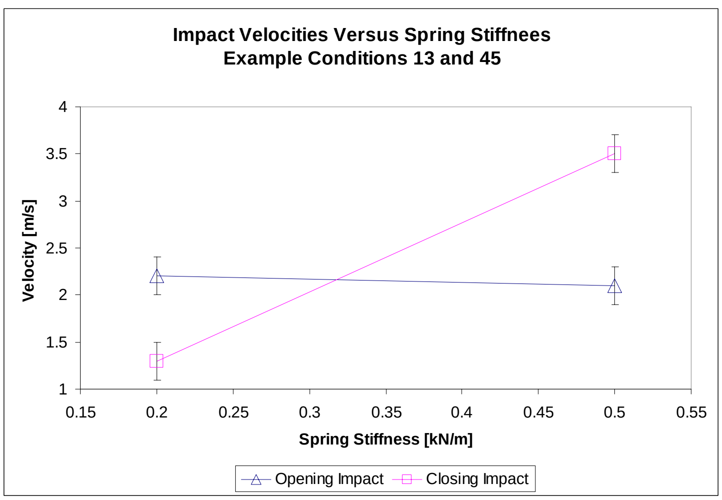

Figure 4-30. Variation in Spring Stiffness for Opening and Closing Impact Velocities

4.2.5 MATERIALS ANALYSIS

In addition to understanding the three-dimensional valve motion and predicting the transient stress, a stress-life analysis of the valve plate material is being performed. This analysis will be used to convert the predictions of the transient stress model into an estimation of valve life using S-N material curves. Fatigue life testing is being performed on PEEK material samples to determine the number of cycles to failure at various stress levels. Stress versus life curves (S-N curves) are then created for each set of tests.

Fatigue testing of PEEK was performed at room temperature at stress levels of 4.5, 11.3, 13, and 16.3 KSI at a frequency of 5 Hz. The testing was consistent with manufacturer predictions, but also indicated that fatigue life is nearly infinite for very low stress levels. The experimentally determined S-N curve based on the SwRI testing is shown in Figure 4-31. The materials analysis did determine the yield stress of typical plate valve materials based upon the number of level of cycles measured. The results were combined with valve motion data and the finite element model to estimate valve life using the plate valve life application tool developed in the current research.

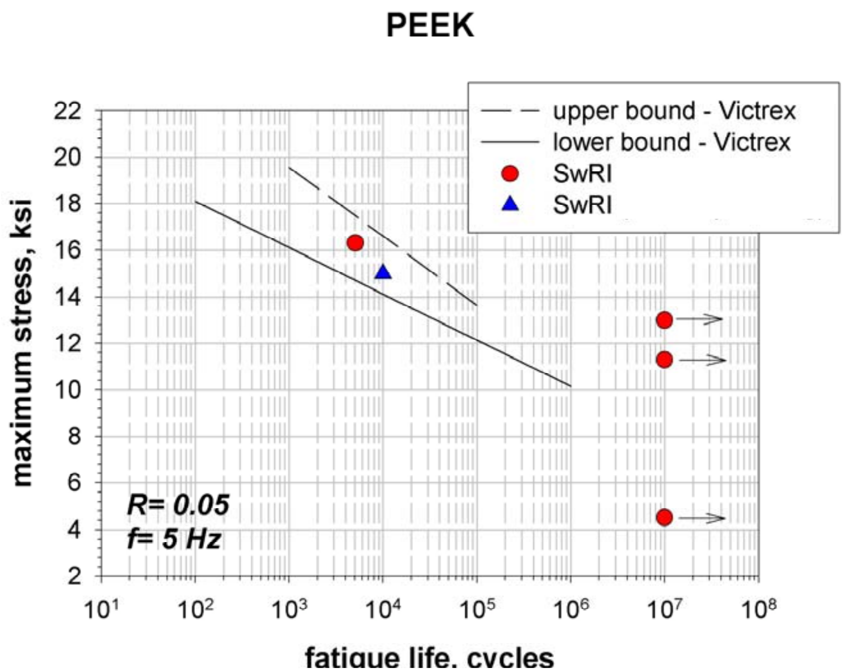

Figure 4-31. Experimental S-N Curves Developed Through Fatigue Testing

4.2.6 PLATE VALVE LIFE ANALYSIS TOOL

The finite element model of plate valve stress and the material analysis were combined with the empirical relationships developed through the 3-D valve motion assessment in a plate valve life analysis (VLA) tool. The VLA allows the user to easily evaluate the effect of compressor performance parameters, valve geometry and design parameters, and valve operation on the effective life of the plate valve. A "screen-shot" of the application interface screen for the VLA is shown in Figure 4-32. An existing compressor performance simulation tool is used by the application to generate a one-dimensional plate motion profile based upon the compressor

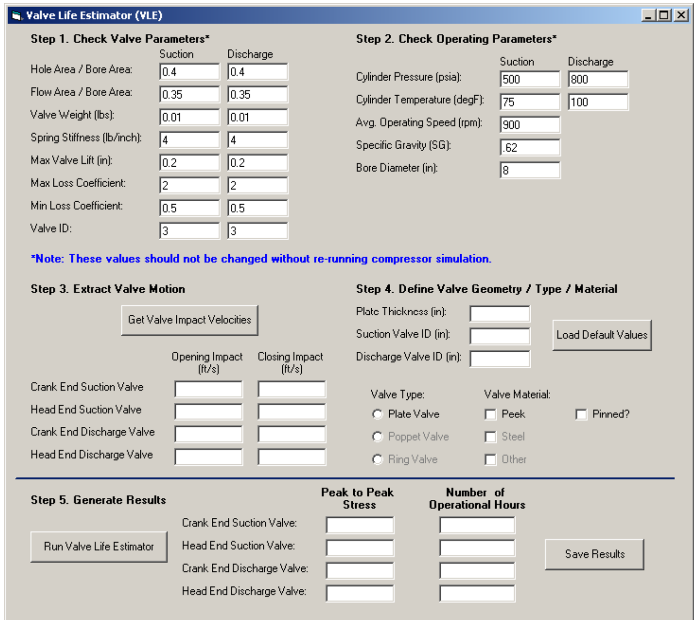

Figure 4-32. User Interface for Valve Life Analysis Tool

performance. The VLA is then called by the valve life program application to determine the valve impact velocities. The VLA automatically extracts the opening and closing impact velocities for each compressor valve location.

Once the impact velocities are determined based upon the specific operating conditions of the compressor and the valve geometry, the VLA calculates three plate bounces per impact velocity and two angular plate impact events per bounce. For one cycle of the compressor, the VLA determines 12 individual impact events for a particular valve location. Finally, the VLA formulates a 3-D model of the valve velocity and impact angle for each impact event. This model is the basis for the stress prediction using the finite element model previously developed. The VLA tool determines peak-to-peak stress levels based upon the finite element model. Using the material properties for peek, stainless steel (SS316) or nylon, the VLA tool estimates valve life, using the experimentally developed S-N curves and the assumption of continuous operation of the plate valve. The VLA tool is particularly useful in that it can easily perform calculations of valve stress and valve life based upon varying parameters, such as the valve lift, spring stiffness, or compressor speed. The application tool has the ability to compute compressor efficiency, which is a direct design trade-off to valve lift. The application will be further developed to serve as a valve analysis tool for two other basic valve designs (ring valves and poppet valves).

4.2.7 VERIFICATION OF PLATE VALVE LIFE MODEL

Though the plate valve life analysis model was developed based on experimental data and verified model predictions, the analysis tool itself must be verified through actual field measurements. Initial testing with the beta version of the model indicates that the VLA tool predicts trends that are consistent with existing relationships. As the 3-D valve motion model is improved through field testing of different valve types, the VLA application will be updated. The intent of the research is to develop an application tool that can be used by operating companies to determine the optimal valve performance and compressor operation, through a study of the net effect of design trade-offs and compressor operational changes. The research does not aim to predict the valve life in terms of absolute accuracy.

4.2.8 VALVE ANALYSIS SUMMARY

The reciprocating compressor valve research aims to improve compressor efficiency and reduce downtime by enhancing valve performance. A dynamic finite element model has been developed and validated against single impact test data for plate transient stress analysis. Combining these stresses and valve motion predictions with material stress-life characteristics will yield valve plate life predictions. The data gathered through the analysis of valve motion and materials testing were combined in a working application tool for valve users. The resulting prediction capability should enable the reciprocating compressor user to tune the machine performance to reduce valve stress and prolong valve, using the plate valve life estimation tool. The predictions of the plate life estimation tool will allow users to balance valve life and performance to meet a particular operation/business need for the compressor. The tool will be primarily useful in weighing the competing performance characteristics of a plate valve in terms of its life. Lastly, the insight gained during the passive valve investigation provides guidance for requirements for new improved valve concepts.

4.3 VALVE SPRINGS OPTIMIZATION

4.3.1 SPRING FAILURE ANALYSIS

Industry estimates indicate that approximately 50% of all valve plate failures are preceded by spring failures. Thus, the focus of this study is to understand the dynamics of spring motion in a typical plate valve and predict the relationship between this motion and the resulting stresses. Once this relationship is established, current and new spring geometries can be optimized for reduced peak-to-peak stresses. This will result in fewer spring related valve high cycle fatigue failures.

4.3.2 FINITE ELEMENT MODEL

A spring geometry was chosen which is used in the ARCT Test Bed that employs an Ariel 250 HP reciprocating compressor. A 3-dimensional finite element model of the spring was generated. The resulting mesh is shown in Figure 4-33. Since there are four springs used with this plate valve, one-fourth of the plate mass is added to the end of the spring. The other end is fixed. The flats at the ends of the spring are also modeled.

Figure 4-33. Finite Element Model of Spring

First, a modal analysis is performed, which calculates the natural frequencies of the system. The first two natural frequencies are shown in Table 4-3. The mode shapes are shown in Figure 4-34. The first mode has greatest motion near the free end of the spring while the second mode oscillates near the middle. The compressor operating speed is 900 RPM (15 Hz). Therefore, both modes are well above running speed.

Table 4-3. Natural Frequencies of Spring System

Mode	Natural Frequency (Hz)
1	35
2	477

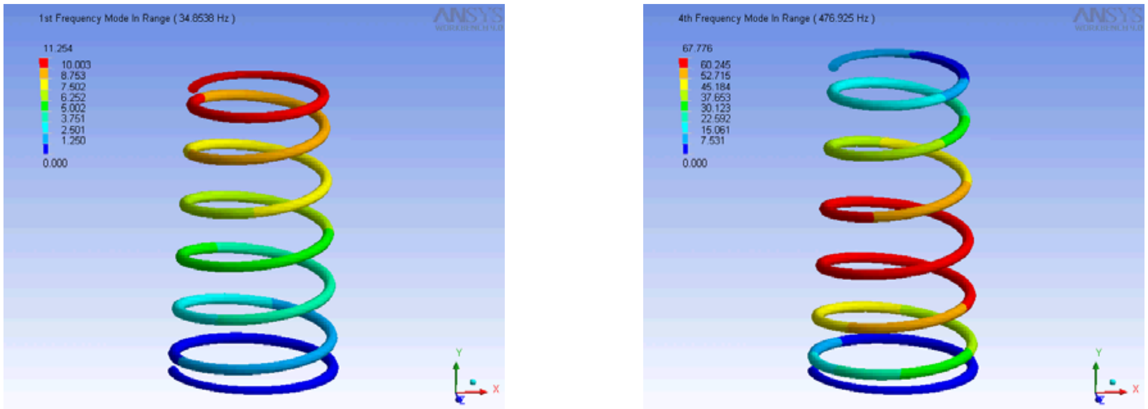

Figure 4-34. First and Second Mode Shape of Spring

The plate motion prescribes the harmonic motion of the plate. To improve the realism of the analysis, actual plate motion measurements are used as shown in Figure 4-35. A Fast Fourier Transform (FFT) of this harmonic motion is made to break the motion into its frequency components as shown in Figure 4-36. The multiples of running speed are labeled (1x, 2x, etc.).

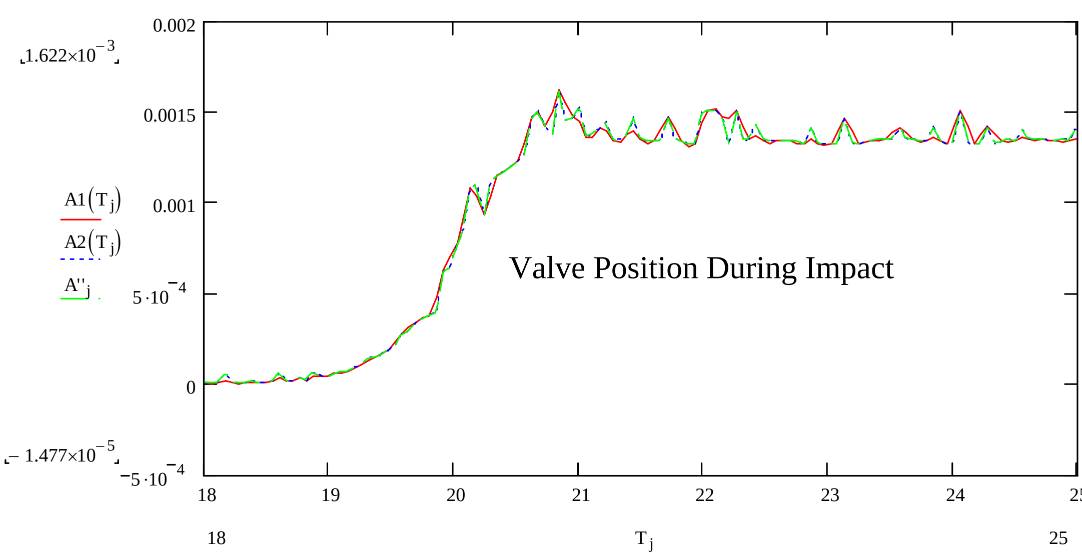

Figure 4-35. Measured Valve Motion During Opening

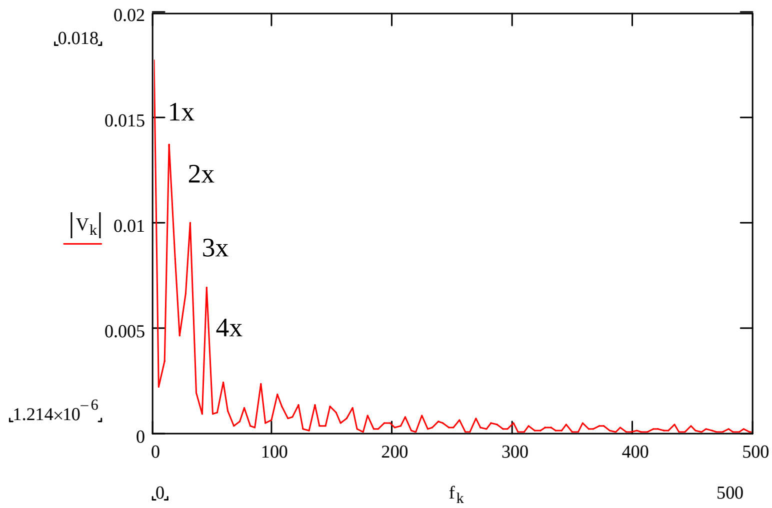

Figure 4-36. FFT of Measured Valve Plate Motion

Using the magnitude and frequency of these major FFT components, a harmonic solution of the finite element is performed by prescribing the motion at the free end of the spring to be equal to the plate motion components. The solution is found for each component and summed together to obtain the total maximum stress in the spring as shown in Figure 4-37. As expected, the stresses in the spring are within the yield strength of the material (approximately 180,000 PSI), since no failure has occurred with this spring.

Future work will perform parametric studies on the effect of impact velocity. Also, transient calculations will be made to predict the stress waves traveling through the spring due to this impact. Guidelines for improving spring reliability will also be developed.

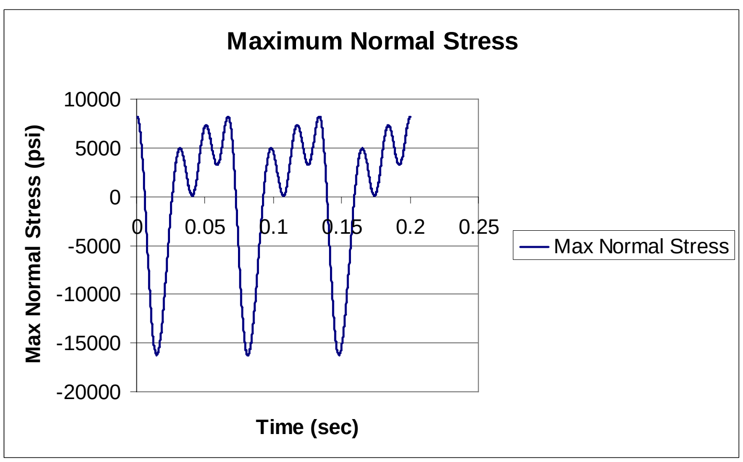

Figure 4-37. Total Maximum Normal Stress in Spring

4.4 ADVANCED VALVE CONCEPTS

The "Active Valve New Concepts" phase of this project is aimed at developing technologies for robust, efficient, and long-life compressor valves. Active valve ideas encompass ways to improve on existing valve technology by improving life and/or efficiency. The initial effort concentrated on documenting the current state of technology in compressor valves, brainstorming on new methods, and developing advanced solutions for improving the state of technology in reciprocating compressor valves.

The current state of technology in compressor valves shows that the majority of the valves in service are strictly passive in design. There have been attempts at actively controlling the valve motion; however, most of the control mechanisms are in service to help control compressor capacity, not improve valve life or efficiency. There are three main types of valve actuation configurations. These configurations are listed in Table 4-4 along with definitions.

Table 4-4. Valve Actuation/Control Terminology

Passive	Movement controlled purely by pressure force and mechanical spring force. Optimized for a single operating condition.
Semi-Active	Pressure force driven with motion assist – no pressure transducer or shaft encoder required. Able to be optimized for a wide range of operating conditions due to motion assisted control.
Active	Allows full control of valve motion by a totally external force, controlled with shaft encoder or a pressure sensor. Allows for the widest range of operating conditions.

Most of the valves in use belong to the passive category of valve actuation. However, the design of these passive valves inherently relies upon a constant valve behavior (constant valve actuation, movement, and compression volume), which is usually not the case in valve operation, given the wide operating range and profile requirement of gas compressors. Valve losses associated with flutter, late or early closure, and imperfect sealing can lead to increased inefficiency of the compressor. Clearly, adaptive valve behavior through active valve control would be more beneficial at preventing valve failure and improving compressor efficiency.

4.4.1 CONCEPT GENERATION

Through the use of a brainstorming session, 61 possible ideas emerged. The ideas were then grouped and ranked in accordance with a preliminary set of criteria. The list of concepts was shortened to around twenty possible candidates and a better description was added to help define each idea. This abbreviated and defined list are shown below in Table 4-5.

Table 4-5. Abbreviated Table of Valve Concept Descriptions

VALVE IDEAS	DESCRIPTION OF VALVE CONCEPT
Electromagnetic Plate Valve	Shaft with electromagnet coil, used to sense and control plate
Improved Spring Technology	Springs with increased durability or variable stiffness
Combo E-M Rotary Valve	Valve opens according to rotation, controlled by motor
Screw/Cam Thread Rotary Valve	Rotational valve on threaded screw or cam
Self-Damping Valve	Return flow to guard side of valve to achieve self damping
Slider Valve	Sliding plate opens/closes along cylinder surface
Reinforced Passive Valve	Passive valve reinforced in specific areas found to have higher frequency of impact
Passive Valve with Removable Guard	Traditional valve with a removable guard for increased flow area
Fully Active Plate Valve	Traditional plate valve controlled with hydraulic power across entire motion

VALVE IDEAS	DESCRIPTION OF VALVE CONCEPT
Multiple Guillotines	A series of perforated plates controlled actively
Variable Choke Valve	Variable nozzle throat ID to choke flow at varied flow rates
Micro-Flapper Valve	Multiple small flaps used to open valve, individual pivoted slats
Guillotine Valve	Uses a flat plate to cut into flow
Valve in Piston	Valve is located in piston for improved gas flow
Variable Orifice Valve	Variable open area through plate valve to vary flow amount
Flapper Valve	A flapper element is used to block passage area
Valve Installed in Cylinder	Valve is located in cylinder for improved gas flow
Variable Permeability	Material permeability changes to control flow
Rotary Restriction Camera Shutter	Camera like shutter to open and close valve
Membrane Valve	Valve allows flow to come in based on pressure
Multi-Stage Bladder Valve	Multiple bladders used, with varied openings
Flexible Bladder Valve	Bladder used to expand (to close) or deflate (to open)

The abbreviated list was then ranked on the categories shown in Table 4-6. Each category has a definition as it relates to the valve concepts.

Table 4-6. Weighting Criteria and Definitions as they Pertain to Valves

CATEGORY	DEFINITION
Life & Reliability	Ability to withstand normal wear and tear with low leakage. Ranking from 1-5, with 1 being less than current designs.
Effectiveness	Modification or new design improves the life or sealing capability. Ranking from 1-5, with 1 being no improvement.
Efficiency	Modification or new design improvements decreases valve loss. Ranking from 1-5, with 1 being least efficient.
O&M Cost	Maintenance cost to keep new design running. Ranking from 1-5, with 1 being most maintenance cost and time required.
Retrofitable	Ability to be used with little to no modification. Ranking from 1-5, with 1 being least retrofitable.
Developmental Risk	Amount of risk involved in bringing idea to usefulness. Ranking from 1-5, with 1 being large risk.
Installed Cost	Relative cost compared to present passive valve technology. Ranking from 1-5, with 1 being most costly to implement.
Maturity	Idea maturity, an estimate of developmental work needed. Ranking from 1-5, with 1 being least mature/most development.

The ranking for each concept varies from one to five, with one being the worst. The highest-ranking concepts were then selected and refined to provide for quantitative analysis. The top three concepts were then selected to move forward on possible designs.

4.4.2 SHORT LIST OF CONCEPTS

The three concepts selected for refinement span the passive, semi-active, and active categories. The "Electromagnetic Plate Valve," "Combo E-M Rotary Valve," and "Screw/Cam Thread Rotary Valve" all have attributes that lead to a better total compressor. Table 4-7 shows how each idea on the short list ranked when using the weighting criteria.

Table 4-7. Short List Valve Idea Ranking

Valve Ideas	Life & Reliability	Effective-ness	Efficiency	O&M Cost	Retrofit-able	Synergy	Develop-mental Risk	Installed Cost	Maturity	Total Score (Out of Possible 5)	Type of Actuation
Weighting Factor	18%	15%	14%	13%	10%	9%	9%	7%	5%	100%	
Electromagnetic Plate Valve	5	4	3	5	5	4	4	3	3	4.15	Semi-Active
Combo E-M Rotary Valve	4	4	5	3	3	4	2	3	2	3.58	Active/Semi-Active
Screw/Cam Thread Rotary Valve	2	4	5	3	3	3	2	3	2	3.12	Passive/Semi-Active

4.4.2.1 Electromagnetic Plate Valve

The semi-active concept consisting of a standard plate valve with a shaft and coils came to the top of the short list. Figure 4-38 shows the first iteration at this type of design. The basic principal behind the valve is that control is only needed at certain times in the valve's cycle. The pressure forces (just like a typical valve) start the opening; however, once in motion, the electromagnets take over and limit the impact velocity.

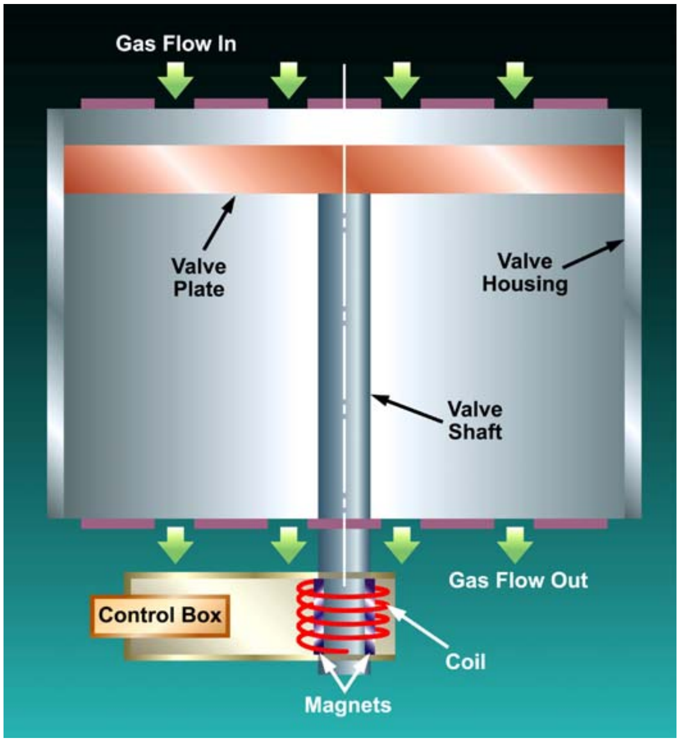

Figure 4-38. Electro-Magnetic Linear Valve

Since the impact velocity can be controlled actively, the lift may be increased, thus, positively impacting the efficiency. With the basic operation of the valve unchanged, there is minimal impact to the fleet of reciprocating compressors already in service. To most installations, the modifications needed would be a simple valve replacement. The concept also has a built-in failsafe. If the coils fail or the shaft breaks, the assembly reverts to a conventional passive valve.

4.4.2.2 Combo E-M Rotary Valve

The second conceptual idea is a fully active design that incorporates a motor and a cam type actuation. The opening and closing events are controlled by crankshaft timing or by pressure sensing. Figure 4-39 shows the first iteration of this idea.

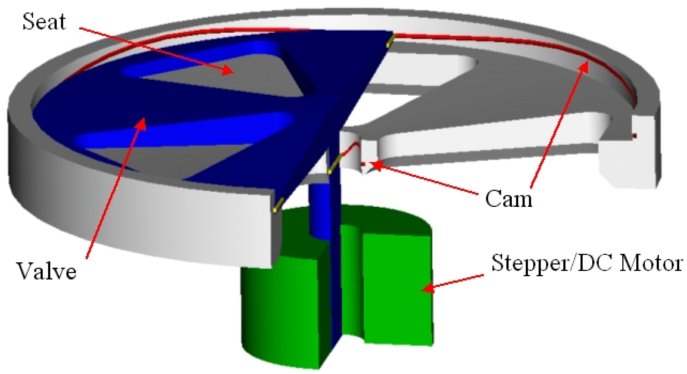

Figure 4-39. Electro-Magnetic Rotary Valve

This idea is similar to the Cozzani Company rotary valve design; however, the main difference is the addition of the cam to overcome the oil sticktion that can occur in oil-lubricated compressors. The cam lifting action also allows the cylinder pressure forces to assist in the opening of the valve. The controlling motor could then accelerate and decelerate the valve to minimize abrasion and impact velocities. The rotation of the plate allows for unturned flow through the seat, valve plate, and guard. The efficiency of this idea is appealing because the losses through the valve would be minimized.

4.4.2.3 Screw/Cam Thread Rotary Valve

The final conceptual idea on the short list is a passive design that incorporates a screw or cam type actuation. The profile involved looks much like a tangent curve in that the extremities of the movement are purely linear. The rotation occurs during the valve stroke to open the passageway and allow for better flow coefficients. Figure 4-40 shows the first iteration of this idea along with a conceptual representation of the cam profile.

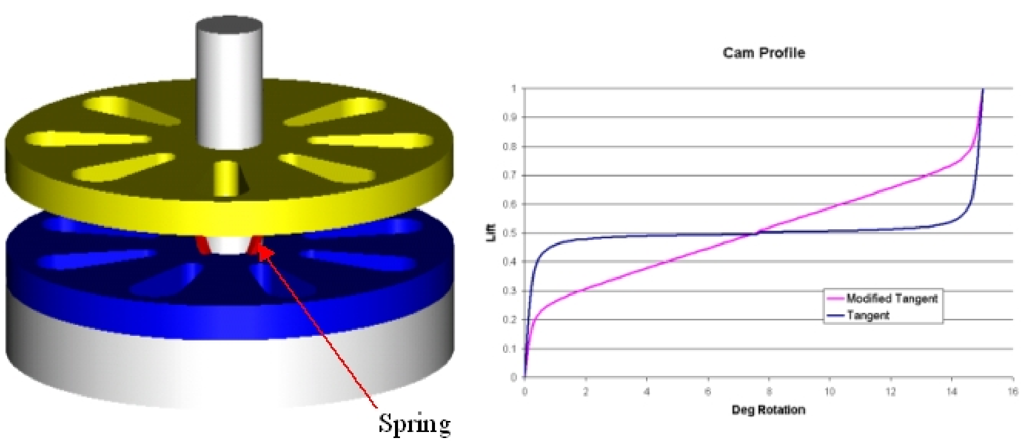

Figure 4-40. Passive Rotary Valve with Tangent Cam Profile

The efficiency of this type of valve should be high because flow does not undergo rapid turning and, thus, the loss coefficients are reduced. This rotary valve may be able to change the standard valve configuration away from using a guard to stop the valve plate. If the forces on the plate are low enough, then the plate could be free-floating on a shaft with nothing but a spring to control the stroke. With no guard, the main culprit in valve failures is removed because the plate undergoes half as many impacts as a valve with a guard.

These three concepts have been distilled from a list of sixty-one. They each involve technical challenges that would (if solved) be helpful to the industry in general. The rest of this project is focused at quantitatively answering these challenges and providing industry with the valuable knowledge.

4.5 ADVANCED VALVES PROOF-OF-CONCEPT

As previously documented, a reduction in impact velocities can greatly increase the life of a valve. The semi-active valve concept primarily aims at reducing valve plate impact velocities by actively applying an opposing electromagnetic force to dampen the motion of the plate prior to any guard/seat impact. The semi-active valve acts passively throughout most of the plate motion cycle and only applies a short duration electromagnetic damping force prior to the plate's impact to reduce plate velocities. This concept can be implemented by using the same electromagnetic coil for both sensing and control, i.e., the coil-magnet performs a linear sensing function to measure the plate's position. Once the voltage output from coil indicates that the plate is about to strike the guard or seat, the coil is switched to become an electromagnetic actuator for motion control by applying a large opposing voltage to the coil. This concept is inherently fail-safe since, should the control or coil fail, the valve reverts automatically to operate as a fully passive valve.

The design concept was initially tested using a single impact shock tube. For this testing, a valve element was coupled to the voice coil of a loudspeaker to provide a controllable reaction force. Results from the single impact tests showed that the reaction force applied by the coil was able to measurably reduce the impact velocities of the valve plate. The reduction in impact velocities resulted in 40% to 50% reduced stress levels and, thus, a relative life gain of 3 to 11 times (based on high cycle fatigue calculations) versus that of a standard valve. Based on these results, a full-scale test breadboard model was designed, fabricated, and implemented into the ARCT test bed compressor. The design included a standard valve configuration with small modifications to couple it to an electromagnetic loudspeaker voice coil.

4.5.1 SEMI-ACTIVE VALVE PROOF-OF-CONCEPT DESIGN CONSIDERATIONS

A number of critical design considerations must be addressed to implement the semi-active valve concept to an operating benchscale model in a full size compressor. The constraints include the magnitude of the opposing force required, the added weight to the valve plate, and sealing of the assembly to prevent process gas from leaking. Also, to reduce time and cost, off-the-shelf components were principally used in the design and fabrication.

The first design consideration is the force that is needed to react against the gas loads when the plate valve opens or closes. Generally, the more delayed within the valve opening/closing movement the reaction force is applied to the plate, the more effective is the impact damping effect, but also the more peak power is required from the controller. In order to have control flexibility and minimize cost, the coil and amplifier were oversized by approximately 100%, which resulted in relatively large loudspeaker geometry. Another design consideration is to reduce the weight added to the valve's moving elements to avoid mass inertia to cause delayed valve action. A nylon material was used for most of the valve-to-coil coupling parts. Also, the translation rods were constructed from aluminum to minimize weight while allowing for stiff force translation.

The sealing between the valve and atmosphere was another critical design concern. As tolerances add in the assembly, the through-holes in the valve's cap had to be oversized to avoid friction heating on the rods. Namely, the internal clearance required for non-stick operation of the valve assembly required that the sealing of the system pressure be external to the actual valve assembly. Furthermore, the speaker itself cannot withstand the differential pressures during valve normal operation. Thus, a pressure containment vessel was constructed around the speaker to allow for system sealing with no pressure differential across the loudspeaker cone.

The use of off-the-shelf component for the design significantly reduced the development time and costs of the semi-active valve assembly. Using a standard loudspeaker solved numerous design issues that could have easily added significant cost and time to the program. However, for future more advanced designs, the coil should be tightly integrated with the valve so that the interface is seamless, compact, and the large external sealing structure is not required.

4.5.2 Semi-Active Valve Assembly

The design of the semi-active plate valve is based on existing plate valve technology. A schematic of the entire assembly is shown in Figure 4-41. Only slight modifications are needed to facilitate the addition of the electromagnetic loudspeaker coils. Four rods coupled the valve plate directly to the loudspeaker for motion control, and the valve was rigidly mounted to the loudspeaker externals using the valve's cage. To best facilitate the installation of the valve into the compressor, the cage was integral with the valve assembly.

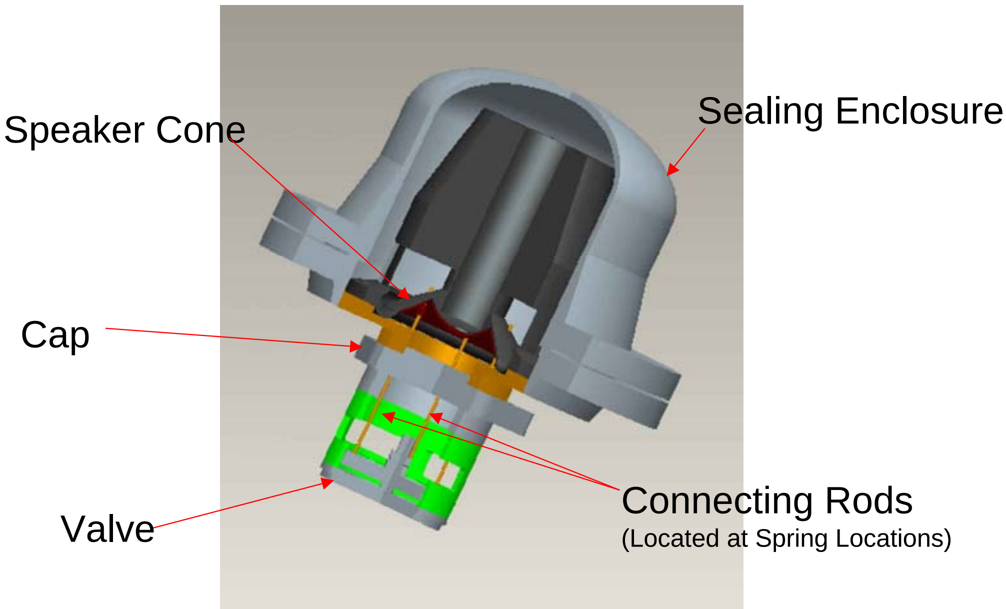

Figure 4-41. Semi-Active Plate Valve Design

The Cone style sealing enclosure couples the standard valve cap to a standard compressor flange on the semi-active valve assembly and provides sealing to the atmosphere. The base is also used to rigidly mount the speaker. As previously mentioned, the speaker was connected to the plate using aluminum rods. The rods were sized such that the speaker provided preload to the valve. Once assembled, the entire assembly was closed to ensure proper sealing of the process gas. Power and sensing wires were also fed through the cap. Thus, the entire semi-active valve is a rigid assembly that is easily mountable on the compressor and provides good process gas sealing.

4.5.3 SEMI-ACTIVE TESTING AND CONTROL SCHEME

Initial testing consisted of a square wave signal sent to the coil while controlling the amplitude on the coil. This initial test helped to resolve clearance issues and allowed to adjust the travel of the valve/speaker to assure complete opening and closing motion of the plate. The valve was then installed into the ARCT test bed compressor. A picture of the installation is shown in Figure 4-42.

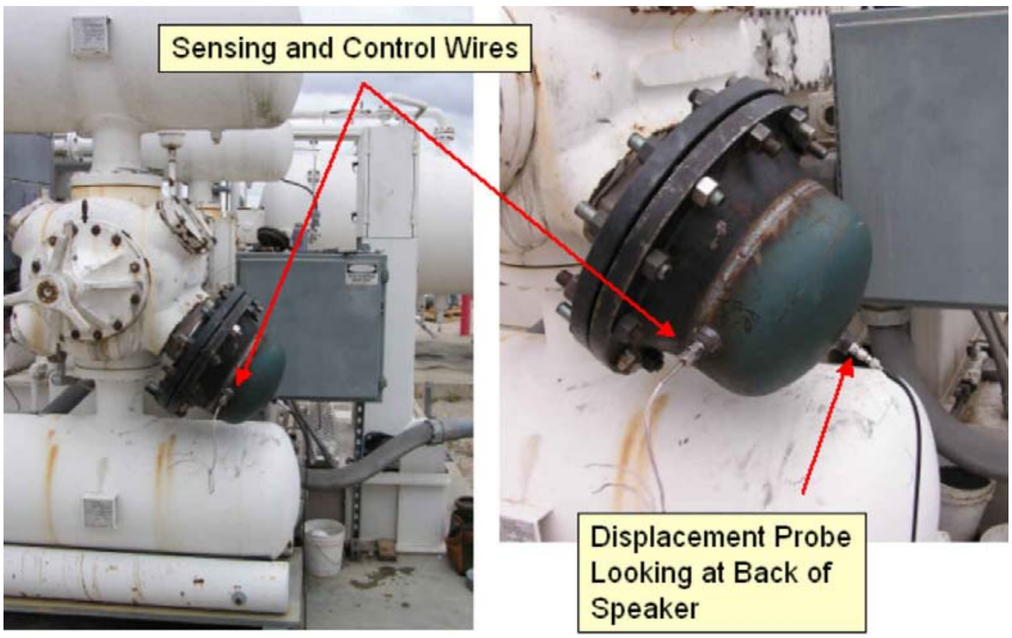

Figure 4-42. Semi-Active Plate Valve Installed in Compressor

The control of the plate movement is accomplished using a simple PC with AD and DA card connected to the electromagnetic loudspeaker coil and an acoustic amplifier. A schematic of the control strategy is shown in Figure 4-43.

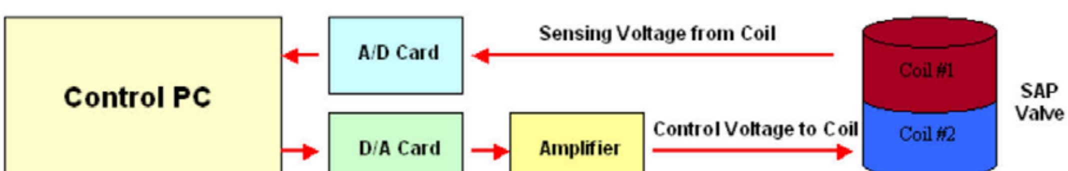

Figure 4-43. Control Philosophy

The first coil is used for sensing only via an AD card. The computer uses the signal generated from this coil to time the output. Initially, the control algorithm records the coil's output for a period of time with no control applied to the second coil to determine peak-to-peak voltages and valve cycle duration. This data is subsequently used to set reaction control amplitude and timing for the control coil. Namely, when the sensing coil measures a preset value of the output voltage, the control algorithm reacts with a reaction pulse through the amplifier to the second coil. By adjusting the control timing and strength of the reaction pulse, the plate impact velocities can be damped and the motion profile is optimized for either increased life or efficiency.

