TECHNOLOGIES TO ENHANCE OPERATION OF THE EXISTING NATURAL GAS COMPRESSION INFRASTRUCTURE

Quarterly Technical Progress Report

Reporting Period Start Date: 10/01/04

Reporting Period End Date: 12/31/04

Prepared by

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**DOE Award No. DE-FC26-02NT41646
SwRI® Project No. 18.06223**

Prepared for

**U.S. Department of Energy
National Energy Technology Laboratory
3610 Collins Ferry Road
P.O. Box 880
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January 2005

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ABSTRACT

This quarterly report documents work performed under Tasks 10 through 14 of the project entitled: *Technologies to Enhance Operation of the Existing Natural Gas Compression Infrastructure*. The project objective is to develop and substantiate methods for operating integral engine/compressors in gas pipeline service, which reduce fuel consumption, increase capacity, and enhance mechanical integrity. The report first documents tests performed on a KVG103 engine/compressor installed at Duke's Thomaston Compressor Station. This is the first series of tests performed on a four-stroke engine under this program. Additionally, this report presents results, which complete a comparison of performance before and after modification to install High Pressure Fuel Injection and a Turbocharger on a GMW10 at Williams Station 60. Quarterly Reports 7 and 8 already presented detailed data from tests before and after this modification, but the final quantitative comparison required some further analysis, which is presented in Section 5 of this report. The report further presents results of detailed geometrical measurements and flow bench testing performed on the cylinders and manifolds of the Laboratory Cooper GMVH6 engine being employed for two-stroke engine air balance investigations. These measurements are required to enhance the detailed accuracy in modeling the dynamic interaction of air manifold, exhaust manifold, and in-cylinder fuel-air balance.

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1. INTRODUCTION

This report presents results from a series of tests performed on a KVG103 integral engine compressor in natural gas transmission service, installed at Duke's Thomaston compressor station in South Texas. This is the first series of tests performed on a four-stroke engine under this program. The engine in question (Station Unit 7, Duke Identification Number 40207) was installed in 1955. The report also completes some comparisons on a GMW10, Unit 6 at Williams' Station 60 (St. Francisville station), which was previously tested in February and August 2004 to provide data before and after modification (including addition of high-pressure fuel and a turbocharger). Previous quarterly reports (Nos. 7 and 8) provided detailed data from these tests, but the final quantitative comparison required some additional analysis, which the report presents. The report further presents results of detailed geometrical measurements and flow bench testing performed on the cylinders and manifolds of the Laboratory Cooper GMVH6 engine being employed for two-stroke engine air balance investigations. These measurements are required to enhance the detailed accuracy in modeling the dynamic interaction of air manifold, exhaust manifold, and in-cylinder fuel-air balance.

1.1 THE U.S. GAS TRANSMISSION COMPRESSION INFRASTRUCTURE

The gas transmission industry operates over 4,000 integral engine compressors, which play a major role in pumping natural gas through the U.S. pipeline system. Although the use of centrifugal compressors in the U.S. pipeline industry has grown, these integral reciprocating units still represent over 70% of the fleet in numbers and over half the installed power. These "slow-speed" integral engine compressors have been the workhorses of the industry for over 50 years, providing the reliable gas compression needed by the pipeline system. Figure 1-1 shows two such units: a 48-year old TLA6 and a 50-year old GMW10.

Figure 1-1. TLA6 (2,000 HP) and GMW10 (2,500 HP) in Pipeline Service

Figure 1-2 shows the age distribution of the current infrastructure. Over half the fleet is well over 40 years old, but replacing all these units with currently available technology would incur a huge cost and disruption to service with insufficient improvement in overall performance of the pipeline system to justify this cost and disruption. For these reasons, wholesale replacement remains unlikely (although selective replacement driven by factors, such as environmental regulations can be expected). Growth to a 30-TCF-plus gas market in the U.S., anticipated over the next 10 to 20 years, must come on the backs of the existing compression infrastructure; it will, therefore, depend on continued integrity, enhanced capacity, and efficiency of the existing integral engine/compressors under all loads. The industry needs demonstrated technology options and operating methods, which will cost-effectively maximize the capacity of these old units, and reduce their fuel consumption, while respecting or improving their integrity.

Figure 1-2. Install Dates: Over 50% of Pipeline Compressors Exceed 40 Years Old

Figures 1-3, 1-4, and 1-5 exemplify these needs of the existing infrastructure.

Figure 1-3 shows how annual fuel consumption at individual compressor stations in the pipeline system varies with the number of horsepower hours delivered by the engine to the compressor cylinders at that station. Points on the high side of the mean slope represent stations, which are burning more than the industry average. In addition, with a regressed slope of 7.7 CF/BHP-hours for Figure 1-3, the industry burns significantly more fuel than the most efficient current technology natural gas engines (as little as 6 CF/BHP-hr.). As a slightly different performance measure for the industry, Smalley, et al. [1], calculate an industry average (ratio of total fuel volume to total BHP-hours) of 8.25 SCF/BHP-hours.

Figure 1-3. Industry Fuel Consumption
 (~7.7 MCF/HP-Hr. $\pm 20\%$ - Need to Lower the High Values)

Figure 1-4 presents a distribution of compressor thermal efficiency for the industry created by the Gas Machinery Research Council (GMRC) from a quantitative survey a number of years ago. This is the efficiency with which the compressors convert HP-hours to useful compression. The width of the range and the 12 points by which the 79% median lies below the best achieved (91 to 92%) represents not only gas which is burnt rather than delivered, but also engine capacity, which must overcome losses rather than deliver useful compression of the transported gas.

Figure 1-4. Compressor Thermal Efficiency Histogram Based on GMRC Survey

Figure 1-5 shows a number of failed crankshafts. This problem continues to occur at an undesirable rate for the pipeline industry as a whole (about one failure per thousand engines per year). This rate may not seem excessive, but for the compressor station and company, which incurs such a failure, the disruption, cost, and loss of capacity at the time is significant. The chance of this rate increasing as a penalty for improved performance and increased capacity must be avoided, as well as any increase in problems such as bearing failure, or damage caused by detonation, or unintended overload.

Figure 1-5. Integrity: Crankshaft Failure Examples – Need Methods of Avoidance

1.2 THE COMPRESSION INFRASTRUCTURE PROJECT

Three years ago, the U.S. Department of Energy (DOE) initiated a Natural Gas Infrastructure (NGI) program, whose goals include increasing capacity of the current pipeline infrastructure (10%), and reducing operational costs (50% by 2010). As part of this program, Southwest Research Institute[®] (SwRI[®]) is undertaking a project entitled, “Technologies to Enhance the Existing Natural Gas Compression Infrastructure.” This project is managed for DOE by the National Energy Technology Laboratory (NETL). The project objective is:

To develop and substantiate methods for operating integral engine/compressors in gas pipeline service, which reduce *fuel consumption*, increase *capacity*, and enhance mechanical *integrity*.

1.3 PROJECT ACCOMPLISHMENT

This project continues to document and demonstrate feasibility of technologies and operational choices for companies who operate the large installed fleet of integral engine compressors in pipeline service. Applying project results will enhance integrity, extend life, improve efficiency, and increase capacity, while managing NO_x emissions. These benefits will translate into lower cost, more reliable gas transmission, and options for increasing deliverability from the existing infrastructure on high demand days. In the process, the project has assembled a powerful suite of instruments and a data system with which it has characterized behavior of the units tested under a wide range of conditions. This suite will remain available for

characterization and optimization after completion of the project. At the project mid-point, the following documents its ongoing value and contribution to DOE goals.

1.3.1 Integrity

Increasing integrity and reducing statistical likelihood of component failure reduces transmission cost and enhances aggregate deliverability. Detonation represents a damaging threat to an engine. Applying the detonation detection technology tested under the project will mitigate this threat, which widely inhibits potentially beneficial operation with advanced timing. The newly defined CPR balancing method, which has proved quick and convenient to apply, will help equalize air/fuel ratio across cylinders and reduce the tendency to detonate. The low cost control method demonstrated for maintaining a global equivalence ratio set point provides another option for maximizing margin between misfire and detonation limits, using commercially available controllers. The crank Strain Data Capture Module (SDCM) applied on all engines tested has shown its value for defining conditions when crank damage rate increases. Measuring crankshaft torsional velocity has complemented the SDCM, particularly in documenting the influence of speed changes, showing also that torsional velocity data responds detectably to loss of torque from a misfire. The rod load monitor evaluated and enhanced on every test so far promises to avoid overload of engines and resulting damage, by improving consistency of load torque values used in load step control.

1.3.2 Efficiency

As much as three percent of the natural gas consumed goes as fuel gas for engines and turbines to drive compressors. This fuel gas would cost \$3 billion at current rates—the single most significant cost of gas transportation. Increasing the aggregate efficiency with which engine/compressors convert fuel energy into useful compression work will reduce this cost and leave more of the gas in the pipeline system available to the end user. The project has already documented how high-pressure fuel injection coupled with addition of a turbocharger on old GMW engines reduces heat rate by about seven percent. The demonstrated air/fuel ratio control on a rich burn, carbureted, four-stroke engine can replace manual adjustment and use of indirect measurement, allowing optimization for minimum fuel, for minimum emissions without a three-way catalyst, or for optimum catalyst performance if one is installed. The rod load monitor discussed above will allow engine operation at the point of highest efficiency (100% torque), with greatly reduced risk of overload. The detonation detector will safely allow more efficient engine operation with timing advanced. Comparison with the heat rate *versus* load characteristic has revealed small potential benefits in brake thermal efficiency by applying CPR balancing. Project mapping of system efficiency has made clear the importance of considering both compressor and engine when evaluating how operational decisions will impact fuel conversion efficiency; speed/load combinations which favor heat rate may at the same time hurt compressor efficiency, so that maximizing efficiency requires careful choices based on data. The project will continue to identify ways to enhance this efficiency, with emphasis on the compressor and pulsation control. The limitations in relying on differential indicated power for compressor efficiency calculations has emphasized the need for an effective method of measuring compressor flow and temperature rise, and the project will start evaluating alternatives. The project has also made clear the need for more information about mechanical losses and will seek

to add to this knowledge with new interpretation of the rod load data. Valve leaks represent a significant loss of compressor efficiency system-wide. Engine/compressor operators know the sensitivity of temperature rise to valve leaks, but the project has re-emphasized this sensitivity; the data normalization and statistical process control techniques already promoted by McKee, et al. [2], would lend themselves very effectively to monitoring of cylinder temperature rise and associated decision making based on economic significance. The project has documented air imbalance between cylinders as a widespread condition that can limit combustion efficiency; new tasks will characterize air imbalance in more detail and seek cost-effective solutions.

1.3.3 Capacity

As discussed above, integrity enhancement and reduced component failure probability will enhance aggregate deliverability. In addition to improving the efficiency of fuel conversion, all increases in compressor efficiency will reduce the fraction of available engine power that must go to overcome losses and will, thereby, also add to deliverability. Tests so far have shown a compressor efficiency range from 84% to 91%, adding to an earlier GMRC survey with a range from 52% to 92%! The high compressor efficiency values found present a benchmark that will add greatly to system capacity if more widely achieved. The remainder of the project will seek to re-emphasize compressor efficiency by characterizing and reducing compressor losses, both mechanical and thermodynamic. Measurements of flow, temperature rise, and dynamic pressure in the cylinder nozzles (as well as in the cylinders themselves) will help quantify and characterize inherent thermodynamic losses—a first step in their reduction. Previous tests have shown the likely contribution of pulsations to these losses, yet pulsation control methods such as acoustic filters and orifices must take account also of associated resistive pressure losses.

1.4 FIELD TEST PROGRAM OVERVIEW

The tests and analyses have been performed so far on two different two-stroke engine models from two manufacturers and on one four-stroke engine model: a Cooper GMW10 with three compressor cylinders, a Dresser-Clark HBA-6T with four compressor cylinders, and an Ingersoll-Rand KVG103. The HBA is a straight six with turbocharger. The GMW is a V-10 and has been tested both with and without the combination of a turbocharger and high-pressure fuel injection system. The KVG is a V-10 with three compressor cylinders. The engine selection was based on detailed quantitative analysis of the engine population using a database prepared for the pipeline industry, which shows all three of the tested models are in the top seven, measured by horsepower installed, and in the top six by number of units installed. Thus, marked diversity has been achieved in the process of testing three widely deployed engine models. This quarterly report focuses on initial (though quite extensive) results from a series of tests on a KVG103 at Duke's Thomaston station (Station Unit 7, Duke Engine Number 40207). The report also completes the comparison of performance on a specific GMW10 (Unit 6) at Williams' Station 60 (St. Francisville, Louisiana) from data obtained before and after modification to add high-pressure fuel and a turbocharger.

The data presented in this report on the KVG103 includes:

- Performance of engine (heat rate and normalized NO_x) as a function of timing, load, and fuel rate.
- Performance of the compressor, with distinction of compressor cylinder losses and installation losses.
- System efficiency as a function of timing, load, and fuel rate.
- Unique data and analysis of compressor piston friction losses.
- Functional demonstration of closed loop control of fuel rate to maintain an equivalence ratio set-point.

Further results will be presented in a subsequent report.

2. EXPERIMENTAL

2.1 SENSORS AND DATA CHANNELS FOR FIELD MEASUREMENT

Sensors and data acquisition capability have been assembled to record the following data on large integral engine compressors in the field.

Dynamic Pressure in the Compressor Cylinders – These measurements are used for compressor horsepower and flow determination. Both ends of each compressor cylinder have been instrumented for dynamic pressure in each test series. The sensors are Sensotec piezo-restrictive transducer. They are calibrated prior to each test by deadweight loading to generate known force per unit area in the test fluid applied to the sensing element.

Dynamic Pressure in the Engine Cylinders – These measurements are used for engine horsepower determination, engine balancing, and to calculate engine statistics. All power cylinders have been instrumented for dynamic pressure in each test series. The sensors are Kistler quartz piezoelectric transducers. Being dynamic sensors, they are calibrated prior to each test by suddenly applied deadweight loading to generate known force per unit area in the test fluid applied to the sensing element.

Dynamic Pressure in the Engine Air Intake Manifold – These measurements are used to correlate dynamic effects in the manifolds, which deliver air for each cylinder with the dynamic statistics within each cylinder. They also provide the time-averaged value for air manifold pressure whose influence on engine heat rate and emissions is assessed. Air manifolds have been instrumented in each test series. The sensors are Kistler piezo-resistive pressure transducers with factory provided calibration.

Dynamic Pressure in Engine Exhaust – These measurements are used to determine dynamic variation of pressure in the manifolds, which capture hot exhaust gas from each cylinder, and to correlate these dynamic pressure variations with the dynamics within each cylinder. The sensors are Kistler piezo-resistive transducer with factory provided calibration; they are water-cooled to reduce uncertainty resulting from temperature influence on the sensor readings.

