Improvement to Pipeline Compressor Engine Reliability through Retrofit Micro-Pilot Ignition System – PHASE I

ANNUAL TECHNICAL REPORT

For the Period:

9/19/01 – 9/18/2002

Prepared by:

Ted Bestor, EECL Director

Submitted/ Revised:

Originally Submitted: January 10, 2003
Revised: March 4, 2003 (Revision1)

DOE Award Number:

DE-FC26-01NT41162

Submitted by:

Colorado State University
Mechanical Engineering Department/ Engines & Energy Conversion Laboratory
Fort Collins, CO 80523-1374

DISCLAIMER

ABSTRACT

This report documents the first year's effort towards a 3-year program to develop micropilot ignition systems for existing pipeline compressor engines. In essence, all Phase I goals and objectives were met. We intend to proceed with the Phase II research plan, as set forth by the applicable Research Management Plan.

The objective for Phase I was to demonstrate the feasibility of micropilot ignition for large bore, slow speed engines operating at low compression ratios. The primary elements of Micropilot Phase I were to develop a single-cylinder test chamber to study the injection of pilot fuel into a combustion cylinder and to develop, install and test a multi-cylinder micropilot ignition system for a 4-cylinder, natural gas test engine. In all, there were twelve (12) tasks defined and executed to support these two (2) primarily elements in a stepwise fashion. Task-specific approaches and results are documented in this report.

Research activities for Micropilot Phase I were conducted with the understanding that the efforts are expected to result in a commercial product to capture and disseminate the efficiency and environmental benefits of this new technology. An extensive state-of-art review was conducted to leverage the existing body of knowledge of micropilot ignition with respect to retrofit applications. Additionally, commercially-available fuel injection products were identified and applied to the program where appropriate. This approach will minimize the overall time-to-market requirements, while meeting performance and cost criteria.

The four-cylinder prototype data was encouraging for the micro-pilot ignition technology when compared to spark ignition. Initial testing results showed:

- Brake specific fuel consumption of natural gas was improved from standard spark ignition across the map, 1% at full load and 5% at 70% load.
- 0% misfires for all points on micropilot ignition. Fuel savings were most likely due to this percent misfire improvement.
- THC (Total Hydrocarbon) emissions were improved significantly at light load, 38% at 70% load.
- VOC (Volatile Organic Compounds) emissions were improved above 80% load.
- Coefficient of Variance for the IMEP (Indicated Mean Effective Pressure) was significantly less at lower loads, 76% less at 70%.

These preliminary results will be substantiated and enhanced during Phase II of the Micropilot Ignition program.

TABLE OF CONTENTS

APPENDICES

EXPERIMENTAL

The objective for Micropilot Phase I was to demonstrate the feasibility of micropilot ignition for large bore, slow speed engines operating at low compression ratios. Two experimental efforts were conducted to support this objective: 1) study the injection of pilot fuel into a combustion cylinder using a single-cylinder test chamber, and 2) develop, install and test a multi-cylinder micropilot ignition system for a 4-cylinder, natural gas test engine.

The data from CTC experimentation were primarily visual in nature, consisting of various image types, such as digital still photos, high-speed digital video images, and laser techniques. These images were used to quantify spray angle and spray penetration for the pilot fuel. A separate report describing the methods is attached as Appendix 5. Pictures and data analysis is included in Appendix 6.

Data for the on-engine testing was obtained with standard, laboratory-grade emissions analyzers and combustion analysis systems. Results and discussion of the methods used are included in Appendix 10.

RESULTS AND DISCUSSION

Improvement to Pipeline Compressor Engine Reliability through Retrofit Micro-Pilot Ignition System – PHASE I

Introduction

This report documents the first year's effort towards a 3-year program to develop micropilot ignition systems for pipeline compressor engines. In summary, all Phase I goals and objectives were met. We intend to proceed with the Phase II research plan, as documented by the applicable Research Management Plan, transmitted under separate cover.

Account of Progress

The primary tool used for predetermining the research activities is the Research Management Plan, Appendix 1. The individual tasks and original timeline are shown below, followed by a description of the deliverable produced for each task.

PHASE I TEST PLAN

Phase I Project Tasks	**PRIMARY RESPONSIBILITY**	**O ‘01**	**N ‘01**	**D ‘01**	**J ‘02**	**F ‘02**	**M ‘02**	**A ‘02**	**M ‘02**	**J ‘02**	**J ‘02**	**A ‘02**	**S ‘02**
Task 1: Research Management Plan	CSU	🟦											
Task 2: Review Prior Research	CSU	🟦	🟦										
Task 3: Develop System Specification	Woodward		🟦	🟦									
Task 4: Design/Build 1-Cyl. Prototype	CSU			🟦	🟦								
Task 5: Test 1-Cylinder Prototype	CSU				🟦								
Task 6: Analyze Results 1-Cyl. P-type	CSU				🟦	🟦							
Task 7: Develop 4-Cyl. Product Spec.	Woodward			🟦	🟦	🟦	🟦						
Task 8: Design/Build 4-Cyl. Prototype	CSU					🟦	🟦	🟦	🟦				
Task 9: Install 4-Cylinder Prototype	CSU								🟦	🟦	🟦		
Task 10: Test 4-Cylinder Prototype	CSU											🟦	🟦
Task 11: Phase I Report	CSU											🟦	🟦
Task 12: DOE Contractors Meeting	CSU												🟦

Task 1: Research Management Plan

This document was submitted at the project's start and was updated for each quarterly progress report. The final version for Phase I is attached as Appendix 1.

Task 2: Review Prior Research

A report documenting our literature review was submitted along with the first quarterly progress report and is attached as Appendix 2. This report detailed the existing body of knowledge regarding micropilot ignition systems for reciprocating engines. It was concluded that retrofit micropilot ignition technology, defined as pilot fuel consuming less than 1.0% of total energy content, is nonexistent for large, stationary engines and virtually undeveloped for most other applications. Micropilot ignition systems are, however, commercially available for some new engines as a purchased option and the benefits associated with the technology have been demonstrated. The literature review also served as a starting point for modeling and other analytical efforts.

Task 3: Develop System Specifications

This document was submitted with the first quarterly report and is attached as Appendix 3. This task served as a starting point for further system enhancements as experimental information was obtained. The system specification was developed using information from the literature review and input from EECL and Woodward personnel.

Task 4: Design/ Build 1-Cylinder Prototype

The objective of this task was to create an experimental apparatus in order to evaluate micropilot injection pressure, quantity, and spray patterns. The final deliverable for this task was a Combustion Test Chamber (CTC) which was designed and assembled by EECL personnel. Drawings and photos were submitted with Quarterly Report #2, and are attached to this report at Appendix 4.

Task 5: Test 1-Cylinder Prototype

The data from CTC experimentation were primarily visual in nature, consisting of various image types, such as digital still photos, high-speed digital video images, and laser techniques. These images were used to quantify spray angle and spray penetration for the pilot fuel. A separate report describing the methods is attached as Appendix 5.

Task 6: Analyze Results from 1-Cylinder Prototype

CTC studies verified that the capability of the prototype performed well against the specification set in Task 3. Pictures and data analysis is included in Appendix 6.

Task 7: Develop 4-Cylinder Prototype

Prototype hardware for the GMV-4 test engine was developed using a commercially available pilot fuel injection system manufactured by Delphi Corporation. Identification and procurement of an appropriate, “off-the-shelf” system was critical to meeting the cost objectives of the program. Pertinent specifications are included in Appendix 7.

Task 8: Design/ Build the 4-Cylinder Prototype

Certain modifications were necessary to adapt the Delphi system to the GMV-4 test engine, the most notable being the electronic valve controller. An “InPulse” electronic valve driver manufactured by the Woodward Governor Company (the commercialization partner) was programmed to properly control the Delphi components. Also, custom fuel storage tank and delivery tubing was fabricated.

Minor modifications to the engine were also designed. Since the Delphi fuel injectors were designed for an automotive engine, an adapter was designed by EECL personnel and fabricated accordingly. The GMV engine model typically uses 2 spark plugs per cylinder and one spark plug port per cylinder head was used for the injector/ adapter. Another modification involved the design and fabrication of bolt-on "pancakes", or contoured plates that were used to increase the height of the pistons and thereby increase the compression ratio in the combustion cylinders. Schematics and drawings for this task are included in Appendix 8.

Task 9: Install the 4-Cylinder Prototype

The system was relatively simple to install, since most of the control system components were integrated previously for the 1-Cylinder prototype (CTC) studies. Engine modifications were limited to machining of the spark plug ports to accept the pilot fuel injector and installing the piston "pancakes" described above. Photographs of the system as installed on the GMV-4 test engine are included in Appendix 9.

Task 10: Test the 4-Cylinder Prototype

Preliminary testing was performed in December, 2002. A description of the experimental data, data reduction methods, and conclusions is included in the Test Report contained in Appendix 10. The four-cylinder prototype data was encouraging for the micro-pilot ignition technology when compared to spark ignition. Initial testing results showed:

- Brake specific fuel consumption of natural gas was improved from standard spark ignition across the map, 1% at full load and 5% at 70% load.
- 0% misfires for all points on micropilot ignition. Fuel savings were most likely due to this percent misfire improvement.
- THC (Total Hydrocarbon) emissions were improved significantly at light load, 38% at 70% load.
- VOC (Volatile Organic Compounds) emissions were improved above 80% load.
- Coefficient of Variance for the IMEP (Indicated Mean Effective Pressure) was significantly less at lower loads, 76% less at 70%.

These preliminary results are consistent with the program objectives as originally proposed and will be substantiated and enhanced during Phase II of the Micropilot Ignition program.

Task 11: Phase I Report

Contained herein.

Task 12: DOE/ NETL Contractor's Meeting

The EECL will attend this meeting upon notification in order to present the results of this research.

Problems Encountered

Most of the problems were associated with the 1-cylinder prototype, or combustion test chamber (CTC). This was a new experimental apparatus designed and constructed primarily for the micropilot program. Difficulties with CTC itself included cracking of the quartz material used for optical windows which was solved by design changes. The optical imaging method selected for the plume characterization also required significant development. Image techniques used in previous research were not applicable to the temperatures and pressures of this study.

Many techniques were investigated and determined to be insufficient. These included:

1. Back lighting / shadowgraph with diffused light source
    a. This technique required a very quick shutter speed that required more light than we were able to create with available equipment.
2. Back lighting / shadowgraph with laser light source
    a. Using the laser light as a point source allowed us to stop the motion of the spray with a fast shutter speed or a pulsed laser. The quick shutter speed was attempted first but during setup the intense laser light damaged the camera.
3. Shlieren with laser light source
    a. Shlieren was a very promising technique. Very good images could be taken with ambient conditions in the CTC. However, when the CTC was heated, large convective currents and small changes in air density within the chamber dominated the image hiding the fuel spray.
4. Mie scattering with halogen light source
    a. This technique required a very fast shutter speed to stop motion and the white light washed much of the plume out.

Minor start-up problems with the 4-cylinder prototype were also encountered. It was discovered that too much material was removed from the cylinder heads during modification to accept the pilot injectors, thus small cracks occurred in the affected areas. Spare heads were machined with a revised procedure in order to preclude further failures. Another start-up problem involved the programming of the InPulse fuel injector controller that was solved by on-site assistance by Woodward personnel.

Significant Accomplishments

1. Design of CTC
2. Complete assembly of CTC
3. Procedure for injector quantity mapping
4. Advances is heating techniques allowing greater test temperatures in CTC
5. Producing high resolution spray images with Laser illuminated mie scattering
6. Running 1 cylinder of the Cooper Bessemer GMV-4 on micro-pilot ignition
7. Running all four cylinders of the Cooper Bessemer GMV-4 on micro pilot ignition

Publications and Presentations

No reports that contain data or results, other than those submitted to NETL per the Federal Assistance Reporting Checklist, have been published. Progress reports have been presented to NETL, in Morgantown, on 2 occasions:

1) November 15, 2001 – “Pipeline Infrastructure Contractor’s Kickoff Meeting”
2) September 16, 2002 – “Natural Gas Infrastructure Reliability Industry Forum”

Plans for Next Reporting Period

We are proceeding with the tasks defined for Phase II of the program as listed in the schedule below:

PHASE II PROJECT SCHEDULE

Phase II Project	**PRIMARY RESPONSIBILITY**	**O ‘02**	**N ‘02**	**D ‘02**	**J ‘03**	**F ‘03**	**M ‘03**	**A ‘03**	**M ‘03**	**J ‘03**	**J ‘03**	**A ‘03**	**S ‘03**
Task 13: Research Management Plan	CSU	■											
Task 14: Evaluate Compression Ratio	CSU	■											
Task 15: Evaluate Pilot Fuels	CSU		■										
Task 16: Analyze Prototype Results	CSU	■	■	■									
Task 17: Revise Product Specifications	Woodward			■	■	■							
Task 18: Revise Dsn to Optimize Perf.	Woodward					■	■	■					
Task 19: Evaluate with Optical Engine	CSU							■	■				
Task 20: Lab Test to Verify Performance	CSU								■	■	■		
Task 21: Finalize Design for Field Test	Woodward										■	■	■
Task 22: Phase II Report	CSU											■	■
Task 23: DOE Contractors Meeting	CSU												■

These tasks are defined further in the Research Management Plan for Phase II, attached as Appendix 11. The first Quarterly Report for Phase II will include Tasks 13-16.

Assessment of the Prospects for Future Progress

The testing conducted for Task 10, described earlier, provided substantial evidence that the micropilot ignition technology being developed under this program will be successful in meeting the objectives. The trends for reductions in both emissions and fuel consumption are in the right direction, and the magnitude of these reductions will be enhanced during Phase II.

As the schedule above indicates, we are currently 3 months behind schedule, but intend to recover lost progress during the second and third quarters.

CONCLUSION

Phase I of the Retrofit Micropilot Ignition System was successful in demonstrating that:

1. Micropilot ignition systems are technically capable of delivering efficiency and emissions improvements when compared to spark ignition systems
2. Appropriate hardware and control system components are commercially available now, providing an expeditious path to market.
3. The technology can be applied to existing pipeline compressor engines on a retrofit basis.

APPENDIX 1

“Research Management Plan – Phase I”

Improvement to Pipeline Compressor Engine Reliability through Retrofit Micro-Pilot Ignition System – PHASE I

RESEARCH MANAGEMENT PLAN

Work to be Performed

The project team, CSU and Woodward, have adopted a technical approach that produces the highest probability of success and the shortest time-to-market. Woodward has an existing business and technical background in fuel system control for large, stationary, reciprocating engines and gas turbines; CSU has a well-developed engine laboratory for the research and development of combustion technology that is applicable to pipeline, reciprocating compressor engines. Woodward has many years of experience in product development and the commercialization of technology for the natural gas industry and natural gas pipelines. In Phase I of the program, we will design a common rail micro-pilot ignition system by combining Woodward controls and electronics with commercially available common-rail injectors. The injectors will be modified to allow mounting in the test engine and the nozzle tips will be modified to produce the desired spray pattern. The entire system will be controlled with existing Woodward electronics. In Phase II of the program, the system will be refined and optimized. In Phase III of the program, the system will be field-tested.

Phase I Project Scope – 2001

The objective of the project in year one is to demonstrate the feasibility of micropilot ignition for large bore, slow speed engines operating at low compression ratios. The project is expected to include single cylinder tests without combustion, refinement of the system parameters, and testing of a 4-cylinder prototype in the laboratory.

Task 1: Research Management Plan

STATUS: Complete

Develop a work breakdown structure and supporting narrative that concisely addresses the overall project as set forth in the proposal by CSU. Provide a concise summary of the technical objectives and technical approach for each Task and, where appropriate, for each subtask. Provide detailed schedules and planned expenditures for each Task including any necessary charts or tables, and all major milestones and decision plans.

Phase I Project Tasks	**PRIMARY RESPONSIBILITY**	**O ‘01**	**N ‘01**	**D ‘01**	**J ‘02**	**F ‘02**	**M ‘02**	**A ‘02**	**M ‘02**	**J ‘02**	**J ‘02**	**A ‘02**	**S ‘02**
Task 1: Research Management Plan	CSU	🟦											
Task 2: Review Prior Research	CSU	🟦	🟦										
Task 3: Develop System Specification	Woodward		🟦	🟦									
Task 4: Design/Build 1-Cyl. Prototype	CSU			🟦	🟦								
Task 5: Test 1-Cylinder Prototype	CSU				🟦								
Task 6: Analyze Results 1-Cyl. P-type	CSU				🟦	🟦							
Task 7: Develop 4-Cyl. Product Spec.	Woodward			🟦	🟦	🟦	🟦						
Task 8: Design/Build 4-Cyl. Prototype	CSU					🟦	🟦	🟦	🟦				
Task 9: Install 4-Cylinder Prototype	CSU								🟦	🟦	🟦		
Task 10: Test 4-Cylinder Prototype	CSU											🟦	🟦
Task 11: Phase I Report	CSU											🟦	🟦
Task 12: DOE Contractors Meeting	CSU												🟦

Task 2: Review Prior Research

STATUS: Complete

Benefit from the significant body of work already conducted by other researchers and manufacturers. A preliminary literature review has been performed and will be expanded by CSU. Further work on this project will be grounded in the technology documented in the complete literature review. A review paper will be published documenting important findings from the literature review.

RESULTS: submitted previously.

Task 3: Development of System Specification

STATUS: Complete

Create a system concept specification as a commercialization goal and benchmark against which research and development progress can be measured. Woodward will provide the direction and leadership for this task. The project team is proposing a high-pressure common-rail approach. With the use of this technology, pilot injection quantities as small as 1 mm^3 are achievable. We anticipate experimenting with pilot injection quantities from $1mm^3$ to $20mm^3$ and injection pressures between 400 to 1300 bar (6000-20,000 psi). Nozzle hole size and orientation will be altered to achieve pilot fuel penetration of 50-250 mm. Based on the electrical characteristic of the common-rail injectors, Woodward will adapt the power electronics in their current In-Pulse™ system to drive the injectors. For the initial laboratory evaluations, the fuel injectors will be mounted through one of the spark plug hole in a set of dual-spark plug heads. This method of mounting, if successful, would allow the most efficient implementation of the system in field tests and subsequent commercialization. High-pressure fuel will be supplied for the test program with a variable displacement, high-pressure pump driven with an electric motor.

RESULTS: submitted previously.

Task 4: Design and Build Single-Cylinder Prototype

STATUS: Complete

Build a simple, low cost, single-cylinder test prototype to evaluate conceptual principles. Based on the specification developed in the previous task, hardware will be built/ procured for a single cylinder evaluation prototype. Design and construction will be led by CSU. This system will utilize identical components to those anticipated for the full 4-cylinder laboratory test. A set of nozzles will be built with different hole diameters and orientations to verify the analytical predictions developed in the specification stage.

RESULTS: submitted previously.

Task 5: Performance Test of Single-Cylinder Prototype

STATUS: Complete

Evaluate prototype performance against system specification. The CSU EECL has a single-cylinder stationary test rig that will be modified and used for this test program. This test rig duplicates the geometry in the cylinder at the time of injection but has transparent walls so in-cylinder phenomena can be observed. The test rig can be pressurized to duplicate the pressure in the cylinder at the time of pilot injection. The cylinder will be pressurized with nitrogen to avoid any possibility of combustion in the test rig. Utilizing the single cylinder hardware developed in the previous section, a series of tests will be conducted to verify proper penetration and dispersion of the fuel spray. These tests will evaluate the effects of nozzle design, injection pressure, injection quantity / duration, and cylinder pressure.

During the single cylinder evaluations, high-speed video imaging (10,000 frames per second) will be utilized to document the spray behavior. Other techniques at the EECL that could potentially be used include Schlieren photography and laser fluorescence. The goal of this stationary test is to “shape” the fuel plume such that adequate penetration and spread of the fuel injection system are achieved.

RESULTS: submitted previously.

Task 6: Analysis of Single-Cylinder Results

STATUS: Complete

Determine the capability of the prototype performance against specifications. CSU will determine the shape and penetration (plume length vs. time) for the injectors evaluated in the single cylinder tests. These results will be compared to analytical predictions to allow empirical correction of the models used. From the test program and any subsequent modeling, the project team will finalize important system parameters, including: injection pressure, injection duration, hole size, number of holes, and hole orientation.

RESULTS: submitted previously.

Task 7: Development of 4-Cylinder Product Specifications

STATUS: Complete

Revise specifications based on the prototype test results as appropriate. Woodward will review the initial product specifications modify them based on the results of the single-cylinder test program.

RESULTS: submitted previously.

Task 8: Design / Fabrication of 4-Cylinder Product Prototype

STATUS: Complete

Build a full-scale, operating, test prototype to evaluate system performance against specifications. Using the new product specifications, CSU will complete the design for the full 4-cylinder, GMV prototype in the EECL. For this task, installation issues such as fuel rail design and routing of the supply and return fuel lines are incorporated into the design decisions for the fuel injection system. The fuel injection system will be driven by a Woodward In-Pulse™ system, which will in turn be controlled by a human-machine interface (HMI) – an additional computer that will also be used for data acquisition. This HMI interface will allow the research team to vary the injection timing and duration.

For research purposes, each cylinder of the test engine will be equipped with a piezoelectric combustion pressure transducer. Analysis of the combustion pressure in each cylinder is one of the most important techniques for monitoring system performance. Direct analysis of the combustion pressure in each cylinder allows us to determine such important parameters as misfire, peak pressure, and location of the peak pressure of each combustion event. By further processing of the combustion pressure signals, we can determine the rate of combustion ("burn rate") and rate of heat release. These are fundamental parameters for evaluating an ignition system. Through statistical analysis of combustion parameters we can determine the cycle-to-cycle variability of the combustion process. One of the primary hypotheses of this project is that we should be able to achieve significant improvements in combustion stability.

RESULTS – see Task 9.

Task 9: Installation of 4-Cylinder Product Prototype

STATUS: Complete

Prepare the GMV prototype installation for testing. For laboratory evaluation, the micro-pilot fuel ignition system will be installed on the Large Bore Engine Testbed (LBET) at CSU's EECL. The construction of this facility was funded, to a large extent, by GTI and members of the U. S. natural gas pipeline industry. The purpose of the LBET is to "facilitate the development of new technologies for reducing emissions and fuel consumption from large bore engines." The facility has been instrumental in the development of several new products that have been commercialized and are now being implemented on pipeline engines. The LBET allows very flexible operation, allowing researchers to create the conditions found in a wide variety of different pipeline engines, from low BMEP piston-scavenged engines to highly turbocharged, high BMEP engines. The nozzles for the micropilot system will be installed in spark plug adapters in each cylinder. The engine will be equipped with heads that contain two spark plug holes per cylinder. A conventional spark plug will be used in the other spark plug hole to allow the engine to be easily started. During operation, we anticipate starting the engine on spark alone and then adding the micropilot as the engine comes up to temperature. During the shakedown phase of the project, we will determine whether the engine can be cold-started without spark assist. Once the engine is operating, the spark system will be turned off.

RESULTS: submitted previously.

Task 10: Test of 4-Cylinder Product Prototype

STATUS: Complete

Collect product performance data on a full-scale running GMV engine to compare results against product specifications. The LBET in the EECL at CSU is equipped with a wide variety of instrumentation for monitoring operating variables (speed, load, manifold pressure, manifold temperature, etc.), emissions, fuel consumption, and combustion parameters. Emissions at the EECL are measured in two ways. Criteria pollutants (NO_x, CO, hydrocarbons) are measured using a reference method 5-gas emissions bench. The production of hazardous air pollutants (HAPs) will also be monitored during the program. HAPs are a class of pollutants that have come under increased scrutiny over the past few years due to new standards on HAPs production which are scheduled for announcement by the EPA in late 2000. HAPs production will be measured using an FTIR (Fourier Transform Infrared) spectrometer.

After completion of shake down, all cylinders will be equipped with the fuel injection system and a set of tests will be done to determine the optimal injection timing at variety boost pressures for the engine. We plan to run these injection timing/ boost maps under different fuel injection scenarios, as described below.

Data to be gathered will include: engine operating parameters, standard 5-gas emissions (O_2, NOx, CO, CO_2, THC), HAPs emissions (primarily CH_2O); individual cylinder exhaust temperatures; individual cylinder combustion data: raw P-θ traces; ensemble averaged P-θ, peak pressures; location of peak pressures, θ_{0-10}, θ_{0-90}, locations of 10% burn, 50% burn & 90% burn; IMEP, standard deviation of combustion parameters, etc.

We plan to test system performance under a wide variety of operating conditions and utilizing different pilot injection parameters. Items to be examined experimentally include nozzle design, injection pressure, injection quantity, and injection timing. We will evaluate the effect of pilot fuel and compression ratio as part of the test program, although it is probable that these tests will be performed at the beginning of Phase II.

RESULTS: submitted previously.

Task 11: Phase I Report

STATUS: nearing completion, to be submitted by January 15, 2003.

In the year one report, CSU/Woodward will document activities leading up to the performance tests. The report is expected to document the literature review, development of specifications, testing of the single cylinder prototype, and startup / shakedown of the 4-cylinder product prototype.

Task 12: DOE Contractors Meeting

STATUS: CSU will be prepared to present Phase I results upon notification by NETL. This will be the most effective way to convey the results from the first year's work.

CSU/Woodward will report on the project results at the annual DOE Contractors Meeting in Morgantown, WV.

APPENDIX 2

“Micro-pilot Ignition
Prospects, Problems and Solutions – A Review”

Engines and Energy Conversion Laboratory
Department of Mechanical Engineering
Colorado State University

Micro-pilot Ignition
Prospects, Problems and Solutions – A Review

2001

ABSTRACT

A review is made of some of the main advantages and problems associated with the use of micro-pilot ignition in dual fuel engines. It is shown that such applications represent in principle a very attractive alternative to the spark ignition method due to the reduction in downtime, increased efficiency and lower emissions. Some of the more relevant and most recent research on pilot ignition is discussed together with an outline of the main factors that influence the ignition process. This is a basis for further research on micro-pilot ignition that will be carried out at the Engines and Energy Conversion Laboratory (EECL) at Colorado State University.

DUAL FUEL ENGINE

In a dual fuel engine, a gaseous fuel called the primary fuel is inducted along with the intake air and is compressed like in a conventional diesel engine. This mixture does not auto-ignite due to its very high self-ignition temperature. A small amount of diesel called the pilot is injected near the end of the end of the compression stroke to initiate combustion of the gas-air mixture. The combustion of the pilot leads to flame propagation in the gas-air mixture. A wide range of gaseous fuel can be used in dual fuel engine, with natural gas being the most used.

A major advantage of the dual fuel engine is that it produces less CO_2 compared to conventional engines. Further more, natural gas, which is thought to be abundant in Earth, may be regarded as a fuel of reliable long-term supply. Previously published reports [1] have confirmed that lean burn engines, while having high thermal efficiency, emit fewer pollutants. As a result, lean burn gas engines have been the subject of intensive research.

PILOT IGNITION

It is well known that methane, the main constituent of natural gas, has relatively excellent knock resistant properties that makes it very well suited fuel for high compression applications. Resorting to pilot injection to provide ignition will produce a very powerful and voluminous source of ignition that is well matched with the lean mixtures of methane and air. The somewhat slow flame propagation of methane is therefore speeded up through the high compression and the large ignition energy provided by the ignition of the pilot. Moreover, some modifications of a chemical nature to the methane-air mixture will be provided through the presence of the diesel vapor. Thus, leaner mixtures operation is possible to levels that are unheard of in spark ignition applications even of the high compression ration.

MICRO-PILOT IGNITION

The micro-pilot terminology relates to the energy-based percentage of pilot fuel that is used relative to the total amount of fuel injected into the cylinder. The quantity of pilot fuel that is regularly used in dual fuel engines varies between 4% and 10 %. Data obtained by Karim (Figure 1) has shown that the emissions are lower as the amount of pilot fuel decreases. Therefore, the term ***micro-pilot*** is introduced, which is defined as the pilot fuel, which amounts to 1% or less of the total energy contained by the fuel injected per cycle.

