Technologies to Enhance Operation of the Existing Natural Gas Compression Infrastructure

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ABSTRACT

This quarterly report documents work performed in Phase I 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 documents the second series of tests performed on a turbocharged HBA-6T engine/compressor. It also presents baseline testing for air balance investigations and initial simulation modeling of the air manifold for a Cooper GMVH6.

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1. INTRODUCTION

This report presents results from a series of tests performed on an HBA-6T integral engine compressor in natural gas transmission service. This is the second series of tests on this engine model and was guided by previous results at this site and at other test sites.

The gas transmission industry operates over 4,000 integral engine compressors, which play a major role in pumping natural gas through the US pipeline system. Although the use of centrifugal compressors in the US 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 (2000 HP) and GMW10 (2500 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 US, 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-hour 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-hour). 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-hr.

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

Figure 1-4 presents a distribution of compressor thermal efficiency for the industry created by 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 1-5 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 1-4. Compressor Thermal Efficiency Histogram based on GMRC Survey

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

Three years ago, the US 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 a part of this program, SwRI is undertaking a project entitled “Technologies to Enhance the Existing Natural Gas Compression

Infrastructure”. The 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*.

To support this objective, SwRI has assembled a suite of instruments for measuring key performance parameters on these engine/compressors, and has integrated the data channels from these instruments into a powerful Data Acquisition System (DAS). On the widely accepted premise that effective improvement depends first on accurate measurement, the project has established the capability to concurrently measure and record a range of parameters (including cylinder pressure, fuel flow, torsional vibration, lateral vibration, crankshaft strain, manifold dynamic pressures, and global equivalence ratio). Using these measurement capabilities, the project has performed tests on a number of integral engine compressors. Data analysis capability has been added to provide operational measures such as efficiency of engine, compressor, and the system they form, cycle-to-cycle variance statistics, cylinder-to-cylinder variation in peak-firing pressure, and measures of crankshaft cumulative damage.

The general approach for the project has been to design a test program which links integrity, capacity, and efficiency to operation and balance; to acquire data by test on operational engine compressors; to analyze data and develop relationships; to apply these relationships and identify methods to optimize capacity and/or efficiency (subject to integrity constraints); and to evaluate optimization methods and benefits.

With this approach and the measurement capability described, the program has already accomplished the following:

- A new method of achieving balanced combustion across the cylinders has been invented and demonstrated.

- The need for improved cylinder-to-cylinder air balance has been documented. An investigation to accomplish this through manifold design has been added as a task to this program.
- The relationship between engine balance and crankshaft integrity has been shown.
- Knock detection capability has been evaluated and demonstrated as a means to allow more efficient engine operation, while protecting engine integrity.
- The use of closed-loop global air fuel ratio control has been identified and is to be evaluated as a means to maintain consistent engine operation as ambient and operating conditions change, subject to knock protection.
- The use of system efficiency to measure the conversion of fuel energy to useful compression of transported gas has been demonstrated and used to create benchmarks for the industry.
- Further benchmarks in component efficiency for engine and compressor have been documented.
- The relationship between system mechanical efficiency for transmission of engine power to compressor power and oil temperature has been quantified for two engines.
- High-pressure fuel injection, in combination with turbo-charging, has been shown to allow leaner operation and to achieve both lower heat rate, lower emission of NO_x, and to improve engine integrity by lowering crankshaft strain for a given compression power.
- Guidelines for achieving the benchmarks have been extracted from the data and its analysis.

- A means of directly monitoring and controlling compressor cylinder load by measurement of rod load variation has been tested and refined during the program. While still needing some final refinements, this technology has the potential to maintain the engine at its most efficient full load condition, while protecting against overload and the associated threat to integrity.

Based on these demonstrations, the project is bringing forward technology options that promise to increase capacity, and reduce operating costs in the following ways:

- Improving efficiency of power transmission, and of the compression process, makes more of the installed engine power available for useful compression.
- Enhancing integrity increases availability and capacity by avoiding forced downtime for maintenance.
- Enhancing integrity also reduces operating costs required for maintenance.
- Improving system efficiency reduces fuel consumption and associated fuel use costs, and makes more of the gas entering the pipeline system available to the end consumer.

Thus, project results are very consistent with DOE's goals of increasing capacity and reducing operating costs.

The tests and analyses have been performed so far on two different two-stroke engine models from two manufacturers: a Cooper GMW10 with three compressor cylinders, and a Dresser-Clark HBA-6T with four compressor cylinders. 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 engine selection was based on detailed quantitative analysis of the engine population using a database prepared for the pipeline industry, which shows both models are in the top six, measured by horsepower installed or by number of units installed. Thus, marked diversity has been achieved in the process of testing two

widely deployed engine models. This quarterly report focuses on results from a second series of tests on an HBA-6T. The results include:

- Demonstration of a closed-loop system to control the turbocharger waste gate and maintain global equivalence ratio set points.
- Refinement and evaluation of Combustion Pressure Ratio (CPR) balancing.
- Demonstration of knock detection as a means to protect against damaging detonation when timing is advanced.
- Demonstration of the first successful self-powered test of the Rod Load Monitor (RLM).
- Demonstration of strong local dependence of heat rate and, hence, the need for the RLM.
- Demonstration of how timing advance influences heat rate, emissions, crankshaft strain.
- Quantified comparison of power values from RLM and cylinder pressure.

In addition, this quarterly report describes work conducted on the task for controlling engine air balance through manifold design. This task is utilizing a laboratory Cooper GMVH-6 engine for detailed measurement of manifold pulsations and individual power cylinder performance. Prior data from the GMVH engine has indicated an air imbalance, similar to that witnessed in the field engines tested on this project. The major subtasks involved with this task are as follows:

- Add additional dynamic manifold pressure sensors and acquire data across the operating range of engine speed and load. Testing is to be performed in both open-chamber and pre-chamber configurations. Open-chamber testing is focused on documenting the baseline engine performance. Pre-chamber testing is focused on acquisition of engine data for computational modeling.
- Computational modeling is to be performed to develop a conceptual design of inlet and/or exhaust manifolds to improve the engine air balance between cylinders. The first step is to configure the computational model to match the detailed geometry of the GMVH engine. The second step is to tune and validate the model with baseline engine data in the pre-chamber configuration. The last step is to utilize the validated model to develop a conceptual

manifold(s) design that should minimize deviations in the trapped air mass between power cylinders.

- A cost-benefit analysis will be performed to determine if the predicted benefits (efficiency, emissions, etc.) of the conceptual manifold(s) design outweigh the potential cost of manufacture and installation.
- If the cost-benefit analysis of the conceptual manifold(s) design is deemed favorable, prototype components will be designed and constructed for evaluation on the GMVH engine.
- The prototype manifold(s) will be installed on the GMVH engine and the baseline test matrix will be repeated to validate the predicted performance.
- A final cost-benefit analysis will be conducted with actual performance measurements and manufacturing cost estimates.

2. EXPERIMENTAL

2.1 SENSORS AND DATA CHANNELS

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 period 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 [2]. 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* – 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.

