TECHNOLOGIES TO ENHANCE THE OPERATION OF EXISTING NATURAL GAS COMPRESSION INFRASTRUCTURE

Quarterly Technical Progress Report

Reporting Period Start Date: 10/01/05

Reporting Period End Date: 12/31/05

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
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January 24, 2006

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ABSTRACT

This quarterly report documents work performed under Tasks 15, 16, and 18 through 23 of the project entitled: *Technologies to Enhance the Operation of 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 presents results of design analysis performed on the TCVC10 engine/compressor installed at Dominion's Groveport Compressor Station to develop options and guide decisions for reducing pulsations and enhancing compressor system efficiency and capacity. The report further presents progress on modifying and testing the laboratory GMVH6 at SwRI for correcting air imbalance.

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1. INTRODUCTION

This quarterly report summarizes results from the design analysis on the compressor manifold of the TCVC10 tested. This design analysis has developed options and guided decisions on modifications to reduce pulsations, enhance thermal efficiency, and increase compressor capacity. The report also presents progress on the installation of changes to a laboratory GMVH6 engine, aimed at controlling air balance and air manifold dynamics.

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 one-half of 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 one-half of 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; therefore, it will 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

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

Figure 1-3 shows how annual fuel consumption at a number of 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-Hr 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], calculated an industry average (ratio of total fuel volume to total BHP-Hr) of 8.25 SCF/BHP-Hr.

**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 piston face HP-Hr 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 included 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 Operation of 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 that reduce *fuel consumption*, increase *capacity*, and enhance mechanical *integrity*.

1.3 PROJECT ACCOMPLISHMENT

This project continues to document and demonstrate the 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. The following documents the project’s 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 the margin between misfire and detonation limits and using commercially available controllers. The crank Strain Data Capture Module (SDCM) applied on all engines tested in the first two phases of the project 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 respond detectably to loss of torque from a misfire. The Rod Load Monitor evaluated and enhanced on every major 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 3% of the natural gas consumed goes toward fuel gas for engines and turbines to drive compressors. This fuel gas would cost over \$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 the addition of a turbocharger on old GMW engines, reduces heat rate by about 7%. 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 previously 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 of the heat rate versus load characteristic has shown value as a graphical method to compare fuel conversion efficiency achieved by different engine technologies or operating decisions. This heat rate versus load comparison has revealed small potential benefits in brake thermal efficiency by applying CPR balancing. Mapping of overall system thermal 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 that favor heat rate may, at the same time, hurt compressor efficiency, so 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 project has prototyped and demonstrated a methodology for use of differential indicated power (DIP) to distinguish valve and installation losses and their contribution to compressor efficiency. Results show DIP based efficiency and enthalpy based efficiency track each other quite closely. The project has also made clear the need for more information about mechanical losses and has added to this knowledge with a 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, and 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 the economic significance of valve leakage.

The project has documented air imbalance between cylinders as a widespread condition that can limit combustion efficiency. New Tasks 15 and 16 are characterizing air imbalance in more detail and are evaluating options for cost-effective solution; a side branch absorber is being built and installed and will be evaluated as an add-on modification to help reduce the contribution of fuel manifold dynamics to air imbalance.

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, thereby, will add to deliverability. Project tests so far have shown a compressor efficiency range from 76 to 91%, adding to an earlier GMRC survey for a larger base of compressors with a range from 52 to 92%! The highest compressor efficiency values found present a benchmark that will add greatly to system capacity, if more widely achieved. Current efforts under the project 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 and recent survey tests on the project have confirmed this contribution; yet, pulsation control methods, such as acoustic filters and orifices must also take account of associated resistive pressure losses.

1.4 FIELD TEST PROGRAM OVERVIEW

Detailed 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 a 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 these 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. Survey test results now also include the Dresser Clark TCVC10, part of the TCV engine family, which has the highest aggregate installed power of any engine in the pipeline system.

1.5 CURRENT PROJECT EMPHASIS

Observations from the project and from a 1990s GMRC survey (discussed previously) indicate that many low speed engine/compressor units have compressor efficiencies, which could be significantly increased. It is believed that the compressor manifold system and lateral piping between the unit and the headers contribute significantly to low compressor efficiency. On this basis, reducing installation losses (i.e., losses outside the compressor cylinder) represent an opportunity to improve compression efficiency in the U.S. pipeline system and, thereby, to increase system capacity (by reducing energy spent overcoming compressor and piping losses and making this energy available for useful compression work).

For the ongoing and final project phase, SwRI seeks to locate, characterize, and guide appropriate modifications to two slow-speed integral units whose compressor thermal efficiency suggests significant room for improvement (mid-80's or below), with a significant fraction of the losses in the installation piping (so that losses could be reduced by changes to installation or operational practice). The Industry Advisory Committee (IAC) for the project has helped SwRI to identify and characterize two candidate units, and survey site tests have now been performed on an HBA-6 at Duke Energy's Bedford Station and on a TCVC10 at Dominion's Groveport

Station. Each unit tested demonstrates significant room for improvement in overall compression efficiency. Guided by the results, design analysis has been performed for both units to quantify the benefits, which can be achieved by changes in the installed piping outside the compressor. The design analysis has already been performed for the HBA-6, with engineering data and review of options provided by the host company. The host has implemented the planned changes. Further tests will then be performed to quantify improvements resulting from the changes. The design analysis for the TCVC10 has been completed and is presented in this report.

2. EXPERIMENTAL

2.1 OVERVIEW

The majority of this section describes the suite of instruments, which have been used in tests so far for intensive testing of the power and compression sides of integral engine compressors. This description is included in this report for completeness and for reference. Two survey tests have also provided data to guide design analysis, installation changes, and further testing with emphasis on efficiency and capacity of the compressor, its compressor manifold system, and its attached piping.

In the following list of sensors and data channels (Section 2.2), which comprises the full suite used in field tests so far, a pair of asterisks and specific discussion denote those from the full list which make up the much reduced set of sensors and data channels used for the “**survey tests**”.

An additional section (2.3) briefly summarizes changes in the instrumentation suite, which are under discussion for use in further testing to emphasize compressor side performance.

2.2 SENSORS AND DATA CHANNELS FOR FIELD MEASUREMENT

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

- ****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 transducers. 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.

For the **survey site tests** discussed in this report, “roving” pressure transducers are used. Rather than install, calibrate, checkout, and concurrently acquire data from a transducer on every end of every cylinder, data is acquired from one cylinder at a time, and then the set of transducers is removed from that cylinder and re-installed on the next cylinder to be tested. The benefit is a much faster set-up for a screening test; as a penalty for this benefit, the survey site data does not provide concurrency and longer term concurrent trending. Recent enhancement employs a heater on each sensor, which keeps the sensor at a uniform temperature and helps reduce the uncertainty caused by calibration drift resulting from temperature variation. As discussed below, these heated sensors are also being applied on the suction and discharge nozzles and suction and discharge lateral lines (“laterals”).

- **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. Because they are 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 inlet 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 the Engine Exhaust Manifold* – These measurements are used to determine dynamic variation of pressure in the engine 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 transducers with factory provided calibration; they are water-cooled to reduce uncertainty resulting from temperature influence on the sensor readings. It has not been possible to install these transducers on exhaust manifolds with water jackets.
- *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 because of dynamic load variation. Rather than digitally time the variation in the 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 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 have 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 (TDC) 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 that the mark on the flywheel coincides with the stationary mark.