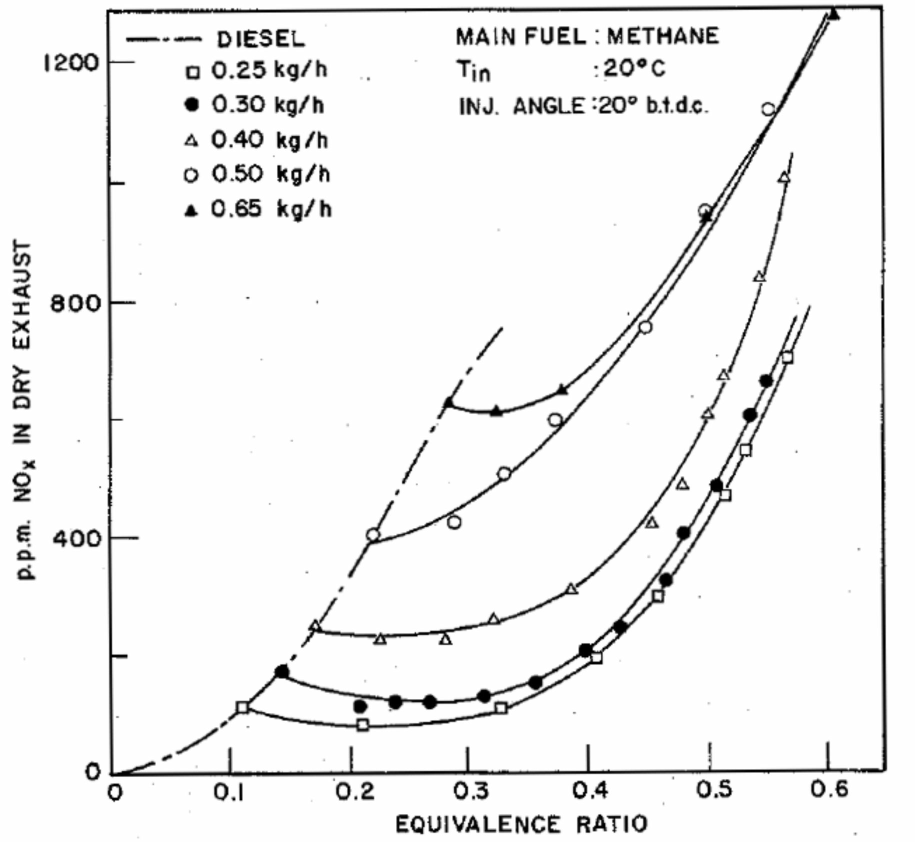


Figure 1 – *Effect of pilot quantity on dry exhaust NO$_x$ concentration in dual fuel operation with methane. Corresponding normal diesel operation is also shown.*

The use of a *micro-pilot* ignition method in a dual fuel engine, while reducing emissions, it raises new technological hurdles and, furthermore, makes more acute some of the present problems that the *pilot* ignition systems have. Therefore, it is imperative to have a very good understanding of the phenomena that influence the "regular" pilot ignition process and determine how they apply to the case of a micro-pilot system.

The approach for such an endeavor, due to the very large number of factors that need to be taken into account, is a combination of preliminary theoretical investigation backed by experimental data. Finally, data collected from the direct implementation and testing of a micro-pilot system is needed. In Figure 2, a typical dual fuel engine that uses micro-pilot as a source of ignition is shown.

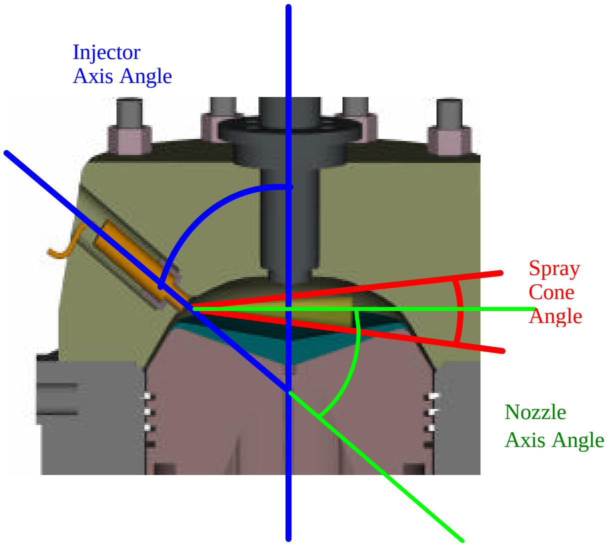


Figure 2 – *Cylinder of a dual fuel engine with micro-pilot ignition*

PROBLEMS. PROSPECTS. SOLUTIONS.

A micro-pilot fuel jet, in spite of being smaller than a "regular" pilot fuel jet, it still provides several times more energy than a spark ignition. When the micro-pilot fuel is injected into the combustion chamber and autoignition occurs, many individual ignition sources are created, allowing for more complete and rapid combustion of the natural gas-air mixture than with a single spark. However, limitations specific to dual fuel engines create a new set of issues that need to be overcome.

Combustion in a Dual Fuel Engine

The combustion process in a dual fuel engine tends to display a complex combination of features of both diesel and spark ignition engine operations, with elements that are unique to dual fuel operation. Combustion in the dual fuel engine is similar to that in an Otto-cycle engine, in that the bulk of the energy is produced by the combustion of a more-or-less homogeneous, preformed mixture of air and fuel. Dual fuel engines differ from spark-ignition Otto-cycle engines in that the ignition source for the premixed charge is not a spark, but the compression ignition and combustion of the diesel pilot fuel. Another distinction between many dual fuel engines and common spark ignition Otto-cycle engines is that dual fuel engines are seldom equipped with throttles to control the power output. Instead, power is controlled by reducing the concentration of natural gas in the premixed charge. From this point of view, as well as many other technical aspects, dual fuel engines resemble diesels more than spark ignition engines.

The introduction of a gaseous fuel with the air in the cylinder modifies greatly the mixture formation and the combustion process of the pilot fuel spray, which has the role of providing a deliberate source for ignition. Flames from the various ignition centers originating from the pilot fuel can propagate to varying degrees and rates throughout the surrounding gaseous fuel-air mixture. Problems encountered in dual fuel engine operation, such as poor light load performance, extent of variations in the length of the ignition delay, the incidence of knock at high load operation and high exhaust emissions, vary largely with the quantity of the pilot fuel employed, the type of the gaseous fuel used and its concentration in the cylinder charge.

The development of comprehensive combustion models for dual fuel engine operation has been so far very limited mainly due to the complex combustion processes involved. The application of most of these models is restricted to a limited range of operating conditions. For example, a single-zone combustion model developed by Thyagarajan et al. [Thyagarajian, V. and Babu, M.K.G., "A Combustion Model for a Dual Fuel Direct Injection Diesel Engine", Diagnostics and Modelling of Combustion in Reciprocation Engine, Proc.of COMMODIA Sym., P607, Tokyo, 1985] could only be used to predict the general combustion performance of the dual fuel engine such as pressure and power output.

Gao, et al. [Gao, X, Chen, J., Je, Z, Foster, D. and Borman, G.L., "Ignition Delay and Heat Release Analysis of an Ethanol Fumigated Turbocharged Diesel Engine", ASME, Paper No.83-DGP-1, 1983] developed a three-zone model to simulate the performance of a fumigated fuel engine. The cylinder was divided into three zones: a zone containing a homogeneous mixture of unburned fumigated fuel and air, a second zone containing unburned diesel fuel, and a third zone containing the products of combustion. This model, which assumed that the diesel spray entrained the same amount of ethanol and air mixture at all conditions, was used to predict the oxides of nitrogen emissions under a limited range of operating conditions. In the case of a dual fuel engine with a micro-pilot ignition system, the hydrodynamics of the spray shows that due to the small quantity of fuel injected, the entrainment of the surrounding air is very small. However, as the size of the pilot increases, the significance of the air entrainment for atomization, ignition delay and combustion increases significantly. Hence, Gao’s model has a limited application.

The quasi-two zone model developed by Karim et al. [Karim, G.A. and Liu, Z. "A Prediction Model for Knock in Dual Fuel Engines", Transactions of SAE, SAE 921550, 1992; Karim, G.A. and Zhaoda, Y., "Modelling of Auto-ignition and Knock in a Compression Ignition Engine of Dual Fuel Type", Proceedings of the Institution of Mech. Engineers, IMECHE, C430/035, PP141-147, 1991-11] is a relatively simple model, which was used to predict the autoignition and knock characteristics and overall engine performance of dual fuel engines near full load. The model cannot not be applied to predict exhaust emissions, nor the operation at light load when lean mixtures are employed. This is due mainly to the absence of measures for predicting variations with time of the temperature and composition within the cylinder. The production of exhaust emissions is a strong function of the distribution of the charge temperature and concentration within the cylinder. Hence, alternative approaches need to be developed involving multi-zone computational models to better simulate the complex nature of the combustion processes in dual fuel engines, especially at light load. Such models can provide in-cylinder temperature and concentration variations with time and better predict power output and efficiency.

The following describes a multi-zone thermodynamic model that was developed by Liu and Karim [Karim, G.A. and Liu, Z., "A Predictive Model for the Combustion Process in Dual Fuel Engines", Transactions of SAE, SAE 952435, 1995] to describe the combustion processes of dual fuel engines and predict aspects of their performance. The consequences of the interaction between the gaseous and diesel fuels and the resulting modification to the combustion processes are considered. A detailed kinetic scheme is employed to describe the oxidation of the gaseous fuel right from the start of compression to the end of the expansion process. The associated formation and concentrations of exhaust emissions are also established. The model not only can predict the onset of knock but also attend to the more demanding case of predicting the low load engine performance with the associated partial oxidation reactions and the production of exhaust emissions. Some corresponding experimental data [Khan, M.O., "Dual Fuel Combustion Phenomena", Ph.D Thesis, Mech. Engr., London University, 1969; Azzouz, D., "Some Studies of Combustion Processes in Dual Fuel Engines: The Role of Pilot Liquid Injection Characteristics", M.Sc. Thesis, Mechanical Engineering, University of London, 1966] are used to compare with predicted values obtained using the model.

Liu and Karim describe the dual fuel engine phenomena as follows. Once the gaseous fuel is admitted into the cylinder and mixed with the intake air, the premixed gaseous fuel-air charge is subjected increasingly with time during compression to higher temperatures and pressures as top dead center is approached. Some significant reaction activity of the gaseous fuel may proceed during the compression process and produce some intermediate species such as radicals, carbon monoxide and formaldehyde. These can have profound effects on the subsequent combustion processes of the engine. In order to describe the preignition reactions of the gaseous fuel before the injection of the pilot fuel, the whole charge of the homogeneously premixed gaseous fuel and air in the cylinder is treated as a single zone. The detailed chemical reaction kinetics of the gaseous fuel-air charge are then employed to follow the progress of its reaction right from the start of the compression process.

When the pilot diesel fuel is injected into the combustion chamber under high injection pressures, the pilot fuel is atomized and distributed within its spray cone. Some gaseous fuel and air are entrained into the pilot fuel spray due to the jet action of the pilot fuel. Hence, the ignition and combustion processes of the diesel fuel are modified significantly by the possible participation of the gaseous fuel. The entrainment of the gaseous fuel into the pilot fuel and the flammable regions is assumed to start at pilot fuel jet break-up. The entrainment rate and the amount of entrained gaseous fuel-air charge depend on the injection conditions, the quantity of the pilot fuel and the concentration of the gaseous fuel in the cylinder charge.

Ignition of the fuel charge is assumed to take place first within a flammable region where the overall reaction rates of the diesel and gaseous fuel mixture are greatest. Then, following ignition the combustion of the mixture is viewed to develop in two directions. The first is through flame propagation within the flammable zone. The other is through diffused combustion of the pilot diesel fuel, which takes place within the core of the pilot fuel. Thus, the mixture in the cylinder can be viewed to be divided into a number of zones, as shown schematically in Figure 3.

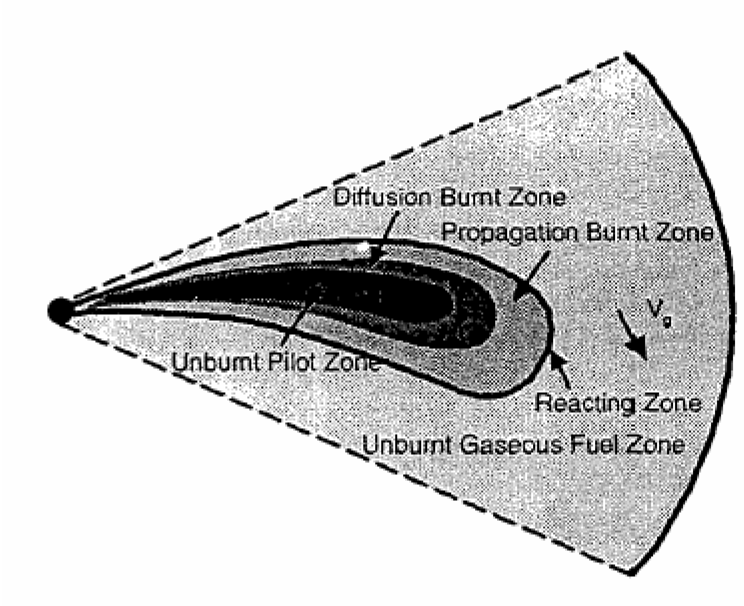


Figure 3 - *A Schematic Zone Division During the Combustion Process*

There are two "unburned zones" in the combustion chamber. The first is a "pilot fuel unburned zone" in which the distribution of fuel/air ratio is too rich to burn immediately. The unburnt gaseous fuel-air mixture within the surroundings is considered to form a "gaseous fuel unburnt zone" which is compressed and heated by the combination of the movements of the piston and flame front. Furthermore, there are two "burned zones" within the combustion chamber. The diffusion combustion of the pilot diesel fuel and part of the gaseous fuel, which takes place towards the core of the pilot fuel, forms a "diffusion burned zone" which is assumed to burn essentially stoichiometricly. Fresh mixtures of diesel and gaseous fuels and air are entrained from the surroundings to this burned zone. The flame propagation towards the flammable region of the mixture forms a "propagation burned zone". Once the fuel charge is entrained from the unburnt zone to the burnt zone, its energy is assumed immediately to be released at the edge of the burnt zone. The division of the cylinder charge into these zones, which undergo different combustion processes, produces different charge temperatures and combustion products within the cylinder.

For normal flame propagation, there exists a very thin reaction zone within the flame front as shown in Figure 4. The reaction rate of the gaseous fuel-air mixture within this reacting zone is sufficiently fast that the gaseous fuel-air mixture releases immediately its energy at the edge of the flame front and gets converted to combustion products after passing through the reacting zone.

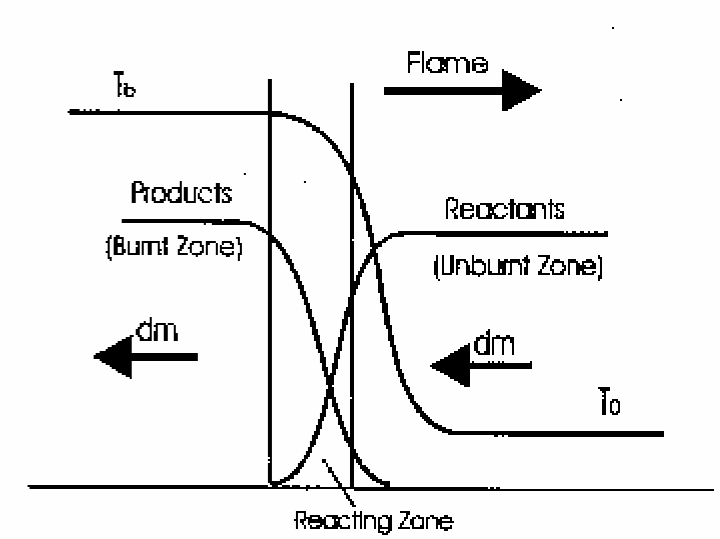


Figure 4 – *A schematic flame propagation and reacting zone*

However, when the dual fuel engine is operated at very light load while using a relatively small pilot quantity, combustion is confined to the pilot fuel spray zone since the flame cannot propagate throughout the very lean fuel mixture. With the development of the expansion process, the charge mean temperature in the burnt zone is decreased initially slowly and later rapidly, while the reaction rates of the over lean gaseous fuel-air mixture in the reacting zone of the flame front is increasingly slowed down, resulting in the partial oxidation of the gaseous fuel within the reacting zone. It would be expected that increasingly more unconverted gaseous fuel and carbon monoxide will accumulate in the reacting zone due to the incompleteness of the reaction processes and will survive eventually to the exhaust stage. In order to describe the partial oxidation reactions of the gaseous fuel at light load operation involving very lean mixtures, a reacting zone that lies on the boundary between the flame propagation burnt zone and the surrounding unburnt zone is added, as shown schematically in Figure 4. Under normal engine operating conditions, this zone tends to be very thin and will have a negligible mass due to its high charge temperature and fast reaction rates. Only when the charge temperature in the reacting zone decreases to a certain level, the reaction rates of the gaseous fuel are sufficiently slowed down that the partial products produced can survive in the reacting zone. As a result, the mass of this zone accumulates and was taken into consideration in the model by Liu and Karim. The overall structure of the computational model is shown in Figure 5.

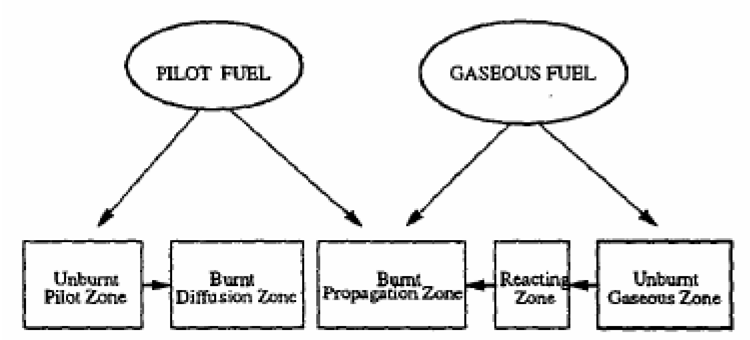


Figure 5 - *The Scheme of the Five-Zones Combustion Model*

The energy released by the combustion of the pilot fuel is assumed to be divided into two parts. The first part is due to the premixed combustion of the pilot diesel fuel, which takes place in the flammable zone. The second part is due to the diffusion combustion of the pilot diesel fuel that takes place in the diffusion burned zone.

Based on the distribution of the gaseous fuel-air mixture in the cylinder, the combustion rates of the gaseous fuel-air mixture in the charge can be considered to consist of three parts (Figure 6) as described by Karim et al. [Karim, G.A. and Liu, Z. "A Prediction Model for Knock in Dual Fuel Engines", Transactions of SAE, SAE 921550, 1992; Karim, G.A. and Zhaoda, Y., "Modeling of Auto-ignition and Knock in a Compression Ignition Engine of Dual Fuel Type", Proceedings of the Institution of Mech. Engineers, IMECHE, C430/035, PP141-147, 1991-11]. They are the combustion of the gaseous fuel-air mixture in the diffusion burned zone due to the diffusion combustion of the pilot fuel (I), the premixed combustion of the gaseous fuel-air mixture in the flammable zone (II) and the combustion of the remaining gaseous fuel-air mixture outside the pilot fuel spray zone due to flame propagation and turbulent mixing (III).

At light load, when very lean gaseous fuel-air mixtures are employed, the bulk of the combustion energy release comes about from the ignition and combustion of the pilot zone (I) and from the energy release associated with the combustion of some of the gaseous fuel-air mixture that is entrained into the burning pilot combustion zone and from the immediate surroundings of such a zone where higher temperatures and relatively richer mixture regions may evolve. As shown in Figure 6, only relatively little contribution to the energy release may be expected from the pilot zone, since with very lean mixtures no consistent flame propagation can take place from these ignition centers. An increase in the quantity of the pilot injected for very lean mixtures operation will tend to increase more than proportionally the total energy released and its associated rates. Greater amounts of gas-air mixtures will then be oxidized due to the larger amount of mixtures entrained within the pilot combustion zone and as a result of the thickening of the burning regions in their vicinity. Greater energy release and rate will also be evident due to some partial flame propagation and increase preignition reaction activity of the rest of the charge.

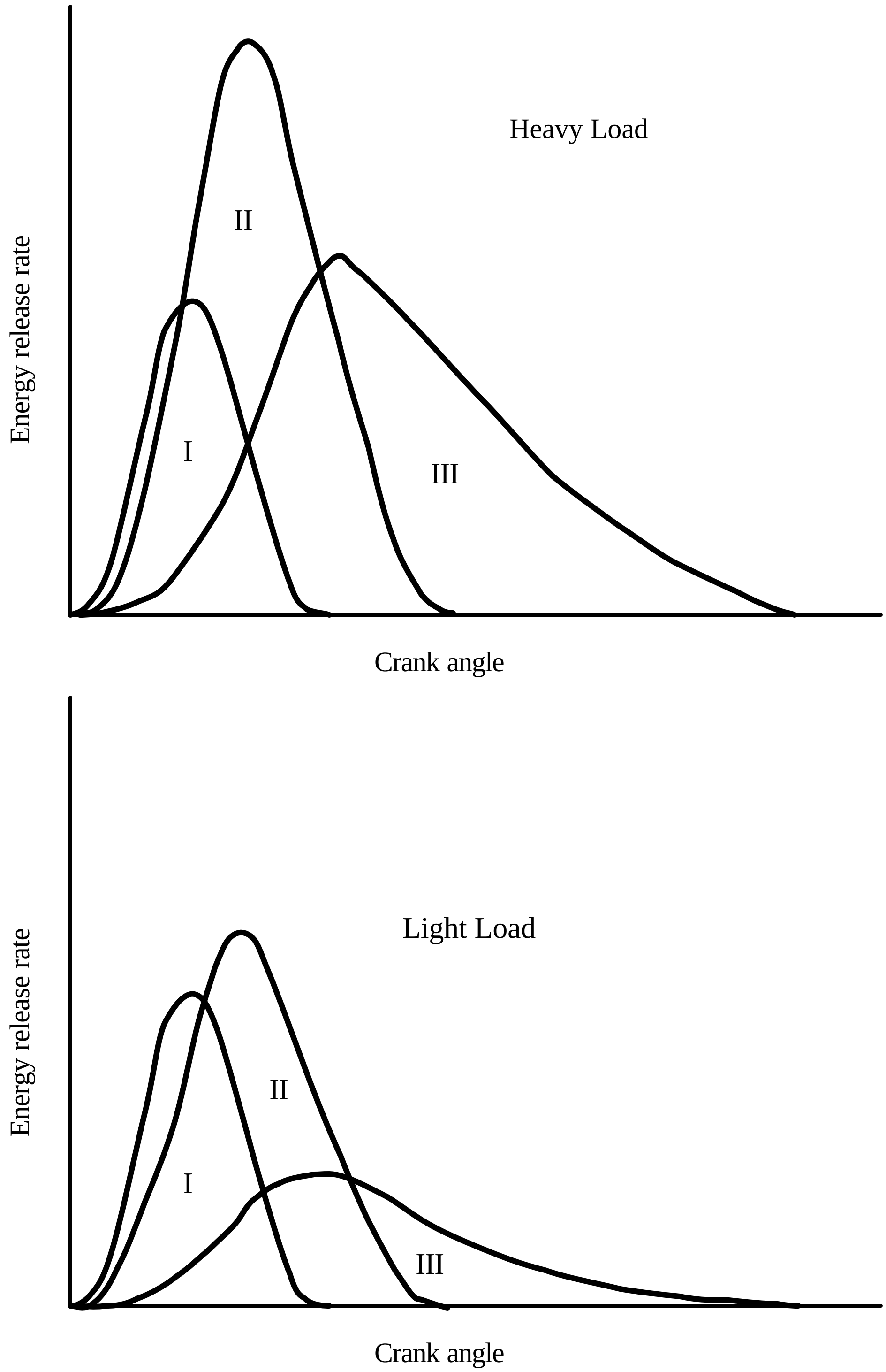


Figure 6 – *Schematic representation of the contribution of the various parts of the combustion process to the energy release rate under heavy load conditions and light load conditions, in a typical dual fuel engine*

Increasing the concentration of the gaseous fuel further will permit eventually, after pilot ignition, flame propagation to proceed throughout the rest of the charge resulting, as shown in Figure 7 [Karim, G.A., "A Review of Combustion Processes in the Dual Fuel Engine – The Gas Diesel Engine", Prog. Energy Cobust. Sci., Vol. 6, pp. 277-285, 1980], in a sudden increased contribution to the total energy release.

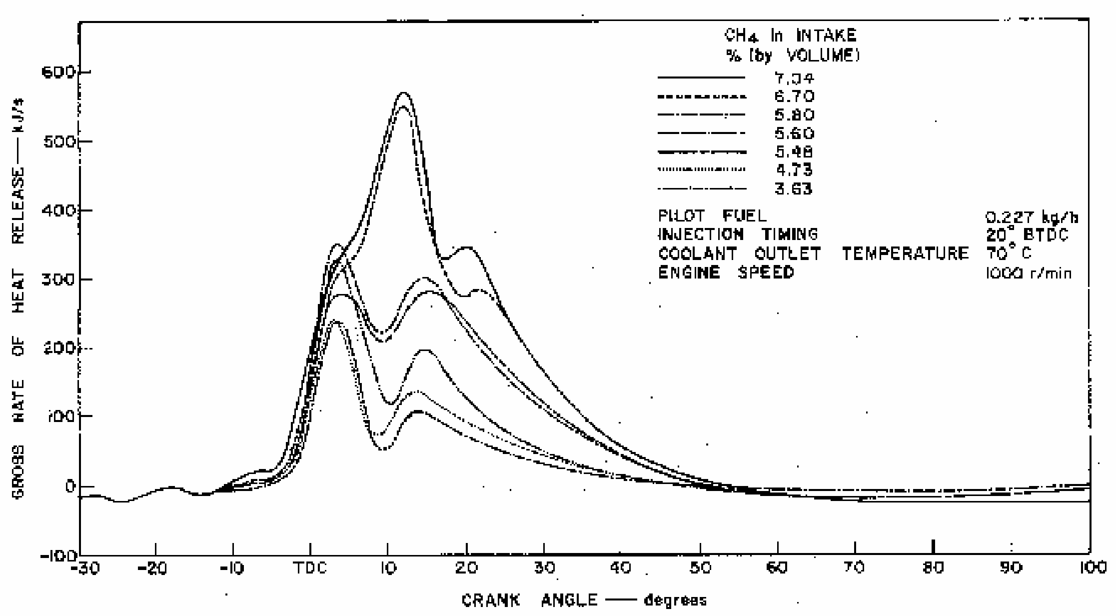


Figure 7 – *Derived "gross heat release" data versus crank angle for a variety of concentrations of methane in the intake at constant pilot fuel quantity and injection timing*

Continued increase in the concentration of the gaseous fuel in air will result in a greater overlap between the second and third energy release regions and will lead to their amalgamation, further releasing much of the energy immediately following the commencement of the autoignition of the pilot. When the energy release rates become very rapid and associated with the autoignition of the charge they have been considered to indicate the onset of knock. The incidence of knock represents usually the practical limit for power output in dual fuel engines.

The combustion rates of the gaseous fuel-air mixture in the burned zones are under the direct influence of the combustion of the diesel fuel and will have very similar heat release rates to those observed for the diesel fuel. The combustion rate of the gaseous fuel-air mixture in the surrounding zone is dependent mainly on the concentrations and quality of the pilot and gaseous fuels in the cylinder charge. With an increase in the concentration of the gaseous fuel, the size of the flammable zone is enlarged and the mixing rates from the unburned zone to the burned zone are increased. A continued increase in the concentration of the gaseous fuel allows the flammable zone to extend into the whole gaseous fuel region. This may even lead to the onset of autoignition of the gaseous fuel in the surrounding zone before the flame front arrives.

Knock Phenomena in Dual Fuel Engines

When very high power outputs or very high intake temperatures and pressures are involved, the problem of knock may be encountered. Karim [Karim, G.A., "A Review of Combustion Processes in the Dual Fuel Engine – The Gas Diesel Engine", Prog. Energy Cobust. Sci., Vol. 6, pp. 277-285, 1980] determined that knock phenomena in dual fuel engines is of autoignition nature, most likely of the gaseous mixture in the neighborhood of the ignition centers.

For a stable operation, the dual fuel engine feed mixtures lie normally within a narrowing range that changes with the charge temperature as shown in Figure 8.

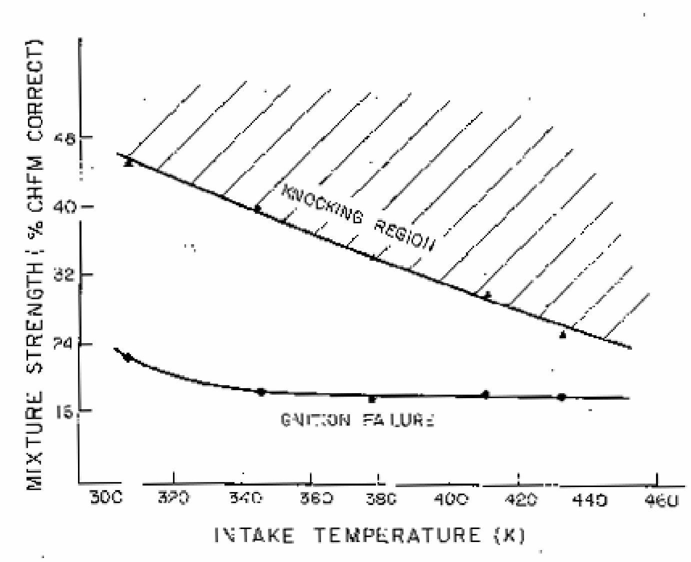


Figure 8 – *A typical variation of the knock limited mixture strength with charge temperature, with hydrogen as a fuel. The pilot quantity is constant*

The working region is bound on one side by mixtures that produce excessive rates of pressure rise and knock, and on the other by mixtures that produce erratic engine running, and may lead ultimately to ignition failure. Experimental evidence indicates that the transition from non-knocking to knocking is sharp, well defined, repeatable, and accompanied by an abrupt change in the shape of the pressure diagram, with high frequency pressure oscillations. As shown in Figure 7, important changes in the derived rate of equivalent "heat release" by combustion from pressure data can also be observed.