2.2 LABORATORY GMVH MEASUREMENTS

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.
- *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.

2.3 COMPUTATIONAL MODELING FOR AIR BALANCE INVESTIGATION

The computational modeling for the air balance investigation is to be 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.

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

3. DATA ACQUISITION

3.1 FIELD DATA SYSTEM

Figures 3-1 and 3-2 show photographs of the 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 below. 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 Data Acquisition System (DAS)

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

Figure 3-3. Laboratory GMVH Instrumentation and Control Panel

4. RESULTS AND DISCUSSION

This section of the quarterly report presents results obtained from the second series of tests performed on an HBA-6T at El Paso's Kinder station (Station 823). Test objectives included:

- Feasibility Analysis for Waste Gate/Global Equivalence Ratio Control
- Evaluation of Timing Advance
- Evaluation of a Detonation Detection in Conjunction with Timing Advance
- Refinement and Evaluation of Combustion Pressure Ratio Balancing
- Further Testing of the Rod Load Monitor
- Documenting the Need for a Rod Load Monitor

The following sub-sections present installation photographs; data obtained with permanently installed station instruments; data obtained with instrumentation installed temporarily for the tests; strain data capture module data; heat rate summaries; analysis of heat rate as a function of load; rod load monitor evaluation; knock detection; potential integrity indicators based on instantaneous rotational velocity; display of system and component thermal efficiencies; demonstration and refinement of combustion pressure ratio balancing; and comparison of station BHP to BHP from cylinder pressure.

4.1 INSTALLATION PHOTOGRAPHS

Figure 4-1 shows the four compressor cylinders with suction manifold bottle above them and cabling to the cylinder pressure sensors apparent. Close inspection shows the temperature sensors in the discharge nozzles below the cylinders. Figure 4-2 shows the power cylinders. Close inspection shows the pressure sensors installed. Figure 4-3 is a back view of the unit. It shows a full bed grout – some units at the station including the one previously tested have been regROUTED on chock mounts.

Figure 4-1. Compressor Cylinders; HBA-6T at El Paso Station 823; April 2004

Figure 4-2. Power Cylinders; HBA-6T at El Paso Station 823; April 2004

Figure 4-3. Back View of Engine; HBA-6T at El Paso's Station 823; April 2004

Figure 4-4 shows pressure transducers installed on one compressor cylinder – head and crank ends. Figure 4-5 shows a power cylinder pressure sensor. Figure 4-6 shows the air manifold and the air manifold pressure sensor installed at one end. Figure 4-7 shows the flow measurement installation. Figure 4-8 shows one of the accelerometers on the unit at the flywheel end.

Figure 4-9 shows the encoder which triggers data acquisition synchronized with crankshaft rotation in 512 increments. Figure 4-10 shows the data acquisition system by the test unit.

Figure 4-11 is a composite of four photographs showing the elements of the waste gate control to maintain a global equivalence ratio set point.

Figure 4-4. Compressor Cylinder with Pressure Transducers Installed on Head and Crank Ends; HBA-6T at El Paso's Station 823; April 2004

Figure 4-5. Power Cylinder Pressure Transducer; HBA-6T at El Paso's Station 823; April 2004

Figure 4-6. Air Manifold Pressure Transducer; HBA-6T at El Paso' Station 823; April 2004

Figure 4-7. Fuel Flow Measurement using Emerson Flobes 103 Transmitter; April 2004

Figure 4-8. Frame Vibrations Measurement using Accelerometer; HBA-6T at El Paso's Station 823; April 2004

Figure 4-9. Data Acquisition Encoder; HBA-6T at El Paso's Station 823; April 2004

Figure 4-10. Data Acquisition System and Display; HBA-6T at El Paso's Station 823; April 2004

Figure 4-11. Element of Waste Gate Control for Equivalence Ratio; HBA-6T at El Paso's Station 823; April 2004

4.2 DATA FROM PERMANENTLY INSTALLED STATION INSTRUMENTS

After installation and checkout of instrumentation, two productive days of testing were performed. Much of the data will be presented as pairs of plots as a function of time for day 1 and day 2. Following this, some supplementary analyses will be presented.

Data from most temporarily installed instruments was recorded as usual with the project data acquisition system. In addition, data from the detonation detector and from the rod load monitor was separately recorded. The crank strain data capture module has its own data acquisition system installed on the crankshaft, and this data has been separately analyzed.

Data from the station instruments was manually logged and entered into a spreadsheet. This data will first be presented and discussed below.

Figures 4-12 and 4-13 show the variation of several station temperatures – lube oil outlet temperature, air manifold temperature, gas suction header and gas discharge header temperatures. On both days, the lube oil outlet temperature rises by about 20 degrees to between 155 and 160°F over the first one to two hours as the sump heats up. Once the initial rise is over, this temperature stays relatively constant over the day. On day 1, the other temperatures are remarkably constant. On day 2, the air manifold temperature rises by about 8 degrees during the middle of the day; also, the discharge temperature drops because the ratio across the machine drops between 12:30 and 1:00 PM, as will be seen in subsequent figures.

Figure 4-12. Station Data Temperatures; Kinder Test 2; Day 1 (April 21, 2004)

Figure 4-13. Temperature Deg. F; Day 2; Kinder Tests; April 22, 2004

Figures 4-14 and 4-15 show the variation of suction and discharge pressures for the unit, together with their ratio and unit speed. The suction and discharge pressures vary by small amounts during the day, and this shows up as noticeable changes in compression ratio across the unit. On day 2, the pressure changes are more pronounced. At around 1:00 PM, the ratio drops, and then between 2:30 and 4:00, it steadily rises again.

Figure 4-14. Suction Pressure, Discharge Pressure, Speed during Day 1 Test (April 21, 2004); Kinder Tests

Figure 4-15. Suction Pressure, Discharge Pressure, Speed, and Ratio during Day 2 Test; April 22, 2004; Kinder Tests

Close examination of the speed plots shows the speed changes implemented as part of the test plan. These speed changes will be presented on an expanded scale based on revolution timing by SwRI's data acquisition system in subsequent plots.

Figures 4-16 and 4-17 show the variation of compressor BHP and "fuel curve" (often referred to as load step). The compressor BHP from the station data is the result of a calculation using the suction and discharge conditions and an ideal model corrected for losses. This it is approximate. The load step is adjusted to keep the unit loaded at 100% torque or close thereto. It tends to respond closely to ratio changes and suction temperature; thus, reasonably close correspondence between load step and ratio can be seen by comparing Figures 4-14 to 4-16 and 4-15 to 4-17.

The station BHP wanders a little at fixed speed. It also drops as speed drops around 1:15 and 2 on day 1, and at about 10 and 11 on day 2.

