Report 11: Cost-Effective Reciprocating Engine Emissions Control and Monitoring for E&P Field and Gathering Engines

Technical Progress Report

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Abstract

This quarterly report discusses the results from a testing phase of the project that evaluates emission control technologies applied to a two-stroke cycle natural gas-fueled engine. In this phase, a one cylinder Ajax DP-115 (a 13.25 in bore $\times$ 16 in stroke, 360 rpm engine) is used to assess a sequential analysis and evaluation of a series of engine upgrades. As with most of the engines used in the natural gas industry, the Ajax engine is a mature engine with widespread usage throughout gas gathering industry. The end point is an assessment of these technologies that assigns a cost per unit reduction in NO_x emissions.

This report describes potential emission reduction technologies followed by a battery of tests that demonstrate synergies between some of the more promising technologies. While the end-goal is a closed loop control, low cost NO_x retrofit package, additional work remains.

The battery of tests include pre-combustion chambers, in-cylinder sensors, the means to adjust the air-to-fuel ratio, and modification of the air filter housing. During several phases of the tests, the ignition timing also was varied to determine the optimal point for ignition timing. The results from these tests suggest that an optimum exists where fuel consumption is minimized along with NO_x emissions.

By using the Ajax DP-115 these tests are completed in a low-cost and efficient manner. The various technologies can be quickly exchanged with different hardware, and the cost to operate the engine is very inexpensive.

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Introduction

The objective of this project is to identify, develop, test, and commercialize emissions control and monitoring technologies that can be implemented by exploration and production (E&P) operators to significantly lower the cost of environmental compliance and expedite project permitting. The project team will take considerable advantage of the emissions control research and development efforts and practices that have been underway in the gas pipeline industry for the last 12 years. These efforts and practices are expected to closely interface with the E&P industry to develop cost-effective options that apply to widely-used field and gathering engines, and which can be readily commercialized.

The project is separated into two phases. Phase 1 work establishes an E&P industry liaison group, develops a frequency distribution of installed E&P field engines, and identifies and assesses commercially available and emerging engine emissions control and monitoring technologies. Current and expected E&P engine emissions and monitoring requirements will be reviewed, and priority technologies will be identified for further development. The identified promising technologies will be tested on a laboratory engine to confirm their generic viability. In addition, during Phase 2 a full-scale field test of prototype emissions controls will be conducted on at least ten representative field engine models with challenging emissions profiles. Emissions monitoring systems that are integrated with existing controls packages will be developed. Technology transfer/commercialization is expected to be implemented through compressor fleet leasing operators, engine component suppliers, the industry liaison group, and the Petroleum Technology Transfer Council.

Research Progress

Forecasts of future U.S. natural gas demand of 30 trillion cubic feet (Tcf) /yr by 2015 require 36% production growth from 2001 levels, with the majority of this increase expected from three primary areas: Offshore Gulf of Mexico, Rocky Mountains, and Canadian imports. Mature basins in the Southwest and Mid-Continent areas also will contribute to the total domestic supply, and maximizing their output will be necessary to meet the aggressive 30 Tcf gas demand target. Oil and gas production operations in the United States face a wide variety of environmental regulations that are imposed by multiple, sometimes overlapping, jurisdictions. In particular, onshore production must grapple with existing and emerging regulations that address National Ambient Air Quality Standards for NO₂, ozone, fine particulates, acid deposition and regional haze, while pending air toxics regulations will address formaldehyde emissions from compressor engines. The scope of these regulations includes the wellhead and field gathering reciprocating engine-driven compressor equipment that is ubiquitous in E&P operations; current estimates are that approximately 15 million horsepower are presently operating in upstream production applications (Hanover Compressor Company 2001 10-K Annual Report filing). At an average size of 250 HP, this implies a total E&P fleet of 60,000 engines.

Though in many oil and gas production areas the airshed emissions inventory is dominated by coal power plants, regulatory agencies continue to pursue incremental reductions in the total pollutant loading. Reciprocating engines have been identified as a meaningful source category. These engines are used to produce electricity for a leasehold, compress and re-inject natural gas for increased oil production, or compress natural gas so that it can be delivered to local gathering systems that feed ultimately into gas transmission pipelines.