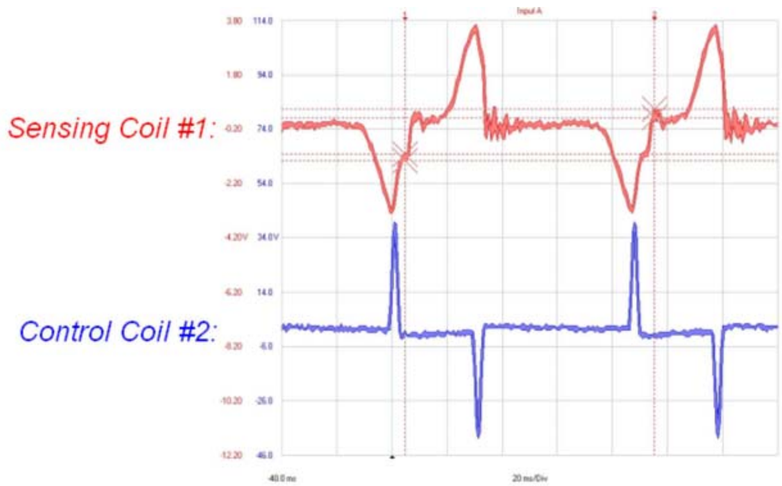

Figure 4-44. Time Trace of Sensing and Control Coils

Figure 4-44 shows the traces that represent the signal voltage (red) and the resultant output voltage (blue). This shows that the controller is acting late in the plate movement (just prior to hitting the seat or guard) to react against the pressure forces. Reaction force is not applied until the plate nearly reaches the guard/seat to reduce valve leakage losses and minimize power requirements. A plot of the controlled versus uncontrolled displacement is shown in Figure 4-45.

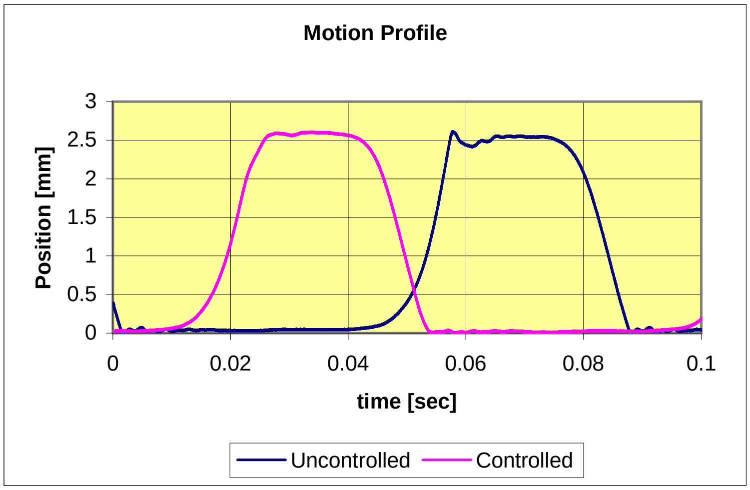

Figure 4-45. Controlled and Uncontrolled Plate Motion

The uncontrolled displacement shown in blue has the characteristic valve motion behavior that has previously been documented. The uncontrolled impact velocities are on the order of 3.0 m/s. As the pressure ratio and lift change, the impact velocities change as well. The controlled (red) displacement plot shows the valve motion under the same operating conditions with the control system operating. One can see in the figure above that the plate impact profile becomes rounded, demonstrating reduced impact velocities. This data was taken using a non-intrusive infrared optical probe to not interfere with the motion of the valve plate. Table 4-8 below shows the impact velocities, percent peak stresses, and relative life comparison of the controlled versus uncontrolled valves.

Table 4-8. Relative Life Comparison

(700 rpm, p2/p1=1.2, p1=20 psig, N_2)

Opening	Impact Velocity	Peak Stress	**Life**
No Control	3.0 m/s	100%	1
Control	1.4 m/s	~40%	~150

Closing	Impact Velocity	Peak Stress	**Life**
No Control	2.4 m/s	100%	1
Control	1.3 m/s	~50%	~100

Results show a significant reduction of impact velocities when the semi-active valve control is operational versus uncontrolled plate motion. Reduction of impact velocities by 50% are easily achieved. These reduced impact velocities can effectively eliminate high cycle fatigue as a cause of valve failures. Clearly, the semi-active valve has the potential to significantly reduce plate impact velocities and thus increase valve fatigue life.

Thus, the semi-active valve concept was functionally demonstrated. The next program step is to develop a compact prototype version with reduced mechanical complexity and possibly a self-powering mechanism.

4.5.4 PASSIVE ROTATING VALVE PROOF-OF-CONCEPT

The "Screw/Cam Thread Rotary Valve" concept is a novel valve idea with the potential to improve on existing valve technology by reducing pressure drop and increasing compressor efficiency. The objective of the current work included completing the detailed design and fabrication of a breadboard rotating passive valve based on the "Screw/Cam Thread Rotary Valve" concept. In addition, mechanical testing was performed on the breadboard rotating passive valve to verify mechanical integrity at realistic compressor speeds. Finally, flow capacity testing was performed to verify the improved aerodynamic performance provided by the rotating passive valve in comparison to a traditional plate valve.

The objective of the "Screw/Cam Thread Rotary Valve" (i.e., "rotating passive valve") concept is to improve upon the efficiency of the traditional passive plate valve design by providing for a flow path with reduced resistance (pressure drop) in the valve open position. The essential feature of the rotating passive valve concept is a valve plate that simultaneously lifts and rotates to provide a straight-through flow path when the valve is open. The straight-through flow path will reduce pressure drop through the valve and increase compressor efficiency. In comparison, the flow in traditional plate valves is forced to complete a more complex path, as it must complete two 90-degree bends as it travels through the seat, around the lifted plate, and out the guard.

Conceptual models of the rotating passive valve concept are shown in Figure 4-46. In addition, pictures of a rapid prototype of the passive valve concept are shown in Figure 4-47.

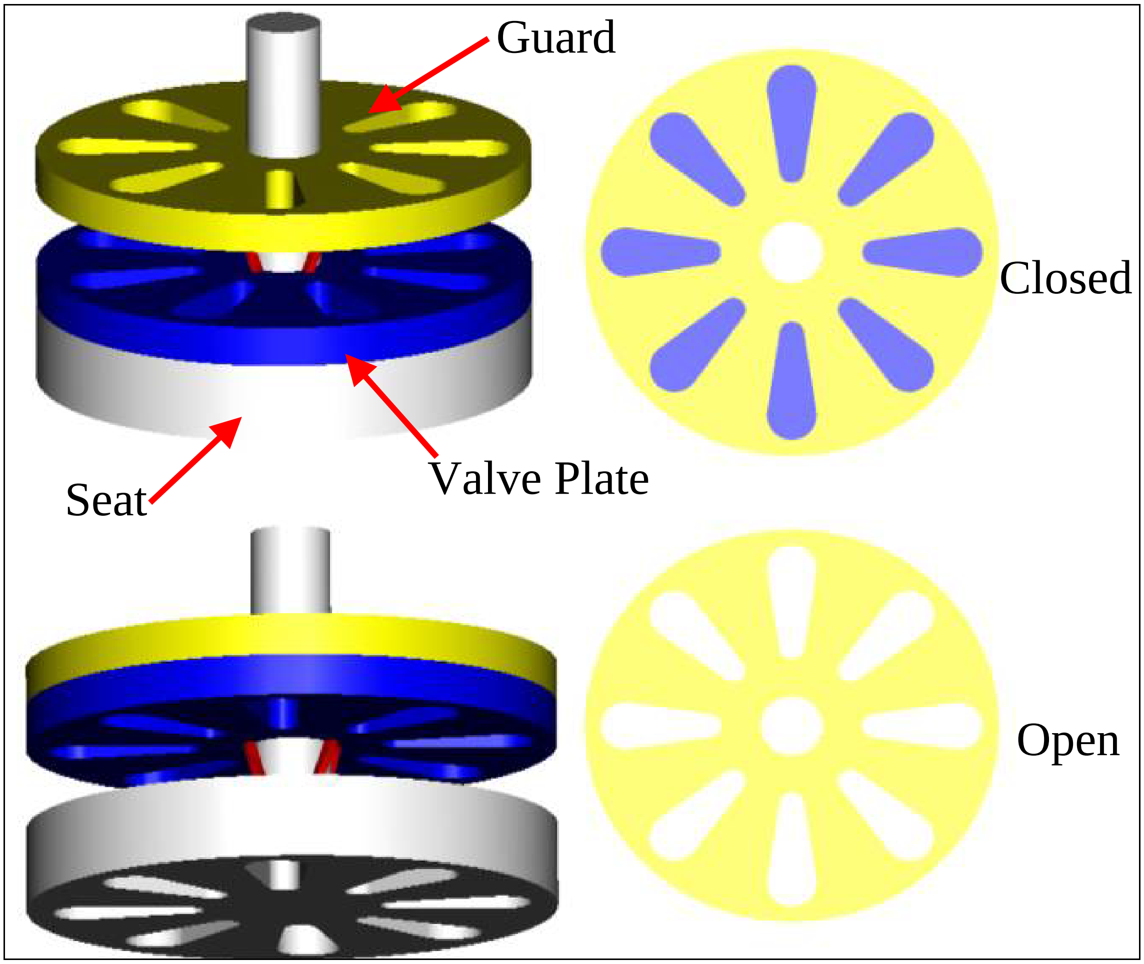

Figure 4-46. Models of the Rotating Passive Valve Concept

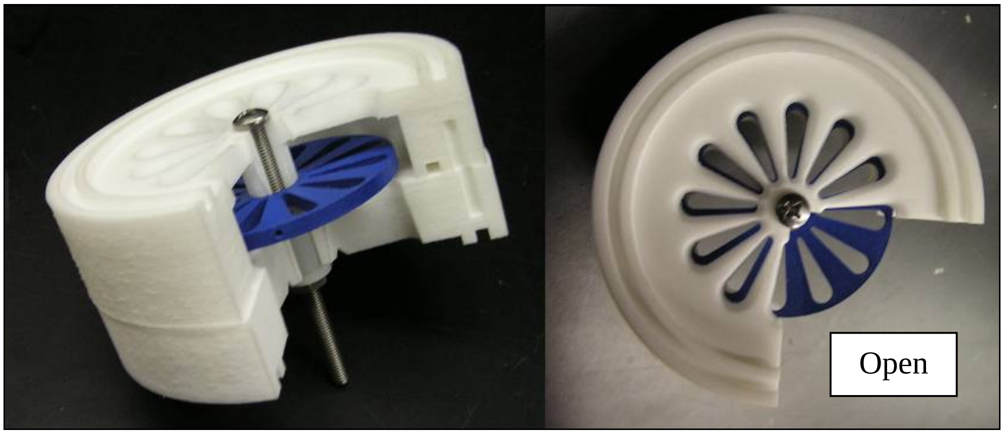

Figure 4-47. Pictures of a Rapid Prototype of the Rotating Passive Valve Concept

4.5.5 Rotating Valve Design

The design constraints considered in the detailed design of the breadboard rotating passive valve included: the valve must be fully passive; the valve plate must lift and rotate simultaneously to the fully open position; the flow path must be straight and unobstructed in the fully open position; the valve will require a restoring force to return the plate to the closed position; and the design must be mechanically robust. In addition, the design should allow for retrofit to current valves.

The current breadboard design includes a valve guard component. However, future passive rotating valve designs may not require a guard to stop the translation of the valve plate as the aerodynamic forces tending to push the valve open become small (and balanced by the spring restoring force) when the valve is in fully open position. The elimination of the guard could significantly increase valve plate life by removing the opening impact against the guard currently experienced in traditional plate valves.

A 3-D CAD model and drawings of the breadboard rotating passive valve design are given in Figure 4-48 and Figure 4-49, respectively. In addition, pictures of the fabricated breadboard rotating passive valve are given in Figure 4-50. As called for by the rotating valve concept, the essential feature of the breadboard rotating passive valve is a valve plate that simultaneously lifts and rotates to provide a "straight-through" flow path when the valve is open. The breadboard valve was sized based on the envelope dimensions of a traditional 6-inch plate valve.

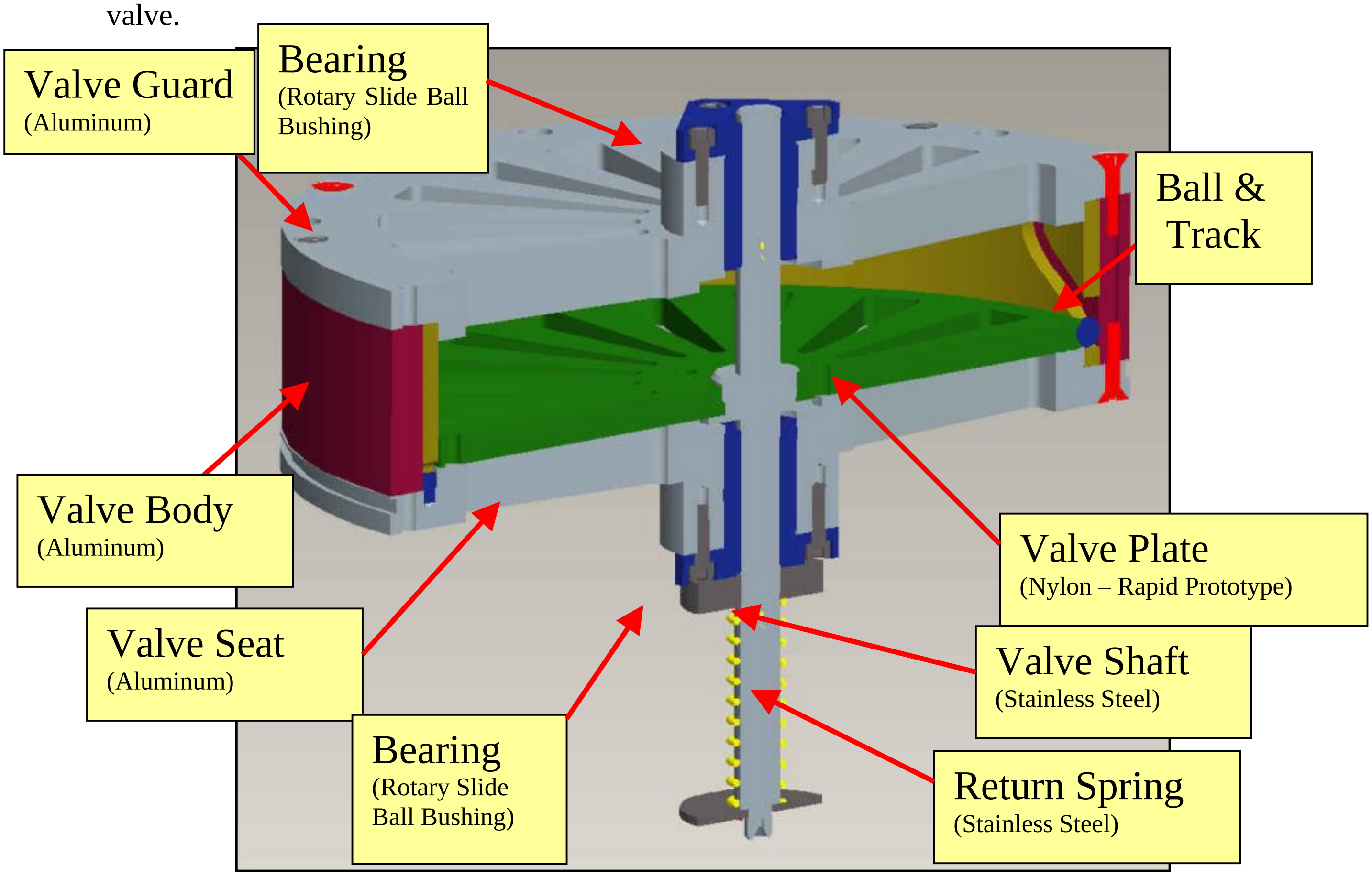

Figure 4-48. 3-D CAD Model of the Breadboard Rotating Passive Valve

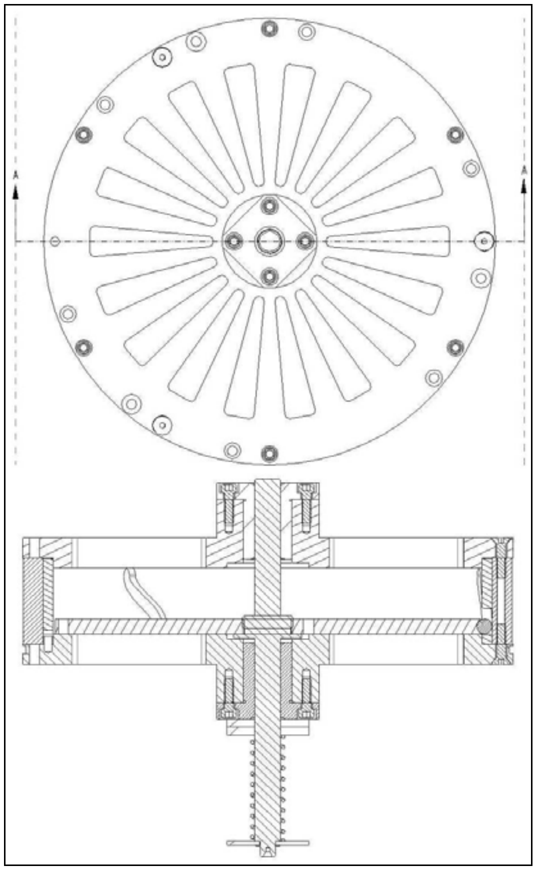

Figure 4-49. Drawing of the Breadboard Rotating Passive Valve

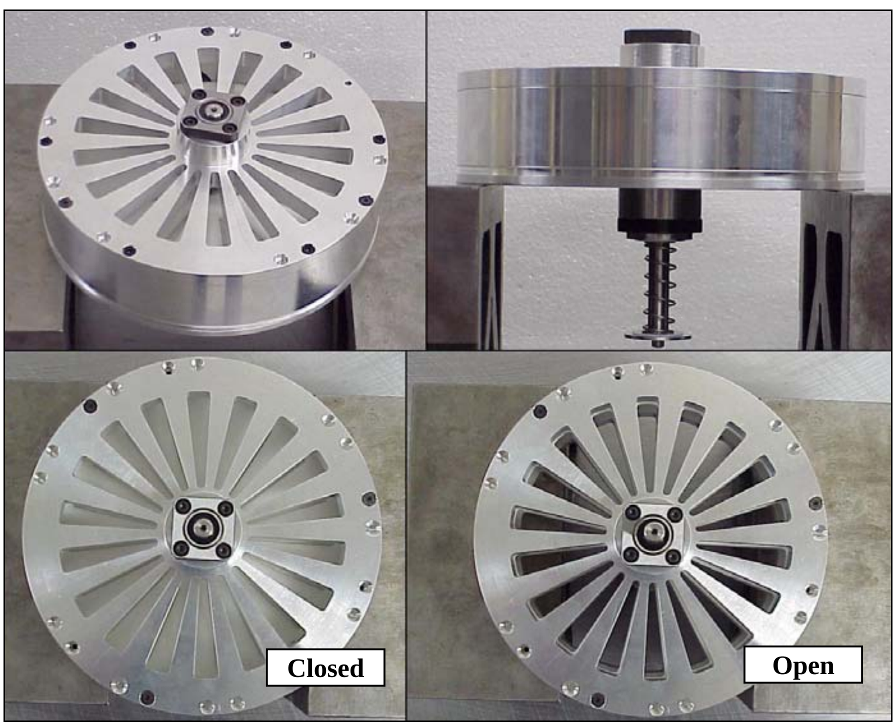

Figure 4-50. Pictures of the Breadboard Rotating Passive Valve

The valve plate is fixed to a stainless steel valve shaft with a threaded connection. The valve shaft is supported by two rotary slide ball bushings (ISOTECH P/N SREK 8) with one bushing installed in the valve seat and one bushing installed in the valve guard. The rotary slide ball bushings are a commercial, off-the-shelf item, which allow both translational (linear) and rotational motion of the shaft and plate relative to the fixed seat and guard. The purpose of the two rotary slide ball bushings is to ensure that the shaft and plate assembly remains aligned and free to move as the valve is actuated. Three ceramic balls trapped between the outside diameter of the plate and the valve body control the rotation of the valve shaft and plate assembly. These balls roll in tracks (equally spaced in 120 degree intervals around the valve), which prescribe the linkage between the valve plate lift (translation) and rotation.

The current breadboard rotating passive valve has a lift of 0.625 inches and a corresponding rotation of 10 degrees to fully open. This 10-degree rotation allows the 18 “flower-petal” shaped openings in the valve plate to alternatively cover (i.e., valve closed) and uncover (i.e., valve opened) similarly shaped openings in valve seat. The majority of the breadboard valve components were fabricated of aluminum to facilitate rapid and cost-effective fabrication of the breadboard valve. In addition, the valve plate was made of a nylon-based rapid prototype material finish machined to size. However, the plate could be made of more realistic valve plate materials (i.e., PEEK, nylon, steel).

A standard, coil spring installed around the valve shaft provides the restoring force to return the plate to the closed position. In the breadboard valve, the return spring was packaged external to the valve to facilitate easy spring preload adjustment (with spacers) or spring rate adjustment (by replacement springs), if needed.

4.5.6 TESTING OF ROTATING VALVE

Mechanical testing was performed on the breadboard rotating passive valve to verify mechanical integrity at realistic compressor speeds. The rotating passive valve assembly was mounted to a loudspeaker, which was used as a linear actuator for the mechanical testing. The speaker’s driven cone was attached to the valve shaft with a ball joint coupling so that the speaker would drive the valve shaft translation but not impede the valve shaft rotation. The valve spring was removed during the mechanical testing.

The loudspeaker was driven using a standard DC power supply and amplifier with a standard function generator to providing the input frequency and signal shape (sine or square) command to the amplifier. Pictures of the valve assembled to the loudspeaker and the test setup are given in Figure 4-51 and Figure 4-52.

The mechanical testing demonstrated good valve operation. The valve was operated at short durations at various frequencies from 1 to 20 Hz and then operated at 20 Hz (square wave input, equivalent to 1,200 RPM) for about 10 minutes. The valve opened and closed fully and the plate moved smoothly with no tilting and/or sticking. No mechanical failures or deteriorations were noticed. The full speed valve motion was observed visually with the aid of a strobe light. In addition, normal-speed and high-speed videos of mechanical testing were also captured.

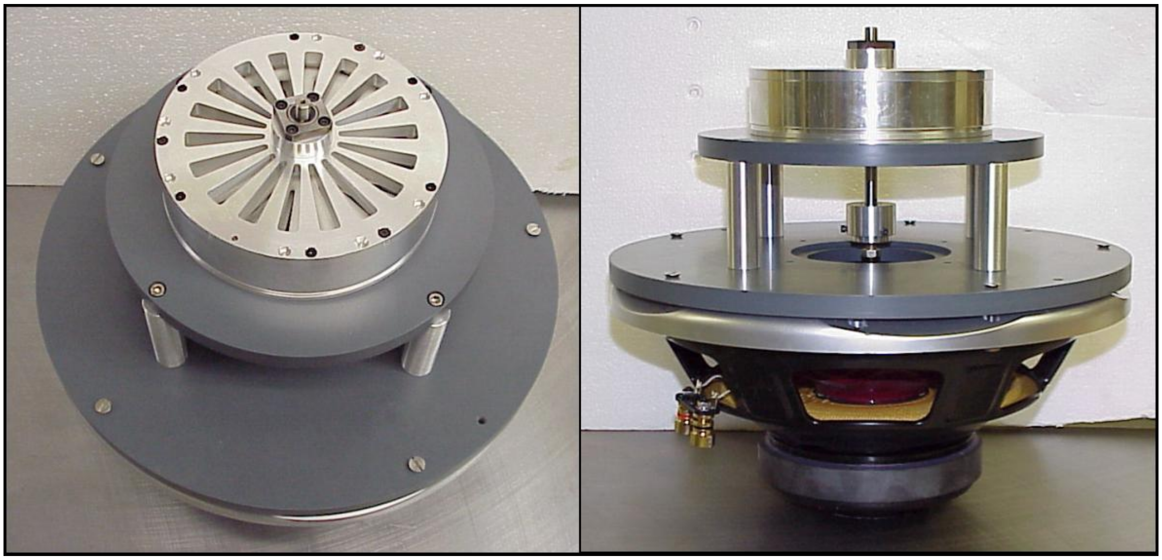

Figure 4-51. Pictures of the Breadboard Rotating Passive Valve Installed on Loudspeaker

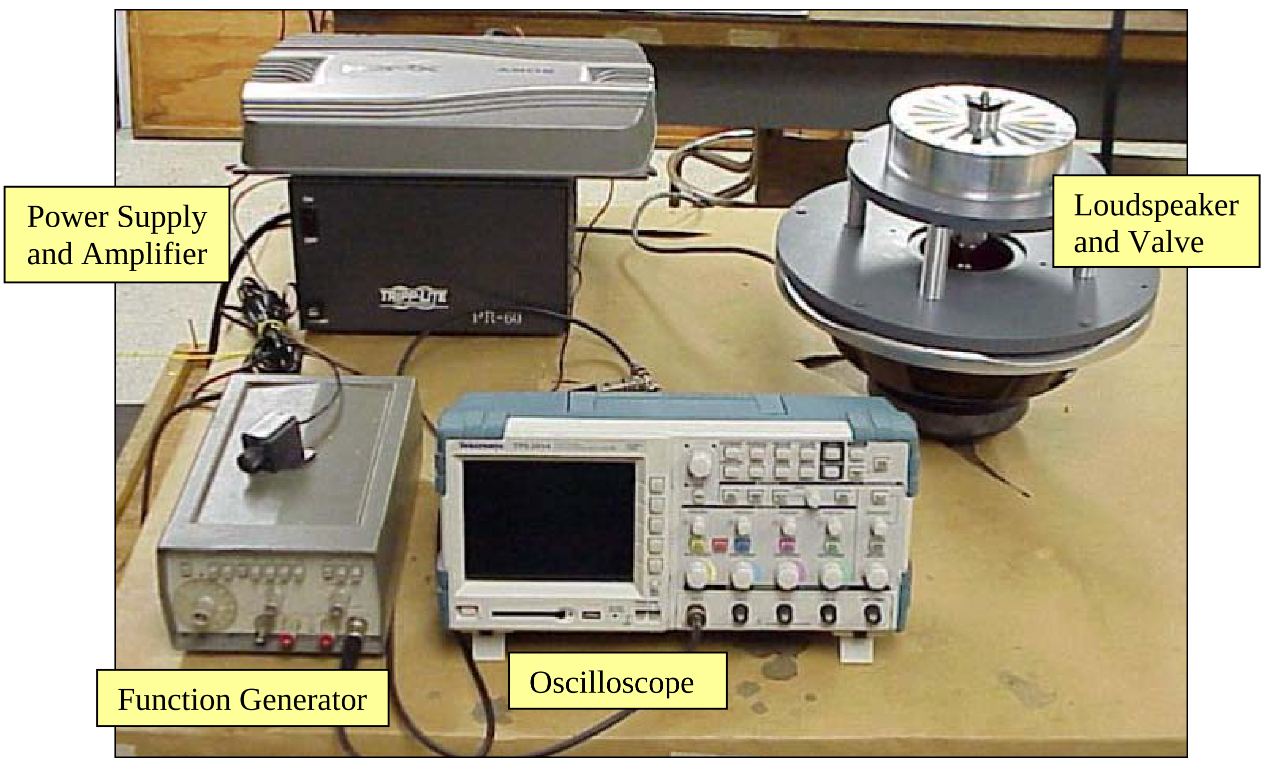

Figure 4-52. Picture of the Mechanical Testing Setup

Flow capacity (flow rate versus pressure drop) testing was performed on the breadboard rotating passive valve to verify improved aerodynamic performance in comparison to a traditional plate valve. The flow capacity testing was completed with the valve plate held fixed at 25% intervals of valve lift (translation). The flow testing was conducted at the low-pressure flow facility using nitrogen as the test fluid. Pictures of the flow test fixture and test set-up are given in Figure 4-53. Flow capacity testing was also completed for a similarly sized traditional plate valve with about 0.10-inch lift in the same facility.

The flow testing demonstrated that the rotating passive valve has the potential for significant aerodynamic improvement in comparison to traditional plate valves. The flow versus pressure drop results, presented in Figure 4-54, demonstrate that the rotating passive valve exhibits on the order of half the pressure drop of a traditional plate valve with both valves in their fully open positions for similar, high flow rates.

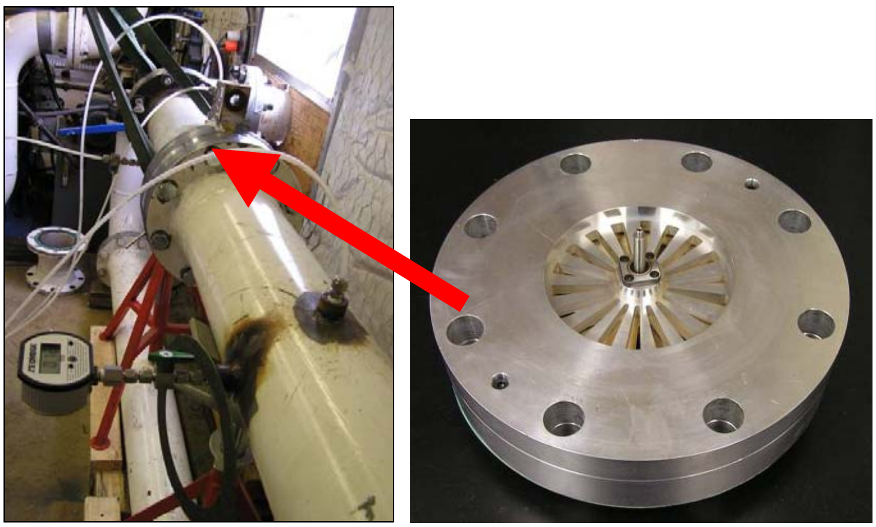

Figure 4-53. Flow Test Setup and Test Fixture

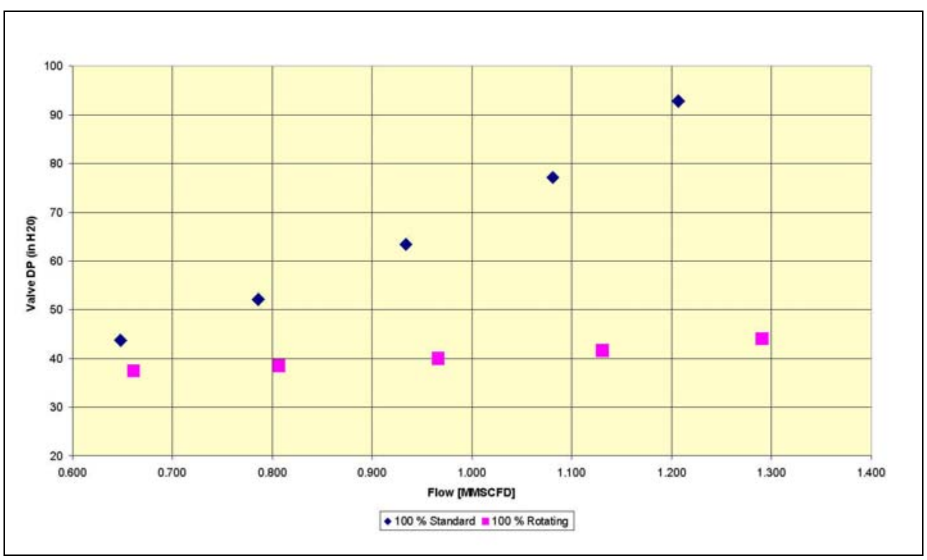

Figure 4-54. Flow Testing Results

4.5.7 ROTATING VALVE SUMMARY

The breadboard version of the rotating passive valve designed, fabricated, and tested in the current work has been demonstrated to function mechanically as desired and has shown potential to have significantly improved aerodynamic performance in comparison to traditional plate valves. The next steps required in this technology development effort include designing and fabricating a prototype to be tested at ARCT test bed compressor and to develop an analytical model to predict the performance and function of the rotating passive concept.

4.5.8 ACTIVE ROTATING VALVE PROOF-OF-CONCEPT

The passive rotating valve concept (described in the previous section) could also be envisioned as an active rotating valve concept. In the active rotating valve concept, an external driver (such as a stepper motor controlled based on a shaft encoder or pressure sensor) rather than pressure-induced valve plate translation could provide the valve plate rotation. The active rotating valve could have very little plate lift (translation) accompanying the valve rotation (i.e., the valve rotation could be largely decoupled from valve translation in the active valve concept). In addition, the mechanical elements responsible for valve plate rotation could be removed from the flow stream. However, a conceptual/preliminary design of this concept has not been considered, and many technical challenges remain. Nonetheless, the active rotating valve concept has significant potential benefits that warrant its consideration in future work.

4.6 RECOMMENDATION FOR FUTURE VALVE WORK

The long-term goal of this program is to develop four specific, market-oriented valve products that aim at improving current and future reciprocating compressor performance as well as operational economics. These products are (1) a valve life prediction program, (2) a semi-active plate valve, (3) a passive rotating valve, and (4) an active rotating valve. They are described in detail in this report. Over the next three to four years, these individual compressor valve products will be advanced from their current development state to fully commercial products to be utilized by the industry.

4.6.1 LIFE VALVE PREDICTION PROGRAM

Over the next 18 months, the current beta version of this code will become a fully operational stand-alone program. To achieve this, a number of tasks must be completed; specifically:

- Field testing of valve plate motion will be performed at, as a minimum, four compression stations using optical probes. During these tests, the three-dimensional motion profile, plate impact velocities, plate impact angles, and plate impact locations will be sampled. The results from these tests will be implemented into the existing statistical model for valve motion versus operating condition and geometry. Based on the results, the current valve motion algorithms will be refined and generalized.
- Additional material properties of PEEK, Nylon and metals must be implemented into the plate life prediction program. This information will be collected from the public domain and SwRI in-house testing.

- The interface of the current valve life code must be improved and more closely integrated with the compressor performance code to make it more user-friendly. A short user-guide and operating instructions for the program must also be developed. The entire program will be made available so that compressor operators can perform plate life and optimization analysis.
- The valve life program must be verified using field data on failed plate valves. A number of parametric studies should be undertaken to assure that general trends of the analysis results agree with physical observations.

4.6.2 SEMI-ACTIVE PLATE VALVE

A physical model for the semi-active plate valve function must be developed, optimized, and verified with test data. A design team, including electronic controls, power, and mechanical engineering experts, will be formed to develop a prototype semi-active plate valve design. This valve design will be compact, incorporate an integrated controller, and should be self-powering. Once a detailed design has been completed, prototype fabrication will begin. The proper functioning of the prototype semi-active valve will first be verified and refined in the ARCT test bed compressor. Once the prototype has been fully tested, it will be released for field trials at a commercial installation, where a long-term endurance testing will also be performed. During the prototype development phase, a commercialization partner should be identified and integrated into the development team such that the semi-active plate valve can become a commercial product at the end of the field trials.

4.6.3 ROTATING PASSIVE VALVE AND ROTATING ACTIVE VALVE

Various concepts to design a rotating passive and active valve benchscale models will be evaluated for testing in the ARCT test bed compressor. Once benchscale testing has been completed, a design team will be formed to develop a fully functional prototype for additional laboratory testing and field trials. Simultaneously, a physical model of the rotating valve concepts will be developed to optimize the future valve geometry design process.

4.6.4 SPRING RESEARCH

The current FE spring models will be expanded to a fully transient analysis. This model will then be employed to provide a better physical understanding of the failure mechanisms of valve springs. The model can also be employed to perform parametric spring design studies and to develop springs with minimized local spring peak-peak stress levels (i.e., to maximize strengths and life of the valve spring).

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5. SENSORS AND INSTRUMENTS TECHNOLOGY RESULTS

5.1 SENSOR TECHNOLOGY ASSESSMENT

The development of key sensors/instruments to enhance the flexibility, performance and reliability of current and future reciprocating compressors is a crucial component of the ARCT project. Automation in the traditional sense of a compressor installation involves control of systems (lubrication, starting, shutdown, unit valves, loading), locally and remotely allowing reliable unit starts and stops, as well as responding to changes in station ratio and unit torque to meet gas control needs. Modern installations of high speeds, as well as retrofit integral are currently capable of achieving these requirements. A significant cost of implementing the components of current automation requirements is sensors and wiring for local, station, and remote control. Wireless communications has the potential to reduce the cost of control implementation. Automation in the context of this project will likely involve local closed loop control of active systems (i.e., active pulsation control). Finally, development of advanced compressor technology will require performance measurement systems and instrumentation systems not currently available to evaluate and optimize the design. Technology, which will improve the efficiency of the cylinder, either directly (i.e., valves) or indirectly (bottles), will require continuous compressor HP determination, as well as loss evaluation of the specific component. Currently, cylinder based HP determination of high-speed compressors is seriously hampered by channel distortion effects that can only be overcome with flush mounted sensors. Installation losses from bottle designs cannot be determined from differential pressure measurements alone, but require accurate unit flow knowledge, which is difficult to obtain in a highly modulating flow environment. This chapter examines in more detail the instrument and measurement issues associated with compressor operation, as well as examining the likely measurement system needs to achieve many of the goals of the ARCT project.

5.1.1 LOAD CONTROL

Load control is a critical component to compressor operation. From a pipeline operation perspective, variation in station flow is required to meet pipeline delivery commitments, as well as implement company strategies for optimal operation (i.e., line packing, load anticipation). From a unit perspective, load control involves reducing unit flow (through unloader or speed) to operate as close as possible to the design torque limit without overloading the compressor or driver. Critical limits on any load map curve are rod load limits and HP/torque limits for any given station suction and discharge pressure. Gas control generally will establish the units within a station that must be operated to achieve pipeline flow targets. Local unit control will establish load step or speed requirements to limit rod loads or achieve torque control.

The relationship between torque (driver limited) and unit operation (pipeline operation driven) can be established in a variety of ways. For the existing fleet of integral units, performance testing has generally been completed on the units to establish relationships between unit operation and HP/torque. Either this testing can be very extensive, allowing interpolation models or reduced to generate physics based predictive models. Accuracy in the ultimate prediction of unit HP is varied. Recent testing as part of the DOE Infrastructure project has indicated errors associated with reduced testing, and extrapolation can exceed 15% to 20%. Recent testing on extensive interpolation models indicate HP differences between real time compressor cylinder data and station load control predictions of less than 5%. Minor unit overloads can generally be accommodated by margins in rod load design limits. Under-loading translates directly into lost capacity. In either case, error in HP prediction will result in unit operation off ideal design targets.