Torsional Vibrations (IRV) – This measures the dynamic variation in speed of rotation of the flywheel. The sensor is a BEI 512 pulse encoder driven through a flexible coupling by a shaft connected by a friction drive to the flywheel. The frequency of its output pulse train directly reflects instantaneous flywheel angular velocity, which varies within each cycle of the engine as a result of dynamic load variation. Rather than digitally time the variation in periods between pulses (which imposes unrealistic period discrimination requirements), a frequency to voltage analog circuit is used to determine the continuous variation in flywheel speed. The speed variation is assessed as a surrogate for mechanical integrity. The frequency-to-voltage measurement is calibrated by supplying the analog circuitry with a pulse train of known frequency from a signal generator. The torsional vibration has been measured in this way on all tests. The torsional vibration data has been assessed as a potential indicator of engine dynamic loading severity.

Data Acquisition Triggering – The BEI encoder signal is also used to trigger acquisition of samples from all dynamic transducers. The phasing of the pulse train to top dead center is important. A pre-established top dead center mark for power Cylinder 1 is used as a reference, and the angular setting within the DAS corresponding to Cylinder 1 TDC is adjusted, as the engine runs, until a strobe light triggered by the DAS at this angle shows the mark on the flywheel coincides with the stationary mark.

Bearing Centerline Vibration – This measurement is assessed as an indicator of engine dynamic loading severity. The sensors are PCB velocimeters with factory provided calibration. The sensors have been located to measure lateral vibration at each end of the engine/compressor frame.

Crankshaft Dynamic Strain – This measurement is used as a direct indicator of shaft loading, and to provide a link between engine statistical quantities and potential for crankshaft fatigue damage [3]. The strain gage is placed on the crankshaft web as close as possible to the crank pin—at the point most sensitive to opening and closing of the crank throw faces under load from engine and compressor rods. Data is acquired by the Strain Data Capture Module (SDCM), which rides on the shaft within the engine during each day of testing, and from which data is downloaded at the end of each day. This is calibrated using a calibration resistance. The SDCM has worked with complete reliability for all tests so far.

Engine Fuel Flow – This is used to document overall engine efficiency. The sensor is an Emerson Flobas 103 transmitter, which implements the AGA3 flow measurement based on a differential pressure measurement and is factory calibrated with a certificate. This sensor is connected to taps on the already installed engine fuel flow orifice, which has been available on all engines tested so far. The fuel flow, coupled with a gas analysis, provides the basis for determining fuel energy consumed by the engine and for determining heat rate and overall system efficiency. At the first test, the flow measurement functioned, but the flow range was not properly matched to the engine, and satisfactory data was not obtained. At subsequent tests, the fuel flow has been successfully measured and used for the intended purposes.

Suction Header and Discharge Header Pressures and Temperatures – These measurements are used for installation efficiency determination. Pressures are measured with Sensotec piezo-restrictive transducers.

Engine Exhaust O₂ Level – This measurement is used to determine global equivalence ratio, both as an independent variable influencing engine performance, and where the loop is closed to the turbocharger waste-gate for active control. The sensor used is an NGK fast-response transducer, which provides a continuous variation of voltage with exhaust oxygen level. It is calibrated against a standard.

Engine Exhaust NO_x Level – This measurement is used to provide comparative emissions data. The sensor used is an NGK fast-response transducer, which provides a continuous variation of voltage with exhaust NO_x level. It is calibrated against a standard.

Compressor Rod Load – This measurement is used for both mechanical integrity and loading optimization. The sensor uses a pair of strain gages mounted on either side of the rod,

which are bridged additively to cancel bending and produce a signal proportional to axial load on the piston rod. The signal is transmitted using RF from a moving antenna to a stationary antenna. The strain gage and signal transmission can be powered by a battery or by a generator driven by rod motion. The battery power is adequate and simpler to set up for short-term tests, but for continuous monitoring and control, self-powering is needed. Calibration issues are not fully resolved yet for this device [termed the “rod load monitor” (RLM)]. So far, the horsepower measurement from the compressor cylinder, based on cylinder pressure transducer, has been used for calibration.

Knock Detection – Evaluation of a sensor provided by Metrix, which counts occurrences of dynamic acceleration levels above a threshold.

Particular experimental aspects of the KVG103 tests include some tests with dynamic pressure transducers installed in the compressor nozzles, use of the rod load monitor with the engine load reduced almost to zero to isolate piston friction losses, and some initial attempts to measure compressor gas throughput using a “strap-on” ultrasonic meter, as illustrated in Figure 2-1. This work on compressor throughput measurement did not produce flow data, because of excessive noise from the engine at the frequencies of interest but helped define the challenges involved, which should be surmountable in the future with additional filtering and signal processing. It is planned to pursue this throughput measurement further as an important additional basis for evaluating reciprocating compressor performance.

2.2 LABORATORY GMVH MEASUREMENTS FOR AIR BALANCE TASK

The GMVH engine was highly instrumented prior to utilization for the air balance investigation. However, additional dynamic pressure measurements were required for proper simulation with the computational model. The additional instrumentation is as follows:

Dynamic Pressure in Exhaust Manifold Runners – Prior to the air balance investigation, only Cylinder 1L was instrumented for dynamic exhaust pressure. Additional dynamic pressure sensors were added to the remaining five cylinders to capture the dynamic pressure pulsations of the exhaust from each cylinder’s ports. These sensors are of a thin-film strain gage type, typically used for absolute pressure measurement of manifold pressure in automotive electronic engine control systems. Each sensor was calibrated, and a comparison test to a Kistler piezo-resistive sensor was performed on the running engine to validate transient response.

Dynamic Pressure in Exhaust Manifold Plenum – A new sensor was installed in the exhaust manifold plenum near the turbocharger. This measurement is required to capture the dynamic pressure pulsations in the exhaust manifold plenum and provide data to characterize the dynamic flow through the exhaust manifold. A Kistler piezo-resistive absolute pressure transducer was utilized for this measurement. This sensor was calibrated via a deadweight tester. A photograph of the exhaust plenum sensor, as installed for testing, is provided in Figure 2-2.

Figure 2-1. Trial Installation of Strap-On Ultrasonic Sensor for Throughput Measurement

Figure 2-2. Photograph of Dynamic Exhaust Pressure Sensor in Exhaust Plenum

Dynamic Pressure in Inlet Manifold Plenums – Prior to the air balance investigation, only the left inlet manifold was instrumented for dynamic inlet plenum pressure. An additional dynamic pressure sensor was added to the right inlet manifold plenum to capture the dynamic pressure pulsations of the exhaust from each cylinder's ports. These sensors are of a thin-film strain gage type, like those utilized in the exhaust manifold runners.

The complete instrumentation package on the laboratory GMVH engine is listed in Table 2-1.

Table 2-1. Time-Averaged and Crank-Angle Resolved Measurements on GMVH

Time-Averaged Measurements	
Engine Speed	Oil Pressure
Turbocharger Shaft Speed	Turbocharger Oil Pressure
Turbocharger Wastegate Position	Coolant Inlet & Outlet Pressure
Engine Torque	Pre-Turbine Pressure
Total Fuel Flow	Stack Pressure
Pre-Chamber Fuel Flow	Compressor Inlet Temperature
Fuel Gas Composition	Compressor Left & Right Outlet Temperatures
Fuel Gas Heating Value	Inlet Manifold Left & Right Temperatures
Total Air Flow	Fuel Header Temperature
Barometric Pressure	Pre-Chamber Header Temperature
Ambient Temperature	Individual Cyl. Exhaust Runner Temperatures
Ambient Humidity	Pre-Turbine Temperature
Exhaust NOx Concentration	Post-Turbine Temperature
Exhaust CO Concentration	I/C Inlet Left & Right Water Temperatures
Exhaust HC Concentration	I/C Outlet Left & Right Water Temperatures
Exhaust CO2 Concentration	Oil Sump Temperature
Exhaust O2 Concentration	Oil Inlet Temperature
Exhaust Equivalence Ratio	Turbocharger Oil Inlet Temperature
Inlet Manifold Left & Right Pressures	Coolant Inlet & Outlet Temperatures
Fuel Header Pressure	Individual Cyl. Head Temperatures
Pre-Chamber Header Pressure	Dynamometer Inlet & Outlet Temperatures
Crank-Angle Resolved (Dynamic) Measurements	
Cylinder 1L Firing Pressure	Cylinder 1L Exhaust Runner Pressure
Cylinder 2L Firing Pressure	Cylinder 2L Exhaust Runner Pressure
Cylinder 3L Firing Pressure	Cylinder 3L Exhaust Runner Pressure
Cylinder 1R Firing Pressure	Cylinder 1R Exhaust Runner Pressure
Cylinder 2R Firing Pressure	Cylinder 2R Exhaust Runner Pressure
Cylinder 3R Firing Pressure	Cylinder 3R Exhaust Runner Pressure
Left Inlet Manifold Plenum Pressure	Right Inlet Manifold Plenum Pressure
Cylinder 1L Pre-Chamber Firing Pressure	Exhaust Manifold Plenum Pressure

In addition to the many measurements for engine performance and emissions, several static measurements are being made of the engine geometry. These geometric measurements have been determined to be of critical importance for proper simulation of the engine. The key geometric parameters to be determined are compression ratio, port timing, and port area for each cylinder of the test engine. In order to conduct the many detailed measurements, the engine was disassembled. A list of the many static measurements taken on each cylinder is provided in Table 2-2. From these measurements, several calculated parameters were derived and discussed in the next section.

Table 2-2. Static Measurements on Each Cylinder of GMVH

Piston Stroke (BDC to TDC)	Cylinder Bore (~1" from top)
Connecting Rod C-C (cyl 1L only)	Piston TDC Height (from cylinder top)
Pre-Chamber Volume	Piston Top Ring Land Diameter
Cylinder Inlet Volume (inc.ports)	Piston Top Ring Land Height
Cylinder Intake Flange Width	Piston Dome Angle
Cylinder Intake Flange Height	Piston Dome Height from edge
Cylinder Exhaust Flange Width	Piston Bowl Depth
Cylinder Exhaust Flange Height	Piston Bowl Volume (inc puller-hole)
Cylinder Head Volume	Piston Pin Center to Crown Height
Cylinder Head Gasket Step	Top Int Port to Gasket Step - A
Cylinder Head Gasket Thickness	Top Int Port to Gasket Step - B
Exhaust Port "Shape" - A	Top Int Port to Gasket Step - C
Exhaust Port "Shape" - B	Top Int Port to Gasket Step - D
Exhaust Port "Shape" - C	Top Int Port to Gasket Step - E
Exhaust Port "Shape" - D	Top Int Port to Gasket Step - F
Exhaust Port "Shape" - E	Top Int Port to Gasket Step - G
Top Exh Port to Gasket Step - A	Top Int Port to Gasket Step - H
Top Exh Port to Gasket Step - B	Intake Port to Edge Width - A
Top Exh Port to Gasket Step - C	Intake Port to Edge Width - B
Top Exh Port to Gasket Step - D	Intake Port to Edge Width - C
Top Exh Port to Gasket Step - E	Intake Port to Edge Width - D
Exhaust Port Edge Width - A	Intake Port to Edge Width - E
Exhaust Port Edge Width - B	Intake Port to Edge Width - F
Exhaust Port Edge Width - C	Intake Port to Edge Width - G
Exhaust Port Edge Width - D	Intake Port to Edge Width - H
Exhaust Port Edge Width - E	Intake Port Edge Height - A
Exhaust Port Min Width - A	Intake Port Edge Height - B
Exhaust Port Min Width - B	Intake Port Edge Height - C
Exhaust Port Min Width - C	Intake Port Edge Height - D
Exhaust Port Min Width - D	Intake Port Edge Height - E
Exhaust Port Min Width - E	Intake Port Edge Height - F
Exhaust Port Edge Height - A	Intake Port Edge Height - G
Exhaust Port Edge Height - B	Intake Port Edge Height - H
Exhaust Port Edge Height - C	Intake Port Angle - A
Exhaust Port Edge Height - D	Intake Port Angle - B
Exhaust Port Edge Height - E	Intake Port Angle - C
Exhaust Port Min Height - A	Intake Port Angle - D
Exhaust Port Min Height - B	Intake Port Angle - E
Exhaust Port Min Height - C	Intake Port Angle - F
Exhaust Port Min Height - D	Intake Port Angle - G
Exhaust Port Min Height - E	Intake Port Angle - H

Two of the six cylinders, representing a high and low compression pressure on a given bank, were to be flow tested. During disassembly it was found that Cylinder 1R had a different exhaust port shape from the other cylinders and was removed to be flow tested. Therefore, Cylinders 1L, 3L, and 1R were removed from the engine. The flow testing was conducted to measure the discharge coefficient of both intake and exhaust ports versus open area. Accurate discharge coefficients are required for accurate simulation. In addition, a review of allowable port shape on the manufacturing drawings gave concern that variance in port shape from cylinder to cylinder could be a large contributor to flow imbalance. The effects of port shape also needed to be characterized and accounted for in the simulations.

A flow test rig was assembled specifically for this effort. This test rig featured a compressed air storage and regulation system, meter run, data acquisition, and cylinder stand. Photographs of the flow bench rig are shown together in Figure 2-3. The compressed air system featured 3 to 1,050 gallon cylinders, charged to 250 psig. The outlet of the compressed air cylinders was connected to a regulator and control valve for setting the desired pressure versus mass flow of air into the flow bench. The meter run was fabricated from Schedule 40 PVC pipe and featured an ASME nozzle for flow measurement. Two sizes of flow nozzles, 2 and 4 inch, were interchangeably used for low and high flows. Mass flow was calculated from the volumetric flow measurements using standard equations given in ASME codes. The cylinder stand was fabricated to hold and seal the cylinder during testing. An adjusting screw protruded from the bottom of the stand to allow for adjustment of piston height to achieve the desired port open fraction. A vernier scale mounted on the bottom of the stand was used for measuring piston travel. A fixture was later fabricated to mount on the cylinder studs to lock the piston and prevent lifting due to air pressure leaking past the rings and under the piston. The data acquisition system acquired data at a rate of 6 Hz and included the measurements given in Table 2-3.

Figure 2-3. Photographs of GMVH Cylinder Flow Bench

Table 2-3. GMVH Cylinder Flow Bench Measurements

Supply Static Pressure (upstream)	Ambient Pressure
Supply Temperature (upstream)	Ambient Temperature
4" Nozzle delta-Pressure	Ambient Dewpoint Temperature
2" Nozzle delta-Pressure	Air Tank Pressure
Plenum Static Pressure (downstream)	Piston Travel from BDC
Plenum Temperature (downstream)	

2.3 COMPUTATIONAL MODELING FOR AIR BALANCE INVESTIGATION

The computational modeling for the air balance investigation is being performed with software purchased from Optimum Power Technology. The particular software package is titled *Automated Design with Virtual 2-Stroke*. This software is a one-dimensional cycle-simulation model that focuses on the fluid dynamics in an internal combustion engine.

A model of the GMVH engine was configured using the dimensions both provided by Cooper Compression and obtained through direct measurement. Being a one-dimensional computational model, many of the complex three-dimensional geometries were simplified to representative pipes, plenums, junctions, and orifices. A schematic of the computational model of the GMVH-6 engine is provided in Figure 2-4.