At present, the region with the greatest confluence of emissions concerns is the Rocky Mountain and Intermountain West area. In these regions, significant concerns about regional haze control accelerated the implementation of NO_x and fine particulate regulations that are only pending in many other producing areas. However, the incremental adoption of regulations state-by-state, as well as the proximity of many remote production areas in the Southwest to National Parks and Class I Wilderness Area (which are protected airsheds) may likely stimulate aggressive compressor engine controls in that and other production regions, as well. Finally, the East Texas and Louisiana regions are subject to conventional ambient ozone concerns, and have promulgated strict NO_x controls for reciprocating engines. Oil and gas production from all states will be required for the U.S. to meet the expected 30 Tcf/year gas demand and to minimize the ongoing slide in domestic oil production, and impediments to production that are created by air quality permitting must be alleviated through focused R&D efforts.

Gas compressor operations are an essential element of oil and gas production. Increased emissions constraints on compressor operations affects oil and gas production in four distinct ways:

- 1) The length of time to obtain an emissions permit is increased as multiple jurisdictions evaluate the effects of various pollutants and attempt to define a mutually acceptable permit level for a given engine;

- 2) The capital and operating costs of compressor engine operation are increased as this equipment is physically modified and/or operated differently to comply with the air permits;
- 3) The capital and operating costs of compressor engine operation are increased when expensive and maintenance-intensive continuous emissions monitors are required, as is the case in parts of California. In many settings, the cost of this monitoring exceeds the cost of NO_x control; and
- 4) Compressor operators may be forced to limit the annual hours of operation to avoid exceeding a fixed annual ceiling on allowed emissions.

Each of these situations impedes oil and gas production by:

- 1) Deferring the start of wellhead production thereby increasing the general business risk in current price-volatile markets and increasing the carrying costs of various lease and development fees;
- 2) Directly increasing the cost of compression services used at the wellhead; or
- 3) Artificially limiting the annual take from a well due to constrained operations.

The net effect is reduced oil and gas production for a given cost within a fixed time period. Multiplying this through thousands of production sites will most certainly have a significant negative impact on the ability of U.S. operators to meet domestic energy demands, and on the general productivity of the U.S. hydrocarbon resource base.

These economic and operating burdens to oil and gas operations can be reduced through a focused effort to develop cost-effective retrofit components, engine combustion controls, and engine performance monitoring options. The proposed project will significantly improve the cost-effectiveness of implementing NO_x and formaldehyde controls and monitoring on compressor engines, thus ensuring that compliance with air regulations does not prevent oil and gas operations from achieving their maximum productivity at competitive production costs.

Basis of the Project

This project draws heavily on the experience gained from the interstate gas pipeline industry's experience with NO_x emissions reductions, and their efforts to develop cost-effective options for extensive deployment throughout their systems. A number of gas pipelines faced EPA statutory deadlines in 1994/1995 to achieve and certify dramatic reductions in compressor engine NO_x emissions across a very wide range of ageing and diverse, but critical, equipment. Even though typical pipeline reciprocating compressor engines range in size from 600 HP to 8,000 HP and are largely two- and four-stroke cycle integral compressors, there is some commonality in equipment types and operational concerns with the wellhead and gathering facilities under study in this project. Beginning in 1990, the pipeline industry embarked on a comprehensive R&D program that targeted significant (50%+) reductions in the cost of NO_x controls without any significant engine performance compromises. All of the technologies developed had to be field-retrofitable and commercially-supported. That program was a significant success and created a number of technical options that allowed up to 80% NO_x reductions in a cost-effective and operationally-acceptable manner. The individuals involved with this current project were key participants in that prior pipeline NO_x and formaldehyde reduction program.

The gas pipeline emissions control technology development effort was instructive in that it employed the following six distinct phases of activity, each of which was necessary for success:

- 1) Obtain an industry consensus for the:
 - a) specific engine types and models on which to focus development efforts;
 - b) installed cost targets; and
 - c) realistic emissions levels to be achieved under all operating conditions.
- 2) Develop an inventory of installed horsepower to confirm initial industry guidance and to create a useful tool for impact analysis;
- 3) Create a coordinated, core team of engine technologists, regulatory experts, and industry representatives to ensure that engine design issues, regulatory drivers, and practical operating considerations always were addressed simultaneously;
- 4) Aggressively field test component and controls developments;
- 5) Characterize the fundamental relationships between engine operating parameters and exhaust emissions so that accurate, non-instrumented emissions monitoring systems could be deployed; and
- 6) Transfer technology results to organizations with an existing presence in the industry so that equipment could be provided on commercial terms, with emissions guarantees, and supported on an ongoing basis.

This project followed a similar broad outline with the expectation that the end product is a set of cost-effective emissions control and monitoring options that can be applied to a wide range of compressor engines in common use in oil and gas production. Operators will enjoy reduced costs of compliance, greater permitting certainty, reduced costs of emissions monitoring, and possible improved compressor performance due to improved combustion stability. All of this will sum to increased production as wells are brought online more rapidly, compression equipment is run harder and longer to facilitate increased production, and lifting cost savings are reallocated toward additional resource base development.