For slow speed integral compressors, establishment of the load curves based on performance measurements is a well-established, mature procedure. Cylinder PV (Pressure-Volume) cards are required, and commercially available systems are available to acquire the data. Distortions associated with measured PV cards resulting from mounting the pressure sensor on access valves (commonly referred to as channel measurement error) are linear in nature for most pipeline applications, and methodologies for channel correction have been developed and incorporated into the available measurement instruments. For high-speed units, traditional cylinder based PV card determination using conventional transducers on access valves result in significant non-linear distortion [Harris and Edlund, 1999][22]. The distortion associated with cylinder pressure measurements in high speed, high ratio (hs/hr) units can be substantially different from the common "ringing" found on the suction and discharge ends of the pressure record on low speed/low ratio units. On hs/hr units, the cylinder pressure measurements can be significantly distorted in phase, with the phase distortion not uniformly distributed around the PV card. The overall effect is to produce a PV card, which is "skinnier" and which will calculate less horsepower than the compressor cylinder is actually consuming, and less flow than the cylinder is actually producing (Figure 5-1). In this situation, load curves will be significantly in error and large overloading can occur. The only reliable approach to PV based measurements in high-speed applications is to ensure that the sensors are "flush mounted" to the cylinder, eliminating the channel and the associated distortion. Using current technology, flush mounting the sensors requires that the sensor be extended into the measurement channel with the unit off-line and blown down. This approach also requires that the sensors be very carefully calibrated with temperature effects considered. Since most PV card type performance measurements are done on a roving basis with a reduced number of probes, this approach is not generally implemented by industry. Ariel Corporation has offered and delivered a few large compressors with advanced

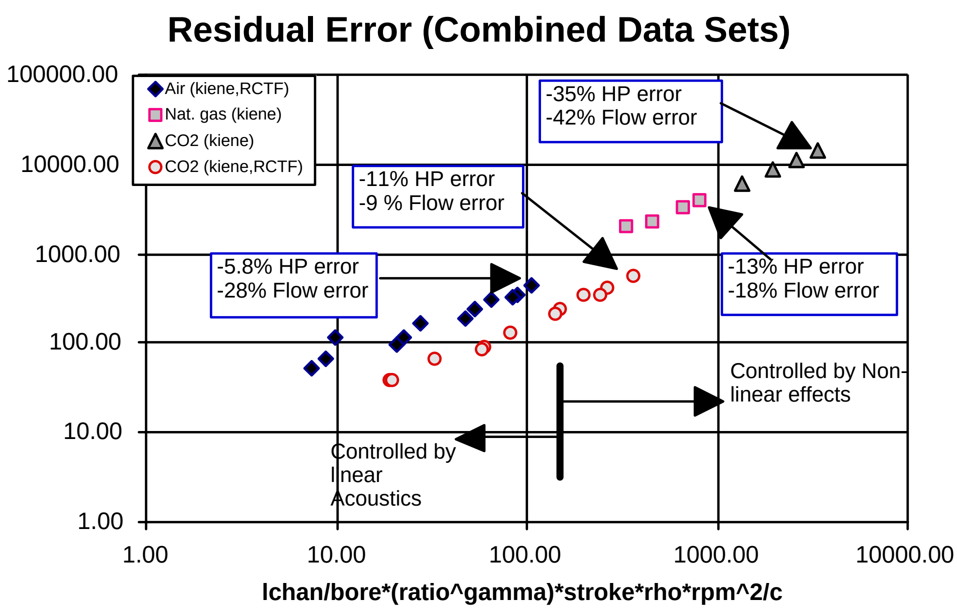

Figure 5-1. Channel Measurement Errors

measurement systems using flush mounted cylinder pressure measurements and real-time HP determination. Unfortunately, cylinder pressure sensor reliability, and the reluctance of industry to incorporate the technology into the control strategy technology resulted in Ariel removing this as an option for future units.

An enabling technology for real-time HP determination in both high-speed and low-speed applications is the development of the Rod Load Monitor (Figure 5-2). This developing technology directly measures the compressor rod forces and allows real-time determination of rod HP. This HP estimate will be slightly higher than cylinder PV based measurements due to cylinder frictional losses (on the order of 2% to 4%). This emerging technology has yet to be fully evaluated by industry.

Figure 5-2. Self-Powered Rod Load Monitor on Slow-Speed Integral Unit

Load control will continue to be a critical component of future compression. Horsepower replacement projects, which will replace multiple smaller units with fewer larger units, will require more accurate real-time HP/torque and flow knowledge of the units.

5.1.2 SYSTEM EFFICIENCY DETERMINATION

A wide range of unit efficiencies exists in the pipeline industry (Figure 5-3). This variation reflects differences in compressor design, unit operation, and pulsation manifold design. A systematic program to improve the lower performing units can result in significant reductions in lost HP and gains in pipeline capacity. For future compressor designs and

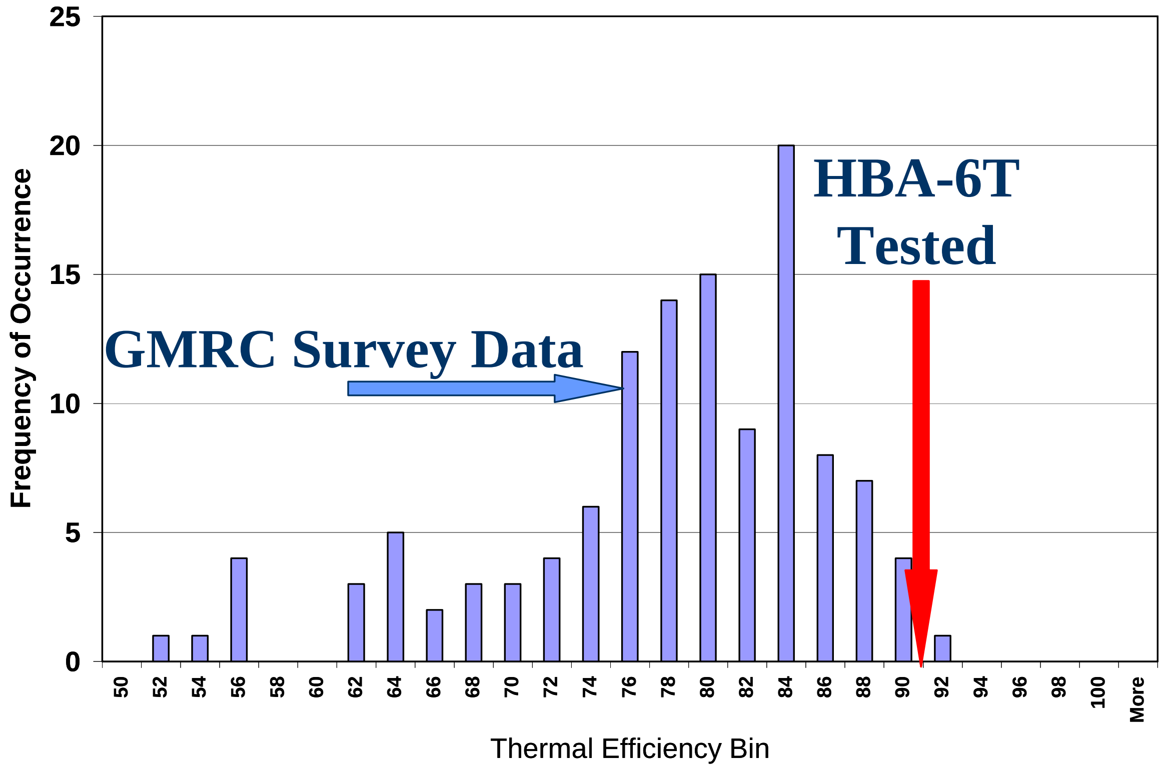

Figure 5-3. Survey Summary of Industry Low-Speed Compressor Efficiency

installations, manifold design must be improved to increase the operating efficiency, without negatively affecting vibration and unit reliability. Manifold losses are very difficult to quantify and document. Simple pressure drop measurements cannot be used since flows are highly modulated close to the compressor. Development of optimal manifold designs (a significant component of the ARCT objectives), particularly those with minimal pressure drop will require measurements of both cylinder HP, and delivered gas HP to refine and compare actual performance with predicted values. Enthalpy rise HP is obtained from temperature and pressure rise across the unit, combined with unit flow. Typically, individual unit flows are not measured at compressor stations. Accurate unit flows are difficult to obtain since in many cases flow modulations are still present upstream and downstream of the manifold system. These flow modulations will produce well documented errors in flow rate estimates, potentially degrading the desired benefit of documenting unit flows. However, the state-of-the-art in flow measurement has advanced greatly over the last ten years. In pipeline applications, highly accurate custody transfer meters are used to measure the pipeline flow with an accuracy of ±1.0% or better. These meters typically require upstream flow conditioning and a steady, non-pulsating flow. With a natural gas flow calibration, turbine flow meters and ultrasonic flow meters are capable of an accuracy of 0.50% or better, with a repeatability of 0.05% to 0.25%. The flow stream near a natural gas reciprocating compressor typically has a significant level of pulsations, with a disturbed, undeveloped velocity profile. A flow meter in this environment must either be capable of discerning the pulsation-induced error on the flow rate or be insensitive to pulsation-induced errors. In addition, the flow meter must be fairly robust because the natural gas may contain varying levels of corrosives and contaminants. Non-intrusive flow measurement methods are preferred for this application, but small flange-mounted meters are also an option. In addition, the compressor environment is typically very noisy, making ultrasonic transmission more difficult for ultrasonic flow measurement. Various flow measurement methods that may work at a reciprocating gas compressor station environment are considered in the discussion to follow.

Turbine flow meters offer highly repeatable flow measurement, but the output from a turbine meter must be adjusted for the modulations caused by pulsating flow. Similar to other meters, turbine meter error may vary as much as 40% due to pulsations. A method was developed for determining the effect of pulsations on the turbine meter flow measurement [McKee and Sparks, 1992][23]. This method requires measuring the time between pulses to determine the rotor period modulations. The difficulty in the method is in correlating the meter error due to pulsations with the rotor period modulation. A baseline measurement of the turbine flow meter performance in non-pulsating flow is necessary, in order to determine the pulsation-induced error on the turbine meter. Despite these constraints, turbine meter measurement is worth considering for this application. Additional research in determining the pulsation level from rotor pulse modulation may prove to be beneficial, in improving the accuracy of the method and its practical applications.

Ultrasonic flow meters are also a consideration, if fast response transmitters could be used to provide a measure of instantaneous velocity. The current ultrasonic flow meters used in natural gas applications rely on multiple pairs of transducers to transmit and receive ultrasonic noise across the flow stream. For a given pipe diameter, the velocity is determined by the transit time difference for the ultrasonic signal to travel in the forward and reverse flow directions. Figure 5-4 shows an example of a single pair of ultrasonic transducers and the typical path that ultrasonic noise travels across the flow stream. An example of how the transit time is derived from the ultrasonic signals is shown in Figure 5-5 [Southern Gas Association Short Course, 2004][24].

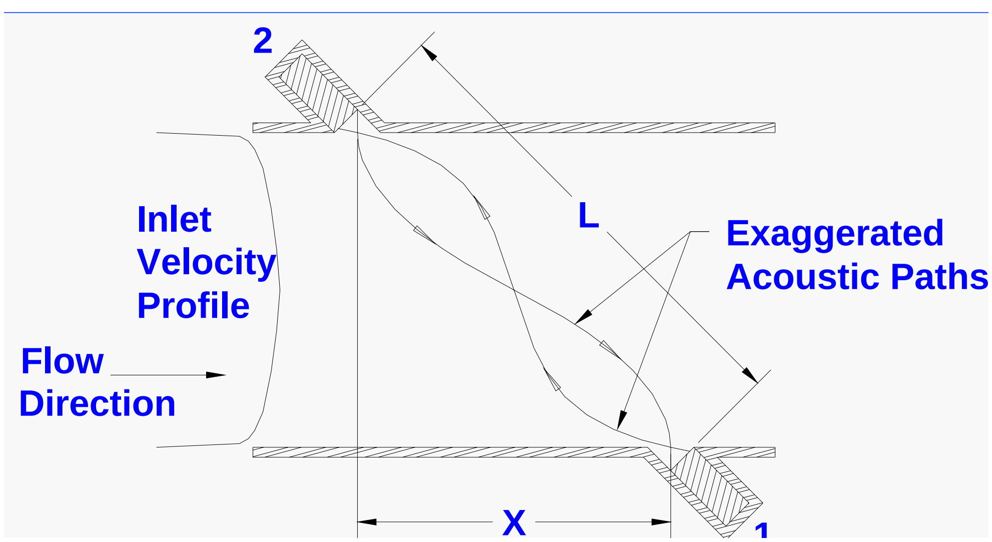

Figure 5-4. Ultrasonic Transducers and Ultrasonic Acoustic Path

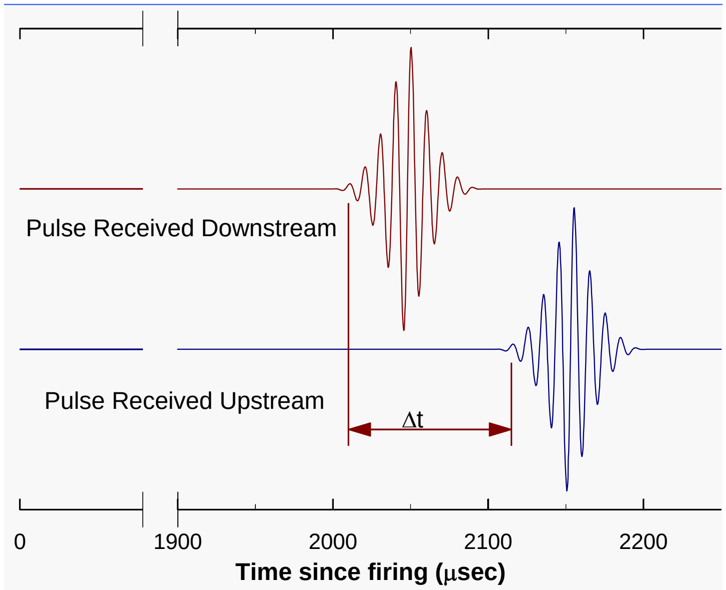

Figure 5-5. Ultrasonic Signal Processing with Associated Transit Time

In measuring pulsating flow with an ultrasonic meter, the instantaneous measurement of the velocity from each transducer could be measured in conjunction with the average flow rate determined by the meter. The instantaneous measurements would allow the periodic variations in flow due to pulsations to be determined. Ultrasonic flow meters are either flange-mounted or made to "clamp on" to the outside pipe wall in a non-intrusive application of the technology. Since ultrasonic flow measurement has not been tested successfully in the compressor station environment, it still requires some development. In addition, the ultrasonic meter must be capable of filtering out the environmental noise in order to determine the transducer ultrasonic signal.

In considering the current custody transfer flow measurement technology, standard orifice meters should also be considered for a compressor station flow application. If a Square Root Error Indicator (SREI) is used in combination with proper gage line installation, field-testing has shown that the orifice meter can be used reliably to measure flow and predict the pulsation level [Durke and McKee, 1986][25]. This method relies on the inertial error being somewhat negligible, which is the case in some industrial applications. However, the orifice meter typically will not offer the same level of accuracy and repeatability as a turbine or ultrasonic flow meter.

Other non-intrusive flow measurement techniques should also be considered. These techniques are not used for pipeline custody transfer metering because they typically measure to within ±3% to 5% accuracy. However, they may be applicable for measuring flow near a compressor, because they do not require changing the piping and may still provide a fair representation of the flow velocity. The non-intrusive techniques that should be considered for the compressor flow measurement application are the use of a multi-hole pressure probe, rugged hot wire anemometry, drag probe, tracer gas, and fiber optic sensors.

A multi-hole pressure probe can be used to measure disturbed or turbulent flow with fairly good accuracy. The multiple ports in the probe are used to sense the pressure and corresponding velocity component at different angles in the flow stream. This method requires fast response pressure transducers in order to characterize the periodic variations in pressure due to the presence of pulsations. In addition, calibration of a flow measurement device of this type must be performed in steady flow conditions. Another disadvantage to this approach is the relatively high cost of additional pressure sensing instrumentation.

Hot wire anemometers are often used to measure local flow velocity. As the gas flows over the sensor, the heat loss is measured. In a wheatstone bridge configuration, the voltage measured in the wire is proportional to the velocity of the gas. This relationship allows the temperature loss to be associated with the local flow velocity. The frequency response of hotwires can be very high based on the size of the wire used in the application. Hot wires work well in less harsh environments and traditionally have not been used for challenging field applications. The small elements (films) of a typical hotwire will need to be mounted on a protective substrate of support for reliable station or pipeline use.

Use of a tracer gas is another viable flow measurement option. Tracer gas testing has been successful in a variety of applications. Typically, a gas is injected into the flow stream. One of two methods may be used to determine flow rate. The first involves calculating the dilution concentration of the injected gas. This method assumes ideal gas behavior, where the flow rate of the stream can be calculated as according to $Q = P/C$. In this relationship, *(P)* represents the flow rate of the injected tracer gas and *(C)* represents the equilibrium tracer gas concentration in a sample taken downstream of the injection point. Typically, sulfur hexa-fluoride (SF_6) is injected using a critical flow nozzle. The samples are analyzed for SF_6 content using electron-capture gas chromatography. The sulfur hexa-fluoride serves as an excellent tracer gas because it is inert, non-reactive, and can be detected in quantities approaching one part per trillion. This method was tested successfully in a number of applications for measuring gas flow rate through a compressor and the precision accuracy was found to be approximately 1.5% [Lagus, et al., 1991][26].

In a second application of a tracer gas method, the time of flight for the tracer gas particles is measured by using detectors near the injection point and at a point further downstream. Radioactive isotopes are used with gamma ray detectors. This method is commonly referred to as a pulse velocity method. The time between the pulses detected by two gamma ray detectors is measured. The distance between the detectors is then used to determine the velocity of the flowing stream. Figure 5-6 shows an illustration of this method, used by Quest TruTec in determining the flow rate of gas in flare lines.

In a recent test at the Metering Research Facility at Southwest Research Institute, Quest TruTec used a radioactive isotope, Argon-41, as a tracer gas. Argon-41 served as a good tracer gas in this application, because it has a very short half-life of 1.8 hours and it is a very strong gamma ray emitter. The proprietary flow measurement equipment demonstrated an error of less than ± 1.0% for high velocities (10-120 ft/sec) and near ±1.1% in lower velocities (less than 5.0 ft/sec). Figure 5-7 shows the results of the tracer gas test at the MRF conducted in June 2004. Similar to the other non-intrusive techniques, this method has the advantage of not disturbing the existing compressor station piping. However, the pulse velocity method for a tracer gas has not been tested in flow streams with high pulsation levels.

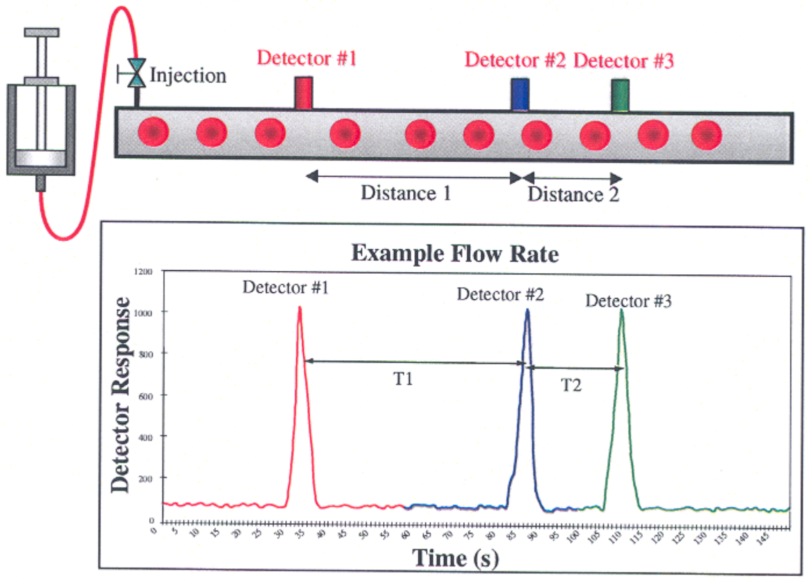

Figure 5-6. Illustration of Pulse Velocity Method for Determining Flow Rate with a Tracer Gas

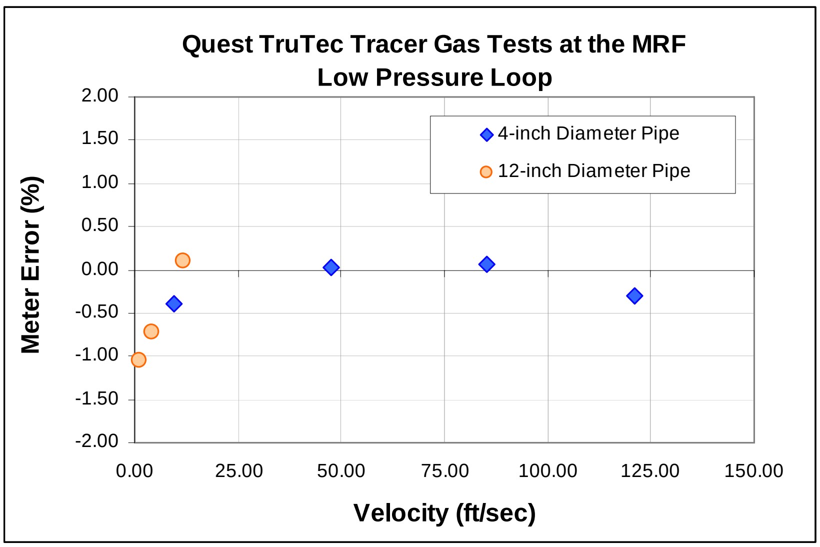

Figure 5-7. Tracer Gas Tests at the Metering Research Facility at Southwest Research Institute

Finally, fiber optic sensors also offer a method of measuring flow through an insertion probe. One such fiber optic sensor uses a fused tapered coupler inside of a probe. The probe is inserted into the flow stream where a miniature transducer is used to detect when the looped coupler bends, as a result of the fluid flow. The flow measurement can be accomplished rather inexpensively and with a reduced amount of signal processing [Chapman, et al., 2003][27]. However, the method may not be reliable in highly turbulent flow, since it has only been tested in controlled laboratory conditions. Based on its simplicity and cost, the method is worth considering for adaptation to the more rigorous flow measurement environment of a compressor station.

A field deployable unit flow measurement system will be required for the load control development effort. It is expected that this system will also be required to evaluate bottle designs from an installation efficiency perspective.

Drag probes measure the force on a bluff body supported in the flow stream. The force (or drag) is measured by a bridge arrangement of strain gages. This technology has the advantage of ruggedness, with the frequency response controlled by the dynamics of the bluff body and support design. This technology is currently being evaluated for surge detection as part of the DOE/GMRC Direct Surge Control project.

5.1.3 System Measurements and Control Development for Active Pulsation Control

ARCT technology ideas for pulsations control include active components, which may shift the frequency response of the filter design to minimize transmitted pulsation for current operating compressor conditions. Local smart sensors and control technology, including predictive and feedback based control algorithms are anticipated. As part of the ARCT development effort, prototype controllers and smart algorithms will be developed and evaluated. It is expected that this development effort will proceed as part of the sensor/automation portion of the ARCT program.

5.1.4 Technology Options

Based on the preceding discussions, several sensor/measurement systems have been identified that may significantly enhance reciprocating compressor operation, or technology development in the ARCT program. These include:

- Field deployable RLM system for load control evaluation.
- High capacity crank strain monitoring system for alignment (possible continuous) evaluation.
- Internal component temperature sensors (self-powered, RF transmitted).
- Continuous shaft torque measurement for HP and load determination on separable units. This may require local RPM variation measurement as well.
- Liquid sensors (more applicable to upstream operation).
- Cylinder pressure probe for PV card determination for high speeds.
- Field deployable valve diagnostic sensors (impact, velocity).
- Self-powering concepts (from strain, vibration, and flow).
- Unit flow measurements for project needs.

5.2 SENSOR NEW CONCEPTS DISCUSSION

The top priority sensor selected for investigation was the Continuous Torque Sensor (CTS). This sensor is described in more detail in the following section.

5.2.1 CONTINUOUS TORQUE SENSOR (CTS)

Real time compressor torque acquired from high speed separable machines would improve both the ability to optimize loading, and allow closer control to peak torque limits. Both potential benefits can improve capacity and efficiency. Accurate cylinder based performance/torque curves are difficult to obtain from high-speed machines. The basic concept for the CTS is to measure mean operating torque from existing coupling designs and installations. The basic design features of the CTS concept are:

- Measure torque using well-established strain gage technology on existing couplings.
- Use self-powering concepts previously developed and refined in the Rod Load Monitor (RLM).
- Integrate sensor and load path into the instrument removing the need to mount gages on the coupling. This will allow the sensor to be completely integrated and field-mounted on the coupling without the need for instrument technicians to lay sensitive strain gages on the coupling.
- Use existing wireless technology and transmit the mean torque levels to the fixed receiver mounted near the coupling.
- Leverage existing technology from previously developed and proven devices, specifically, the Strain Data Capture Module (SDCM) and the Rod Load Monitor (RLM) to allow development and evaluation of a rapid prototype of the device.

Figure 5-8 illustrates the key components to the CTS. On the rotating element (coupling), the CTS transmitter extracts power from the magnetic array, powers the gage and rotating electronics, and transmits the mean torque levels. Key features of the CTS are the self-powered concept, integration of the sensor gage into the housing assembly, and use of a continuous, non-resonant antenna to transmit the data. The self-powering mechanism leverages the previously developed linear magnetic array technology developed for the RLM. A key difference in this

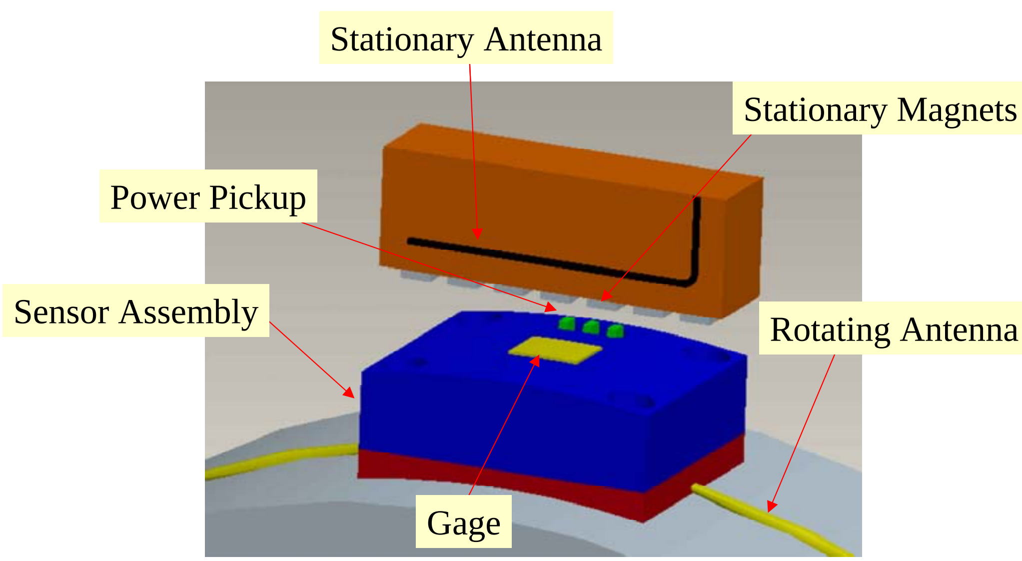

Figure 5-8. Torque Sensor Concept

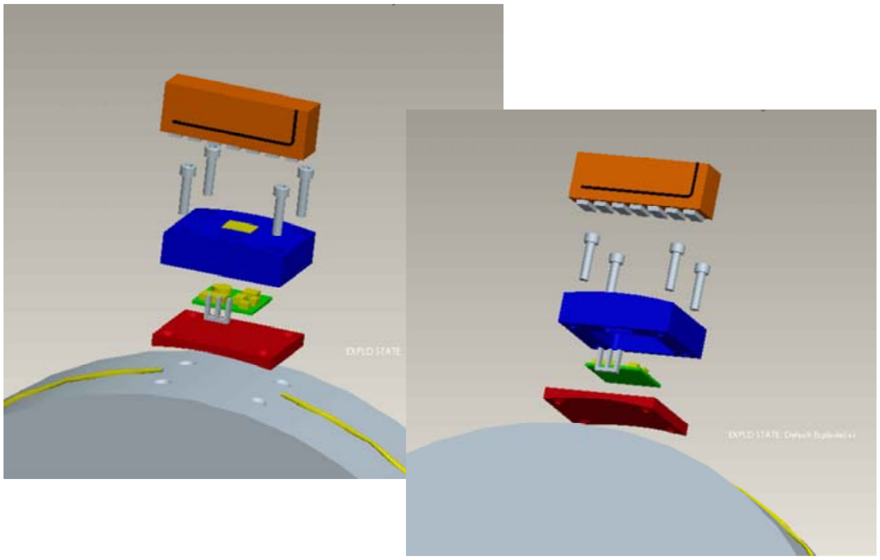

Figure 5-9. Assembly View

device is that the magnetic array system must allow for axial growth of the rotating shaft without loss of power generation. Figure 5-9 presents an exploded view of the device. Note that the housing and gage use a fixed geometry, while the base mounting plate (transition piece) will change based on the geometry of the coupling. The housing is mounted to the base plate, which is simultaneously mounted to the coupling using the same through bolt arrangement. The power pickup coil (Figure 5-10) generates a large amplitude voltage pulse for each north/south pair of the stationary magnetic array (Figure 5-11). The receiver side antenna is mounted on the stationary array and will allow for continuous pickup of the mean torque levels, since the transmitter side antenna will extend around the coupling. Figure 5-12 illustrates the design allowing shear stress transfer from the coupling to the instrument shear gage pair. As shown in Figure 5-13, the sensor housing and integral shear sensitive gage is mounted to the transition piece, which must be custom sized for each coupling.

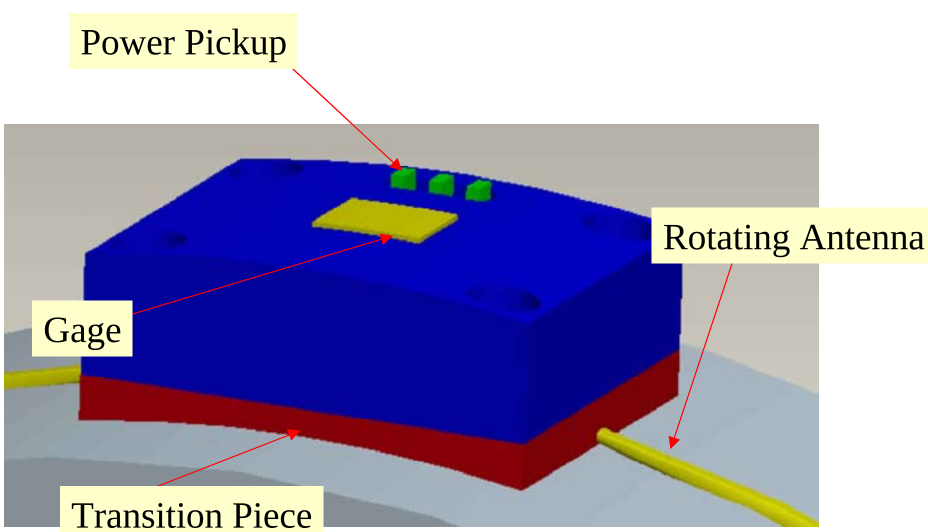

Figure 5-10. Rotating Instrument

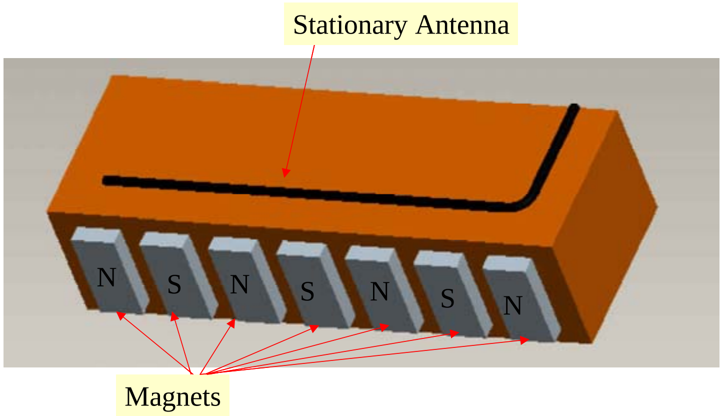

Figure 5-11. Stationary Instrument

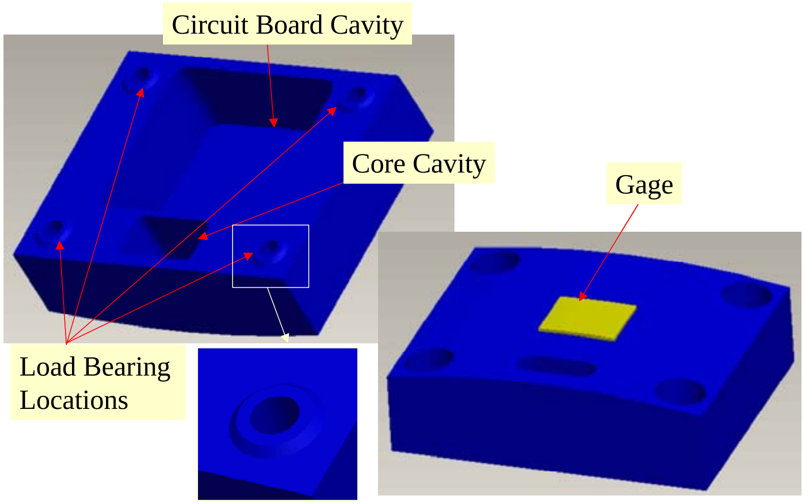

Figure 5-12. Sensor Mount Layout

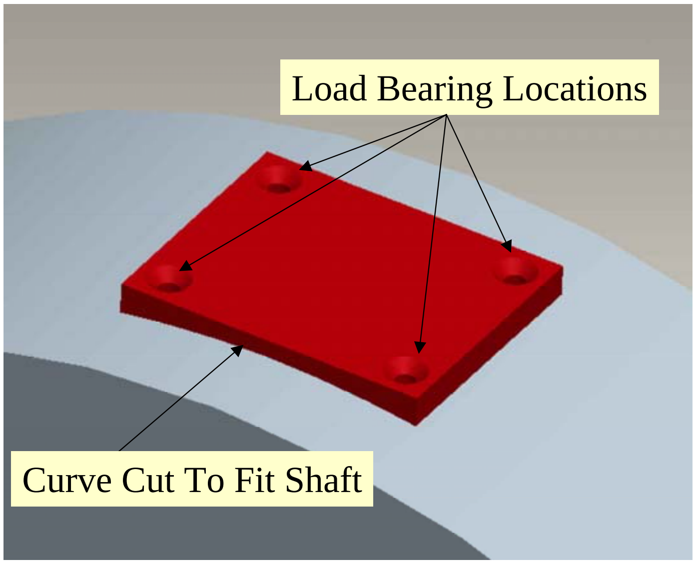

Figure 5-13. Transition Piece

The design of the prototype CTS leverages several existing and functioning instruments recently developed at SwRI. This leverage of existing instrument technology will allow development and evaluation of a prototype with significantly less cost and risk. The leveraged technology includes:

- Existing power generation magnetic array technology. The CTS will utilize significantly less continuous power than the RLM. It is anticipated that the proposed CTS will measure and transmit strain gage based torque levels on the order of once every few seconds. This will allow the unit to power down key components, generate and store power from the coil, power up, acquire and transmit the data, then go back to low power, conserve mode. This power down sequence is not incorporated into the RLM, which is continuously sampling and transmitting data.
- The software microprocessor and associated software will be very close to that in the RLM. This will significantly reduce the effort to generate and evaluate a functioning prototype.
- The RF link required to transmit the data has already been developed and evaluated in the RLM.
- A 16-bit analog to digital converter technology will be used, which has been developed and implemented in the second-generation strain data capture modules (used for crankshaft stress determination).

A unique feature of the CTS will be the continued transmission of strain data after the compressor is off. This will occur until the storage capability of the onboard super capacitor is consumed. This transmission of the mean torque level after the compressor is off will allow for offset (drift) compensation.

Since the device must be high gain to measure mean torque levels on the existing coupling, careful attention to drift consideration must be made. The design will use a self-calibrating 16-bit analog to digital converter with built-in low pass filters. This converter will also allow a ratio-metric based excitation/voltage reference to be incorporated. This will minimize the effects of temperature-based drift in the electronics affecting the mean strain levels at the gage.

A key feature of the design, which must be carefully evaluated, is the concept to incorporate the gage into the housing and transmission of the shear stress from the coupling housing to the instrument itself. The significant advantage of this portion of the design is that the instrument is self-contained, without the need to install strain gages on the coupling housing itself. This approach has been evaluated on a separate instrument project for pure axial loading and has shown good linearity.

5.3 SENSOR CONCEPT SELECTION

The ARCT sensor technology are categorized as follows:

- Traditional automation concepts for control including temperature, pressure, status (i.e., valve position), and the possibility of rapidly evolving wireless technology reducing the cost of installing sensors of this type for enhanced diagnostic analysis and process control.

- Sensor technology to enhance performance, control, and/or operation of reciprocating compressors. This technology, as discussed in the previous section, includes enhanced cylinder PV card determination (currently not possible for high speed compressors), accurate unit flow measurements, accurate HP determination, independent of pressure measurements, and local measurement/control systems for active components. Foreseeable active components that will require integrated measurement/control include, but are not limited to, the tunable side branch absorber, and the active or semi-active valve technology.

In the area of traditional automation concepts, available RF technology was identified that would potentially allow low cost temperature measurement throughout a compressor system. This smart, low powered battery powered technology allows networking for low bandwidth communications. This is defined as "mote" technology, which is an intelligent communication architecture allowing arrays of sensors, which generate a self-reconfigurable network for reliable communications. Application for pipeline service would include low cost individual valve cap temperature measurement for valve leak determination. Significant RF noise exists inside the building of integral compressor units. Reliable RF links are difficult to establish and maintain. For this reason, recommendations were made to develop a test program for commercially available "mote" based temperature sensors inside compressor building. The recommended program would piggyback existing field test efforts and evaluate the reliability for multiple sensors communicating simultaneously in a "confined" RF rich environment.

5.3.1 CONTINUOUS TORQUE SENSOR (CTS) PROOF-OF-CONCEPT

The breadboard CTS is a smart, microprocessor (μP) based instrument. It is built around the Microchip 16F88 RISC based family of processors. The μP controls the data acquisition sample rate, the communications baud rate, and power down sleep interval. The CTS is a combined 3.3, 3-volt device, which periodically powers the full bridge strain gages, the RF communications link, and onboard 10-bit digital temperature sensor IC.