The same encoder and triggering methodology are used for the **survey site tests** in conjunction with the transducer set installed on each cylinder in turn.

- *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 (Harris, et al., [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 are 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 are 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. Its main drawback is the need for daily download, which can cut into test time; a refinement is under consideration that increases storage and energy capacity by a factor of ten or more.
- *Engine Fuel Flow* used to document overall engine efficiency – This sensor is an Emerson Flobas 103 transmitter that implements the AGA3 flow measurement based on a differential pressure measurement and is factory calibrated with a certificate. It 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.
- ***Pressures and Temperatures in Headers and Laterals (Suction, Discharge)* – These measurements are used for installation efficiency determination. Pressures are measured with Sensotec piezo-restrictive transducers. Permanently installed station sensors have also been used to provide these data at some sites.

For the **survey site tests** (and for several of the full scale tests undertaken), pressure and temperature data in the suction and discharge headers has also been obtained from permanently installed station instruments. The standard station instruments are transmitters without dynamic pressure response capability, but when well calibrated, they provide accurate data on the operating conditions for the tested unit.

To supplement cylinder pressure and station header pressure data, the **survey site test** reported herein has also used dynamic pressure measurement in the unit laterals and in the suction and discharge nozzles. This enables interaction of pressures at these locations and of cylinder power to be evaluated. The heated sensors used for this purpose have been discussed above and will be illustrated in the results and discussion section (Section 4) of this report.

- *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 (two-stroke) or fuel rate valve (four-stroke) 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 that 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 to 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* – This sensor, provided as a loan to the project by Metrix, counts occurrences of dynamic acceleration levels above a threshold, to detect detonation.

2.3 POTENTIAL INSTRUMENT CHANGES FOR COMPRESSOR SIDE TESTING

The following potential changes to the instrument suite make-up are under consideration for the remaining intensive testing in which it is planned to emphasize compressor side performance.

- *Nozzle Dynamic Pressure Measurement* – This has been discussed above in relation to the survey site tests. Knowledge of dynamic pressure variation in the nozzles acquired coherently with dynamic pressure variation in cylinder, laterals, and headers allows for more specific assessment of the time integrated pressure drop across the compressor valves between cylinder and nozzles, and also provides a reference for assessing pressure drop through compressor manifold and lateral piping between nozzles and headers. Effective interpretation of these pressures demands accurate and consistent calibration for all the pressure transducers involved.
- *Compressor Natural Gas Flow Measurement* – This is a very challenging measurement because of flow modulations and local noise, particularly if dynamic variation of flow over a compressor cycle is to be distinguished. If it can be accomplished, the knowledge will help define the influence of operational parameters on compressor capacity and will better define the power loss (flow weighted pressure drop) across sections of system piping.
- *Compressor Suction and Discharge Temperature Measurement* – This measurement is within the existing state of the art. A well-calibrated temperature measurement, coupled with reliable and co-located pressure measurement, with the knowledge of compressed gas composition and accurate thermo-physical properties for the operating conditions, enables deviations from isentropic compression to be

accurately assessed, and the influence of operational and configurational changes on these deviations to be evaluated.

- *Basis for Compressor Mechanical Loss Assessment* – The Rod Load Monitor evaluated at each detailed test undertaken so far has shown its potential for distinguishing the mechanical friction losses incurred by the compressor piston rings and rider bands. While piston friction is not readily amenable to design changes, the knowledge of how operation affects piston friction losses can become significant when operational changes are under consideration for other purposes.

2.4 LABORATORY GMVH MEASUREMENTS FOR AIR BALANCE TASKS

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 2-1.
- *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.

Figure 2-1. Photograph of Dynamic Exhaust Pressure Sensor in Exhaust Plenum

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 NO _x 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 CO ₂ Concentration	Oil Sump Temperature
Exhaust O ₂ 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 were 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 in 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 2-2. The compressed air system featured three 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 2-2. 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)	

Results from recent simulations with the additional measurements incorporated have shown that the actual inlet air temperature, passing through the ports, is significantly hotter than that measured in the inlet manifold. This increased inlet air temperature is due to a portion of the air mass coming from the large plenum in the base, where the air is heated closer to oil temperature. Additional temperature sensors have been installed directly into the air box of two cylinders to validate the model predictions of elevated inlet air temperature.

2.5 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 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 current computational model of the GMVH-6 engine is provided in Figure 2-3. This model now incorporates the base plenum and more precise geometric dimensions derived during engine teardown.

Figure 2-3. Current GMVH Computational Model Schematic

3. DATA ACQUISITION

3.1 FIELD DATA SYSTEM

Figure 3-1 and Figure 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.

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

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

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 DATA ACQUISITION FOR SURVEY SITE TESTS

A PC-based data acquisition system is being used for the survey site tests. This system does not have the extensive channel capacity of the data acquisition system used in detailed testing at sites documented in previous reports. However, it is adequate for the reduced number of channels required for concurrent data acquisition on cylinder head- and crank-end, suction nozzle, discharge nozzle, and suction and discharge laterals (i.e., 6 channels). A transducer “break-out” box is used, which conditions the signal from the pressure transducers, together with an analog to digital (A to D) converter between the break-out box and the computer. Sampling by the A to D card is triggered by pulses from the encoder, which is driven by a quill shaft connected to the crankshaft at the flywheel.

The processing software is identical to that used by the higher capacity system in previous tests. Normally, this software is designed to acquire data at 512 angular subdivisions of 360 degrees of rotation, over 32 successive revolutions of the crankshaft, and to average the 32 values obtained at each of the 512 rotation angles. This averaging or “comb-filtering” process tends to minimize or eliminate random cycle-to-cycle variations and to reinforce persistent characteristics of the pressure variations.

During the first survey site test, the need was identified to characterize systematic variation in the pressure data, which was occurring at a slow frequency (fractions of a Hz). The averaging process, which normally aids the data acquisition process, was found to work against the need for this characterization, and a field modification was made to allow the capture and storage and analysis of individual pressure records, yielding information on how instantaneous power and pulsations were varying.

3.3 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 2.4, 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 3-3. Laboratory GMVH Instrumentation and Control Panel

4. RESULTS AND DISCUSSION: REDESIGN OF TCVC10 PIPING

4.1 SUMMARY OF OBSERVATIONS FROM TCVC10 SURVEY TEST

The survey test on a TCVC10 showed that the unit had an isentropic efficiency of 75.77% at 330 RPM, which increased to 82.89% when speed was lowered to 270 RPM. Based on cylinder pressure data, the total differential indicated power (DIP) as a percentage of total cylinder power was calculated as 22.64% at 330 RPM and 15.77% at 270 RPM. The valve DIP was 10.28% at 330 RPM and 7.9% at 279 RPM. Table 4-1 summarizes these efficiency and loss numbers, with some measured pulsation values, and adds that the loss outside the valves (Installation DIP) is 12.36% at 330 RPM and 7.87% at 270 RPM. The isentropic efficiency at 330 RPM is about three points below the median for the GMRC survey referred to elsewhere, and over 15 percentage points below the 91% benchmark demonstrated both in that survey and in the present study.