Figure 2-4. Current GMVH Computational Model Schematic

3. DATA ACQUISITION

3.1 FIELD DATA SYSTEM

Figures 3-1 and 3-2 show photographs of the Field Data Acquisition System (DAS). The system comprises an industrially hardened computer, a flat screen for display, and a separate box with connectors to which cables from individual sensors are connected. The DAS box has analog-to-digital converters of appropriate speed for over 50 different channels.

The individual power cylinder transducers (up to 10) are connected to a box with connectors on the deck near the cylinders. A single cable from this box carries the signals from all the power cylinder transducers to the main data acquisition box. A similar approach is used for the compressor cylinders. In this way, the complexity of the cabling and system checkout is minimized. Signals from rod load monitors, from other system pressures, and from temperature sensors are acquired by the DAS, concurrently, and a database of the sensor values throughout each test is created by the DAS.

3.2 LABORATORY GMVH ENGINE

A photograph of the laboratory GMVH instrumentation and control panel is depicted in Figure 3-3. The data acquisition system is PC-based, and features custom software written by SwRI. In addition to recording and displaying the measurements listed in Section 2.2, the data acquisition software is programmed with many calculated parameters that are displayed in real-time for monitoring performance and setting specific operating conditions.

Figure 3-1. Front View of Field Data Acquisition System (DAS)

Figure 3-2. Rear View of Field Data Acquisition System (DAS)

Figure 3-3. Laboratory GMVH Instrumentation and Control Panel

4. RESULTS AND DISCUSSION: TESTS ON A KVG103

4.1 OVERVIEW OF KVG103 TESTS

This report section presents results from recent tests on a KVG103, Unit 7, at Duke's Thomaston Compressor Station in South Texas. This unit was installed in 1955 and is one of several operated by Duke, with two of this configuration at Thomaston. It has ten power cylinders in Vee configuration and three compressor cylinders. This is the first four-stroke cycle engine tested under the program. It is a rich burn engine naturally-aspirated, operated at the leanest point at which it will run reliably—in the range of 0.8 to 0.85 equivalence ratio on the days tested. Up to now it is essentially in the same configuration as when installed—with the exception of the addition of an upgraded ignition system and twin carburetors. This unit is scheduled for the addition of a catalyst, for NO_x reduction. To make this conversion effective, the engine will need automatic control of its stack equivalence ratio, which is presently the result of indirect manual, open loop, adjustment by station operators, based on manifold vacuum.

The unit was tested for three successive days (Wednesday, Thursday, Friday; November 3, 4, 5). On the first day, engine and compressor characteristics were recorded over a number of imposed test conditions. The second day was to demonstrate closed loop equivalence ratio control. The third day was devoted to evaluation of loss characteristics: the compressor valve (DIP) losses, installation losses, and losses from the mechanical friction, which opposes piston motion.

As a carbureted four-stroke engine without individual fuel valves for each cylinder, it would not allow the process of cylinder-to-cylinder balancing to change mixture in individual cylinders (as evaluated previously on two-stroke engines). Thus, the main parameters investigated during Day 1 testing were load, timing, and fuel rate.

4.2 SITE AND INSTALLATION PHOTOGRAPHS

Figures 4-1 through 4-19 present a number of photographs showing the station, the unit, and the installed instruments.

Figure 4-1 shows compressor building at the station, with yard piping, and seven stacks. The first five stacks belong to the small SVG units at this station, and the last two belong to the pair of KVG103 units, the second of which (Unit 7, at the end of the building) was tested.

Figure 4-2 shows suction piping, discharge piping, and the stack for the tested unit.

Figure 4-3 overviews the tested unit from the back (non-compressor) side, looking from the oil pump end towards the flywheel. The photograph shows the left bank of power cylinders, with the fuel/air manifold running the length of the unit, outboard of the cylinders, just above the walkway decking level.

Figure 4-1. Photograph of Compressor Building, Duke's Thomaston Station

Figure 4-2. KVG103 – Suction Piping, Discharge Piping and Stack

Figure 4-3. KVG103 – Overview of Unit – Backside

Figure 4-4 overviews the data acquisition system, the compressor side of the unit, and the right bank of power cylinders. Close examination will show that the power cylinders are not evenly spaced axially; the first two on each side are closer together than the others, because the compressor cylinders are located in between power cylinders 2-3, 3-4, and 4-5, requiring wider spacing.

Figure 4-5 shows the panel for the unit, from which a number of parameters measured with permanently installed instruments were logged regularly during the tests.

Figure 4-6 provides a view of the three compressor cylinders from the side at the flywheel end.

Figure 4-7 shows Cylinder 1 “end-on” emphasizing the two unloaders for changing capacity and compressor load on the engine. Both are manually operated; the head end has a central pocket, connected or disconnected by a valve to the head end clearance, to change clearance volume; the crank end has a pocket visible on the left of the cylinder (about half the head end pocket volume), similarly connected or disconnected to the crank end clearance volume to change its clearance. By these mechanisms, the volume of clearance space for the entire compressor can be adjusted in steps, thereby, changing the rate of change of pressure with piston motion and controlling the volume compressed per cycle and the load on the engine. Only Cylinder 1 has two unloaders—Cylinders 2 and 3 have only a head end unloader.

Figure 4-4. Overview of Unit, Data System, and Compressor Cylinders During Day 1 Tests

Figure 4-5. KVG103 – Station Data Panel for Unit 7

Figure 4-6. KVG103 – Compressor Cylinders

Figure 4-7. KVG103 – Compressor Cylinder 1 Unloaders

Figure 4-8 shows the flow meter attached across the taps for the station fuel gas meter. The station does not have individual unit flow meters, but to accommodate the testing, only the tested unit was running, so the station flow equaled the unit fuel flow.

Figure 4-8. KVG103 – Fuel Flow Meter

Figure 4-9 shows the three stack sensors or taps installed, the UEGO sensor, the tap for the ECOM analyzer used for NO_x measurement, and the separate exhaust gas oxygen sensor used by the MEC-L Equivalence ratio controller.

Figure 4-9. KVG103 – Stack Sensors

Figure 4-10 shows the ECOM portable emissions analyzer and the laptop being used to acquire data from this analyzer.

Figure 4-10. ECOM Analyzer and Associated Data Acquisition Computer

Figures 4-11 and 4-12 show the encoder installation in overview and in more detail. The encoder shaft is pressed into a hole in the flywheel so it rotates at the same speed as the flywheel, driven through a friction drive. The encoder's 512 pulses per revolution each trigger acquisition of voltages from the dynamic sensors (compressor cylinders, power cylinders, manifold dynamic pressure, and the Rod Load Monitor).

Figure 4-11. KVG103 – Encoder Overview

Figure 4-12. KVG103 – Encoder Detail

Figure 4-13 shows the pressure transducers attached to the valved Kiene connections in the head and crank end of compressor Cylinder 3.

Figure 4-13. KVG103 – Compressor Cylinder 3 with Pressure Transducers

Figure 4-14 shows the pressure transducer attached to the Kiene valve connection in power Cylinder 6. In the background, the detonation detector can also be seen mounted on Cylinder 7.

Figure 4-14. KVG103 – Power Cylinder 6 with Pressure Transducer

Figure 4-15 shows the dynamic pressure transducer installed in one of the fuel/air manifolds; a pair of such sensors was actually installed, one on either side of the unit, since the engine has a manifold for each of two cylinder banks.

Figure 4-15. KVG103 – Fuel Manifold Pressure Sensor

Figure 4-16 shows the detonation detector loaned by Metrix in more detail than in Figure 4-14.

Figure 4-16. KVG103 – Power Cylinder 7 with Detonation Detector Mounted

Figure 4-17 shows the Cylinder 1 “dog-house” and the leads for the rod load monitor.

Figure 4-17. KVG103 – Dog House and Rod Load Monitor Leads

Figure 4-18 shows the installation of the RLM on the rod of Cylinder 1.

Figure 4-18. KVG103 – Rod Load Monitor Installation

Figure 4-19 shows one of the vibration sensors installed on the back side of the unit at each end.

Figure 4-19. KVG103 – Vibration Sensor – Flywheel End

4.3 DATA FROM PERMANENTLY INSTALLED STATION INSTRUMENTS

The next 11 figures are plotted from data logged from the unit panel shown in Figure 4-5. This is data measured with permanently installed station instruments. In addition, two figures present ambient temperature data from the ECOM portable analyzer records. To provide context, the variation in test conditions is first discussed.

Tables 4-1 and 4-2 show the log of imposed test conditions for the first two days. The first days of testing involved first baseline data, in as-found condition, with load as close to 100% as discrete pocket settings would allow without going over. The earliest data involved the unit fully loaded, but according to load charts, it was above Duke's load limit (105%). One head end pocket was opened at about 9:20, giving about 97% load according to station charts. Then a second pocket was opened on Cylinder 2 head end to reduce load to about 86%. The second pocket was then closed again to bring the load back up as close to 100% as possible. Then the timing was advanced by two or three degrees and data was taken. The timing was then returned to as-found condition (about 20 degrees BTDC). The fuel rate was reduced, giving an equivalence ratio of about 0.79. As a final test that day, the fuel rate was increased to significantly richen the unit's operation to about 0.93, with the unit then brought back to nominal baseline conditions.

**Table 4-1. KVG103 – Log of Test Conditions
Day 1, Wednesday, November 3, 2004
Duke Thomaston Station**

From	To	Conditions
8:41:09	9:20:04	Initial conditions; all pockets closed; load slightly over Duke's limit; baseline timing (20 DBTDC).
9:24:44	9:58:09	One pocket open; load at about 97%; baseline timing; fuel rate *reduced by station personnel.
10:01:33	10:33:30	Load reduced to 86%; same timing; fuel rate reduced by station personnel.
10:35:24	10:39:07	Load returned to 97%.
10:42:42	11:02:12	Timing advanced 2 or 3 DBTDC relative to baseline; load about 97%.
11:22:58	11:36:33	Timing returned to baseline; load about 97%.
11:38:46	11:47:44	Fuel rate reduced to give equivalence ratio of about 0.79; load about 97%.
11:54:29	12:23:44	Fuel rate increased to give equivalence ratio of about 0.93; load about 97%.
12:26:02	12:28:15	Return to baseline.

**Table 4-2. KVG 103 – Log of Test Conditions
Day 2, Thursday, November 4, 2004
Duke Thomaston Station**

From	To	Conditions
10:39:38	11:10:03	Baseline – open loop; 97%load.
11:13:47	11:21:20	Unloaded to 81% in open loop.
12:31:49	12:54:32	First closed loop operation.
13:05:16	13:17:27	Unloaded to 80% in closed loop.
13:21:57	13:29:32	Baseline closed or open loop.
13:44:42	13:58:20	Open loop fuel rate increased – equivalence ratio ~99%.

On Day 2, the testing was directed mainly at demonstrating closed loop control of global equivalence ratio in anticipation of Duke's intentions to add a catalyst, which is likely to operate most effectively when the exhaust maintains a small and constant excess oxygen. To test functionality of the closed loop control, the load was reduced to about 80%, first without the controller (open loop), and later in the day with the controller active (closed loop). At the end of the second day's testing, the fuel rate was increased. The second day was mainly a day of operational adjustments to the closed loop controller, its support data channels, and the valve control signal, to achieve functionality, but data was taken during the day, which provides documentation of how the engine behaved during this logistical and demonstration process.

Figures 4-20 and 4-21 show the values for suction and discharge pressure across the unit during two days of testing (Wednesday, November 3 and Thursday, November 4). These figures document that pipeline conditions changed very little during these two days of testing. On the first day, the suction pressure remained between 692 and 704 PSI—less than 2% variation; discharge pressure lay between 1,157 and 1,168 PSI—a 1% variation. On Day 2, suction pressures remained between 676 and 685 PSI, and discharge pressures between 1,135 and 1,147 PSI. Thus, the absolute levels of both suction and discharge dropped lower in the second day's testing. Expressed as a ratio, in Figures 4-22 and 4-23, the variation is between 1.63 and 1.67, less than 3% variation in ratio over the two days of testing.

Figures 4-24 and 4-25 show similarly small variation in suction and discharge temperatures over the two days.

**Figure 4-20. KVG103 – Suction and Discharge Pressure – Day 1 Testing.
Wed., Nov. 3, 2004 – Duke Thomaston Station**

Figure 4-21. KVG103 – Suction and Discharge Pressure – Day 2 Testing
Thurs., Nov. 4, 2004 – Duke Thomaston Station

Figure 4-22. KVG103 – Compression Ratio – Day 1 Testing
Wed., Nov. 3, 2004 – Duke Thomaston Station

Figure 4-23. KVG103 – Compression Ratio – Day 2 Testing
Thurs., Nov. 4, 2004 – Duke Thomaston Station

Figure 4-24. KVG103 – Suction and Discharge Temperature – Day 1 Testing
Wed., Nov. 3, 2004 – Duke Thomaston Station

**Figure 4-25. KVG103 – Suction and Discharge Temperature – Day 2 Testing
Thurs., Nov. 4, 2004 – Duke Thomaston Station**

Although the conditions changed little, an effort was made to identify the extent to which the data reflected a relationship between temperature rise and compression ratio. Figure 4-26 shows a distinct relationship, with positive slope, as would be expected. The regression model, with slope of 71.7 degrees per unit ratio change has a standard error of 0.74 degrees. A more extensively mapped relationship could form the basis for detecting and discriminating valve leaks, if digital logic were implemented to trap repeated temperature rise values, which were, say, three standard errors outside the established mean relationship.

**Figure 4-26. KVG103 – Temperature Rise as a Function of Ratio for Days 1 and 2
Wed. and Thurs., Nov. 3 and 4, 2004 – Duke Thomaston Station
Regression Standard Error = 0.74 Degrees**

Figures 4-27, 4-28, 4-29, and 4-30 show the exhaust temperatures for the ten cylinders, five to a page for Days 1 and 2. The average temperature under nominal conditions is in the range of 1,230°F to 1,250°F. The spread in exhaust temperature is about 170 degrees (close to 15%). Closer review indicates the highest exhaust temperature is clearly from Cylinder 10, with the next highest, Cylinder 4, about 20 to 30 degrees lower. The temperatures tend to track each other as test conditions change, with the exception of Cylinder 10. Late in each day's testing, the test logs of Tables 4-1 and 4-2 shows the fuel rate was increased, with a direct increase in global equivalence ratio. On Day 1, as a result, exhaust temperatures for Cylinders 1 through 5 go up about 50 degrees, and for Cylinders 6 through 9 by close to 70 degrees; however Cylinder 10 goes up by only about 20 degrees. Similar differences in the temperature rise when the unit operation was richened are observed in Day 2's data. This anomalous behavior of Cylinder 10 will be discussed again when pressure measurements are presented. Since Cylinders 1 through 5 are on one side and Cylinders 6 through 10 are on the other side, the possibility of bank-to-bank differences may be noted.