The device is powered by the rectification of the AC waveform generated by the magnetic array assembly as shown in Figure 5-14. The magnetic array assembly or generator consists of three rows of alternating pole (north-south-north…) permanent magnets mounted near the rotating shaft or coupling. An "E" core assembly is part of the CTS housing mounted on the rotating element. The "E" core is wound with approximately 1,000 turns of wire with total core resistance of 10 Ohms. In the current generator configuration, the device generates six full cycles of AC voltage each revolution of the shaft or coupling. The peak-to-peak amplitude of the voltage generated is linearly proportional to the surface velocity of the "E" core. The gap between the magnets and the "E" core is a key parameter in controlling the voltage amplitude. In the current design, at a rotational speed of 300 RPM, with the "E" core at a 4-inch radius and a 0.010-inch gap, the coil produces on the order of 20 volts peak-to-peak.

The AC waveform is rectified by a low voltage drop full bridge diode rectifier. Since the AC waveform is generated for a fraction of one rotation of the shaft, this power must be stored and managed to allow continuous (or semi-continuous) operation of the CTS. The original CTS breadboard used a switcher type 3.3-volt output regulator to supply the circuit board. Step down switcher supplies allow efficient use of excess voltage (input voltage exceeding the 3.3 volt output). This efficiency, however, decreases as the input voltage approaches the minimum

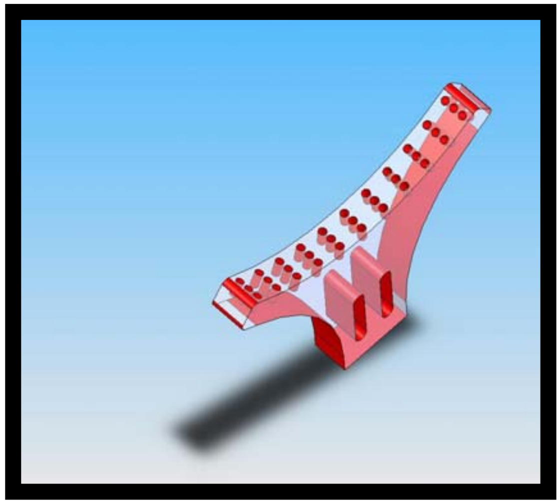

Figure 5-14. Solid Model of Magnetic Array

threshold voltage for stable output operation (say 3.6 volts). This loss of efficiency requires that smaller gaps be used to ensure that sufficient power is available at lower RPM ranges. At higher RPM when the generator is increasing the peak-to-peak voltage output, the switcher efficiency increases; however, this improvement in efficiency is not needed at these higher speeds. For this reason, and based on detailed power measurements from bench testing, the breadboard design was changed to replace the switcher regulator with a traditional low noise, low drop out 3.3 volt regulator, with a 5 volt protection Zener diode on the input of the regulator. This change resulted is a significant drop in power consumption at low shaft RPM's, as well as a reduction in overall board noise.

Since the CTS extracts power from the shaft rotation, available voltage and stored charge builds up over time as the unit starts. A power monitor IC is used to hold the μP off until the output of the 3.3-volt supply is stable for a predefined period of time. This ensures that the μP devices starts up and powers down properly. The regulated 3.3 volts powers the μP directly. In addition, the 3.3 volts is further regulated to 3 volts; this low noise 3-volt regulator is enabled and disabled by the μP.

Periodically, the CTS wakes ups, turns the 3-volt regulator on, powers the strain gages, powers the communication link, powers the analog front end, acquires data, transmits the data, then powers down the 3 volt section. With the current design, using a 1,000-Ohm full bridge, the average power consumption of the CTS is 15 mWatts.

The analog front end consists of a very high gain DC path, low bandwidth instrument amplifier. This is followed by a 16-bit analog-to-digital converter, which is currently sampled at 64 Hz. The sample duration is one second, and provisions have been made to allow synchronous acquisition using the AC voltage waveform as a timing marker. The A/D data is accumulated onboard the μP, with the running sum transmitted at the end of one acquisition/wake cycle. The CTS incorporates a 10-bit digital temperature device, which is sampled once per wake cycle. This device is quoted as 0.5°C accurately. The communication link uses a LINX 900 MHz, FM transmitter, operating at 19,200 baud. Including framing bytes, three strain based data bytes, and two temperature bytes, the total transmission time is less than 6 m-sec.

The CTS breadboard device was prototyped to allow bench testing and evaluation on Ariel compressor in the RCTF. Board gain calibration was tested using battery power and shut resistance. Figure 5-15 presents the board calibration curve. As shown, the CTS is very sensitive, with a slope of 615 A/D counts per micro-strain. Note that the equivalent strain was applied on a single leg of the bridge. For a full bridge in a shear configuration, a gage configuration factor of 4 would result, increasing the apparent sensitivity further.

Figure 5-16 presents the solid model of the CTS housing. Note that the housing and associated parts (magnetic array holder and housing) were rapid prototyped directly from the solid model. Figure 5-17 presents the solid model overview of the CTS housing, magnetic array and mounting configuration for the Ariel. Figure 5-18 presents a photograph of the magnetic array. Figure 5-19 presents a photograph of the CTS board installed in the housing. The board is not potted as would be required in the final product, limiting the upper RPM that can be tested on the Ariel. Note that in the final design, the connector for the strain gages would not be required as the CTS final design is to incorporate and mount the gages on the housing itself. Figure 5-20 presents the "E" core side of the housing. Figure 5-21, Figure 5-22, and Figure 5-23 shows the breadboard installed on the compressor side of the coupling of the Ariel in the CTS. Note that for test purposes, the antenna is mounted on the housing, and not around the coupling. It is anticipated, but not yet verified, that a "leaky" style antenna will be required for this device.

End-to-end testing of the breadboard CTS was completed in the RCTF. Strain gages were mounted on the output shaft of the electric motor in a full bridge, (shear) configuration. Cylinder indicated horsepower was combined with RPM to determine indicated torque, CTS output at various cylinder speed, and unit ratios were obtained. Note that at zero RPM, the unit was battery powered to determine a zero speed bridge balance output. Figure 5-24 presents the various test conditions. Figure 5-25 presents the CTS strain versus indicated cylinder based torque. The device output is linear (as expected). Note that the best-fit line through the data does NOT pass through the zero speed zero torque condition. This is very significant in that this small difference in the torque intercept at zero speed are losses in the system associated running the compressor. As seen, these losses can be assigned a fixed torque (offset), which would then translate to a horsepower loss linearly proportional to speed. Once the breadboard has been fully tested, then lessons learned should be integrated into a prototype version to be formatted at the ARCT test bed compressor at the SwRI MRF. This should be followed by a next generation used in field trials.

5.4 CAPACITY MEASUREMENT NEW CONCEPTS

Improved flow measurement methods are needed for advanced reciprocating compressor technologies for a number of reasons. In the area of capacity control, better flow measurement technology is needed as a counterpart to advanced capacity control methods, in order to meet varying flow demands. To improve load control and prevent overload, advanced flow meters are needed to acquire flow rate measurements quickly, with less susceptibility to pipeline noise, pulsation-induced errors and installation effects (due to upstream disturbances). For capacity and load control, the new flow measurement methods should consider a fixed meter installation that can be located fairly near the compressor.

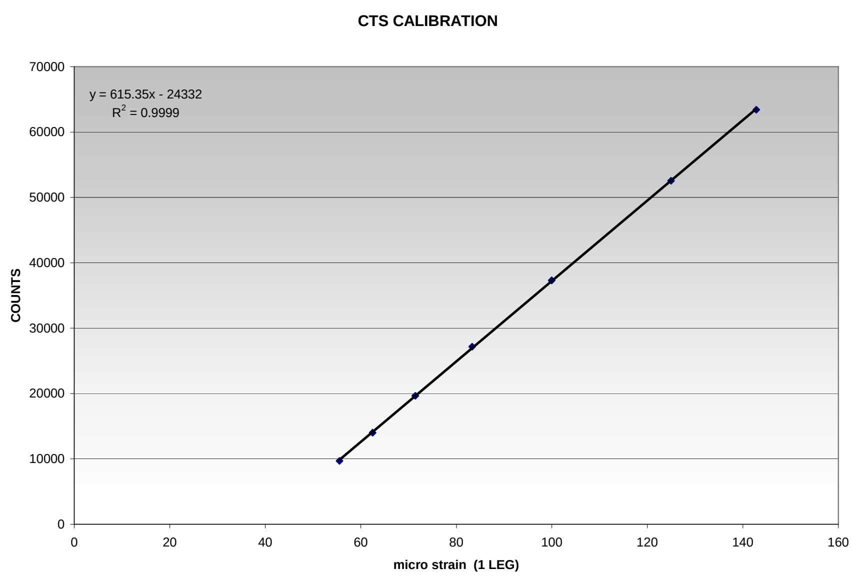

Figure 5-15. Bench Calibration

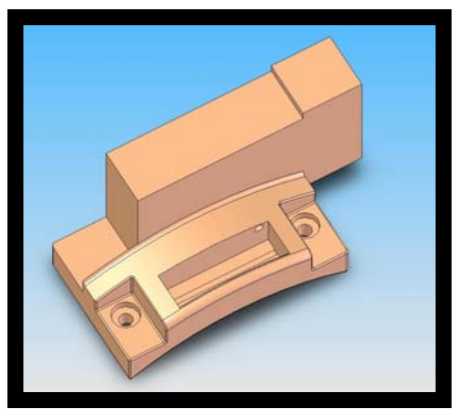

Figure 5-16. Solid Model of CTS Housing

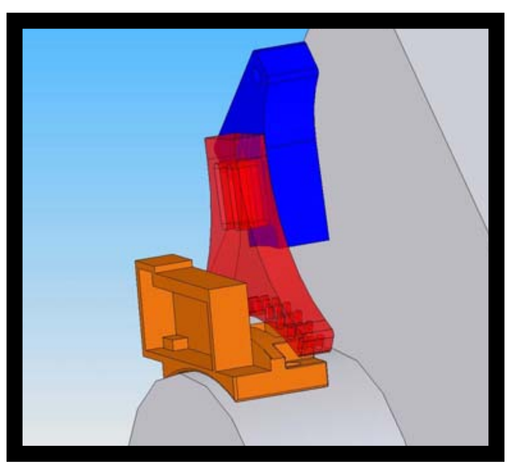

Figure 5-17. Solid Model of CTS for RCTF Testing

Figure 5-18. Magnet Array for RCTF Testing

Figure 5-19. CTS Breadboard in Housing

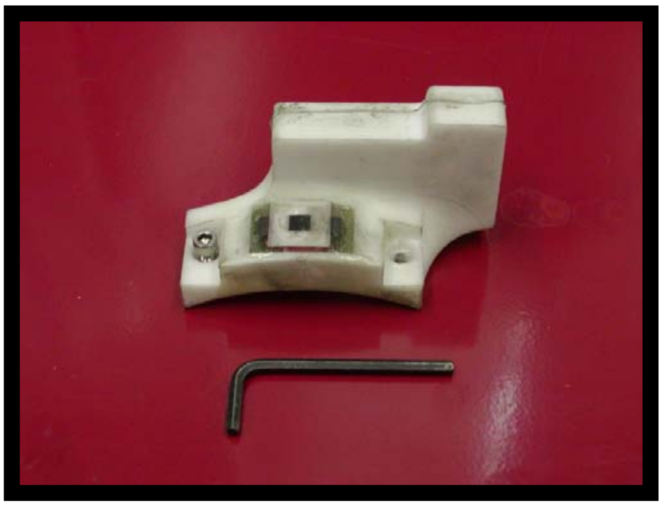

Figure 5-20. CTS "E" Core

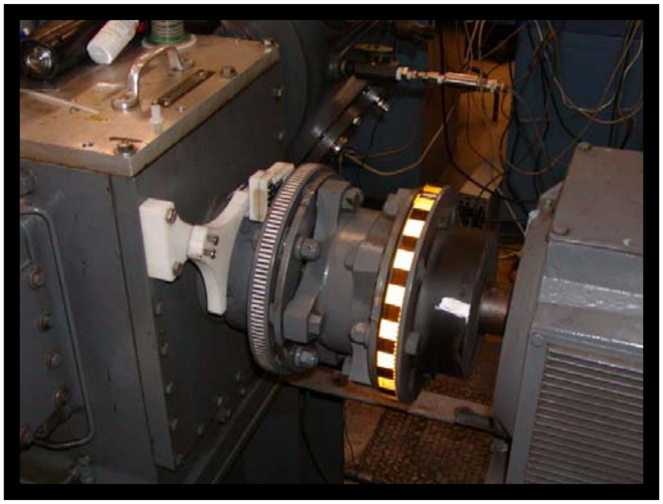

Figure 5-21. CTS Installed in RCTF

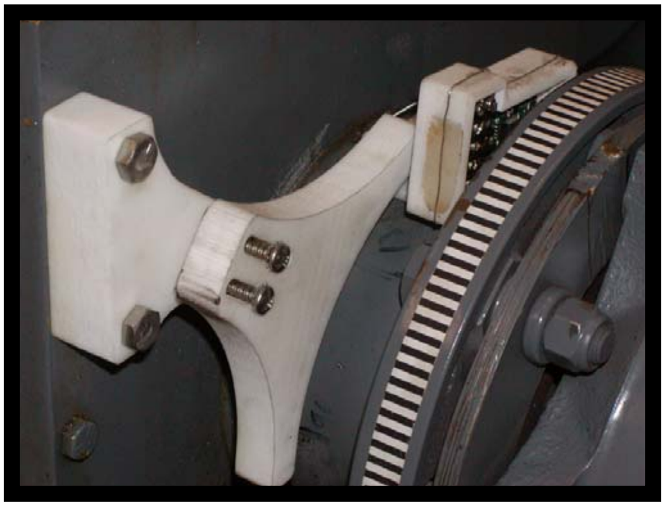

Figure 5-22. CTS Installed in RCTF

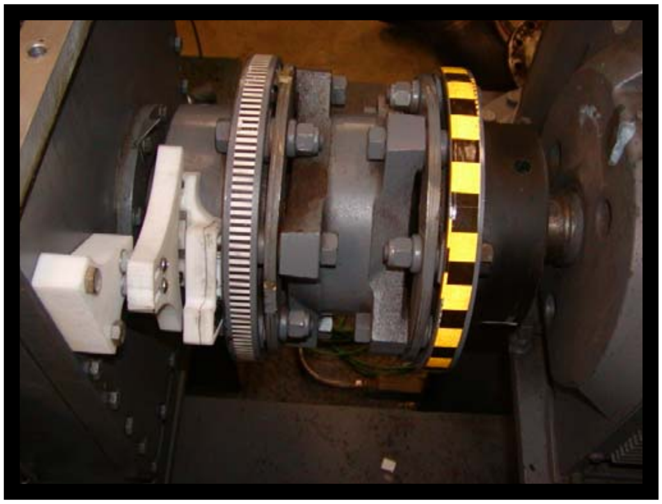

Figure 5-23. CTS Installed in RCTF, Magnetic Array and "E" core Aligned

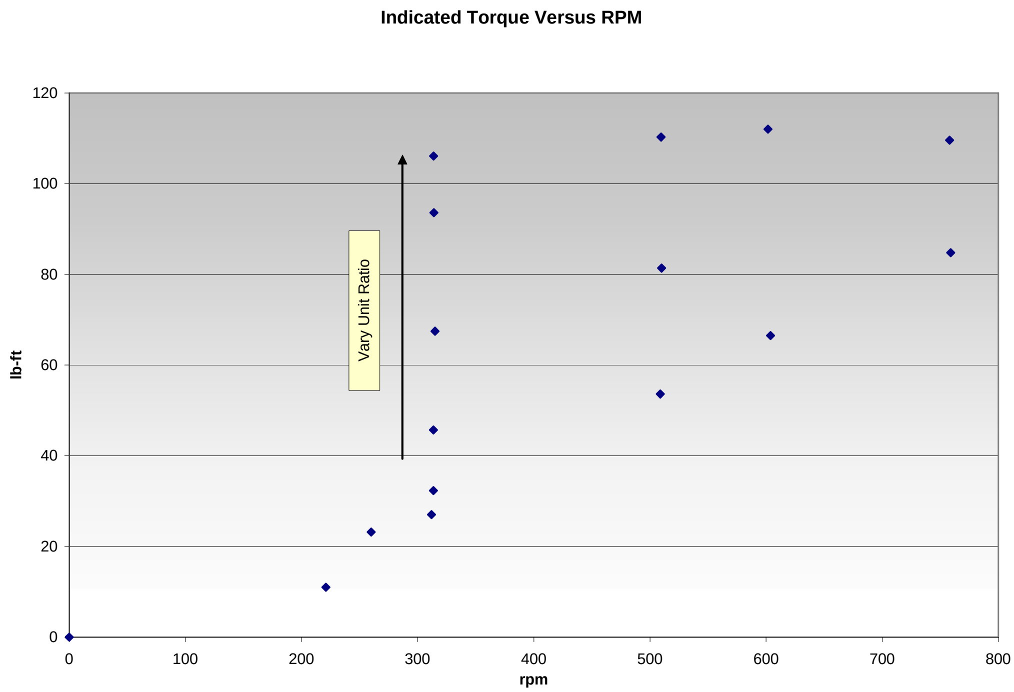

Figure 5-24. Indicated Torque versus RPM

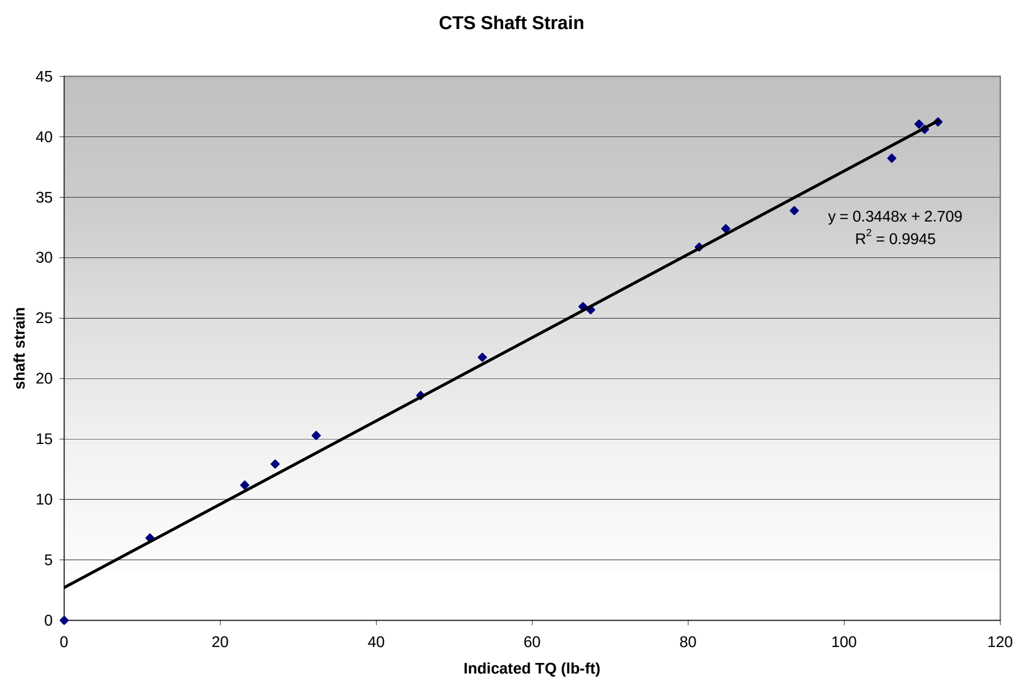

Figure 5-25. CTS Shaft Strain versus Indicated Torque

In the area of station optimization and sensor integration, higher speed flow measurement would provide an estimation of velocity modulations resulting from pulsations within the system. More accurate measurements of flow rate will improve the state of the art in performance monitoring of compressors and allow users a comprehensive measure of efficiency. As a result, the more precise, the determination of compressor performance and pulsation levels will improve the tools and modeling methods used in reducing pulsations. For this application, the flow measurement method could be more advantageous as a movable device and will most likely be an insertion-type device that can be used at multiple locations at a compressor station.

5.4.1 Concept Generation

The flow stream near a natural gas reciprocating compressor typically has a significant level of pulsations (2-5% velocity modulation) with a disturbed, undeveloped velocity profile. A flow meter in this environment must either be capable of discerning the pulsation-induced error on the flow rate or be insensitive to velocity modulations caused by pulsations. In addition, the flow meter must be fairly robust because the natural gas may contain varying levels of corrosives and contaminants. Non-intrusive flow measurement methods or insertion type devices are preferred. In addition, the compressor environment is typically very noisy, making ultrasonic transmission more difficult for ultrasonic flow measurement. Various flow measurement methods that may work at a reciprocating gas compressor station environment were considered in the initial stages of concept generation.

A state-of-the-art review of existing flow measurement technology was conducted to generate possible ideas that could be adapted to flow measurement at a reciprocating gas compressor station. Although many flow measurement technologies exist, the review was focused on finding concepts that do not require extensive upstream piping and that have been proven to work in fairly rugged natural gas applications. The review resulted in an initial set of twelve different flow measurement concepts. These concepts and descriptions of the flow metering principle behind each concept are listed in Table 5-1.

The initial list of concepts spans a range of technologies, accuracies, and installation requirements. Among the more traditional concepts are the multi-hole pitot tube, and the measurement of differential pressure across a valve. The application and acceptability of both of these concepts is highly dependent on the response time of the transmitters used to measure the pressure or differential pressure across the flow stream. Meters that provide a measure of the mass flow rate through the measurement of a proportional force include the Coriolis meter, the vortex meter, reaction force meter, and the target/drag meter. Some meters measure different properties of the flow stream such as induced vibrations or vortex shedding frequencies and correlate the measured property to a mass flow rate. Other flow meters in the list of concepts measure velocity more directly, such as the insertion turbine (rotor rotates due to flow force), high frequency ultrasonic meter (ultrasonic signal time differences measured), laser Doppler velocimetry (LDV) (reflected frequency correlated to gas particle velocity), and tracer gas methods (tracer gas particle velocity measured). Finally, the rugged hot film anemometer and the thermal mass meter use measurements of the temperature change to determine the proportional mass flow rate.

Table 5-1. Initial Concepts Considered for Advanced Flow Measurement at a Reciprocating Compressor station

METER TYPE	DESCRIPTION OF HOW METER WORKS
Coriolis Meter	Fluid flow produces force on vibrating tube, proportional to mass flow rate.
High Frequency Ultrasonic Meter	Ultrasonic signal transmitted across flow stream, in both upstream and downstream direction. Transit time or Doppler frequency shift linked to flow velocity.
LDV Measurement	Lasers transmitted across flow stream, reflected gas particles emit frequency proportional to speed.
Low Inertia Insertion Turbine Meter	Low inertia turbine, where relatively light meter rotor turned by in low flow/low viscosity flow stream in order to determine flow rate.
Multi-Port Pitot Tube	Multiple measurements of pressure allow determination of velocity component at different angles in flow stream.
Rugged Hot Film	Heat loss from fluid passing over anemometer is set proportional to flow rate.
Target/Drag Meter	AKA “drag meter,” flow induced strain on insertion tube creates imbalance in resistance circuit.
Thermal Mass Meter	Mass flow rate determined by rise or fall of sensing fluid temperature.
Reaction Force Meter	Measurements of torque set proportional to mass flow rate.
Tracer Gas	Radioactive/detectable tracer gas input to flow stream. Velocity determined through tracer particle transit time or decay rate.
Valve DP	Measure DP across valve and equate to flow rate (similar to principle for orifice meter).
Vortex Meter	Frequency of vortex shedding surrounding inserted body in flow stream used to determine mass flow rate.

These concepts were ranked against a preliminary set of ranking criteria, given in Table 5-2, which included the technology cost, reliability, risk, accuracy and development stage. All criterion were given equal weighting factors in the initial ranking process in order to eliminate concepts which were clearly unacceptable in any one area. The technology cost considered the cost of installing the flow measurement concept at a typical reciprocating gas compressor station. The concept’s reliability included an assessment of the ruggedness/complexity of the design and its durability in a corrosive, dirt natural gas environment. The third factor in the ranking criteria, risk, accounted for the potential detriments in operating successfully at the compressor station. The concept’s accuracy considered the overall accuracy of the device, considering the installation and operating conditions at a typical station. The final factor, development stage, included an assessment of the current state of the art for each concept and the adaptations that would be required to use the flow measurement concept at a compressor station.

5.4.2 SHORT LIST OF CONCEPTS

The initial flow measurement concepts were assessed according to the criteria shown in Table 5-2. The initial ranking process resulted in many of the concepts scoring very similarly. Many of the concepts scored well on reliability, risk, and development stage because the technologies are fairly advanced and have been used in natural gas applications previously. The primary differences in the concept ranking arose in the areas of accuracy and cost. The more costly concepts included high frequency ultrasonic meters, LDV measurement, Coriolis meters, and valve differential pressure measurements.

Table 5-2. Preliminary Ranking Criteria and Associated Weighting Factors Applied to Initial Flow Measurement Concepts

INITIAL CRITERIA	WEIGHTING FACTOR	DEFINITION
Cost	20%	Cost to implement on Compressor System
Reliability	20%	Expected reliability based on system complexity and susceptibility to environment
Risk	20%	Expected potential for success in implementing into compressor system
Accuracy	20%	Overall system capacity measurement accuracy in light of operating environment
Development Stage	20%	Current status relative to level needed for system implementation

As a result of the competitiveness of the flow measurement technologies, only three concepts were eliminated from the larger list. The list of concepts and the result of the ranking process is shown in Table 5-3. The concepts highlighted in red were not advanced for further analysis. The LDV measurement was eliminated primarily based on cost and its impracticality for use in natural gas compressor stations. Though the high frequency ultrasonic meters were also costly, clamp-on ultrasonic meters have been used successfully in many natural gas applications, and this technology is continually advancing to meet the growing needs of the industry. In addition to LDV measurement, reaction force meter and valve differential pressure measurement were eliminated based on the combined factors of cost and accuracy. Both concepts offered lower accuracies than other concepts and required a higher relative cost.

Table 5-3. List of Flow Measurement Concepts Advanced for Quantitative Analysis

Meter Type	Installation Requirements / Measurement location	Description of How Meter Works	Concept Advanced?
Coriolis Meter	Permanent installation, in system anywhere	Fluid flow produces force on vibrating tube, proportional to mass flow rate.	Yes
High Frequency Ultrasonic Meter	Insertion device (clamp-on) or permanent installation, requires less noisy condition	Ultrasonic signal transmitted across flow stream, in both upstream and downstream direction. Transit time or Doppler freq shift linked to flow velocity.	Yes, though high noise tolerance and high frequency required.
Low Inertia Insertion Turbine	Insertion device, system or header	Insertable turbine, where meter rotor turned by flow stream in order to determine flow rate.	Yes
LDV Measurement	System	Lasers transmitted across flow stream, reflected gas particles emit frequency proportional to speed.	No - Concept too costly, not practical.
Multi-Port Pitot Tube	Insertion device, system or header	Multiple measurements of pressure used to develop velocity measure.	Yes
Rugged Hot Film	Insertion device, system or header	Heat loss from fluid passing over anemometer is set proportional to flow rate.	Yes
Target/Drag Meter	Insertion device, anywhere in system	Aka "drag meter", flow induced strain on insertion tube creates imbalance in resistance circuit.	Yes
Thermal Mass Meter	System anywhere, location should maintain nearly constant fluid temp	Mass flow rate determined by rise or fall of sensing fluid temperature.	Yes
Torque measurements	System	Measurements of torque set proportional to mass flow rate.	No - Concept too costly for accuracy offered.
Tracer Gas	System	Radioactive tracer gas input to flow stream. Velocity determined through tracer particle transit time or decay rate.	Yes
Valve DP	Discharge valve	Measure DP across valve and equate to flow rate (similar to principle for orifice meter).	No - Concept too costly for accuracy offered.
Vortex Meter	Insertion device, anywhere in system	Frequency of vortex shedding surrounding inserted body in flow stream used to determine mass flow rate.	Yes

5.5 CAPACITY MEASUREMENT CONCEPT SELECTION

The state-of-the-art review generated ideas that could be adapted to flow measurement at a reciprocating gas compressor station. Although many flow measurement technologies exist, the review was focused on finding concepts that do not require extensive upstream piping and that have been proven to work in fairly rugged natural gas applications. The major flow measurement concepts evaluated for use in compressor capacity measurement included the target flow meter, thermal mass flow meter, ultrasonic meter, insertion turbine meter, multi-port pitot tube, tracer gas, vortex meter, Coriolis meter, and the ruggedized hot-film anemometer. The flow measurement concepts were evaluated further in terms of range, accuracy, repeatability, time response, and estimated installation cost.

5.5.1 RANKING OF CONCEPTS

The short list of the nine most promising flow measurement concepts are shown in Table 5-4. The concepts were evaluated in more detail in order to determine a finalized ranking. The ranking process used the criteria provided by the industry Project Supervisory Committee (PSC). A typical reciprocating compressor station was used as the design case in evaluating the flow measurement concepts in each of these areas. The typical design case assumed that a flow meter is installed in a suction or discharge compressor line with flow velocities ranging from 20 to 50 ft/sec. The meter should be capable of accurately determining flow rate, with a velocity modulation in the flow stream of 2% to 5% at a frequency of 10 Hz to 20 Hz for low speed machines or 20 Hz to 40 Hz for high-speed machines. The meters specified accuracy, repeatability, and time response were used to evaluate the technology's overall accuracy. The estimated cost of the flow meter was the central factor in evaluating technology cost, though additional costs for external instrumentation and wiring were also considered in the cost category. Each meter's rangeability was evaluated by considering the meters specified and expected range compared to the ideal range for the design case. Risk and developmental aspects were evaluated using the documented experience with the various technologies and the likely performance in natural gas.

Table 5-4. Summary of Nine Flow Meter Concepts Performance Criteria

FLOW METER TECHNOLOGY	RANGE	ACCURACY	REPEATABILITY	TIME RESPONSE	EST. COST
Target Meter	20-50 fps	±1.0% of reading	±0.15% of reading	0.1 sec	$8-10k
Thermal Mass Meter	< Required range	±1.0% of reading	±0.50% of reading	10 sec (high flow), 17 sec (low flow)	N/A
Ultrasonic Meter	2-150 fps	±1.0% of reading	±0.20% of reading	1 sec	$30k
Insertion Turbine Meter	2.5-55 fps	±1.0% of reading	±0.25% of reading	Undetermined	$12k
Multi-Port Pitot Tube	2.5-55 fps	±0.9% of reading	±0.10% of reading	0.1 sec	$14k
Tracer Gas	Undetermined	±2.0–4.0% of reading	Undetermined	Not continuous	$50k plus
Vortex	4-300 fps	±1.5% of reading	±0.15% of reading	0.8 sec	$11.5k
Coriolis	0-50 fps	±0.35% of reading	±0.20% of reading	1 sec	$47k
Ruggedized Hot-Film	Not commercially available, all factors should be tested.				

5.5.2 DESCRIPTION OF SELECTED TOP CONCEPTS

The decision matrix used to rank the flow meter concepts is shown in Figure 5-26. As the chart showed, the three concepts with the highest summary scores that fell into the "good" category were the insertion target meter, the ultrasonic flow meter, and the multi-port pitot tube. The target meter and the ultrasonic flow meter had better overall scores than the multi-port pitot tube. The target meter was viewed as very versatile and effective, which lends itself well to an application where the flow is not necessarily well conditioned with a high amount of pulsations. In addition, the target meter has a very fast response time, which would be capable of distinguishing pulsations in the 10Hz to 50 Hz frequency range. The meter installation is also very well suited to the application, because the meter can be inserted into the pipe flow using a flange or NPT connection, rather than bolting in spool piece meter. The ultrasonic meter is also well suited to the application because the meter takes many ultrasonic samples per second, which makes it capable of tracking the probable pulsation levels induced in the flow from the reciprocating compressor. The overall response time of the ultrasonic meter is slower than the target meter, but the response is an average of many high frequency samples. In addition, the ultrasonic meter is very reliable and accurate. It has been field tested for natural gas flow at both low and high pressures. Based on the industry committee input, the insertion target meter and the ultrasonic flow meter were chosen as the concepts that should be advanced to a proof-of-concept test.

The insertion target meter works by using a combination of four variable resistance strain gages mounted on a connecting support rod. The support rod contains a "target" or drag body that is inserted into the flow to magnify the drag force. The strain gages are interconnected to form a bridge circuit on the non-exposed end of the support rod. The force of the fluid (drag force) causes a change in the resistance of the strain gages mounted on the target support. The two strain gages on the forward side of the support are put in tension, while the two strain gages on the reverse side of the support are compressed by the fluid flow force. At zero flow, the change in resistance is zero. As flow is increased, the strain gage circuit becomes unbalanced in

	Effectiveness	Efficiency	Life & Reliability	O&M Cost	Retrofitable	Development Risk	Synergy	Installed Cost	Economic Viability	Maturity	Summary
Target/Drag Meter	Good	Good	Excellent	Good	Good	Good	Good	Excellent	Good	Good	Good
High Frequency Ultrasonic M	Excellent	Good	Good	Fair	Fair	Excellent	Good	Poor	Good	Excellent	Good
Multi-Port Pilot Tube	Excellent	Fair	Excellent	Poor	Good	Good	Fair	Good	Fair	Good	Good
Low Inertia Insertion Turbine	Good	Good	Poor	Fair	Good	Good	Good	Excellent	Good	Good	Fair
Vortex Meter	Fair	Poor	Excellent	Good	Good	Poor	Good	Excellent	Fair	Excellent	Fair
Coriolis Meter	Excellent	Good	Poor	Fair	Poor	Fair	Fair	Poor	Good	Excellent	Fair
Thermal Mass Meter	Poor	Fair	Excellent	Good	Good	Good	Fair	Good	Fair	Good	Fair
Rugged Hot Film	Good	Poor	Good	Fair	Good	Good	Good	Fair	Fair	Fair	Fair
Tracer Gas	Poor	Poor	Poor	Poor	Excellent	Good	Fair	Poor	Fair	Fair	Poor

Figure 5-26. Decision Matrix Used in Ranking the Flow Measurement Concepts

proportion to the flow velocity squared. The output of the bridge circuit is proportional to the product of the density and velocity head ($V^2/2g$). As such, the temperature and pressure of the flowing stream must be measured near the target meter because the density must be determined. Assuming constant density, flow will be proportional to the square root of the determined strain.

The manufacturer must calibrate or range the target sensor to meet the desired temperature, pressure, and flow rate of the particular application. The meter does not contain any moving parts that can wear out, which makes it fairly rugged. It can withstand a sizeable over range flow above its stated capacity without damaging the sensor. In addition, the fast response time of the meter allows many measurements over time, which can reduce the precision error. Its disadvantages are the density dependence in determining flow rate and the somewhat small flow range that a single sensor can offer. In addition, this flow measurement technology has not been fully tested in pulsating natural gas flows. Its ability to work over a range of gas pressures is somewhat unknown.

An example of one type of insertion target meter manufactured by Aaliant is shown in Figure 5-27. The picture on the left in shows the permanently mounted insertion meter, which uses a flange connection on the top of the pipe. The transmitter for the meter may be mounted directly onto the strain gage sensor. The picture on the right in shows a diagram of the bridge circuit with the four strain gages used by the target meter.

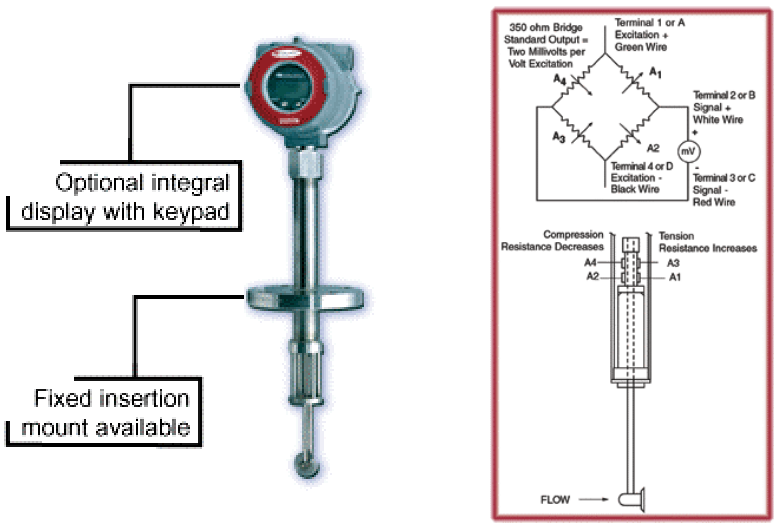

Figure 5-27. Aaliant Insertion Target Meter, Photo Courtesy of Aaliant Website (www.aaliant.com)

The current ultrasonic flow meters used in natural gas applications rely on multiple pairs of transducers to transmit and receive an ultrasonic signal across the flow stream. For a given pipe diameter, the velocity is determined by the transit time difference for the ultrasonic signal to travel in the forward and reverse flow directions. The ultrasonic transducers take many high frequency measurements of the flow velocity, which could be used to provide an instantaneous flow velocity. In measuring pulsating flow with an ultrasonic meter, the instantaneous measurement of the velocity from each transducer could be measured in conjunction with the average flow rate determined by the meter. The ultrasonic meter concept provides other advantages in that it is highly reliable and does not require temperature and pressure compensation through live measurements of the flowing stream. In addition, the ultrasonic meter has been shown to be a highly accurate and repeatable measurement of flow rate for natural gas for both low- and high-pressure applications.

Currently, the ultrasonic meter technologies provide a single average flow rate after processing many instantaneous flow velocities. Since ultrasonic flow measurement has not been tested successfully in the compressor station environment, this flow meter technology would also require some development in terms of response time. In addition, the ultrasonic meter must be capable of filtering out the environmental noise (which is likely in the vicinity of a reciprocating compressor) in order to detect the transmitted ultrasonic signal.

An example of an ultrasonic meter manufactured by Instromet Ultrasonic Technologies is shown in Figure 5-28. This meter is a Q-Sonic three-path meter that has been adapted for low-pressure applications (50 to 250 PSI). The meter is a spool-mounted design, where the ultrasonic transducers are placed in the flow stream. The meter features two pairs of swirl-path transducers and a single pair of transducers that use an axial flow path. A single path flow meter would use transducers that send the ultrasonic signal along the axial path.

Figure 5-28. Instromet Q-Sonic Three Path Ultrasonic Flow Meter

5.6 CAPACITY MEASUREMENT PROOF-OF-CONCEPT

For the proof-of-concept test, an Aaliant flange-mounted insertion target meter and an Instromet 4-inch ultrasonic meter were installed in the ARCT test bed at the SwRI Metering Research Facility. The test loop utilizes a two cylinder, 250 horsepower reciprocating compressor. The meters were installed in the suction tube entering the primary pulsation control bottle. The suction line comes out of the secondary pulsation control bottle on each side and feeds each of the two primary pulsation bottles. A photograph of the installation location is shown in Figure 5-29.