Table 4-1. Summary of Efficiency Loss (DIP) and Pulsation Data from TCVC10 Survey Test

	270 RPM	330 RPM
Isentropic Efficiency	82.89%	75.77%
Total DIP	15.77%	22.64%
Valve DIP	7.9%	10.28%
Installation DIP	7.87%	12.36%
Max S-Nozzle Pulsation	3.44%	4.59%
Max S-Lateral Pulsation	1.38%	0.66%
Max D-Nozzle Pulsation	4.47%	5.12%
Max D-Lateral Pulsation	1.28%	1.57%

The pulsations were higher than desirable, but not dangerously so. The highest nozzle pulsation is 5.12% for line pressure and the highest lateral pulsation is 1.57% (both on discharge at 330 RPM). The API 618 limit of three times the ratio would be about 3.9% for nozzle pulsations, and so the highest measured nozzle pulsation is 1.31 times the API 618 limit.

4.2 GENERAL GOALS OF REDESIGN

The goal of the design study addressed in this section of the report is to increase compression isentropic efficiency of this TCVC10 and to keep pulsations under control. Increasing compressor isentropic efficiency will reduce fuel burnt for a given amount of useful compression, and will increase capacity whenever power available for compression is at its limit. This limit may arise because shaft torque is at a maximum for unit integrity, or because at low ratio no load step exists which can fully load the engine.

The losses summarized in Table 4-1 are strongly influenced by speed. Since speed directly influences flow, this implies a strong dependence of losses on flow. Flow dependence is a normal characteristic of “Flow Squared” losses, which occur at orifices, as well as at valves, at expansions, at contractions, and at tees. In addition, the majority of the DIP comes from outside the valves (55% if we calculate the 330 RPM ratio of installation DIP to total DIP). Thus, to locate and remove sources of significant “Flow Squared” losses in the present installation piping should significantly increase the isentropic efficiency.

4.3 DISCUSSION OF EXISTING INSTALLATION DESIGN AND AVAILABLE DATA

Figure 4-1 shows the TCVC10 with its three compressor cylinders, and large, 54-inch diameter, suction bottle. There is no explicitly obvious source of losses apparent in this photograph, but review of Figure 4-2 and Figure 4-3, which present schematics of the suction and discharge bottles (obtained from the station staff), indicates the presence of a number of orifices. There is an orifice at the end of the center (Cylinder 3) nozzle on both suction and discharge sides and at the ends of the “pipe-backs” from the two end cylinders (1 and 5) to the center of the suction and discharge bottles. There is also a line orifice at entrance to the suction bottle and at exit from the discharge bottle, giving a total of 8 orifices.

Figure 4-1. Photograph of TCVC 10 as Tested Showing Compressor Cylinders, Suction Bottle, and Suction Piping

The purpose of piping back each end cylinder to the center of the bottle is solely to minimize the bottle acoustic shaking forces. The existing bottles do not provide acoustic filtering for the attached piping.

In addition to these schematics, some data on existing piping lengths from a previous SwRI study of this station are available for the present study, but these data do not completely define orientation of the piping. Thus, the acoustics of the existing system can be well defined, but the impact on lateral pipe lengths of planned changes has some uncertainty, which will be discussed further in subsequent report sections.

Figure 4-2. Schematic of Existing Suction Bottle, Showing Internals

Figure 4-3. Schematic of Existing Discharge Bottle, Showing Internals

If a compressor manifold and piping system can be configured, which controls the pulsations satisfactorily while removing the orifices, some significant sources of loss will have been removed, and the primary goals of the redesign will have been accomplished: (1) to reduce the expenditure of fuel for a given amount of useful compression; (2) to increase capacity

whenever engine maximum torque or compressor load step range limit the shaft power, which can be delivered to the compressor cylinders; and (3) to keep pulsations under control.

4.4 MODEL AND PREDICTIONS FOR THE EXISTING SYSTEM

Figure 4-4 presents the acoustic model one line diagram for the existing system. This figure also defines the test points—that is, nodes in the system at which data has been plotted and is presented in subsequent figures. These test points are as follows and remain the same for the existing system and for the redesigned system:

1HS, 3HS, 5HS	Locations just outside the suction valves for the three cylinders
2	Suction bottle line connection
4	Suction lateral quarter point
5	Suction lateral midpoint
6	Suction header cap (compressor end)
8	Suction header cap (scrubber end)
BP	Suction side of bypass valve
BV-1	Suction block valve Unit 1
BV-3	Suction block valve Unit 3
1HD, 3HD, 5HD	Locations just outside the discharge valves for the three cylinders
2'	Discharge bottle line connection
4'	Discharge lateral quarter point
5'	Discharge lateral midpoint
6'	Discharge header cap
BP'	Discharge side of bypass valve
BV-1	Discharge block valve Unit 1
BV-3	Discharge block valve Unit 3

Note the three cylinders are numbered 1, 3, 5 (as opposed to 1, 2, 3)

Figure 4-4. Acoustic Model of Existing System

Figure 4-5 shows the passive response at a few locations on the existing system. In the passive response calculation, the flow just outside one of the cylinder valves is varied sinusoidally, with unit amplitude, at a frequency which is swept from a low value to a high value (from 1 Hz to 100 Hz in this case), and the pulsation response at different test points is recorded. The units displayed on the vertical axes are PSI per Lb/Sec as a function of frequency in Hz. The passive response serves to show sensitivity to excitation, potential frequencies of high response, and the degree of damping around such frequencies. In Figure 4-5, the suction source location is 1 HS, with suction response recorded at 1 HS and at the center of the suction lateral line (Node 5); the discharge source location is 1 HD with discharge response recorded at 1 HD and at the center of the discharge lateral line (Node 5').

Figure 4-5. Passive Response of Existing System

Of course, the passive response tends to be strongest at the source locations. The 1 HS response shows a very low frequency resonance with low sensitivity (about 0.2 PSI per Lb/sec) and a substantial resonance at a higher frequency (50 HZ), with a sensitivity of 1 PSI per Lb/Sec. The damped nature of this 50 Hz resonance resulting from the nozzle orifice damping is clear. Using the half power point calculation for amplification factor ($AF = f_n/\Delta f$; where f_n is the resonant (peak) frequency; and Δf is the frequency difference between points above and below f_n , at which the response amplitude is 0.707 times the peak amplitude) gives an amplification factor in 1 HS response at 50 Hz of about 3.5.

For 1 HD response, there is a nozzle resonance at 40 Hz with sensitivity of 1 PSI per Lb/sec, and the damping is stronger than for suction, with an amplification factor calculated from half-power points of about 2.7. There is also a resonant peak near 2.5 Hz with sensitivity of 0.4 PSI per Lb/Sec.

At Node 5, the overall scale of passive response at any frequency is much lower (by a factor of 10) than at 1 HS, and the response near 50 Hz is just over 0.1 PSI per Lb/Sec. The low frequency response at about 2 Hz has a sensitivity of 0.09 PSI per Lb/sec.

At Node 5', there is little evidence of the nozzle resonance; the 2.5 Hz resonance exhibits about 0.17 PSI per Lb/Sec sensitivity.

Table 4-2 compares the measured cylinder performance and the corresponding predictions of the model; the agreement is encouragingly close.