According to Karim, the onset of knock can be delayed somewhat through the lowering of the induction temperature and pressure, water jacket temperature and pilot quantity. Lower compression ratio and slightly later fuel injection can also be employed. Enhancing the quality of the diesel fuel by increasing its cetane number will have a relatively minor effect. However, the quality of the gaseous fuel employed will have a very significant role, especially through the presence of various hydrocarbon impurities or through the presence of some hydrogen gas. The presence of diluents in the methane such as carbon dioxide, steam or nitrogen will suppress the onset of knock. Moreover, the use of smaller pilot quantities or delayed injection will aid in suppressing the onset of knock. This would clearly point out to the desirability of having a variable diesel pilot to methane ratio over the whole load range. At light loads, a relatively large pilot is used. At the higher loads, the pilot quantity can be reduced, providing that adequate cooling is maintained for the micro-pilot fuel injection system. As it can be seen in Figure 9 [Karim, G.A., "A Review of Combustion Processes in the Dual Fuel Engine – The Gas Diesel Engine", Prog. Energy Cobust. Sci., Vol. 6, pp. 277-285, 1980], considerably more power could be obtained at the rich mixture than at the lean mixture knock limit.

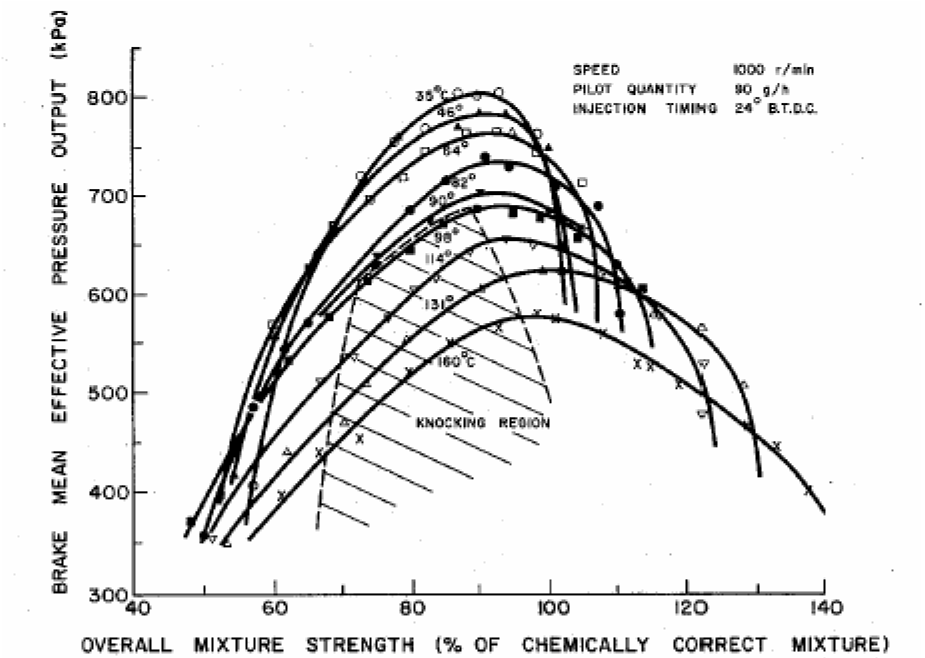


Figure 9 – *Variation of power output with the overall mixture strength for different intake temperatures but fixed pilot quantity; the knocking region is shown for methane as a fuel*

Ignition Delay

The variation in the length of the ignition delay have a profound and controlling effect on the subsequent combustion processes and hence on almost every feature of engine performance.

By definition, the ignition delay decreases with an increase in the diesel fuel cetane number (Figure 10 – [Gunea, C. Razavi, M.R.M., Karim, G.A. – “The Effects of Pilot Fuel Quality on Dual Fuel Engine Ignition Delay”, SAE 982453, 1998]).

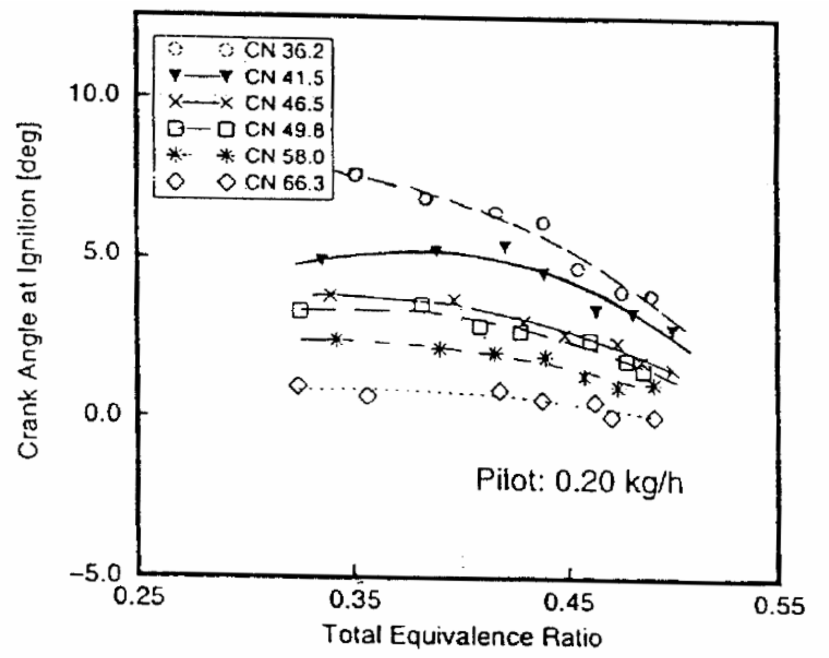


Figure 10 – *Variation of the ignition point with total equivalence ratio for dual fuel operation with methane when using a range of cetane number diesel fuels with a pilot quantity of 0.2kg/h*

As expected, increasing the pilot fuel quantity, as shown in Figure 11, resulted in an overall decrease in the ignition delay for the whole range of dual fuel operation with gaseous fuels. On the other hand, for small gaseous concentrations, lowering the intake mixture temperature showed an alarming effect in delaying the ignition of the pilot. This is due to the decrease of the of the compression temperature which has a detrimental effect on fuel vaporization in the first part of the delay period and on the chemical component of the ignition delay later on.

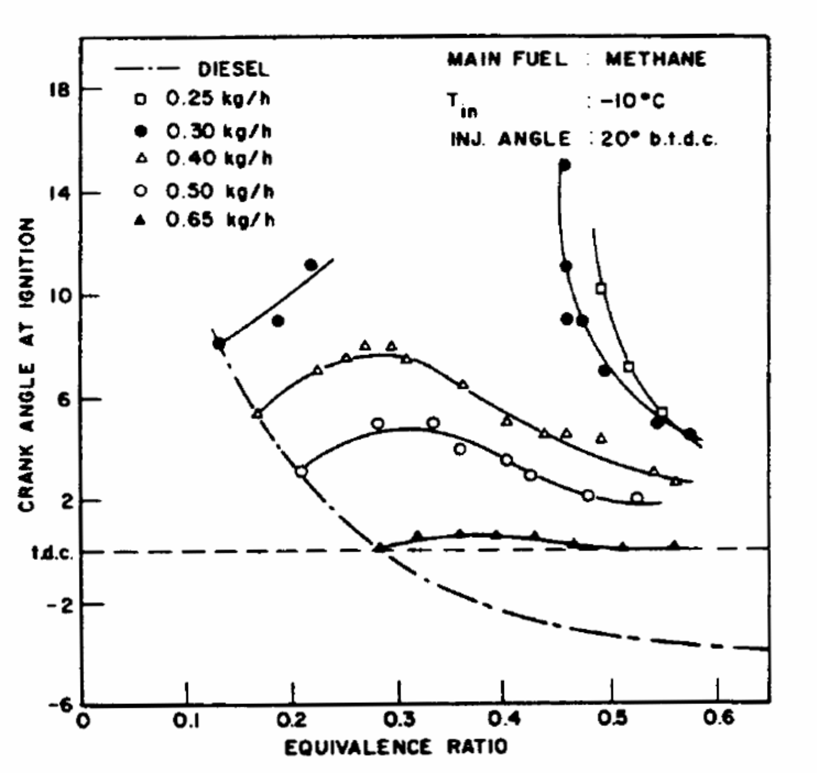


Figure 11 – *Effect of pilot quantity on ignition delay in dual fuel operation with methane. Corresponding normal diesel operation is also shown*

An advance in the injection timing generally, resulted in earlier ignition. Advancing injection to avoid erratic running and ignition failure was of limited success particularly at low intake temperatures.

The presence of the gaseous fuel influences both the pre-ignition and post-ignition processes. The ignition dependence on the gas properties is a function of the fuel used, its concentration, and the operating conditions. Karim [Karim, G.A., "A Review of Combustion Processes in the Dual Fuel Engine – The Gas Diesel Engine", Prog. Energy Combust. Sci., Vol. 6, pp277-285, 1980] determined that the ignition delay of the pilot fuel increases considerably with the addition of the gaseous fuel, and reduces later with further gas addition, as shown in Figure 12.

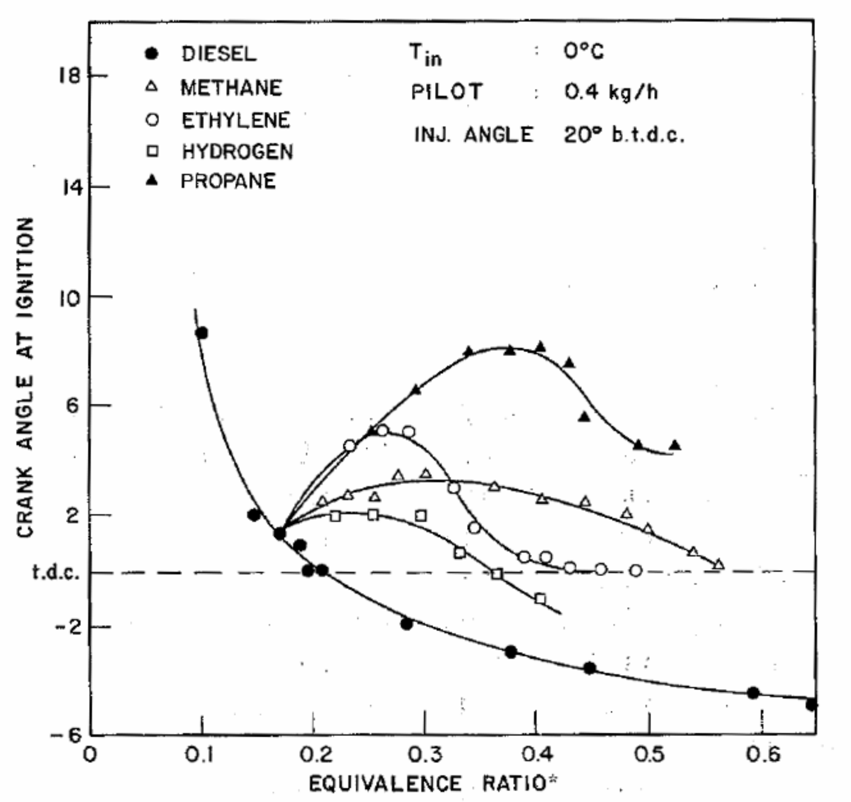


Figure 12 – *Comparison of ignition delay for an intake mixture temperature of 0^{o}C in dual fuel operation with various fuels (*Equivalence ration is based on total fuel employed, i.e. (m_{pilot}+m_{gas}/m_{air}) relative to the corresponding value of the chemically correct mixture.)*

This increase is far in excess of that caused by the reduction of the partial pressure of oxygen by the addition of the gaseous fuel (i.e., methane) or by the reduction in the temperature of the charge at around the top dead center position, as a result of the higher overall specific heat of the charge. As shown in Figure 12, the induction of hydrogen in the intake air appeared to have the smallest effect on prolonging the ignition delay among the four fuels considered. Propane addition resulted in the largest increase, while methane and ethylene were relatively moderate. Two inert gases, nitrogen and carbon dioxide were also introduced in the intake air to determine the relative roles of the physical and chemical parts of the ignition delay. Figure 13 shows that neither the nitrogen nor the carbon dioxide additions could produce a comparable increase in the ignition delay.

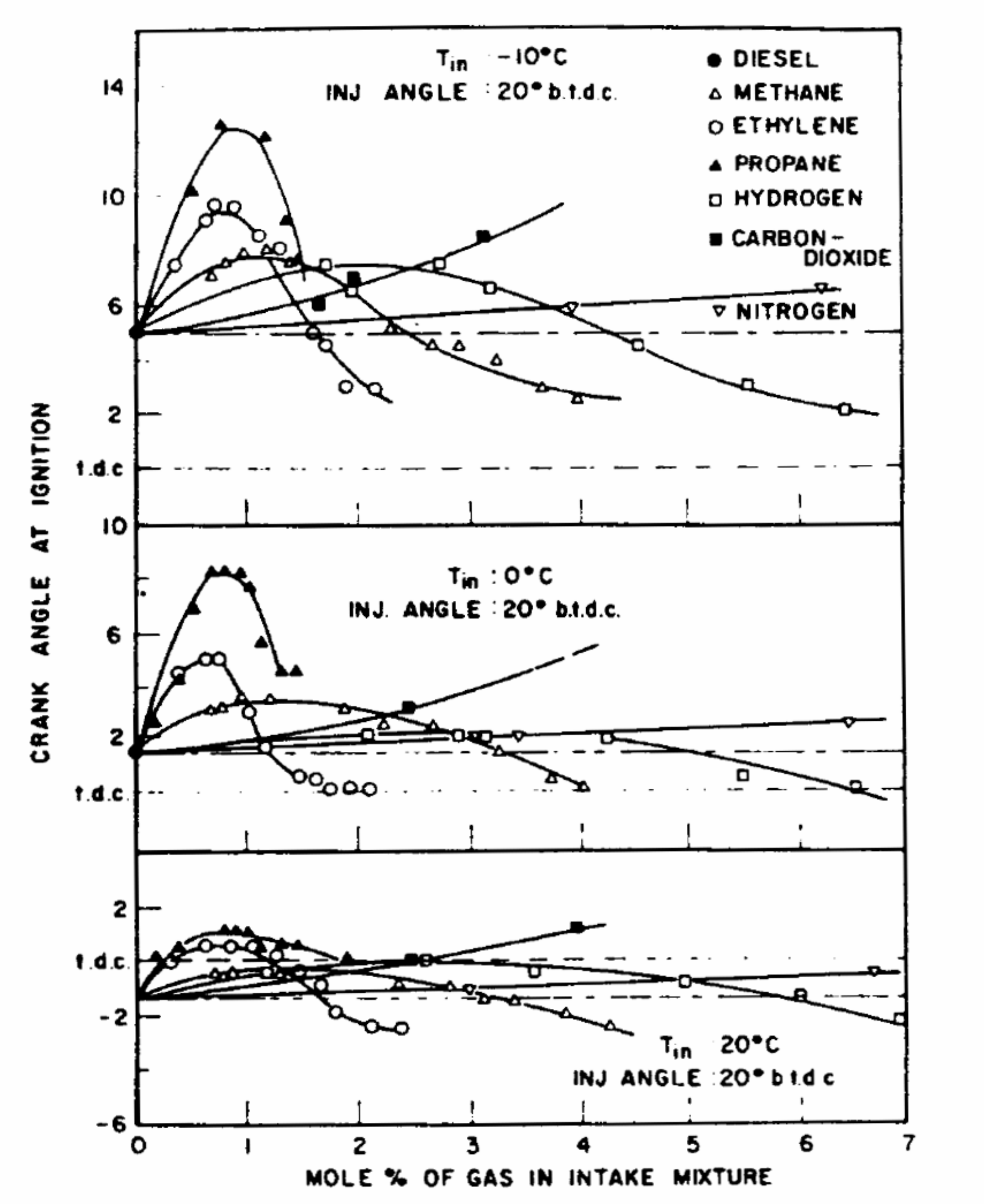


Figure 13 – *Variation of ignition delay of the pilot fuel when introducing various gases with intake air. Corresponding normal diesel operation is also shown*

As a result, Karim concludes that the gaseous fuel must participate actively in an unknown manner in the pre-ignition chemical processes of the pilot fuel, to bring about these variations in the delay. Later on, Karim [Karim, G.A., Abraham, M., Jensen, L. – "An Examination of the Role of Formaldehyde in the Ignition Processes of a Dual Fuel Engine", SAE 912367, 1991] mentioned that the reasons for this relate to the pre-combustion chemistry of the gas-air mixture. During the compression, the gaseous fuel undergoes pre-flame reactions. The gaseous fuel and its partial oxidation products participate actively in the pre-ignition chemistry of the pilot fuel. The presence of a small amount of formaldehyde –formed by partial oxidation of methane in the charge – has been shown to increase the ignition delay experienced by the diesel fuel due to the competition between the diesel fuel vapor and the gaseous fuel for active radicals. When a larger amount of the gaseous fuel is added, the pre-combustion reactions produce significant amounts of energy and radical species during compression, adding the ignition of the pilot fuel.

The ignition delay in a dual fuel engine depends strongly on both the quantity and quality of the pilot fuel used. Dual fuel engine performance is improved with the employment of high cetane number pilots. Their use permits the employment of smaller pilot quantities and can improve engine operation and emissions.

Emissions

An analysis of the exhaust gas of a dual fuel engine normally indicates that considerable proportions of the fuel gas can survive the combustion process when it is fed to the engine at either well below or above some limiting concentrations. These limits, which are generally identified with the effective flammability limits of the mixture, are a function of both the fuel and operating conditions used.

Figure 14 [Badr, O., Karim, G.A., Liu, B., "An Examination of the Flame Spread Limits in a Dual Fuel Engine", Applied Thermal Engineering, Vol. 19, No. 10, 1999] shows the variations with total equivalence ration of the concentrations of unconsumed methane and carbon monoxide in the exhaust gas of the engine for different pilot quantities. It can be seen that there is a limiting equivalence ration beyond which the exhaust emissions of the carbon monoxide and the unconverted methane become virtually unaffected by the pilot quantity. This is indicative of the equivalence ratio limit for successful flame propagation from the pilot ignition centers.

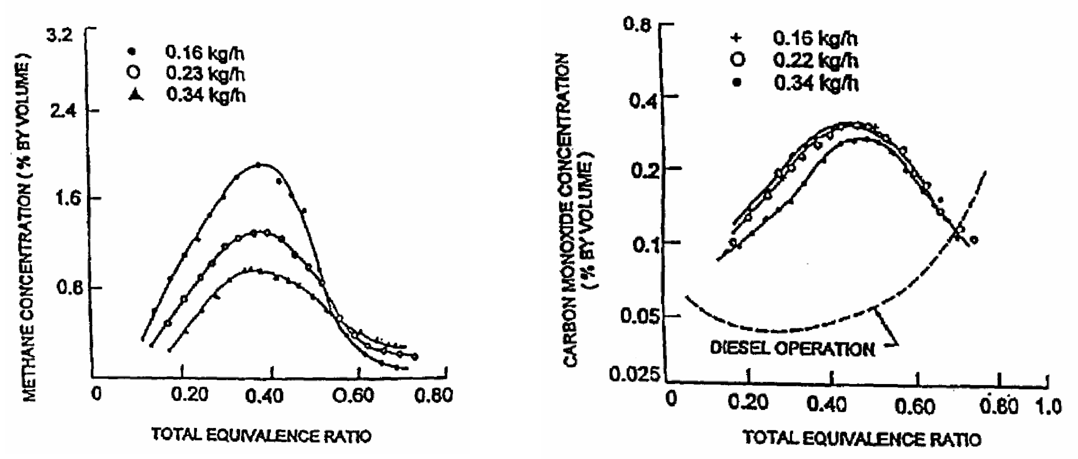


Figure 14 –*The variations of the exhaust gas concentrations of methane and carbon monoxide with total equivalence ratio for different fuel quantities at ambient intake and 1000 rpm*

Figure 15 is a typical representation of the observed exhaust smoke density, in Hartridge Units, from dual fuel operation with methane addition where there appears to be hardly any detectable smoke even for low intake temperatures and even with high loads. Similar trends were observed by Karim for different pilot quantities and different gaseous fuels.

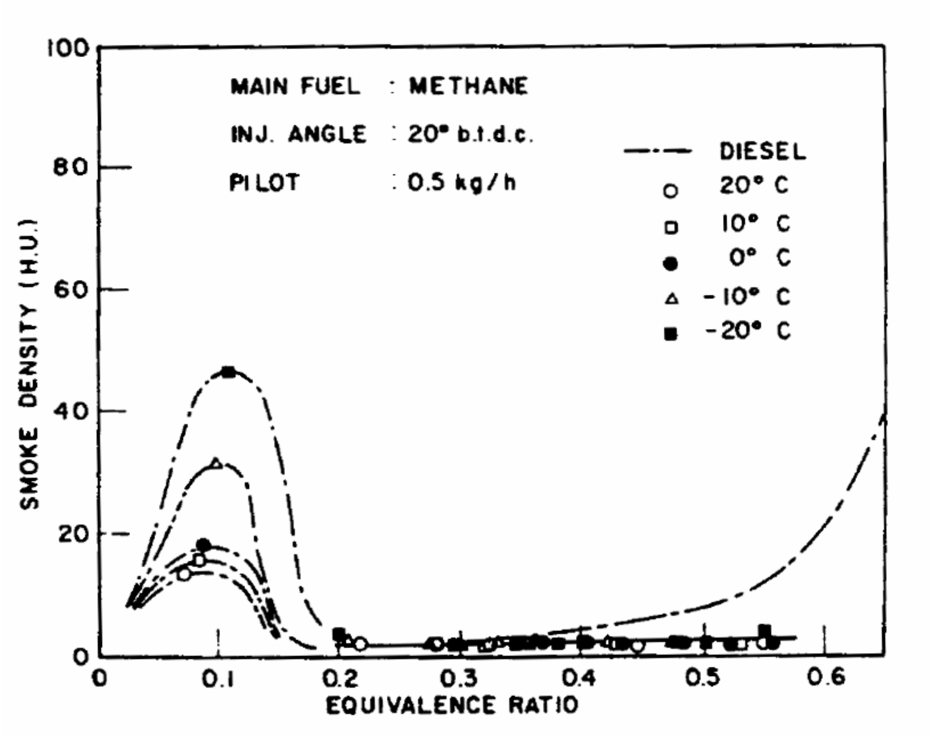


Figure 15 – *Variation of apparent exhaust smoke density for a range of intake mixture temperature in dual fuel operation with methane. Corresponding normal diesel operation is also shown*

Figure 16 shows a schematic variation of the exhaust emissions of carbon monoxide and methane with the overall equivalence ratio for a fixed pilot quantity. Several main operational regions can be identified.

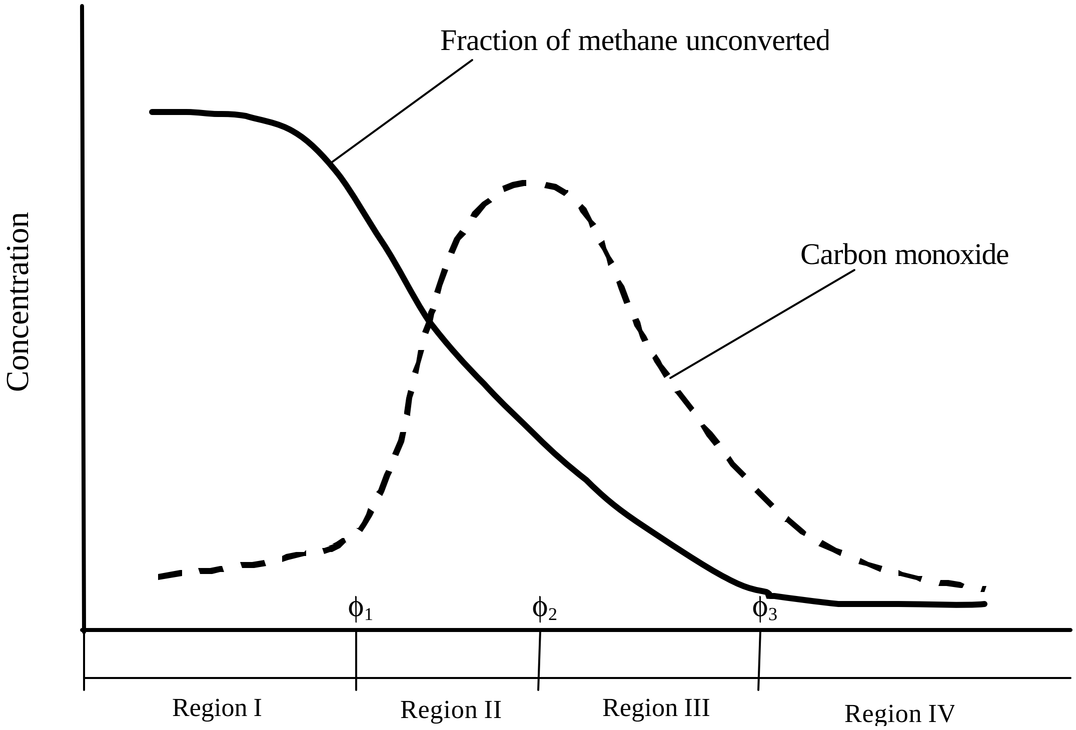


Figure 16 - *Schematic variation of the exhaust emissions of carbon monoxide and methane with the overall equivalence ratio for a fixed pilot quantity*

The first region is associated with extremely low gaseous fuel admission where the exhaust emissions of carbon monoxide and the fraction of the methane consumed are very small. In the second region, following an increased admission of the gaseous fuel, the consumption of the methane and the production of carbon monoxide begin to increase rapidly with the continued increased admission of the methane. Later on, these begin to decrease in regions III and IV. These limiting values of equivalence ratio can be identified as ϕ_1, ϕ_2, and ϕ_3. Their significance is as follows:

- ϕ_1 – the start of significant local partial oxidation;
- ϕ_2 – flame initiation;
- ϕ_3 – the spread of propagating flames within the gaseous fuel-air charge.

The complex chemical and physical interactions that take place to produce these regions require the consideration of a number of related processes. These would include:

- Preignition reaction activity of the gaseous fuel-air mixture during compression.
- Pilot injection processes and subsequent formation of the flammable envelope.
- Progressive reactions during the ignition delay of the pilot.
- Formation of ignition centers and subsequent reactions with the gas-air mixture that may lead to partial or complete flame propagation.

When operating with very fuel lean mixtures at light load most of the energy release comes from the combustion of the pilot and the gaseous fuel entrained within its envelope as well as adjacent reacting zones where high temperature may evolve. The contribution of the bulk surrounding lean gaseous fuel-air mixture to the energy release remains small [see Karim, "An examination of some measures for improving the performance of gas fuelled diesel engines at light load"]. For less lean mixtures, the concentration of the gaseous fuel may become sufficiently high to permit flame propagation throughout the entire charge within the time available to contribute significantly at a more gradual rate to the overall energy release.

Generally, the oxidation of a fuel such as methane proceeds sequentially via the formation of formaldehyde followed by carbon monoxide and the subsequent conversion to carbon dioxide and water vapor. For sufficiently fuel rich mixtures yielding high temperatures, good conversion of the methane-air mixture to completion takes place with little carbon monoxide and unconverted methane appearing in the exhaust. For less rich mixtures producing combustion with moderately high temperatures, a substantial amount of the carbon monoxide produced cannot be converted in the time available to carbon dioxide. However, for sufficiently lean mixtures, the charge temperature may be so low that no significant reactions proceed, leaving the bulk of the methane unconverted and producing insignificant amounts of carbon monoxide in the exhaust.

In region I, associated with very low equivalence ratios, carbon monoxide is produced at very low levels and comes mainly from the combustion of the pilot. The contribution made by the surrounding zone of the gaseous fuel-air mixture is very small. In region II, some of the exhaust carbon monoxide zone originates increasingly from the preignition reactions of the gaseous fuel within the unburned zone, yet they are incapable of leading to flame propagation within the time available, in spite of the presence of ignition centers. Further increases in equivalence ratio beyond the value ϕ_2 permit some flame propagation within the methane-air mixture. In region III, further increases in the admission of the gaseous fuel produces extensions to the size of both the pilot envelope and the adjacent reacting zone and extends the flame propagation into a larger fraction of the cylinder charge. With increased gaseous fuel admission the flame propagation continues to extend into further regions of the charge until at ϕ_3 it extends essentially to all parts of the combustion chamber. Further increases in the gaseous fuel concentration within region IV produce essentially proportionally high rates of heat release, leading to high cylinder pressures and increased power output.

Light Load

Dual fuel engines are seldom throttled, which requires the use of very lean fuel-air mixtures at light load. At light load conditions, the bulk of the energy released comes from the micro-pilot fuel combustion and any gaseous fuel-air mixture that is entrained in the pilot zone at the time of the ignition. The remaining gaseous fuel-air mixture is very lean which slows propagation of the flame front and makes complete combustion difficult, if not impossible to attain. Poor combustion brings about losses in thermal efficiency as well as increased exhaust emissions. In order to ensure complete combustion of the gaseous fuel-air mixture, it then becomes necessary to increase the pilot quantity to a greater amount than would be required at high load conditions [10]. However, according to Figure 1, a higher quantity of micro-pilot fuel increases the emissions level.