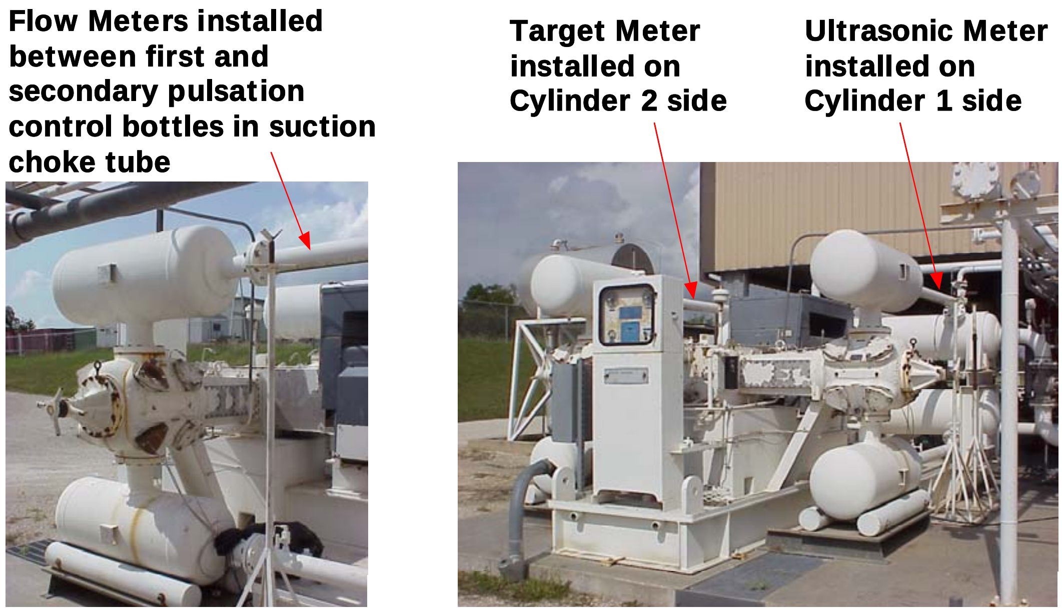

Figure 5-29. Proof-of-concept Test Installation in ARCT Test Bed

Prior to running the proof-of-concept test, each meter was tested against the MRF standard flow reference using the critical flow Venturi nozzles. The meters were installed in the test section of the low-pressure flow loop with at least 50 diameters of straight pipe upstream. The baseline test characterized the meter performance in a steady, well-conditioned flow over a flow velocity range of 20-100 ft/s at 100 and 150 PSIA. In addition, the target meter current output was calibrated as part of the MRF system to assure that the data acquisition system correctly recorded the flow meter current output.

The meters were then moved to the test installation, next to the compressor suction bottles. Pressure volume data was taken on each cylinder concurrent with the proof-of-concept test for the meter on the same cylinder side, i.e., the Cylinder 1 measurements were recorded, as the ultrasonic meter test was being run and the Cylinder 2 measurements were taken during the target meter test. The pressure volume data was recorded using high frequency response pressure transmitters and a shaft encoder (to determine the location of the piston in the cylinder). The meters were tested at two pressure ratios (1.33 and 1.67) over a range of compressor speeds, 500, 600, 700, and 900 RPM. The various compressor speeds resulted in different test velocities for the meters. The suction pressure was 105 PSIA for the higher ratio test and 150 PSIA for the lower pressure test. The target meter and ultrasonic meter flow velocities were adjusted with a constant linear correction factor based upon the flow-weighted mean difference between the meter and the PV card. This adjustment was used to compensate for any installation effects due to the short entry length and upstream 45-degree elbow.

In the proof-of-concept test, the flow meter performance was compared to the measurements of flow rate calculated using the pressure-volume cards. The loop reference flow rate could not be used for this comparison because the meters were installed in the suction pipe leading into a single cylinder (approximately half of the loop flow rate because of the two cylinder machine). For this reason, the proof-of-concept testing was used to demonstrate meter performance and general agreement the PV cards. The testing was not used to determine absolute accuracy of the selected flow meter technologies in the non-steady flow environment near the compressor.

5.6.1 Target Meter Proof-of-concept

The target meter baseline test was used to characterize the meter performance in a steady flow condition and to calibrate the serial (current) output from the meter. The meter showed a slight change of approximately 0.5% in its calibration factor at the two test pressures of 150 and 100 PSIA. The proof-of-concept installation of the insertion target meter is shown in Figure 5-30. Temperature and pressure measurements of the gas in the suction tube were used to calculate the density compensation for the meter. The meter was tested in the six flow rate and test pressure combinations. The meter response was on the order of 100 samples per second for the testing. A 30-second average of the meter samples was used to compare against the PV card measurements for the same period of time.

Figure 5-30. Insertion Flange-Mounted Target Meter in Suction Choke Tube at ARCT Test Bed

Figure 5-31 shows the results for the target meter proof-of-concept test at the two test pressures. The percent difference between the target meter and PV card measurement varies from –4.0% to +3.0%. The percent difference was notably less in the case of the highest test velocity at 80.0 ft/sec. In addition, the target meter consistently measured less flow than the PV cards calculated, on the order of –1.0 to –6.0%.

Though the true accuracy of the target meter cannot be determined based on the PV card measurements, the precision error can be calculated based on the average velocity for six 30-second runs. The precision error is proportional to the standard deviation of the velocity for a constant number of runs. The standard deviation in the PV card measurements and the target meter is shown in Figure 5-32. This figure demonstrates the more precise measurement of velocity that the target meter provides. The standard deviation of the target meter was less than 0.20% in all cases, whereas the PV card measurements showed a standard deviation of 0.5% to 1.0%. (In determining the standard deviation, the pressure data recorded in the cylinder was linearly corrected to reduce the scatter in the transducer measurement as much as possible.)

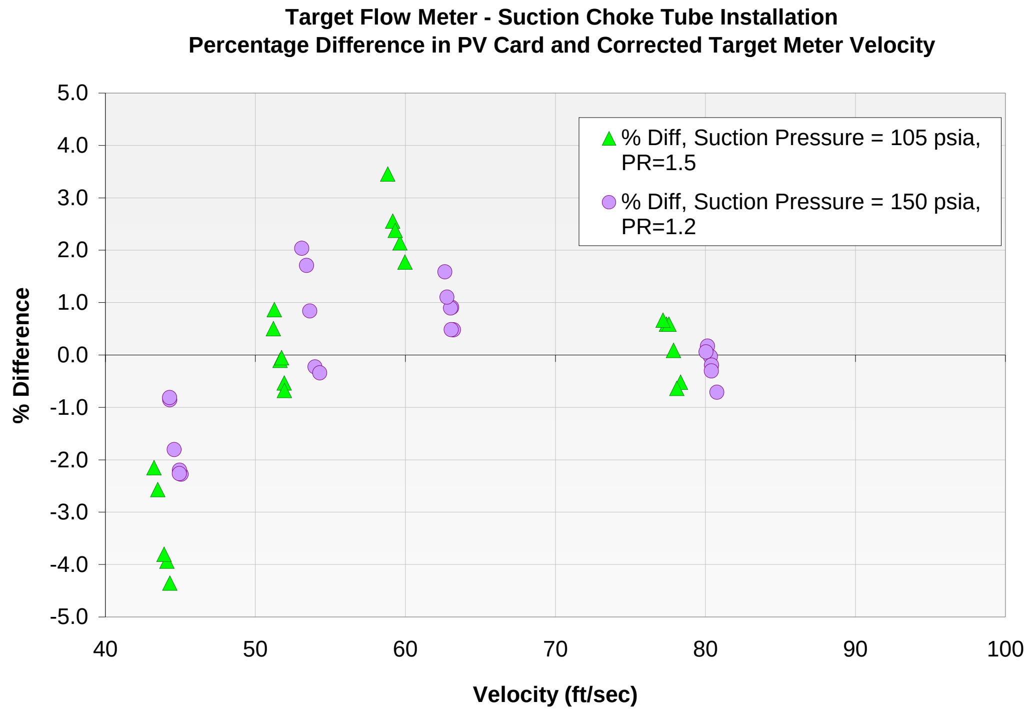

Figure 5-31. Percentage Difference in Determination of Flow Velocity Between PV Card Measurements and Target Flow Meter

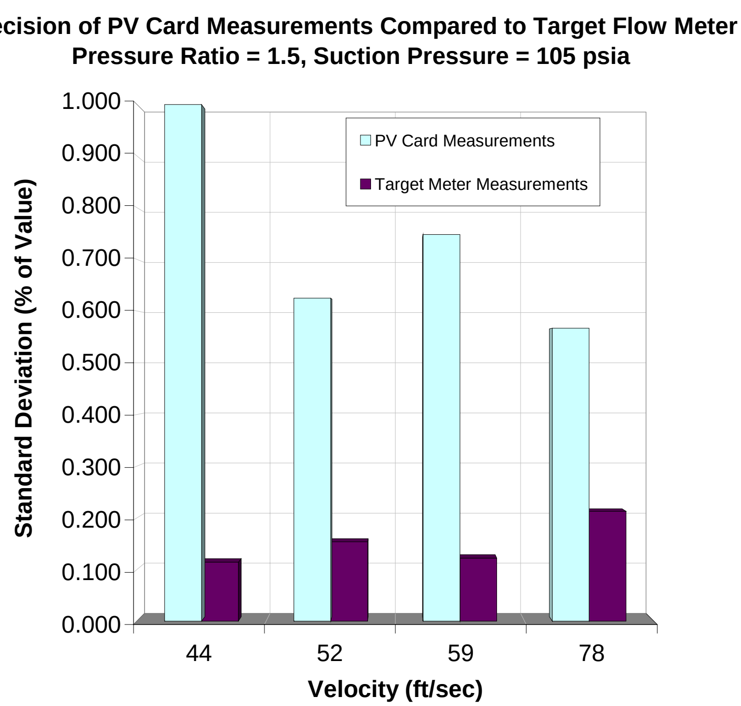

Figure 5-32. Comparison of Precision Error in PV Card Measurements and Target Meter

5.6.2 ULTRASONIC METER PROOF-OF-CONCEPT

The ultrasonic meter baseline test was also used to characterize the ultrasonic meter performance in a steady flow condition. The meter results showed a very close agreement of 0.25% to 0.50% with the MRF critical flow Venturi nozzles in the baseline test. The meter did not show a change in the calibration curve at the two test pressures. The meter was then installed upstream of the primary pulsation bottle on the Cylinder 1 side of the compressor, as shown in Figure 5-33. Temperature and pressure measurements were also recorded for the gas flow through the meter, though the ultrasonic meter does not require a "live" compensation for density. The measurements allowed the density at the meter to be calculated to assure that the gas density remained fairly constant during each test run. A 30-second average of the meter instantaneous velocity readings was used to compare against the PV card measurements for the same period of time.

Figure 5-33. 4-inch Instromet Ultrasonic Flow Meter in Suction Choke Tube at ARCT Test Bed

The meter was tested at the same four velocities as the target meter by varying the speed of the compressor to change the velocity through the suction choke tube. The percent difference in the ultrasonic meter and the PV card measurements is shown in Figure 5-34 for the two test pressures. Except in the case of the highest test velocity, the ultrasonic meter consistently measured less flow to the compressor than the PV card measurement. This result was similar to the target meter test. The percentage difference between the ultrasonic meter and the PV card measurement ranged from –3.0 to +2.0%. The velocity curve shown in Figure 5-34 has a similar shape to the curve for the target meter and the PV card measurements shown in Figure 5-31, where the percent difference increases as velocity increases. The difference in the two suction pressures did not affect the results, which suggests that the flow measurement device is not strongly affected by changes in pressure (or density).

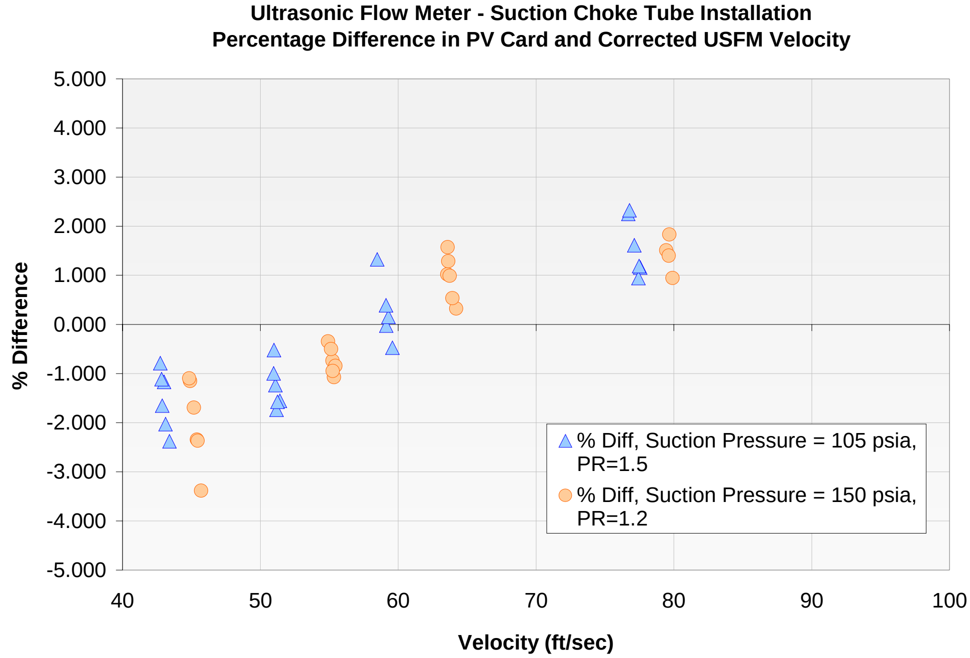

Figure 5-34. Percentage Difference in Determination of Flow Velocity Between PV Card Measurements and Ultrasonic Flow Meter

The ultrasonic proof-of-concept test was taken further in testing the ultrasonic meter as a single path design. To perform this test, the two swirl paths on the Instromet three-path meter were deactivated, and only the axial path was used to determine the flow velocity. The test showed that the single path meter design worked as well in this application as the multi-path ultrasonic meter design. The single path meter results were adjusted for the installation effect using the same method of determining a constant flow-weighted mean difference between the meter and the PV card measurements. The percent difference between the corrected single path ultrasonic meter and the PV card measurement is shown in Figure 5-35. The percent difference was within a similar range as the multi-path ultrasonic meter. In addition, the single path meter measured a consistently lower flow rate than the PV card measurements, like the target meter and the multi-path ultrasonic meter. The single path meter design is more affordable because less ultrasonic transducers are required.

The precision of the ultrasonic flow meter was compared to the precision in the PV card measurements to compare the precision uncertainty in the two measurements of flow. For both flow measurements, the average velocity was calculated each 30-second. The standard deviation of the average velocity was then taken as a measure of precision uncertainty. The results for the four test velocities are shown in Figure 5-36. The ultrasonic meter showed a standard deviation of 0.15% to 0.30%, while the PV card measurements contained a higher standard deviation of 0.32% to 0.70%. Clearly, the ultrasonic meter provides a more precise measurement of flow in this particular application.

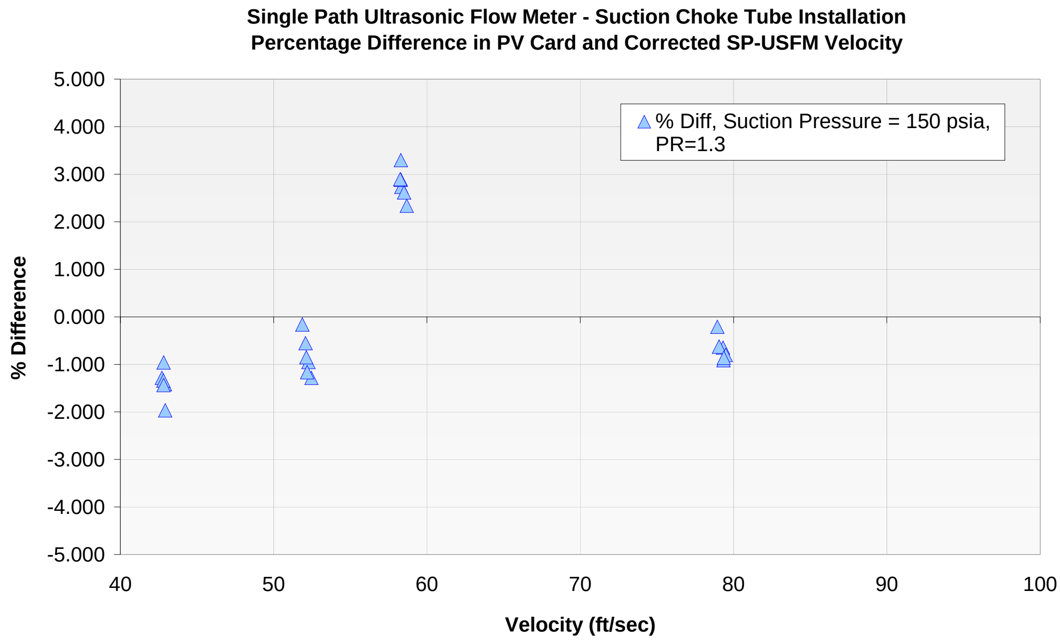

Figure 5-35. Percentage Difference in Determination of Flow Velocity Between PV Card Measurements and Single Path Ultrasonic Flow Meter

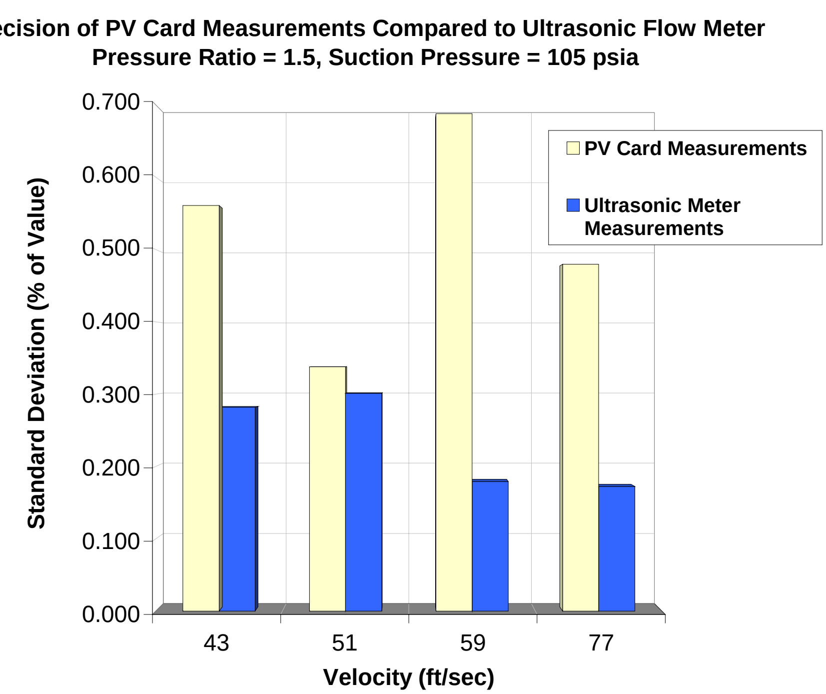

Figure 5-36. Comparison of Precision Error in PV Card Measurements and Ultrasonic Meter

5.6.3 RECOMMENDATION FOR FUTURE WORK

The proof-of-concept test demonstrated that the ultrasonic meter and the insertion target meter are viable flow measurement technologies. The technologies are both capable of a high frequency response that can be used to determine the steady component of the flow velocity in a non-steady flow application. The meter error due to pulsations must still be determined and possible modifications to the meter flow output may be necessary in a highly pulsating, non-steady flow. The two flow meters proved to be consistent and reliable in determining velocity near a reciprocating compressor. The PV card measurements determined a higher flow rate in both proof-of-concept tests, which suggests that the PV cards may induce a positive error on the flow rate through the compressor. A follow-on test of the two flow meter technologies is recommended to determine absolute accuracy in the non-steady flow environment near a reciprocating compressor.

6. SYSTEM INTEGRATION AND OPTIMIZATION RESULTS

6.1 SYSTEM TECHNOLOGY ASSESSMENT

Although the actual reciprocating compressor itself is the central component of any compression system, the term "gas compressor" (or compressor package) often refers to the collection of equipment needed to receive gas at one pressure and deliver it at a second, higher pressure. In addition to the compressor, such a system typically includes a driver (often an internal combustion gas engine or electric motor), coolers, separators (scrubbers), pressure vessels used for acoustic filtering, piping, and various instrumentation and control hardware. Figure 6-1 is a photograph showing the components of a typical small compressor package.

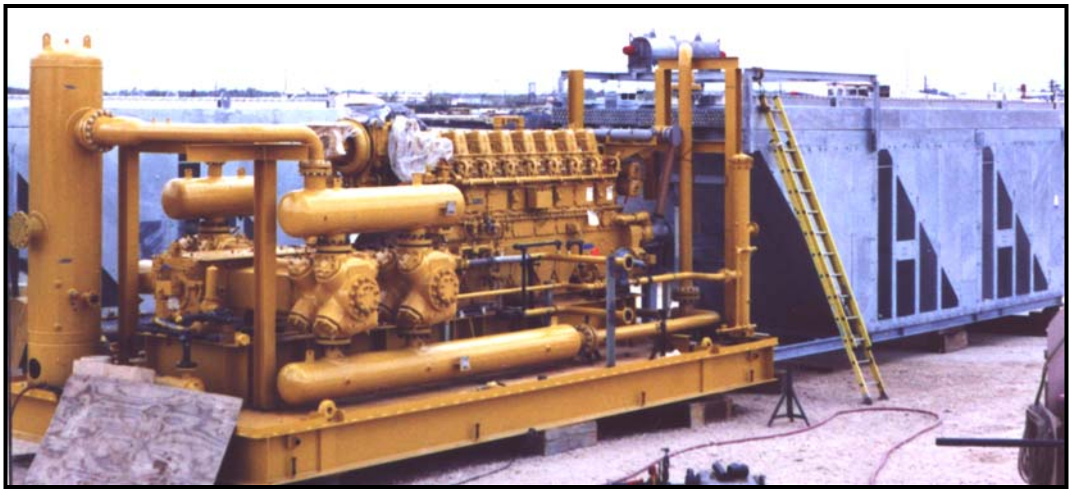

Figure 6-1. Photograph of a Typical Compressor Package Showing Compressor, Engine, Cooler, and Associated Piping and Pressure Vessels

6.1.1 COMPRESSOR MOUNTING

The basic requirement for any compressor mounting system is to support and restrain the compressor, driver, and other associated equipment over the lifetime of the machine (likely 30-40 years). Regardless of the particular design details, any mounting must withstand both static and dynamic loads. The static loads come from the weight of the equipment being supported. In addition, thermal loads that vary slowly over time can be viewed as an additional static load. The dynamic loads are a result of unbalanced shaking forces generated by the reciprocating machinery. These dynamic loads can be quite large, resulting in forces as large as several hundred thousand pounds acting on each of the machine's anchor bolts [Smalley, et al., 1999[28]; Harrell, et al., 2001[29]]. Reciprocating compressors rely on the mounting system to support the equipment, to maintain alignment between the compressor and driver, and to resist the dynamic loads present when the compressor is operating.

Figure 6-2 shows two types of compressor mounting designs that are commonly used: skid and block (or foundation). In the former, the compressor, driver, piping, vessels, and associated equipment are all located on a steel base or skid. Normally, the skid is meant to be portable so that the compressor package can be assembled in a shop and the whole unit delivered to the installation site. In situations with larger, heavier equipment, the skid may be delivered to the site with only some of the equipment and piping in place, and the larger items, such as the engine and/or compressor, are mounted on the skid after it is installed at the site. In the case of block mounting, all of the equipment is attached directly to a concrete foundation that has been constructed at the field site.

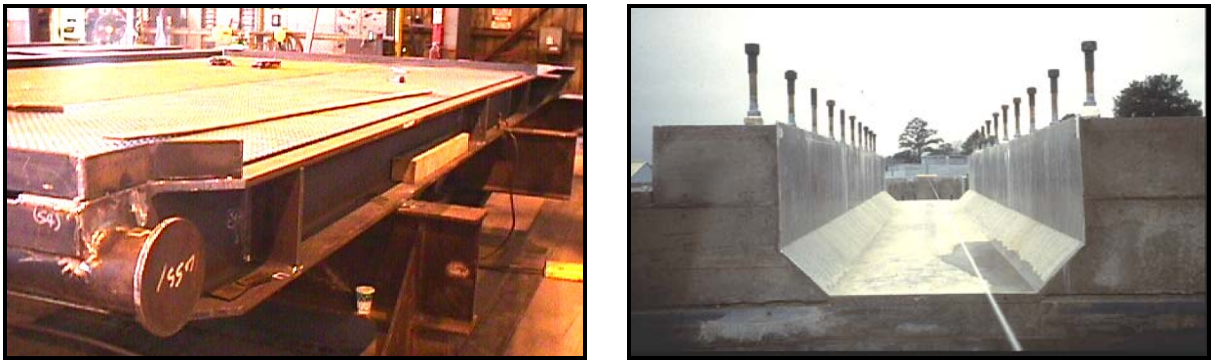

Figure 6-2. Two Compressor Mounting Methods: Steel Skid (Left) and Concrete Block Foundation (Right)

The integrity of the mounting system has a direct influence on equipment reliability and life by minimizing vibration and maintaining alignment between the compressor and driver. In addition, the mounting must provide sufficient rigidity to prevent high stresses in nozzles, piping, etc., due to relative motion between components. An inadequate skid or foundation design can lead to excessive vibration, breakage of anchor bolts, and cracking of foundation material. Besides creating ongoing operation and maintenance issues, compressor-mounting problems can ultimately lead to bearing, connecting rod, and crankshaft failures. Figure 6-3 shows two examples of crankshafts that failed. A survey [Smalley, 1995][30] of 26 companies in 1995 showed that they cumulatively suffered an average of two to three broken crankshafts per year, with mounting and alignment problems identified as the leading cause of the failures. This report gave an average repair cost of $264,000 per incident. For larger machines, the cost can be much higher than this average.

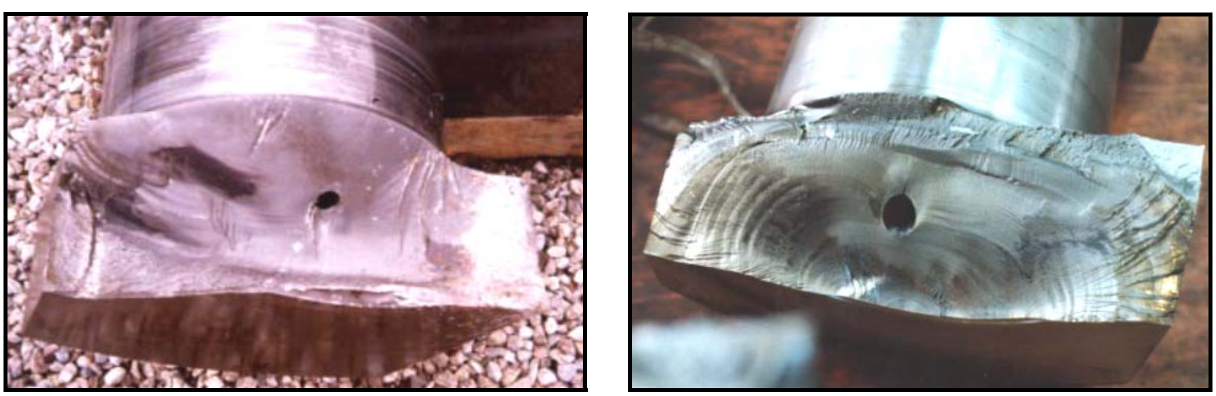

Figure 6-3. Photographs of Two Crankshafts that have Failed Due to Excessive Bending Deformation [Smalley, 1997][31]

6.1.1.1 Skid Mounting

Skid mounting of compressor packages is a common practice for high-speed units (1,000 to 1,800 RPM) up to about 4,700 HP. As shown in Figure 6-4, a compressor skid usually consists of a welded I-beam frame onto which the compressor, driver, and associated equipment is mounted. Skids for large compressor packages can be as large as 15 feet wide and 40 feet long. In some cases, skids are filled with concrete to add mass and damping to the structure. Skids can be designed to be set directly onto compacted soil at the field site or they can be installed on a concrete pad. In the case of a concrete pad installation, the skid is often set one to two inches above the concrete and grouted in place [Leary, 2004][32]. After grouting, the skid is secured with anchor bolts set in the concrete.

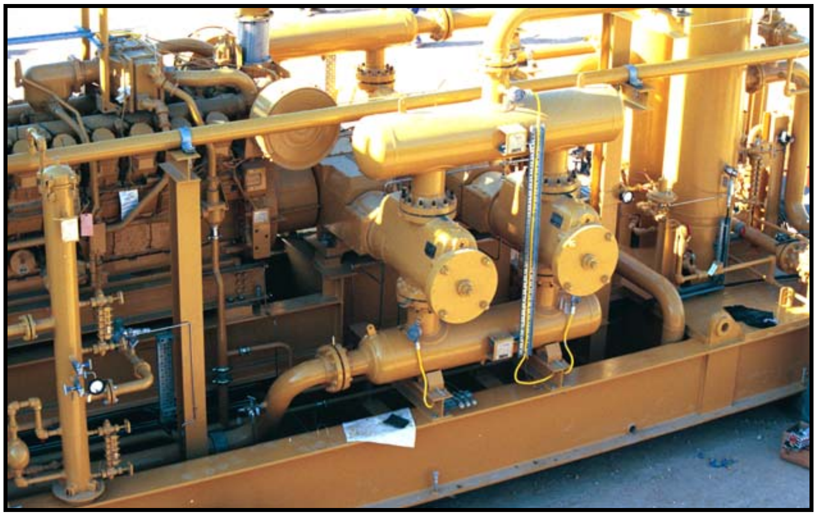

Figure 6-4. A Skid-Mounted Compressor Package

The main advantages of skid-mounting compressors are ease of assembly and portability. At least up to some size and weight limit, a skid-mounted compressor package can be completely assembled in the shop where skilled labor and equipment are available and then transported to the field and installed with minimal on-site work. Skid mounting is also frequently used for units intended for offshore installation.

A 1992 report by Mandke and Troxler [Mandke, et al., 1992][33] estimated that there were approximately 5,000 skid-mounted compressors in service among 27 companies surveyed. Some of the key trends and issues identified by Mandke and Troxler from their survey are as follows:

- Over a 20-year period, 21 of the reporting companies experienced skid vibration problems with five or more units.
- Ten of the reporting companies had no specific design code compliance requirement for skid fabricators.
- Areas of most concern were computation of shaking forces and resonant frequencies of the skid and attached components.
- The companies expressed a need for the development of guidelines for skid design and analysis.

- Corrective actions for vibration problems included bracing components, tightening or modifying tie-down bolts, and reinforcing the skid structure to shift its natural frequency.

As suggested by the above survey results, excessive vibration is a major concern with skid mounted compressors. These vibrations are often due to resonances that result from skid or equipment (e.g., scrubber or piping) natural frequencies in the running speed range. The survey found that while most companies analyze their skid designs for lifting, most do not perform the type of dynamic analyses needed to identify structural resonances. There are indications that this situation has changed since this survey was conducted. Harrell [2004][34] also notes that successful skid designs are often reused without verifying the suitability of the design for the new application.

There also appears to be no uniform practice in skid design across the industry. Skid design guidelines are available from a variety of sources such as manufacturer's specifications, company in-house specifications, previous American Petroleum Institute (API) Specifications [API Specification 11P, 1989][35], and research reports [Mandke, et al., 1992][33].

6.1.1.2 Block Mounting

Foundation blocks are the mounting method used for most slow-speed integral compressors. Block foundations have the advantage of being stiffer and adding more damping than skids. Figure 6-5 is a drawing of a typical concrete block compressor foundation. As described in a GMRC research report [Harrell, et al., 2001][29], a typical modern foundation (built after 1980's) will have three to five times the mass of the supported compressor and will consist of high-strength concrete (compressive strength up to 5,000 PSI) with a dense rebar grid through its entire section. The concrete is capped by a layer of epoxy grout, which provides a tough, level, durable, and chemically impervious surface for the foundation.

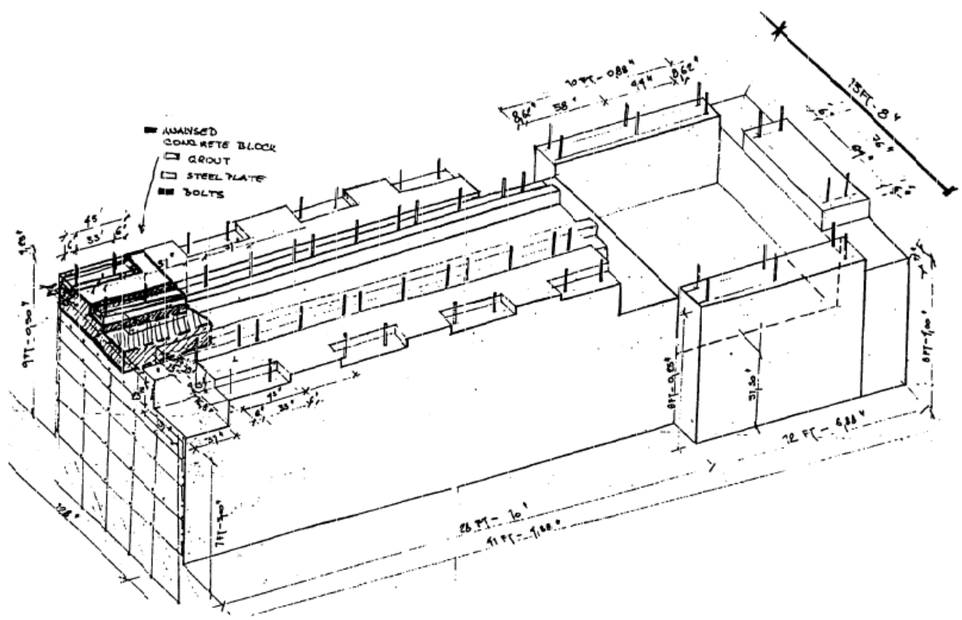

Figure 6-5. Drawing of a Block Foundation [Mandke, et al., 1994][36]

Reciprocating equipment is attached to the foundation using some variation of the arrangement shown in Figure 6-6. On top of the foundation, the compressor and driver are mounted on a number of metal or epoxy chocks and secured to the foundation with anchor bolts. In some designs, the chocks rest on a rail or soleplate that is mounted in the foundation, and in others, the chocks are set directly on the epoxy layer covering the foundation. Anchor bolts are typically terminated deep in the foundation, and with this length, they can be stretched sufficiently to maintain their preload under dynamic loading conditions.

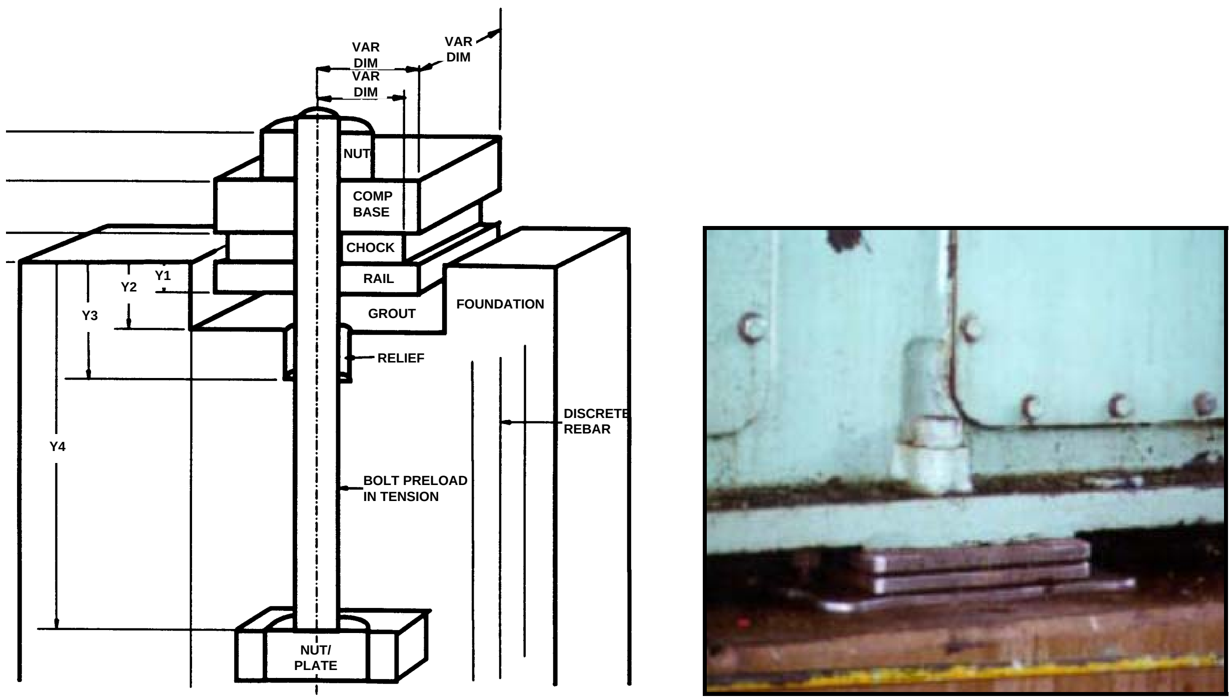

Figure 6-6. Mounting Equipment on a Concrete Block Foundation: Schematic Shown on the Left [Pantermuehl, et al., 1997][37], and Photo of Mounting Using Soleplates and Shimmable Steel Chocks Shown on the Right [Smalley, 1995][30]

The use of chocks provides several advantages, including definite support points, adjustability, ease of maintenance and replacement, and isolation of the equipment from the foundation. The 1-2 inch air gap between the equipment and the foundation created by the chocks has proven useful in reducing heating of the foundation and minimizing the resultant thermal distortion of the block and attached equipment. Although installing equipment on a full bed of grout with anchor bolts at regular intervals around the base was once a common practice, chock mounting of compressors and engines has become recognized as the preferred mounting method [Harrell, et al., 2004[38]; Mandke, et al., 1990[39]].

The primary threat to the integrity of concrete block foundations is cracking [Harrell, et al., 2001][29]. When cracks form in the foundation, the dynamic loads imposed from the reciprocating machinery cause the cracks to grow and propagate. Over time, these cracks can coalesce to form cracks large enough to separate the block into pieces and cause complete foundation failure.

While newer foundations benefit from design guidelines based on years of experience and research [Smalley, et al., 1997[40]; ACI 351.3R-04, 2004[41]], many of the older foundations currently in service have design features that make them susceptible to cracking. Among these are foundations that only have rebar around the periphery, rather than completely through the entire block. Without a high density of rebar, there is more opportunity for cracks to propagate unabated through the foundation. The “J” and “L” shaped anchor bolts once commonly used are now known to induce cracks in the concrete immediately upon tensioning. In addition, since these older anchor bolts are often made of materials with considerably less strength than that now available, they are not capable of being tightened sufficiently to produce adequate clamping force to restrain the machinery when high dynamic forces are present. Another common design feature of older foundations that is a threat to foundation integrity are sharp 90-degree corners, such as those found in the crankcase oil pan recess. These geometries cause stress concentrations, which are often seen to be the location of crack initiation. Figure 6-7 shows a crack in a foundation block that initiated from a 90-degree interior corner. When foundation integrity is compromised, typical foundation repair methods currently being used include placing post-tensioned bolts through the block, removing and replacing the top 18-24 inches of the foundation, and complete foundation demolition and replacement.