Table 4-2. Comparison of Measured and Predicted Cylinder Performance: Existing System

Location	IHP Measured	IHP Predicted	Valve % DIP Measured	Valve % DIP Predicted
C1	1262.20	1232.00	10.39	8.9
C3	1241.51	1221.70	10.39	8.5
C5	1382.44	1252.30	9.52	9.8
Total	3886.15	3706.00	10.28	9.07

Table 4-3 presents the calculated losses at the cylinder and line orifices. The line orifice loss in Table 4-3 is calculated based on steady flow, whereas the nozzle orifice losses account for the time varying flow through the nozzles.

Table 4-3. Predicted Power Losses (HP) at the Cylinder and Line Orifices

	Suction	Discharge
Cylinder 1 Orifice	56.27	83.07
Cylinder 3 Orifice	32.26	58.64
Cylinder 5 Orifice	78.71	103.24
Line Orifice	39.58	28.44
Individual Totals (S, D)	206.82	273.39
Total Loss for Suction and Discharge		480.21

In summary, this calculation indicates that there is 480 HP being spent in overcoming flow resistance in the eight orifices shown in Figure 4-2 and Figure 4-3. Reducing this power by

eliminating all these orifices should achieve a correspondingly substantial improvement in compressor isentropic efficiency.

4.5 REDESIGN APPROACH

The redesign approach followed is to enhance the filtering of the compressor manifold system, as shown in the bottle schematics of Figure 4-6, and in the one line acoustic diagram of Figure 4-7. For the suction bottle, gas enters the left hand “common” chamber (Chamber 1), which has no cylinders directly connected to it. Gas flows from Chamber 1 via a 10.25-foot long choke tube, 7.625” in diameter, to Chamber 2 from which gas flows to Cylinder 1. Gas also flows from Chamber 1 via a 17-foot long choke tube, 11.062” in diameter, to chamber 3 from which gas flows to both Cylinders 3 and 5.

On the discharge side, the bottle internals are geometrically identical, although flows are reversed from the suction flow description. Thus, Cylinders 3 and 5 deliver discharge gas to Chamber 3 of the discharge bottle, and Cylinder 1 delivers discharge gas to Chamber 2. Gas flows from both Chambers 2 and 3 via separate choke tubes (identical to the suction choke tubes) to the “common” discharge chamber (Chamber 1). Gas flows from the common discharge bottle chamber to the discharge lateral. Thus, flow to and from every cylinder now passes through an acoustic filter. The Helmholtz frequencies associated with the volume-choke-volume acoustic filter configurations were placed between the first and second orders. There are no orifices in the nozzles or in the lines.

To achieve this enhanced filtering, the connection points from both suction and discharge bottles to the laterals are shifted from the center of the bottles to the common chamber at the left hand end. It is, in fact, an essential feature to remove the direct connections between suction line, the discharge line, and the central cylinder seen in the existing bottles.

Figure 4-7 shows a resonator or Side Branch Absorber (SBA) located at the mid-point of the suction lateral to reduce second order pulsation energy. This resonator will only be necessary if the lateral length remains resonant once the piping is modified to accommodate the new bottle design. While existing pipe lengths are known, the piping configuration outside the bottles is not clearly defined at this time, limiting the precision to which the lateral length can be defined with the new bottle design installed. Results for the modified system include the influence of this resonator, and the lateral pipe length has been kept at the “worst” possible value in the studies.

4.6 PASSIVE RESPONSE OF REDESIGNED SYSTEM

Figure 4-8 compares the passive response of the redesigned compressor manifold system (referred to as Mod 4) with that of the existing system. The response for the original system is shown in red in this figure, and a green line represents the redesigned system. The response to 1 HS excitation at location 1 HS shows increased sensitivity to the suction nozzle resonance, by a factor of four, as well as a slight shift upwards in frequency of the resonance. The amplification factor based on half-power points is about 14, that is, higher than for the original system by a factor of four, just as for the peak amplitude. This difference in peak amplitude and in amplification factor results from the removal of the orifices on the suction side, which eliminates associated losses, but also reduces damping.

The response to 1 HD excitation at location 1 HD also shows increased sensitivity to the discharge nozzle resonance, relative to the existing system, by a factor of five. The amplification factor is about 13, also five times the value for the original system. The frequency of the discharge nozzle resonance goes up from 40 Hz to over 50 Hz.

Figure 4-6. Schematic of Redesigned Bottles

Figure 4-7. Acoustic Model of Redesigned Bottles and Piping System

Passive Response

Project Title **IPPS Digital Analysis**

Company **Dominion Transmission**

Project **18.06223.01.306**

Site **Groveport**

Test **Test 1**

Figure 4-8. Passive Response – Comparison of Existing and Redesigned Systems

The passive response at test point 5 (mid-point of the suction lateral) does not show the same increase in sensitivity near 50 Hz—in fact, the maximum sensitivity at this point around 50 Hz is lower for the redesigned system than for the existing system. This is because of the filtering provided by the redesigned suction bottle. Sensitivity below 10 Hz for the redesigned system at test point 5 is higher than for the exiting system by about 60%. This is the result of some response at an acoustic resonance in the suction lateral line.

The passive response at point 5' (mid-point of the discharge lateral) is slightly higher around 50 to 65 Hz than the original system, by about 30%. The sensitivity in the 5 to 10 Hz range is higher at this location than in the original system by almost a factor of 2. Again, this is a result of response at the discharge line length resonance, but actual pulsations will show this discharge response to be of no concern.

4.7 PREDICTED PULSATIONS WITH REDESIGNED SYSTEM

Figure 4-9, Figure 4-10, and Figure 4-11 present the suction side pulsations as a result of actual, phased, flow excitation from all the cylinders at the test points listed previously. These figures again compare the existing system (red) to the redesigned system (green). Pulsations at the suction valve locations (1 HS, 3 HS, 5 HS) remain controlled at low frequency. They exhibit some increase in pulsation level in the 50 to 60 Hz range by a factor of 2 to 3. For all cylinders, predicted peak pulsation at any frequency is below 18 PSI, and based on suction pressures during test would be below 3% of line pressure, thereby meeting the API 618 limit of 3.9% discussed earlier.

Figure 4-9. Comparison of Predicted Pulsations; Existing and Redesigned Systems; Suction Cylinders and Node 2

Piping Pulsation

Project Title **IPPS Digital Analysis**

Company **Dominion Transmission**

Project **18.06223.01.306**

Site **Groveport**

Test **Test 1**

Figure 4-10. Comparison of Predicted Pulsations; Existing and Redesigned Systems; Suction Nodes 4, 5, 6, and Bypass

Piping Pulsation

Project Title **IPPS Digital Analysis**

Company **Dominion Transmission**

Project **18.06223.01.306**

Site **Groveport**

Test **Test 1**

Figure 4-11. Comparison of Predicted Pulsations; Existing and Redesigned Systems; Suction Block Valves 2 and 3, and Node 8

Pulsations in the suction lateral line (Nodes 2, 4, 5) show some tuning, or build-up, at the low end of the speed sweep range, reaching 14 PSI at 9 Hz for the location just outside the suction bottle and 5 PSI at the mid-point of the lateral. These pulsations are excited by the second order of shaft rotation. (These cylinders do not single-act so first order does not have much excitation energy). Higher order pulsations in the suction lateral are low as a result of the enhanced filtering provided by the suction bottle.

At the suction side of the bypass valve, when it is in its normal closed condition, pulsations of about 8 PSI are seen at several frequencies, and at BV-2 the low end of the speed sweep brings pulsations up to about 9 PSI. Peak pulsations at the other block valve and closed ends are also below 9 PSI (all below 1.5% of line pressure).