Methods have been investigated to improve dual fuel engine performance and emissions at light load. Gebert et al. [11] have tried several methods, including injection timing optimization, skip firing, and turbocharger air bypass. It was found that advancing injection timing increased NO_x production, while retarding the injection timing resulted in reduced NO_x, but increased hydrocarbon (HC), CO, and smoke emissions as well as reduced thermal efficiency. Skip firing requires the engine to run on a reduced number of cylinders at light load to bring the fuel-air ratio closer to stoichiometric in the cylinders that are used. This was determined to be very effective method to improve light load operation and even allowed idle operation with a 95% diesel substitution of the natural gas. An observed problem with skip firing was rough operation that caused visible shaking of the engine on the dynamometer.

At some engine speed and load points it is desirable to redirect the turbocharger boost to decrease the mass of air filling the combustion chamber and therefore increase the equivalence ratio of the mixture. Turbocharger boost bypassing was tried and shown to reduce HC and CO emissions. Daisho et al. [12] used hot and cooled exhaust gas recirculation (EGR) to increase the fuel-air ratio and thus improve combustion at light loads. It was found that hot EGR at light loads improved the thermal efficiency due to the charge temperature increase. Hot EGR also reduced NO_x and smoke formation. Cooled EGR gave slightly lower thermal efficiency but provided an even greater NO_x reduction than hot EGR. Along similar lines, Poonia et al. [13] and Karim's [14] research of light load dual fuel operation produced some more suggestions for improving performance and emissions. They suggested fuel charge stratification to produce a slightly richer gaseous mixture in the areas surrounding the pilot zone. This would allow a greater percentage of the gaseous mixture to be burned before the lean limit is reached. The use of auxiliary fuels such as hydrogen or gasoline vapor was recommended. This, however, adds great complexity to a dual fuel engine. Finally, reducing the engine operational speed was suggested to increase the time that the piston remains near the top dead center and therefore increase the time for ignition and flame propagation through the lean fuel-air charge. Because the ignition delay increases as the fuel-air mixture becomes leaner, reduced engine speed would minimize the detrimental effects of this delay increase.

Pilot Fuel Quantity

The quantity of pilot fuel injected affects dual fuel engine performance and emissions, especially at light loads (Figure 1, 11, 14). According to Gebert [Gebert, K., Beck, N. J., Barkhimer, R. L., Wong, H., " Development of Pilot Fuel Injection System for CNG Engine," SAE paper 961100, 1996] the micro-pilot fuel quantity (Q_p, mm^3/inj) influences engine performance as following :

1. Gaseous emissions decrease with a decrease in Q_p, especially CO and HC. If lambda-gas is controlled and held constant (using turbo-air bypass valve, for example), NO_x emissions are decreased with a reduction of pilot delivery.
2. Visible smoke is significantly decreased with reduces in Q_p.
3. Combustion duration is increased when Q_p is decreased.
4. Ignition delay increases when Q_p is decreased agreeing with other studies on ignition delay in dual fuel engines [18].
5. Maximum cylinder pressure decreases when Q_p is decreased.
6. Maximum cylinder pressure rate of rise is reduced when Q_p decreases.
7. Maximum rate of heat release decreases when Q_p decreased.
8. Engine thermal efficiency was not significantly affected by the pilot fuel quantity. In some cases, a slight reduction in Break Specific Energy Consumption (BSEC) was observed when Q_p decreased. At minimum Q_p, BSEC increased slightly, probably due to deteriorated spray quality.

The volumetric concentration of the gaseous fuel in air at the flame spread limit (FSL) which corresponds to the equivalence ratio ϕ_3 (Figure 16), as indicated earlier, identifies the boundary for the commencement of satisfactory engine operation and improved emissions. This limit represents the minimum concentration of the gaseous fuel in air for which flame propagation appears to spread throughout the entire cylinder charge. On the other hand, the equivalence ratio of the charge associated with the observed peak value of the concentration of carbon monoxide exhaust emissions, corresponding to ϕ_2 in Figure 16, may be considered as indicative of the commencement of some limited flame propagation into the adjacent mixture. Figure 17 [Badr, O., Karim, G.A., Liu, B. –"An Examination of the Flame Spread Limits in a Dual Fuel Engine", Applied Thermal Engineering 19 (1999), pp. 1071-1080] shows the observed variations of the flame spread limit (FSL) with changes in the pilot quantity derived from experimental data.

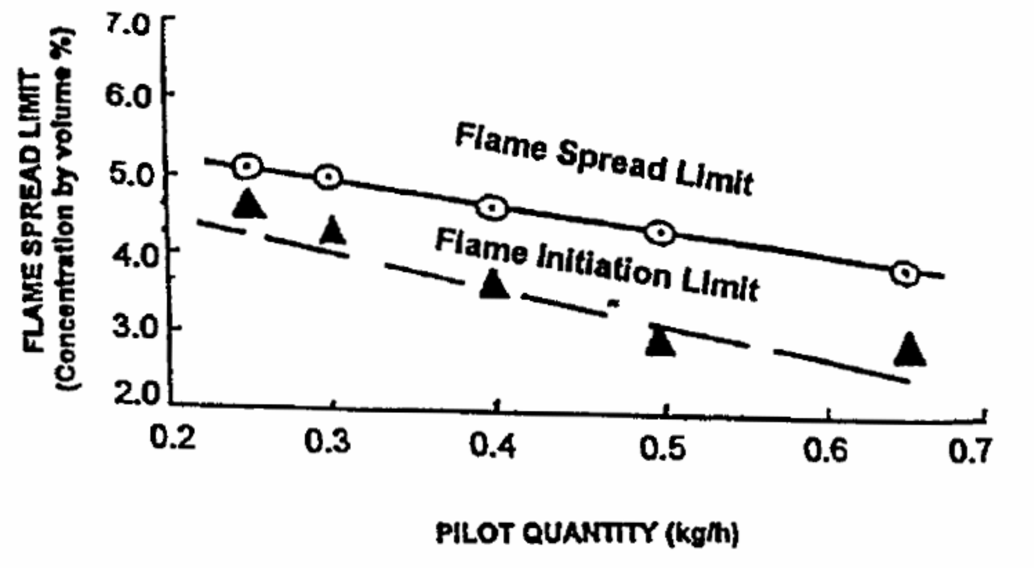


Figure 17 – *Change of the flame spread limit (FSL) with the pilot quantity for methane. The corresponding flame initiation limit (FIL) values are also shown*

The decrease of the limit with the increase in the pilot quantity is related to a number of contributing factors. These include a greater energy release at ignition time, improved pilot characteristics, a larger size of pilot mixture envelope with a greater entrainment of the gaseous fuel, a larger number of ignition centers requiring shorter flame travels, higher rates of heat transfer to the unburnt gaseous fuel-air mixture and an increased contribution of hot residual gas. The flame initiation limit (FIL) exhibits a similar trend.

Micro-pilot Implementation Review.

With research being done to reduce pilot quantities, which in turn reduces the emissions level, there is an increased need for reducing the flow of the diesel injectors to miniscule amounts. Most diesel fuel injection systems have a turndown ratio of about 10, which means that the injector's minimum reproducible flow per injection can only be about 10% of the maximum flow per injection. For this reason, BKM Inc. has developed the Servojet electrohydraulic, accumulator type unit fuel injection system that is capable of delivering down to 2 mm^3 per injection, representing 2% of the total energy required to run a 7.6 liter Navistar DT 466 engine at full load [17]. According to Gebert, the requirements of the pilot fuel injection system (FIS) were met most efficiently by modifying stock injectors to incorporate an internal accumulator and the application of other standard Servojet FIS components (Figure 18).

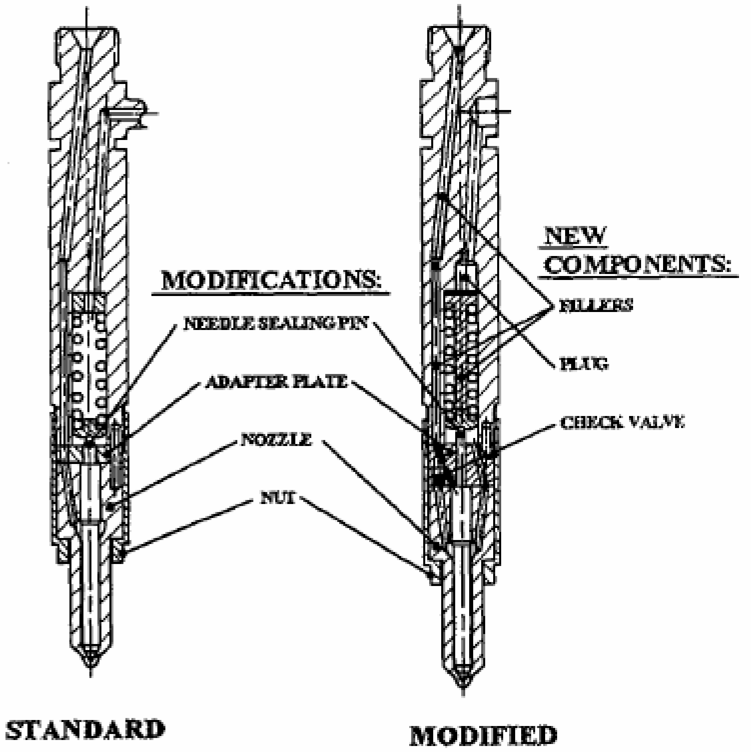


Figure 18.

The final concept of the micro-pilot unit injector consisted of the following components:

- Three-way, normally closed, solenoid operated
- Control valve
- Rail pressure intensifier
- Injector holder
- Nozzle
- Fuel supply rail.

During the engine tests performed by Gebert, the minimum fuel delivery was determined by following criteria:

- Stable injection pressure.
- Absence of engine misfire, determined by simultaneous monitoring of the cylinder pressure traces and THC emission levels.
- Exhaust gas temperature at each port.

The use of a diesel prechamber and separate fuel injection system instead of the conventional direct injection system for the pilot fuel offers substantial advantages, especially in large dual fuel engines. One such implementation was done by Cooper-

Bessemer, and the results were very promising. The dual engines equipped with diesel prechambers have achieved BMEP and fuel efficiency levels comparable with the highest rated diesel engines, and NO_x emissions as much as 90% less than diesel levels. The prechamber used in the design is small, occupying only 3% of the combustion chamber volume. The diesel pilot is injected into the chamber [Blizzard, T. Donald, Schaub, S. Frederik, Smith, Jesse G. – "Development of the Cooper-Bessemer Cleanburn Gas-Diesel (Dual-Fuel) Engine", ASME, ICE-Vol. 15, 1991], where it undergoes selfignition and begins to burn. As the combustion proceeds, the pressure buildup causes the content of the prechamber to shoot into the main chamber in a flaming jet, in a process similar to turbulent jet combustion. The excellent mixing and widespread distribution of the burning pilot charge due to this turbulent jet ensure extensive and complete combustion, even in very lean mixtures. Since the main charge is very lean, the flame temperature and NO_x production are low. The pilot fuel makes up less than 1% of the total fuel input at full load, and its combustion takes place under very rich conditions, so that the contribution to NO_x production by the pilot fuel is also low. Table 1 compares the emissions performance of a large Cooper-Bessemer dual-fuel engine before and after the incorporation of the "Cleanburn" prechamber chamber.

	ENGINE VERSION	
	Standard	CleanBurn
Diesel Pilot (%)	5.7	0.9
Firing Pressure (Kpa)	1301	1277
Fuel Cons. (kJ/kW-hr)	8810	8950
Total HC (g/kW-hr)	1.3	6.6
NO_x (g/kW-hr)	15.4	1.2
Smoke Opacity	20	<5

Table 1

CONCLUSIONS

The work described in this paper highlights the complexity of the phenomena that take place in a micro-pilot ignition system. It is necessary to underline the fact that many of these processes are not very well understood. Therefore, in general, the current approach to this research area comprises a preliminary modeling of the phenomena coupled with experimental data. The goal of further research should be a global treatment and understanding of the problems associated with micro-pilot ignition and with compression ignition in general.

REFERENCES

1. T. Sakuraui, et al., “Study on the lean burn gas engine using pilot oil for ignition source”, JSAE No. 87-89, 1998, P9-12

2. Nurick, W.H. “Orifice Cavitation and Its Effects on Spray Mixing,” J. Fluids Eng., Vol.98, pp.681-687, 1976.

3. Chaves, H., Knapp, M., Kubitzek, A., Obermeier, F. and Schneider, T. “Experimental Study of Cavitation in the Nozzle Hole of Diesel Injectors Using Transparent nozzles,” SAE Paper 950290, 1995.

4. Soteriou, C., Andrews, R. and Smith, M. “Direct Injection Diesel Sprays and the Effect of Cavitation and Hydraulic flip on Atomization,” SAE 950080, 1995.

5. He, L. and Ruiz, F. “Effect of Cavitation on Flow and Turbulence in Plain Orifices for High-Speed Atomization,” Atomization and Sprays, Vol. 5, pp. 569-584, 1995.

6. Knox-Kelecy, A.L. and Farrell, P.V. “Internal Flow in a Scale Model of a Diesel Fuel Injector nozzle,” SAE 922308, 1992.

7. Kato, M., Kano, H., Date, K., Oya, O. and Niizuma, K. “Flow Analysis in Nozzle Hole in Consideration of Cavitation,” SAE 970052, 1997.

8. Tamaki, N., Nishida, K., Hiroyasu, H. and Shimizu, M. “Effects of the Internal Flow in a Nozzle Hole on the Breakup Process of a Liquid jet,” Proc. ICLASS-´97, pp. 417-424, 1997.

9. Fath, A., Muench, K. and Leipertz, A. “Spray Break-Up Process of Diesel Fuel Investigated Close to the Nozzle,” Proc. ICLASS-´97, pp. 513-520, 1997.

10. Karim, G.A., Liu, Z., Jones, W, “Exhaust Emissions from Dual Fuel Engines at Light Load”, SAE paper 932822, 1993

11. Gebert, K., Beck, N. J., Barkhimer, R. L., Wong, H., “Strategies to Improve Combustion and Emission Characteristics of Dual Fuel Pilot Ignited Natural Gas Engines,” SAE paper 971712, 1997

12. Daisho, Y. Takahashi, Y.I., Nakayama, S., Kihara, R., Saito, T., “Controlling Combustion and Exhaust Emissions in a Direct-Injection Diesel Engine Dual-Fueled with Natural Gas,” SAE paper 952436, 1995.

13. Poonia, M.P., Ramesh, A., Gaur, R. R., "Effect of Intake Air Temperature and Pilot Fuel Quantity on the Combustion Characteristics of a LPG Dual Fuel Engine," SAE paper 982455, 1998.

14. Karim, G.A., "An Examination of Some Measures for Improving the Performance of Gas Fuelled Diesel Engines at Light Load", SAE paper 912366, 1991.

15. Li, X., Chippior, W. L., Gulder, O., "Effects of Cetane Enhancing Additives and Ignition Quality on Diesel Engine Emissions," SAE paper 972968, 1997.

16. Gunea, C., Razavi M. R. M., Karim, G. A., "The Effects of Pilot Fuel Quality on Dual Fuel Engine Ignition Delay", SAE paper 982453, 1998.

17. Gebert, K., Beck, N. J., Barkhimer, R. L., Wong, H., " Development of Pilot Fuel Injection System for CNG Engine," SAE paper 961100, 1996.

18. Liu, Z., Karim, G.A.: "The Ignition Delay Period in Dual Fuel Engines", SAE 950466; Nielsen, O.B., Quale, B., Sorenson S.: "Ignition Delay in Dual Fuel Engine", SAE 870589.

19. Hupperich, P., Durnhoiz, M., "Time-Controlled Pilot Injection for Stationary and Heavy-Duty Gas Engines", SAE paper 971713, 1997.

20. Liu, Z., Karim, G., "A Predictive Model for the Combustion Process in Dual Fuel Engines", SAE paper 952435, 1995.

21. Bergwerk, W. "Flow Pattern in Diesel Nozzle Spray Holes," Proc. Inst. Mech. Eng., Vol.173, pp.655-660, 1959.

22. Benedict, R.P. Fundamentals of Pipe Flow, Wiley, New York, 1980.

23. Chaves, H., Knapp, M., Kubitzek, A., Obermeier, F. and Schneider, T. "Experimental Study of Cavitation in the Nozzle Hole of Diesel Injectors Using Transparent nozzles," SAE Paper 950290, 1995.

24. Mises, R.V., von Ausfluß, B and Überflußzahlen, "Zeitschrift des Vereins deutscher Ingenieure," Band 61, Nr 21, pp 447-452, 1917.

25. Soteriou, C., Andrews, R. and Smith, M. "Direct Injection Diesel Sprays and the Effect of Cavitation and Hydraulic flip on Atomization," SAE 950080, 1995.

26. Tamaki, N., Nishida, K., Hiroyasu, H. and Shimizu, M. "Effects of the Internal Flow in a Nozzle Hole on the Breakup Process of a Liquid jet," Proc. ICLASS-´97, pp. 417-424, 1997.

27. BERGWERK W. Flow Pattern in Diesel Nozzle Spray Holes. Proc Instn Mech Engrs Vol 173, No 25 1959.

28. Celia Soteriou, Richard Andrews, and Mark Smith, "Direct Injection Diesel Sprays and the Effect of Cavitation and Hydraulic Flip on Atomization", SAE paper 950080, 1995.

29. Tamaki, N., Shimizu, M., Nishida, K., and Hiroyasu, H., "Effect of Cavitation and Internal Flow on Atomization of a Liquid Jet", Atomization and Sprays, Vol. 8, pp.179-197, 1998.

30. Browne, K. R., Partridge, I. M., and Greeves, G., "Fuel Property Effects on Fuel/Air Mixing in an Experimental Diesel Engine," SAE paper 860223, 1986.

31. Kamimoto, T., Yokota, H., and Kobayashi, H., "Effect of High Pressure Injection on Soot Formation Processes in a Rapid Compression Machine to Simulate Diesel Flames," *Transactions of the SAE*, Vol. 96, Sect. 4, pp. 4.783-4.791, 1987.

32. Espey, C. and Dec, J. E., "The Effect of TDC Temperature and Density on the Liquid-Phase Fuel Penetration in a D.I. Diesel Engine," *Transactions of the SAE*, Vol. 104, Sect. 4, pp. 1400-1414, 1995.

33. Long Zhang, Tadashi Tsurushima, Takahiro Ueda, et al., "Measurement of Liquid Phase Penetration of Evaporating Spray in a DI Diesel Engine, SAE paper 971645, 1997.

34. Chen, S. Kevin and Arthur H. Lefebvre, "Spray Cone Angles of Effervescent Atomizers", Atomization and Sprays, Vol.4, pp.291-301, 1994.

35. Tabata, M., Arai, M. and Hiroyasu, H., "Effects of Fuel Viscosity and Surface Tension on Diesel Spray Drops," ICLASS-85, IIB/1/1-IIB/1/8, 1995.

36. C.T. Chang and P.V. Farrell, "A Study on the Effects of Fuel Viscosity and Nozzle Geometry on High Injection Pressure Diesel Spray Characteristics", SAE paper 970353, 1997.

37. P.G. Felton, G. –P. Pan, and F.V. Bracco, "Additive Effects on Atomization and Evaporation of Diesel Fuel Under Engine Conditions", SAE paper 970795, 1997.

38. Evert Rosseel and Roger Sierens, "The Physical and the Chemical Part of the Ignition Delay in Diesel Engines" SAE paper 961123, 1996.

39. Brian Higgins and Dennis Siebers, "Diesel-Spray Ignition and Premixed-Burn Behavior", SAE paper 2000-01-0940, 2000.

APPENDIX 3

“System Specifications”

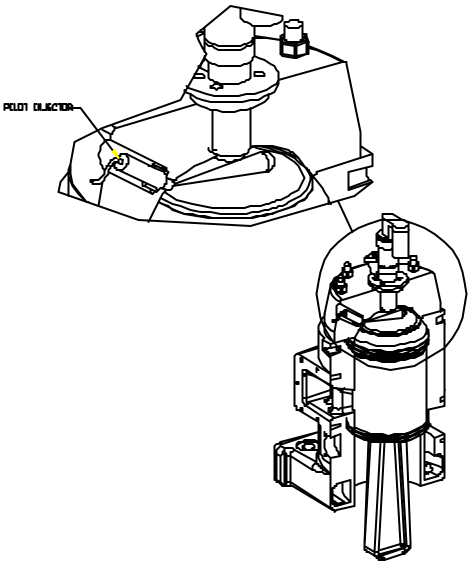
Colorado State University

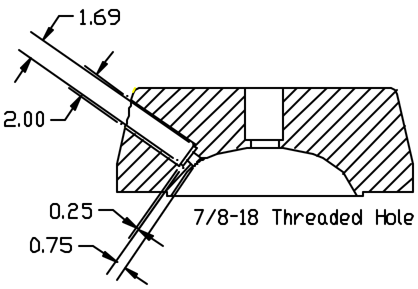
Engines and Energy Conversion Laboratory

Department of Mechanical Engineering
Fort Collins, Colorado 80523-1374
(970) 490-1418
FAX: (970) 493-6403
http://www.engr.colostate.edu/eecl

Micropilot Ignition Project

The Micropilot Ignition Project being conducted at Colorado State University will assess the benefits of retrofitting diesel pilot ignition systems on large-bore, lean burn, natural gas engines. A three-year program has been developed to integrate the necessary research and development efforts. Under Phase I of the program, a series of experiments will be conducted at the Engines and Energy Conversion Laboratory (EECL) of Colorado State University (CSU) to establish key design parameters and quantify the efficiency and emissions benefits from the use of pilot ignition.





DETAIL OF EXISTING HOLE
FOR INJECTOR MOUNTING
(Dimentions in Inches)

The EECL seeks a common rail ignition system that meets the following speculations:

Pulse Width – Injector should be capable of delivering 5-10 mm^3 of fuel in a minimum of 0.5 milliseconds. Nominal operation will be between 0.5 and 4 msec. 10 msec will be the maximum pulse width.

Rail Pressure – A rail pressure of 20,000 psi

Orifice Diameter – The injector orifice hole will be produced by a specialty nozzle manufacture if possible. This will allow the lab to evaluate identical injectors with different orifice diameters. Orifice holes will be between 0.1mm and 0.2 mm.

Injector Size – The fuel injector must be small enough to allow installation through an existing 18mm spark plug hole. The injector hole dimensions are shown above.

Electronics' Specifications

Pulse Width –Nominal operation will be between 0.5 and 4 milliseconds. Ten milliseconds will be the maximum pulse width.

Manual Controls on the ECU – The ECU must be able to manually adjust rail pressure between ~10-20k psi, operate in a "single shot" mode, and have the ability to accurately adjust injection volume.

– Electronics must be able to support the slow speed of the engine (300rpm), and large number of teeth on the flywheel (411).

Current Signal – Achieve 18-20 amps in 0.2 milliseconds.

Additional Hardware

The EECL seeks a complete system, which in addition to the injectors and electronic control until, may include, the following:

Low Pressure Stage

- Pre-supply pump with pre-filter
- Fuel Filter

High Pressure Stage

- Injectors
- High Pressure Pump with Pressure Control Valve
- Pressure Limiter Valve
- Flow Limiter

Bosch Recommendations

The following hardwear has been recommend by Bosch:

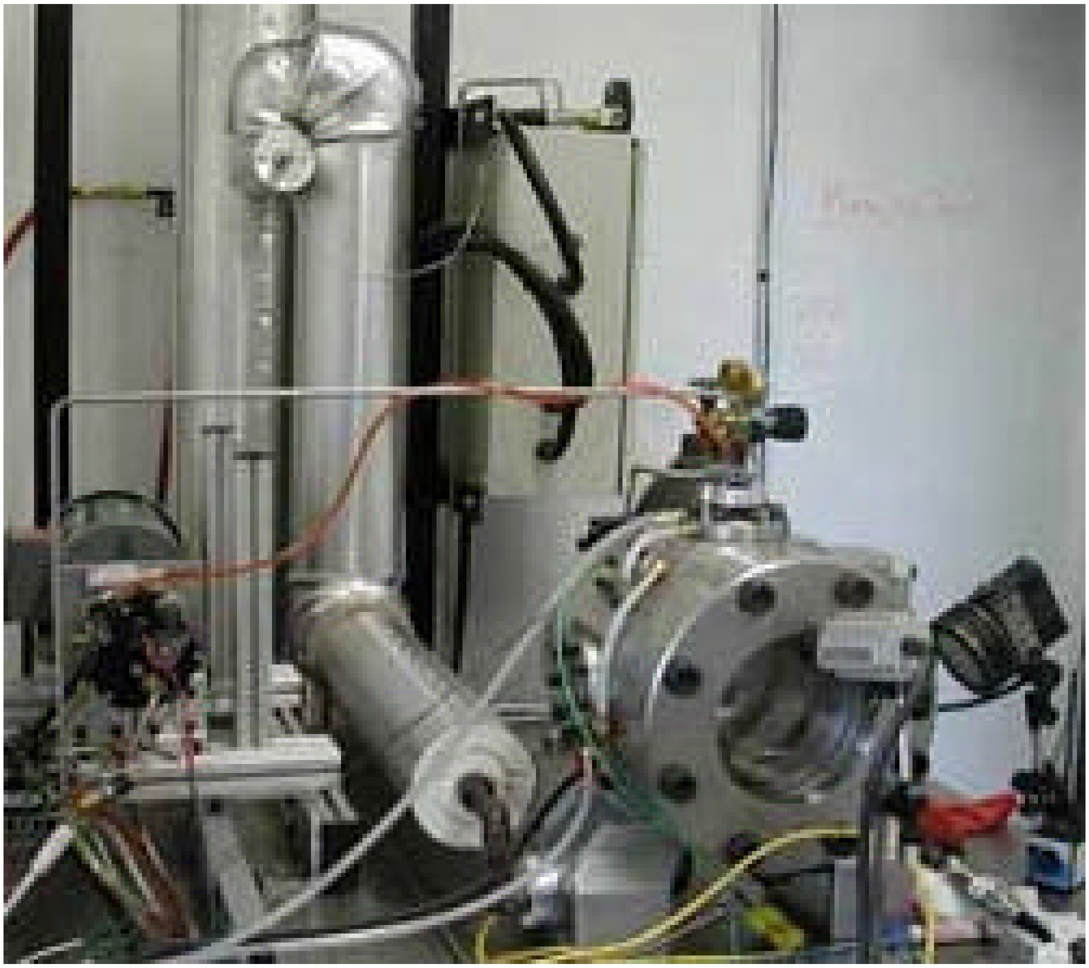
Injector # 0445110062

Injector # 0445110002

CP3.3 High Pressure Pump

APPENDIX 4

“1-Cylinder Prototype (CTC - combustion test chamber)
Drawings and Photographs”



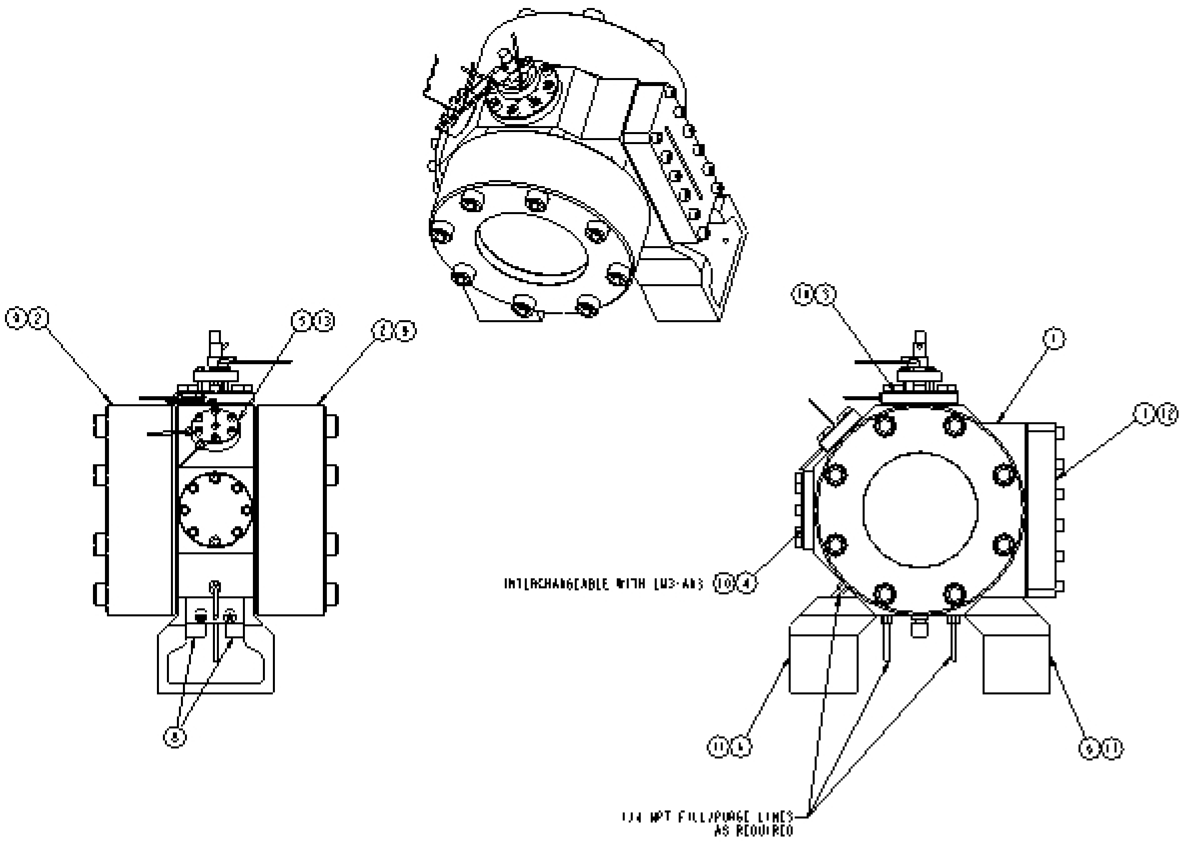
1-Cylinder Prototype (CTC) with Delphi fuel system installed, CTC heating system, and imaging system installed.