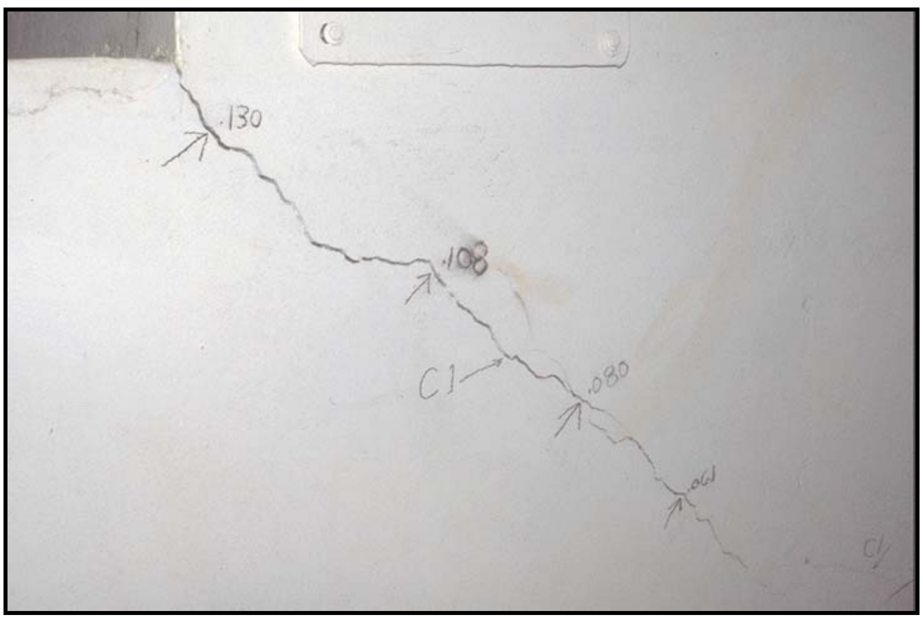

Figure 6-7. Crack in a Concrete Foundation Block Starting from a 90-degree Interior Corner [Harrell, et al., 2001][29]

A number of pipeline companies have installed or are planning to install new generation high-speed separable compressors in the 4,000 to 10,000 HP range. As an example of the range of high-speed separable machines now being considered by the industry, Harrell, et al. [2004][38], present the design of a foundation for an 8,100 HP compressor driven by a natural gas engine. This report points out that these very large machines exhibit compressor frame mode shapes and frequencies not seen in smaller units. It appears that the higher forces and the lower relative structural stiffness associated with the larger equipment size may have important implications for the design of foundations for these units.

6.1.1.3 Optimization Goals

The traditional optimization tasks for a particular compressor package or station are minimization of operating parameters, such as fuel/electricity consumption, power cost (which may include fuel and electric power costs), cost of operating unneeded machines, and compressor startup/shutdown costs [Carter, et al., 2001[42]; Jenícek, et al., 1995[43]]. It is predicted that fuel optimization will become a more important parameter in the coming decade as an increased emphasis on reducing emissions causes companies to look at reducing fuel consumption as one way to limit carbon dioxide emissions. An added benefit of optimization is that maintenance costs are often reduced due to better utilization of equipment.

The choice of what is to be optimized for a particular package or station varies for different companies and circumstances. For example, factors, such as local emissions regulations or whether or not fuel costs are passed on to customers, will influence the choice of optimization goals. Regardless of the parameters chosen, the optimization must be done subject to various physical and contractual constraints, such as minimum/maximum allowable pressures and flows, available horsepower, compressor rod load, torque limits, maintenance schedules, and permissible emissions.

Among the possible optimization goals, minimization of fuel consumption (which is not usually directly measured) appears to be the most common objective based on an inspection of the published literature. This may be due, in part, to the tremendous economic gains to be made by saving on fuel. Most compression is powered by natural gas taken directly from the pipeline, and approximately 3% to 5% of the gas transported by the pipeline is used by the compressors on that pipeline [Wu, et al., 2000][44]. It has been estimated that every one percent in fuel savings represents up to $5 million per year in economic benefits [Carter, 1996][45]. When it is possible, operating a compressor as close as possible to full-rated load (maximum torque) without overloading or underloading is one way to ensure maximum efficiency and minimum fuel usage per unit of gas delivered.

Most of the traditional optimization tasks call for steady-state optimization, in which optimal conditions, such as maximum rated torque, is desired for some fixed set of operating conditions (inlet and outlet pressures and flow rate). Steady-state optimization is a fairly mature technology as compared to transient optimization [Carter, et al., 2001][43]. Whereas steady-state optimization is concerned with only the final optimal state, transient optimization address such questions as how best to transition from one operational state to another and how to respond to rapidly fluctuating loads.

6.1.1.4 Approaches

In deciding how to operate a multi-unit compressor station to meet some specified parameter (typically station discharge pressure or gas throughput), there are basically two decisions to be made. The first is to decide which of the available compressors in the station should be running, and the second is to determine how each of the selected compressors should be operated. In deciding which compressors to run, the operator has to consider that a particular station may have a mixture of compressors, each with different performance characteristics and perhaps even different drivers (i.e., gas engine or electric motor). The question of how to operate a given compressor involves making decisions about speed, pocket settings, and possibly cylinder-end deactivation. As a result, the operation of a compressor station to meet a particular flow or pressure is generally a trial-and-error process without any guarantee of optimality [Chapman, et al., 2004][46].

Developing an algorithm for use in determining how to operate a compressor station is not a simple task. The main source of the complexity lies in the need to solve an integer variable problem (to determine which units should be on or off) simultaneously with a continuous variable problem (to determine how to operate the selected units). Furthermore, as the number of compressors is increased, the range of available options to be considered gets very large. This can be a severe limitation to inefficient optimization techniques, especially when attempting to optimize an entire network rather than a single station. Some approaches that are being used by the industry are as follows [Carter, 1996[46]; Wright, et al., 1998[47]]:

- Heuristics – Heuristics are essentially "rules of thumb" implemented by the station operator for deciding which compressors to run and how to run them. In the field, this approach is the most commonly used technique. The simplest heuristic is to bring as many units on line as needed and then back off each in equal proportion until the target condition (pressure or flow) is met. Although more sophisticated heuristics are used, this approach results in solutions that are between 1% and 10% suboptimal [Wright, et al., 1998][48].
- Mixed Integer Linear Programming (MILP) – In this approach, the station configuration is determined from the solution of a mathematical problem that represents the station operations. The appeal of this approach is that commercial software is available to solve problems of this type. However, the drawback of MILP is that fuel is treated as being linearly related to flow (or throughput), which is not a very realistic assumption, and thus suboptimal solutions are usually obtained.
- Mixed Integer Non-Linear Programming (MINLP) – This is essentially a generalization of the MILP technique that removes the unrealistic assumption that fuel is proportional to flow. While good results can be obtained with MINLP, this approach is very computationally intensive, especially for larger numbers of compressors.

The extent to which these techniques are being applied to reciprocating compressors in the field is not clear. Most of the results reported in the available literature are for stations with centrifugal compressors, although the particular techniques used would seem to be applicable to reciprocating compressors as well. Osiadacz [1980][48] and Osiadacz and Bell [1981][49] have presented an optimization technique specifically for reciprocating compressors and shown that this technique can provide a significant reduction in fuel consumption. Also, the usefulness of station optimization is limited by the fact that optimizing the pieces of a gas pipeline network does not optimize the overall system. For example, a network optimization plan might save fuel on a system-wide basis by calling for a particular station to be shut down or for gas to be routed through select parts of a system to take advantage of some ideal combination of compressors. Of course, the appeal of station-level optimization is that it is considerably simpler to implement than system-wide optimization. In addition, optimization of a system in which the selected stations have not been optimized and are assumed to have a certain efficiency or fuel usage can result in incorrect conclusions and suboptimal results.

The optimization of a station is further complicated by the need to account for the condition of each compressor and its driver. For example, if compressor valves are leaking or the engine timing or balance is off, the unit performance will differ from what was assumed in the optimization procedure. How to account for the current condition of equipment is an important issue for station optimization that is not being taken into account with current optimization approaches.

6.1.2 SUMMARY

This section of the report has reviewed the current technology being used in reciprocating gas compressors in the areas of compressor mounting and system operations and optimization.

Skid-mounting of compressor packages is a common practice for high-speed units (1,000 to 1,800 RPM) up to about 4,700 HP, while concrete block foundations are the mounting method used for most slow-speed integral compressors as well as most of the new generation of large, high-speed separables. Skid mounting of compressors offers the advantages of ease of assembly and portability, but problems associated with excessive vibration remain a major concern. On the other hand, block foundations offer more stiffness and damping than skids, but cracking of the block can be a threat to the integrity of these foundations. Many of the older foundations currently in service have design features that make them particularly susceptible to cracking.

Optimal operation of a multi-unit compressor station involves deciding which of the available compressors in the station should be running and determining how each of the selected compressors should be operated to meet some optimization goal subject to constraints on pressure and flow. Typical optimization goals for reciprocating compressor packages and the compressor stations using these packages are the minimization of operating parameters, such as fuel/electricity consumption, power cost, cost of operating unneeded machines, and compressor startup/shutdown costs. Among these, minimization of fuel consumption appears to be the most common in the industry, possibly due, in part, to the tremendous economic gains to be made by saving fuel. Heuristic approaches are the most common method used for determining the operating configuration of a compressor station. Better results can be obtained with mathematical optimization techniques, but the need to overcome some practical limitations makes this an area of ongoing research.

6.2 SYSTEM OPTIMIZATION NEW CONCEPT DISCUSSION

The goal of the system integration and optimization task within the ARCT program is to improve reciprocating compressor operations by developing technologies that optimize compressor system efficiency, reliability, and capacity at the unit and station level. This optimization will also take into account new, advanced technologies that are developed as part of the ARCT project.

Although optimization of entire pipeline networks is an important topic to the natural gas transmission industry, pipeline optimization is considered to be beyond the scope of the ARCT project. However, the present work fills an important gap in existing technology because current methods of pipeline optimization often employ simplistic models of individual compressors and compressor stations. By focusing in detail on the optimization of reciprocating compressors and compressor stations, industry will have information about how to optimally operate the units and stations to meet the requirements of the overall pipeline network optimization strategy.

6.2.1 OBJECTIVES FOR THE SYSTEM OPTIMIZATION CONCEPTS

The specific objectives for the system optimization that were identified as possible goals and constraints for the ARCT optimization work include the following:

- Minimize key operating parameters, such as fuel consumption, costs, or other measures of operating expense for individual compressors given the operating conditions while accounting for available horsepower, rod load, and torque limits.
- Optimize compressor operations for other variables, such as efficiency, emissions, maintenance, number of starts and stops, and other important parameters.
- Include compressor condition factors, such as leaky valves, out of tune engines, high operating hours, or other condition-based factors into compressor optimization.
- Incorporate new advanced technologies into the optimization of the performance for individual compressor packages.
- Optimize the operation of a compressor station by being able to identify the preferred compressors to operate for a given situation.
- Optimize the benefits and value per unit cost of compressor automation.
- Optimize the flexibility of operation (i.e., range of pressures and flow that a compressor configuration can achieve with good efficiency, integrity, reliability, and economy).
- Develop a guideline for parallel operation of different sizes and types of compressors at the same station.

6.2.2 APPROACH AND PLANS FOR SYSTEM OPTIMIZATION

Optimization of a single compressor consists of deciding how to operate the unit in terms of speed, clearance, etc., so that a specified objective is met subject to a number of constraints. In turn, the optimization of a compressor station consists of not only deciding how to operate each individual compressor (at which load steps), but also determining which of the available units should be running. This decision must consider the different compressor sizes and characteristics, the different conditions of the compressors (one may be in poor repair), and the expected pressure and flow conditions.

The approach to optimization for the ARCT program is to develop an optimization software tool that is capable of identifying operating strategies that meet given optimization goals, such an one of those listed above. This tool will be built around accurate and comprehensive models of the compressor and driver thermodynamic and mechanical behavior, and will include the capability to have other models added in the future as needed to meet the various optimization goals. It is anticipated that this tool will be useful in the design stage in making decisions about how to operate selected equipment. Also, the optimization tool is expected to be of value in analyzing installed equipment to identify inefficiencies in order to improve unit and station operations.

The central issue for optimizing a compressor package is to select a primary parameter to optimize and at most one or two secondary parameters or constraints on the problem. As an example, fuel consumption could be optimized, but if valve life or maintenance costs are adversely affected, this is not the optimization result that is required. A proper optimization algorithm will have the ability to optimize fuel consumption or some other selected variable with the constraint that valve life, torque level, and other parameters not exceed an assigned level or deteriorate from the present performance. The approach for optimization in this effort will be for steady state or quasi-steady conditions. Optimization for a compressor will be performed for a given pressure ratio and planned throughput or for a limited range of these operating variables.

Discussions of the optimization issues have been held with users, OEMs, and suppliers to identify the most potentially productive objectives for the ARCT effort so that the technology development can be focused on those aspects of system integration and optimizations that will provide the greatest benefit to the industry. The issues of what to optimize and what constraints to apply were determined in order to provide specific direction for the system integration and optimization effort. Some of the parameters identified as candidates for optimization in the ARCT project are as follows:

- Fuel/Power Consumption
- Efficiency/Power Cost (different from fuel consumption)
- Horsepower/Torque/Unit Load
- Emissions
- Unit Starts/Stops
- Maintenance Costs (Downtime or maintenance hours)
- Total Operating Costs
- Reliability/Availability

Selecting the best available compressors to use at a station and which load steps to use on the assumption that all compressors are equal will not result in optimum performance for the station when, in fact, there are opportunities for better performance from several of the individual compressors. Accounting for the current condition of a compressor is an important factor to consider in order to obtain a truly meaningful and useful optimization. Information on the condition of a compressor will need to be obtained from the operator or from the condition monitoring data for each compressor. For example, if a leaky valve is present on a compressor cylinder, that condition will need to be accounted for while optimizing the compressor for a given flow rate and pressure ratio. However, if a load step can be selected in which the leaky valve is deactivated, then the compressor performance is effectively returned to a good condition. For the development of compressor optimization algorithms, the selected method will have to be able to handle such compressor condition sensitive decisions.

The approach to the ARCT optimization effort will be to develop an initial version of the optimization tool built on simplified compressor and driver models that are a first approximation to the complexity of the actual equipment. Then, the models can be refined, as needed, depending upon the level of complexity necessary to obtain useful results. The goal in the model refinement process is not to develop high accuracy models, but rather to build models with enough sophistication to give good optimization results. Additional models will be incorporated into the optimization tool as required by the specific optimization tasks.

6.2.3 OPTIMIZATION TOOL

An initial version of the optimization tool for a single compressor unit has been developed to demonstrate the operation of the optimization tool and assess the need for more sophisticated component models. Optimization of fuel consumption for a gas engine driver at various speeds and load steps for a unit operating with given suction and discharge pressure and flow conditions was selected as the optimization goal to address first. The objectives of this effort were to obtain a few results to debug the tool, assess if more sophisticated models are needed, demonstrate the value of the results, and prove the feasibility of this approach.

This version of the optimization tool was developed in the MathCad technical calculation software package. MathCad offers a number of advantages over traditional programming languages for doing engineering calculations, especially in situations such as this where a prototype algorithm is being developed. In particular, MathCad has programming language-like features in a visual interface that makes it very easy to enter equations and data and view results interactively. MathCad also has many built-in solving and optimization routines that can be called for specific tasks.

The optimization tool was built using a modular approach so that additional modules could be added in the future as needed for other optimization tasks. Figure 6-8 is a block diagram showing how the various modules are connected in the initial version of the unit optimization tool. The arrows in the figure show the flow of information between the various modules. At the center of the block diagram is the master module, which interacts with the various supporting modules to perform performance mapping and optimization tasks. This module has the capability to produce performance maps, which show horsepower or fuel consumption over a range of speeds, clearances, or capacities, or, alternatively, this module can be used to directly seek a single optimal solution. The supporting modules in the initial version include models of a reciprocating compressor and a gas engine driver along with specified constraints on speed, clearance, and capacity. The results obtained include optimal operating conditions, compressor operation data, performance maps, and compressor cylinder gas mass, temperature, and pressure as a function of crankshaft angle.

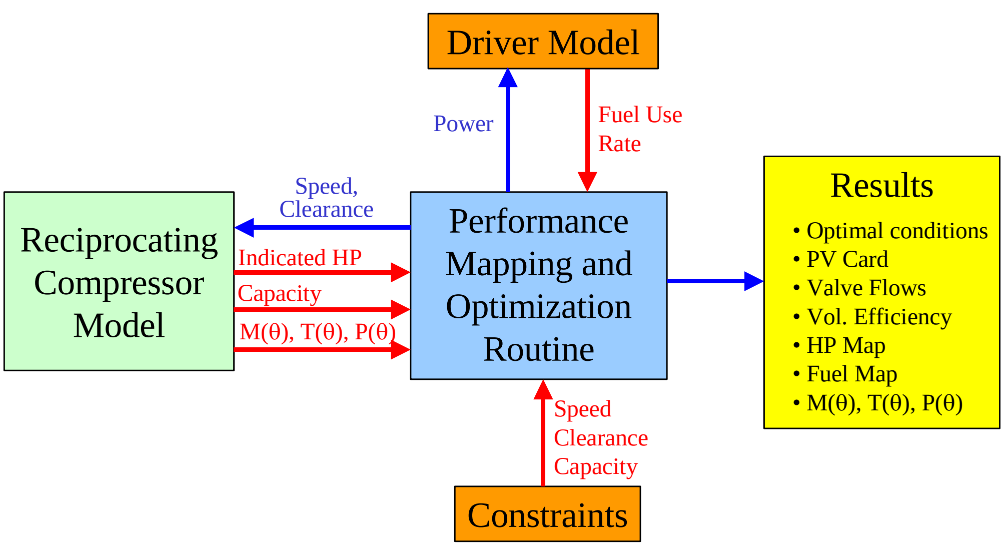

Figure 6-8. Block Diagram of the Unit Optimization Tool

6.2.3.1 Compressor Model

The compressor model that was developed for the initial version of the unit optimization tool is for one single-acting cylinder. This model considers only the behavior of the gas that is in the cylinder at any instant. Pressure losses due to the flow of gas through piping, vessels, or cylinder gas passages are neglected. Also, note that the results obtained for compressor horsepower reflect only the power required to compress the gas (indicated horsepower) and do not include the effects of other loss mechanisms such as friction. It is assumed that the suction pressure, suction temperature, and discharge pressure are constant at some given values.

The model is based on differential equations that express the conservation of mass and energy for the gas contained in the cylinder. These equations are solved to yield the mass and temperature in the cylinder at any instant. All of the other quantities of interest can be computed once these two fundamental variables are known. In other words, the equation for conservation of mass is as follows:

$$\begin{array}{c}\text{Change in}\\ \text{Cylinder Mass}\end{array} = \begin{array}{c}\text{Flow into}\\ \text{Cylinder}\end{array} - \begin{array}{c}\text{Flow out of}\\ \text{Cylinder}\end{array}$$

This equation shows that any change in the mass of gas in the cylinder is balanced by either flow into or out of the cylinder via the valves. The equation for the conservation of energy can be stated in words as follows:

$$\begin{array}{c}\text{Change in}\\ \text{Cylinder}\\ \text{Energy}\end{array} = - \begin{array}{c}\text{Heat Flow}\\ \text{to Cylinder}\\ \text{Walls}\end{array} - \begin{array}{c}\text{Work Done}\\ \text{by Gas in}\\ \text{Cylinder}\end{array} + \begin{array}{c}\text{Energy Flow}\\ \text{into Cylinder}\end{array} - \begin{array}{c}\text{Energy Flow}\\ \text{out of Cylinder}\end{array}$$

The conservation of energy requires that the energy of the gas in the cylinder change in response to heat loss, work done by the gas in the cylinder, and energy that is moved with the gas entering and leaving the cylinder.

There are two approaches to gas property calculations implemented in the compressor model. Originally, the model made use of a 20-component AGA (American Gas Association) equation of state. Due to the excessive amount of computation time required for evaluating the gas properties with the AGA equation, an ideal gas equation of state was implemented. To achieve better accuracy with the ideal gas equation, the AGA equation is called once to evaluate the specific heats and compressibility at the average of the suction and discharge conditions for the given gas composition.

The effect of the compressor valves is included in the model by considering the valves to be an orifice that offers a restriction to gas flow into and out of the cylinder. With this approximation, gas is allowed to enter the cylinder any time the cylinder pressure is less than the suction pressure. Likewise, gas leaves the cylinder when the cylinder pressure exceeds the discharge pressure. The orifice simplification makes it possible for the model to approximate the pressure drop that occurs when there is flow across the actual compressor valves, but it neglects the effects related to valve motion.

Figure 6-9 presents a block diagram showing how the compressor model operates. The model takes the user specified inputs and solves for the mass, temperature, and pressure of the gas in the cylinder over one crankshaft revolution starting from bottom-dead-center. The solution process makes use of the equation of state and valve models discussed above as well as several equations that describe the instantaneous cylinder volume and piston position. When the solution for one revolution is completed, the temperatures at the beginning and end of the cycle are compared. Because of the cyclic operation of the machine, these temperatures should be identical, but in general, they will not be since the initial temperature is not known precisely. To resolve this, the initial conditions are adjusted, and the solution is repeated until satisfactory agreement of the temperatures is obtained. Once the mass, temperature, and pressure solutions are complete, the model computes the identical horsepower and capacity.

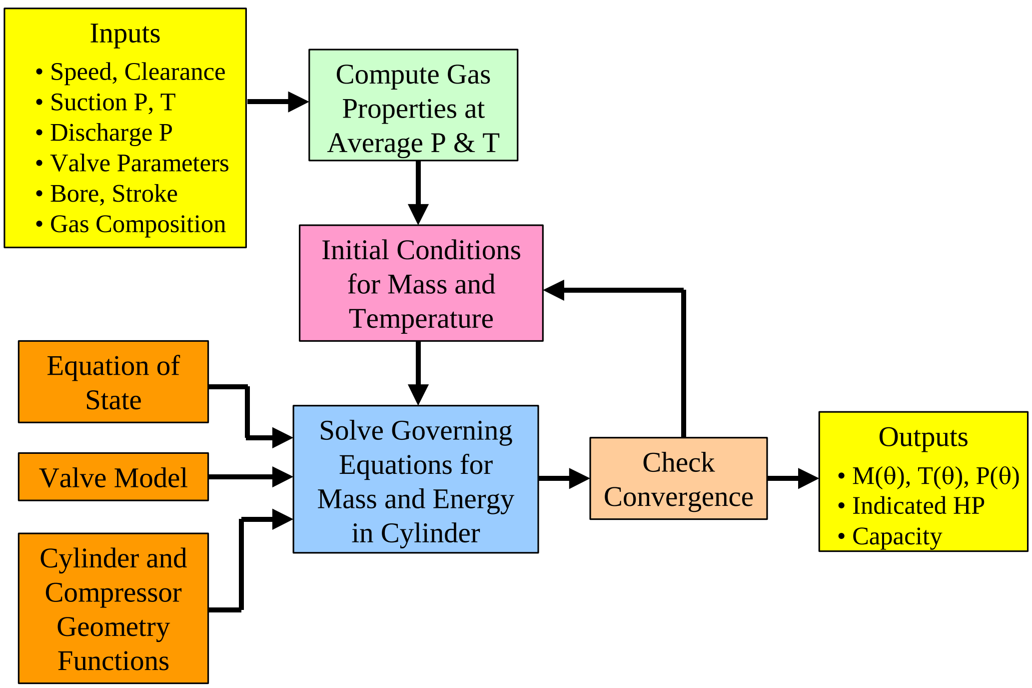

Figure 6-9. Block Diagram of the Reciprocating Compressor Model

To assess how well the compressor model performed, a number of validation cases were run to compare the model results with experimental data. The experimental measurements were taken from the data set obtained for the ARCT capacity control proof of concept work that was described earlier in this report. Figure 6-10 shows the comparisons between the experimental data and the model predictions in terms of indicated horsepower and capacity for six cases. The cases chosen were selected to exercise the model over as large a range of the experimental data as possible. Thus, the points shown cover a range of speeds from 500 to 900 RPM, clearances of 16% and 63%, and pressures ratios from 1.3 to 1.97. In all cases, the model predictions were within 5% of the experimental data for both indicated horsepower and capacity. Considering the assumptions made in the model and the uncertainties inherent in the experimental data, this level of agreement indicates that the model is providing reasonable results.

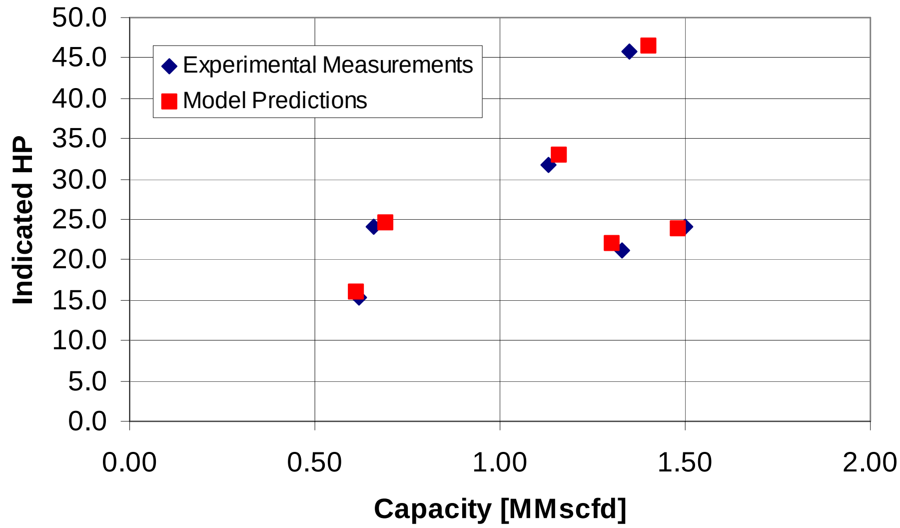

Figure 6-10. Comparisons of Experimental Data and Predictions from the Reciprocating Compressor Model

6.2.3.2 Driver Model

The driver model that was developed for the initial version of the unit optimization tool is based on heat rate curves for gas engine drivers, such as the example shown in Figure 6-11. In this figure, the heat rate is shown as being a function of brake horsepower only, while in the case of the data provided by some manufacturers, the heat rate is presented as a function of both horsepower and speed. Using these curves, it is possible to obtain the driver fuel consumption rate as a function of horsepower or horsepower and speed. In the optimization tool, once the compressor model has been used to determine the horsepower required for compression, the driver model is then used to calculate the fuel consumption rate for that known horsepower.

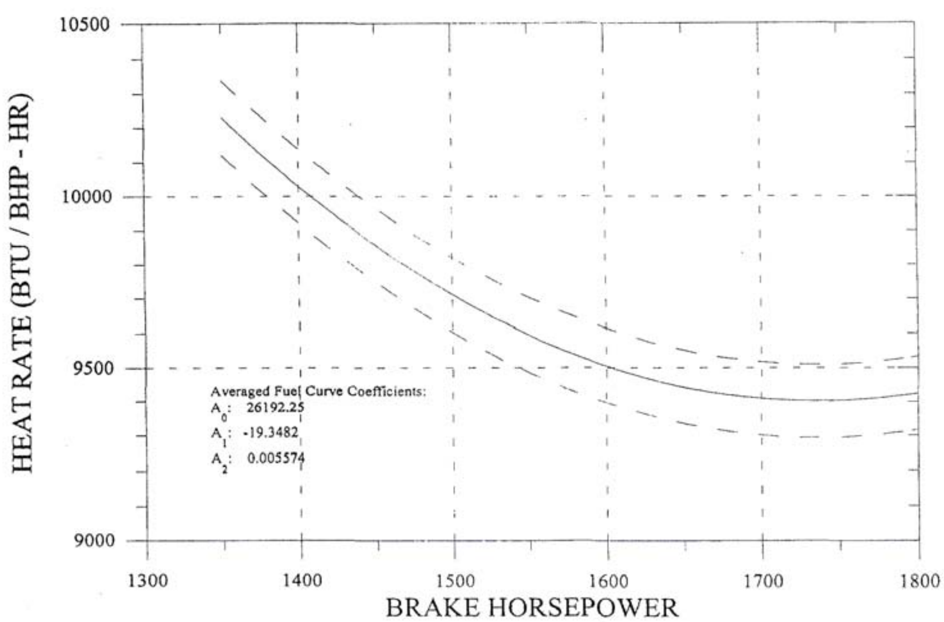

Figure 6-11. An Example Heat Rate Curve Used in the Driver Model to Obtain Fuel Consumption Rate

6.2.3.3 Optimization Examples

As a demonstration of the types of results that can be obtained with the optimization tool, a fuel optimization problem for a typical pipeline compressor was considered. The unit used for the example problems was a single-stage, high-speed separable compressor with a gas engine driver. It was assumed that the unit was operating under steady conditions with given suction and discharge conditions. Since the compressor model in the optimization tool was limited to modeling only one end of a single acting cylinder, the power and capacity measurements were multiplied by 12 to approximate both ends of the six-throw compressor. All of the calculations for the demonstration problems were based on the indicated horsepower for the given conditions.

In the first demonstration problem, the capacity was specified, and the objective was to identify the optimal speed to minimize the fuel consumption. In this scenario, the compressor clearance is varied as the speed is changed, such that the given capacity is met. Figure 6-12 and Figure 6-13 are performance maps showing the indicated horsepower and fuel consumption rate as the speed is varied from 800 RPM to 1,200 RPM. The separate lines in these plots show the results for several different given capacities. As would be expected, the horsepower required for compression is reduced as the speed is slowed (with a corresponding reduction in clearance). The same trend is also seen in the fuel consumption curves, which show that, for the characteristics of this particular driver, the fuel usage is minimized by operating at lower speeds. The same results could also have been obtained directly with the optimization tool by using the optimization functions without the need to create the performance maps. However, the performance maps offer some additional insight into the behavior of the unit.

A second demonstration problem was considered in which the optimization tool was used to determine how to minimize fuel usage in a case where capacity is to be reduced from some given operating point. Figure 6-14 shows the results of two capacity reduction methods: increasing clearance at fixed speed and reducing speed at a fixed clearance. These results indicate that for the characteristics of this particular driver, fuel consumption is less if the capacity is reduced by slowing the unit at a fixed clearance as opposed to increasing the clearance while running at a constant speed.

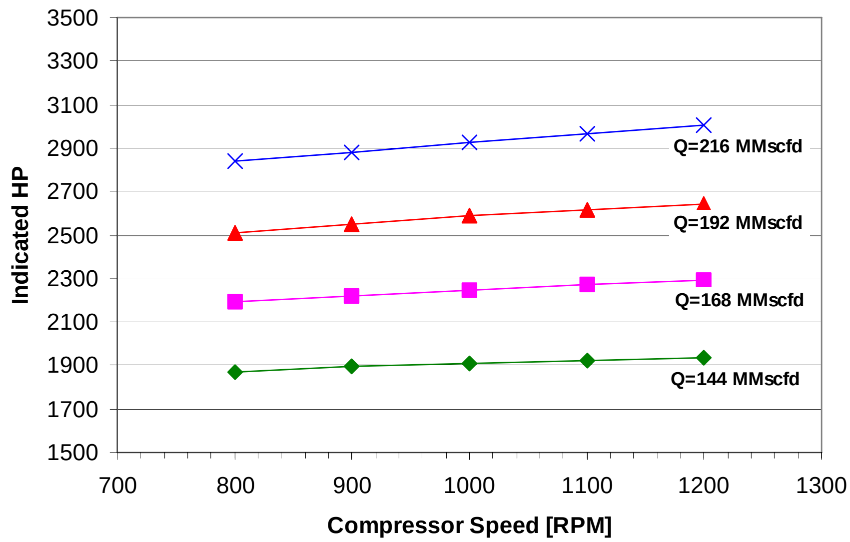

Figure 6-12. Performance Map Showing Indicated Horsepower Over a Range of Compressor Speeds for Several Given Capacities

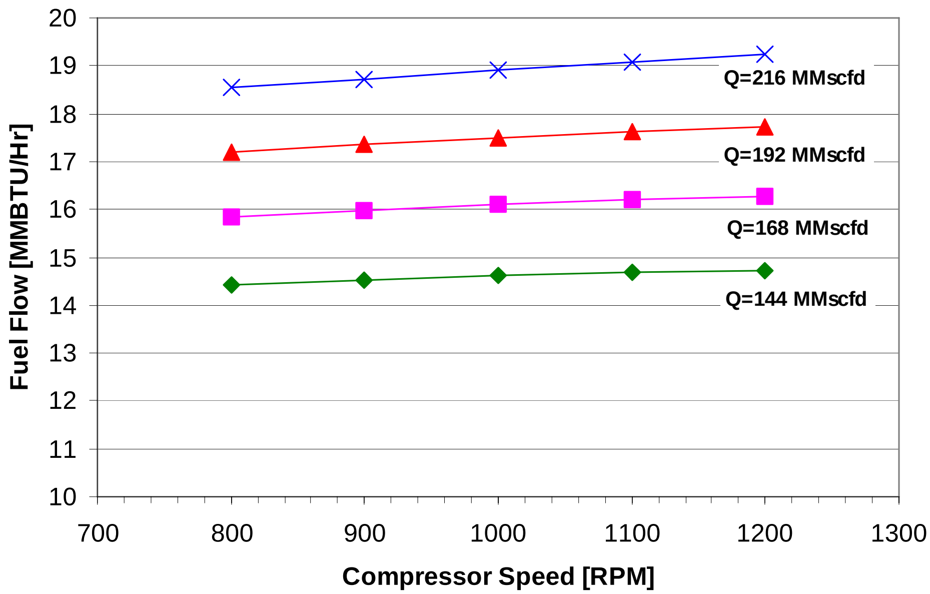

Figure 6-13. Performance Map Showing Fuel Consumption Rate Over a Range of Compressor Speeds for Several Given Capacities

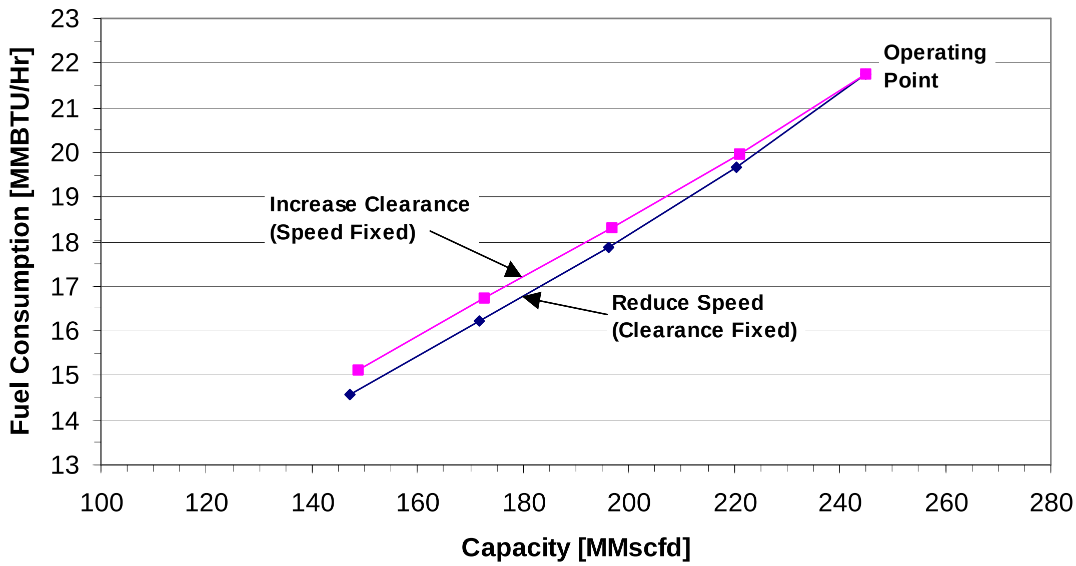

Figure 6-14. Performance Map Showing Fuel Consumption Rate for Two Methods of Reducing Capacity

6.2.4 RECOMMENDATIONS FOR FUTURE WORK

The first version of the reciprocating compressor optimization tool, consisting of compressor and driver models, has demonstrated the potential for this type of tool to provide information that can be used to improve compressor operations. Validation of the compressor model showed good agreement between the model predictions and experimental data. The following are some recommendations for future work that can be done to further the development of the optimization tool:

- Extend the compressor model to double-acting, multi-cylinder compressors.
- Upgrade the driver model to make fuel consumption a function of both horsepower and speed.
- Include more realistic constraints such as allowable clearances, horsepower versus speed, etc.
- Continue comparisons with experimental data and validate the compressor model against data for a complete compressor.
- Assess the need for further refinement of the compressor model in areas such as gas properties, valves, pressure and mechanical losses.
- Improve the valve model to include the effects of the reduced pressure drop valves being developed as part of the ARCT project.
- Integrate the ARCT valve life model to permit valve life optimization.

6.3 SYSTEM MOUNTING DESIGN GUIDE DISCUSSION

This section reports on work accomplished on the subject of mounting guidelines for modern and next generation pipeline reciprocating compressor installations. The advisory group for this effort has members from a number of industry sectors, including El Paso Corporation, Ariel, Caterpillar, and CSI..

This guideline activity will add to a well-established area of GMRC core competence, and address issues of current and growing significance as large separables are more and more widely deployed. These issues include:

- Excessive machinery vibration and movement.
- Difficulty in maintaining alignment (coupling and crankshaft).
- Wide variation in mounting methods with variable results.
- Incomplete understanding of proper analytical methods and lack of uniform approach.

It is intended that the Guidelines will:

- Document considerations implicit in mounting decisions.
- Help ensure more reliable installations by best practice methods of design, analysis, and installation.
- Document successes and underlying practice as well as problem areas.
- Foster effective communication of technical information between OEMs, packagers, and pipeline customers and between pipeline engineering and operations.
- Address cost/benefit issues.

The work so far has included a careful planning effort and initiation of an information gathering process. Success depends on a wide range of input sources, as well as some carefully directed analysis; thus, the content, sequence, and process of information gathering called for well-defined plans from the start. With plans set down and an outline prepared for the eventual guidelines, an initial information gathering effort was initiated. The information gathering seeks to build an experience base with input from pipeline companies and to develop initial guidance from this experience base. A set of questions was prepared, and along with the draft an outline of the guidelines were sent to knowledgeable individuals in a series of pipeline and storage companies, including El Paso, Duke, Williams, Kinder Morgan, Panhandle Energy, Centerpoint, Nisource, Sempra, and AGL Resources. Within some companies, several individuals received the list of questions. A number of responses have been received and more are expected. Some individuals have expressed the wish to provide input just in a telephone call; others have sent written responses; one respondent organized a panel of experts; this panel provided interactive input, which has been documented, and is now being reviewed by the panel for accuracy.

Contact with OEMs has been initiated—including engine, compressor, and motor. As an initial step in gathering information from compressor OEMs, each (four companies in all) has provided their standard packager specifications, which in particular identify the requirements for skid design, function, and stiffness characteristics from the OEM's perspective.