Emission Reduction Literature Review

This section of the report provides an overview of typical emission generation mechanisms as well as techniques to reduce emissions from two-stroke cycle (2SC) engines.

Combustion Temperatures, Residence Time, and Reaction Rates

A commonly known fact is that a reduction in combustion temperature leads to a reduction in the production of NO_x . Heywood (1988) explains the chemistry on why this happens. During the combustion process the majority of the NO_x is formed after the flame front (Borman and Ragland, 1998). Oxides of nitrogen formed after the flame front is termed thermal NO_x . The

following equations are referred to as the Zeldovich mechanisms, and are the primary chemical mechanisms for thermal NO production, (Borman and Ragland, 1998):

The reaction mechanisms are shown in the order in which they occur. The first equation dominates the reaction, and the remaining two reactions cannot occur until the first one is complete.

As stated above, NO_x formation is a function of temperature. When the combustion temperature is lowered, the first equation reaction rate reduces significantly. The rate of formation for the first reaction is given by the following two equations (Heywood, 1988):

$$\begin{aligned} \frac{d[\text{NO}]}{dt} = & k_{+1}[\text{O}][\text{N}_2] - k_{-1}[\text{NO}][\text{N}] + k_{+2}[\text{N}][\text{O}_2] - \\ & k_{-2}[\text{NO}][\text{O}] + k_{+3}[\text{N}][\text{OH}] - k_{-3}[\text{NO}][\text{H}] \end{aligned} \quad (4)$$

$$\begin{aligned} \frac{d[\text{N}]}{dt} = & k_{+1}[\text{O}][\text{N}_2] - k_{-1}[\text{NO}][\text{N}] + k_{+2}[\text{N}][\text{O}_2] - \\ & k_{-2}[\text{NO}][\text{O}] + k_{+3}[\text{N}][\text{OH}] - k_{-3}[\text{NO}][\text{H}] \end{aligned} \quad (5)$$

The variables $k_{\pm}$ are the forward and reverse rate constants.

Temperature, molecular surface area, and concentration all influence the activation energy level (Kuo, 2005). Since the goal is to reduce NO, NO₂, and NO_x, and the physical surface area of the molecule cannot be changed, temperature and concentration become the controlling factors. Consequently, lower temperatures result in lower activation energy in the mixture, thereby lowering the NO that is formed.

When the reaction rate is reduced by temperature, the first mechanism reduces the concentration of NO and N. This reduces the potential reactants in the second mechanism. Therefore, a reduction of the temperature in the first reaction introduces a limiting factor for NO_x emissions. For that reason, the goal of the technology to be applied is to help reduce the thermal NO_x by lowering the reaction temperature.

There are several technologies on the market today that can control thermal NO_x. Such technologies are: pre-combustion chambers, turbocharging, humidity control, exhaust gas recirculation, air-fuel ratio control, and ignition timing. The more appropriate of these technologies is briefly discussed in the following paragraphs.

Pre-Combustion Chamber (PCC)

There are several advantages to using a PCC over standard spark ignition. One advantage is that the PCC delivers a relatively large amount of thermal energy that is necessary to ignite very lean air-to-fuel mixtures. With leaner mixtures, the main combustion chamber temperature will be lower, which results in lower NO_x formation. Pre-combustion chambers have been used with a high degree of success on pipeline-size engines.

High Energy Ignition System (HEIS)

A high energy ignition system (HEIS) places a high-energy spark within the combustion chamber. This high-energy spark, which ranges from 125 mJ to 185 mJ, creates a larger than normal flame kernel that leads to the development of self-sustaining flame kernel within very lean fuel/air mixtures.

Air / Fuel Ratio

The effects of air/fuel ratio play a larger role when compared to ignition timing. Figure 1 shows the familiar diagram that demonstrates the effect of air-to-fuel ratio on emissions. In this particular figure, the horizontal axis is lambda, which is greater than one for lean mixtures. This can be compared to a rich mixture where there is more fuel than air. When the value for lambda

Figure 1. NO_x emissions as a function of trapped excess air in the cylinder

Figure 2. Emissions as a function of lambda (<http://www.remtechnology.com/applications/leancon2.htm>)

is 1.0 the mixture is stoichiometric. The majority of two stroke engines burn a lean mixture. As a result, CO and unburned hydrocarbon (HC) emissions are lower than for the stoichiometric case. Emissions of CO are lower because there is excess oxygen and the CO oxidizes with the excess O_2 into CO_2 . Figure 1 indicates that the NO_x emission rate appears as a bell curve in reference to the rich and lean air-fuel mixtures.