Figure 4-12, Figure 4-13, and Figure 4-14 present the predicted discharge system pulsations. Outside the discharge valve locations (1 HD, 3 HD, 5 HD) the pulsations approach 20 PSI in the redesigned system in the region of 50 Hz, but are lower than in the existing system in the low frequency range (below 15 PSI). The predicted higher frequency pulsations remain below 2.5% of line pressure and are, therefore, well below the API 618 limit of 3.9%.

Piping Pulsation

Project Title **IPPS Digital Analysis**

Company **Dominion Transmission**

Project **18.06223.01.306**

Site **Groveport**

Test **Test 1**

Figure 4-12. Comparison of Predicted Pulsations; Existing and Redesigned Systems; Discharge Cylinders and Node 2'

Piping Pulsation

Project Title **IPPS Digital Analysis**

Company **Dominion Transmission**

Project **18.06223.01.306**

Site **Groveport**

Test **Test 1**

Figure 4-13. Comparison of Predicted Pulsations; Existing and Redesigned Systems; Discharge Nodes 4', 5', 6', and Bypass

Piping Pulsation

Project Title **IPPS Digital Analysis**

Company **Dominion Transmission**

Project **18.06223.01.306**

Site **Groveport**

Test **Test 1**

Figure 4-14. Comparison of Predicted Pulsations; Existing and Redesigned Systems; Discharge Block Valves 2' and 3'

The discharge lateral pulsations are well controlled over the entire frequency range and are consistently well below the predicted pulsations with the existing system, including at the bypass and block valves. This result confirms no resonator is needed for the discharge side.

Figure 4-15 shows the shaking forces for the suction and discharge bottles. The redesigned system exhibits lower peak forces than the existing system over the entire frequency range. Bottle shaking forces still exceed 5,000 lbs in the 50 to 80 Hz frequency range due to cylinder nozzle and chamber length resonances that remain in the system. However, forces of this magnitude are not unusual for units of this type and can be controlled using mechanical restraint, if necessary.

4.8 SUMMARY OF PULSATIONS FOR REDESIGNED SYSTEM

In summary, all pulsations are well controlled on the discharge side outside the manifold system. The suction filtering generally controls higher frequency pulsations outside the bottles. Nozzle pulsations in the 50 to 65 Hz frequency range are predicted higher than in the existing system for both suction and discharge but satisfactorily meet API criteria. As previously discussed, a resonator or Side Branch Absorber (SBA) has been incorporated in the modified suction piping, and is joined to the suction lateral at its mid-point. The SBA is shown in Figure 4-16. Its purpose is to minimize second order pulsation energy based on the assumption that the lateral length has not changed and is the worst possible length. The influence of this device is most visible at Node 5 where 10 Hz pulsations exhibit a minimum.

Test Test 1

4.9 PREDICTED PERFORMANCE BENEFITS WITH REDESIGNED SYSTEM

Table 4-4 presents the predicted losses in the redesigned system. Here, without any orifices, only the choke tube entrance and exit flows contribute to square law losses. These are based on steady flow calculations, and checks show that dynamic flow contributions at these locations would be small, which justifies the steady flow treatment.

Table 4-4. Predicted Choke Tube HP Losses in Redesigned System

	Suction	Discharge
Cylinder 1 Choke	9.43	6.78
Cylinder 3 & 5 Choke	16.08	11.56
Individual Totals (S, D)	25.51	18.34
Total for Suction and Discharge		43.85

There is clear performance benefit to the replacement of the nozzle and line orifices with a filter system, even with its associated choke losses. The losses of Table 4-4 for the redesigned system show a 90% reduction from the losses of Table 4-3 for the existing system. Thus, considering uncertainties in the analysis, at least 5% and as much as 10% increase in isentropic efficiency can be expected. Whenever the engine torque limit, or the compressor load step limit at low ratio, constrain the amount of useful compression horsepower, which can be delivered to the flowing gas, then this increase in isentropic efficiency will increase the units capacity by 5 to 10%.

4.10 RECOMMENDATION

Based on the preceding results, it is recommended to replace the suction and discharge compressor manifold bottles with the bottle configurations shown in Figure 4-6, to move the connections between these bottles and the lateral piping to the locations indicated in Figure 4-7, and to use no orifices in nozzles or lateral piping.

The required changes to the lateral piping configuration remain to be reviewed with the operating company personnel and the resulting lateral pipe lengths defined. The Side Branch Absorber shown in Figure 4-16 should be included at the mid-point of the suction lateral only if the resulting piping changes result in tuning of the suction lateral piping length near second order.

These recommended changes are expected to achieve an increase in the available compression power of over 400 HP on each unit, which is so modified. This indicates a potential compression horsepower recovery at the station of over 1,200 HP, with acceptable pulsations.

5. RESULTS AND DISCUSSION: CONCEPTUAL MANIFOLD DESIGNS

At the beginning of Air Balance effort, it was anticipated that a new manifold(s) design would be developed to correct dynamic flow problems and balance the airflow and trapped mass between cylinders. This focus changed somewhat after testing, geometric analysis, and simulation. Simulation and geometric analysis results indicated that a large portion of the perceived air imbalance, as deduced from cylinder pressure data, was caused by geometric variation. However, there is still a discrepancy between test data and analysis that has yet to be explained and felt to be due to flow dynamics that could be caused by current manifold designs. It was also thought that different engine models might have worse and more obvious flow dynamic problems than the even firing GMVH-6. Therefore, developing a methodology for analysis and design is valuable even if the solution for the GMVH-6 is nominal.

The GMVH-6 features an even 60-degree firing interval, with each crankshaft throw coupled to a compressor. The exhaust manifold features a large diameter log, or plenum, connected to the cylinders at the bottom through runners. The exhaust manifold plenum features vertical support plates that effectively split the plenum at the runners, and are slightly longer than the width of the exhaust runners with an open volume between the cylinder pairs. Therefore, the exhaust from each cylinder passes through the runners into individual volumes that merge into common volumes between cylinders and at the turbocharger turbine entrance. There are two intake manifolds, one for each bank, with each comprised of a common log with rectangular runners to each cylinder airbox. The airbox in each cylinder is a volume that has connections to the intake runner, base air chest, and intake ports. The inlet air can pass through the airbox into either (or both) the base air chest or ports.

The exhaust ports are the last to close and the last to communicate with the combustion chamber during a cycle. Therefore, dynamics in the exhaust would seem to be more critical to scavenging than the intake side. Initial focus on conceptual manifold designs, therefore, focused on the exhaust. Though later, some of the focus switched to the intake side due to potential negative influences by having the intake manifolds and cylinder airboxes communicating through the base air chest. The resulting matrix of concepts for solution of the potential breathing issues involved a few different viewpoints on the solution. One viewpoint was that the concept should not only improve air balance but also provide improved performance. Another viewpoint was that the influence from one cylinder on another should be removed and each cylinder isolated. Yet, another viewpoint was that if the original manifold designs are fine as they are, then develop retrofit concepts to mitigate the geometric influences. The original viewpoint was to design modifications, or a new concept to mitigate flow imbalance and achieve equal breathing among cylinders. Designing devices to address a particular cylinder's geometric variation was thought to be too cumbersome and not pursued, as the variation would be random, not easily quantified without disassembly for measurement, and change when engine components are replaced. Thus, a series of concepts were developed and are shown in a flowchart in Figure 5-1.