Rather than write this report section by expanding the documentation of activities outlined above, it has been decided to start writing guidelines material based on existing knowledge and information already gathered. It must be clearly understood that the resulting content is an initial and incomplete draft with known gaps; however the result of this process reflects current thinking and knowledge and enables review by other SwRI team members, by the PSC, by potential co-funders, and by the Industry Advisory Group for the mounting guidelines project. The result meets DOE reporting requirements, and maximizes the value of the current reporting effort by moving towards the final guidelines document rather than simply listing and describing incomplete activities. This report section will end with a list of planned future work designed to consolidate the guidelines and to fill the known knowledge gaps.

6.3.1 SCOPE OF GUIDELINES

These guidelines will address the mounting of reciprocating compressors. Mounting includes support of the compressor, the driver, compressor cylinders, dampener bottles, and other appurtenances; it covers the functions provided by the skid or skids, reinforcements, grout and concrete, the mat and foundation, anchor bolts, chocks, piers, clamps, cylinder supports, frames, and other support structures. The guidelines will emphasize modern high-speed separable compressors, with power of approximately 2,000 HP and above, with either fixed or variable speeds in the range from 500 to 1,200 RPM. The guidelines will seek to address decisions, which will influence the short and long term integrity of the installation—whose expected life for pipeline service can be expected to exceed at least 20 years. These decisions include both hardware choices and methods used to assure integrity of the hardware choices and to guide detailed design decisions.

As stated, the guidelines will emphasize pipeline service; those responsible for installations in other services may choose to apply many of the general aspects of the guidelines but should recognize the special considerations of pipeline service implicit in the guidelines (typically single stage, ratio below 1.5, high availability requirements, with need for high thermal efficiency, and long service life).

The guidelines will consider alternative mounting types including block mounting, skid mounting and hybrid mounting (which may combine features of skid and block mounting). In general, the guidelines do not address slow speed integral compressors, although a number of lessons learned from experience over the lifetime of the large number of slow speed integrals in pipeline service will be adapted to the present guidelines.

6.3.2 DEFINITIONS AND TERMINOLOGY

6.3.2.1 Mounting

Mounting is the location, weight support, alignment, mechanical load management, and tie-down, of the compressor, its driver, associated systems, and appurtenances.

6.3.2.2 High and Medium Speed

For the purposes of these guidelines high speed is considered to be 900 RPM and above. Medium speed is 500 to 900 RPM. These definitions are somewhat arbitrary and do not greatly influence the mounting guidelines.

6.3.2.3 Package

In principle, a package is the compression system—everything between the suction flange and the discharge flange; a turnkey system which just has to be set in place, connected to compressor station gas headers, connected to fuel or power lines, started, and operated. In fact, it is this turnkey concept, which has needed some "qualification" and rethinking when applied to some of the large pipeline compressors, which have been installed in the last few years.

6.3.2.4 The Packager

The Packager is the single source and point of contact for the procurement, engineering, assembly, transportation, installation, startup, commissioning, and problem resolution for the compression system. The packager purchases and assembles the compressor, driver, coupling, vessels, coolers, controls, oil system, etc., on a skid, delivers the package to the end user, installs it, starts it up, and assures its function as a system for a finite period, often with availability guarantees.

6.3.2.5 Integral Engine Compressor

A compressor and engine in the same frame, normally burning the same gas that it compresses, whose single crankshaft transmits power from the power cylinders to the compressor cylinders. Often referred to as slow speed integral engine compressors, their speed is most commonly 250 to 350 RPM.

6.3.2.6 High Speed Driver with Low Speed Separable Compressor

This configuration has been proposed on occasion for pipeline service with the potential advantages of combining the high thermal efficiency of a low speed compressor with the low heat rate of a modern high-speed engine. It includes a gearbox to reduce speed from the engine to the compressor. A small number of examples exist in other services.

6.3.2.7 Grout

A pourable material, which cures and hardens with time to form a structurally stiff layer, and fills the previously empty space into which it has been poured. The hardened grout forms a structural element of the mounting system for skids, sole plates, chocks, compressors and their drivers. Grouts in this application are most commonly epoxy material formed from long chain polymer compounds, which can be formulated to provide a Young's modulus of 1 Million PSI or more. The modulus has a time dependent characteristic sometimes referred to as creep, which can cause slow changes in deflection under load.

6.3.2.8 Anchor Bolts

Anchor bolts for compression equipment normally pass through a hole in a plate or flange, and via a nut and washer, apply a vertical downwards force on the upwards facing surface of a component to be mounted (of which the plate or flange forms an integral part). Anchor bolts can be embedded in concrete or may be terminated at their lower end by a head, which bears against a surface of the steel structure onto which the component is to be mounted.

6.3.2.9 Chock

A chock is a stiff, flattish element, rectangular in plain view, with typical dimensions of 6 to 12 inches on a side, and 1 to 3 inches of thickness. A chock separates a component to be mounted from the structure to which it is to be mounted. Normally, a single anchor bolt passes through the chock, and the tensioning of the anchor bolt produces a high normal force between the interfaces involved (e.g., the compressor to chock interface, the chock to skid interface, or the chock to concrete interface).

6.3.2.10 Sole Plate

A sole plate performs some similar functions to a chock, but normally has a larger area in plain view. The sole plate is normally grouted in place on top of a concrete block and may provide several points of support for a compressor or driver.

6.3.3 HISTORICAL PERSPECTIVE

The natural gas transmission industry has made heavy use of reciprocating compressors at stations along the pipeline system. These compressors move the gas and overcome frictional resistance to flow between stations. When first applied in this industry, the compressors were slow speed—first, the 180-RPM horizontal compressors, and then the 250 to 350 RPM engine-compressor combinations, now termed slow speed integrals. These integrals still form the backbone of the pipeline system, and economics will drive most pipeline companies to keep these integrals operating for many years to come.

Over the years, centrifugal compressors have also become a popular choice and a significant fraction of the system's expansion since 1960 has come through deployment of gas turbine driven centrifugal compressors. However, most operating companies must deal with widely varying conditions and have determined that a mix of reciprocating and centrifugal compressors gives them the operating flexibility they need. Thus, they will continue to operate their old, slow speed, integral, engine compressors even with the added constraint of environmental regulation, and will add (or replace) horsepower with a balanced mix of centrifugal and reciprocating compressors.

New reciprocating compressors, in general, will not be slow speed. While an occasional slow speed unit may be chosen, present trends show clearly that the majority of new reciprocating compression in pipeline service will be medium and high speed separable compressors, with a mix of gas engine and electric drives. The sustained emphasis on reciprocating compressors results in part from economics as well as operating considerations—a very competitive market exists for high and medium speed separable engine compressor packages; the engines now have very attractive heat rates (6,500 BTU/BHP-Hr and below); and under specific ideal conditions, the positive displacement reciprocating compressor can achieve thermal efficiencies at 90% or more. The shorter stroke of a high-speed compressor reduces the required span of the cylinders and frame in the horizontal direction perpendicular to the crankshaft. This reduces the amount of steel and cast iron needed to achieve the kinematic requirements of reciprocating piston motion, and significantly reduces the cost of a medium or high-speed separable package.

The shorter cylinders of medium and high-speed separables also reduce the natural area available for gas to flow in and out of the cylinder for a given capacity. This reduced flow area increases the flow resistance because the gas flows through the valves with higher velocity, and pressure drop tends to vary with the square of flow velocity. The result of this higher flow resistance is seen especially in pipeline compressors across which the pressure rise tends to be a fairly small fraction of the line pressure (compression ratios as low as 1.1 are quite common). Compressor manufacturers have expended much effort to maximize the flow area for a given cylinder size for pipeline application, but the physical configuration constrains how far this can go.

The growth of the market for packaged separables has come from the needs of the gas industry upstream of the regulated pipelines. Compression needs exist at the wellhead, in gathering systems, in gas processing, and in boosting of gas pressure to meet pipeline inlet requirements. The needs to link gas supply sources to gas markets were (and continue to be) fast moving and opportunistic. Numerous packager companies formed to meet these "upstream" needs—the packager would provide turnkey compression, taking care of the many logistical issues involved in compression and would offer a range of competitive lease, rental, or contract operation options. Some compressor manufacturers would provide their products only through packagers.

In the process, the packagers learned how to engineer, procure parts for, assemble, transport, deliver, and install at site an integrated, operating package for the lowest possible cost—a natural and innovative result of competition. The skid mount was a natural choice from the start—it could be designed, cut from I-beam, and welded up in the packager's shop. There, it provides a base in a relatively clean environment to assemble and component test the oil system, cooling system, electric and control systems, mount the compressor, and align it to its driver. To provide a base and carry the weight of components, an I-beam structure is ideal, but under lateral forces, the I-Beam cross-section tends to deform and needs reinforcement.

First, cost was a driving factor, together with availability guarantees; long life was less of a requirement and in the upstream industry, capacity and availability normally have more perceived value than efficiency. Sometimes the integrity and rigidity of the package fell victim to cost savings; sometimes the action of reciprocating kinematics and intermittent gas flows dynamic forces on the skid, the vessels, and the piping led to damaging vibrations, excessive motion, fatigue failures, and potentially harmful deterioration in alignment.

Over time, weights, sizes, and power densities of packaged compressors grew, driven by economies of scale; in general, size increases exposed new rigidity and integrity issues. Problems have generally been solved with more steel in the package, with more intensive engineering and management of forces and pulsations, with more rigid compressor frames, and with targeted reinforcement. Yet, problems still arise and these guidelines seek to consolidate past experience, good and bad, and to define approaches and principles, which will minimize the probability of future problems.

The leading compressor manufacturers whose products are used in packages each have developed a set of standards for packagers. In general, these require that the skid shall be adequately stiff to carry loads induced by the compressor and its operation, to carry the weight of the installed equipment when being hoisted, and to keep the natural frequencies of skid and installed vessels outside the inherent excitation frequency range of the compressor. These are very important requirements and, in general, the guidelines developed under this project will complement these standards and provide specific details to achieve these and other requirements.

With very few exceptions, slow speed integrals are mounted on concrete blocks. Through the 40's, 50's, 60's and 70's, the standard installation involved a full bed grout; a rectangular concrete block was first installed, the compressor frame was held in position and aligned with jacking screws, and then the base of the compressor frame was uniformly grouted to the concrete block using a cementitious grout. Vertical anchor bolts set in the concrete passed through holes in the flange at the bottom of the frame and nuts tied the frame down to the block.

A beneficial evolution, which started in the 1970s and progresses to this day for slow speed integral units, is the replacement of cementitious grout with epoxy and the use of chocks between the compressor frame and the concrete block. The epoxy grout could be formulated with substantially higher strength (three or more times that of cementitious grout); it was a more pourable material; it could be stronger in both tension and compression than cementitious material; and it offered more thermal resistance. Chocks provided clearly defined support points under each anchor bolt rather than a continuous support over the wide base area of the compressor frame. The chock materials were variously steel, grout, or a composite material. Some chocks offered height adjustability, through tapered interfaces, or through replaceable shims. A further benefit of the chocks was the air gap, which added thermal resistance to conductive heat transfer, between the bottom of the frame and the concrete (except at the chocks) and also allowed cooling air to flow and carry heat away by convection. As a result, the temperature difference from the top to the bottom of the block and the associated tendency to "hump" or bow upwards were reduced.

New installations increasingly would use chock mounts and most pipeline companies instituted a regrout program. In recognition of the deterioration of their older foundations and mounting systems, these programs replace the concrete block partially (near the top) or entirely, and remount the compressors on chocks instead of a full bed grout. Regrouting a compressor represents a major maintenance cost item outside the scope of normal annual maintenance for a unit. As a result, regrouting is generally scheduled for units with mounting in the worst conditions of the fleet, and a regrout program for a large fleet would extend over many years.

In parallel with this evolution, organizations such as GMRC (then PCRC) and PRCI took a critical engineering look at the physics of compressor mounting: the heat transfer and thermal distortion issues just mentioned; the role of gas and inertia forces inherent to operation of the compressor; and the transfer of these forces through the frame, the connecting rods, crankshaft and bearings, the frame-chock joints, and into the concrete.

Some inherent flaws in the most widespread methods of analyzing and designing for the transmission of shaking forces and moment were identified and emphasized by these studies. With appropriate arrangements of crankshaft throw angles, the compressor manufacturer could offset forces and moments from some throws with opposing forces and moments from other throws, leading to net forces and moments substantially lower than the forces and moments from individual throws. The manufacturers of the compressors (both integral and separable) would normally provide the integrated shaking forces and moments for the entire compressor (eight quantities in all: (vertical/horizontal)X(primary/secondary)X(forces/moments)) to those responsible for engineering the foundation and mounting system.

With information provided on overall shaking forces and moments, the natural and often unstated or unrecognized assumption in foundation engineering was that the compressor frame would act as a rigid body under the action of these forces and moments. The apparent load-carrying requirement for the tie down locations was sometimes disarmingly low, and mounting systems designed to comfortably carry these overall forces and moments could in fact have inadequate local load carrying capability. The frame is far from rigid and without restraints designed to carry a major fraction of the forces from individual throws at or near the line of action of the force, the frame was unable to meet the implicit expectations for load carrying; relative motion could occur between frame and foundation at the tie-down point. The frame could move under the dynamic loads and some “after the fact” increase in diameter, number, and tightness of anchor bolts were required to restore integrity to the installation.

Although tie-down forces were sometimes inadequate for the compressor driven loads, the speeds and speed related excitation frequencies of these old compressors were sufficiently low that they did not excite natural frequencies of the compressor frame as a “box” on its mounting. This is by no means to say no dynamic problems existed with slow speed integrals—the gas flow dynamics and acoustic natural frequencies of the gas in the piping created substantial challenges; torsional vibrations periodically occurred with new configurations of crankshaft and engine-compressor and led to some notable crankshaft failures and solutions; and the cylinders would sometimes experience lateral resonances excited by gas forces at 6 to 10 times their slow operating speed. However, the compressor was attached rigidly enough to the block that its lowest system natural frequency involving mounting flexibility was high enough to avoid coincidence with any strong excitation frequencies, as was the cylinder stretch natural frequency.

These characteristics of slow speed integrals are mentioned at this point because with medium and high speed compressors, the frame-block natural frequency and cylinder stretch natural frequency can fall into the range of strong excitations at about 8 to 12 times running speed. Vertical system vibration modes involving cylinders and suction bottles can also become more troublesome than with slow speed units.

In the process of regrouting, many old integral units and installing new ones, some practices, old and new, were questioned and slowly changed, specifically:

The use of J and L Bolts—these have an asymmetric termination which naturally produces a local overload on the concrete; the preferred termination is a symmetrical nut or nut and washer.

The use of short anchor bolts, sometimes only 2 feet long—this can make the stretch in the anchor bolt sensitive to any small relaxation at the chock, in the grout, or at the termination point. The termination is also a point of high stress, both compressive and tensile, and when near the compressor, the termination stresses may interfere with stresses induced by the transmitted dynamic forces. The longer the anchor bolt, the better, and extending anchor bolts into the mat below the foundation block minimizes any termination point cracking, and if cracking does occur, it will make it invisible and as far removed as possible from dynamic stresses and leaking oil. Anchor bolts as long as 12 feet have been successfully installed.

Low rebar density, sometimes only 1% or less by volume. The static and dynamic stresses in the concrete for a reciprocating compressor installation can be high and both compressive and tensile. The tensile stresses can cause cracking—rebar has a limited benefit in reducing the tendency to crack, and its main benefit is that it limits and arrests the progress of cracking. Some practitioners, based on experience, recommend #8 rebar on 8-inch centers. GMRC research showed that about 1% rebar density was desirable, which is roughly consistent with this practice.

Anchor bolts embedded in the concrete over their length; any stretch of an embedded anchor bolt produces relative motion and/or stress in the concrete. Sleeving the anchor bolt over its length makes it free to stretch over its length.

Low strength anchor bolts; the yield stress sets the limit to which an anchor bolt can be stretched. The GMRC and ACI recommend ASTM A193 Grade B7 anchor bolts. These provide 105,000 PSI yield strength for 2.5-inch diameter and below and 95,000 PSI yield strength for larger diameters.

Bolt Stretch to a small fraction of yield strength. Meeting tie down requirements with a limit on bolt diameter often requires the maximum anchor bolt force possible. This is because the horizontal restraint at grout-metal interfaces comes from friction alone and coefficients of friction can fall to 0.2 or below, so the force normal to the interface produced by the anchor bolt must be five times or more the required horizontal force to be restrained. A bolt can be stretched to a high fraction of yield stress without danger and where needed 80% of yield is allowable, giving over 145,000 lbs anchor bolt preload for an effective area of 1.75 in^2 with ASTM A193 Grade B7 bolts.

Thick Epoxy grout layers. An overly thick epoxy grout layer, besides being costly, can impose high thermal stresses and distortion because its coefficient of thermal expansion exceeds that of concrete by a factor of 2 or more. In addition, it can exaggerate the loss of preload in anchor bolts experienced as a result of creep in the compressed grout. The recommended thickness is two to four inches, and as much as six is normally acceptable. Also thick epoxy layers can develop excessive internal heat during curing.

Use of Expansion Joints. Without expansion, joint grout tends to crack because it has such a thermal expansion mismatch with concrete. An expansion joint every three to four feet is generally recommended.

Thus, much was learned from experience and analysis of slow speed integral engine compressors, and today a well engineered new foundation or upgrade to an existing foundation can be installed for a slow speed integral reciprocating compressor. However, even for gas transportation service, the economics of the situation dictate that most future installations will use medium or high speed reciprocating compressors. The capital cost savings inherent in a medium or high-speed separable package compared to a slow speed integral is substantial and probably growing. This savings, coupled with the other benefits discussed, ensures slow speed integrals will rarely, if ever, be deployed for new installations, and the knowledge developed on slow speed integrals must be adapted to modern separable compressors if it is to be of value to new pipeline compressor installations.

As discussed earlier, there has been a trend to increasing frame size in medium and high speed separable compressors; a substantial number of units in the power range from 4,000 HP to 8,000 HP have been installed on skids in pipeline and storage service. For the bigger units, the combination of size, speed, and power density, with a skid mount has introduced a number of new problems and challenges, some of which are directly related to mounting practice. Transportation is an issue above a certain size, and the major components must be disassembled before shipping to a site from the packager's shop. Even smaller units down to 2,000 HP have had substantial problems associated with the mounting, and the need for guidelines to avoid these problems is clear.

Some contend that the answer is to block mount medium and high speed separable compressors above a certain size, although the size cut-off is open to debate; others contend that the advantages of skid mounting are so substantial that they would choose it universally; others suggest derating the application to avoid the highest speeds and power densities.

There are two known examples of a block mounted high speed separable in other service, and at this time an 8,000 HP storage unit is being installed with a block mount in Louisiana at the Jefferson Island Storage facility. However, to this date the number of block mounted medium and high-speed separables remains very small.

6.3.4 Situation Statement

The current situation can be briefly summarized as follows:

- All or almost all new pipeline reciprocating compressors are now "medium or high" speed separables.
- Essentially all are skid mounted.
- Capital cost savings help to drive this trend.
- Significant dynamic problems have arisen in pipeline units above 4,000 HP.
    - Frame/Block First Rotational Model (typically 95 to 120 Hz)
    - Cylinder Stretch Mode (150 to 190 Hz)
    - System Vertical Modes (100 Hz Up)
    - Shell Modes (250 Hz Up)
- These problems represent coincidence of high excitation and natural frequencies not encountered with slow speed integrals.

- A number of these problems are system issues influenced by mounting of compressor, driver, bottles, cylinders, and other vessels.
- In smaller units with 8-cylinder 4-stroke drivers and 4-cylinder compressors, pronounced problems have been observed at 4X rotating speed.
- In addition to considering high frequency vibration modes, it remains important to manage 1x and 2x loads on the mounting system even if they do not excite resonances.
- Not all large medium/high speeds report dynamic problems.
- Initial misalignment and deterioration of alignment have been noted on some skid mounted units.
- Packages are not transportable as a single unit over a certain size, thus, they show up at the installation site in three separate pieces.
- For some large pipeline examples, drivers and compressors are shipped straight to the site from the manufacturer.
- Large pipeline applications have raised the question as to whether a "size" or other criterion exists where block mounting the compressor directly is preferable.
- A very few examples of block mount designs for high-speed compressors now exist.
- High-speed compressor efficiency is more difficult to maintain over all conditions (and has been reported as approximately 5 full percentage points below a competing slow speed).
- Drive options are more limited as size increases; engines are available to about 8,000 HP, but a wider choice of gas engines exists under 5,000 HP.
- Motor drives are available in all needed ranges of HP, VFD, or fixed speed.

6.3.5 TYPICAL PIPELINE COMPRESSOR APPLICATIONS

In today's market, pipeline companies use reciprocating compressors in a variety of applications. These can include mainline transmission, storage injection and withdrawal, compression added on a lateral line, or a combination of these functions.

Mainline applications generally involve larger horsepowers, and examples exist of medium speed separable reciprocating compressors as large as 8,000 HP in mainline transmission, and many in the range of 4,000 to 8,000 HP. (Existing frame sizes and rod load capacities could produce compression power as high as 12,000 HP). These applications are particularly critical, for a number of reasons. First, mainline transmission is the primary business of the pipelines, and deliverability at all take off points on the system is dependent on the reliability and availability of compression assets all along the line. With more capacity packed into a single unit, the more the system depends on that unit. On a high load day, all compression capacity becomes essential. Secondly, firm transportation contracts require regulated interstate pipelines to assure capacity, which is a function of available compression (whether it is used or not).

Storage can require similar horsepowers to mainline transmission. Duke's Moss Bluff storage facility has an 8,000 HP separable, and AGL's Jefferson Island Storage facility is installing a similar sized unit. Energy Transfer Partners now own the Bammel storage field installed originally by Enron, with a number of 7,000 HP motor driven compressors. Storage service can also be critical to operation of an individual company, and to the pipeline system as a whole; it is offered to a pipeline's clients as means for these clients to manage their gas purchases and to assure deliverability on high demand days.

The most likely example of mixed service is a combination of transmission and storage. This is a complicated service, which requires the flexibility to switch from single stage operation to two or more stage operation. Nisource Wellington Station has an example with a 6-cylinder 4,500 HP high-speed separable, motor driven at 1,200 RPM, designed for at least eight different flow and pressure conditions.

Lateral line compression most typically provides extra compression for a power plant. As an example, Williams has installed two 5,000 HP electric motor driven medium speed separables running at a fixed speed of 720 RPM on a lateral line. This is relatively large for this type of service, and examples exist in the range of 2,000 HP. The service tends to be more variable, often driven by demands from a power plant. The variability in power plant gas deliveries can be up and down daily on very short notice. In most cases, the gas is to drive a peaking or combined cycle gas turbine plant, with high inlet pressure requirements—sometimes approaching 1,000 PSI for the most efficient aeroderivative gas turbines, such as the Rolls Royce Trent.

The characteristics of mainline transmission are typically low ratio (1.1 to 1.4), single stage operation, with a typical service factor of 60%. High availability is a requirement to meet regulatory requirements and swings in actual demand. Lifetimes of 20 years and above are required.

Storage service requires a wider range of operation—normally multi-stage with discharge pressures as high as 3,000 PSI or above. It used to be that storage units operate only part of the year, with a service factor of 40% or below, but the demand for storage injection and withdrawal is less seasonal now and more market driven. Lifetimes of 20 years and above are required.

Mixed storage and transmission is the most demanding service of all, and requires great care and attention to the integrity and long-term reliability of the installation.

Lateral line service is likely to be the most variable and may require a higher ratio than mainline transmission to achieve the required gas turbine fuel inlet pressure. The lifetime requirement is likely to be less than for mainline transmission and storage.

6.3.6 EXPERIENCE BASE: SUCCESSES AND PROBLEMS WITH SKID MOUNTED PIPELINE RECIPROCATING COMPRESSOR SYSTEMS

A series of questions was prepared and transmitted to representatives of the pipeline industry as a planned approach to gathering experience with mounting of high-speed separable compressors. Contact was made with a total of 12 major pipeline or storage companies, and between one and five representatives from each company. Written responses to be used in the experience base is eventually expected from 11 or 12 individuals from eight companies. From two or three, the response will be limited to the documentation of an extended phone conversation. In one case, the respondent convened a panel of experts who participated in an interactive conference call. This call was documented, and a final edited version is expected. Initial response has been received in relation to a new 8,000 HP block mounted installation.

The responses cover 28 different units, 18 engine driven, and 10 motor driven. The units range in size from a Cat 3608 of around 2,200 HP range to a nominal 8,535 HP motor driven unit. Twenty-seven units are skid mounted and one 8,180 HP engine driven unit (just being installed at this time) is block mounted.

6.3.7 FORCES TO BE RECOGNIZED AND MANAGED IN DESIGN AND ANALYSIS OF THE MOUNTING SYSTEM

The forces to be managed or carried by the mounting system can be distinguished broadly as weight forces, inertia forces, gas forces, drive forces, installation forces, and thermal forces. More specifically, they can be categorized as follows:

- Weight of skid
- Weight of compressor
- Weight of driver
- Weight of cylinders and crosshead guides
- Weight of discharge bottle(s)
- Weight of suction bottle(s)
- Weight of other vessels
- Local rotating shaking forces
- Local reciprocating shaking forces at rotating frequency
- Local reciprocating shaking forces at 2x rotating frequency
- Local gas forces in cylinder at 6x and below
- Local gas forces in cylinder above 6x rotating frequency
- Differential cylinder stretch forces
- Axial gas shaking forces in discharge bottles
- Lateral gas shaking forces in discharge bottles
- Axial gas shaking forces in suction bottles
- Lateral gas shaking forces in suction bottles
- Vertical gas forces
- Vertical XH forces on the vertical forces due to the rod load on the slider crank mechanism
- Torques which produce differential vertical forces
- Installation forces
- Thermal forces

These are discussed briefly below.

6.3.7.1 Skid Weight

The weight of a skid with concrete can easily exceed 200,000 lbs and may be as high as 300,000 lbs. The weight of a skid and, of course, all that sits on the skid must eventually be carried and located by the concrete foundation below the skid. This makes clear that whether or not a unit is directly mounted on the block, its weight together with any skid weight must be satisfactorily supported by the block.

6.3.7.2 Weight of Compressor and Driver

The weight of the compressor and its driver must be accommodated on the skid (and/or block) with some attention paid to the location of the CG of the components and to the distribution of the weight about the CG.

6.3.7.3 Weight of Cylinders and Crosshead Guides

The cylinders and crosshead guides are cantilevered from the compressor frame and must be provided additional support at points of significant weight load. Some manufacturers will provide extra A-frame supports for the crosshead guides, and these supports must themselves be appropriately mounted to skid or block.

Some cylinders are supported on the nozzle attached to the discharge bottles, and these depend for weight support on the method used to support the discharge bottle. This practice, which adds stress on the discharge nozzles and their reinforcement, raises significant question for heavier cylinders and high energy density equipment. Independent support at the end of the cylinder specifically designed for the purpose of carrying weight and restraining vertical and lateral vibration is desirable, and many larger cylinders are provided with bolt holes at their head for a support to be attached. Independent cylinder end support provides more effective management of vertical system vibrations, which can cause damage to suction bottle nozzles. The challenge is that the discharge bottle’s outer diameter typically extends to the end of the cylinder or beyond. Thus, the discharge bottle and piping connected to the discharge bottle may interfere with the natural location for a cylinder end support.

Any cylinder end support must be designed to accommodate expected cylinder stretch, so it cannot be excessively stiff in response to forces acting along the cylinder axis or it will itself be overstressed.

6.3.7.4 Discharge Bottle Weight

The geometry of larger discharge bottles typical of gas transmission leads to them being most commonly (though not universally) supported below skid level on the foundation block and normally with wedges and/or clamps. Use of an extra chamber on the discharge bottles is quite common (i.e., chambers for three cylinders on one side), and this extra length and weight must be accommodated in the discharge bottle supports

6.3.7.5 Suction Bottle Weight

The suction bottles are sometimes free standing, and in principle, their weight can be supported by the suction nozzles, and most commonly is, but the concern exists that with large bottles typical of transmission, the suction bottle lateral resonance may be excited if they are not more completely constrained. Some suction bottles employ an extra chamber—four chambers for three cylinders on one side, and in this case, the added mass and rotary inertia makes it desirable to support the suction bottles more fully for dynamic considerations as well as for weight support. An A-frame structure or vertical vessels closely attached to the suction bottle may provide the needed support.

6.3.7.6 Other Vessels

Other vessels, such as scrubbers, whether used to support suction bottles or free standing, require weight support and may need bracing or well-reinforced base support to manage dynamic loads and natural frequencies.

6.3.7.7 Rotating Unbalanced Forces

The compressor and engine's crankshafts can generate rotating unbalanced shaking forces at 1x rotational speed. However, most commonly the rotating forces are carefully balanced to tight limits by the compressor OEM, which eliminates the need to carry and manage unbalanced dynamic loads attributable to crankshaft unbalance alone.

6.3.7.8 Reciprocating Unbalanced Forces

The shaking forces from reciprocating motion of the connecting rod, crosshead, piston rod, and piston are not so readily balanced. The effective frequency of these forces is predominantly at 1x and 2x rotating speed. The approach used in most medium and high speed separable compressor designs is to use the opposed motion of adjacent throws to achieve as much balance as possible between the reciprocating unbalanced forces of these throws. Looked at as a system, an adjacent pair of throws, oriented at 180 degrees to each other on the crankshaft with cylinders on opposite sides of the crankshaft, combine to produce a rotating force and rotating couple at 1x and at 2s rotating speed, which act on the bearings supporting these throws. Equalizing all the weights on this pair of throws can make the 1x and 2x rotating forces for the pair of throws zero, but with any axial offset between the throws the rotating couples cannot be reduced to zero. In fact, the conservative way to look at this system is that the forces acting locally to each throw must be restrained locally; if the frame were very flexible, such local support would be essential. In fact, the frame has significant inherent stiffness and not all the local force must be supported locally, but the fraction of the local forces, which is carried by the frame and shared across other support points, is not readily quantified without detailed analysis (i.e., FEA). For large slow speed compressors, the rule of thumb established from GMRC research was that the individual tie-downs for the compressor frame should be able to carry at least half the maximum lateral local shaking force of the throw to which they are adjacent. The load carrying mechanism is friction induced by the normal force in the anchor bolt.

6.3.7.9 Required Engineering Data

To ensure integrity of the skid and mounting system under action of local forces generated by rotating and reciprocating motion, the compressor OEM needs to provide the necessary values for local throw by throw forces or the details of reciprocating weights for piston, piston rod, crosshead, connecting rod, and any rotating counterweights and inherent crankshaft unbalance from which they can be calculated.

6.3.7.10 Fully Balanced Compressor

One model of separable compressor can achieve true lateral balancing of rotating forces and couples by arranging throws in groups of three, which lie in a common plane, with the outer two paired on one side of the crankshaft, and the third "center" throw on the opposite side of the crankshaft. The center throw drives one piston through a normal slider crank mechanism, and the outer pair of throws drives a second piston via a yoked pair of connecting rods, which connect to a common crosshead. This configuration with its added complexity achieves both force and moment balance.

6.3.7.11 Cylinder Gas Forces

Gas forces are generated in each end of a compressor cylinder during the compression process. Considering first the head end of the cylinder, neglecting small high frequency acoustics within the cylinder chambers, the pressure is uniform and acts equally in all directions, particularly with equal forces outwards on the head of the cylinder and inwards on the piston face for the head end. For the crank end, the pressure acts with equal force inwards on the fixed inwards-facing head of the cylinder and outwards on the piston face for the crank end. At any instant, the net force outwards from the difference between forces on the outwards and inwards facing cylinder "heads" equals the net force acting inwards on the piston from the difference between forces on the two piston faces.

The different paths through which these gas forces on cylinder and piston act then become important. Both paths through steel or cast iron are elastic with distributed flexibility and each possesses distributed mass. A universal characteristic of the force transmission is cylinder dynamic stretch. Stretch is normally observable with a strong 1x component. In addition, from the complexity of the in-cylinder pressure variation over a revolution, there are higher order components of cylinder stretch. Cylinder stretch cannot and should not be restrained. The potential for cylinder stretch resonance should also not be neglected. A frequency spectrum of the stretch forces would reveal levels drop with increasing order of running speed. For slow speed compressors, the excitation level is believed to drop to an inconsequential level before the frequency reaches the first stretch resonance of the cylinder, but this may not be so for high-speed separables. In fact, on some large (8,000 HP) medium speed compressors, distinct cylinder stretch resonances at about 150 Hz have been documented; this shows that sufficient energy to excite these resonances exists to at least the 12th order of running speed.

In the past, for slow speed compressors, it has traditionally and validly been assumed that gas forces internally balance within the compressor. Certainly, as discussed above, such balancing occurs within the cylinder's trapped volume, but the different force transmission paths (one path through the stationary frame, and one path through the piston rod, bearings, and throws), the distributed flexibility and inertia of each path, and the potentially high excitation energy acting at higher frequencies requires this assumption to be revisited. An internal study by SwRI showed that the magnitude of vibration (above 0.5 IPS) on a JGV compressor frame at a frequency of about 100 Hz when unsymmetrical load steps were used could be explained only if the assumption of internal balancing became invalid at these frequencies. The mode of vibration excited by the inferred forces is an unsymmetrical first mode of the compressor system (frame, cylinders, bottles) rotating about a vertical central axis. A proven approach to avoiding this problem is to avoid unsymmetrical load steps (i.e., always load and unload the end cylinders on each side of a 6- or 4-throw equally). If this is not possible, then the support system should be analyzed for its response to gas forces at the frequency of this mode of vibration as if the gas forces were not internally balanced (i.e., calculate the gas force frequency spectrum and apply it as acting on the bearings without an offsetting force on the cylinders.

6.3.7.12 Differential Stretch Forces

The cylinder stretch differs in phase from cylinder to cylinder. As a result, bottles connected to three or more cylinders of single stage unit can be exposed to differential motion along its length, which will induce stress in the bottle and nozzles. If the nozzles are too short, this differential motion can overstress welded joints and nozzles, and even if tolerable in themselves, can add to the stresses from pulsation induced vibration at other frequencies.

6.3.7.13 Dynamic Gas Forces in Suction and Discharge Bottles

The intermittent gas flow produces the potential for pulsations in the compressor manifold and attached piping system, which is partially mitigated by the filtering, by center feeding of bottles, and by orifices. A primary goal of filtering is to minimize the level of pulsations above the filter cut-off frequency, which pass to the laterals and headers of the compressor station. Orifices dampen peak responses at resonances, which cannot be fully filtered; center feeding minimizes the potential for excitation of acoustic modes.

In general, it should be assumed that exciting forces would exist and act in the axial, lateral, and vertical directions for suction and discharge bottles. These forces should be quantified from the pulsation study and the suction and discharge bottle supports should be designed and evaluated for their ability to carry these loads with minimal or tolerable deflection. Discharge bottles are easier to support than suction bottles and should be provided with wedge and clamp supports, which will restrain them effectively in all three directions. It should further be recognized that, without cylinder supports or a suction bottle support structure, the discharge bottle support system might be the primary restraint against vertical motion of the cylinders, of the suction nozzles, and of the suction bottles. As the experience base shows, the vertical system vibrations and their influence on stresses at the joint between suction nozzle and suction bottle can be significant and have led to failures of welds at this joint. Thus, recognition of the potential for forces to act vertically on the system at various locations, including the suction bottle, is important. The ability to restrain suction bottle motion and to control nozzle joint stresses through the combined influence of discharge bottle supports, cylinder supports, and suction bottle support structure needs to be evaluated.

6.3.7.14 Bracing for Suction Bottle Lateral Forces

As discussed, there is unavoidable residual shaking and gas forces at 2x and above, which have directional components in the lateral direction relative to the suction bottles. To preempt the damaging influence of these lateral forces, it is often beneficial for single stage compressors, with individual bottles on either side, to join the two bottles with one or more braces straddling the compressor, particularly with heavy extra chamber bottles. If planned from the outse1t, such bracing can be integrated effectively and aesthetically into the design. The bracing effectively avoids the excitation of symmetrical modes. Since such bottles often have their own piping and support systems arranged symmetrically side to side, it can also be beneficial to brace across the piping and bottle support systems as well.

6.3.7.15 Differential Vertical Forces

Differential vertical forces exist between the two sides of a compressor and a motor because the drive torque and load torque must be reacted by the support system. In addition, any dynamic variation in the drive torque shows up as dynamic variation in the vertical forces on the sides of the support system. The system must be able to tolerate these differential forces.

6.3.7.16 Installation Forces

Known and predictable installation forces occur as the skid with mounted components is hoisted into place during installation. These forces act to bend the skid and can cause excessive loads on already mounted or coupled components if the bending is excessive. Rigidity of the skid must be shown by analysis to be sufficient to avoid such damage.

Installation forces may also occur if joints cannot be made up with the pipes, bottles, and nozzles in an unstrained condition. "Come alongs," or a backhoe may be used to close the gap between components to be joined leaving residual tensile, compressive, or transverse forces in the components joined.

6.3.7.17 Thermal Forces

Thermal forces result from expansion of piping at its temperature of operation. Overly close clamps on discharge piping can impose excessive stresses. A thermal piping analysis in conjunction with criteria in ANSI piping codes should be applied, and the necessary piping and support modifications made to reduce such forces and stresses to acceptable levels.

Additional thermal forces can be experienced at compressor and engine tie-downs. As the compressor heats up, it will naturally expand relative to the skid, particularly in the axial direction. It is not possible or desirable to restrain thermal growth, and so some sliding will occur at tie-down locations to accommodate the relative changes in length.

6.3.8 INITIAL GUIDELINES FOR MOUNTING MEDIUM AND HIGH SPEED SEPARABLES

6.3.8.1 Responsibility

- Engineering and program management are critical to the success of any high or medium speed separable compressor.
- While the single point of contact, which a packager provides for a skid, mounted compressor is attractive, it should not lessen the responsibility of the operating company's project manager to assure that quality engineering is performed and that full consideration is taken of those who will have to operate and maintain the installation.
- The operating company's project engineer should encourage and enable communication between operations and engineering during the design and procurement.
- The operating company's project engineer should also ensure communication between designers of skid, foundation, pulsation control, piping and bottle supports, installers and suppliers, and those analyzing and evaluating these aspects of the configuration.

6.3.8.2 Foundation Design and Installation

- The design and installation should start with a sound, well-engineered, concrete foundation block on which skid and/or compressor system will be mounted.
- Ideally, this block will be on a solid, thick mat.
- The block or mat should be mounted on sound soil material with properties established by a geotechnical study.
- If the consistency of the soil is in doubt, consider supporting the concrete block on driven piles.
- Never consider supporting a compressor skid directly on compacted soil or gravel for pipeline service.