			4	D17	MAIN ACCESS WINDOW GSKT			
			1	D16	INTERNAL TRNSDCR SLV GSKT			
			2	D15	EXTERNAL SLEEVE GASKET			
			2	D14	INTERNAL SLEEVE GASKET			
			1	D13	LOWER DELCO INJCTR SLEEVE	1	A07	LASER IGNITION SLEEVE
1	D30	CUMMINS INJECTOR RETAINER	2	D12	MAIN FRAME BRACKET	2	A06	MAIN OPTICAL ACCESS ASSY
1	D29	LWR CUMMINS INJCTR SLEEVE	2	D11	MAIN ACCESS WNDW RETAINER	1	A05	PRESSURE TRNSDCR MTG ASSY
1	D28	UPR CUMMINS INJCTR SLEEVE	2	D10	MAIN ACCESS WINDOW	1	A04	DELCO INJECTOR MTG ASSY
1	D27	LASER IGNITION WINDOW	1	D09	UPPER DELCO INJCTR SLEEVE	1	A03	COOLING PROBE MTG ASSY
2	D26	LASER IGNITION GASKETS	1	D08	PRESSURE TRANSDUCER SLEEVE	1	A02	LASER ACCESS ASSY
1	D25	LOWER LASER SLEEVE	1	D07	LASER ACCESS MTG GASKET	1	A01	MASTER ASSY
1	D24	UPPER LASER SLEEVE	1	D06	PRESSURE TRANSDUCER COVER	QTY	SUFFIX	NAME
1	D23	UPPER PROBE SLEEVE	1	D05	LASER ACCESS WINDOW			
1	D22	LOWER PROBE SLEEVE	1	D04	LASER ACCESS SLEEVE			
2	D21	LASER ACCESS WINDOW GSKT	1	D03	DELCO INJECTOR RETAINER			
1	D20	LASER ACCESS COVER	1	D02	BLANK ACCESS SLEEVE			
1	D19	EXTERNAL TRNSDCR SLV GSKT	1	D01	MAIN FRAME			
QTY	SUFFIX	NAME	QTY	SUFFIX	NAME			

TITLE:
MASTER LIST
COMBUSTION TEST CHAMBER

SCALE: NONE	DWG #: EM3-LST		LAST REVISED: 07/25/02
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 12/14/01

CSU ENGINES & ENERGY CONVERSION LAB



#	QTY	COMPONENT	NAME
1	1	EM3-D01	MAIN FRAME
2	2	EM3-A06	MAIN OPTICAL ACCESS ASSEMBLY
3	1	EM3-A04	FUEL INJECTOR MOUNTING ASSEMBLY
4	1	EM3-D02	BLANK ACCESS SLEEVE
5	1	EM3-A05	PRESSURE TRANSDUCER MTG ASSEMBLY
6	2	EM3-D12	MAIN FRAME BRACKET
7	1	EM3-A02	LASER ACCESS ASSEMBLY
8	2	PURCHASE	OMEGALOCK SSLK-14-14 FITTING
9	16	PURCHASE	SHCS 3/4-10 X 4-1/2 LG
10	16	PURCHASE	SHCS 3/8-16 X 1-1/4 LG
11	4	PURCHASE	SHCS 3/8-16 X 1 LG
12	10	PURCHASE	SHCS 3/8-16 X 1-3/8 LG
13	6	PURCHASE	SHCS 1/4-20 X 1 LG

TORQUE SPECIFICATIONS:
MAIN OPTICAL ACCESS ASSEMBLY: 80-100 FT-LB
LASER ACCESS ASSEMBLY: 20-25 FT-LB
FUEL INJECTOR MOUNTING ASSEMBLY: 15-20 FT-LB
BLANK ACCESS SLEEVE: 15-20 FT-LB
PRESSURE TRANSDUCER MOUNTING ASSEMBLY: 30-40 IN-LB

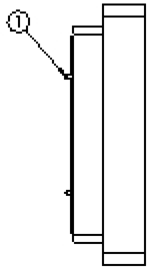
UNLESS OTHERWISE SPECIFIED.
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

MASTER ASSEMBLY
COMBUSTION TEST CHAMBER

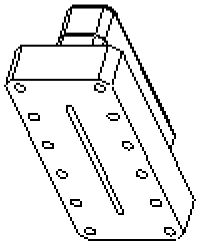
1:5	EM3-A01	05/23/07
WALL	WALL	1 OF 1	01/09/07

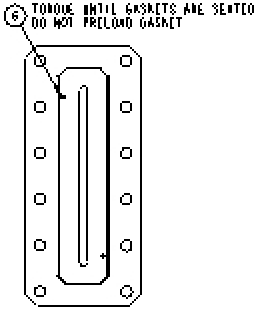
CSU ENGINES & ENERGY CONVERSION LAB

#	QTY	COMPONENT	NAME
1	1	EM3-004	LASER ACCESS SLEEVE
2	1	EM3-005	LASER ACCESS WINDOW
3	1	EM3-020	LASER ACCESS COVER
4	1	EM3-007	LASER ACCESS MOUNTING GASKET
5	2	EM3-021	LASER ACCESS WINDOW GASKET
6	2	PURCHASE	SHCS #4-40 X 1-1/2 LG
7	2	PURCHASE	DOWEL PIN 1/8 X 1/2 LG

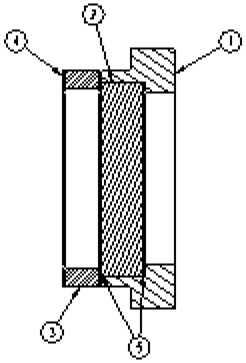


VIEW IN DIRECTION OF ARROW 7
ITEM 3, 4, AND 5 REMOVED

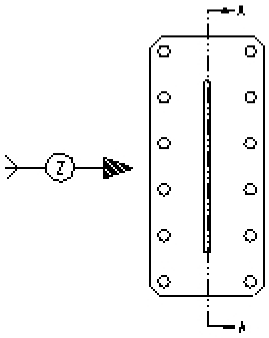




ITEM 4 MOUNTING GASKET REMOVED



SECTION A-A



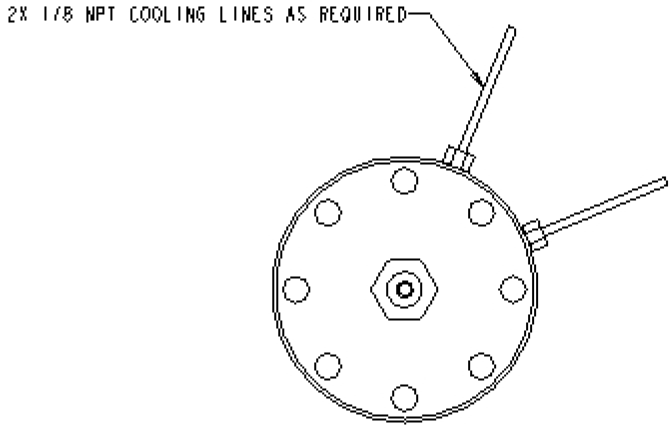
UNLESS OTHERWISE SPECIFIED.
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2B
--ALL EXTERNAL THREADS CLASS 2A
--ALL DIMENSIONS ± .01
--ALL ANGLES ±1°

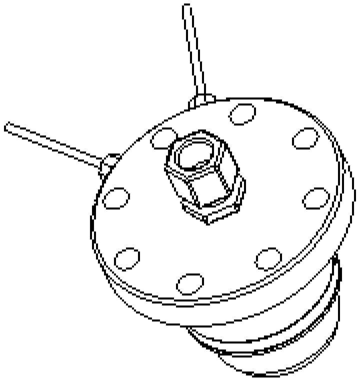
LASER ACCESS ASSEMBLY
COMBUSTION TEST CHAMBER

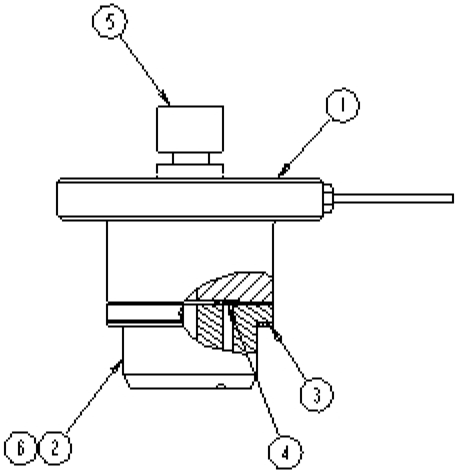
1:2	EM3-A02		09/29/07
NULL	NULL	OF 1	01/08/07

CSU ENGINES & ENERGY CONVERSION LAB

1	1	EM3-D23	UPPER PROBE SLEEVE
2	1	EM3-D22	LOWER PROBE SLEEVE
3	1	EM3-D15	EXTERNAL SLEEVE GASKET
4	1	EM3-D14	INTERNAL SLEEVE GASKET
5	1	PURCHASE	OMEGALOCK SSLK-14-14 FITTING
6	3	PURCHASE	SHCS #6-32 X 1-1/4







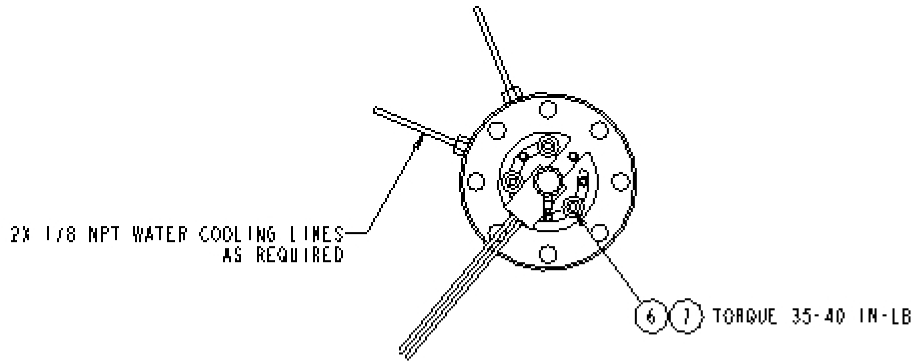
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

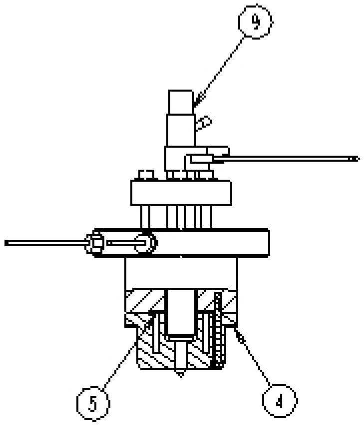
TITLE:
COOLING PROBE MTG ASSEMBLY
COMBUSTION TEST CHAMBER

SCALE: 3:4	DWG #: EM3-A03		LAST REVISED: ---
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 01/21/02

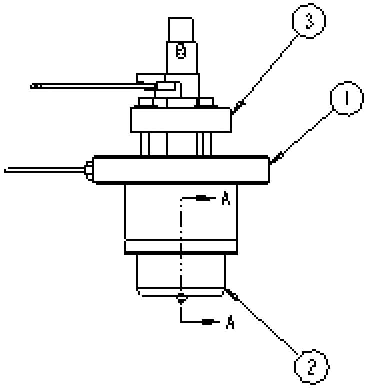
CSU ENGINES & ENERGY CONVERSION LAB

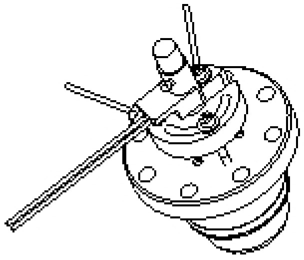
1	1	EM3-D09	UPPER INJECTOR SLEEVE
2	1	EM3-D13	LOWER INJECTOR SLEEVE
3	1	EM3-D03	FUEL INJECTOR RETAINER
4	1	EM3-D15	EXTERNAL SLEEVE GASKET
5	1	EM3-D14	INTERNAL SLEEVE GASKET
6	3	PURCHASE	SHCS #10-24 X 1-1/2
7	3	PURCHASE	FLAT WASHER #10
8	3	PURCHASE	SHCS #6-32 X 1-1/4
9	1	PURCHASE	AC DELCO 930BZB08 INJECTOR

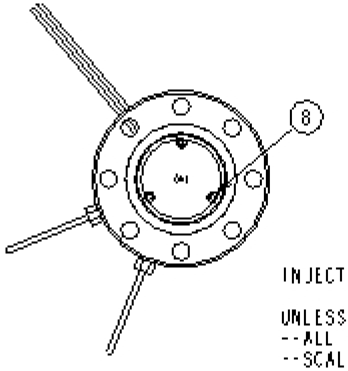




SECTION A-A







INJECTOR ORIENTATION AS REQUIRED

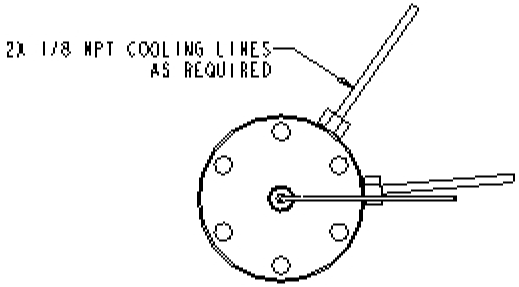
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

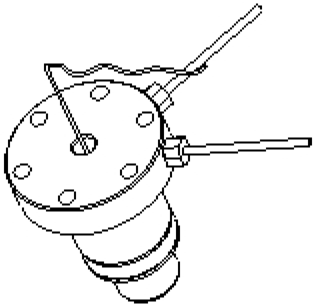
TITLE:
DELCO INJECTOR MTG ASSEMBLY
COMBUSTION TEST CHAMBER

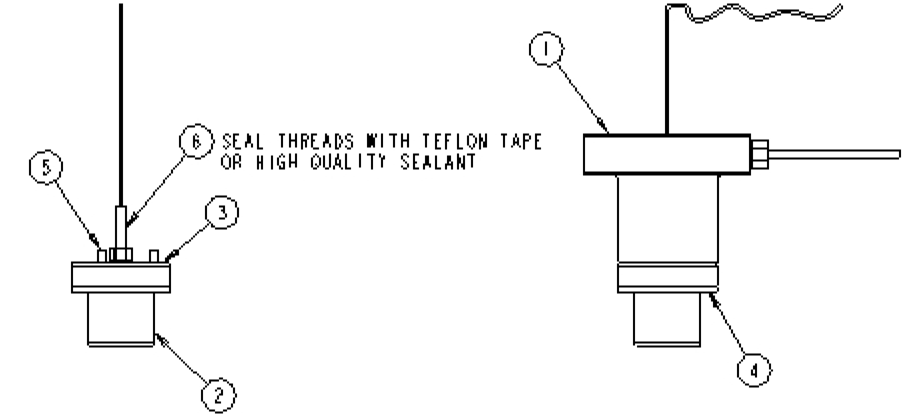
SCALE: 1:2	DWG #: EM3-A04		LAST REVISED: 02/27/02
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 12/14/01

CSU ENGINES & ENERGY CONVERSION LAB

1	1	EM3-D06	PRESSURE TRANSDUCER COVER
2	1	EM3-D08	PRESSURE TRANSDUCER SLEEVE
3	1	EM3-D16	INTERNAL TRANSDUCER SLEEVE GSKT
4	1	EM3-D19	EXTERNAL TRANSDUCER SLEEVE GSKT
5	2	PURCHASE	DOWEL PIN 1/8 X 1/2 LG
6	1	PURCHASE	KISTLER PRESSURE TRANSDUCER 6125A21







EM3-D08 TRANSDUCER COVER AND
COOLING LINES REMOVED

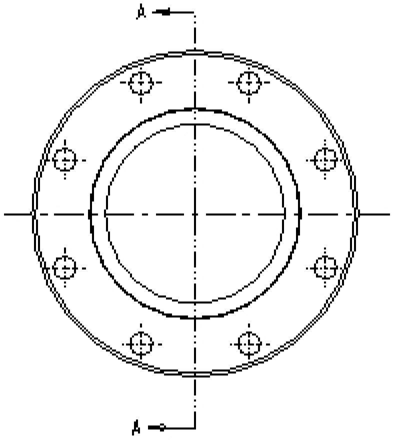
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

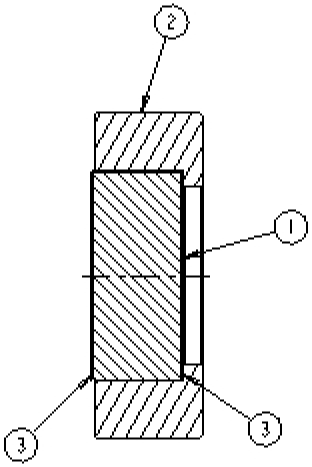
TITLE:
PRESSURE TRANSDUCER MTG ASSEMBLY
COMBUSTION TEST CHAMBER

SCALE:	DWG #:		LAST REVISED:
3:4	EM3-A05		01/14/02
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 01/03/02

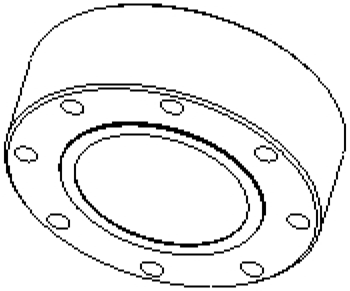
CSU ENGINES & ENERGY CONVERSION LAB

1	1	EM3-D11	MAIN ACCESS WINDOW RETAINER
2	2	EM3-D10	MAIN ACCESS WINDOW
3	2	EM3-D17	MAIN ACCESS WINDOW GASKET





SECTION A-A



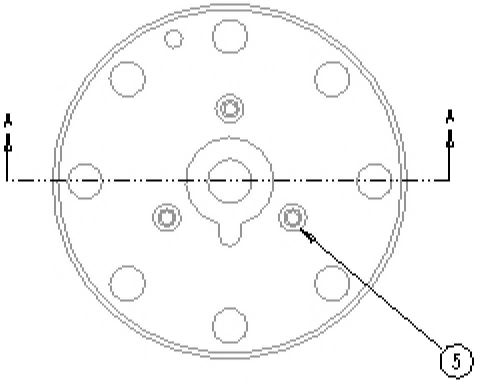
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

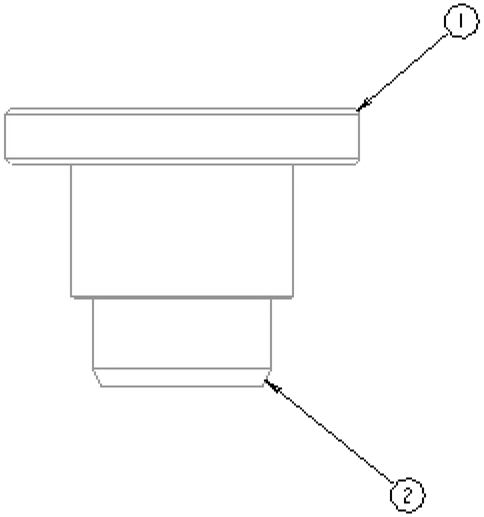
TITLE:
MAIN OPTICAL ACCESS ASSEMBLY
COMBUSTION TEST CHAMBER

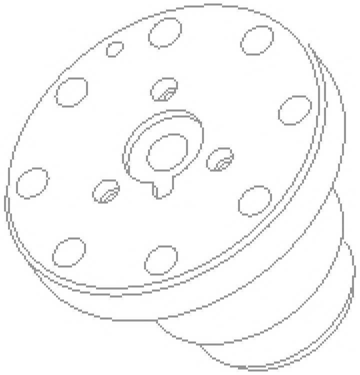
SCALE: 1:3	DWG #: EM3-A06		LAST REVISED: 01/28/02
DESIGN: WHULL	DETAIL: WHULL	SHEET 1 OF 1	DRAWN: 11/18/01

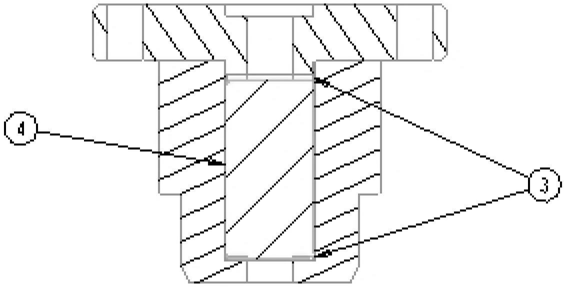
CSU ENGINES & ENERGY CONVERSION LAB

1	1	EM3-D24	UPPER LASER SLEEVE
2	1	EM3-D25	LOWER LASER SLEEVE
3	2	EM3-D26	LASER IGNITION GASKETS
4	1	EM3-D27	LASER IGNITION WINDOW
5	3	PURCHASE	SHCS #10-24 X 1









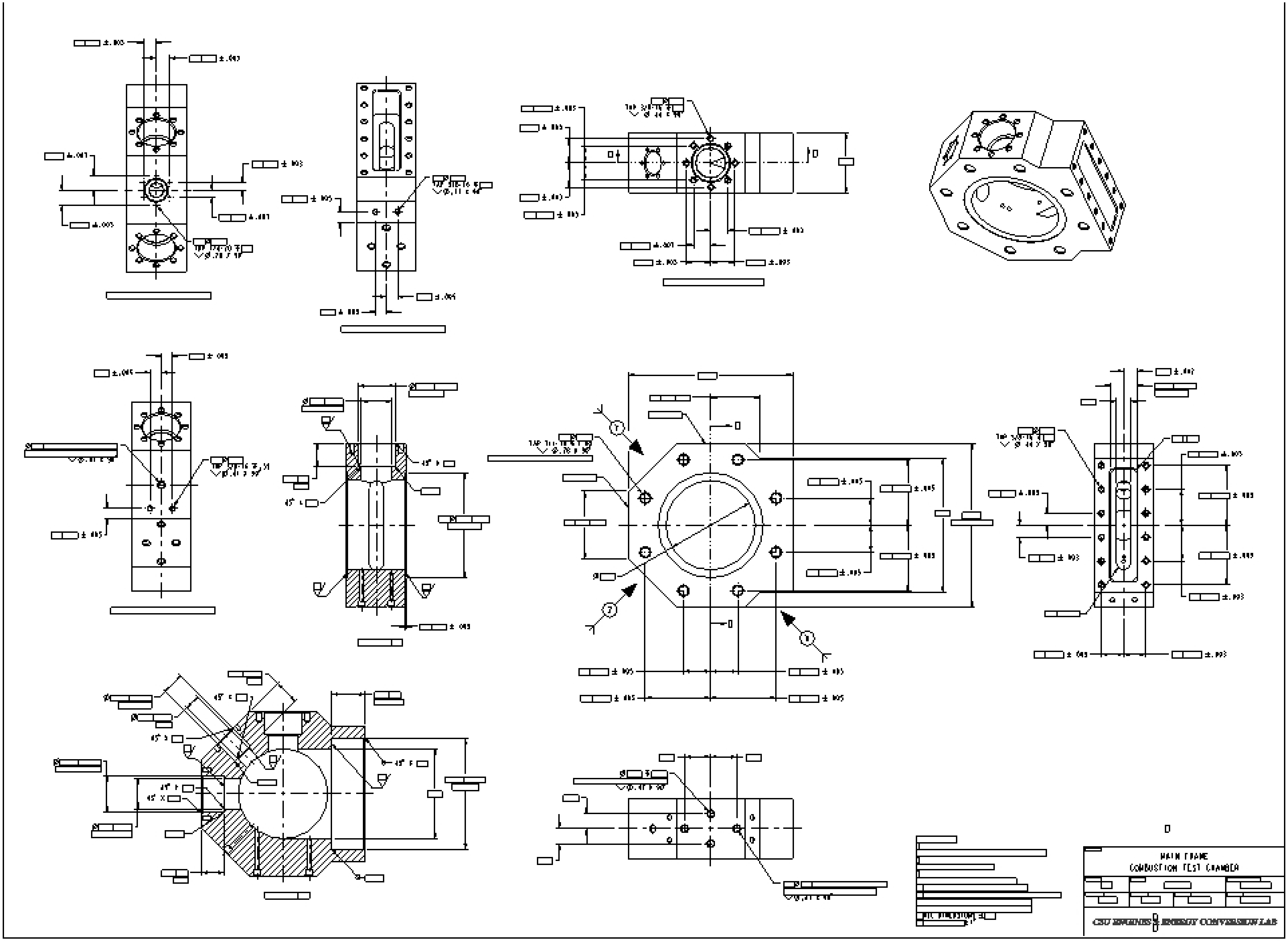
SECTION A-A

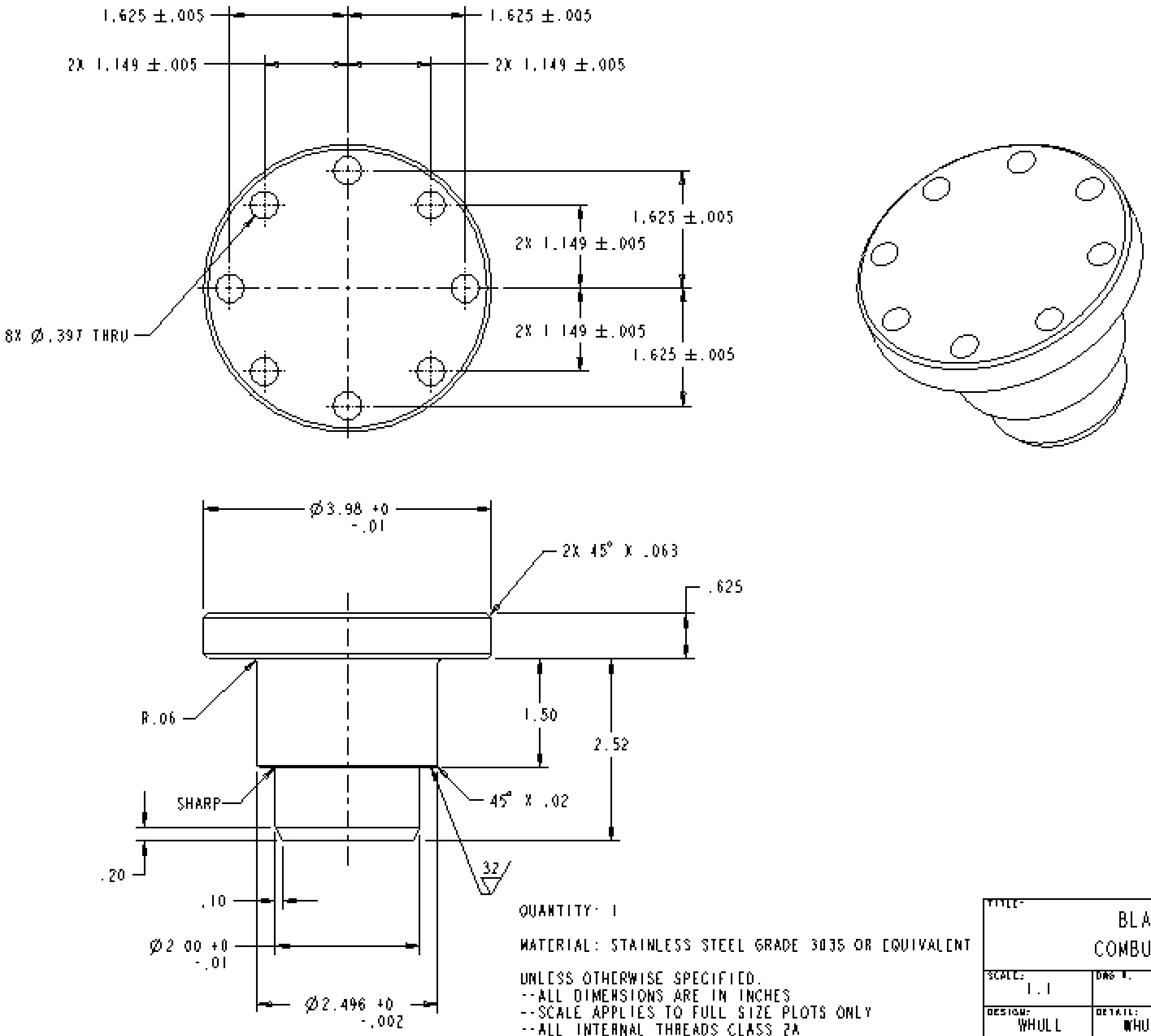
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