Figure 5-1. Flowchart of Current Conceptual Manifold Designs

5.1 INDIVIDUAL TUNED EXPANSION CHAMBER CONCEPT

The Individual Expansion Chamber concept involves the design and application of tuned expansion chambers for each cylinder. This concept would provide both cylinder isolation and could improve performance. Since this is a turbocharged engine, the individual expansion chambers would need to connect to a common plenum to maintain use of the constant pressure turbocharger. An expansion chamber is designed to first create a suction pulse that aids scavenge airflow, and then reflects a plugging pulse back to the exhaust ports at a time near port closing to supercharge or “pack” fresh charge mass into the cylinder. This causes an increase in trapped mass that is typically utilized for increased power output. An excellent description and schematic of an expansion chamber for a single cylinder is provided by Dr. Gordon Blair in his text, “Design and Simulation of Two Stroke Engines” [5]. This concept is most often applied to two-stroke performance or racing engines in applications, such as motorcycles and snowmobiles. These applications are mostly single or two-cylinder engines. An application specific to gas compression engines was a design produced by Cooper Compression for the AJAXTM engine family [6]. Example photographs of Individual Expansion Chamber manifolds are provided in Figure 5-2.

Figure 5-2. Photographs of Expansion Chambers (Motorcycle Application at Left & AJAX™ Application on Right [7])

The design of an expansion chamber has been simplified by Dr. Blair and the equations from this text were utilized for an initial design of individual expansion chambers for the GMVH-6 engine [5]. Figure 5-3 shows the various sections (lengths and diameters) specific to this design. The results of the initial design showed the overall length (LT) would be 60.2 feet, and the major diameter (d4) would be 16.9 inches.

Figure 5-3. Design Parameters of an Individual Expansion Chamber Manifold

The advantages of an Individual Expansion Chamber manifold for the GMVH-6 engine are as follows:

- Trapped mass increased for either leaner operation (if turbo-limited) or increased power.
- Isolation between cylinders from adverse dynamics caused by combustion instability would be achieved.
- Potential exists for tunable section to compensate for cylinder geometric variability, potentially allowing for balancing of cylinders in terms of trapped mass.
- Basic design is applicable to all two-stroke engines, more so for non-turbocharged engines.

The disadvantages of an Individual Expansion Chamber manifold for the GMVH-6 engine are as follows:

- Common plenum required to connect the turbocharger to all expansion chamber outlets. Packaging would be extremely challenging, likely requiring off-engine turbocharger mounting.
- Size and complexity would create very expensive product.
- A tuned manifold has a very narrow operating band for efficient application. Performance may be worse than the original manifold design at off-rated engine speed and load range.

The disadvantages for this concept outweigh the advantages, specifically in terms of cost and complexity. There was also concern that performance would be significantly degraded at operating conditions off-rated speed and load where the chambers would be tuned for maximum performance. Therefore, detailed design and optimization was not conducted for this concept.

5.2 MULTI-CYLINDER COUPLING TUNED MANIFOLD CONCEPT

The Multi-Cylinder Tuned Manifold concept follows the design for V-6 two-stroke engines typically used for outboard marine applications. This tuned manifold concept still utilizes a reflected pulse to supercharge the cylinder near port closure. The difference from the previous design is that this pulse is derived from another cylinder. The most attractive configuration would be three cylinders feeding into two plenums, which then feed into one junction to the turbocharger. This configuration, called the 3-2-1 arrangement, is depicted in Figure 5-4. This design would conveniently couple the cylinders by right bank and left bank due to the firing order and phasing inherent to the GMVH-6 design.

Figure 5-4. Multi-Cylinder Coupling Tuned Manifold Concept

As mentioned previously, cylinder firing occurs evenly, every 60 crank angle degrees in the GMVH-6. The phasing of blowdown events is shown in Figure 5-5. In this image, the coupling of the left and right bank cylinders can be seen. The design process would, therefore, focus on runner lengths and diameters to achieve the optimum timing of the pulse from one cylinder to the next (i.e., 1L to 3L to 2L to 1L, etc.).

Figure 5-5. Phasing and Coupling of Cylinders on the GMVH-6 Engine

The design process for this concept utilized the Virtual 2-Stroke[®] software and the optimization process was conducted by Optimum Power Technology. The results of this process showed an expected increase in engine BMEP of approximately 4% at rated conditions (330 RPM). The increase in BMEP is due to increased trapped mass and can be viewed as an ability to operate leaner at the same power output. The dimensions for this design are shown in Figure 5-6 for one bank (identical for opposing bank). The EMAN3 pipes from each bank would be coupled for a single EMAN4 pipe to the turbocharger.

Pipe	Length (inch) - EXR's from cyl flange	Diameter (inch)
EXR1	$10+20+48 = 78$	8.6
EXR2	$10+18 = 28$	8.6
EXR3	$10+22 = 32$	7.8
EMAN2	48	8.6
EMAN3	48	8.6
EMAN4	19.7	15.75

Figure 5-6. Dimensions for Optimized Multi-Cylinder Coupled Exhaust Manifold Concept

The advantages of the Multi-Cylinder Coupled exhaust manifold for the GMVH-6 engine are as follows:

- Trapped mass increased for either leaner operation (if turbo-limited) or increased power.
- Smaller size and dimensions than the Individual Expansion Chamber design.
- Potential exists for tunable section to compensate for cylinder variability, allowing for balancing of cylinders in terms of trapped mass.
- Basic design is more suited for turbocharged engines and should be applicable to all two-stroke engines, more so for Vee engines.

The disadvantages of the Multi-Cylinder Coupled exhaust manifold for the GMVH-6 engine are as follows:

- Completely new design would be an expensive product. Would likely require modification to turbocharger location/mounting, and in turn the compressor outlet pipes.

- Since pulse charging depends on a previously firing cylinder, the effect of a misfire in one cylinder would more greatly affect the next cylinder dependent on this pulse.
- Design seems applicable to V-6 engines. Design for in-line engines would be more complex in terms of packaging due to cylinder spacing.
- A tuned manifold has a narrow operating band for efficient application. Performance may be worse at off-rated engine speed and load range.

The disadvantages for this concept appear to outweigh the advantages, specifically in terms of cost and potential performance degradation with unstable combustion seen in most open chamber engines operating lean for low NO_x emissions. There is also concern that performance would be degraded at operating conditions off-rated speed and load.

5.3 EXHAUST SIDE BRANCH ABSORBER (SBA)

As mentioned in the discussion of the tuned manifold concepts, unstable combustion will affect the dynamic pressures and flows in the exhaust manifold, and there is risk with a design that is dependant on these pulses. Therefore, a different approach was considered that would attempt to isolate the cylinders from the gas dynamics. One such approach was to incorporate a Side Branch Absorber (SBA) designed to dampen the specific frequency of pressure pulses caused by cylinder blowdown.

Data measured from the GMVH-6 for Cylinder 1 Left is plotted in Figure 5-7. The in-cylinder, intake runner, and exhaust runner pressures versus crank angle are shown. Noted on this graph are the pulses in this one exhaust runner caused by the blowdown of all the other cylinders. Also seen on the exhaust runner pressure trace are higher frequency pulsations, specifically during the scavenging region.