Figure 4-27. KVG103 – Exhaust Temperature for Cylinders 1 Through 5 – Day 1 Testing
Wed., Nov. 3, 2004 – Duke Thomaston Station

Figure 4-28. KVG103 – Exhaust Temperature for Cylinders 6 Through 10 – Day 1 Testing
Wed., Nov. 3, 2004 – Duke Thomaston Station

**Figure 4-29. KVG103 – Exhaust Temperature for Cylinders 1 Through 5 – Day 2 Testing
Thurs., Nov. 4, 2004 – Duke Thomaston Station**

**Figure 4-30. KVG103 – Exhaust Temperature for Cylinders 6 Through 10 – Day 2 Testing
Thurs., Nov. 4, 2004 – Duke Thomaston Station**

Figures 4-31 and 4-32 present the variation of ambient temperature during Wednesday and Thursday's tests. This data was recorded by the ECOM portable analyzer. It shows temperatures rising throughout the testing of each day. It also shows higher temperatures during part of Day 2's testing, although this seems to have occurred because Day 2's tests started later and continued later. Close examination indicates ambient temperatures reached a similar level at corresponding times on each day.

**Figure 4-31. KVG103 – Ambient Temperature – Day 1 Testing
Wed., Nov. 3, 2004 – Duke Thomaston Station**

**Figure 4-32. KVG103 – Ambient Temperature – Day 1 Testing
Thurs., Nov. 4, 2004 – Duke Thomaston Station**

4.4 LOAD AND SPEED DATA

Figures 4-33 and 4-34 present the variation of load, expressed as compressor indicated horsepower, together with the indicated engine cylinder horsepower. The engine and compressor power maintain a rational relationship to each other, with an implied efficiency of power transmission from power to compressor cylinders between 92% and 95%. Since the nature of pressure data acquisition through Kiene valves for the engine probably loses some accuracy, this engine load should be treated as approximate.

Figure 4-33. KVG103 – Engine and Compressor Indicated Power – Day 1 Testing
Wed., Nov. 3, 2004 – Duke Thomaston Station

Figure 4-34. KVG103 – Engine and Compressor Indicated Power – Day 2 Testing
Thurs., Nov. 4, 2004 – Duke Thomaston Station

Between 8:41 and 9:56 on Day 1, baseline data was acquired, with the unit as close to 100% load as possible. In actuality, the data indicated an intermediate reduction after the compressor load to keep it below Duke's limits at about 9:22. Then, at 9:56, a further distinct drop in load is apparent to about 830 HP. At about 10:30, the load was increased again to just under 1,000 HP and stayed nearly constant for the remainder of the day.

On Day 2, the load was maintained reasonably constant throughout the tests with the exception of two intentional reductions, the first around 11:11 for open loop testing, and the second around 12:50 for closed loop testing.

Figure 4-35 shows the speed records for Day 1 and indicates a reasonably constant speed with perhaps 5 to 7 RPM variation about the nominal 330 RPM. There are a significant number of zero RPM points, which indicate bad triggering. Speed on Day 2 also stayed very close to 330 RPM throughout the testing

**Figure 4-35. KVG103 – Speed – Day 1 Testing
Wed., Nov. 3, 2004 – Duke Thomaston Station**

4.5 GLOBAL EQUIVALENCE RATIO

Figures 4-36 and 4-37 present the variation of global (stack) equivalence ratio on the first two days of testing, with compressor brake horsepower superimposed. For most of Day 1, global equivalence ratio lies between 0.8 and 0.83, except that around 11:46, an air/fuel ratio sweep was started, first the mixture was intentionally made leaner (about 0.79) and then, starting at 11:54:29 the fuel rate was intentionally richened, causing equivalence ratio to increase to about 0.93. Finally, for a few minutes equivalence ratio was reduced back to 0.83.

**Figure 4-36. KVG103 – Compressor ICHP and Equivalence Ratio – Day 1 Testing
Wed., Nov. 3, 2004 – Duke Thomaston Station**

Review of the superimposed load variation for Day 1 in Figure 4-36 shows that after each reduction in load, the engine tends to richen. This is apparent around 9:24 and again at about 10:03, each of which are points of distinct load reduction. Station operating procedure after each load adjustment is to adjust the fuel to air ratio from the carburetors according to standard charts, using manifold vacuum as a guide (because they have no measure of O₂ or equivalence ratio), with the intent of keeping operation at a desirably lean level. The momentary increase in global equivalence ratio followed by a distinct and intentional reduction can be seen corresponding to each of the load reductions. When load is adjusted back up at about 10:44, a momentary reduction in equivalence ratio can be seen, followed by an increase as the station personnel adjusted the carburetor again, this time to richer operation for the higher load.

Figure 4-37 shows compressor BHP and global equivalence ratio on Day 2. As the log of Table 4-2 documents, load was reduced twice during the day, the first time on open loop and the second time on closed loop. After the first load reduction, the station personnel did not adjust the carburetor after load was reduced; the natural richening of the mixture was allowed to occur and the equivalence ratio shows a richening response in the first load reduction. For the second load reduction, the MEC-L Flo-Tech Control Valve was allowed to adjust fuel rate to maintain a set point on the signal from its oxygen sensor. As will be subsequently discussed, the controller appears to have executed this function successfully because the oxygen sensor on the ECOM analyzer shows a constant setting on closed loop control. However, the NGK UEGO presents a less distinct picture—after load reduction, the equivalence ratio appears to start rising on closed loop control, just as it did on open loop control, then to spike down, then to increase again.

**Figure 4-37. KVG103 – Global Equivalence Ratio and Compressor ICHP – Day 2 Testing
Thurs., Nov. 4, 2004 – Duke Thomaston Station**

4.6 ENGINE PEAK FIRING PRESSURE CHARACTERISTICS

Figures 4-38 and 4-39 shows the variation of peak firing pressure during the first two days of testing, with equivalence ratio superimposed. Nominally, the average peak firing pressure is about 275 PSIG, and for Cylinder 10 nominal peak firing pressure is about 190 PSIG. On Wednesday, Day 1, there appears to be some level of tracking of peak firing pressure with equivalence ratio, particularly during the air/fuel ratio sweep which started 11:38. As is to be expected, a leaner mixture reduces peak firing pressure and a rich mixture increases peak firing pressure. The response of Cylinder 10 to the change in air/fuel ratio is much more pronounced than for the average of all cylinders. Cylinder 10 also shows a much lower peak firing pressure than the other cylinders, by as close to 100 PSI for most of the day, until the mixture is richened.

**Figure 4-38. KVG103 – Peak Firing Pressure Averaged for All Cylinders and for Cylinder 10 Alone,
with Global Equivalence Ratio Superimposed – Day 1 Testing
Wed., Nov. 3, 2005 – Duke Thomaston Station**

In fact, Cylinder 10 barely generates power until the mixture is richened—at which point its peak firing pressure rises above the average. During the early part of the day, the relationship between equivalence ratio and peak firing pressure is much less distinct, because load and timing were being changed as well as equivalence ratio. After 11:38, only air/fuel ratio was changed.

Figure 4-38 shows the reduction in average peak firing pressure with reduced load up to about 10:30, followed by an increase as load is returned to 97%, and then a further increase in average peak firing pressure as timing is advanced two or three degrees. Cylinder 10 shows some response to load and timing changes, but not as pronounced.

On Thursday, Day 2, Figure 4-39 clearly shows the average peak firing pressure drops both times the load is reduced. However, Cylinder 10 is much less affected by the load reductions. As on the previous day, Cylinder 10 PFP responds very strongly to a richening of the air/fuel ratio. The average PFP also increase with a richer mixture again, but the influence is much less than for Cylinder 10. On Day 2, the typical PFP value for Cylinder 10 is about 175 PSIG, and the average is about 280 PSIG—more than 100 PSI difference.

Figure 4-39. KVG103 – Peak Firing Pressure Averaged for All Cylinders and for Cylinder 10 Alone, with Equivalence Ratio Added – Day 2 Testing – Thurs., Nov. 4, 2005 – Duke Thomaston Station

Figures 4-40 and 4-41 show the variation of the standard deviation of cycle-to-cycle variability during the first two days of testing, normalized with respect to peak firing pressure, both for all cylinders and for Cylinder 10 alone. The quantity plotted is, essentially, the coefficient of variance. The nominal value for this coefficient of variance is about 10%, dropping to about 7% for the richest conditions tested late in the first day. Cylinder 10 indicates a standard deviation of 15%, which implies, since peak firing pressure for Cylinder 10 is about 65% of the average, that the standard deviation for Cylinder 10 in PSI is similar to the average. On Day 1, the air/fuel ratio sweep influences Cylinder 10's coefficient of variance most strongly. Richening the mixture drops the coefficient of variance for Cylinder 10 from 15% to 10%.

Figure 4-40. Cycle-to-Cycle Standard Deviations in Peak Firing Pressure – Average and Cylinder 10 – Day 1 Testing – Wed., Nov. 3, 2004 – Duke Thomaston Station

Figure 4-41. KVG103 – Standard Deviation in Peak Firing Pressure (PFP) as a Percentage of PFP – Average and Cylinder 10 – Day 2 Testing – Thurs., Nov. 4, 2004 – Duke Thomaston Station

On Day 2, Thursday, Figure 4-41 indicates the same nominal coefficient of variance, 10%, but here, Cylinder 10's coefficient is more like 13.5%, distinctly lower than the day before. Decreasing load in open loop at about 11:13 slightly drops the average coefficient of variance to 9%, but increases it for Cylinder 10 to about 16%. In closed loop (13:03 to 13:17), the story is less distinct, but Cylinder 10's coefficient of variance again appears to increase under reduced load. The most pronounced trend in Figure 4-40 occurs late in the day with a richer mixture,

which greatly reduces the coefficients of variance to about 10% for Cylinder 10 and to about 8% for the average.

Figures 4-42 and 4-43 present data on the spread across the cylinders, together with the value of peak firing pressure averaged across the cylinders. As in previous engine studies, the spread is presented in two ways—the spread in the cycle averaged peak firing pressure for each cylinder, and the instantaneous spread in peak firing pressure averaged over a number of cycles. For all two-cycle engines tested so far, there was a wide difference between these two spread numbers, whereas here they are within 20 PSI of each other out of 150 PSIG, and track very closely. Here, the spread characteristics are probably dominated by Cylinder 10's behavior, because its PFP is nominally 90 to 100 PSI lower than the other cylinders out of the 130 spread shown. Both spreads reduce with decreasing load at about 10:03 on Day 1. The most pronounced trend in Figure 4-42 is the reduction in both spreads with richening of the mixture. This is predominantly a result of the richened mixture bringing up Cylinder 10's peak firing pressure from its low value.

**Figure 4-42. KVG103 – Spread in Cycle Averaged Cylinder Pressure and the Average Instantaneous Spread, with Peak Firing Pressure
Day 1 Testing – Wed., Nov. 3, 2004 – Duke Thomaston Station**

On Day 2, both the average spread and the instantaneous spread are significantly higher, 185 and 200 PSI, respectively, but again track very closely. Decreasing load appears to have a more pronounced influence on Day 2 than on Day 1, whether in open or closed loop, but this appears to be because the nominal spread was higher on Day 2; after load reduction, the spread is similar to that on Day 1 with reduced load. As on the previous day richening, the mixture sharply reduces both spreads (by a factor of 2), while increasing PFP to over 30 PSIG.

**Figure 4-43. KVG103 – Spread in Cycle Averaged Cylinder Pressure and the Average Instantaneous Spread, with Peak Firing Pressure
Day 2 Testing – Thurs., Nov. 4, 2004 – Duke Thomaston Station**

Figures 4-44 and 4-45 simply reproduce Figures 4-42 and 4-43 in normalized format, where the spread is divided by the peak firing pressure value superimposed in Figure 4-42 and Figure 4-43. These figures show the high percentage spread of 45% to 55% for Day 1, and 65% to 70% for Day 2, dropping to 20% for spread in averaged pressure, and 30% for instantaneous spread with the richer mixture.

**Figure 4-44. KVG103 – Normalized Spreads – Day 1 Testing
Wed., Nov. 3, 2004 – Duke Thomaston Station**

Figure 4-45. KVG103 – Normalized Spreads
Day 2 Testing – Thurs., Nov. 4, 2004 – Duke Thomaston Station

In spite of the high spread, no balancing was possible on this engine because the carburetor controls the nominal air/fuel ratio for the entire engine, and there is no control over the air/fuel ratio for individual cylinders. The behavior of Cylinder 10 is anomalous, because it is nominally supplied with the same fuel mixture as the other cylinders. This will be subsequently discussed further, with possible explanations.

4.7 COMPRESSION PRESSURE AND COMBUSTION PRESSURE RATIO CHARACTERISTICS

A characteristic recorded and noted on two-stroke cycle engines under this project has been the compression pressure and its variation with time and across the engine. For two strokes, this was taken as the pressure 20 degrees before Top Dead Center (TDC). For four strokes, the same definition has been used during the compression stroke, prior to ignition. Of course, for the four-stroke, there is a separate pumping cycle before the compression stroke, in which exhaust gas is pushed out by the piston and fresh mixture is induced. There is not the overlap of functions for a four-stroke that there is for a two-stroke. Figures 4-46 and 4-47 present compression pressures for all ten cylinders for the two days of testing. In the first day, there are two anomalous spikes, which need to be ignored. The relatively low spread in compression pressure should be noted—less than 5 PSI, whereas it was as much as 20 PSI during two-stroke testing. The magnitude of compression pressure is 80 to 85 PSIG, quite low compared to the two stroke engines tested (with recorded values of 200 PSIG, 300 PSIG, or even higher). Compression pressures track each other closely through all changes in conditions of the two days testing, including Cylinder 10. Throughout the day, the spread does not exceed 5 PSI.

**Figure 4-46. KVG103 – Compression Pressure (20 Degrees Before TDC)
Day 1 Testing – Wed., Nov. 3, 2004 – Duke Thomaston Station**

**Figure 4-47. KVG103 – Compression Pressure (20 Degrees Before TDC)
Day 2 Testing – Thurs., Nov. 4, 2004 – Duke Thomaston Station**

Figures 4-48 and 4-49 present the combustion pressure ratio (CPR) for the Wednesday and Thursday tests. This quantity is the ratio of peak firing pressure to compression pressure. This quantity averages about 3.5, with a range from 3.2 to 4.2 over all test conditions. This is higher than observed on any two strokes tested, which have exhibited CPR values of 1.8 to 3. On Day 1 a small tendency to drop with reduced load may be observed. On Day 2, the open loop tests for which no fuel rate adjustment was made do not reflect any noticeable change in

CPR with load. However, the closed loop tests do show some reduction in this ratio; thus, it appears, from this observation, that the closed loop operation does approximate the manual adjustments. As would be expected the ratio appears to increase to about 3.8 when timing is advanced. It also increases to as high as 4.1 or 4.2 when the mixture is richened.