TITLE:
LASER IGNITION SLEEVE
COMBUSTION TEST CHAMBER

SCALE: 1:1	DWG #: EM3-A07		LAST REVISED ---
DESIGN: F JAMISON	DETAIL: F JAMISON	SHEET: 1 OF 1	DRAWN: 06/06/02

CSU ENGINES & ENERGY CONVERSION LAB


±.005
TAP 3/8-16
45° X
MAIN FRAME
COMBUSTION TEST CHAMBER
CSU ENGINES & ENERGY CONVERSION LAB



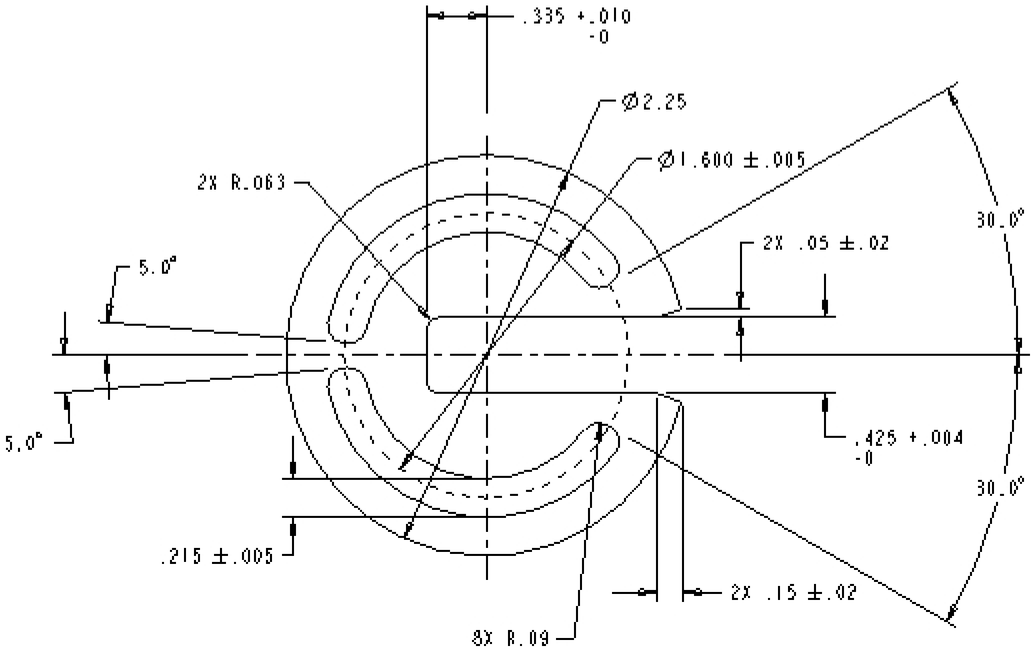
QUANTITY: 1

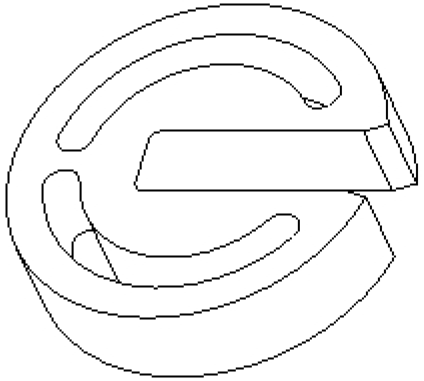
MATERIAL: STAINLESS STEEL GRADE 303S OR EQUIVALENT

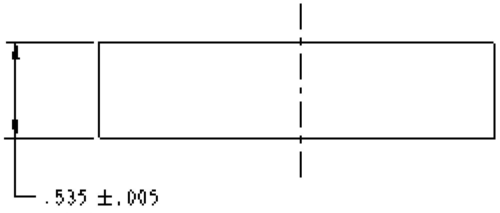
UNLESS OTHERWISE SPECIFIED.
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

TITLE: BLANK ACCESS SLEEVE COMBUSTION TEST CHAMBER			
SCALE: 1:1	DWG #: EM3-D02		LAST REVISED: 01/11/02
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 11/17/01

CSU ENGINES & ENERGY CONVERSION LAB







QUANTITY: 1

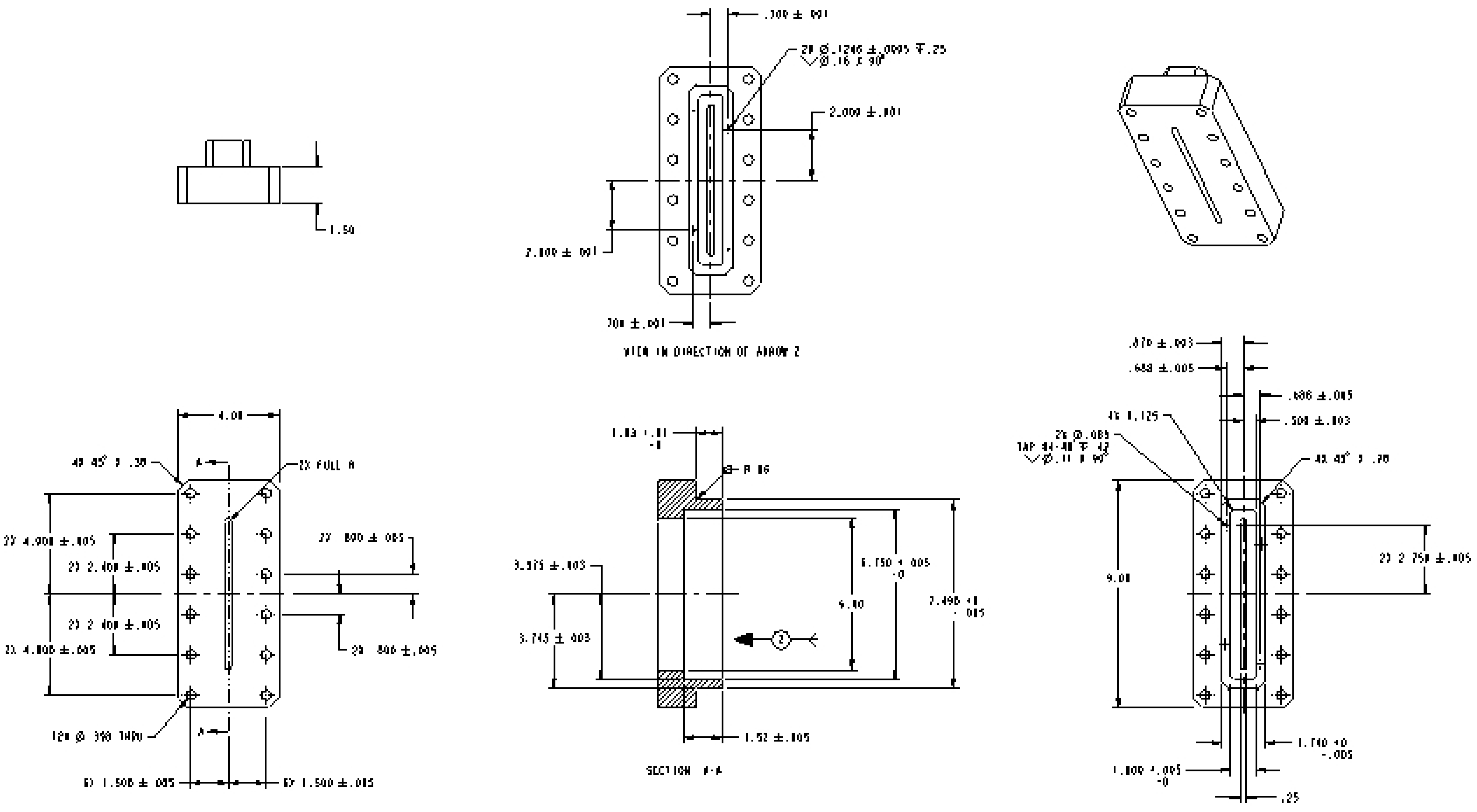
MATERIAL: ALUMINUM 6061

UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

TITLE:
DELCO INJECTOR RETAINER
COMBUSTION TEST CHAMBER

SCALE: 2:1	DWG #: EN3-D03		LAST REVISED: 05/21/02
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 11/27/01

CSU ENGINES & ENERGY CONVERSION LAB



QUANTITY: 1

MATERIAL - STAINLESS STEEL GRADE 303S

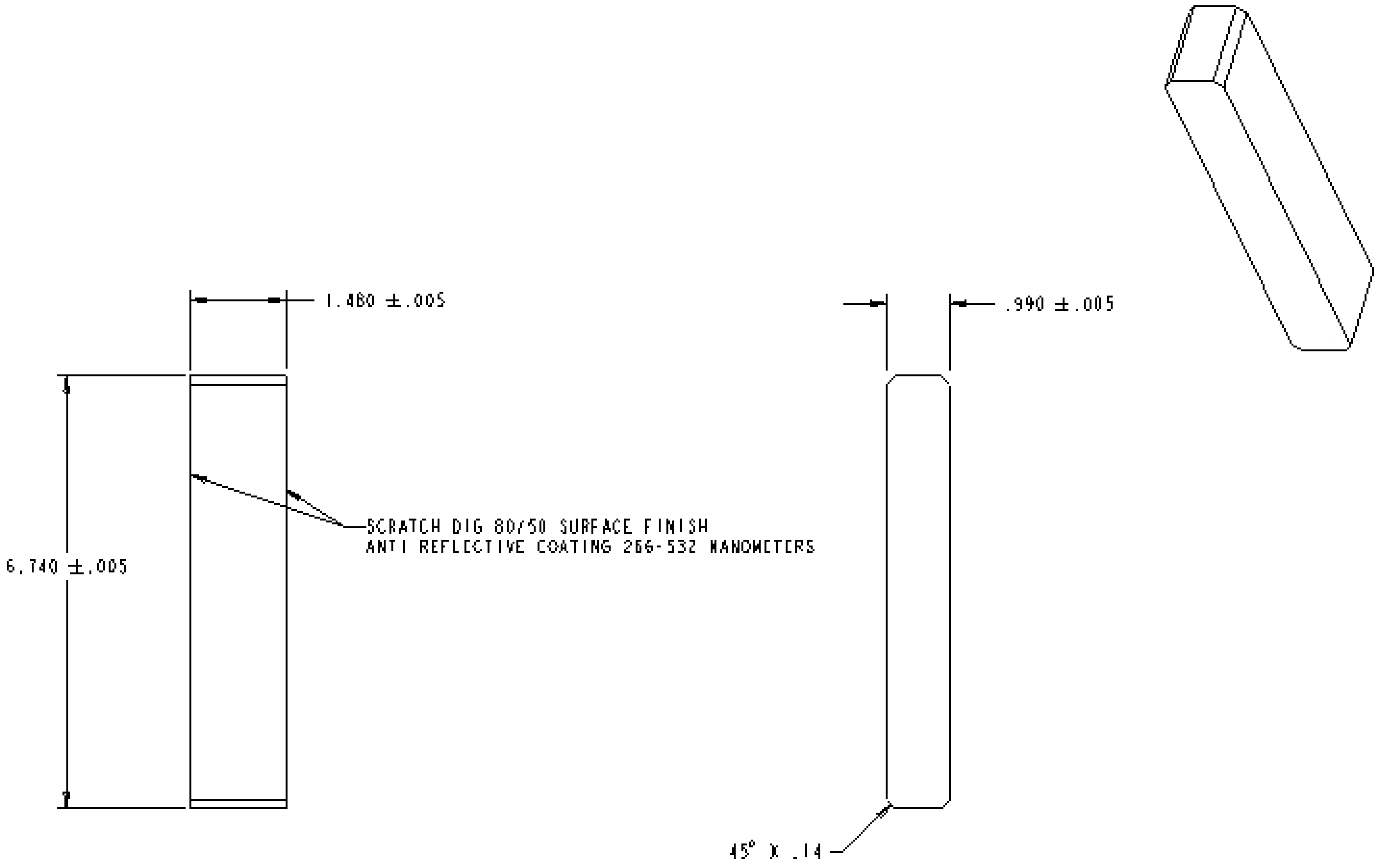
BREAK ALL SHARP EDGES

UNLESS OTHERWISE SPECIFIED.
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

LASER ACCESS SLEEVE
COMBUSTION TEST CHAMBER

1:2	EM3-004	09/20/07
NULL	NULL	OF 1	1/19/01

CSU ENGINES & ENERGY CONVERSION LAB



QUANTITY: 1

MATERIAL: FUSED SILICA TYPE GE124 OR EQUIVALENT

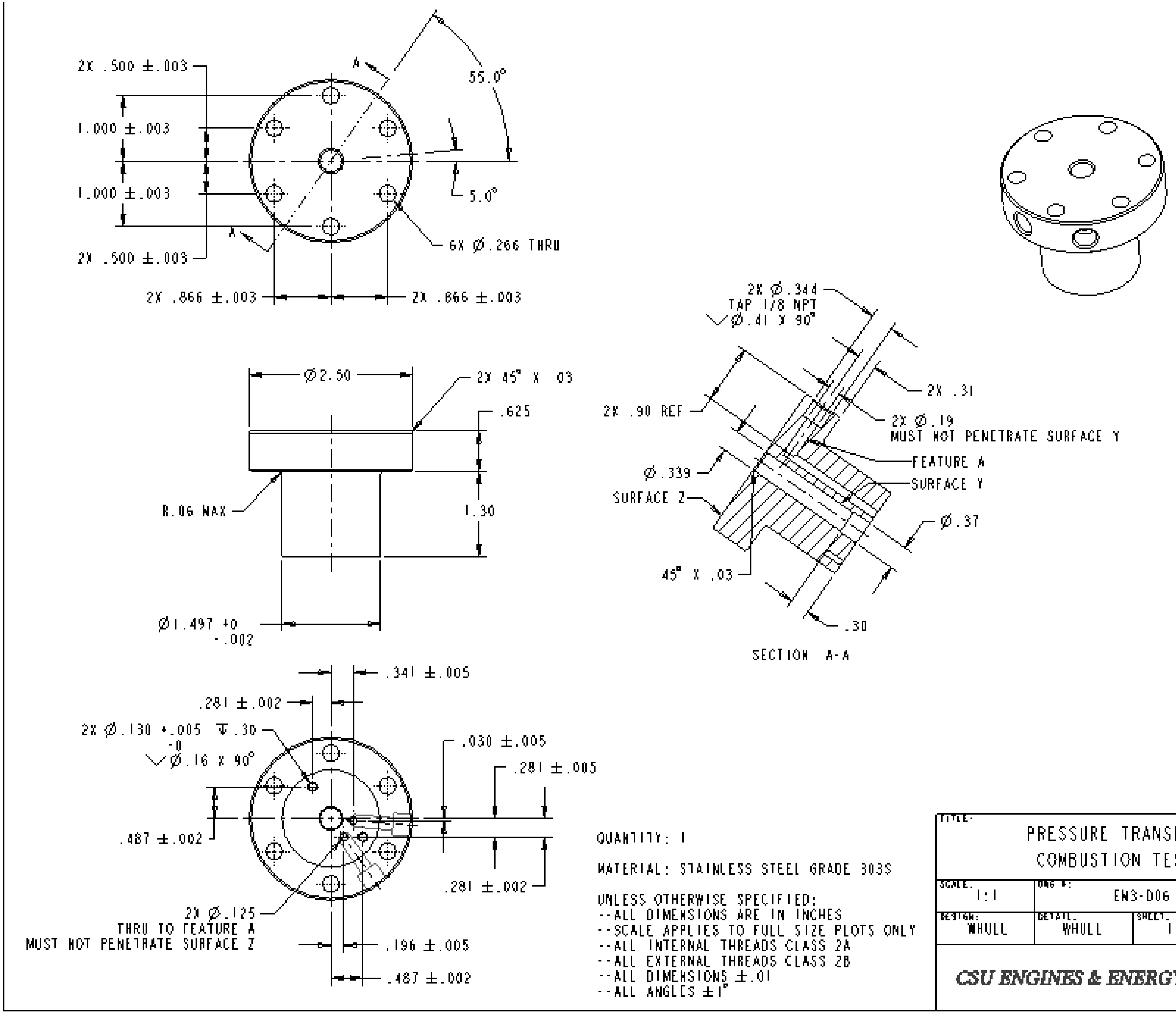
BREAK ALL EDGES .01 MIN

UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

TITLE:
LASER ACCESS WINDOW
COMBUSTION TEST CHAMBER

SCALE: 3:4	DWG #: EW3-D05		LAST REVISED: 01/23/02
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 11/19/01

CSU ENGINES & ENERGY CONVERSION LAB



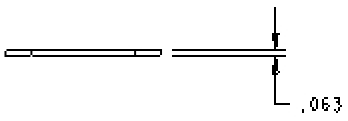
QUANTITY: 1

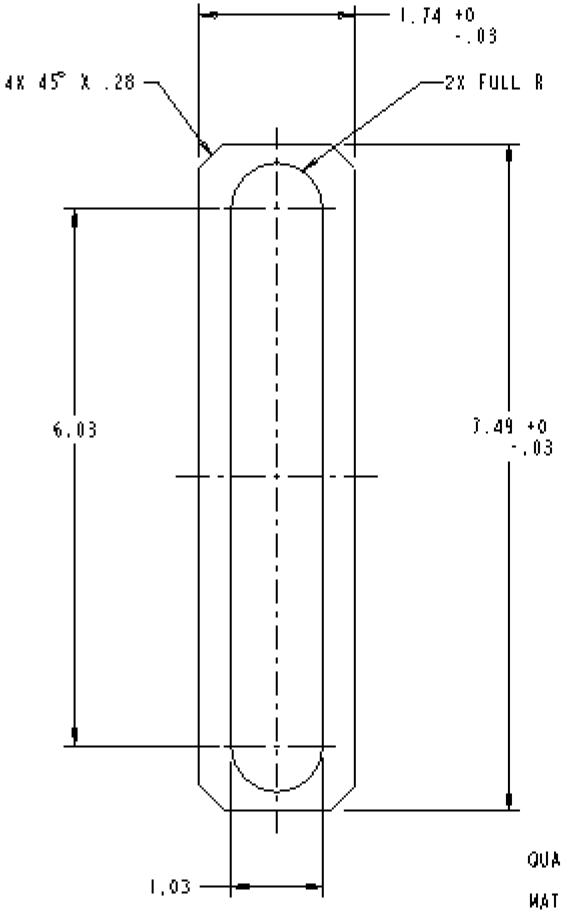
MATERIAL: STAINLESS STEEL GRADE 303S

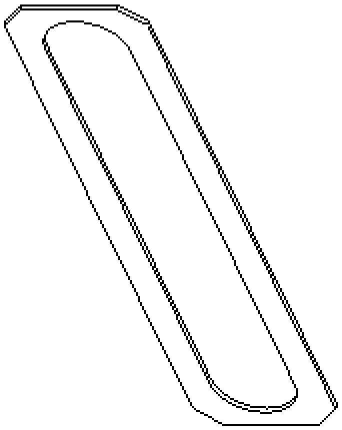
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

TITLE: PRESSURE TRANSDUCER COVER COMBUSTION TEST CHAMBER			
SCALE: 1:1	DWG #: EN3-D06		LAST REVISED: 02/22/02
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 11/19/01

CSU ENGINES & ENERGY CONVERSION LAB







QUANTITY: 4 (ONLY 1 RQD)

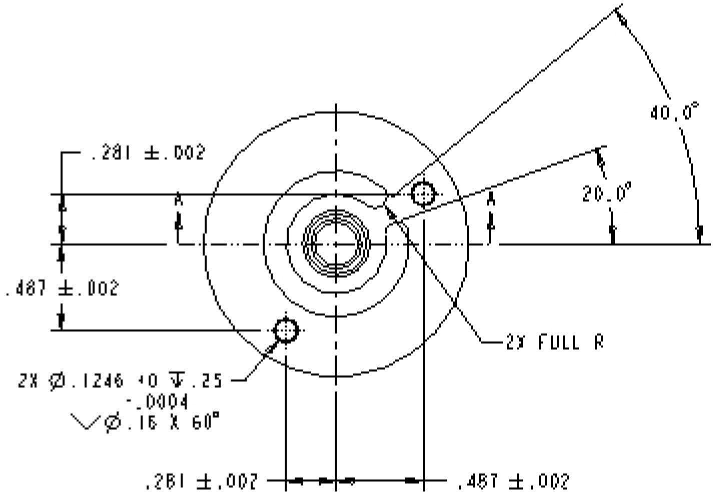
MATERIAL: BSSC GRAPHITE

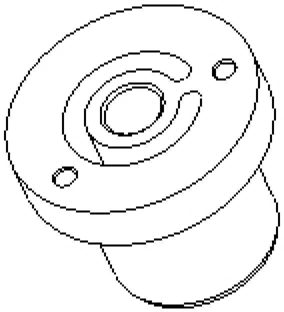
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.015
--ALL ANGLES ±1°

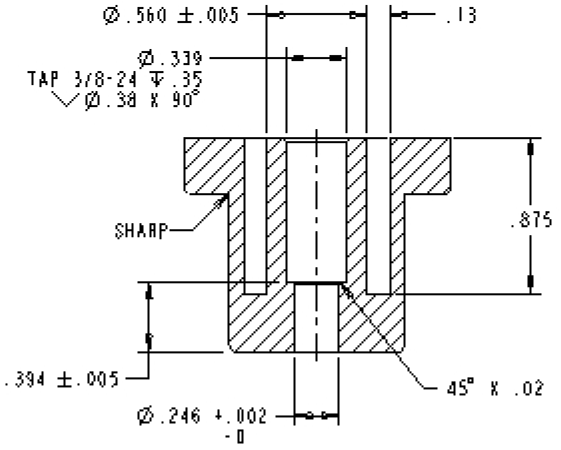
TITLE:
LASER ACCESS MTG GASKET
COMBUSTION TEST CHAMBER

SCALE:	DWG #:		LAST REVISED:
1:1	EN3-D07		01/25/02
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 11/17/01

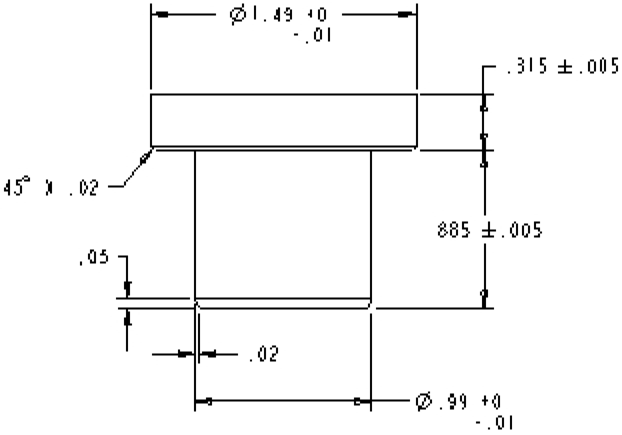
CSU ENGINES & ENERGY CONVERSION LAB







SECTION A-A



QUANTITY: 1

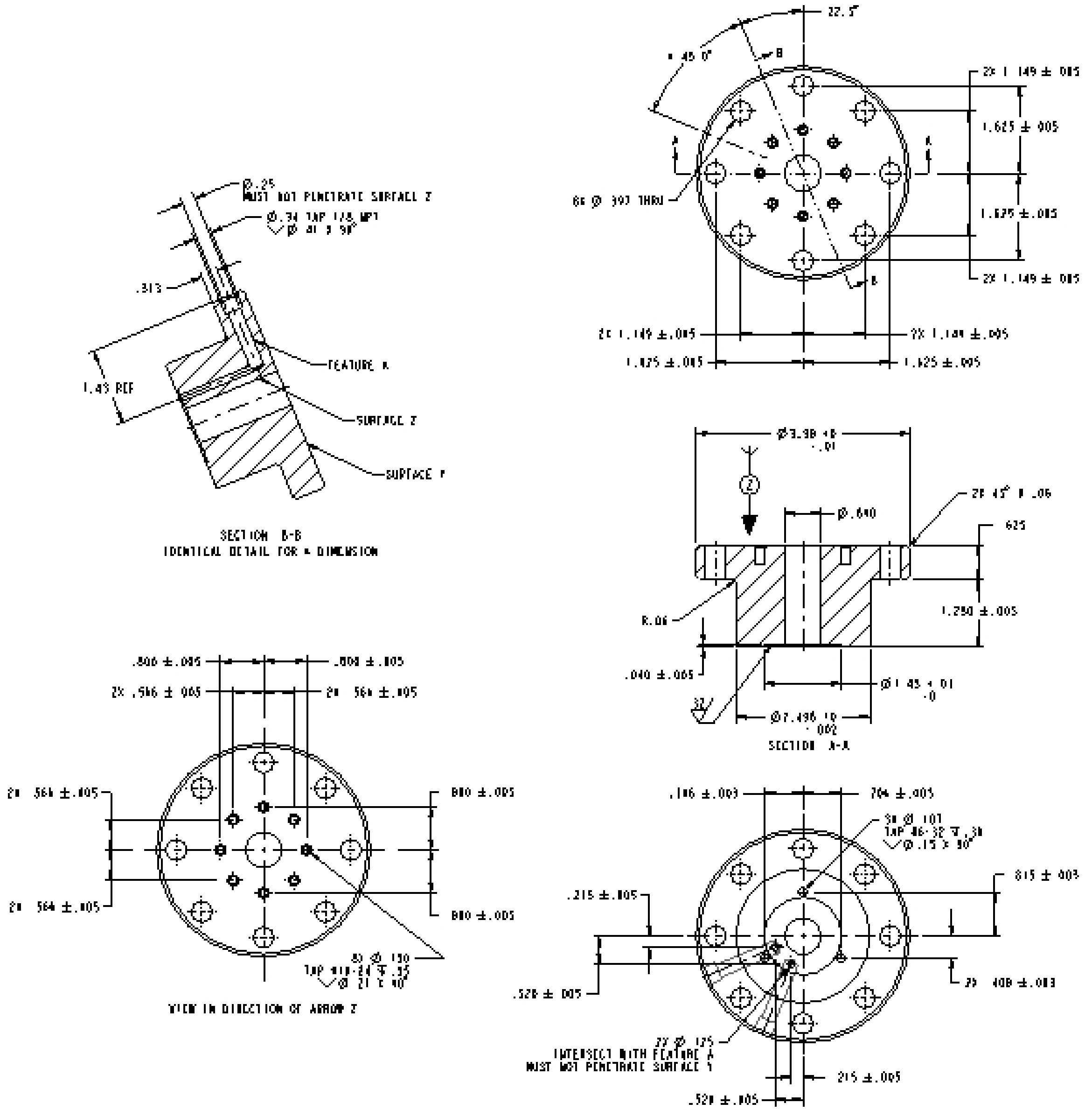
MATERIAL: STAINLESS STEEL GRADE 303S

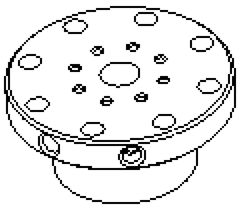
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

TITLE:
PRESSURE TRANSDUCER SLEEVE
COMBUSTION TEST CHAMBER

SCALE: 2:1	DWG #: EN3-D08		LAST REVISED: 01/15/02
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 01/04/02