Figure 5-7. Dynamic Pressure Data Recorded on Cylinder 1L of the GMVH-6 Engine

An analysis on the exhaust dynamics was conducted with the SwRI developed IPPS model, which included derivation of the frequency modes and amplitudes of the measured exhaust pressures. Results of the data analysis are shown in Figure 5-8 for the left bank cylinders. The first mode of the length response of the exhaust chamber is seen in both the IPPS and engine data (near Cylinders 1L and 3L). The IPPS model predicted 64 Hz while the engine data showed 60 to 68 Hz. The second mode of the response is seen at the center of the chamber (near Cylinder 2 left). IPPS predicted 124 Hz, and the engine data shows a response at approximately 132 Hz. These results indicate that an SBA can be used to essentially eliminate the first mode of the response. Elimination of the pressure fluctuations during scavenging would be advantageous to increase scavenging and better isolate cylinders from each other for more consistent scavenging.

Acoustically, an SBA tries to create a velocity maximum at a point where a velocity minimum (pulsation maximum) exists. An SBA alters the acoustics such that the response associated with the frequency to which the SBA is tuned is “split” into two responses. The SBA design developed from IPPS modeling to address the exhaust dynamic pressure measurements is shown in Figure 5-9. The IPPS simulation results with the SBA incorporated are shown in Figure 5-10 for Cylinder 1L.

Figure 5-8. Frequency Analysis of Recorded Engine Data for Left Bank Cylinders

Figure 5-9. Conceptual Exhaust SBA Design for GMVH-6

Figure 5-10. IPPS Model Results for SBA Concept

The advantages of the Exhaust SBA for the GMVH-6 engine are as follows:

- Cost-effective retrofit of existing manifold.
- Concept design should provide better cylinder isolation for more consistent scavenging and combustion.
- Some improvement in scavenging and, therefore, performance expected with pulsation attenuation.
- Can be designed to be tunable at different modes for best efficiency.
- Noise reduction would be expected as a side benefit.

The disadvantages of the Exhaust SBA for the GMVH-6 engine are negligible, if the advantages can be achieved. If the mode changes at off-rated operating conditions, the SBA may become ineffective but will not alter engine performance from the original configuration. A more complex, but tunable design could be developed to address off-rated conditions.

The advantages for this concept make it very attractive. The next step was to conduct detailed analysis and optimization with the engine model. Initial modeling with the engine simulation showed only a small reduction in exhaust pulsations with the SBA. Iterations with SBA dimensions showed slight improvement in pulsation attenuation. It is currently uncertain if the engine model is accurately simulating the SBA. These devices have been designed and utilized in pipeline applications where they have successfully reduced resonant pulsations. SBA's are also used by the automotive industry on inlet air systems, such as in the late 1990's, the Chevrolet pickups with V-8 engines. Therefore, more investigation was desired due to the cost-effective potential and testing of the concept is in progress. A prototype SBA has been designed for easy changing of lengths and diameters and with only minor modifications required on the engine. Demonstration testing will be conducted in the next task of this project.

5.4 INTAKE MANIFOLD MODIFICATIONS

The last concepts considered were modifications to the intake manifold. One of the intake concepts included blocking the base air chest to ensure all inlet air is derived from the intake manifold and then modifying the intake manifold volume to dampen the resulting pulsations that will occur. The second intake concept is a modification to the plenum entry where the manifold is connected to the aftercooler.

The base air chest is a legacy design in the GMV from earlier versions where scavenging was accomplished with either pistons or blowers. In these earlier designs, air was fed to each cylinder through the air chest. The GMVH design is turbocharged with external intake manifolds, but the base air chest was left active and connected to all cylinders. In actuality, this air chest provides excellent dampening of intake pulsations. However, the air being fed into and out of this air chest undoubtedly is heated due to the very large surface area at or near oil temperatures. A concept was derived based on this to block the base air chest and reduce the actual air temperature entering the cylinder.

The first analysis was to determine what percentage of the total mass flow entering each cylinder is fed from the air chest. The engine simulation model was utilized for this analysis and

results shown in Figure 5-11, where the instantaneous mass airflow from the air chest (plenum) and intake manifold runner is plotted with cylinder pressure. An integration of these traces shows that approximately 33% of the total mass airflow comes from the air chest, with the remainder coming from the intake manifold. This portion of the total mass flow was felt significant enough that eliminating the air chest flow would cause a significant reduction in inlet temperature. Simulation has shown that the air temperature entering the ports is approximately 20°F hotter than the air in the intake manifold log. A reduction in inlet air temperature would provide reduction in both NO_x emissions and the tendency to knock. Prior test data from lean-burn natural gas engines indicates that a 10°F drop in inlet air temperature could achieve approximately 0.2 g/BHP-Hr NO_x reduction in the GMVH-6. While not a significant reduction in total NO_x, an engine bordering on meeting permit levels would benefit in terms of efficiency with this amount of cushion or margin from the limit.

**Figure 5-11. Simulated Mass Airflow from Base Plenum,
Intake Manifold, and Through the Intake Ports**

The next step in this analysis was to redesign the intake manifold to prevent the pulsations that will occur with the elimination of the base plenum. Earlier simulations with the base plenum not incorporated into the model were used to derive the expected pulsation amplitude and frequency. The IPPS model was then utilized to determine if this pulsation was resonant and could be addressed with an SBA, or if simple enlargement of the volume is required. The pulsation was determined to not be resonant and, therefore, the additional volume required for the intake manifold was derived. With the new intake volume determined, the engine simulation model was utilized to determine the potential temperature reduction with this concept.

The simulation results showed a disappointing reduction of only 4°F. This reduction is not felt sufficient to justify the cost of removing each power cylinder and capping the base plenum with special gaskets. An alternative approach was to reduce the large volume (large surface area) inside each cylinder's airbox, which is suspected to still cause significant air heating even with the base plenum disconnected. This alternative, however, would require a new cylinder casting and would become even more costly.

The next concept for intake manifold modification was not derived from simulation, but rather engineering judgment. The intake manifolds on each bank are composed of a log with runners branching perpendicular to each cylinder. The entrance to each log is connected via bolted flange to a transitional duct on the aftercooler. The issues are with the bolted flange, which is internal to the manifold log, and the short transitional duct from the aftercooler. The transitional duct does not appear optimum for flow, and the internal flange in the manifold log creates an orifice that is very close to the first cylinders' (3L and 3R) runners. A photograph and sketch of the left bank intake manifold and aftercooler duct are shown in Figure 5-12. The orifice created by the internal flange is likely creating a vena-contracta that may be restricting flow to the first cylinder runner. Cylinders 3L and 3R are the two lowest cylinders in terms of compression pressure.

Figure 5-12. Photograph and Sketch of Intake Manifold Showing Region of Concern for Flow Disturbance and Restriction

If the vena-contracta is creating a flow structure that limits the flow to the first cylinders, then those cylinders will likely draw more air from the base air chest and may not trap as much total mass as the other cylinders. The air from the base is hotter than from the intake manifold air. This theory could explain the difference between measured data and simulation with geometric affects, which still cannot predict completely the spread in compression pressures. The flow phenomena suggested at the intake manifold entrance is highly dependent on the three-dimensional geometry. It is, therefore, suspected that a one-dimensional model may not completely capture the affects. The effect can either be simulated with a three-dimensional CFD program, or testing on the engine can determine if the suspected flow disturbance is significant in affecting the flow.