**Figure 4-48. KVG103 – Average Combustion Pressure Ratio
Day 1 Testing – Wed., Nov. 3, 2004 – Duke Thomaston Station**

**Figure 4-49. KVG103 – Average Combustion Pressure Ratio
Day 2 Testing – Thurs., Nov. 4, 2004 – Duke Thomaston Station**

4.8 NORMALIZED NO_x CONCENTRATION

Figures 4-50 and 4-51 show the normalized NO_x concentration for the Wednesday and Thursday tests. On Day 1, reduced load reduces NO_x once the manual fuel adjustment is made. Advanced timing increases NO_x, as does richening the mixture.

Figure 4-50. KVG103 – Normalized No_x Concentration – Day 1 Testing
Wed., Nov. 3, 2004 – Duke Thomaston Station

Figure 4-51. KVG103 – Normalized No_x Concentration – Day 2 Testing
Thurs., Nov. 4, 2004 – Duke Thomaston Station

On Day 2, levels are generally similar to Day 1. NO_x rises slightly during the open loop load reduction test and drops starting midway through the closed loop test.

4.9 FUEL CONSUMPTION AND HEAT RATE

Figures 4-52 and 4-53 present the fuel flow measured during the Wednesday and Thursday tests. The units are thousands of standard cubic feet per day. The drop with reduced load is clear on Days 1 and 2. The levels appear slightly lower on Day 2.

**Figure 4-52. KVG103 – Fuel Flow – Day 1 Testing
Wed., Nov. 3, 2004 – Duke Thomaston Station**

**Figure 4-53. KVG103 – Fuel Flow – Day 2 Testing
Thurs., Nov. 4, 2004 – Duke Thomaston Station**

A more meaningful measure of combustion efficiency may be seen in the next two figures (Figures 4-54 and 4-55) for heat rate. This represents the fuel energy burned per unit of useful engine power delivered to the shaft, in the units of BTU/HP-Hr. These figures have the general test conditions superimposed, although the logs in Tables 4-1 and 4-2 provide a more definitive breakdown. On Day 1, an increase in heat rate from about 9,200 to as high as 9,600 can be seen when load is reduced to 87%, followed by a similar drop in heat rate when the unit is reloaded. A drop to about 8,800 can be seen when timing is advanced, and this is also reversed with an increase to 9,100 when timing is returned to its baseline value. The air/fuel ratio sweep starts at about 11:38 and a steady increase in heat rate follows, even though the sweep starts with a reduction in air/fuel ratio to 0.79 before an increase to 0.93. For these tests, changing air/fuel ratio either up or down is accompanied by an increase in heat rate.

Figure 4-54. KVG103 – Heat Rate (LHV = 979) – Wed., Nov. 3, 2004 – Duke Thomaston Station

Figure 4-55. KVG103 – Heat Rate (LHV = 979)
Day 2 Testing – Thurs., Nov. 4, 2004 – Duke Thomaston Station

On Day 2, the most noticeable trend in Figure 4-55 is the increase in heat rate with load reduction. The reduced load on Day 2 went down to about 80%, and the heat rate went as high as 9,700. No distinction can be drawn between closed loop or open loop as far as heat rate is concerned. Richening the mixture essentially to stoichiometric (excess oxygen = 0.3%) on Day 2 causes the highest heat rate of the tests at 9,900. The Day 1, highest equivalence ratio was 0.94, with excess oxygen of 0.9%.

Figures 4-56 and 4-57 present heat rate as a function of load. Past tests have shown that this presentation can better discriminate positive or negative trends, which load changes might otherwise mask or confuse when heat rate values alone are compared. Seven different test conditions are identified on Figure 4-56. The global trend—a reduction in heat rate with load—is re-confirmed for this four-stroke engine. However, it is less clear what other trends can be conclusively discerned from the different groupings. The lack of repeatability between the two nominal loads of 96% suggests some inherent scatter. The lowest load case of 86% does not increase heat rate as much as the slope of higher load cases would suggest. Advanced timing does produce some reduction in heat rate from other points with the same load, but the trend is not as pronounced as has been seen in two-stroke tests.

Looking at the extremes on this chart (8,700 to 9,800 change in heat rate for a 1,025 to 800 change in load) a nominal value for normalized rate of change:

$$(Load/Heat Rate) * d (Heat Rate) / dLoad$$

of about -0.4 can be inferred. This implies approximately that for a one percent drop in load, the heat rate goes up by 0.4%. For the HBA-6T tested earlier in 2004, the same normalized rate of change of heat rate was about -0.6. Thus, the sensitivity appears somewhat less for this four-stroke engine compared to some two strokes, but still significant and providing strong motivation to maximize load up to a safe limit.

**Figure 4-56. KVG103 – Heat Rate Versus Load, with Different Test Conditions Identified
Day 1 Testing – Wed., Nov. 3, 2004 – Duke Thomaston Station**

**Figure 4-57. KVG103 – Heat Rate Versus Load with Test Conditions Identified
Day 2 Testing – Thurs., Nov. 4, 2004 – Duke Thomaston Station**

The entire Day 2 heat rate versus load data in Figure 4-57 is closely consistent with the Day 1 data. The closed loop data indicates slightly lower heat rate than the open loop data.

4.10 COMPRESSOR LOSSES AND EFFICIENCY

Figure 4-58 presents preliminary information related to compressor efficiency—the differential indicated power (DIP) calculated from the difference in area on the pressure-volume diagram between the cylinder pressure trace and the lines drawn horizontally through the suction and discharge toes. This is intended to distinguish valve losses, but requires an assumption that nozzle pressure remains constant.

In anticipation of a more focused evaluation of compressor performance in future tests, the third day of testing at Thomaston included nozzle measurements. The availability of nozzle pressures avoids the need to assume constant nozzle pressure, and thereby allows more accurate DIP calculation. Without nozzle traces, such discrimination was not possible in previous tests. The detailed processing and corrections applied to cylinder and nozzle traces is discussed further in Section 4.13 of this report. The refined cylinder pressure to nozzle DIP was determined to be 5.71% (53.42 HP total DIP compared to 935.13 total piston face [indicated] HP for the three compressor cylinders). A separate analysis, based on enthalpy change between suction and discharge headers indicates a total loss through valves and attached piping equal to 10.81% of the piston face HP, as presented below. It is the DIP/valve loss based on nozzle traces, which is used in this analysis, rather than the data of Figure 4-58.

Figure 4-58. Compressor Differential Indicated Power (DIP) Values

The analysis process involves a calculation of enthalpy power as the product of flow and enthalpy rise across the compressor, from header to header. This enthalpy rise is calculated from the measured suction pressure, discharge pressure, suction temperature, and discharge temperature in the headers, shown in Figures 4-20, 4-21, 4-24, 4-25.

$$\text{Enthalpy Power} = K(h_d - h_s) * \dot{m}$$

$$\text{Where } h_s = f_h(P_s, T_s), \quad h_d = f_h(P_d, T_d) \text{ BTU / Lb}$$

K is the constant, which converts BTU/Sec to HP.

f_h is the thermophysical property function for enthalpy (e.g., from the BWR thermophysical property relationships).

And $\dot{m}$ is the mass flow in Lb/sec, calculated as the product of swept volume per second multiplied by suction volumetric efficiency and suction density.

The difference between the indicated HP and this enthalpy power is inferred as the total loss equal to valve loss added to loss in the piping between the compressor and the headers. The installation loss is calculated by subtracting valve loss from the total loss. The numerical calculation is as follows:

$$\begin{aligned}
\text{Enthalpy HP} &= 834.05 \\
\text{Indicated HP} &= 935.13 \\
\text{Total DIP (S\&D)} &= 5.71\% = 53.42 \text{ HP} \\
\text{Thus, Total Loss} &= 935.13 - 834.05 = 101.08 \text{ HP} \\
\text{Total DIP HP} &= 53.42 \\
\text{Installation Loss} &= 101.08 - 53.42 \text{ HP} \\
&= 47.66 \text{ HP} \\
&= 47.66/935.13 = 5.10\% \text{ of piston face or indicated power}
\end{aligned}$$

$$\begin{aligned}
\text{And Compressor Thermal Efficiency} &= (\text{Enthalpy Power/Indicated Power})\% \\
&= 834.05/935.13 = 89.19\%
\end{aligned}$$

We can calculate a “Compressor Piston face heat rate” (HR_{cpf}) as the fuel flow thermal power per unit compressor indicated power and, knowing compressor thermal efficiency, we can calculate an overall system thermal efficiency as follows:

$$\begin{aligned}
HR_{cpf} &= 1000 * MSCFD * LHV / (IHP) / 24 \\
\eta_{system} &= \eta_c * 2546.6 / (HR_{cpf})
\end{aligned}$$

Where η_c is the compressor thermal efficiency,

η_{system} is the overall system thermal efficiency,

MSCFD is the fuel mass flow rate in thousands of standard cubic feet per day,

LHV is the lower heating value of the fuel gas in BTU/Standard Cubic Foot,

IHP is the indicated compressor Horsepower.

4.11 SYSTEM OVERALL THERMAL EFFICIENCY

The compressor piston face heat rate is higher than the engine heat rate of Figures 4-54 and 4-55 by a factor equal to the inverse of the mechanical efficiency of the compressor. An assumed value for compressor mechanical efficiency of 95% has been used to convert compressor indicated (piston face) power (a measured quantity) to brake power in the calculation of engine heat rate; however calculating compressor piston face heat rate avoids the need for this assumption, since directly measured quantities (fuel flow, LHV, and indicated power) are used to determine it. In turn, overall system thermal efficiency can be determined from the piston face heat rate combined with the compressor thermal efficiency, making it a performance parameter, which depends only on measured quantities or quantities derived from measured quantities, and requires no assumed efficiencies. This is one of several valuable attributes of using overall system thermal efficiency to assess performance of the overall engine/compressor fuel conversion process.

Figures 4-59 and 4-60 present system overall efficiency calculated in this way for Days 1 and 2. They show values for this tested unit in the range 22.2% to 24.8% for Day 1 and from 21.8% to 24.6% on Day 2. The changes directly follow from the engine heat rate variations discussed previously. Even with the relatively high compressor efficiency, the high heat rates of this engine result in less than 25% of the fuel gas energy burnt being converted into useful compression work. Some of the efficient two strokes tested have been shown to achieve as high as 30% overall system efficiency or more.

**Figure 4-59. KVG103 – Overall System Efficiency – Day 1 Testing
Wed., Nov. 3, 2004 – Duke Thomaston Station**

**Figure 4-60. KVG103 – Overall System Efficiency – Day 2 Testing
Thurs., Nov. 4, 2004 – Duke Thomaston Station**

4.12 PRESSURE TRACES

The ICHP values are derived from cylinder pressure traces. Figures 4-61 and 4-62 show typical traces under different cylinder loading conditions. The red traces correspond to cylinder head ends, and the green traces to cylinder crank ends. Examination of Figure 4-61 shows five of the six ends following closely parallel compression and re-expansion lines. In addition, one head end follows a shallower slope on compression and re-expansion because its head end unloader pocket is open. As a result, this end consumes less power than the other ends and provides less flow. Opening and closing the head and crank end pockets on three compressor cylinders provides this unit its flexibility to adjust load when suction and discharge pressure change, so it can operate as close as possible to 100% load without exceeding this limit. While capacity is directly affected by clearance volume changes, the primary purpose of changing clearance is to maintain loading as close to 100% as possible, because, as shown, the engine's heat rate increases significantly as load is reduced.

**Figure 4-61. KVG103 – Compressor Cylinder Traces – One Head End Pocket Open
Wed., Nov. 3, 2004 – Duke Thomaston Station**

Load control flexibility is limited by the discreteness of the pockets; each pocket increases or decreases the clearance volume to which it is connected by a predetermined amount. Thus, the highest load for most of the testing of this unit during Days 1 and 2 was 96% with one pocket open—the condition of Figure 4-61. According to station loading charts, closing this pocket would have pushed the total load over 105%, which would jeopardize the unit according to Duke's standards. This limit was a function of pipeline conditions, which, as previously shown, changed very little during these two days.

In Figure 4-62, it can be seen that two of the head end traces follow the more gradual slope, because two head end pockets are now open. The crank ends remain fully loaded. Opening the second pocket achieved the minimum load tested on Day 1 of about 86%.

**Figure 4-62. KVG103 – Compressor Cylinder Traces – Two Head End Pockets Open – Day 1
Wed., Nov. 3, 2004. Duke Thomaston Station**

Figures 4-63 and 4-64 show traces from the power cylinder pressure transducers. All ten power cylinders are shown in both figures. The first of these two figures (4-63) was recorded on Day 1 (Wednesday, November 3) at 9:24, which was under nominal lean operating conditions for the engine. The second of these two figures (4-64) was also recorded on Day 1 at 12:07 when the unit was running rich, with equivalence ratio at about 0.92 or 0.93.

Cylinder 10, the second trace in these figures, is clear—under normal lean operation, its peak pressure is much lower than for any of the other cylinders—170 PSIG compared to about 250 PSIG. Thus, the power contributed by combustion which raises pressure above compression pressure is much less than for the other cylinders.

It is to be noted from the logged data discussed earlier that Cylinder 10 runs noticeably hotter than the other cylinders. This is apparently a consistent feature of this cylinder according to reports by station operators.

Richened operation tends to lift the peak firing pressure of all cylinders from around 250 to about 285 PSIG or higher. The peak pressure of Cylinder 10 in particular is raised from 170 PSIG to 290 PSIG—a 120 PSI increase compared to the 35 PSI increase experienced by the other cylinders.

This behavior of Cylinder 10 is anomalous, because all cylinders should be provided with the same mixture in a carbureted four-stroke engine, whether lean or rich, so all cylinders should be expected to respond similarly to changes in air/fuel ratio. No proven explanation can be

**Figure 4-63. KVG103 – Power Cylinder Traces – Lean Operation – Day 1
Wed., Nov. 3, 2004 – Duke Thomaston Station**

**Figure 4-64. KVG103 – Power Cylinder Traces – Rich Operation – Day 1
Wed., Nov. 3, 2004 – Duke Thomaston Station**

Figure 4-65. KVG Fuel System, Showing Backfire Relief Line

provided, but some possible explanations can be offered. There is a backfire relief pipe attached to the air/fuel manifold in a non-symmetrical manner (see Figure 4-65). This pipe has a closed end under normal conditions. It is about 12 feet long and could cause some unsymmetrical dynamic flow behavior. Cylinder 10 is at the flywheel end of the unit on the side closest to this backfire line. The volume of the manifold is fairly small, which might exaggerate any acoustic response to the pulsations induced by valve action and intermittent cylinder flows.

As a possible alternative explanation, this cylinder might have a weak coil, which would be less apparent under rich than lean conditions, because a rich mixture reduces resistance. This, however, is less likely if this cylinder has always exhibited anomalous temperatures.

A further possibility is that, as result of Cylinder 10's position at one end, there is a maldistribution of residual exhaust gas.

4.13 DAY 3 TESTING FOR PISTON FRICTION

The third day of testing focused on the compressor (including some incomplete but encouraging efforts to measure unit throughput).

On the third day, nozzle pressure transducers were installed on one cylinder. In addition, some tests were performed with the unit suction and discharge lines blocked off which allowed isolation of compressor frictional losses.