**Figure 4-16. Station Data for Brake HP and Load Step ("Fuel Curve");
Day 1, April 21, 2004; Kinder Station**

**Figure 4-17. Station Data for Brake HP and Load Step ("Fuel Curve");
Day 2; April 22, 2004; Kinder Station**

Figures 4-18 and 4-19 present the compressor cylinder discharge temperature values from the station data. A characteristic common to both days is that the highest temperature rise at any time is 3 to 4 degrees higher than the lowest temperature rise. Compressor cylinder temperature rise when accurately measured is a very meaningful indicator of compressor efficiency – the higher the temperature rise for a given pressure rise, the less efficient the compressor cylinder. Even when the temperatures change with time, they remain in a very similar relationship to each other. The percentage difference between the highest and the lowest is a measure of the added losses in the inefficient cylinder – the indication is that cylinders 2 and 3 incurs about 8 percent higher losses than cylinder 4. The most common contributors to losses are valves which leak when closed, and monitoring cylinder temperature rise is an effective way of identifying when a valve replacement would save energy (and increase capacity). To take full advantage of the information, there is a need for appropriate statistical analysis of the data, and for quantification of the benefits which will result from reducing the implied inefficiency. These benefits can then be compared to the labor, materials, and downtime cost of the maintenance action to replace the valve. For example, 8 percent on two cylinders out of 4 on a nominal 1550 BHP compressor add up to a possible fuel energy penalty of about 12 MCF per day. An alternative way to look at the possible penalty is that the unit has potentially 4 percent more capacity available when torque limited, since that amount of the torque is going to overcome losses.

**Figure 4-18. Compressor Cylinder Temperature Rise in Deg. F;
Day 1; April 21, 2004; Kinder Tests**

**Figure 4-19. Compressor Cylinder Temperature Rise in Deg F;
Day 2; April 22, 2004; Kinder Tests**

The hour-by-hour variation seems very consistent in these figures – all temperatures move together. The driving force behind changes in temperature rise is the change in compression ratio (P_d/P_s). Thus, Figures 4-20 and 4-21 have been prepared; these present temperature rise as a function of compression ratio, and show consistent linear relationships over the ratio range

covered. These relationships could be used as part of a condition monitoring process to detect statistically and economically significant deviations from good condition. The reference from which deterioration can be identified, established as a regression model with all valves in good condition would then be used to detect repeated deviations which are more than (say) three standard deviations from the regression model. Deviations from individual valve regression models can be supplemented by quantification of the range of temperature rise in nominally identical cylinders such as Figures 4-18 and 4-19 show.

**Figure 4-20. Cylinder Temperature Rise as a Function of Ratio
(Series Number = Cylinder Number); Kinder Tests; Day 1; April 21, 2004**

Figure 4-22 plots average cylinder discharge temperature against discharge header temperature. The two track very closely with normal small scatter. Thus, little heat (at most fractions of a degree) is lost between cylinders and header. If flow could be measured accurately, the temperature rise and pressure rise from suction to discharge would give enthalpy rise, and provide, with flow, an alternative basis for assessment of system performance.

Figure 4-21. Cylinder Temperature Rise as a Function of Ratio
(Series Number = Cylinder Number); Kinder Tests; Day 2; April 22, 2004

Figure 4-22. Average Cylinder Td versus Header Td; Days 1 and 2

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Figure 4-24. Power Cylinder Exhaust Temperatures; Kinder Tests; Day 2; April 22, 2004

4.3 DATA FROM TEMPORARILY INSTALLED TEST INSTRUMENTATION

Figures 4-25 and 4-26 show the speed history for the two days, based on SwRI revolution timing. The speed generally stays close to 300 RPM, except in the periods when it was cut to 290 and 280 to test the waste gate controllability for maintaining global equivalence ratio.

Figure 4-25. Unit Speed; Kinder Second Test; Day 1; April 21, 2004

Figure 4-26. Unit Speed; Kinder Second Test; Day 2; April 22, 2004

Figures 4-27 and 4-28 show the global equivalence ratio and normalized NO_x concentration for the two days. On day 1, the equivalence ratio drops from about 0.428 to about 0.403 when speed is reduced to 280 RPM. This is because the global equivalence ratio was on open-loop control – though maintaining air manifold pressure constant. On day 2, the waste gate position was under closed-loop control to keep the global equivalence ratio at a set point. The equivalence ratio was measured by the NGK Universal Exhaust Gas Oxygen Sensor (UEGO) installed in the stack. Matching up Figures 4-26 and 4-28 shows that when speed is changed with global equivalence ratio controlled to a set point, the equivalence ratio stays at its set point. The changes in global equivalence ratio from its set point between 2:00 and 3:00 PM occurred when deliberate efforts were made to over-ride the controller and richen the fuel air mixture to the cylinders during tests of the detonation detector.

**Figure 4-27. Global Equivalence Ratio and Normalized NO_x Concentration;
Kinder Second Test; Day 1; April 21, 2004**

The elements of this control system have been shown in Figure 4-11, including the stack sensor, the waste gate, and actuator arm, the pneumatic controller, and the connections to the station control system. The comparison of Figures 4-27 and 4-28 provide clear evidence of the feasibility of this closed loop control – it is very simple and effective.

**Figure 4-28. Global Equivalence Ratio and Normalized NOx Concentration;
Kinder Second Test; Day 2; April 22, 2004**

Figures 4-29 and 4-30 show fuel flow variation during the two days. This clearly reflects the speed changes, but assessment of fuel flow relative to other parameters will be addressed more closely when discussing the heat rate.

Figure 4-29. Fuel Flow; Kinder Second Test; Day 1; April 21, 2004

Figure 4-30. Fuel Flow; Kinder Second Test; Day 2; April 22, 2004

Figures 4-31 and 4-32 show the variation of engine and compressor indicated horsepower during the two days, together with the ratio of engine to compressor power. Since test data acquisition did not start until some time after the unit started, this data is not available for the starting transient of day 1. However, on day 2, there is an indication of a drop in the ratio of engine to compressor power over the first hour of operation. A similar drop for a more complete cold start was previously observed for another unit at this station – hypothesized at the time as a result of increasing oil temperature.

Figure 4-31. Measured IHP for Engine and Compressor and their Ratio; Kinder Second Test; Day 1; April 21, 2004

Figure 4-32. Measured IHP for Engine, Compressor, and their Ratio; Kinder Second Test; Day 2; April 22, 2004

On both days, there are some changes in power on compressor or engine, which are attributable to test condition changes, and some that appear to be the result of other causes. On day 1 after about 12:30, the changes are directly attributable to speed changes. On day 2, after the speed

variation tests were completed (about 12:00), there is a reduction in compressor power about 1:00 PM, which is not directly matched by a reduction in engine power, and so the ratio of engine to compressor power jumps. The reduction in compressor power seems to correspond to a reduction in ratio across the compressor. Later in the day about 2:35, the engine power drops with only a slight matching drop in compressor power. This is approximately when the fuel valves were opened wide on all power cylinders. Soon after 3:00 PM, the compressor power increased – apparently coinciding with an increase in ratio. This time there is not a matching increase in engine power, and so the ratio of engine power to compressor power drops to its lowest of the test series – about 1.14 or below. There is an indication that the mechanical efficiency of power transmission from power cylinders to compressor cylinders changes with conditions, but the reasons are not clear at this time.