Air Supply

Excess air in the combustion cylinder creates a leaner mixture. This leaner mixture essentially serves as a diluent for the fuel in the cylinder, thereby decreasing the peak in-cylinder gas temperatures during combustion that then lead to a lower NO_x formation rate (Sharrah and Bomar, 1996).

Figure 2 demonstrates the effect of excess air on NO_x production. A small amount of air beyond stoichiometric does not decrease NO_x . This figure clearly illustrates that the trapped air-to-fuel ratio within the cylinder must increase beyond 10-20% excess air before NO_x emissions are significantly reduced. However, if the excess air is too great, then combustion becomes unstable.

Consequently, it is important to create a balance between air-to-fuel ratio and combustion stability.

Summary

This brief literature review clearly shows that NO_x emissions from a 2SC engine are a function of in-cylinder temperature. The in-cylinder temperature is a function of the trapped air-to-fuel ratio (or equivalence ratio) within the cylinder. Consequently, varying the trapped equivalence ratio is one of the primary keys to reducing NO_x emissions from the two-stroke cycle engine.

Ajax Two-Stroke Cycle Engine

While the overall project includes rich-burn and lean-burn four stroke cycle (4SC) engines, and lean-burn 2SC engines, this report focuses on the lean-burn two-stroke cycle engines used in the E&P industry. This engine classification is comprised almost exclusively of the Ajax engine. The Ajax engine, which is still in production today, is a horizontal reciprocating engine. The integral engine-compressor comes with 1, 2, 3 or 4 power cylinders depending on the power requirements.

Figure 3 shows the one-cylinder Ajax DP-115 engine that was used for this project and report. The DP-115 has a bore of 13.25 inches and a stroke of 16 inches. Figure 4 shows the inside of the cylinder. Of note is that the cylinder is scavenged by intake and exhaust ports. The 2SC engine must have a forced air flow rate through the engine. In many 2SC engines, this forced air flow is created by crank-case compression, pump scavenging, or turbocharging. The Ajax series of engines uses an air chest that surrounds the lower portion of the cylinder and a set of reed valves in the air intake system.

The Ajax Engine Cycle

As the piston moves toward top dead center during the compression portion of the cycle, air is sucked in through the reed valves. The typical reed valve assembly is shown in Figure 4. The reed valve consists of thin metal plates that are spring loaded to maintain contact with the reed valve seat. As the pressure decreases in the air chest, the reed valve metal plates are pulled down and air flows through the reed valves into the air chest. As the piston begins its power stroke, traveling from top dead

Figure 3. The NGML one-cylinder Ajax engine test bed during installation

Figure 4. Ajax engine reed valve assembly

center toward bottom dead center, the underside of the piston compresses the air in the air chest. As the air pressure increases, the reed valve metal plates seal against the reed valve seat to prevent air from escaping through the reed valves. The air pressure in the air chest reaches about 10 psig when the piston uncovers the top portion of the exhaust ports. At this point, exhaust blowdown occurs, and essentially depressurizes the cylinder. As the piston moves further toward bottom dead center, the intake ports become uncovered and the air-chest air, pressurized to 10 psig, flows into the cylinder to loop scavenge the exhaust products and create a fresh air charge for the next engine cycle. As the piston then moves from bottom dead center toward top dead center, the intake and exhaust ports are covered, and the combustion air that remains in the cylinder is compressed. At some point prior to top dead center, the gas injection valve in the center of the head, shown in Figure 5, admits gas into the cylinder. Since the Ajax is designed to operate with fuel pressures less than 30 psig, the gas must be admitted before the cylinder pressure reaches 30 psig. At about 7° before top dead center (BTDC), the two spark plugs fire to ignite the mixture inside the cylinder.

The One-Cylinder Engine Test Bed

The Ajax engine essentially represents a one-cylinder engine test bed that is similar in many ways to all 2SC engines used in the E&P industry. The rated speed of the test engine is 360 rpm and the rated torque is 1,605 lb-ft. The key difference is that the air flow rate through the Ajax engine cannot be easily increased because the cylinder uses a pressurized plenum to scavenge the cylinder. Since the accepted methodology to reduce NO_x emissions in a 2SC engine is to further lean the in-cylinder trapped charge, one way to accomplish this on the Ajax is to decrease the fuel flow rate into the engine. By keeping the air flow rate the same, the equivalence ratio necessarily decreases with decreasing fuel flow rate. A second way is to increase the engine speed.