Figures 4-66 and 4-67 show pressure traces from Day 3 for compressor Cylinder 1 head and crank end, together with traces from the nozzles and from the lateral line from compressor to header. The first figure has raw data and the second figure has had some corrections applied to the nozzle and cylinder data. Specifically, the data has been “channel corrected” and “toe offset” corrected. Channel correction means correcting for the response of the measurement channel between the transducer and the cylinder or pipe where the pressure of interest occurs. Because pressure changes dynamically and the dynamic variation is coherent with crankshaft rotation (plus some random cycle-to-cycle variation), it excites an acoustic response of the measurement channel and the signal at the transducer normally has a “ringing” superimposed on it, which is simply an artifact of the measurement configuration. This may be clearly seen in Figure 4-66. Past research by SwRI for GMRC (Harris and Beeson, 1990) provides software for correcting this artifact. Applying this software to the traces of Figure 4-66 produces the traces of Figure 4-67, which clearly have less modulation. What modulation is left results more from pulsations or valve dynamics. Toe correction means offsetting the nozzle traces so the suction nozzle trace matches the cylinder trace at bottom dead center, and the discharge nozzle trace matches the cylinder trace at top dead center. This corrects for any small calibration differences between the transducers, and is physically appropriate, because at BDC and TDC there is no flow into or out of the cylinder (provided there are no valve leaks) and the pressure inside should match the pressure outside.

With these corrections applied, the differential area on the PV card between the cylinder trace and the nozzle traces at suction and discharge gives us the work done per cycle in overcoming resistance to getting the flow from cylinder to the nozzle (normally regarded mainly as valve resistance). Hence, it provides the DIP/valve loss values discussed earlier and used in establishing compressor efficiency. The average DIP value used was 5.71% of indicated power. The installation loss for the manifold and lateral piping between the compressor and the headers was calculated as 5.10% of indicated power.

Figure 4-66. Cylinder 1 Performance Data – Raw (Uncorrected)

Figure 4-67. Cylinder 1 Performance Data – Channel Corrected and Toe Offset Corrected

In transmitting engine brake power to the compressor piston face, another loss occurs—from friction in the piston rings and rider bands. The third day’s testing included an effort to quantify this piston friction loss for this unit. It was possible to close valves, which isolated the compressor from the headers. By running the compressor in this condition, there is a small pumping loss in the cylinders, but this is much lower than the usual compression power.

By comparing the rod load monitor and indicated compressor power, a basis exists for assessing the differential power between the two, which is attributable to piston friction. Figure 4-68 shows the cylinder pressure traces under this condition as well as the rod load monitor signal in volts. Clearly, the pressures are very low with a total range of about 4 PSI, but when integrated over a revolution, it indicates a small amount of work is still being done to move gas back and forth through the valves. The rod load monitor signal converted to pounds and appropriately integrated over a cycle yields the work done per cycle “upstream” of the piston and subtracting the two yields the friction work done.

Figure 4-69 shows how this inferred friction HP varies with speed, for the three speeds at which this evaluation was performed (250, 290, and 340 RPM). Accepting the values obtained in the way a regression with speed may be performed, as indicated on Figure 4-69. For 330 RPM, the regression line gives 9.343 HP per cylinder. Figure 4-70 shows the percentage loss this value would represent for different total compressor HP values if applied unchanged to a loaded compressor cylinder. For 100% loading (333 HP/cylinder), the percentage would be 2.8%, which is not out of line with the generally accepted working value of 5% (corresponding to 95% mechanical efficiency).

Figure 4-68. Representative Unloaded Cylinder & RLM Data (250 RPM)

Figure 4-69. Unloaded Frictional Losses as a Function of Speed

Figure 4-70. Unloaded Frictional Loss as Percent of Compressor Cylinder IHP

In fact, some losses remain which this testing and analysis cannot resolve – losses in the crank pin bearings, the crosshead pin bearings, and the crosshead guide bearings. This could contribute the remaining 2.2% of the generally accepted 5% compressor mechanical loss.

Now this should be regarded as a first experiment to address mechanical losses. It has several implicit assumptions (no consideration of load effects on friction, and strong dependence on accuracy of the RLM), but the fact that it yields a reasonable value indicates that the approach is worthy of further pursuit.

4.14 COMPARISON OF LOSSES AND ENERGY FLOW

Figure 4-71 presents a graphical comparison of the inferred no-load piston friction loss, the valve loss, and the installation loss from the above calculations. For this particular unit, the inferred installation losses are the largest, but the total of less than 14% for piston friction, for DIP, and for installation loss is relatively low.

Figure 4-72 looks at the total system in terms of the flow of energy, where the major consumption of fuel energy occurs, and where opportunities exist for reduction in fuel consumption and increase in capacity. It lists efficiencies of each of five progressive steps in the conversion of fuel energy into useful compression energy, and also lists the size of the losses, in HP, as a percentage of the fuel energy input, and as a percentage of the power available at the preceding step. Note the installation losses are a slightly higher percentage (5.41%) of the power available downstream of the valves than the 5.10% of piston face or indicated power discussed earlier.

Figure 4-71. Comparison of Inferred Values for Friction, Valves (DIP) and Installation Losses as a Percentage of Indicated Power

Figure 4-72. Flow of Power from Fuel to Header, Showing Percentage of Fuel Power on Each Column – Losses Stated in HP, % of Total and % of Available at that Point

For this KVG engine, the combustion loss (which actually include auxiliary losses) is clearly a major factor in the overall system efficiency. One opportunity for minimizing the significance of these losses would be to ensure loading as close to maximum as possible. Since all head end pockets are currently the same size, it would seem advantageous to replace one or two with different sized pockets, so that a more graduated series of load steps could be defined. It also may be possible to use the head and crank end pockets on Cylinder 1 in combination to

reduce load step size. The discrete step size on the day of testing was about 10%. If this could be cut in half, the average improvement in heat rate would be about 2.5%. Of course the benefits of finer load control may not justify the cost for such a small engine. Accurate knowledge of the load on the engine would also be of benefit by allowing safe loading as close as possible to the limit. Thus, the rod load monitor would have value in ensuring minimum heat rate. Another opportunity would be to make Cylinder 10 perform more consistently with the other cylinders while maintaining lean operation. This cylinder runs so hot for so little output that it must represent a significant loss in fuel energy. Diagnosis of the root cause of Cylinder 10's weak performance under lean operation would either enhance operation on a local basis or provide a generic solution for the fleet (if this problem occurs widely). Reducing this cylinder's temperature should help maintenance costs as well. A further benefit to operation would be to automate control of the air/fuel ratio to replace manual adjustment of the carburetor. This capability was demonstrated on Day 2.

The compressor appears to be relatively efficient based on the data. At the same time, any opportunities for reductions in friction losses, valve losses, and installation losses would be beneficial in reducing the amount of available shaft power that is unproductively consumed, leading to more efficient conversion of fuel energy and to an increase in capacity at maximum load.

4.15 HISTORICAL OPERATIONS AND MAINTENANCE COSTS

Figure 4-73 summarizes some historical maintenance and combined O&M (without fuel) cost data for the KVG model, and for other four strokes in comparison with two strokes and some specific two-stroke models. This data, expressed in annual cost per HP, is from Smalley, Mauney, and Ash [Ref. 1] and is based on data, which is now about ten years old. The data was obtained by building a regression model, using FERC detailed compressor station cost data combined with station-by-station engine inventory data from individual companies. Being a regression model, it has scatter and Figure 4-73 includes standard error information. The data indicates trends rather than hard numbers. These trends are:

Figure 4-73. Historical O&M Costs and Maintenance Only Costs for Two Strokes and Four Strokes as a Class – Also for HBA, GMW, and KVG, as Individual Models

- Four-stroke costs tend to be higher than two-stroke costs.
- KVG costs tend to equal or exceed the costs of four strokes as a class.
- KVG costs for both maintenance and combined O&M tend to be higher than those for the HBA model and for the GMW model (two-stroke engines tested under this program).

One reason for their high maintenance costs per unit power is likely the high temperature at which four-stroke engines run, particularly the KVG, hence, reducing the exhaust temperature of Cylinder 10 should be beneficial from this standpoint.

4.16 CLOSED-LOOP CONTROLS DEMONSTRATION ON KVG103

As discussed at the start of this section, this KVG103 is targeted for catalyst installation to reduce NO_x and HAPS. Achieving peak efficiency and reliability of the catalyst requires precise control of the air/fuel ratio. Commercial systems are available to accomplish this precise control. Based on an offer of assistance by Compliance Controls Corporation to make a demonstration system available to the project and to provide some on-site assistance, it was decided to include a test of this commercially available technology within the test program using the Compliance Controls MEC-L system. This system uses a UEGO sensor in the exhaust stack and allows the flexibility of tuned/optimized control ranging from stoichiometric over a range to lean equivalence ratio. Other available systems, including the Compliance Controls MEC-R system, use an O₂ (lambda) sensor that switches about stoichiometric like the sensor on automobile systems.

The MEC-L system uses the following inputs for its control:

- MAP – manifold absolute Pressure
- MAT – manifold absolute temperature
- UEGO – universal exhaust gas oxygen sensor
- RPM – engine speed
- EGT – exhaust gas temperature

The MEC-L output is a signal to an electronic throttle valve in the fuel supply header, which thereby controls the air/fuel ratio from the carburetor.

Adding closed loop fuel/air ratio control would benefit the KVG operation, whether or not a catalyst was added. As previously discussed, the fuel/air ratio must presently be manually adjusted after any load change based on manifold vacuum with the goal of maintaining a consistent lean operation. With the MEC-L or similar system a specified degree of lean operation could be established and maintained without further manual intervention. Since the manifold vacuum is only an indirect measure, the control would be more consistent and

effective, and could be optimized for different goals, such as emissions control or heat rate control.

Figure 4-74 shows the KVG103 fuel system, with dual Impco 600-D carburetors and the fuel supply header. The fuel supply header and the inlet air lead to both carburetors, where they are mixed to produce the fuel/air mixture, which goes via the fuel/air manifolds, to the individual cylinders. The backfire relief line already discussed is also apparent in Figure 4-74. The section of the fuel header to be replaced with the control valve and bypass is indicated in Figure 4-74.

Figure 4-74. KVG Fuel System

Figure 4-75 shows installation of the piping modification, which enabled the closed loop control on this KVG103. Expert pipeline welders prefabricated a replacement yoke, then cut out a section of the header, and welded this yoke in its place. This yoke provides two fuel paths to the carburetors—one via a 2-inch ball valve, which could be opened or closed, and one via a 75 mm Flo-Tech control valve modulated by the signal from the MEC-L controller. This parallel path was utilized only because a 100 mm Flo-Tech, which would be the correct size to achieve the fuel flow requirement with the available pressure setting, was not available for demonstration testing. During the closed loop test, the ball valve was partially open to allow some fuel flow and allow the control valve to stay within an operable range. At the end of the test the control valve could be removed—allowing continued normal operation through one or both legs of the yoke.

Figure 4-76 shows the MEC-L controller, the sensors (MAP, MAT, RPM, UEGO, and EGT), and the connections to the control module.

Figure 4-77 shows the MEC-L Calibration screen, which enables the desired equivalence ratio to be set as a function of speed (and of MAP, if desired).

75mm FloTech
control valve
w/ parallel 2"
ball valve

Figure 4-75. MEC-L Flo-Tech Control Valve

Figure 4-76. MEC-L Controller and Sensors

Figure 4-77. MEC-L Calibration Screen

Figure 4-78 demonstrates that the controller works as desired. The two “strip chart” records presented here show the variation of O₂ and NO_x during unloading of the unit. In the upper chart, operation is in open loop without manual adjustment. The O₂ drops with reduced load, indicating a richer mixture under reduced load. NO_x stays constant.

Figure 4-78. Simple Test of Closed-Loop

In the lower chart, the closed loop controller has been activated and the ball valve partially closed. When load is reduced O_2 stays constant, implying a constant equivalence ratio, and the NO_x drops, indicating that the controller was operating as desired.

In general the MEC-L system was simple to install, configure, and to operate. It was observed that a larger control valve would be desirable to ensure proper control under all conditions and to eliminate the ball valve in the parallel path. The software limits the maximum opening to 85%, and it would be desirable to keep the opening within the range 30% to 75% for best control resolution. The flow coefficient for the 77 mm Flo-Tech is fairly linear between 25-75% open range, which gives best controllability. It was noted that wide swings in valve opening occurred under steady-state operation of the engine, which implies fuel supply pressure was varying to maintain the target equivalence ratio; tighter control of this pressure would be desirable.

The MEC system provides on-board diagnostics, with 44 fault conditions being monitored. This capability aids in the diagnostics of poor performance, intermittent faults, and identifies failed sensors.

A controller of this type should provide accurate control for more automated and efficient engine operation and for peak catalyst performance. The greatest benefit of automatic air/fuel ratio control is the ability to compensate for continuously varying conditions, such as fuel and ambient conditions. The limited demonstration test of this system could not create these conditions, which would require a more long-term evaluation.

Figure 4-79 presents a normalized chart of CO and NO_x as a function of oxygen percentage in the exhaust stream (which is directly related to equivalence ratio). Such a chart can help guide the specific choice of oxygen concentration to use as a set point for normal operation with a three-way catalyst. It shows that NO_x concentration reaches a peak at a point somewhat lean of stoichiometric, and that CO concentration increases sharply as oxygen drops below two percent. Some level of CO is needed to make the catalyst work effectively.

**Figure 4-79. KVG103 – Normalized Dependence of NO_x and CO on O_2
Days 1 and 2, Wed. & Thurs., Nov. 3-4, 2004**

5. RESULTS AND DISCUSSION: PERFORMANCE COMPARISON FOR MODIFIED GMW10

5.1 OVERVIEW

This section of the ninth quarterly report completes the evaluation of performance changes for a specific GMW10 between tests performed before modification to add high-pressure fuel and a turbocharger and testing performed after these modifications. The seventh and eighth quarterly reports presented detailed results from these tests, performed, respectively, in February and August of 2004 at Williams Station 60 (St. Francisville, Louisiana). However, additional analysis was required to unify and align the data from the two tests. This report section presents the analysis and the resulting performance comparisons.

5.2 COMPARATIVE PERFORMANCE ANALYSIS

During both days of testing at Station 60 in August, the engine torque was generally below 100%. Figures 5-1 and 5-2 document this based on data logged, with Williams's assistance, from their station data system. During the first day's testing, there was an overt effort made to change the torque and changes to load step 10, and then to 19 achieved reduction in torque. However, it was believed that the 96% to 97% level was as high as could be achieved under prevailing pipeline conditions. Review of data from February, when the unmodified unit was tested, showed that the torque was then in the 101% to 102% range. Thus, some further analysis and data re-normalization will be required to align the data sets to a common torque and provide the most meaningful quantitative comparison.