CSU ENGINES & ENERGY CONVERSION LAB





QUANTITY- 1

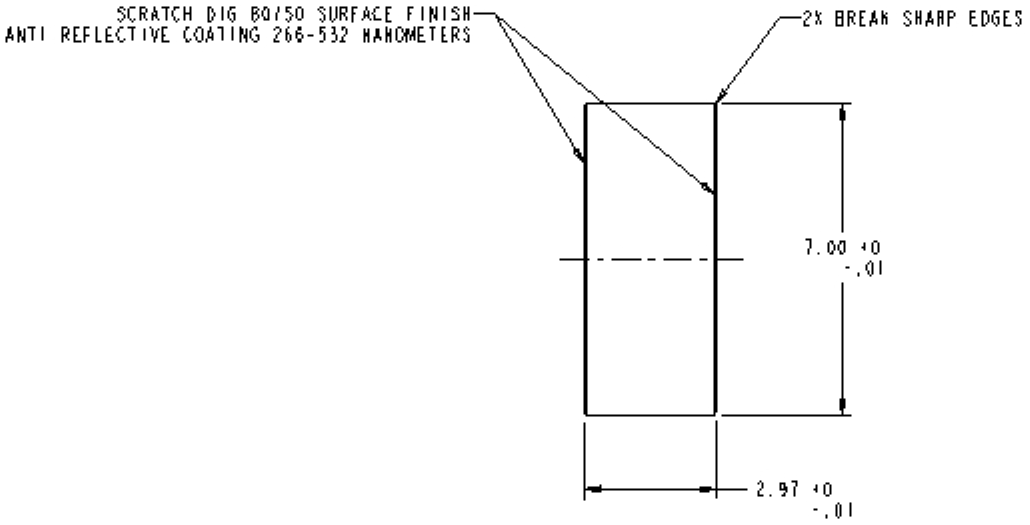
MATERIAL. STAINLESS STEEL GRADE 303S

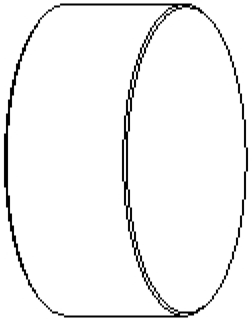
BREAK ALL SHARP EDGES

UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ± .01
--ALL ANGLES ±1°

UPPER DELCO INJECTOR SLEEVE COMBUSTION TEST CHAMBER			
1:1	EN3-009		01/15/02
M.HULL	M.HULL	01 1	12/13/01

CSU ENGINES & ENERGY CONVERSION LAB



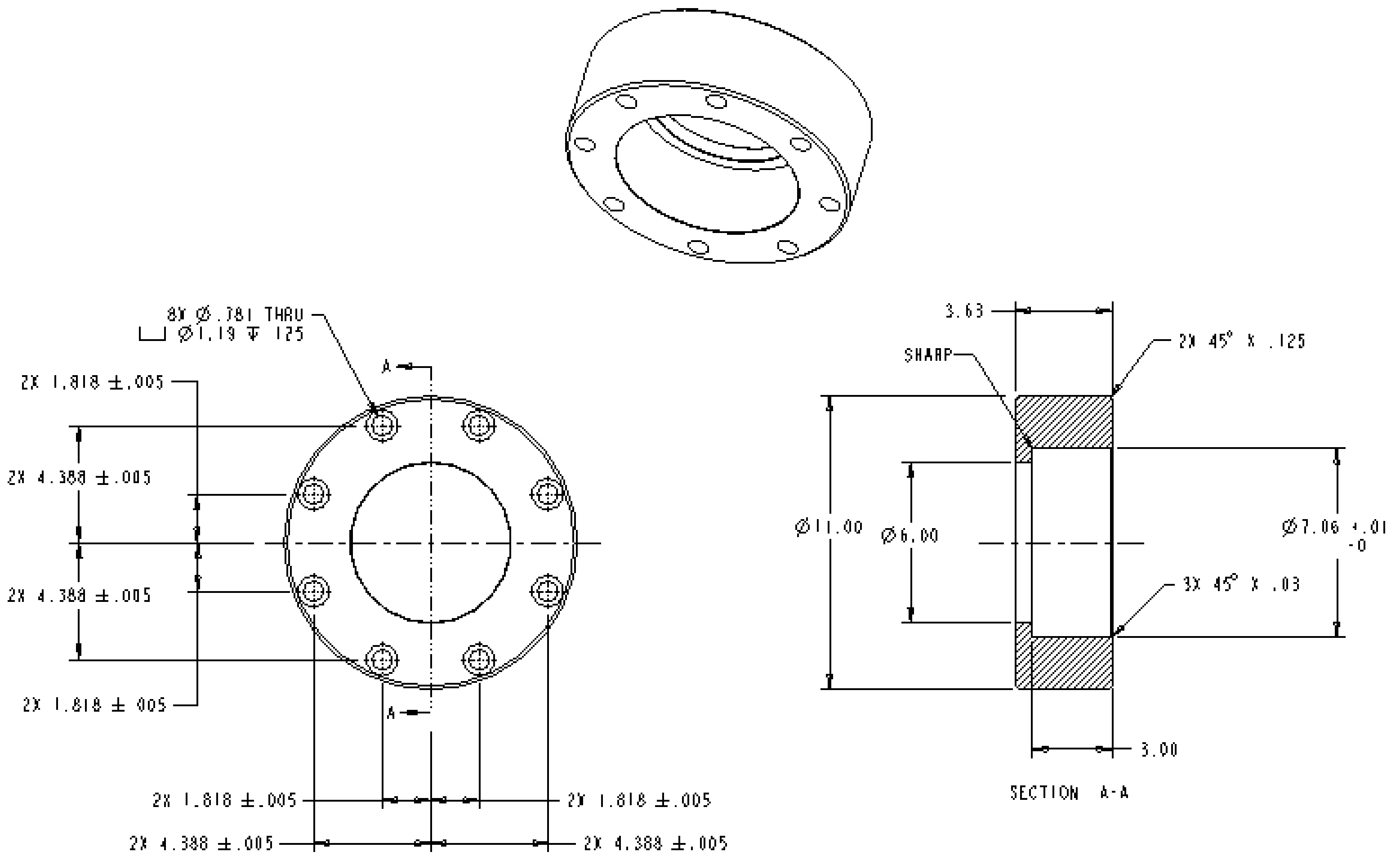


QUANTITY: 2

MATERIAL: FUSED SILICA TYPE GE124 OR EQUIVALENT

UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

TITLE: MAIN ACCESS WINDOW COMBUSTION TEST CHAMBER			
SCALE 1:2	DWG #: EN3-D10		LAST REVISED: 01/11/02
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 11/12/01
CSU ENGINES & ENERGY CONVERSION LAB			



QUANTITY: 2

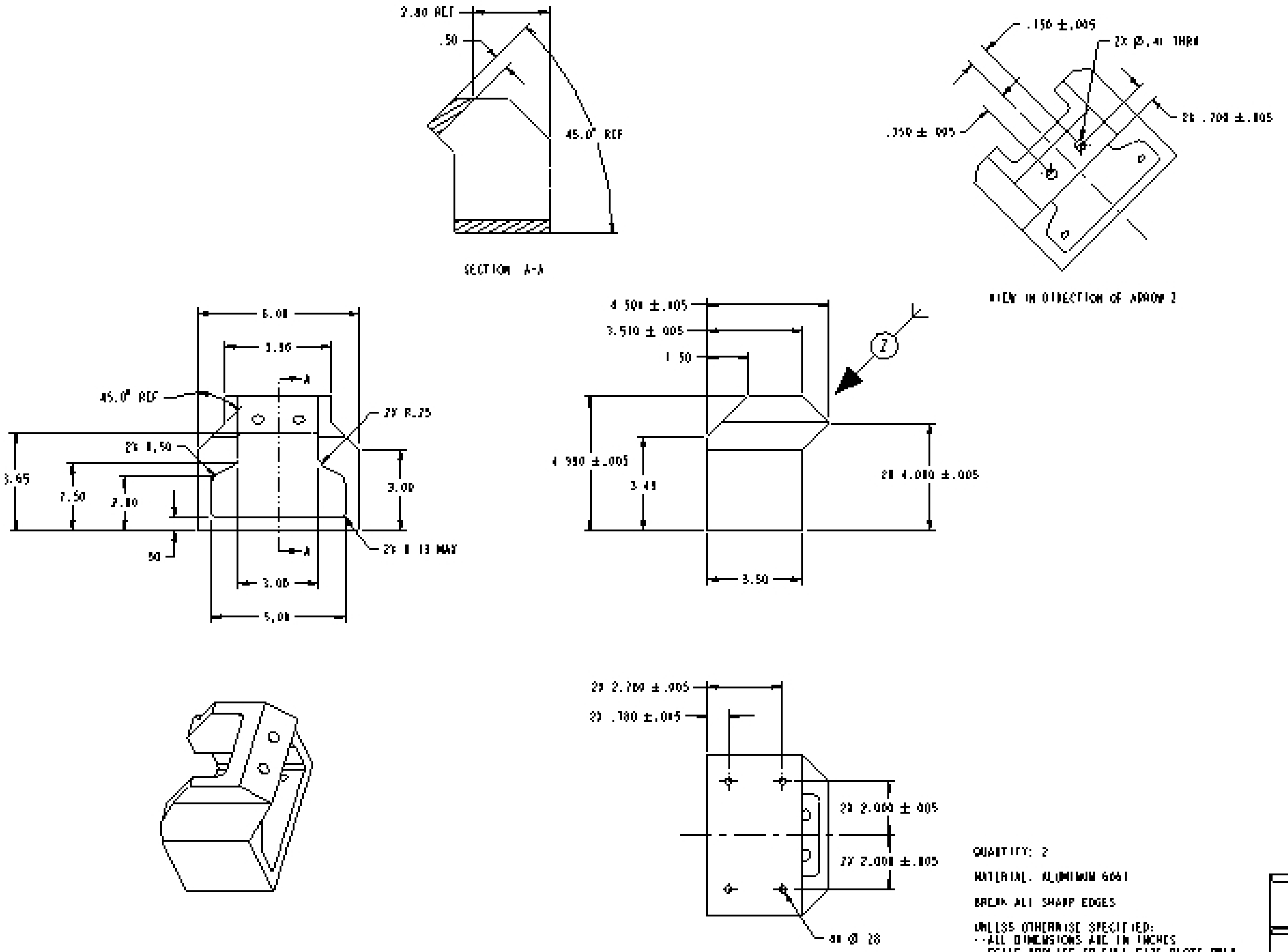
MATERIAL: STAINLESS STEEL GRADE 303S OR EQUIVALENT

UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

TITLE:
MAIN ACCESS WINDOW RETAINER
COMBUSTION TEST CHAMBER

SCALE: 1:3	DWG #: CM3-D11		LAST REVISED: 01/17/02
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 11/17/01

CSU ENGINES & ENERGY CONVERSION LAB



QUANTITY: 2

MATERIAL: ALUMINUM 6061

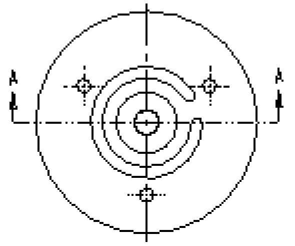
BREAK ALL SHARP EDGES

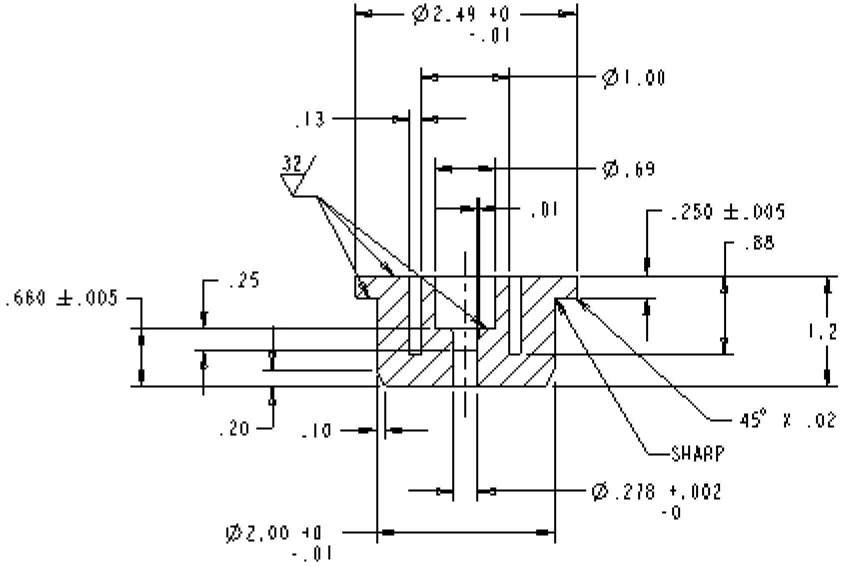
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

MAIN FRAME BRACKET
COMBUSTION TEST CHAMBER

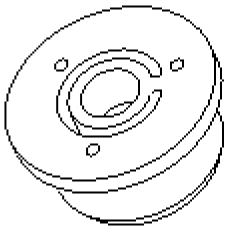
1:2	EMS-012		02/20/02
WALL	WALL	1 OF 1	01/16/02

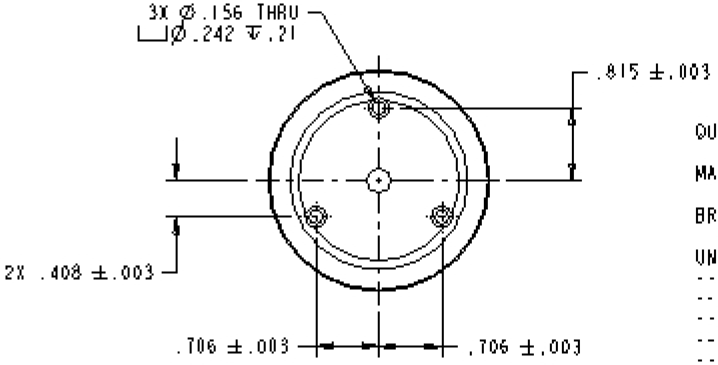
CSU ENGINES & ENERGY CONVERSION LAB





SECTION A-A





QUANTITY: 1

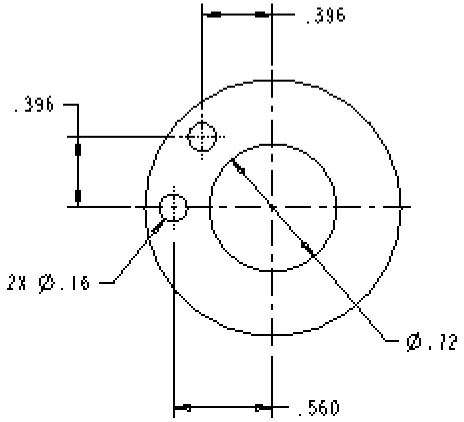
MATERIAL: STAINLESS STEEL GRADE 303S

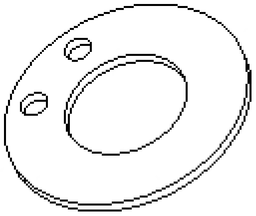
BREAK ALL SHARP EDGES

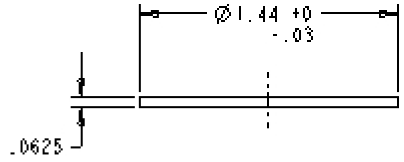
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

TITLE: LOWER DELCO INJECTOR SLEEVE COMBUSTION TEST CHAMBER			
SCALE: 1:1	DWG #: EN3-D13		LAST REVISED: 02/22/02
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 12/14/01

CSU ENGINES & ENERGY CONVERSION LAB







QUANTITY: 8 (ONLY 2 ROD)

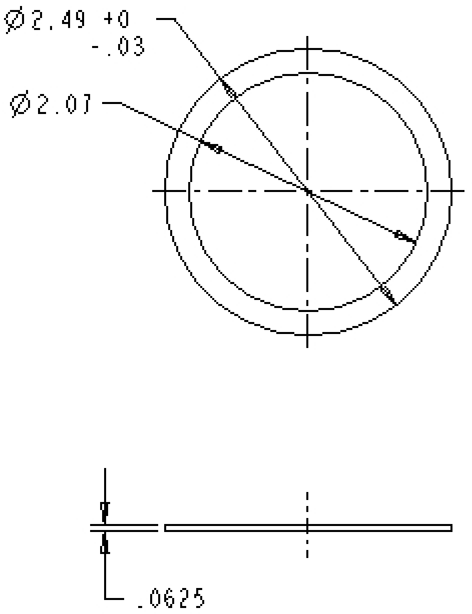
MATERIAL: BSSC GRAPHITE

UNLESS OTHERWISE SPECIFIED.
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.015
--ALL ANGLES ±1°

TITLE:
INTERNAL SLEEVE GASKET
COMBUSTION TEST CHAMBER

SCALE: 2:1	DWG #: EN3-D14		LAST REVISED: 01/25/02
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 01/04/02

CSU ENGINES & ENERGY CONVERSION LAB



QUANTITY: 8 (ONLY 2 ROD)

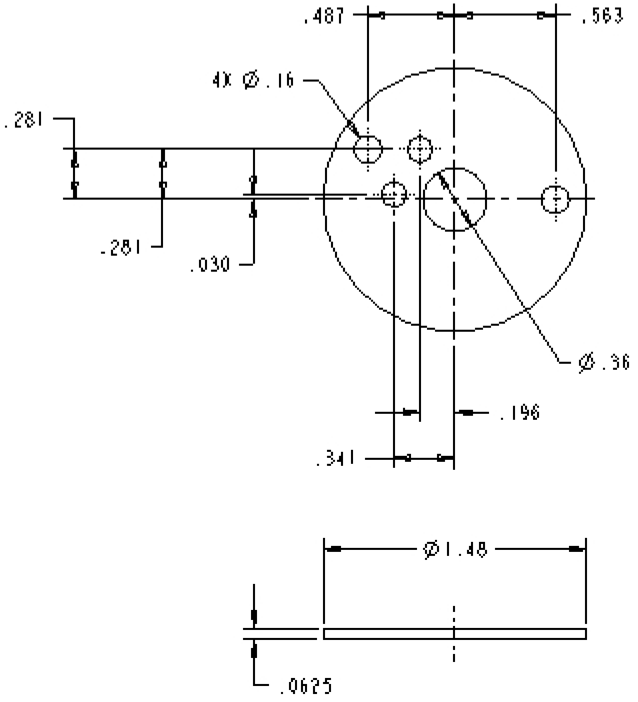
MATERIAL: BSSC GRAPHITE

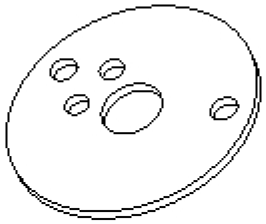
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.015
--ALL ANGLES ±1°

TITLE:
EXTERNAL SLEEVE GASKET
COMBUSTION TEST CHAMBER

SCALE: 1:1	DWG #: EM3-D15		LAST REVISED: 01/25/02
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 01/04/02

CSU ENGINES & ENERGY CONVERSION LAB





QUANTITY: 4 (ONLY 1 RQD)

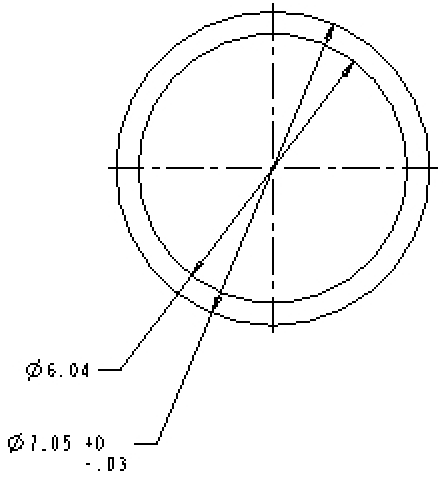
MATERIAL: BSSC GRAPHITE

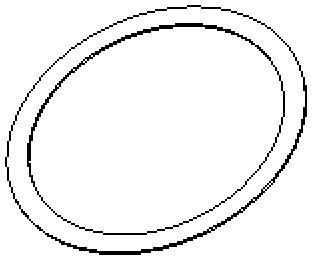
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.015
--ALL ANGLES ±1°

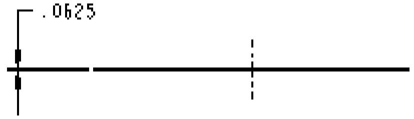
TITLE:
INTERNAL TRANSDUCER SLEEVE GSKT
COMBUSTION TEST CHAMBER

SCALE: 2:1
DWG #: EN3-D16
LAST REVISED: 01/25/02
DESIGN: WHULL
DETAIL: WHULL
SHEET: 1 OF 1
DRAWN: 01/04/02

CSU ENGINES & ENERGY CONVERSION LAB







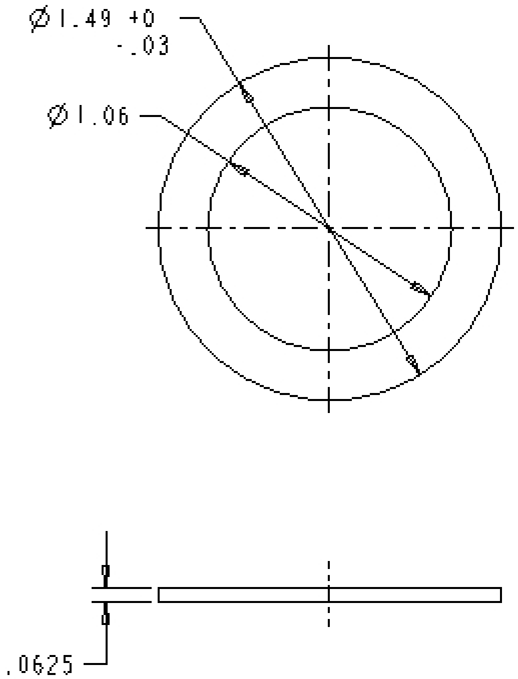
QUANTITY: 16 (ONLY 4 ROD)

MATERIAL: BSSC GRAPHITE

UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.015
--ALL ANGLES ±1°

TITLE: MAIN ACCESS WINDOW GASKET COMBUSTION TEST CHAMBER			
SCALE: 1:2	DWG #: EN3-D17		LAST REVISED: 01/25/02
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 11/17/01

CSU ENGINES & ENERGY CONVERSION LAB



QUANTITY: 4 (ONLY 1 ROD)

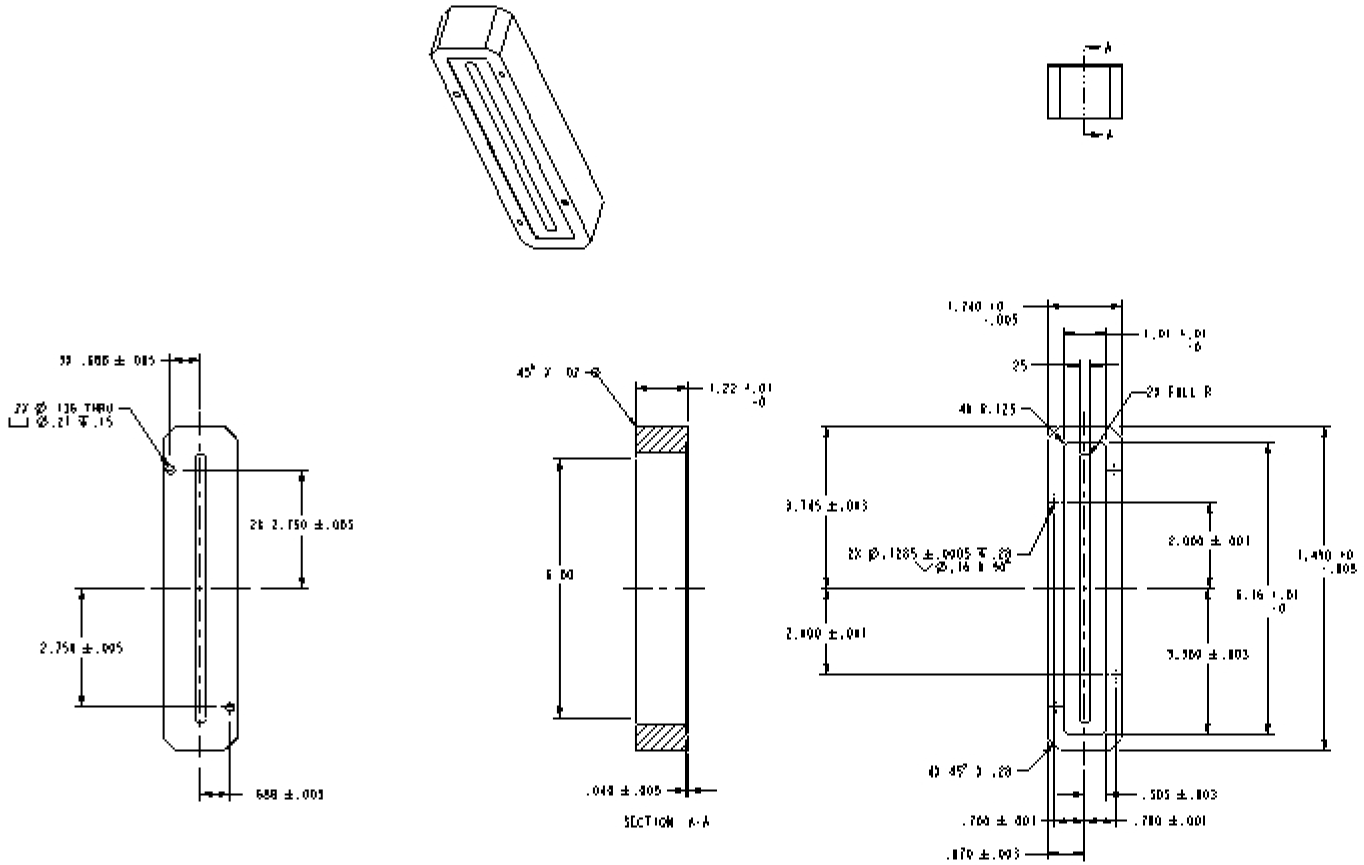
MATERIAL: BSSC GRAPHITE

UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.015
--ALL ANGLES ±1°

TITLE:
EXTERNAL TRANSDUCER SLEEVE GSKT
COMBUSTION TEST CHAMBER

SCALE: 2:1	DWG #: EM3-D19		LAST REVISED: 01/25/02
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 01/04/02

CSU ENGINES & ENERGY CONVERSION LAB



QUANTITY: 1

MATERIAL: STAINLESS STEEL GRADE 304

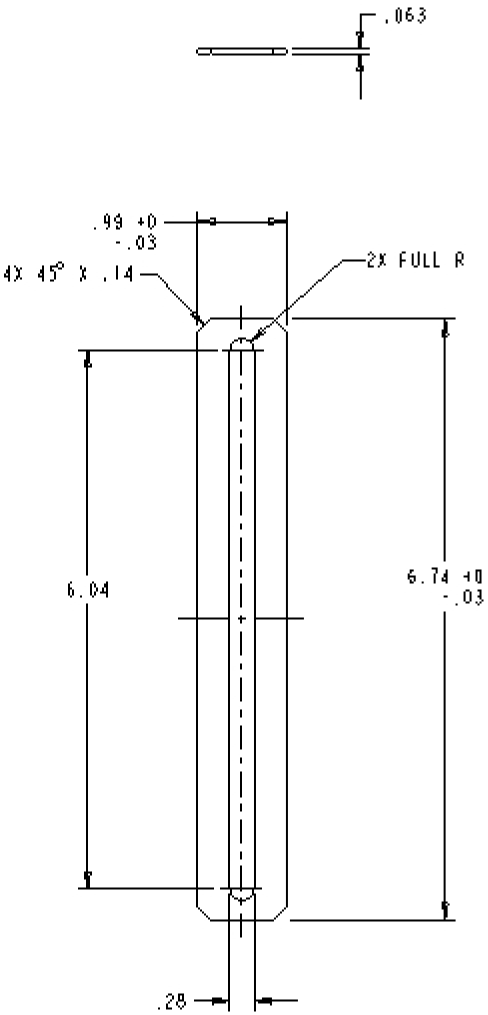
BREAK ALL SHARP EDGES

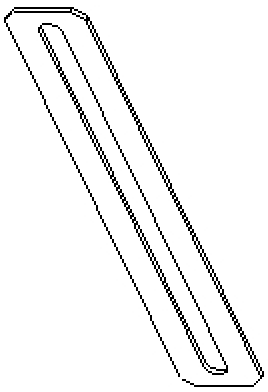
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

LASER ACCESS COVER
COMBUSTION TEST CHAMBER

3:4	EMS-D70		07/10/07
WALL	WALL	OF 1	01/08/07

CSU ENGINES & ENERGY CONVERSION LAB





QUANTITY: 8 (ONLY 2 RQD)

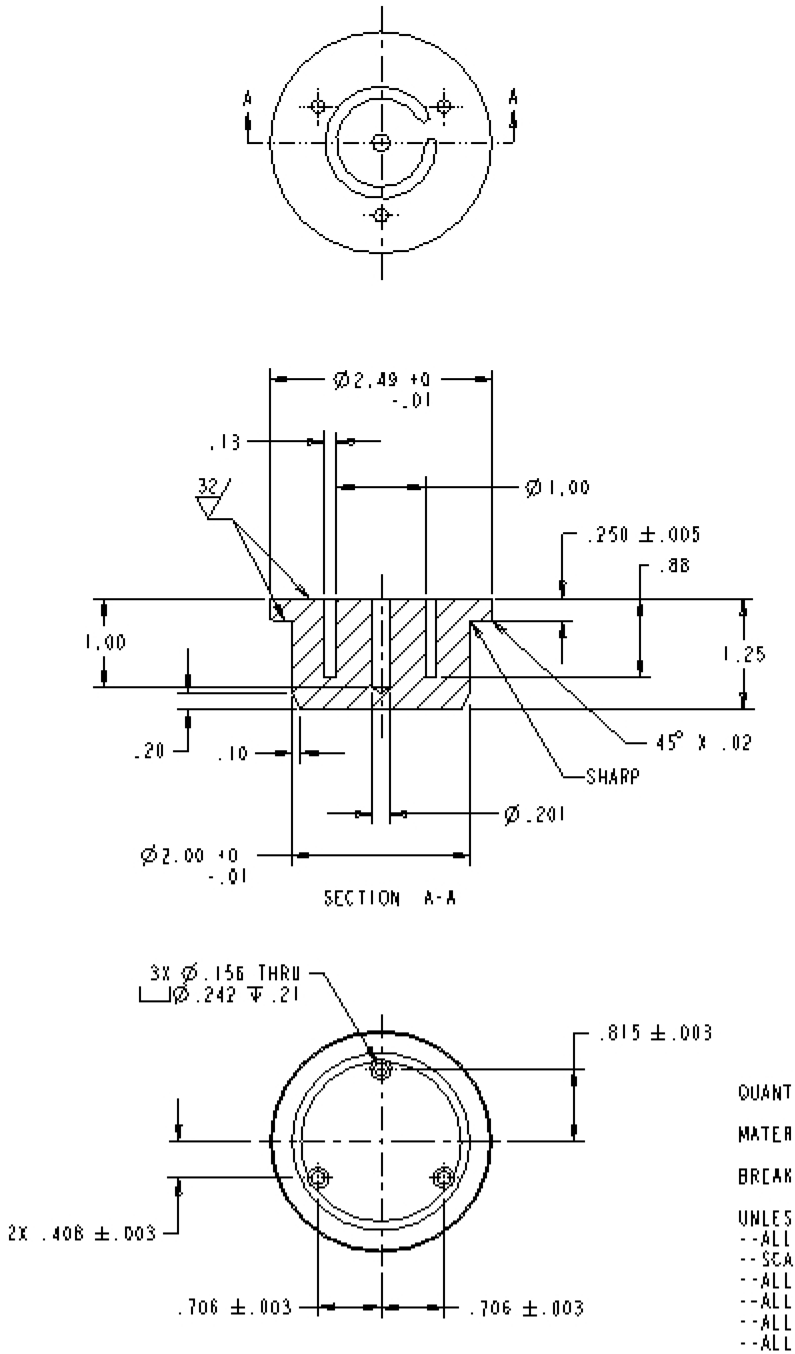
MATERIAL: BSSC GRAPHITE

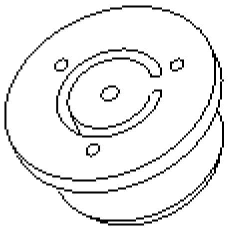
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.015
--ALL ANGLES ±1°