The test engine was chosen because it represents a large percentage of the type of Ajax engines currently used in the E&P industry. This particular engine was manufactured in 1966 and did not contain the low emission combustion technologies that Cooper-Ajax includes on new engines. The primary differences are:

1. The test engine contained an oil-bath air filter and current production engines ship with a paper air filter;
2. Low emissions Ajax engines ship with a pre-combustion chamber head, while the test engine contains an open-chamber head.; and
3. The test engine is fitted with a mechanical governor and the new engine ships with an electronic governor.

Figure 5. The Ajax cylinder head showing the fuel valve and the spark plug

The series of tests for this portion of the project were designed to assess the emission reduction potential of each upgrade, as well

as assess any improvement in fuel consumption.

Engine Test Matrix

The test matrix was created during a research team meeting. Prior to this meeting, several emission reduction technologies were reviewed and ranked in order of emission reduction cost-effectiveness. The final test matrix is shown in Table 1. The left column in the table lists four variables that can be modified with a particular upgrade technology. The row along the top of the table lists the as-found condition, oil bath, along with four engine modifications that can potentially lead to low-cost emissions reduction. The column to the right of the oil bath is the build that replaces the oil bath air filter with the paper air filter. Reading down this column, the series of tests vary the ignition timing, the ignition energy, air-to-fuel ratio, and the amount of exhaust gas recirculation that is pushed into the trapped charge. As stated above, the air-to-fuel ratio is varied by varying the fuel that is admitted to the trapped charge. The three columns to the right of the paper filter column are additional engine modifications to be tested in conjunction with the paper air filter (i.e., the paper filter is not removed for these tests).

Only the shaded tests are reported in this report. The remaining tests are ongoing and will be completed in the next few months. Of note is that the oil bath and paper air filter tests were conducted with the standard ignition system and the two spark plugs shown in Figure 5.

Data Collection and Analysis Methodology

A variety of instrumentation was used to measure engine performance and emissions. The instrumentation and associated accuracy for each sensor is provided in Table 2. The sensor accuracy, however, does not necessarily indicate the uncertainty in the performance and emissions calculations. The uncertainty in a calculated parameter is based on the accuracy of the individual measurements that are used to calculate the parameter.

Table 1. Ajax engine test matrix (test results from shaded boxes are included in this report)

Variables	Builds				
	Vary Air Filter		With Paper Air Filter		
	Oil Bath	Paper Filter	Screw in PCC	Scavenge Port	Fuel Valve 180 psia
	No	Yes	Yes	Yes	Yes
Ignition Timing	No	Yes	Yes	Yes	Yes
Ignition Energy	No	Yes	Yes	Yes	Yes
Air-to-Fuel Ratio (load)	Yes	Yes	Yes	Yes	Yes
Exhaust Gas Recirculation	No	Yes	Yes	Yes	Yes

Table 2. Measurement accuracy of sensors used to assess the Ajax engine performance and emissions

Type/Location	Range	Accuracy
Barometric Pressure	0 to 30 psia	1.5% FS
Main Gas Pressure	0 to 50 psig	1.5% FS
PCC Fuel Pressure	0 to 50 psig	1.5% FS
Air Manifold Pressure	0 to 30 psia	1.5% FS
Exhaust Pressure	0 to 30 psia	1.5% FS
Main Fuel Flow/PCC Δp	0 to 83 inH ₂ O	1.0% FS
Air Flow Δp	0 to 30 inH ₂ O	±1.0% FS
Relative Humidity	5 to 100%	±3.0%
Outside Air Temperature	-20 to 140°F	±1.0% FS
Engine Inlet Water T	0 to 250°F	0.75% Range or ±1.87°F
Engine Outlet Water T	0 to 250°F	0.75% Range or ±1.87°F
Engine Air Inlet T	-20 to 250°F	0.75% Range or ±1.05°F
Fuel T	0 to 250°F	0.75% Range or ±1.87°F
Exhaust T	0 to 1,500°F	0.75% Range or ±4.0°F

For example, the air flow rate through the engine is calculated by:

$$\dot{m}_a = \frac{\pi}{4} k_a \sqrt{2\gamma_a h} \quad (6)$$

With any calculation such as that shown in equation (6), the uncertainty of the calculated parameter is determined from the measurement accuracy and the partial derivative of the measured value with respect to the calculated parameter (Benington and Robinson, 2003). In the case of equation (6), the calculated parameter is the air flow rate and the measured values are the variables k_a , γ_a , and h . Each of these measurements has an associated accuracy.