Figure 5-1. Percentage Torque on Day 1; Unit 6; After Modification; Williams Station 60; August 25, 2004

Figure 5-2. Percentage Torque on Day 2; Unit 6; After Modification; Williams Station 60; August 26, 2004

Timing considerations also complicated this comparison. The timing range covered in February was 6° and 8° before top dead center (BTDC), with the “nominal” or “as found” value at 6°. The timing range covered in the August tests was 6°, 8°, and 9° BTDC, with 9° as the “nominal” or “as found” condition. Comparisons need to be made at comparable torques and timing values, and it requires some further re-normalization to align both torque and timing.

Figure 5-3 shows how heat rate depends on torque, with timing held at 9°, based on the day 1 testing. Figure 5-4 shows how heat rate depends on timing with torque held at about 96.5%. The data set for these figures and others discussed here includes all data points of torque and timing tests acquired during the day except a few clearly transient points, which occur during and immediately after a change in conditions. These figures include a statement of the linear regression models, which the data yields, together with R^2 (the coefficient of determination), and the standard error (which expresses the uncertainty in prediction of heat rate based on these regression models). The values indicate that the heat rate variation is well represented by the regression models for torque and timing.

The regression models for heat rate may be combined to account for both torque and timing variation, as follows:

$$\text{Heat Rate} = 8236.4 - 1481.3 * (\text{Fractional Torque}) - 73.2 * (\text{Degrees Advance} - 9) \quad (1)$$

Equation 1 will allow values to be calculated for the same conditions as were tested with the unit in the unmodified condition.

Figure 5-3. GMW10 After Modifications – Heat Rate Versus Torque
Williams Station 60 – Aug. 2004

Figure 5-4. GMW10 After Modifications – Heat Rate Versus Timing
Williams Station 60 – Aug. 2004

Figure 5-5 presents Normalized NO_x concentration as a function of torque, using the same data set as for Figure 5-3. Figure 5-6 presents normalized NO_x concentration as a function of timing. These figures also include linear regression models with values for coefficient of determination and for standard error. Combining these models in a similar manner as for heat rate in Equation 1 yields Equation 2:

$$\text{Normalized NO}_x = 0.6744 * (\text{Fractional Torque}) - 0.3992 + 0.0428 * (\text{Degrees Advance} - 9) \quad (2)$$

Figure 5-5. GMW10 After Modifications – Normalized No_x Versus Torque
Williams Station 60 – Aug. 2004

Figure 5-6. GMW10 After Modifications – Normalized Nox Versus Timing
Williams Station 60 – Aug. 2004

Figures 5-7 and 5-8 present system efficiency as a function of torque and timing. While the variation with timing is linear and monotonic with a high R^2 and small standard error, the variation of system efficiency with torque is neither linear nor monotonic. For this reason, a different approach has been used to obtain system efficiency values at the conditions of interest.

Figure 5-7. GMW10 After Modifications – System Efficiency Versus Torque
Williams Station 60 – Aug. 2004

Figure 5-8. GMW10 After Modifications – System Efficiency Versus Timing
Williams Station 60 – Aug. 2004

We know that:

$$\text{System efficiency} = (2545/\text{Heat Rate}) * 0.95 * \text{Compressor Efficiency} \quad (3)$$

Where it should also be known that heat rate has been calculated as:

$$\text{Heat Rate} = (\text{Fuel Energy Flow Rate in BTU/Hr})/(\text{ICHP}/0.95) \quad (4)$$

So that:

$$\text{System Efficiency} = (2545/\text{Fuel Energy Flow Rate}) * \text{ICHP} * \text{Compressor Efficiency} \quad (5)$$

Thus, in fact the system efficiency values are not influenced by the mechanical efficiency assumption.

Figures 5-9 and 5-10 show the variation of compressor efficiency as a function of torque and timing. Timing has essentially no effect on compressor efficiency as might be expected and the variation with torque is not monotonic. An average value for compressor efficiency of 83.68% has been superimposed on both Figures 5-9 and 5-10 as a horizontal line. The average is based on all the data in the timing and torque tests of day. Its standard deviation is 0.38% in efficiency. Thus, we can calculate system thermal efficiency as given by Equation 3.

Applying Equations 1, 2, and 3 and including data from the unmodified tests, we obtain the values shown in Table 5-1 before and after the modifications.

**Figure 5-9. GMW10 After Modifications – Compressor Thermal Efficiency Versus Torque
Williams Station 60 – Aug. 2004**

**Figure 5-10. GMW10 After Modifications – Compressor Efficiency Versus Timing
Williams Station 60 – Aug. 2004**

Table 5-1. Comparison of Heat Rate (HR), Overall System Thermal Efficiency (SE%), and Normalized NO_x Concentration for a GMW10 Before and After Modifications for High Pressure Fuel and a Turbocharger

Torque %	Timing	CE	Unmodified Heat Rate	Modified Heat Rate	Unmodified Overall System Efficiency %	Modified Overall System Efficiency %	Unmodified Normalized NO _x	Modified Normalized NO _x
102.00%	6	83.68%	7477	6945	27.31%	29.13%	1.818	0.160288
102.00%	6	83.68%	7526	6945	27.11%	29.13%	1.66	0.160288
101.00%	8	83.68%	7418	6814	27.53%	29.69%	1.825	0.239144
96.00%	6	83.68%	7566	7034	26.99%	28.76%	1.395	0.119824

The following table summarizes the percentage changes resulting from the modifications.

Table 5-2. Percentage Changes from Tests Before and After Modification for High Pressure Fuel and a Turbocharger

Torque%	Timing	Change in Heat Rate %	Change in System Efficiency %	Change In Normalized NO _x %
102.00%	6	-7.11%	6.66%	-91.18%
102.00%	6	-7.72%	7.45%	-90.34%
101.00%	8	-8.15%	7.86%	-86.90%
96.00%	6	-7.03%	6.57%	-91.41%

Thus, as an approximate summary of these modifications produce a 7% reduction in heat rate, a 7% increase in system thermal efficiency, and a 90% reduction in NO_x concentration.

6. RESULTS AND DISCUSSION: AIR BALANCE TASKS

The work completed on the Air Balance task this quarter includes the following:

- GMVH engine has been partially disassembled, key dimensions were measured, and a detailed geometric analysis was conducted. Detailed geometry was required for accurate simulation and to investigate variability of compression ratio and port timings among cylinders.
- Three of the six cylinders were removed for flow bench testing. Flow bench testing was required to get accurate port discharge coefficient data for simulation. Testing of high (Cylinder 1L) and low (Cylinder 3L) compression pressure (presumed high and low airflow) cylinders on the same bank was tested to determine the variability and contribution to airflow imbalance. A third, Cylinder 1R, was tested to determine the variability and contribution to airflow imbalance of the different exhaust port shape.
- A TOC review meeting was held at SwRI on January 24, 2005. A follow up presentation was given to the GMRC PSC on January 25, 2005.

Prior engine testing involved operation in both open chamber and pre-chamber configurations. The engine tested over engine speeds of 231, 264, 297, and 330 RPM. Engine load was also varied from 70, 85, 90, and 100 of rated. Within this map of engine speed and load, the air/fuel ratio and spark timing was varied. In analyzing the data, the spread of compression pressures was seen to remain very consistent regardless of the operating condition. This consistency is shown in Figure 6-1, where the pressure at 20 degrees before top dead center (TDC) for all cylinders at all operating conditions tested is plotted versus air manifold pressure.

Figure 6-1. Comparison of Cylinder-to-Cylinder Compression Pressure at All Operating Conditions

In the engine simulation subtask, a significant amount of work has been invested in creating a geometric model and characteristics of combustion and turbocharger for the GMVH-6 engine. The baseline test data has been analyzed for characterizing the combustion and engine performance. Data provided by Cooper on the turbocharger and measurements on the test engine were analyzed to create detailed maps of compressor and turbine performance. Drawings and external geometric measurements on the test engine were compiled to model the complex geometry of manifolds, ports, and cylinders.

Several iterations of the software have been compiled to address the specifics of the GMVH engine. Prior simulations have predicted mass airflow closely, but did not accurately predict the amplitude and phasing of pulsations in the inlet and exhaust manifolds. It was believed that the inaccurate manifold dynamic predictions were due to the use of inaccurate discharge coefficients and inaccurate representations of the complicated cylinder plenum geometry that could not be measured externally. In addition, cylinder drawings provided by Cooper showed an allowable casting variation for the exhaust ports, as depicted in Figure 6-2, that gave increased concern of component variability contributing more to air imbalance than originally believed. Thus, it was decided to initiate a geometric analysis and conduct flow testing of select cylinders to resolve uncertainty.

Figure 6-2. Allowable Exhaust Port Geometry

The test engine was disassembled and a geometric analysis conducted. The geometric analysis involved many detailed measurements on all cylinders to quantify the variation cylinder-to-cylinder in compression ratio, inlet and exhaust port timing, and inlet and exhaust port geometry. A list of the static measurements was provided in Table 2-2. From these measurements, several parameters were calculated. There were also three critical discoveries made after the engine had been disassembled. These discoveries were as follows:

1. The cylinder inlet plenum volume was significantly larger than estimated from drawings.
2. A very large plenum in the engine base was connected to each cylinder's intake. This base plenum was not thought to be connected to the intake system on this engine and, therefore, had not been incorporated into the simulation model.
3. Cylinder 1R had the exhaust port shape No. 1, as depicted in Figure 6-2, and the other cylinders had the No. 3 allowable shape. Cylinder 1R had a casting number of GMVG-9-A and the remaining cylinders had a casting number of GMVA-2-E.

The cylinder inlet volume is a critical parameter to accurate simulation of the fluid dynamics in the engine. Drawings provided by Cooper were of a 1950's vintage and were originally hand drawn. The true geometry of the cylinder inlet air plenum was difficult to distinguish by the dotted lines and a few cross-sectional views. Therefore, initial estimates of the inlet volume were incorrect due to misinterpretation of the drawing. The cylinder inlet volume was measured by filling the inlet with metered water. This required sealing the ports and base openings. A difference was measured between the two different casting number cylinders installed on this engine, with Cylinder 1R having approximately 4.3% smaller volume. The results of measurements on the three removed cylinders are shown in Figure 6-3.

Figure 6-3. Cylinder Inlet Volume Measurements

The base plenum was originally the inlet air path when an earlier version of this engine was piston scavenged. It was thought that the base plenum was sealed off from the intake system through blocking gaskets. The size and shape of this volume was also unknown during initial modeling of the GMVH engine. After discovering that the base plenum had been connected, base drawings were requested from Cooper and additional measurements were made on the engine. The discovery of the base plenum being connected to the inlet system was a very important finding, as accurate simulation would have never been achieved without this feature being incorporated into the model. A simplified CAD model was then generated to calculate the air volume and allow for visual investigation on the proper method to simulate the flow paths. The base plenum volume was estimated at 65,106 in³ (1067 L). The CAD representation is depicted in Figure 6-4.

Figure 6-4. CAD Model of Simplified Base Plenum

In Figure 6-5, the piston stroke as calculated by kinematic equations is depicted. The profiles shown are for the left and right banks of this articulated engine. The articulated geometry is shown in the inset of Figure 6-5 and causes the connecting rod hinge pins to follow different elliptical paths, which lead to the different piston stroke profiles. The piston stroke for each cylinder was measured, and this data is shown in Figures 6-6 and 6-7 for left and right banks, respectively. As seen in these figures, the piston stroke profiles are as expected. The total stroke distances for each cylinder are shown in Figure 6-8, where very small variations were measured.

Figure 6-5. Piston Stroke Profiles from Kinematic Analysis with Articulated Rod Schematic

Figure 6-6. Measured Piston Stroke Profiles Versus Kinematic Analysis – Left Bank

Figure 6-7. Measured Piston Stroke Profiles Versus Kinematic Analysis – Right Bank

Figure 6-8. Measured Piston Stroke Versus Kinematic Analysis

The intake and exhaust port timings are very critical to cylinder breathing. The distance from the top of the cylinder to the top of the port was measured for each port of each cylinder. From these, and several measurements of the piston measurements, the port timings were derived. The measured distances to the cylinder top for all intake ports are compiled and plotted in Figure 6-9. In this figure, the dimensions for the inlet ports on Cylinder 1R deviated from the other cylinders. As mentioned previously, this cylinder had a different casting number and the different inlet port location was likely for a different engine application. The measured distances to the cylinder top for all exhaust ports are compiled and plotted in Figure 6-10. In the exhaust port data, the port locations for Cylinder 1R were in range of the other cylinders. The locations of two ports in Cylinder 3L and one port in Cylinder 2R were found to be higher (smaller distance) by a small amount than specified. There are eight intake ports and five exhaust ports for each cylinder of the GMVH engine.

Converting the port top to cylinder top dimensions in crank angle degree timing required relating the piston stroke profile to these distances. The nominal timings, based on mean drawing dimensions, for the intake and exhaust ports are shown in Figures 6-11 and 6-12, respectively. In these figures, the timings before (closing) and after TDC (opening) are seen to vary between the left and right banks due to the different piston travel profiles from articulation. The timings shown are when the piston edge crosses the top edge of the ports. Since there is clearance between the piston and cylinder, the top ring crossing may be a better indication of true port timing. However, it is traditional to use the piston edge for port timing.

Figure 6-9. Top of Intake Port Edge to Cylinder Top Distances

Figure 6-10. Top of Exhaust Port Edge to Cylinder Top Distances

Figure 6-11. Nominal Intake Port Timings for Left and Right Banks

Figure 6-12. Nominal Exhaust Port Timings for Left and Right Banks

The port timings calculated from measurements for each intake port are shown in Figures 6-13 and 6-14 for opening and closing, respectively. The average port timings are depicted on these graphs along with the individual port timings. In these figures, the lower intake ports for Cylinder 1R gives a deviation in opening and closing timings of approximately 1.5 degrees from the other cylinders on the same bank. These timings are for the crossing of the piston edge, and the timing difference for the top ring crossing is noted on the graphs.

The port timings calculated from measurements for each exhaust port are shown in Figures 6-15 and 6-16 for opening and closing, respectively. The average port timings are depicted on these graphs along with the individual port timings. In these figures, the two higher exhaust ports for Cylinder 3L and one higher port for Cylinder 2R give a deviation in opening and closing timings of approximately 1.0 degree from the other cylinders on the same bank. These timings are also for the crossing of the piston edge, and the timing difference for the top ring crossing is noted on the graphs.

Figure 6-13. Individual and Average Intake Port Opening Timings for Left and Right Banks

Figure 6-14. Individual and Average Intake Port Closing Timings for Left and Right Banks

Figure 6-15. Individual and Average Exhaust Port Opening Timings for Left and Right Banks

Figure 6-16. Individual and Average Exhaust Port Closing Timings for Left and Right Banks

In addition to port timings, the port areas were determined. The heights and widths of each port in each cylinder were measured. The port areas were then calculated from these dimensions and the assumed similar corner radii. Graphs showing the individual port areas are given in Figures 6-17 and 6-18 for intake and exhaust, respectively. Individual intake port areas for Cylinder 1R were all consistently low. These low area values were due to shorter port heights. The areas of the intake ports in the remaining cylinders were within specification given the maximum possible measurement error. The eight intake ports are angled from vertical with the angle differing in four pairs. Thus, the outer pair of ports has larger areas, and with each pair converging to the middle, the areas become smaller. The five exhaust ports are all similar and symmetric. The exhaust port areas for Cylinder 1R were also smaller due to shorter heights, as shown in Figure 6-18. The remaining exhaust port areas in the other cylinders were within specification.