The heat rate for days 1 and 2 is shown in Figures 4-33 and 4-34. On day 1, the heat rate stays quite constant through the speed variation tests at 7950 BTU/HP-hr. Late in the day, efforts were made with the help of a station operator to reduce load on the unit at 300 RPM by adjustments to load step, and investigate heat rate effects, but this was only slightly successful as registered by the station values for percentage torque. The indicated compressor horsepower numbers in Figure 4-31 show there was very little change in horsepower load, and yet the heat rate went up as high as 8100 BTU/HP-hr. during this period, with short spikes downwards. Fortunately, the day 2 results will show a much wider range in load (which occurred without intentional effort) whose influence on heat rate will be discussed subsequently.

The day 2 heat rates were distinctly lower – at 7800 for a good part of the testing, as opposed to 7950. The waste gate control was active from the start of the days testing, and the setting led to a slightly higher equivalence ratio than on the early part of day 1. This might explain the lower heat rate. The 7800-heat rate does not change much during the speed changes with equivalence ratio control.

Figure 4-33. Heat Rate (LHV); Kinder Second Test; Day 1; April 21, 2004

Figure 4-34. Heat Rate (LHV); Kinder Second Test; Day 2; April 22, 2004

Soon after 1:00 on day 2, the heat rate went up, which corresponds to a decrease in compressor power and brings up the point that studying heat rate in isolation of load for this unit is incomplete and can be misleading.

The effect on normalized NO_x of the waste gate control was to raise it by 10 to 15 percent relative the day 1 testing with air manifold pressure control, as previously shown in Figures 4-27 and 4-28. Thus, an explicit advantage of maintaining low NO_x is not yet demonstrated by the approach. Rather, the feasibility has been shown, and so the low cost ability to control of a global equivalence ratio set point for a particular objective exists. Such an objective could be to maintain lower NO_x concentration.

One benefit of the particular equivalence ratio setting is that the combustion stability does not deteriorate when speed is reduced. Figures 4-35 and 4-36 illustrate this – they show standard deviation for the time variation of peak-firing pressure (averaged over the six power cylinders), together with combustion pressure ratio. When speed is dropped in day 1, standard deviation increases from an average value of 60 with spikes to 68, reaching about 67 or 68 on average, with spikes to 78 PSI. On day 2, with global equivalence ratio maintained at a set point, when the speed is dropped (between 10:02 and 12:14), the standard deviation appears to drop slightly from its average value right around 60 to about 57 spiking to 62.

The average ratio of peak-firing pressure to compression pressure (at 20 degrees Before Top Dead Center (BTDC)) is also presented in Figures 4-35 and 4-36. This quantity (CPR) stays between 2.4 and 2.6 on day 1. On day 2, it shows more variation with a low of 2.25, and a high of 2.85. The day 2 highs result from advancing timing, which occurred between 13:43 and 14:23. It also goes up from 2.5 to 2.65 during the speed decrease on day 2, whereas very little change in CPR is seen on day 1 when speed is changed.

Figures 4-37 and 4-38 show the average and instantaneous spread in peak-firing pressure across the power cylinders for days 1 and 2. On day 1, the average spread stays around 50 to 55 PSI. The instantaneous spread starts around 155 PSI, and appears to rise slightly when speed is decreased, to about 175 PSI at 280 RPM. On day 2, the average spread again starts around 50 PSI, and the instantaneous spread is about 155 PSI. These spreads are essentially unchanged when speed changes on day 2 with equivalence ratio control. The effect of CPR balancing between 12:40 and 13:20 is to create a clear reduction in average spread to about 40 PSI, but no obvious reduction in instantaneous spread. At 2:30, when all fuel valves were opened, average

spread jumps to almost 100 PSI, then drops back to between 40 and 50 as CPR balancing was repeated towards the end of testing.

Figure 4-35. CPR and Standard Deviation for Peak-Firing Pressure; Averaged across the Six Power Cylinders; Kinder Second Test; Day 1; April 21, 2004

Figure 4-36. CPR and Standard Deviation for Peak-Firing Pressure; Averaged across the Six Power Cylinders; Kinder Second Test; Day 2; April 22, 2004

Figure 4-37. Average and Instantaneous Spread; Kinder Second Test; Day 1; April 21, 2004

Figure 4-38. Instantaneous and Average Spread; Kinder Second Test; Day 2; April 22, 2004

In Figures 4-39 and 4-40, peak-firing pressure (averaged across the six power cylinders) and average spread normalized with respect to average peak-firing pressure are presented. The peak-firing pressure on day 1 starts around 520 PSI, then during the speed reduction tests (down to

280 RPM) falls slightly to as low as 505 PSI, then returns to 520 when speed is returned to 300 RPM, with slight variation, again, as efforts were made to vary load with changes in load step.

Figure 4-39. Peak-Firing Pressure and Average Spread Normalized by Peak-Firing Pressure; Kinder Second Test; Day 1; April 21, 2004

Figure 4-40. Peak-Firing Pressure and Average Spread Normalized by Peak-Firing Pressure; Kinder Second Test; Day 2, April 22, 2004

On day 2, Figure 4-40 shows that peak-firing pressure increases noticeably when speed is decreased (between 10:00 and 12:00) under equivalence ratio control. The peak-firing pressure clearly rises again at 13:50 as timing is advanced, then drops to around 500 PSI when the fuel valves are all opened wide.

Figures 4-41 and 4-42 show the variation of compression pressure (pressure 20 degrees before TDC) and the angle of peak-firing pressure. During the tests on day 1, both stay rather constant. Compression pressure is around 210 with a range from 204 to 215 (5 percent variation). There is a slight reduction of 2 or 3 PSI in compression pressure when speed is reduced to 290 and 280 RPM. Peak angle averages about 22° with extremes from about 17.5° to 26° . On day 2, Figure 4-42 shows more variation in compression pressure, which lies between 200 and 223 PSI, although its average stays around 210 PSI. The difference between days 1 and 2 is that the air manifold pressure was held as constant as possible on day 1, whereas it was allowed to vary more widely in order to maintain a global equivalence ratio set point on day 2. Varying boost pressure affects the mass of air trapped in the cylinder and, therefore, affects compression pressure. Thus, on day 2 when speed is reduced, compression pressure falls by between 5 and 10 PSI, compared to the 2 or 3 PSI on day 1. There is also a reduction in compression pressure when timing is advanced at about 13:43. The matching drop in peak angle and compression pressure is clear in Figure 4-42. This is likely a result of changes in boost pressure to maintain equivalence ratio as timing is changed. The high values of compression pressure towards the end of day 2 are likely a result of opening up all the fuel valves and efforts to richen the mixture during detonation tests - the amount of air required to maintain equivalence ratio under these conditions also goes up - hence, a higher compression pressure.