Once the partial derivatives and the measured-variable accuracies are known, the uncertainty for the calculated value is determined by:

$$U_{\dot{m}_a} = \sqrt{\left[\left(\frac{\partial \dot{m}_a}{\partial k_a} \right) U_{k_a} \right]^2 + \left[\left(\frac{\partial \dot{m}_a}{\partial \gamma_a} \right) U_{\gamma_a} \right]^2 + \left[\left(\frac{\partial \dot{m}_a}{\partial h} \right) U_h \right]^2} \quad (7)$$

As an example of one of the partial derivatives, the partial derivative of the mass flow rate with respect to the variable k_a is:

$$\frac{\partial \dot{m}_a}{\partial k_a} = \frac{\partial}{\partial k_a} \left[\frac{\pi}{4} k_a \sqrt{2\gamma_a h} \right] = \frac{\pi}{4} \sqrt{2\gamma_a h} \quad (8)$$

This method is used to estimate the uncertainty in the calculated result. Table 3 summarizes the equations and uncertainties used to calculate the performance and emissions results.

Test Results

The baseline test was conducted with the original oil bath air filter and the ignition timing set at the OEM-recommended value of 11°BTDC. For each test run, data was collected at engine speeds of 360 rpm (rated speed), 330 rpm, and 300 rpm. For each engine speed, measurements were taken at 1,605 lb-ft; 1,500 lb-ft; 1,400 lb-ft; and 1,200 lb-ft of torque. As stated above, the

air flow rate through the engine remains relatively constant with speed. Changing the load/torque, then, effectively changes the trapped equivalence ratio within the engine cylinder. Higher torque is indicative of higher trapped equivalence ratios.

The results presented for each test set include the brake specific fuel consumption (BSFC), brake specific NO_x emissions, the trapped equivalence ratio, and percent standard deviation of peak in-cylinder pressure (SDPP). The SDPP was chosen as an indicator of engine misfire. As will be shown, the SDPP increases at low trapped equivalence ratios since the mixture approaches (or exceeds) the lean limit of the engine.

Baseline Test: Figure 6 compares the BSFC for each of the three speed lines and the four torque settings for each speed line. Also shown in Figure 6 are the measured NO_x values for each data point. The BSFC trends from the baseline tests were within 3% of the values published by Ajax, which indicates the test engine was representative of the Ajax DP-115 engines found in the E&P industry. The full load/full torque brake specific NO_x emission was 4.69 g/bhp-hr. This measured value compares favorably with published OEM Ajax data that the full speed/full torque NO_x emissions are 4.4 g/bhp-hr. Considering the NO_x measurement uncertainty shown in Table 3, these two values are almost the same. When compared to all of the subsequent tests, the baseline NO_x emissions were the highest full speed/full torque NO_x brake specific emissions that were measured.

Starting at the highest torque setting, the BSFC curves may decrease slightly with torque, depending on the uncertainty in the calculation. What is clear from the trends in Figure 6 is that

Figure 6. Baseline BSFC and NO_x vs percent load (torque)

Table 3. Summary of equations and uncertainties

	Uncertainty	Equation
Air Flow though Nozzle	18.0 lb/hr	$\dot{m}_a = \frac{\pi}{4} k_a \sqrt{2\gamma_a h}$
Equivalence Ratio		$\phi = \frac{(m_{tf} / m_{ta})}{(FA)_s}$
Fuel Flow Rate	0.738 lb/hr	$\dot{m}_f = \frac{\pi}{4} k_g \sqrt{2\gamma_g h}$
Trapped Air Fuel Ratio	0.908	$TAFR = \frac{m_{ta}}{m_{tf}}$
Normalized NO _x Flow Rate	0.1757 g/bhp-hr	$\dot{m}_{NO_x} \left[\frac{g}{bhp-hr} \right] = \left(\frac{C_i [ppm]}{10^6} \right) \times \rho_{sa} \times \frac{V_{ex}}{\dot{P}} \times \frac{M_e}{M_a}$

the BSFC increases with decreasing load at a given engine speed. While there is some difference between the different speeds at full torque, this difference is generally within the uncertainty of the measurement.

Starting again at the highest torque setting, the NO_x curves demonstrate a trend that is similar to the BSFC curves. The NO_x values decrease with torque from the full torque point until a minimum is reached. The NO_x emissions then increase with further decreases in torque.