6.3.8.3 Analytical Studies

- Perform structural Finite Element Analysis (FEA), which includes skid, frame, and attached bottles, bottle supports, cylinders, and crosshead guides.
- Model all skid structural I-Beams with plate elements.
    - Beam element representation of the I-Beams is generally inadequate for skid dynamic FEA.
- FEA should include static strength and rigidity analysis for hoisting conditions encountered during installation.
- FEA should include dynamic analysis for natural frequencies and response to excitations.
- Calculate natural frequencies of compressor frame, cylinders, bottles, and other vessels on the support system (skid or block).
- In particular, predict the frame-support system rotary mode and any system vertical modes involving cylinder and suction bottle vertical motion.
- Control the vertical mode involving cylinders and suction bottle with cylinder head end supports and suction bottle support structure.
- In addition to prediction of natural frequencies and mode shapes, the response to all known and potential excitations should be performed, with account being taken of possible uncertainty of natural frequency predictions.
- Even if FEA predicts no frame mounting system natural frequency at or near 1x or 2x rotational frequency, always predict the non-resonant structural response to 1x and 2x unbalanced shaking forces and show resulting movement and stress of the compressor frame and mounting system are acceptable or make necessary changes to the system.
- Specifically assess the adequacy of reinforcement and gusseting of the I-Beam structure in stiffening the structure.
- Also, predict cantilever modes for any vertical vessels and show these lie outside the likely excitation range for the compressor.

- If unsymmetrical load steps are unavoidable, perform a conservative response to the resulting unsymmetrical gas forces applied at and near the frame-support system rotational natural frequency and treated as an unbalanced force.
- Always perform a tie-down load carrying analysis to ensure that frictional joints (between machine base and skid or block) achieve sufficient lateral load restraint by virtue of the applied anchor bolt load normal to the frictional joint.
- Obtain throw-by-throw lateral forces (gas and inertia) from compressor OEM or a basis for calculating them.
- Design tie-downs and skid structure to carry throw-by-throw forces locally (see Structural Analysis Options section).
- Determine bottle forces to be carried by bottle support system from pulsation analysis.
- Always perform a pulsation analysis.
- Always perform a mechanical vibration study.
- Always perform a torsional study. A torsional study should always be performed unless the installation is truly identical in configuration and operation to an installation known to be free of torsional vibrations.
- The content of each of these studies should be spelled out explicitly or according to some existing standard.
- A 3-D solid modeling graphic study of the entire installation is recommended.
- The 3-D solid model graphics should be used for identification and avoidance of interferences.
- The 3-D solid mode graphics should be used to assess operability and maintainability.
- These assessments should specifically involve those who will operate and maintain the compressor system and should be used to encourage their communication.

6.3.8.4 Skid Design and Installation

- Since I-beam cross-section tends to deform under lateral loads, the I-beam should be reinforced with gussets at key load points.
- The skid should be grouted in place on the concrete block and tied down with anchor bolts.
- At installation of the skid, align its top surface at tie-down locations to ensure these surfaces are precisely level to earth within close tolerances.
- Ensure multiple holes through the concrete used to fill the skid under the compressor are provided to ensure adequate grout flow to cover the skid concrete to block concrete interface.
- Put floor at same level as top of skid (i.e., put skid below grade).
- Fill skid with concrete under compressor.

- Be prepared to install a skid, which alone weighs 200,000 to 300,000 lbs, with concrete.

6.3.8.5 Installation of Compressor and Driver

- Align the frame ON-SITE to the mounting system to minimize crankshaft stress.
- Soft foot alignment is unacceptable as an alignment method.
- The driver and compressor must be coupled together with good alignment, which satisfies all manufacturer's requirements and criteria based on reverse indicator measurement.
- Alignment is a critical issue for compressor operation, and if performed by a contractor, must be inspected and signed off by the operating company.
- Use solid shims for alignment instead of multiple shim packs.
- Provide walkways to enable convenient maintenance and performance analysis, including consideration of both suction and discharge valve replacement.
- For skid installation, provide sufficient anchor bolt stretch length (at least 2 feet) with the bolt head bearing against a solid structure.
- For block installation, maximize anchor bolt-stretch length by extending the anchor bolts to the mat.
- Use ASTM A193 Grade B7 anchor bolts (with rolled threads) to tie down the compressor and its driver to the skid or block.
- Use spherical (self-aligning) washers on anchor bolts to minimize stresses during torquing.
- Ideally, provide vertical point-by-point adjustability in the mounting system for the compressor.

6.3.8.6 Support of Cylinders and Crosshead Guides

- The crosshead guide (CHG) must be supported on the mounting system. A manufacturer's supplied CHG support can be used or such a support can be independently designed and built, but it is essential to provide CHG support.
- Devise and install an engineered support at the cylinder end in addition to the discharge nozzles.
- Make sure the cylinder head end support can accommodate maximum possible cylinder stretch without incurring excessive bending stress. This support may be required to de-tune a resonance.
- Also, ensure that tall cylinder supports are evaluated for any vibration natural frequencies of the support itself.

6.3.8.7 General

- Limit the height of the compressor-driver centerline above the concrete foundation (while providing sufficient I-Beam stiffness in the skid). Conversely, do not unnecessarily elevate the compressor system above the top of the concrete foundations; height adds flexibility and amplifies the moment arm from all transverse system forces (at cylinder level and in piping and bottles); with the high energy density of modern high-speed separables, any added flexibility is a potential source of problems.

- Avoid unsymmetrical load steps.
- In engineering for multiple conditions, particularly with multi-stage units, ensure that the process of transition between conditions is considered as well as operation at the specified conditions.
- When evaluating corrective action in the light of defective materials discovered at site, consider carefully the longer term integrity of the installation as well as near term schedule consideration.

6.3.8.8 Support of Bottles and Piping

- Mount discharge bottles to the concrete block structure; use wedges and clamps to provide bottle lengthwise restraint as well as vertical and lateral restraint.
- Brace opposite side suction bottles together across the unit.
- Ensure sufficient piers and pipe support clamps are provided to control natural frequencies associated with free spans piping.
- Ensure discharge bottle supports can carry bottle forces based on pulsation study.

6.3.8.9 Block Mounting

- Seriously consider block mounting for compressors over 5,000 HP—see subsequent section for discussion of issues to be evaluated.
- If block mounting is selected, use a “mini-skid” for the oil system.

6.3.9 OPTIONS FOR STRUCTURAL ANALYSIS TO ASSESS TIE-DOWN INTEGRITY

SwRI’s experience with slow speed compressors showed that the compressor frame bends under the unbalanced forces and that the frame/dog house/cylinder stretches under gas loads, with a direct impact on the loads to be restrained at individual anchor points. It is reasonable to assume comparable bending will occur in large medium and high-speed separables—particularly the longer six-throw units. As a simple (conservative) rule, the frame has to be tied down so that the unbalanced forces from each throw are fully restrained by the anchor bolts and chocks in the immediate vicinity of that throw. To allow a detailed analysis and assessment of tie-down integrity according to this rule, the throw-by-throw gas and inertia loads are needed from the manufacturer (or a basis for their calculation from reciprocating inertias and kinematic details).

With information on loads to be carried by the tie-downs in hand, it is still necessary to decide on a treatment of the frame. The options include:

- Assume frame to be rigid (a non-conservative assumption).
- Assume frame to be totally flexible (most conservative).

- Assume frame to be partially flexible, such that some factor (<1) times the throw forces must be carried locally.
- Prepare a detailed finite element model of the frame and foundation and predict its response to application of inertia and gas forces appropriately applied at their point of action.

Based on a number of finite element analyses for slow speed compressors, it was found to be conservative (but not excessively so) to use a factor of 0.5 for the partially flexible frame method; this avoids the need for FEA of frame and mount for each installation.

These days, FEA has become more routine and is often used to assess other aspects of skid integrity, so this choice may be increasingly attractive and has the advantage of accuracy. However, it must be recognized that to represent compressor frame flexibility accurately, the compressor frame must be modeled using frame structural details from the manufacturer. Thus, in addition to the inertia and gas loads, or a basis for their calculation, the manufacturer must be asked for and must provide frame drawings in sufficient detail for FEA of frame lateral flexibility under throw-by-throw loads.

6.3.10 Issues to Consider When Evaluating a Decision to Skid Mount or Block Mount a Unit

As discussed in other sections, advocates exist for block mounting medium or high-speed separable compressors at some size level. Other advocates exist for skid mounting all units. Thus, the issue is far from clear and will probably remain one of choice. It is suggested that the choice for a particular planned project should be based on consensus building discussion and evaluation involving both those who must engineer and procure the compressor system and those who will end up operating it, maintaining it, and living with the consequences of project decisions. This report section lists some advantages and disadvantages associated with both block and skid mounting for new installations of modern compressors, medium or high speed; no doubt project and installation specific considerations will exist in particular situations, but this list provides a starting point:

- Some Skid Mount Advantages:
  - Unifies and streamlines compressor/driver installation
  - Takes advantage of established, efficient, competitive packager business
  - Minimizes lead time
  - Likely minimizes first cost
  - Provides single point of contact throughout process of procurement, installation and startup
  - The skid is likely to spread the footprint and base for carrying dynamic forces; this may help to reduce tensile stresses in the concrete
- Some Skid Mount Disadvantages:
  - Transportation limits exist for large single packages
  - Skid may be less rigid than block for force management
  - The extra height of the crank forces line of action caused by the skid will aggravate response to these forces.

- Skid mounting requires much unproductive travel for major components if all are shipped from their manufacturer to the packager and then to site.
- If all or most components are shipped direct to site, the skid may be seen as providing an unnecessary extra interface with height disadvantages mentioned above.

❑ Some block mount advantages:

- The block may be more rigid than a skid (if properly engineered)
- Provides a solid reference for alignment
- Can be engineered to carry all vessels and ancillary equipment

❑ Some block mount disadvantages:

- Disadvantages of block mount to be added in the future.

6.3.11 KNOWN LIMITATIONS AND FUTURE PLANS

The preceding report section represents an initial and incomplete draft of the eventual guidelines anticipated to results from the Phase II project on the subject. The work is in its early stages, but as a result of information gathered from a number of operating companies and available knowledge from analysis of both low speed integral compressors and medium and high-speed separable compressors, the report on the subset of the ARCT project on this subject is presented as an initial set of guidelines. The guidelines as presented have known limitations; many of the statements, both qualitative and quantitative will require further critical review and refinement.

The known limitations and needs for further work include:

❑ Further verbal discussion is needed with those who provided installation photographs to identify additional aspects of the images, which help emphasize or refine the guideline.

❑ Criteria for vibration and alignment need to be defined in discussion with OEMs.

❑ Where possible, independent analysis is needed to quantify criteria for alignment.

❑ Independent analysis is needed to better quantify and compare the stiffness of block and skid mounting.

❑ Independent analysis is needed to assess the concrete stresses involved in tie-down of a medium or high-speed frame directly to concrete.

❑ A number of further responses to the questions sent are expected but remain to be obtained from pipeline company representatives.

Debriefings are needed to infer relevant experience from SwRI field engineers involved with field problem identification and with the development of solutions. These engineers have seen some of the most serious problems, which could have been avoided with better upfront practices and analyses. This experience base will provide a basis for addition to and refinement of the guidelines.

Debriefings are needed with SwRI experts in skid analysis to support definition of quality assurance, methodology, and content requirements for structural FEA.

Further input to these guidelines is needed from skid manufacturers, compressor OEMs, Engine OEMs, motor OEMs, and engineering companies involved with the installation of medium and high speed separable compressors.

In particular, the issue of engineering data and the most appropriate format for this data needs discussion with compressor and engine OEMs, such as load information referred to in the initial guidelines section, and options for defining frame flexibility.

The entire draft document needs critical review from and discussion with the advisory group for the project to provide direction, which will increase the practicality and usability of the guidelines in their final form.

The debate over block versus skid mounting needs to be sharpened and opposing views on the subject expressed in moderated face to face discussion.

The practical constraints and issues related to head end cylinder supports need to be defined and refined, with reference to quantitative data from installations, which demonstrably and critically need such supports as illustrative examples. At least one such example has been identified in the information gathering process.

Additional illustrative examples and definitions of specific problems with quantitative data are needed, including:

- First System Rotary Mode
- Vertical Modes involving Cylinders and Suction Nozzles
- Differential Cylinder Stretch
- Cylinder Stretch Vibration Mode

Further input is needed from those who have already provided input to the experience base to shape their observation, comments, and experience in problem solution into practical and applicable guidelines.

This includes refinement of observations on alignment problems.

A better definition of the transportation limits is needed, i.e., at what size and weight levels is it required to remove cylinders and/or major components for transportation.

Further discussion is needed of the role of 3-D solid graphics in ensuring an installation and mounting configuration, which is satisfactory to operators as well as project engineers.

Acceptable and effective methods of adjusting alignment at tie-down locations need further discussion and options need to be presented in conjunction with the guidelines.

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7. EVALUATION OF TECHNOLOGIES

7.1 TECHNOLOGY MATURITY ASSESSMENT

Of the top concepts identified during this program, thirteen were assessed and advanced during Phase I of the ARCT program. Each of these technologies was assigned an initial maturity and then a current maturity as a result of development efforts during the course of the program. Table 7-1 presents the list of technologies investigated with the initial and final TRL for each (TRL levels were discussed in Section 1). The only technology not recommended for future development is the linear motor variable stroke compressor. This recommendation is based on the low maturity level, the estimated benefit, the large development cost, and low industry interest. Each of these technologies has a range of potential technology transfer paths. Some are potential products, some are potential services, and some are in terms of know-how of an expanded application of existing commercial products. The technology transfer path envisioned for each of these technologies is presented in Table 7-2.

Table 7-1. ARCT Technology Readiness Level

TECHNOLOGY	INITIAL	PHASE I
Semi-Active E-M Valve	2	5
Passive Rotary Valve	2	4
Semi-Active or Active Rotary Valve	1	3
Passive Valve Operational Trade-off Software	3	6
Tapered Nozzle	2	4
Infinite Length Nozzle	2	4
Tunable Side Branch Absorber	2	3
Large Clearance Volume Capacity Control	3	5
Target Flow Meter	3	5
Ultrasonic Flow Meter	3	5
Continuous Torque Sensor	1	3
Optimization Software Tool	3	4
Linear Motor Variable Stroke Compressor	1	2

Table 7-2. ARCT Technology Transfer Path

TECHNOLOGY	PATH
Semi-Active E-M Valve	Product
Passive Rotary Valve	Product
Semi-Active or Active Rotary Valve	Product
Passive Valve Operational Trade-off Software	Service
Tapered Nozzle	Service
Infinite Length Nozzle	Service/Prod
Tunable Side Branch Absorber	Service/Prod
Large Clearance Volume Capacity Control	Know-How/Prod
Target Flow Meter	Know-How/Prod
Ultrasonic Flow Meter	Know-How/Prod
Continuous Torque Sensor	Service/prod
Optimization Software Tool	Service
Linear Motor Variable Stroke Compressor	Product

7.2 ADVANCED RECIPROCATING COMPRESSION TECHNOLOGY ROADMAP

The creation of the next generation of compression encompasses the five major subsystems investigated during Phase I of the ARCT program. Figure 7-1 presents the overall roadmap for ARCT illustrating the synergies for these subsystems during the advancement through the different technology maturity levels.

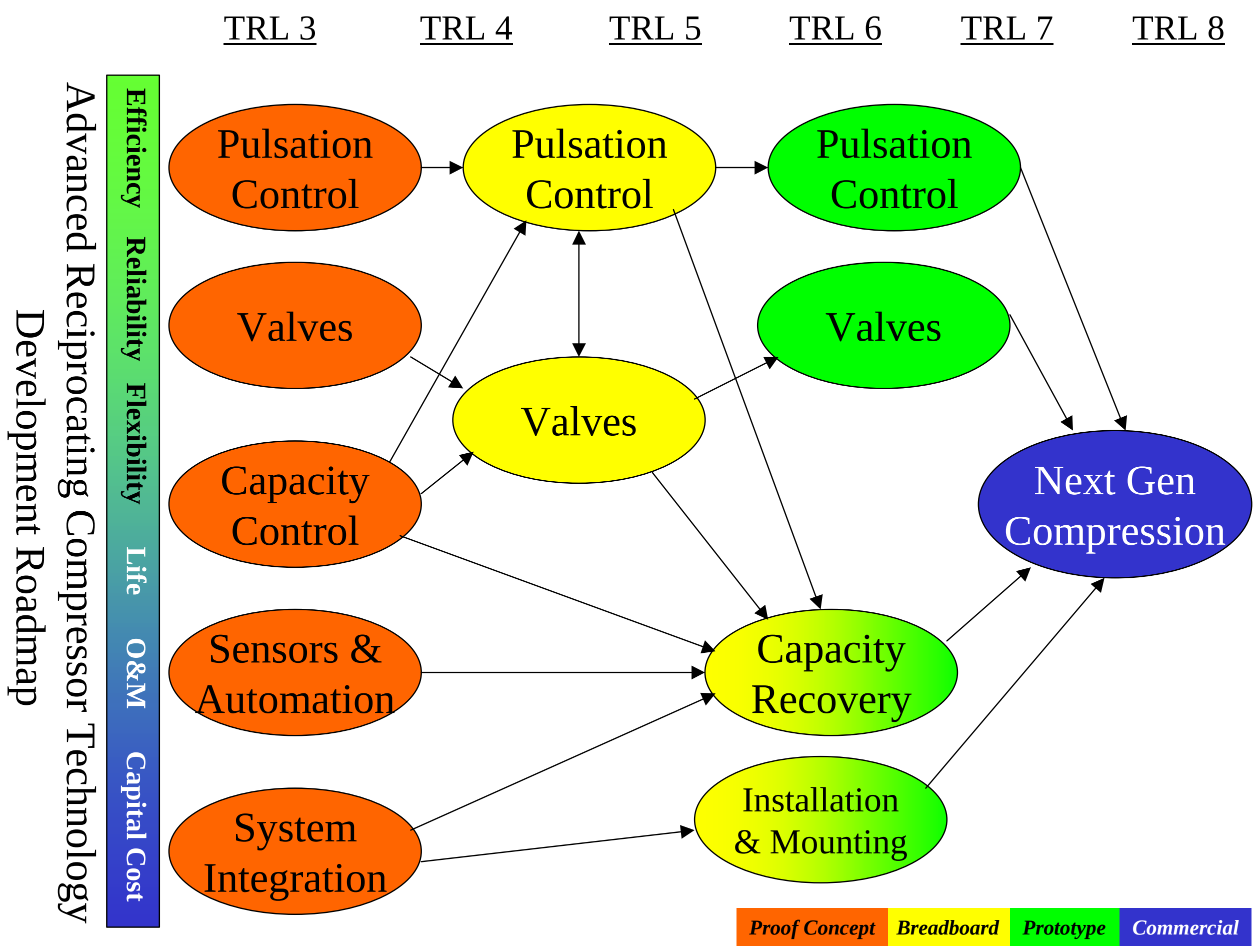

Figure 7-1. Advanced Reciprocating Compression Technology Development Roadmap

The roadmap for five pulsation control technologies that address either the critical cylinder nozzle pulsation problem or the lateral piping pulsation problem is shown in Figure 7-2. Proof-of-concept has been assessed for three of these concepts and two remaining concepts are recommended for proof-of-concept during subsequent years. Both the tapered nozzle and the infinite length nozzle address the cylinder nozzle problem so critical for modern high-speed compression. While the remaining three concepts address the lateral piping needs for more adaptable filter systems associated with large turndown operating strategies.

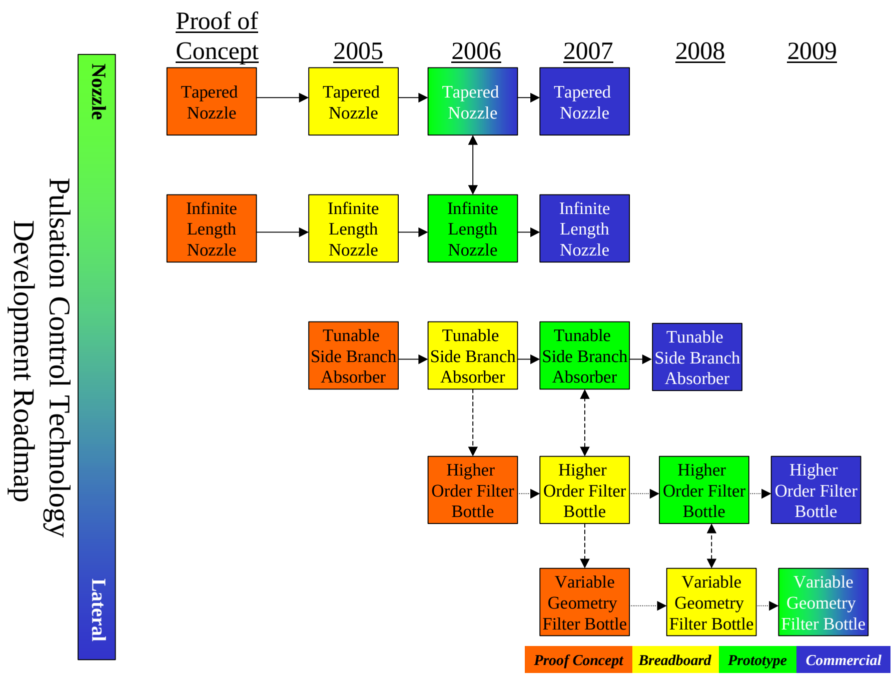

Figure 7-2. Pulsation Control Technology Development Roadmap

The roadmap for five valve technologies that address improved efficiency and/or improved life is shown in Figure 7-3.

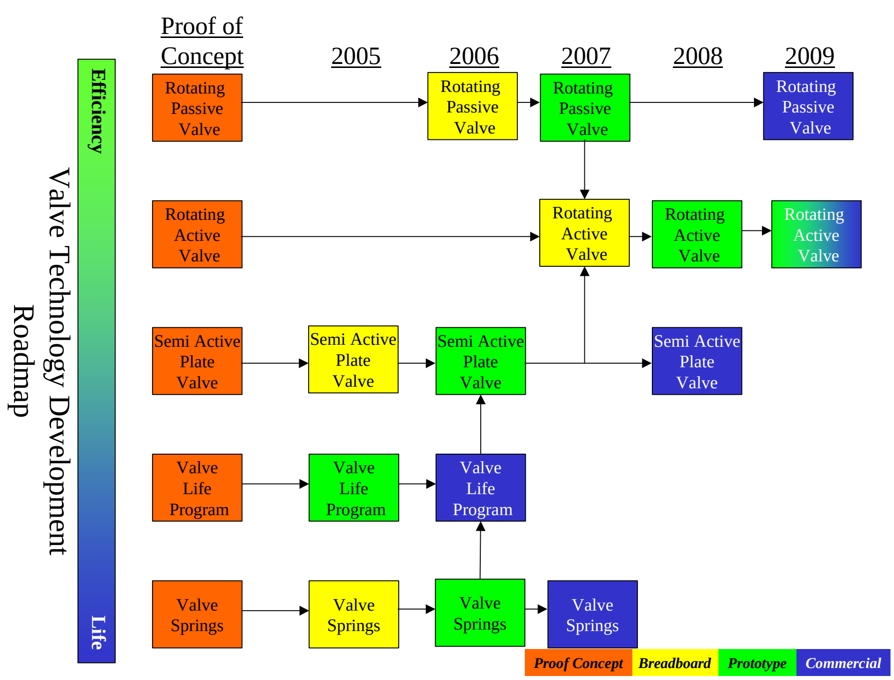

Figure 7-3. Valve Technology Development Roadmap

The various valve concepts address efficiency or life. The rotating passive valve and rotating active valve address the efficiency of operation by greatly reducing the pressure drop through the compressor valve. The semi-active plate valve allows for high-lift and low-impact forces, thereby addressing both efficiency and significantly longer life valves. The valve life software tool addresses the trade-off between efficiency and life of conventional valves by judicious solution of valve lift and valve springs. Lastly, improved valve spring technology know-how should lead to increased life of conventional valves.

The roadmap for capacity technology combines the three remaining subsystems into a focus on capacity recovery for new next generation compression and the fleet of current infrastructure compression. This roadmap is shown in Figure 7-4. The integration of new larger clearance volumes for capacity sensors and system optimization software will lead to optimized assets that recover capacity for a given engine/motor horsepower and emissions limits.

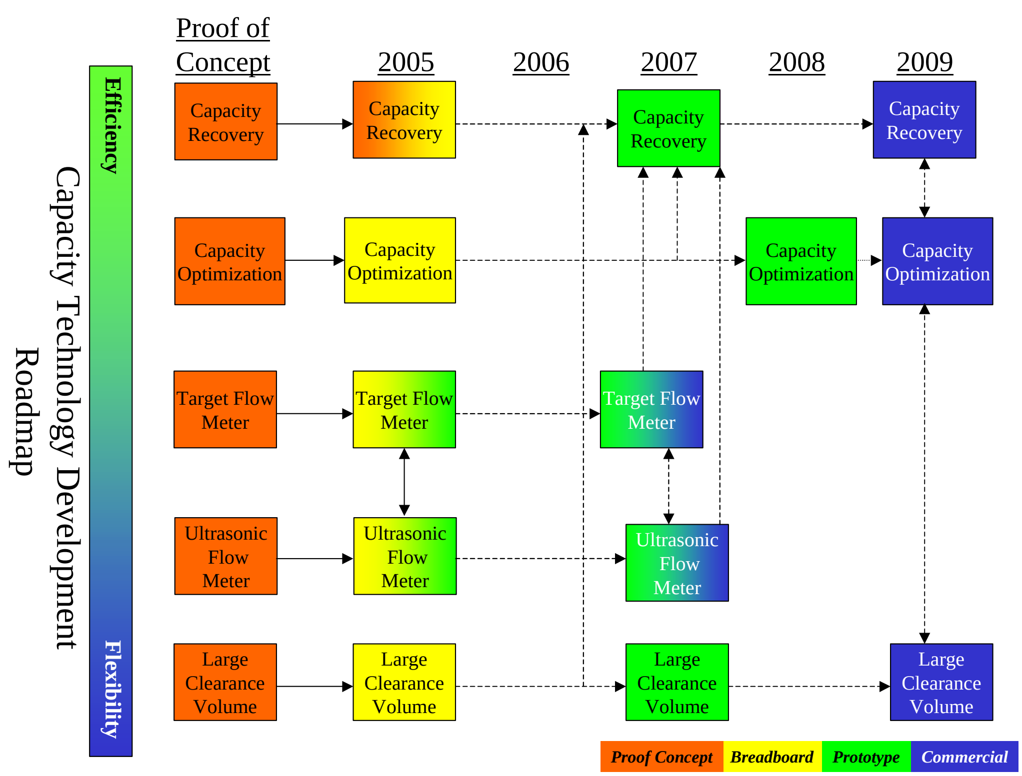

Figure 7-4. Capacity Technology Development Roadmap

The pace of development presented in these roadmaps is affected by the available resources that can be applied. Decisions on priority and development risk should be re-assessed annually, and the resultant roadmap updated. The justification for investing in these technologies is based on the value proposition of the solutions as presented in the next section.

7.3 TECHNOLOGY VALUE PROPOSITION

The estimation of benefits for implementing the technological solutions developed during this program is company and application specific. To illustrate the general approach and magnitude, a few examples are presented on the following pages.

One of the largest O&M costs for a reciprocating compressor is replacement of compressor valves. However, there is an important trade-off in life and performance. To extend valve life, the station owner can elect to reduce lift and the resulting impact stress. This will result in additional pressure drop and incur an efficiency penalty. Another alternative is the installation of the new technology developed during this program. The semi-active plate valve has the potential to increase valve life by one to two orders of magnitude without a pressure drop penalty. This new technology will require somewhat more capital cost per change-out. The critical parameter is the ratio of improved life factor and increased valve change-out cost. If a valve can last twice as long but cost twice as much, then this ratio is equal to one and there is no maintenance cost benefit, unless the cost of downtime, in terms of unavailability, is considered. As this ratio increases, the benefit of long-life valve technology can be realized. An example is shown in Figure 7-5 with an assumed base case of a half-year valve life and an estimated $30,000 per valve change-out for a single compressor unit. A realistic expectation for this new valve technology is a ratio of six, which will result in a $50,000 maintenance savings per year per compressor.

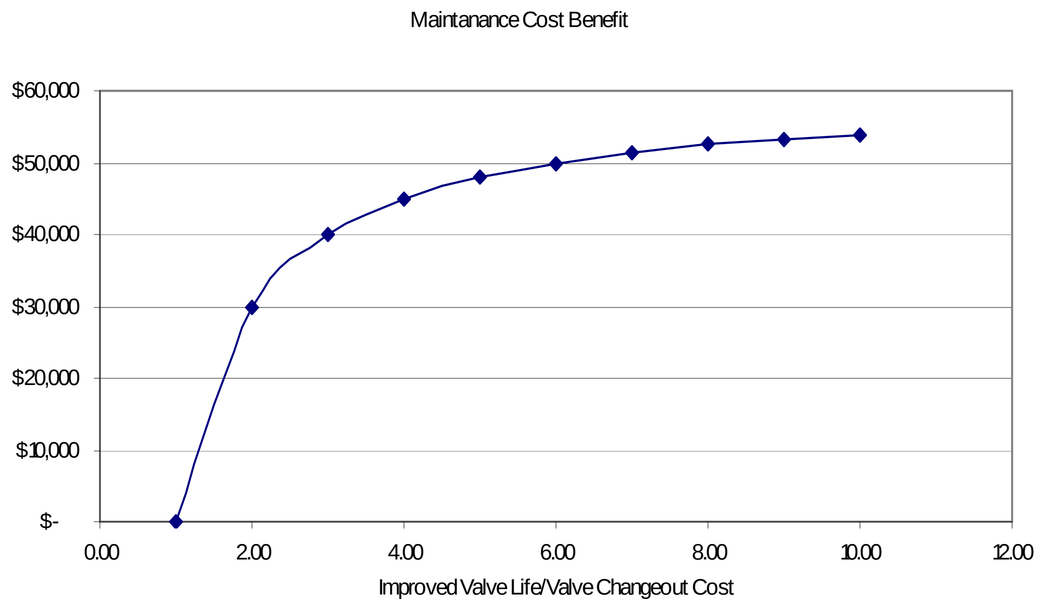

Figure 7-5. Valve Life Extension Benefit

A number of technologies developed during ARCT improve system efficiency by reducing parasitic pressure drop. The rotating valve, tapered nozzle, and infinite length nozzle are a few examples. A survey performed by GMRC showed that for the current slow-speed integral infrastructure there was a large distribution in compressor efficiency. In addition to cylinder designs, three additional causes of reduced efficiency are valves, the pulsation control system, and some methods of capacity control. The mean efficiency was 80% with a peak efficiency of 92%. Recent experience with high-speed units is an even lower mean efficiency. Improved valve, pulsation control, or capacity control technology could recover a significant portion of this efficiency penalty. Figure 7-6 shows an example of annual fuel savings as a function of efficiency improvement for a range of fuel costs. At the current cost of gas at $7 per thousand cubic feet, a 6% improvement would result in a cost savings of $40,000 per year per compressor.

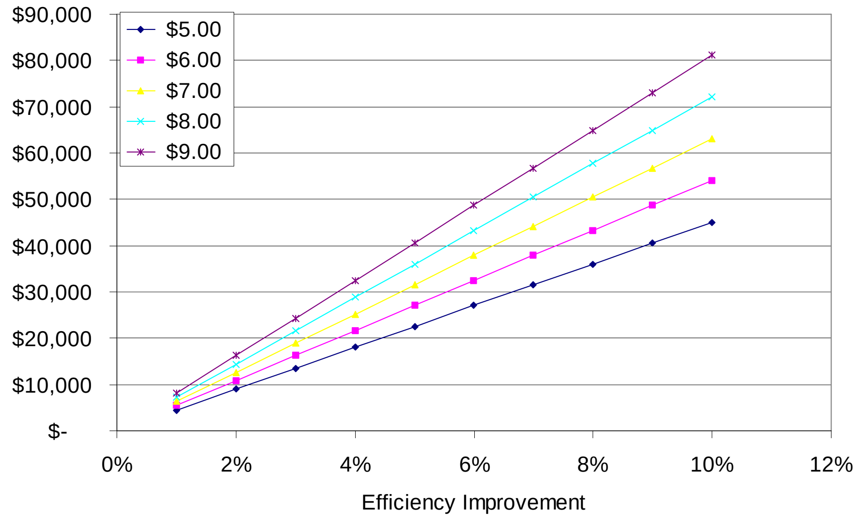

Figure 7-6. Annual Fuel Savings as a Function of Gas Cost

A second approach to an improvement in efficiency is to increase throughput and avoid the need to install new compression. The estimated capital cost reduction can vary over a large range based on the specific circumstances. A reasonable estimate is to assume a $1,000 per horsepower and calculate a simple capital cost savings. Figure 7-7 shows that the same 6% efficiency improvement would recover $120,000 in capital cost.

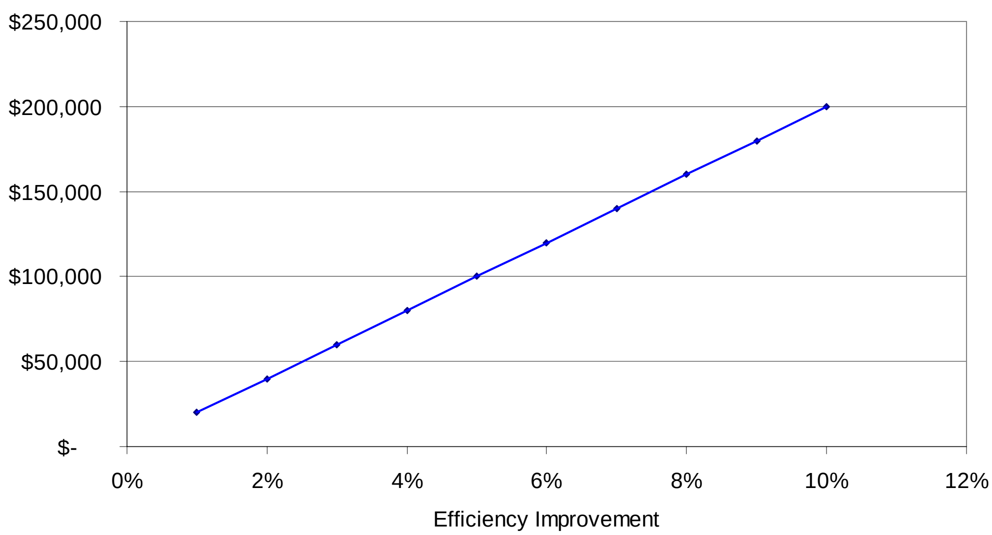

Figure 7-7. Capacity Recovery Benefit

In summary, a conservative estimate for the value of ARCT technologies is on the order of $50,000 per installation per year for reduced O&M or about $100,000 per installation in a one-time reduction in capital cost.

8. CONCLUSION AND RECOMMENDATION FOR FUTURE WORK

During the course of this one-year project, a number of critical needs were identified and eighteen technology solutions were initiated. These technologies have been matured to a proof-of-concept stage. The GMRC PSC has recommended advancing half of these technologies to the next stage.

It appears from the work accomplished to date that the program would develop sufficient technology solutions to address the current limitations of modern high-speed compression, thus enabling this equipment to meet its full potential. If this does indeed occur, the ARCT program will meet its stated objective of creating the next generation of reciprocating compressor technology that provides added pipeline flexibility at reduced capital cost.

The majority of the currently installed fleet of pipeline reciprocating machines are slow-speed integral compressors. Almost all new reciprocating machines are large-horsepower, high-speed compressors. It is, therefore, imperative that any meaningful solutions must address both class of machines.

The major challenges with the fleet of slow-speed integral machines are: limited flexibility and a large range in performance. In an attempt to increase flexibility, many operators are choosing to single-act cylinders, which are causing reduced reliability and integrity due to 1x pulsation problems in the laterals. While the best performing units in the fleet exhibit thermal efficiencies between 90% and 92%, the low performers are running down to 50% with the mean at about 80%. The major cause for this large disparity is due to installation losses in the pulsation control system. In the better performers, the losses are about evenly split between installation losses and valve losses.

The major challenges for large-horsepower, high-speed machines are: cylinder nozzle pulsations, mechanical vibrations due to cylinder stretch, short valve life, and low thermal performance. To shift nozzle pulsation to higher orders, nozzles are shortened, and to dampen the amplitudes, orifices are added. The shortened nozzles result in mechanical couples with the cylinder, thereby, causing increased mechanical vibration due to the cylinder stretch mode. Valve life is even shorter than for slow speeds and is on the order of a few months. The thermal efficiency is 10% to 15% lower than slow-speed with the best performance in the 75% to 80% range.

The objective of this program is to create the next generation of reciprocating compressor technology for both classes of machinery. The goals for the next generation of compression are:

- Improved flexibility (50% turndown)
- Improved efficiency (90%)
- Improved reliability (order of magnitude increase in valve life with half of the pressure drop)
- Improved integrity (vibration less than 0.75 IPS)

The ARCT program has proven the concept of a number of enabling technologies that have the potential to meet these ambitious goals.

Retrofit technologies that address the challenges of slow-speed integral compression are:

- Optimum turndown using a combination of speed and clearance with single-acting operation as a last resort.
- If single acting is required, implement infinite length nozzles to address nozzle pulsation and tunable side branch absorbers for 1x lateral pulsations.
- Advanced valves, either the semi-active plate valve or the passive rotary valve, to extend valve life to three years with half the pressure drop.

This next generation of slow-speed compression should attain 95% efficiency, a three-year valve life, and expanded turndown.

New installation technologies that address the challenges of large-horsepower, high-speed compression are:

- Optimum turndown with unit speed.
- Tapered nozzles to effectively reduce nozzle pulsation with half the pressure drop and minimization of mechanical cylinder stretch induced vibrations.
- Tunable side branch absorber or high-order filter bottle to address lateral piping pulsations over the entire extended speed range with minimal pressure drop.
- Semi-active plate valves or passive rotary valves to extend valve life to two years with half the pressure drop.

This next generation of large-horsepower, high-speed compression should attain 90% efficiency, a two-year valve life, 50% turndown, and less than 0.75 IPS vibration.

The potential to attain these next generation goals are within the industry's reach if the identified technology concepts are fully matured to commercially available products and service during the next stage of development.

9. LIST OF ACRONYMS AND ABBREVIATIONS

AAVCP	Automatic Variable Volume Clearance Pocket
ARCT	Advanced Reciprocating Compression Technology
ASME	American Society of Mechanical Engineers
BDC	Bottom-Dead-Center
CE	Crank End
dB	Decibel
DOE	Department of Energy
FEA	Finite Element Analysis
GE	General Electric
GMRC	Gas Machinery Research Council
GSP	Gas-Controlled Stepless Pocket
HE	Head End
HP	Horsepower
hs/hr	high speed / high ratio
HVVCP	Hydraulic Variable Volume Clearance Pocket
Hz	Hertz
IHP	Indicated Horsepower
ILN	Infinite Length Nozzle
IPPS	Interactive Pulsation Performance Simulation
ISC	Infinite Step Controller
lbf	Pound Force
MMSCFD	Million of Standard Cubic Feet Per Day
MNF	Mechanical Natural Frequencies
O&M	Operating and Maintenance
PSC	Project Supervisory Committee
PSI	Pounds per Square Inch
PSIA	Pounds per Square Inch Absolute
PV	Pressure Volume
RCTF	Reciprocating Compression Test Facility
RPM	Revolutions per Minute
SBA	Side Branch Absorber
SREI	Square Root Error Indicator
SwRI	Southwest Research Institute®
TCN	Tapered Cylinder Nozzle
TDC	Top-Dead-Center
TRL	Technology Readiness Level
VFD	Variable Frequency Drive

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