Figures 4-43 and 4-44 present and compare the maximum standard deviation with time for days 1 and 2. The levels of standard deviation during much of each day are similar at around 70 PSI. Some difference is seen in the speed variation tests. On day 1 with no equivalence ratio control, the maximum standard deviation goes up to about 78 or 80 PSI - starting around 14:00 with the reduction to 290 RPM, and more strongly at 280 RPM from 14:28 to 15:09.

Figure 4-41. Power Cylinder Pressure 20 Degrees BTDC and Angle of Peak-Firing Pressure; Kinder Second Test; Day 1; April 21, 2004

Figure 4-42. Power Cylinder Pressure 20 Degrees BTDC and Angle of Peak-Firing Pressure; Kinder Second Test.; Day 2; April 22, 2004

**Figure 4-43. Maximum Standard Deviation from All Power Cylinders;
Kinder Second Test; Day 1; April 21, 2004**

**Figure 4-44. Maximum Standard Deviation from All Power Cylinders; Kinder Second Test;
Day 2, April 22, 2004**

On day 2, Figure 4-44 shows maximum standard deviation staying very constant under equivalence ratio control, with a slight reduction below 70 PSI during the speed reduction tests 10:00 to 11:00.

Figures 4-45 and 4-46 show the normalized standard deviation of Combustion Pressure Ratio (CPR) across all cylinders. This is actually the coefficient of variance – obtained by dividing the standard deviation in CPR by the average CPR. On day 1, this starts around 0.09 to 0.1, with a wide range in local values from 0.04 to 0.14. During the speed reduction tests, this normalized deviation goes up quite clearly to 0.14, then returns to 0.1 when speed returns to 300.

On day 2, this COV of CPR starts at a similar level to day 1 and is unaffected by speed changes. Of course, it drops sharply as a result of CPR balancing to 0.025 – indicating first how precise a balance condition can be obtained with CPR balancing. It goes up to about 0.08 with timing increase, then to 0.15 or 0.16 when all fuel valves were opened.

Finally, with rebalancing towards the end of the day, the COV of CPR drops to 0.05 or 0.06.

Figures 4-47 and 4-48 present overall system thermal efficiency. Most of day 1, this overall efficiency is around 27.3 percent. On day 2 with the equivalence ratio control system, thermal efficiency is close to 28 percent - right through the speed variation tests. It increases slightly during the reduced speed tests – as might be expected with lower losses, then drops to 27.8 percent after the speed is returned to 300 RPM. After CPR balancing, there is a very slight increase, and then a significant drop to about 26.6 percent, then a smaller increase. Review of earlier figures, particularly for compressor load, shows a sudden small drop in compressor load at 13:09 exactly when this drop in efficiency occurs. This drop in load is 4 to 5 percent, which does not appear large, but as subsequent analysis will show, engine efficiency, heat rate and, therefore, system efficiency are extremely sensitive to load. Thus, it can be misleading to try and assess how other factors appear to influence heat rate when changes in load may be a major contributor.

Figure 4-45. Normalized Standard Deviation of Combustion Pressure Ratio across All Power Cylinders; Kinder Second Test; Day 1; April 21, 2004

Figure 4-46. Normalized Standard Deviation of Combustion Pressure Ratio Across all Power Cylinders; Kinder Second Test; Day 2; April 22, 2004

Figure 4-47. Overall System Efficiency; Kinder Second Test; Day 1; April 21, 2004

Figure 4-48. Overall System Efficiency; Kinder Second Test; Day 2; April 22, 2004

4.4 DATA FROM THE STRAIN DATA CAPTURE MODULE (SDCM)

Figure 4-49 presents output in microstrain from the Strain Data Capture Module (SDCM) during the first day as a function of minutes from when the SDCM started taking data. The figure shows about 130 minutes of “inactivity” since after the SDCM starts to take data, the crank case cover has to be re-installed, and an operator has to take the time to initiate and follow a start-up procedure and reach steady operation. The start occurs at about 130 minutes and initial strain reaches about 88 microstrain before dropping to about 71 in the first five minutes of operation. Strain rises very slightly during the next 3.5 to 4 hours at which point the speed reduction runs start. Crankshaft strain then drops rapidly as speed is reduced. Figures 4-50, 4-51, and 4-52 present three typical 10-revolution snapshots for 300, 290, and 280 RPM. Each presents the variation of strain over about 2 seconds – showing the presence of both first and second order variation in waveform of the approximately ten revolutions. The peak-to-peak variation in each snapshot reduces from over 70 at 300 RPM to 68 at 290 to 62 at 280 RPM. This result is summarized in Table 4-1, giving peak-to-peak microstrain and percentage reduction as a function of speed. The reduction with speed is dramatic, reaching 17 percent, and must be strongly related to the reduction in inertia forces, since the nominal torque does not change significantly with speed. Inertia forces are proportional to the second power of speed, so with appropriate phasing of the response to inertia forces and gas forces, the peak-to-peak force could be very strongly influenced by a speed squared reduction in one of its components. Of course, there is sometimes some influence of near resonant torsional vibration in the variation of crankshaft strain with speed, which might also influence these observations of reduction in peak-to-peak strain. However, the torsional vibration normally shows up in higher orders, since the first crankshaft torsional natural frequency is usually at or above the seventh order.

Figure 4-49. Crankshaft shown as a Function of Time from Start of SDCM Data Acquisition; Kinder Second Test; Day 1; April 1, 2004

Figure 4-50. Crankshaft Strain Snapshot; 300 RPM; Kinder Test 2; Day 1; April 21, 2004

Figure 4-51. Crankshaft Strain Snapshot; 290 RPM; Kinder Test 2; Day 1; April 21, 2004

Figure 4-52. Crankshaft Strain Snapshot; 280 RPM; Kinder Test 2; Day 1; April 21, 2004

Table 4-1. Crankshaft Strain Summary Table; Kinder Test 2; Day 1

RPM	μSTRAIN (PEAK-PEAK)	REDUCTION
300	75	
290	68	9.3%
280	62	17.3%

Figure 4-53 presents similar output from the SDCM for day 2, as a function of minutes from the start of data acquisition. On day 2, the unit started within 30 minutes of starting the SDCM. The same overshoot is seen in the initial start-up with SDCM output then settling down at 70 microstrain – slightly lower than on day 1. During the speed reduction test, the peak-to-peak strain levels reduced again. Figures 4-54, 4-55, and 4-56, show several representative 2-second strain snapshots for 300 RPM, 290 RPM, and 280 RPM operation with global equivalence ratio control. Table 4-2 shows The levels at 290 and 280 RPM were close to the previous days levels, but the value at 300 RPM was somewhat lower (70 *versus* 75 microstrains). Thus, the percentage reduction with speed is lower (11.4 percent reduction from 300 to 280 RPM, but clearly more than linearly proportional to the speed reduction). Table 4-2 also shows strain values for several test conditions matching up with notations in Figure 4-53. As previously documented at Williams’ Station 60, the advance in timing from 10° to 12° led to higher crankshaft strain. The highest strain occurs with the fuel valves open – 78 microstrains – balancing subsequently coincides with a reduction to 75 microstrain.