The trends in the speed curves are explained by the fact that decreasing torque is analogous to decreasing trapped equivalence ratio. As the trapped equivalence ratio decreases, the mixture becomes leaner. Up to a point, leaner mixtures result in lower peak in-cylinder temperatures. These lower in-cylinder temperatures generate lower NO_x emissions. Consequently, as the equivalence ratio decreases, so does the NO_x emitted from the engine. After a certain minimum trapped equivalence ratio is reached, the engine ignition system can no longer ignite the in-cylinder mixture, and the engine begins to experience misfires. Every time the engine misfires, the engine slows down which causes the governor to open the throttle to regain the speed setpoint. This extra fuel increases the equivalence ratio for a short period of time. The subsequent cycle contains fuel from the misfired cycle as well as fresh fuel admitted through the fuel valve. The end result is a much higher equivalence ratio within the cylinder. This higher equivalence ratio leads to a higher in-cylinder peak pressure, which in turn leads to a higher NO_x emission. Therefore, higher misfire rates lead to higher emitted NO_x levels. This is clear from the NO_x trends shown in Figure 6.

The BSFC trends show that as the engine torque decreases at a given speed, the BSFC increases. The reason for this is that the BSFC is a measure of how much fuel energy is actually transmitted

Figure 7. Typical pressure-crank angle trace from the Ajax DP-115

to the engine output shaft. The pressure-crank angle trace within the engine cylinder, shown in Figure 7, can be translated to a pressure volume trace. The area under the curve represents the amount of work/power produced by the expanding gases within the cylinder. This particular work/power is oftentimes referred to as the indicated work/power, and is the maximum work/power that can be transferred to the engine output shaft.

As the indicated work/power is transferred from the piston surface to the output shaft, some of the work is destroyed by friction and used by engine auxiliaries. This lost work is referred to as friction work. The relationship between indicated, friction, and brake work/power is:

$$\dot{P}_b = \dot{P}_{ind} - \dot{P}_f \quad (6)$$

The parameter $\dot{P}$ represents work/power, and the subscripts b , ind , and f represent: brake, indicated, and friction work/power, respectively. Referring to equation (4), the friction work $\dot{P}_f$ is relatively constant for a given speed and is, for the most part, independent of engine torque.

The indicated work $\dot{P}_{ind}$ is a direct function of engine torque. Consequently, as engine torque decreases at a given constant speed, the friction work remains constant while the indicated work decreases. The end result is that the percentage of the fuel energy that is converted to useful work at the output shaft decreases with engine torque.

The trends in Figure 6 show that there is a point where the NO_x emission reaches a minimum, but the expense is a somewhat greater BSFC. For the 360 rpm speed line, the lowest achievable NO_x emission level was 2.75 g/bhp-hr at an engine torque of approximately 90% (Figure 6). The curves can be used to determine the operating costs of achieving a specific NO_x reduction by lowering the engine load. Conversely, the curves may be used to determine the amount of additional air required to operate the engine at 100% load, but with the trapped equivalence ratio that exists at the tested load of 90%. This is discussed further in the next set of tests.

New Air Filter: Figures 8 and 9 compare the data from the baseline test to the data collected with the oil bath filter replaced by the paper air filter. All these data were collected at an ignition timing of 11°BTDC.

Figure 8 shows the same information as that shown in Figure 5. When the measurement uncertainty is considered, the data within these two curves is virtually identical. The only potentially different data are the BSFC at full load. The difference in the BSFC between the two tests is on the fringe of the uncertainty.

Figure 8. Comparison of baseline and new air filter: BSFC and NO_x vs percent load (torque)

Figure 9 compares the trapped equivalence ratio and the SDPP for each of the tests. As explained earlier, the SDPP is lowest at full load, but increases rather significantly as the load is decreased. By comparing the two graphs, three observations are evident:

1. The trapped equivalence ratio measured with the oil bath filter is a strong function of engine speed;

2. The trapped equivalence ratio measured with the paper air filter shows a leaner trapped mixture within the cylinder than that with the oil bath filter; and
3. The trapped equivalence ratio measured with the paper air filter is not a strong function of engine speed.

The impact from the decreased trapped equivalence ratio caused instability in the combustion process, which increases the rate of engine misfires. This was demonstrated in Figure 8, where the SDPP with the paper air filter increases over the baseline test.