The last critical parameter to derive from measurements is the effective compression ratio. The effective compression ratio is the ratio of total cylinder volume at the time of exhaust port closure to the clearance volume at TDC. The clearance volume was determined by measuring the volumes of the cylinder heads, pre-chambers, piston bowls, and crevice. The crevice volume is the volume bound by the distance between the piston and cylinder, and top ring to piston edge. The piston protrusion into the cylinder head volume was also measured. From these measurements, the clearance volume was derived. The total cylinder volume at exhaust port closing is the clearance volume plus the displaced volume from the effective stroke and bore area. The effective stroke is the portion of the piston stroke from exhaust port closing to TDC.

Figure 6-17. Individual Intake Port Areas

Figure 6-18. Individual Exhaust Port Areas

The resulting clearance volume and effective compression ratio for each cylinder is given in Figure 6-19. The spread in clearance volumes is approximately 2.5% of the mean, which is close to the stack up of potential error in the many measurements utilized to derive these volumes. The effective compression ratios show a spread of 0.27 points between the highest—Cylinder 1L and the lowest—Cylinder 2R. Again, there is a potential measurement error, which was estimated to be ± 0.1 point. Thus, the spread in clearance volume and effective compression ratios are fairly small, and not thought significant enough to be the sole cause of cylinder-to-cylinder spread in compression pressure (i.e., air balance).

After completing the measurements from each cylinder, three cylinders were flow tested. Initially, Cylinders 1L and 3L were to be removed from the engine to be flow tested. However, after finding that Cylinder 1R had the different exhaust port shape and different intake port dimensions, it was removed to be flow tested as well. Flow testing was desired to obtain accurate discharge coefficient data for modeling and secondly to determine the variation between cylinders due to geometry of the ports.

Several measurements on each set of ports (intake or exhaust) for each cylinder were taken at different pressure ratios and port openings. The test stand allowed for setting piston height to set a prescribed opening area and adjustments to the compressed air outlet regulator and control valve allowed for prescribed pressure ratios. The two different size flow nozzles were used for high and low flows and installed in the meter run to measure the airflow. Two nozzle sizes were needed to either keep from choking across the nozzle or keep the delta-pressure in a good measurable range over the flow range tested. Since the air supply was from a

Figure 6-19. Clearance Volumes and Effective Compression Ratios

compressed air storage facility, the time for conducting this testing was longer than planned due to time required to recharge the tanks between runs. The air flow capacity of the system was also insufficient for normal testing at the larger port openings, although a few quick measurements at the lowest pressure ratio were acquired at full opening to ensure the extrapolation at other pressure ratios was accurate. In addition, the extensive test range at mid to low port openings gave further confidence in the extrapolations.

Cylinder 1L was the first tested and extensively mapped. Cylinder 3L was only tested at select points to give necessary indication of flow and discharge coefficient variances and to reduce total test time. Cylinder 1R was tested more extensively than 3L, due to the more significant geometric differences, although a few conditions were eliminated for time reduction. The raw (as-measured) data for cylinder 1L intake are given in Figures 6-20 and 6-21. Figure 6-20 shows the mass flow versus piston location from bottom dead center (BDC) for several pressure ratios. In this figure, the flow at higher piston positions is very low because the piston edge is above the port, but the top ring is below, and flow is through the crevice area. As the piston edge uncovers the ports, the flow then increases linearly with reducing piston location or increasing port opening by the piston edge. The increasing flow with pressure ratio is clearly present. Figure 6-21 shows the mass flow versus plenum pressure, which is the pressure upstream of the ports for several piston positions. The flow is shown in this figure to increase rapidly at first with increasing pressure (pressure ratio) and taper off as pressure increase continues.

Figure 6-20. Cylinder 1L Intake – Mass Flow Versus Piston Travel from BDC

Figure 6-21. Cylinder 1L Intake – Mass Flow Versus Plenum Pressure

The raw (as-measured) data for Cylinder 1L exhaust are given in Figures 6-22 and 6-23. Figure 6-22 shows the mass flow versus piston location from BDC for several pressure ratios. In this figure, the flow at higher piston positions is again very low because the piston edge is above the port, but the top ring is below, and flow is through the crevice area. Fewer points were acquired on the exhaust with the piston edge above the port top since the trend was documented with the limited data taken at these positions. As the piston edge uncovers the ports, the flow then increases linearly with reducing piston location as similarly seen with the intake test data. Again, the increasing flow with pressure ratio is clearly present. Figure 6-23 shows the mass flow versus plenum pressure for several piston positions. The flow is shown in this figure to increase similarly as with the intake tests with increasing pressure.

The geometric analysis showed that there is some variation between ports and between cylinders. Therefore, it is important to view the measured data in terms of actual pressure and area ratios. The area ratio is based on the open area (vertical exposed area at port edge in cylinder). The pressure ratio is the ratio of pressure across the ports (upstream/downstream). A comparison of the intake mass flow versus area ratio is shown in Figure 6-24, for the three tested cylinders at a pressure ratio of 1.05. This figure shows that both Cylinders 3L and 1R have lower mass flow versus area ratio relationships than Cylinder 1L. The Cylinders 1L and 3L shared similar dimensions pertaining to the inlet plenum and ports, so the cause for reduced flow for 3L is unclear at this time. Cylinder 1R has several differences in plenum volume and port areas (height-width relationship), which likely explains its reduced flow.

Figure 6-22. Cylinder 1L Exhaust – Mass Flow Versus Piston Travel from BDC

Figure 6-23. Cylinder 1L Exhaust – Mass Flow Versus Plenum Pressure

Figure 6-24. Comparison of Intake Mass Flow Versus Area Ratio (Pressure Ratio=1.05)

A comparison of the exhaust mass flow versus area ratio is shown in Figure 6-25 for the three tested cylinders at a pressure ratio of 1.05. This figure shows that both Cylinders 1L and 3L have similar flow characteristics. Cylinder 1R has a lower mass flow than Cylinders 1L and 3L. Although cylinder 1R has a lower total port area, the port timings and widths were similar and should flow similarly at part openings. However, the shape of the exhaust ports in Cylinder 1R is different and is likely the reason for its reduced flow. This is more apparent in the discharge coefficient calculations.

The data was further reduced to discharge coefficients versus area ratio and pressure ratio. The discharge coefficient was determined from isentropic flow equations with compensation for the moisture content in the air. The discharge coefficient versus area ratio for Cylinder 1L intake at various pressure ratios is given in Figure 6-26. A decreasing trend of discharge coefficient versus pressure ratio can be seen in this figure. The discharge coefficient versus area ratio for Cylinder 1L exhaust at various pressure ratios is given in Figure 6-27. No trend with pressure ratio is exhibited in the exhaust port data.

Comparisons of the intake discharge coefficients for the tested cylinders at pressure ratios of 1.05 and 1.2 are given in Figures 6-28 and 6-29, respectively. Similar trends can be seen under both pressure conditions, with an increased discharge coefficient for Cylinder 1R and reduced values for Cylinder 3L at low area ratios.

Figure 6-25. Comparison of Exhaust Mass Flow Versus Area Ratio (Pressure Ratio=1.05)

Figure 6-26. Cylinder 1L Intake Discharge Coefficient Versus Area Ratio at Tested Pressure Ratios

Figure 6-27. Cylinder 1L Exhaust Discharge Coefficient Versus Area Ratio at Tested Pressure Ratios

Figure 6-28. Intake Discharge Coefficient Versus Area Ratio (Pressure Ratio=1.05)

Figure 6-29. Intake Discharge Coefficient Versus Area Ratio (Pressure Ratio=1.2)

Comparison of the exhaust discharge coefficients at pressure ratios of 1.05 and 1.2 are given in Figures 6-30 and 6-31, respectively. Similar trends can be seen in both figures, with Cylinders 1L and 3L matching and Cylinder 1R having a more significant reduction in discharge coefficient at low area ratios. This significantly different trend for Cylinder 1R is presumed to be caused by the different shape of the top of the exhaust port runners.

The results from the geometric analysis and flow bench testing have been utilized in a revised engine model that now more accurately simulates the GMVH-6 engine. Initial simulations have been performed with most of this new information but still similar cylinders. This was done to check the model predictions with the large base plenum geometry added and more accurate global values of port geometry and discharge coefficient. Tuning of the model is still in progress; however, results from these initial simulations are very promising and a vast improvement of prior predictions that did not include the major discoveries. Figures 6-32 and 6-33, respectively, show comparisons of the measured and predicted cylinder pressures of Cylinders 1L and 3L, zoomed in on the scavenging region. The setting of the turbocharger model was incorrect in these results, thus the step-change in absolute values due to reduced mass flow. However, this data is shown to illustrate the correct prediction of amplitude and phasing of pulsations and how the model appears to be accurately capturing the deviations between these cylinders.

Based on the promising results of initial modeling with the newly measured data, validation is expected soon and the design process will be able to proceed. The model will then be utilized to investigate the effects on cylinder-to-cylinder air balance from not only manifold design, but also the variation in compression ratio, port timing, and port flow that has been documented.

Figure 6-30. Exhaust Discharge Coefficient Versus Area Ratio (Pressure Ratio=1.05)

Figure 6-31. Exhaust Discharge Coefficient Versus Area Ratio (Pressure Ratio=1.2)

Figure 6-32. Cylinder 1L Measured and Simulated Pressure

Figure 6-33. Cylinder 3L Measured and Simulated Pressure

7. CONCLUSIONS

It is to be noted that the second conclusion below is a correction to the second conclusion in the eighth quarterly report, which was misleading in its statement about the influence of timing on NO_x concentration. The need for this correction was discovered by a diligent reviewer from Williams, after the report was issued,

1. As documented in Section 5, a series of tests on a specific GMW10 at Williams Station 60, before and after modifications to add high-pressure fuel injection and a turbocharger, demonstrated a 7% reduction in heat rate, a 7% increase in thermal efficiency, and a 90% reduction in NO_x concentration; crankshaft strain was also reduced by a small amount.
2. As on other engines, retarding timing was shown to increase heat rate while reducing NO_x on the same GMW10.
3. Tests on the rich burn, naturally aspirated, four-stroke KVG103, (with lean operation maintained by manual fuel adjustment and shown to maintain 0.82 or 0.83 equivalence ratio) showed higher heat rate than on all engines previously tested (in the range 9,200 to 9,400 BTU/BHP-Hr with load close to 100%).
4. Heat rate increased to 9,700 with reduced load, with an approximate normalized rate of change of -0.4.
5. Heat rate reduced with advanced timing, to a low of about 8,700.
6. The manual fuel rate adjustment was necessary to minimize heat rate; heat rate increased when the mixture was richened to exhibit a 0.92 or 0.93 equivalence ratio. Further richening almost to stoichiometric on Day 2 showed heat rate as high as 9,900 BTU/BHP-Hr.
7. One cylinder (Cylinder 10) demonstrated anomalous behavior; its peak firing pressure was greatly below that of all the other cylinders; richening the mixture increased this cylinder's peak firing pressure to bring it in line with the other cylinders. No documentable reason was identified, though some possible explanations have been identified and should be further explored. This cylinder ran notably hotter than all the others.
8. Exhaust temperatures on all cylinders were much higher than for the two-stroke engines previously tested. This may explain why maintenance and O&M costs for this class of engine are higher than for two-stroke engines.
9. The compressor thermal efficiency was measured at about 89%, with 5.71% loss coming from the cylinders and valves, and about 5.1% loss coming from the installation.
10. Closed loop control of fuel rate to maintain an equivalence ratio set point, using a UEGO sensor, was successfully demonstrated. This capability presents an option to replace manual adjustment based on an indirect measurement (manifold vacuum) and will enable

alternatives in optimized operation: (minimum heat rate, minimum emissions without a catalyst, or optimum operation of a catalyst if installed).

11. Comparative evaluation of rod load based power and compressor cylinder indicated power under “No Load” conditions (achieved by blocking compressor flow to and from the headers) yielded data for No Load piston friction as a function of speed, reasonably consistent with generally assumed values. This methodology is encouraging and worthy of further application and refinement to provide more accurate data for compressor mechanical efficiency and the potential to better control and maximize this efficiency.

8. REFERENCES

- [1] Smalley, A. J., Mauney, D. A., and Ash, D. I., (1997) Final Report PR-15-9529, “Compressor Station Maintenance Cost Analysis,” prepared for the Compressor Research Supervisory Committee of PRC International, SwRI Project No. 04-7424.
- [2] McKee, R. J., Smalley, A. J., Bourn, G. D., and Young, K. N., (2003) “Detecting Deterioration of Compression Equipment by Normalizing Measured Performance Relative to Expected Performance,” GMRC Gas Machinery Conference (GMC), Salt Lake City, Utah.
- [3] Harris, R. E., Edlund, C. E., Smalley, A. J., and Weilbacher, G., (2000) “Dynamic Crank Web Strain Measurements for Reciprocating Compressors,” presented at the GMRC Gas Machinery Conference (GMC), Colorado Springs, Colorado.
- [4] Harris, R. E. and Beeson, C. M., (1990) “Channel Resonance Correction for Improved Cylinder Performance and Diagnostic Analyses,” Proceedings, PCRC Fifth Annual Reciprocating Machinery Conference, Nashville, Tennessee.

9. LIST OF ACRONYMS AND ABBREVIATIONS

AGA3	Gas Flow Measurement Standard
BEI	Manufacturer's Trade Name
BHP	Brake Horsepower
BTDC	Before Top Dead Center
BTU	British Thermal Unit
CES	Cooper Energy Services
COV	Coefficient of Variance
CPR	Compression Pressure Ratio
DAS	Data Acquisition System
DOE	U.S. Department of Energy
GMC	Gas Machinery Conference
GMRC	Gas Machinery Research Council
GMVH	Cooper Engine Model
GMVH6	Cooper Engine Model with Six Power Cylinders
GMW10	Cooper Engine Model with Ten Power Cylinders
HBA-6T	Clark Engine Model with Six Power Cylinders
HP	Horsepower
IAC	Industry Advisory Committee
IRV	Instantaneous Rotational Velocity
LHV	Lower Heating Value
NGK	Manufacturer's Trade Name
NO _x	Oxides of Nitrogen
OPT	Optimum Power Technology
PCB	Manufacturer's Trade Name
PFP	Peak-Firing Pressure
PSI	Pounds Per Square Inch
RLM	Rod Load Monitor
RPM	Revolutions Per Minute
SCF	Standard Cubic Feet
SDCM	Strain Data Capture Module
SwRI [®]	Southwest Research Institute [®]
TCF	Trillion Cubic Feet
TDC	Top Dead Center
TLA6	Clark Engine Model with Six Power Cylinders
UEGO	Universal Exhaust Gas Oxygen
V-10	10-Cylinder Engine with V Configuration