Figure 4-53. Crankshaft Strain as a Function of Time from Start of SDCM Data Acquisition; Kinder Second Test; Day 2; April 22, 2004

Figure 4-54. Crankshaft Strain Snapshot; 300 RPM; Kinder Test 2; Day 2; April 22, 2004

Figure 4-55. Crankshaft Strain Snapshot; 290 RPM; Kinder Test 2; Day 2; April 22, 2004

Figure 4-56. Crankshaft Strain Snapshot; 280 RPM; Kinder Test 2; Day 2; April 22, 2004

Table 4-2. Crankshaft Strain Summary Table; Kinder Test 2; Day 2

RPM	μ STRAIN (PEAK-PEAK)	REDUCTION
300	70	
290	66	-5.7%
280	62	-11.4%
End of Knock Testing (Timing – 12 Degrees, Rich Fuel)	75	+6.6%
Fuel Valve Open, Baseline Timing	78	
Balanced, Spread Change = 40 PSI	75	-3.8%

4.5 HEAT RATE SUMMARIES

Figures 4-57 and 4-58 superimpose averaged values for fuel flow and compressor IHP on their variations for day 1. Table 4-3 summarizes the heat rates from the day 1 data set. It shows very little change in heat rate between 300 and 290 RPM - a 0.42 percent reduction in heat rate between 300 and 280 RPM, then 1.4 percent increases in heat rate from the first 300-RPM data set to the second 300-RPM data set. Close examination of the two 300 RPM data points also shows the IHP was lower by 27 HP for the second data set compared to the first. This seems a small difference, but the smallness of the IHP change may mask a very strong sensitivity of heat rate to load. In fact, this data implies that for a 1.7 percent decrease in power, there is a 1.4 percent increase in heat rate; i.e., a negative 0.83 non-dimensional rate of change of heat rate with load. Clearly, these are just two data points and a larger statistical survey would be appropriate, but this value suggests the sensitivity of heat rate to load should be investigated further.

Figure 4-57. Kinder 2; Day 1; Fuel Flow History

Figure 4-58. Kinder 2; Day 1; IHP History

Table 4-3. Heat Rate Calculations; Kinder 2; Day 1

RPM	FUEL	IHP	HR	CHANGE
300	325	1554	7947	
290	312	1491	7951	0.05%
280	303	1455	7913	-0.42%
300	324	1527	8062	1.4%
HR = 0.95*(1000*FF*960/24)/IHP				

Figure 4-59 superimposes averaged values on the day 2 fuel flow chart. Figure 4-60 shows indicated power, and Figure 4-61 shows heat rate values for various test conditions of day 2. Table 4-2 summarizes heat rate calculations based on day 2 tests. The heat rate does not change in this table as speed changes, but there are changes in heat rate as timing is changed. The 2-degree timing change from 10 to 12 degrees corresponds to a 2.2 percent reduction in heat rate, and the subsequent change of timing back to 10 degrees corresponds to a 2.2 percent increase in heat rate. Opening the fuel valves appears to correspond to a reduction in heat rate to 7810, and the result of rebalancing corresponds to a further 0.38 percent reduction in heat rate.

Figure 4-59. Kinder 2; Day 2; Fuel Flow History

Figure 4-60. Kinder 2; Day 2; IHP History

Figure 4-61. Heat Rate Calculations; Kinder 2; Day 2

4.6 ANALYSIS OF HEAT RATE AS A FUNCTION OF LOAD

Figure 4-62 presents an alternate way of looking at the heat rate data – with load as an independent variable. There was quite a substantial variation in load during day 2, and Figure 4-62 shows how heat rate varies during the day when presented as a function of the corresponding load. The figure suggests a fairly consistent and near linear relationship between heat rate and load (with negative slope).

The non-dimensional rate of change of heat rate with load is defined as:

$$\left[\frac{\text{Load}}{\text{Heat Rate}} \right] \left[\frac{d(\text{Heat Rate})}{d(\text{Load})} \right]$$

The values for the derivative in the right-hand parenthesis are calculated from the trend line shown on Figure 4-62. The values for load and heat rate in the left-hand parentheses are obtained as approximate averages of the extremes in the difference calculation; i.e., 1550 for HP and 8000 for heat rate.

Figure 4-62. Heat Rate for Day 2 versus ICHP without CPR or Timing Tests or Speed Tests

The result is 0.61 – this means for every percentage reduction in load, the heat rate goes up by 0.61 percent, indicating extreme sensitivity and one reason why such effort is made to keep these units 100 percent loaded.

Figure 4-63 adds the day 1 to the day 2 trends – the added data is reasonably consistent with the day 2 trends.

Figure 4-63. Heat Rate for Days 1 and 2 versus ICHP without CPR or Timing Tests or Speed Tests

Figure 4-64 presents the day 1 heat rate *versus* load chart with advanced timing tests and CPR balancing data added. This presentation provides a much clearer basis for evaluating the benefits of tests such as this on a unit with high sensitivity of heat rate to load. At least during the more highly loaded tests, there is indication of benefit to CPR balancing of about 100 BTU/HP-hr. – that is about 1.25 percent. As compressor load suddenly dropped after the CPR balancing, there is a less distinctive difference between the CPR balancing data and the rest of the data set.

Figure 4-64. Heat Rate *versus* ICHP; Series 2 Adds in the CPR Balancing Data, and Series 3 Adds in the Advanced Timing Data; Kinder Second Test; Day 2; April 22, 2004

Figure 4-65 presents heat rate *versus* load during the reduced speed tests for days 1 and 2. It shows that the heat rate *versus* load line is reduced by reducing speed – presumably as a result of reduced friction losses. It also shows that the heat rate ranges for the tests with equivalence ratio control were lower, but also that the corresponding loads were higher – thus, the benefits of reduced heat rate are not so clear, since a contributor to lower heat rate was clearly increased load. Even so, the slope of the lines joining common speed pairs appears steeper than the trend line of Figure 4-61, so some benefit to the equivalence ratio control probably remains – just less than on the basis of heat rate differences alone.

Figure 4-65. Heat Rate for 280 and 290 RPM as a Function of Load;
Day 2 and Day 1 (= Series 1 and Series 2)

4.7 ROD LOAD MONITOR EVALUATION

Figure 4-66 shows the Rod Load Monitor (RLM) installation for these tests. This was the first successful operation of the RLM with both telemetry of data and self-powering on a low speed engine/compressor. It employed a much longer magnet array for generating power in the moving coils. Previous versions did not cover enough of the stroke (17 inches on this unit) and had too much clearance between the moving coils and the stationary magnets. A rotational test rig at SwRI has been used to calibrate software for predicting power as a function of speed. This software has to correct for the time-varying velocity of relative motion in reciprocating motion. The software was used to size the magnet array and the clearance, and to predict the speed at which the RLM would switch on and self power; this occurred at just over 200 RPM, as predicted.

Figure 4-67 shows the comparison of RLM inferred power values in comparison to power values from the compressor cylinders. These show clearly a significant difference – about 12 percent. This difference is a calibration issue which needs to be addressed, since the determination of power from the RLM involves an assumed value for Young’s modulus of the material.