Figure 9. Comparison of baseline and new air filter: SDPP and ϕ_{tr} vs percent load

The next sets of tests, shown in Figures 10 and 11, compare the data collected with the paper air filter, but at ignition timings of 11°BTDC and 3°BTDC. Figure 10 clearly illustrates that one of the impacts from retarded ignition timing was the well-known increase in the BSFC. The trade-off, however, is that a considerable reduction in the NO_x occurs that decreases from 4.54 g/bhp-hr to 3.01g/bhp-hr, which is a reduction of 35.8% from the baseline test. As stated earlier in the introduction, NO_x emissions are direct function of peak in-cylinder temperature. Although not shown in a graph, the peak in-cylinder pressure decreased with the retarded timing. This decrease leads to a lower in-cylinder peak temperature. It was found by retarding the ignition timing from

Figure 10. Comparison of new air filter 11°BTDC and new air filter 3°BTDC: BSFC and NO_x vs percent load

11°BTDC to 3°BTDC the peak pressure decreased by 13.8%. Also apparent from Figure 10 is that a NO_x reduction can be achieved with almost no increase in BSFC. This region exists until the load is reduced to about 95%.

Figure 11 shows the relationship between the increased BSFC and the trapped equivalence ratio. Since the BSFC increases and the air flow rate remains relatively constant at a given speed, the trapped equivalence ratio must increase, as shown in Figure 11. The somewhat richer trapped

Figure 11. Comparison of new air filter 11°BTDC and new air filter 3°BTDC: SDPP and ϕ_{tr} vs percent load

mixture increases combustion stability as illustrated by the reduced SDPP with the retarded ignition timing. This reduction in SDPP can be seen with all the speed lines and loads.

Pre-Combustion Chamber: With the impact of the ignition timing, it was evident to meet the goal in reduction of NO_x , the ignition timing at 3°BTDC was the best option with the paper air filter. Figures 12 and 13 compare the test results at 3°BTDC ignition timing, but with conventional spark ignition and ignition with a screw-in pre-combustion chamber. Figure 12 shows the impact of the pre-combustion chamber, which not only reduced the BSFC but also

Figure 12. Comparison of conventional and pre-combustion chamber ignition at 3°BTDC : SDPP and ϕ_{ir} vs percent load

reduced NO_x emissions. From the baseline test there was an NO_x reduction to 2.75g/bhp-hr, which represents a reduction of 41.4% in NO_x .

It also can be seen the best operating area was shifted to a higher load. The test data shows that the slope of the BSFC is almost zero from 100% torque to about 95%. This almost zero slope suggests that the NO_x emissions can be reduced with the PCC without a corresponding loss in engine efficiency. Also of interest in Figure 12 is that the NO_x emissions do not demonstrate an increase as the load/trapped equivalence ratio is reduced. The postulated reason for this is that combustion stability is further improved via the PCC.

Figure 13. Comparison of conventional and pre-combustion chamber ignition at 3°BTDC: BSFC & NO_x vs percent load

Figure 13 illustrates the impact of the pre-combustion chamber on combustion stability and trapped equivalence ratio. The data clearly shows that the major impact was on the decreased rate of misfires, which was indicted by the reduction in the SDPP. The PCC not only reduces NO_x, but also provides better combustion stability at much leaner conditions/lower engine loads.

Conclusions and Future Work

Table 4 provides the NO_x emission levels, peak in-cylinder pressures, and NO_x reduction percentages for each test. Several conclusions can be drawn from these tests. These are:

1. The paper air filter provided some benefits. The most notable benefit is an increased air flow rate through the engine. This increased air flow rate, however, actually increased combustion instability as measured by the SDPP. The differences in the NO_x emissions and the BSFC were not significant. The key conclusion from this set of tests is that while the paper air filter increased the air flow rate through the engine, the increased air flow by itself does not improve engine performance or reduce engine emissions.
2. Retarding the ignition timing increased BSFC and reduced NO_x emissions. This is a typical result seen on pipeline-style engines. Additionally, combustion stability was improved by retarding the timing. The reason for this is that in the case of the Ajax engine, the increased BSFC directly increases the trapped equivalence ratio that then leads to improved combustion stability.
3. The test data from the pre-combustion chamber clearly demonstrates a reduction in the BSFC and a reduction in NO_x emissions. The primary benefit from the pre-combustion chamber is that the engine can operate with leaner mixtures and lower loads with a greater level of combustion stability.

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Table 4. Summary of results for the tests at full speed/full torque

	NO _x [g/bhp-hr]	% NO _x Reduction	Peak Pressure [psig]
Baseline	4.69	--	452
New Air Filter/ 11°BTDC	4.54	3.1%	425
New Air Filter/ 7°BTDC	3.73	20.4%	404
New Air Filter/ 3°BTDC	3.01	35.8%	366
PCC/ 3°BTDC	2.75	41.4%	384

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