The engine test planned to address this suspicion is to add thermocouples in the base plenum access doors and probes in the inlet runners of Cylinders 1L and 3L. The thermocouples will provide the needed temperature measurement of base plenum air, and the probes will provide relative measurement of the dynamic airflow entering these cylinders. A significant reduction in airflow or velocity in Cylinder 3L compared to Cylinder 1L would validate that the intake manifold design is affecting flow and, therefore, trapped mass, among the cylinders.

The design concept would then involve removing the internal flange, possibly lengthening the intake log, adding an external flange, and possibly lengthening the aftercooler duct. The flange on the aftercooler duct will need to be redesigned for a larger bolt pattern matching the new external flange on the intake manifold log. This modification would also require a new mount design for the aftercoolers and modified turbocharger compressor outlet pipes.

The advantages of the Intake Entrance Modification for the GMVH-6 engine are as follows:

- Cost-effective retrofit of existing manifold, compared to new exhaust manifold.
- Concept design should provide better cylinder air balance. With cooler, denser charge to end cylinders. Potential for better air/fuel ratio balance.
- Can be coupled with and compliment exhaust manifold modifications.

The disadvantages of the Intake Entrance Modification are as follows:

- Potentially will affect only two of six cylinders, and performance gain would need to be determined for cost-effectiveness. However, on some engines, one poor performing cylinder (i.e., knocking or misfiring) would affect the overall engine performance if global spark timing and/or air/fuel ratio adjustment is required to compensate.

Additional testing is planned in the next task of this project. This testing will focus on quantifying any difference in actual flow from the intake manifold into the cylinders on a given bank. If the test results indicate a flow discrepancy, then design considerations to alleviate the flow imbalance will be determined.

6. CONCLUSIONS

Based on the data presented in Section 4, the following conclusions can be drawn:

1. Survey tests have been performed on engines at two sites: an HBA-6 at Duke's Bedford Station, and a TCVC10 at Dominion's Groveport Station.
2. Both candidates have the potential for compressor efficiency improvement and resultant improvement in capacity and system efficiency. Both candidates have undesirably high pulsations under some operating conditions.
3. The HBA-6 as tested exhibited an efficiency range from 84% when single acting, to 87% when double acting.
4. HBA-6 nozzle pulsations reached over 6% when single acting, and over 4.5% when double acting.
5. HBA-6 lateral pulsations exceeded 3% when single acting and 1.75% when double acting.
6. The TCVC10 as tested exhibited an efficiency of 76% at 330 RPM and 83% at 270 RPM.
7. TCVC10 nozzle pulsations exceeded 5% at 330 RPM and 4.5% at 330 RPM.
8. TCVC10 lateral pulsations exceeded 1.5% at 330 RPM and 1.3% at 270 RPM.
9. Design analysis has been performed on the HBA-6, using an acoustic model of the suction and discharge piping coupled to the compressor cylinders.
10. This model predicts significant pulsations at 1X running speed and at the nozzle resonance.
11. The model predictions show that the installation of new bottles would substantially reduce pulsations at running speed and would also reduce nozzle resonance pulsations.
12. The model predicts that installation of a side branch absorber will also reduce pulsations at 1X running speed but will have limited influence on nozzle resonance pulsations.
13. After discussions between SwRI and the host company, the planned modifications for the Bedford site are to install the side branch absorber together with nozzle orifices.
14. After installation of these modifications, the project will perform tests to quantify performance and pulsation level improvements.
15. A design analysis has also been performed for the TCVC10, guided by the test data from the Groveport Station.
16. The design analysis indicated a significant improvement in efficiency with a new pulsation filter bottle that allows elimination of eight orifices.

Based on the data presented in Section 5, the following conclusions can be drawn:

1. The GMVH-6 test engine has exhibited an 11% difference in compression pressure between the highest and lowest cylinder. A spread of 10% was consistently measured between Cylinders 1L and 3L, which are on the same bank and experience the same articulation effect.
2. The GMVH-6 test engine was disassembled and the geometric variances were quantified. Cylinder 3L, which has the lowest compression pressure, was found to have a 0.3 point lower effective compression ratio, 2 of 5 exhaust ports opening earlier and closing later by 1-1.5 degrees, and lower inlet port discharge coefficient at low port opening positions. These variations all trend with the lower compression pressure. Conversely, Cylinder 2R has the highest compression pressure but was found to have the lowest effective compression ratio. Cylinder 1R was found to be a different casting than the other five cylinders and exhibited several differences in port geometry and exhaust flow coefficient. Cylinders 1L and 3L were swapped and Cylinder 1R replaced during re-assembly to allow further testing of geometric effects.
3. A one-dimensional simulation model was constructed for the GMVH-6 engine using Virtual 2-Stroke[®] software. The model was validated to measured data from Cylinder 1L and used for analysis and design. Results of a model with identical cylinders, matching the mean values from geometric analysis, showed only a contribution from articulation on the compression pressures and no adverse effects of the original manifold designs. The model was used for sensitivity studies of the geometric variations and found they account for approximately 50 to 60% of the compression pressure spread. The remaining discrepancy between the measured spread and that contributed to by geometric variances is being investigated in the next task.
4. Several manifold concepts were developed. Design and simulation was conducted for each concept and several were eliminated due to cost-benefit predictions. Testing of the exhaust SBA concept is planned, and additional measurements are being acquired to validate theories of the intake manifold entrance design issue.

7. REFERENCES

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8. LIST OF ACRONYMS AND ABBREVIATIONS

AGA3	Gas Flow Measurement Standard
BDC	Bottom Dead Center
BEI	Manufacturer's Trade Name
BHP	Brake Horsepower
CPR	Combustion Pressure Ratio
CO ₂	Carbon Dioxide
DAS	Data Acquisition System
DIP	Differential Indicated Power
DOE	U.S. Department of Energy
GMC	Gas Machinery Conference
GMRC	Gas Machinery Research Council
GMV	Cooper Engine Model
GMV6	Copper Engine Model
GMVH	Cooper Engine Model
GMW10	Cooper Engine Model
HBA-6	Clark Engine Model
HBA-6T	Clark Engine Model
HP	Horsepower
Hz	Hertz
ICHP	Indicated Cylinder Horsepower
IRV	Instantaneous Rotational Velocity
KVG103	Ingersoll-Rand Engine Model
KVS	Ingersoll-Rand Engine Model
MMSCFD	Million of Standard Cubic Feet per Day
NGK	Manufacturer's Trade Name
NO _x	Oxides of Nitrogen
O ₂	Oxygen Molecule
PCB	Manufacturer's Trade Name
PPM	Parts Per Million
PSI	Pounds per Square Inch
PSIA	Lb./Sq. Inch Absolute
PSIG	Pounds per Square Inch Gauge
PV	Pressure-Volume
RLM	Rod Load Monitor
RPM	Revolutions per Minute
SDCM	Strain Data Capture Module
SwRI [®]	Southwest Research Institute [®]
TCF	Trillion Cubic Feet
TCV	Family of Dresser Clark engine models
TCVC10	Dresser Clark Engine Model
TDC	Top Dead Center
TGHP	Theoretical Gas Horsepower
TLA6	Clark Engine Model with Six Power Cylinders
V-10	10-Cylinder Engine with V Configuration