Figure 4-66. First Successful Self-Powered RLM Installation; El Paso Station 823; Test 2

Figure 4-67. Raw Data (Kinder 2) Comparison; Cylinder 4; Rod Load and Pressure-Based HP

A straightforward approach to calibration is to use a calibrated pressure transducer to determine cylinder power, then to adjust the RLM calibration factor to best equate the two. This would need to be done only at one or two conditions, and then would yield a reliable power and torque measurement system.

Figure 4-68 shows the results of calibration based on average power from RLM and cylinder pressure measurement. Clearly, the relationship between RLM and cylinder power remains consistent over the speed range, within 1 to 4 HP (less than 1 percent).

Figure 4-68. “Scale Factor”; Adjusted Comparison; ICHP versus RLM (Test 4)

We have learned from the Industry Advisory Committee (IAC) for this project that some companies have devoted substantial effort to horsepower determination over a range of conditions for each unit they operate. The result is an accurate basis for interpolating power based on operating conditions, but the mapping effort is extensive. The calibration of the RLM

for a few points greatly reduces this effort and yields a device which will cover any conditions including valve leaks.

The value of an RLM system will be several fold – the previous section of this report has emphasized how strongly heat rate depends on load, which makes clear how important it is to know load accurately. As regulators require increased documentation of emissions, and their dependence on load, it becomes increasingly important to be able to state the load with confidence. In addition, measuring system efficiency and isolating the contributors to efficiency losses will become more important to support pipeline economics and performance improvement. The RLM will be a contributor to all these evaluations.

4.8 KNOCK DETECTION

During this test series at Kinder, a knock detection sensor was evaluated. This sensor was on loan to SwRI from Metrix, who manufactures it as a product. It had previously been evaluated for the first time on this project on a GMW10 at Williams' Station 60. In that case, the sensor successfully responded to a condition under which audible detonation occurred. The sensor voltage increased from about 2 volts to 5 volts under this condition. For reference purposes, Figure 4-69 reproduces the figure from the previous quarterly report.

For this series of tests, a more refined evaluation was made of the sensor output as conditions likely to lead to knock were aggravated, but before audible knock occurred. Specifically, timing was advanced and the equivalence ratio was over-ridden to yield a richer mixture to all cylinders of the engine, and then the fuel valve for the single cylinder with the knock detector was opened to further richen the mixture locally. Severe audible knock did not occur - though there was an occasional knock observed. Figure 6-70 shows the sensor output gradually increased during this test - reaching just short of 3 volts output – suggesting that a threshold somewhere between 3 and 5 volts would probably be appropriate for use in a protective system coupled with timing advance to enhance fuel efficiency.

Figure 4-69. Knock Detector Output; Williams Station 60

Figure 4-70. Knock Detector Output; El Paso Station 823

4.9 POTENTIAL INTEGRITY INDICATORS BASED ON INSTANTANEOUS ROTATIONAL VELOCITY

During the data analysis, an effort was made to relate combustion events to characteristics of the Instantaneous Rotational Velocity (IRV). A series of “long records” were captured during the second Kinder test - providing a complete history of all measured parameters over 20 revolutions. Figure 4-71 shows power from the six cylinders over 20 successive revolutions, with a clear misfire event in revolution 4, for cylinder 4. Figure 4-72 shows power for the compressor cylinders over the corresponding 20 revolutions. Figure 4-73 compares the total engine power and compressor power over the same 20 – the drop in total power from the misfire is clear. Figure 4-74 shows the crankshaft instantaneous rotational velocity over the same 20 revolutions. IRV on the fourth revolution clearly has a totally different characteristic from the other revolutions, with a substantial momentary slow down. This result provides encouraging evidence that pronounced, momentary, loss in power and torque from the engine would be detectable from the behavior of the crankshaft angular velocity as measured at the encoder. Figure 4-75 shows how crest factor (ratio of maximum to mean) for engine power and for IRV over the second long record correlate with each other. The difference in crest factor between records is not as pronounced as the IRV variation, but if it provides enough discrimination, calculating crest factor involves less intensive data analysis in an integrity monitoring method than tracking the characteristics of individual revolutions.

Figure 4-71. Engine Cylinder Power Over 20 Revolutions

Figure 4-72. Compressor Cylinder Power Over 20 Revolutions

Figure 4-73. Combined, Total Engine, and Compressor Power Over 20 Revolutions

Figure 4-74. Instantaneous Rotational Velocity, as a Function of Crank Angle Over 20 Revolutions

Figure 4-75. Deviations in HP and IRV for Seven Records of 20 Revolutions Each

4.10 SYSTEM AND COMPONENT THERMAL EFFICIENCY DISPLAY

Figures 4-76, 4-77, and 4-78 present comparisons of system and component thermal efficiencies for this tested HBA-6T with values for two other others tested under this program, an unmodified GMW10, and a GMW10 with added high-pressure fuel injection and a turbocharger. These figures seek to place each of these component and system efficiencies on a scale ranging from typical low to typical high limits.

This represents the results of further review and illustrative analysis of benchmarks. At present, the variable efficiencies considered are limited to engine heat rate, compressor efficiency, and overall system thermal efficiency, which ranges from 27.54 for the HBA-6T to 30.35 for the modified GMW10. The ranges indicate the potential ranges for improvement of individual components, and the values on these charts are single values nominally representative of the test data for each engine tested. It will be noted that the measured compressor thermal efficiency is a nominal 90 percent for the tests documented in this report.

Figure 4-76. System and Component Thermal Efficiency Comparison; GMW10 with High-Pressure Fuel Injection and Turbocharger

Figure 4-77. System and Component Thermal Efficiency Comparison; Unmodified GMW10

**Figure 4-78. System and Component Thermal Efficiency Comparison;
HBA-6T with Low-Pressure Fuel**

4.11 REFINEMENT AND DEMONSTRATION OF COMBUSTION PRESSURE RATIO BALANCING

This is the third test at which CPR balancing (Utility Patent applied for) has been performed. All tests so far have been manual, guided by data acquired, and displayed by computer. The method depends on having power cylinder pressure acquired digitally as a function of crank angle, so that the maximum value for each cylinder (peak-firing pressure) can be normalized with reference to the compression pressure for that cylinder to yield combustion pressure ratio. The compression pressure is defined for this purpose as the pressure value at some fixed angle before top dead center for the cylinder.

As implemented in tests so far, the combustion pressure ratio for each cylinder is calculated each revolution and averaged over a finite, specified, number of revolutions. After the averaging process is complete, a bar graph is displayed with averaged CPR for each cylinder side-by-side.

A decision is then made to adjust fuel supply down for the cylinder with the highest CPR, and

once this change has been made, the averaging process is then repeated. The process is repeated until all cylinders indicate acceptably similar values for CPR.

During the present tests, adjustments were made to the number of cycles for averaging, and 100 cycles was determined to provide a reasonable balance between consistency of data and speed of implementation; previously 50 revolutions had been used as the basis for averaging.

Figure 4-79 shows the CPR for six cylinders averaged in this way at the start of the CPR balancing process and at the end of the process. In this case, as Figure 4-46 has shown, the standard deviation across the cylinders of CPR was reduced from about 10 percent to about 2.5 percent (a factor of four reduction) in approximately 20 minutes. Heat rate benefits have been discussed, and the data in Figure 4-64, plotting heat rate as a function of load, suggests that as much as 100 BTU/HP-hr. may have been gained by CPR balancing, although completion of the CPR balancing process was followed very shortly by a drop in power of several percent, which increased the heat rate and made the benefits less clearly defined.

Figure 4-79. Combustion Pressure Ratio “Before and After” CPR Balancing (at Kinder Second Test)

Table 4-4. Heat Rate Summaries; Kinder 2; Day 2

CONDITION	HR	CHANGE
300-280 RPM Change	7800	No Significant Change in HR
Just Before Timing Advance	8164	
Timing to 12 Degrees	7980	-2.2%
Timing Back to 10 Degrees	8160	+2.2%
Fuel Valve Open, 40 PSI Increase in Spread	7810	
Balanced	7780	-0.38%

4.12 COMPARISON OF STATION BHP TO BHP FROM CYLINDER PRESSURE

Figures 4-80 and 4-81 compare brake horsepower taken from the station data to brake horsepower calculated from cylinder pressure for days 1 and 2.

Figure 4-80. Comparison of Station BHP to BHP from Cylinder Pressure (using 0.95 Mechanical Efficiency); Kinder Test 2; Day 1

Figure 4-81. Comparison of Station BHP to BHP from Cylinder Pressure (using 0.95 Mechanical Efficiency); Kinder Test 2; Day 2

The station data is inferred from suction and discharge conditions. It comes from an ideal calculation of cylinder power based on suction and discharge pressure and an assumed expansion coefficient for the gas. The ideal calculation is corrected for non-ideal horsepower losses and adjusted from cylinder power to brake power through a mechanical efficiency (understood to be 0.95).

The brake horsepower from cylinder pressure comes from integrating incremental work done (pressure x area x incremental displacement) around the cycle to give work done per cycle. This work done is multiplied by speed to give cylinder power and adjusted from cylinder power to brake power by a mechanical efficiency (set to 0.95).

The difference between the station data and the horsepower from pressure is apparent from Figures 4-80 and 4-81. In general, the pressure-based power is higher than the station power by 50 to 100 HP (3 to 6 percent), and sometimes by more. This implies a risk of overload, but since heat rate reduces with load, would increase the brake thermal efficiency for the engine relative to what would result from 100 percent load based on cylinder pressure. The apparent power

difference in this case and the implied system-wide potential for inaccurate power based on inferential methods re-emphasize the need to enhance consistency in the engine load used for operational decisions. The rod load monitor, when deployed, should achieve this.

4.13 AIR BALANCE

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

- Kick-off meeting with Task Oversight Committee conducted.
- Engine instrumentation package assembled, installed, and calibrated.
- Testing in open chamber format to acquire performance data is completed and data being processed. GMVH engine re-configured with prechambers and testing with prechambers to acquire simulation data is in progress.
- Optimum Technology software has been procured and installed. Training was conducted on June 16-18 at SwRI.
- GMVH dimensional data provided by CES through drawings and communications. Additional details derived from direct measurement on the laboratory engine. Configuration of the model to match the GMVH-6 engine configuration and dimensions has been completed.

A schematic of the model configured to match the GMVH is provided in Figure 4-82. This initial version of the model is complete, except for the turbocharger. Turbocharger maps have been provided by CES and engine data has been analyzed to characterize the turbocharger performance across the operating range. Once initial model validation is completed, the turbocharger model will be added to the simulation to predict engine performance at operating conditions during the manifold design process.

5. CONCLUSIONS

The second test at El Paso's Station 823 on an HBA-6T integral engine compressor load to the following conclusions:

- 5.1 Feasibility has been demonstrated of a low-cost closed-loop system to control the waste gate setting, and thereby maintain a set point for global equivalence ratio. The benefits demonstrated are improved combustion stability during speed reductions. Heat rate was also lower during the closed-loop control tests, although this may have been caused by load changes. While not demonstrated in these tests, this control could be used for other objectives such as lower NO_x.
- 5.2 Strong dependence of heat rate on load has been demonstrated for the engine tested; for every 1 percent reduction in load, the heat rate was shown to increase by about 0.6 percent. This data re-emphasizes the need for accurate knowledge of the compressor load in order to maintain the engine at its most efficient condition, while not causing any overload-knowledge. This is knowledge which the rod load monitor, once implemented, will provide. Data from the station power analysis and from the test data indicate that loading this unit based on station data can be inconsistent by 50 to 100 HP, and may result in overload.
- 5.3 The rod load monitor has been demonstrated for the first time in end-to-end operation, self-powered, with telemetered data on a slow-speed integral unit. The RLM reached a self-powering condition at approximately the speed predicted, so confirming the prediction software.
- 5.4 The rod load power output tracked measured power closely, but requires a calibration factor based on the measured compressor power.

- 5.5 Combustion pressure ratio balancing was successfully demonstrated on this HBA-6T unit. Use of 100 revolutions for CPR averaging was shown to balance desirable speed of balancing and data consistency. The standard deviation of CPR across the cylinders was reduced from 10 percent to 2.5 percent in 20 minutes. There is some indication of 100 BTU/BHP-hr. reduction as a result, but this is not definitive.
- 5.6 The benefits of advancing timing by 2 degrees were demonstrated in terms of approximately 100 BTU-BHP-hr. reduction in heat rate. This reduced heat rate was accompanied by a 6 percent increase in crankshaft strain, and an increase in normalized NO_x concentration by about 20 percent.
- 5.7 Reducing speed consistently reduced heat rate during the tests by about 100 BTU/BHP-hr.
- 5.8 Reducing speed also substantially reduced crankshaft strain by 11 to 17 percent for 300 to 280 RPM reduction.
- 5.9 The sensitivity of cylinder discharge temperature to compression ratio and its potential value as a quantitative indicator of need for maintenance action has been shown by the station data. Statistical analysis of this data and its use in condition monitoring practice could lead to improved thermal efficiency and to increased capacity (as much as 4 percent for the unit tested).
- 5.10 The effectiveness of the knock detection sensor evaluated during these tests has been demonstrated. The data suggests a threshold exists at which non-audible detonation can be detected.

6. 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] 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.

7. LIST OF ACRONYMS AND ABBREVIATIONS

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
GMVH-6	Cooper Engine Model
GMW10	Pipeline Engine Model
HBA-6T	Clark Engine Model
HP	Horsepower
IAC	Industry Advisory Committee
IRV	Instantaneous Rotational Velocity
LHV	Lower Heating Value
NGK	Manufacturer's Trade Name
NO _x	Oxides of Nitrogen
PCB	Manufacturer's Trade Name
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 ®
TDC	Top Dead Center
UEGO	Universal Exhaust Gas Oxygen