TITLE:
LASER ACCESS WINDOW GASKET
COMBUSTION TEST CHAMBER

SCALE: 1:1	DWG #: EN3-D21		LAST REVISED: 01/25/02
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 01/08/02

CSU ENGINES & ENERGY CONVERSION LAB





QUANTITY: 1

MATERIAL: STAINLESS STEEL GRADE 303S

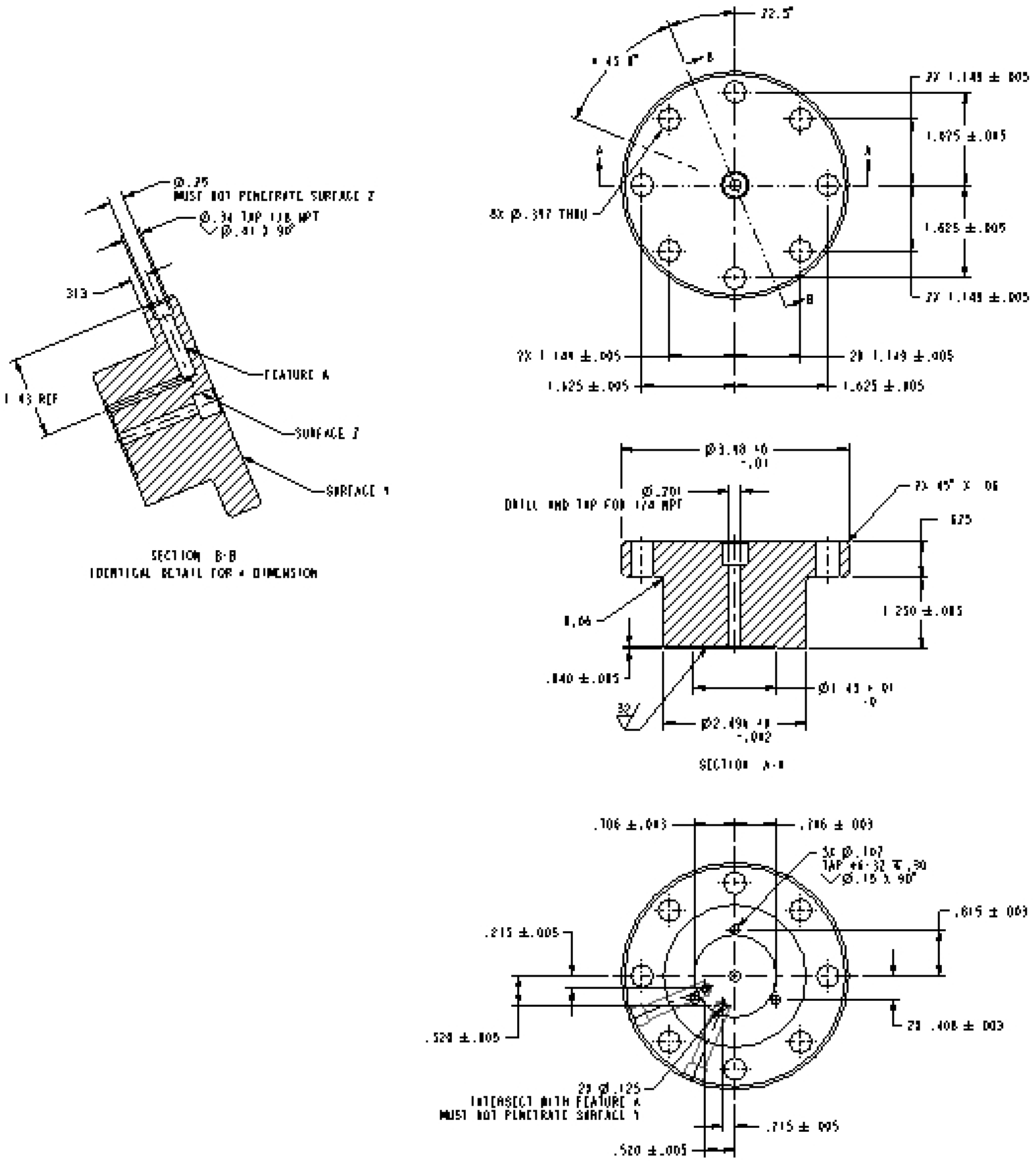
BREAK ALL SHARP EDGES

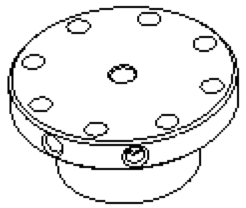
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

TITLE:
LOWER PROBE SLEEVE
COMBUSTION TEST CHAMBER

SCALE:	DWG #:		LAST REVISED:
1:1	EN3-D22		---
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 01/21/02

CSU ENGINES & ENERGY CONVERSION LAB





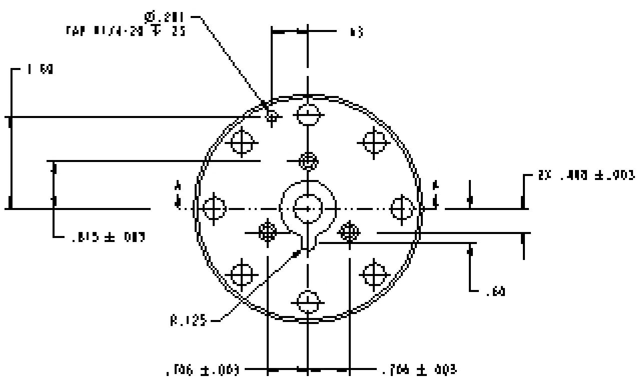
QUANTITY- 1

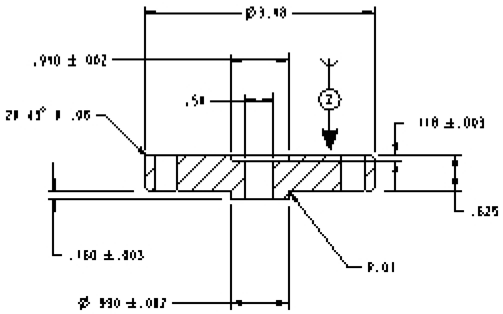
MATERIAL: STAINLESS STEEL GRADE 303S

BREAK ALL SHARP EDGES

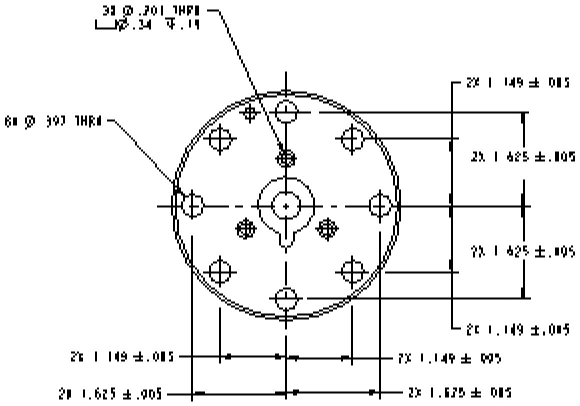
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ± .01
--ALL ANGLES ± 1°

UPPER PROBE SLEEVE COMBUSTION TEST CHAMBER			
1:1	EM3-073		...
NULL	NULL	01 1	01/21/02
CSU ENGINES & ENERGY CONVERSION LAB			

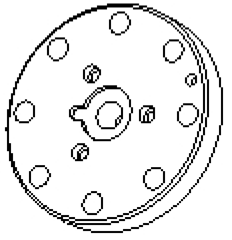


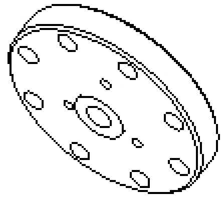


SECTION A-A



VIEW IN DIRECTION OF ARROW Z





QUANTITY: 1

MATERIAL: STAINLESS STEEL GRADE 303S

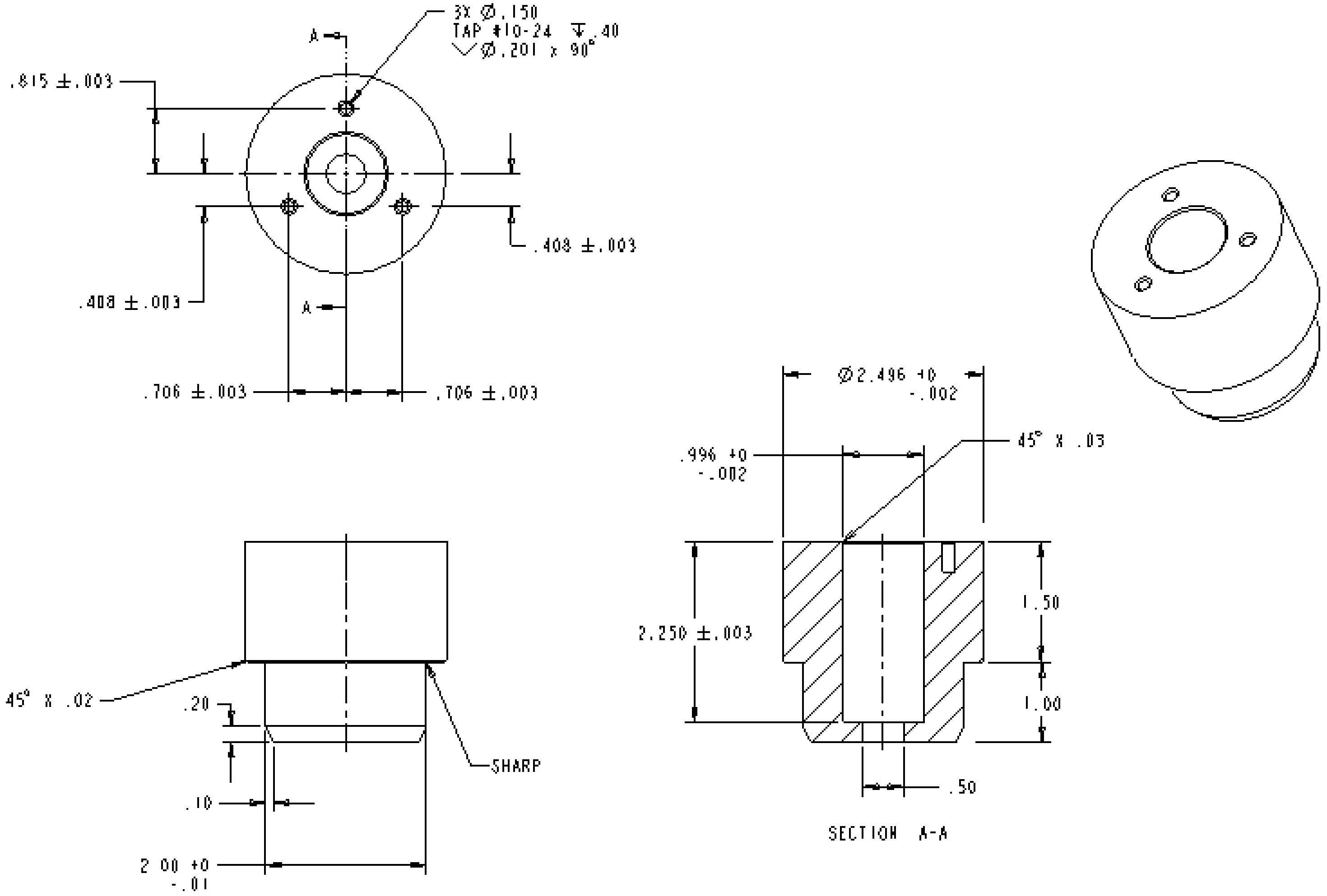
BREAK ALL SHARP EDGES

UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

UPPER LASER SLEEVE
COMBUSTION TEST CHAMBER

1:1	EW3-024		...
TJANISCH	JMAPLEN	OF 1	6/04/02

CSU ENGINES & ENERGY CONVERSION LAB



QUANTITY:1

MATERIAL: STAINLESS STEEL GRADE 303S

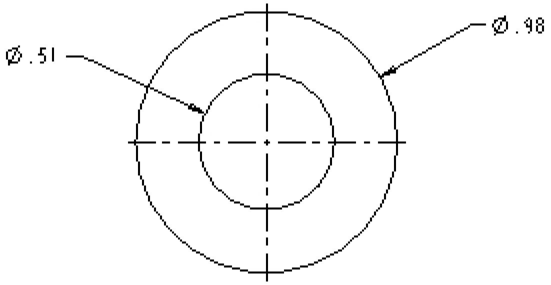
BREAK ALL SHARP EDGES

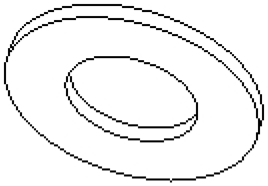
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

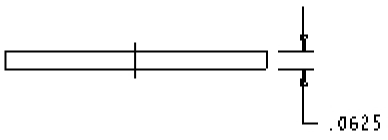
TITLE:
LOWER LASER SLEEVE
COMBUSTION TEST CHAMBER

SCALE:	DWG #:		LAST REVISED:
1:1	EM3-D25		---
DESIGN: FJAMISON	DETAIL: FJAMISON	SHEET: 1 OF 1	DRAWN: 06/05/02

CSU ENGINES & ENERGY CONVERSION LAB







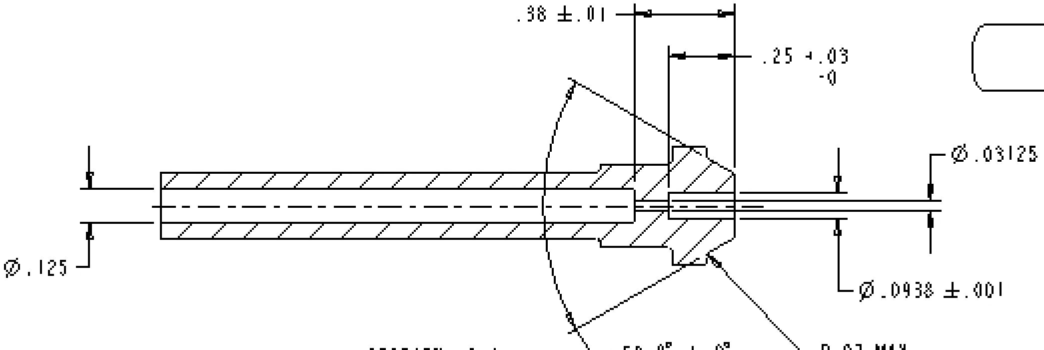
QUANTITY: 8

MATERIAL: BSSC GRAPHITE

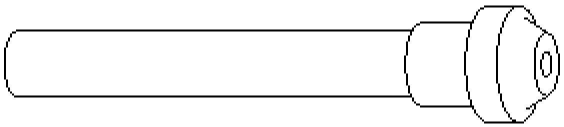
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.025
--ALL ANGLES ±1°

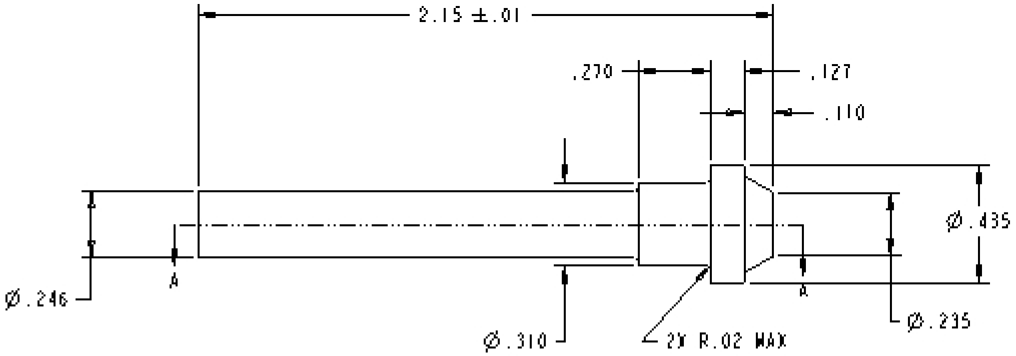
TITLE: LASER IGNITION GASKET COMBUSTION TEST CHAMBER			
SCALE: 1:1	DWG #: EM3-D26		LAST REVISED: ---
DESIGN: F JAMISON	DETAIL: F JAMISON	SHEET: 1 OF 1	DRAWN: 06/06/02

CSU ENGINES & ENERGY CONVERSION LAB



SECTION A-A



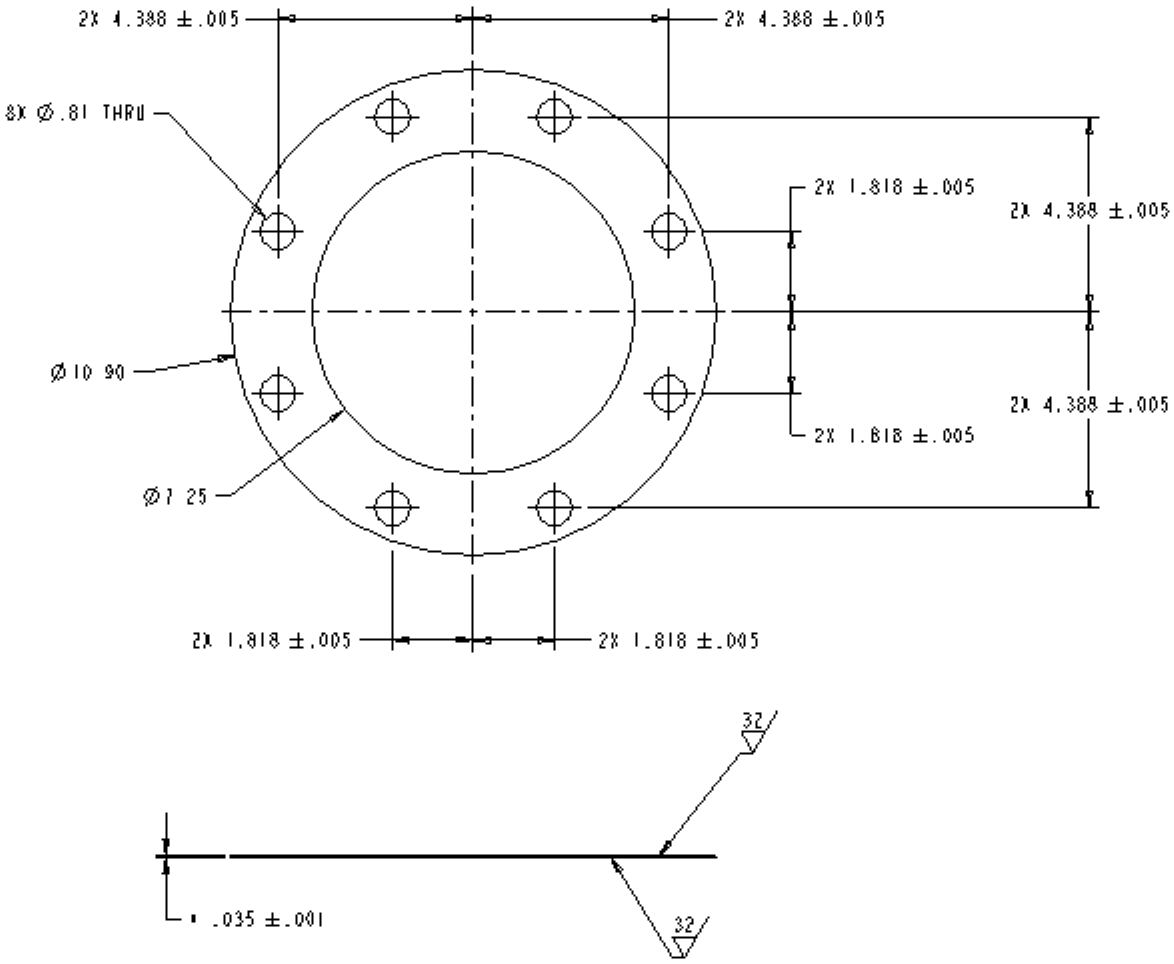


QUANTITY: 2

MATERIAL: AISI 1018 OR SIMILAR LOW CARBON STEEL

UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.003
--ALL ANGLES ±1°

TITLE: ORIFICE RETAINER COMBUSTION TEST CHAMBER			
SCALE: 3:1	DWG #: EN3-D31		LAST REVISED: ---
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 08/06/02
CSU ENGINES & ENERGY CONVERSION LAB			



SET #	* DIM
1	.035
2	.037
3	.039

QUANTITY: 12 TOTAL (3 SETS OF 4)

MATERIAL: SOFT COPPER

UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.015
--ALL ANGLES ±1°

TITLE: MAIN WINDOW INNER SEAL COMBUSTION TEST CHAMBER			
SCALE: 1:2	DWG #: EN3-D32		LAST REVISED: ---
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 08/13/02

CSU ENGINES & ENERGY CONVERSION LAB

APPENDIX 5

“1-Cylinder Prototype (CTC) Test Plan”

COMBUSTION TEST CHAMBER (CTC) TEST PLAN

- CTC heating
  - Start-up check-list for CTC heating

1. Power up control cart
   1.1. Turn on main cart power (2x)
   1.2. Power up computer
   1.3. Turn on IMV power supply to 13.6 V (constant voltage)
2. Turn on shop air supply to 90 psi
3. Cart acquisition verification
   3.1. Run (CTC VI) "CTC Control Interface_v.3"
   3.2. Ensure all inputs function
      3.2.1. Temperature bank (Rail & CTC)
      3.2.2. Pressure transducers (Rail & CTC)
   3.3. Ensure all outputs function
      3.3.1. High pressure bottle
      3.3.2. Exhaust valve
      3.3.3. IMV
4. Ensure adequate insulation
5. Pressure test to 1000 psi
   5.1. Verify no major leaks
6. Test Matrix Verification and setup (is testing equipment setup properly)
   6.1. EXECUTE NECESSARY TEST SETUP VERIFICATION
      6.1.1. Check to make sure the days test is possible BEFORE heating CTC
7. Begin Injector Cooling
   7.1. Hang sign at faucet
   7.2. Turn on supply water
   7.3. modulate valve to maintain 350 Kelvin
8. Initiate Heating
      8.1.1. Turn on Pre-Tower-Heater (2x)
      8.1.2. Turn on Tower Heater
      8.1.3. Turn on Post-Tower-Heater
      8.1.4. Turn on (CTC) cartridge heater
9. Monitor heating process until at required temperature for test

COMBUSTION TEST CHAMBER (CTC) TEST PLAN

- ❖ Injector Characterization
  - ➢ Start-up check-list for mapping diesel injectors

1. Power up control cart
    - 1.1. Turn on main cart power (2x)
    - 1.2. Computer
    - 1.3. Pulse generator
    - 1.4. O-scope
    - 1.5. Woodward control box
    - 1.6. IMV power supply
    - 1.7. Signal generator
    - 1.8. Current probe
2. Cart acquisition verification
    - 2.1. Run (CTC VI) "CTC Control Interface_v.3"
    - 2.2. Ensure all inputs function
        - 2.2.1. Temperature bank (Rail & CTC)
        - 2.2.2. Pressure transducers (Rail & CTC)
    - 2.3. Ensure all outputs function
        - 2.3.1. High pressure bottle
        - 2.3.2. Exhaust valve
        - 2.3.3. IMV
3. Start Woodward impulse software
    - 3.1. Run CTC Control Interface_v.3
    - 3.2. Run Serv-Link
        - 3.2.1. open mpCTC
    - 3.3. Open Watch-Window
        - 3.3.1. Hit Quick Update
4. Setup Test Stand
    - 4.1. Align and turn on Metler Toledo RG 2459 Scale on Optics Table
    - 4.2. Setup Injector Holder
        - 4.2.1. Install injector and record injector number
    - 4.3. Build injector measurement cup and place on scale
    - 4.4. Ensure adequate tip insertion into measurement cup
5. Verify current output to injector using Woodard box and O-scope
6. Start fuel common rail system
    - 6.1. Set eject limit over test pressure
    - 6.2. Bring up to test pressure
7. Test fire
    - 7.1. Check setup
    - 7.2. Purge Line
8. Zero scale
    - 8.1. Take data points

COMBUSTION TEST CHAMBER (CTC) TEST PLAN

- Mie scattering of Injection
  - Start-up check-list for Mie scattering images

10. Record and verify injector orientation in CTC
    - 10.1. One jet must face the laser window
11. Power up control cart
    - 11.1. Computer
    - 11.2. Pulse Generator
    - 11.3. O-Scope
    - 11.4. Woodward control box
    - 11.5. IMV Power Supply
    - 11.6. Laser
12. Verify Connections
    - 12.1. Laser
    - 12.2. Woodward impulse
    - 12.3. Camera
    - 12.4. Current Probe
13. Cart acquisition verification
    - 13.1. Run (CTC VI) "CTC Control Interface_v.3"
    - 13.2. Ensure all inputs function
      - 13.2.1. Temperature bank (Rail & CTC)
      - 13.2.2. Pressure transducers (Rail & CTC)
    - 13.3. Ensure all outputs function
      - 13.3.1. High pressure bottle
      - 13.3.2. Exhaust valve
      - 13.3.3. IMV
14. Pressure test to 1000 psi
    - 14.1. Verify no leakage
15. Exhaust CTC pressure
16. Setup-Align Camera
    - 16.1. Zoom and focus tip
    - 16.2. Aperture = max open
    - 16.3. Shutter = 2 or 3 seconds
17. Start Kodak Program
    - 17.1. Choose saving location
    - 17.2. Take a sample photo
      - 17.2.1. verify proper function
18. Begin laser alignment
    - 18.1. Set laser to 5 Hz low power
    - 18.2. Ensure high intensity tip illumination of laser window
    - 18.3. Set Flash lamp and Q-switch durations and change to external signal
      - 18.3.1. Fire externally and check operation
19. Injection Verification/line purge
    - 19.1. Pressurize Fuel rail to 4000 psi

    19.2. Send several long pulses to injector driver
    19.3. Shut off Lights and verify picture capturing, and oscilloscope operation
20. Shut down common rail system
21. Initiate Heating
    21.1. Follow Heating start-up check-list
    21.2. Wait for test temperature (at least 5 hours)
22. Taking Data Points (Each point has a VI data Tag, Picture, and an O-scope image
    22.1. Set VI data file path to a dated folder
    22.2. Verify Location of picture folder
    22.3. Set VI data tags (Photo number, injector #, duration, etc…)
    22.4. Open high pressure Nitrogen Valve (wait two seconds)
    22.5. Close exhaust Valve
    22.6. At pressure open the camera shutter
    22.7. Fire the injector from the VI
    22.8. Save the O-scope screen to a matching file
    22.9. Move the delay of the injector firing back a few micro-seconds
    22.10. Go back to step 13.7, repeat 3 times
    22.11. Open the exhaust valve and wait for the temperature to rise
    22.12. Go back to step 13.4
    22.13. Repeat until the test matrix has been satisfied
23. Follow CTC shutdown check-list

COMBUSTION TEST CHAMBER (CTC) TEST PLAN

- CTC Shut-down
  - Shut-down check-list for CTC

24. Shut off heaters
    - 24.1.1. Turn off Pre-Tower-Heater (2x)
    - 24.1.2. Turn off Tower Heater
    - 24.1.3. Turn off Post-Tower-Heater
    - 24.1.4. Turn off (CTC) cartridge heater
25. Continue Injector Cooling!!!
    - 25.1. Continue to cool until all temperatures are below max injector temp
26. Shut down control cart accessories
    - 26.1. Pulse generator
    - 26.2. O-scope
    - 26.3. Woodward control box
    - 26.4. IMV power supply
    - 26.5. Signal generator
    - 26.6. Current probe
27. When CTC temperature are < max injector temperature
    - 27.1. Shut off cooling water
    - 27.2. Shut down computer
    - 27.3. Shut off all cart power

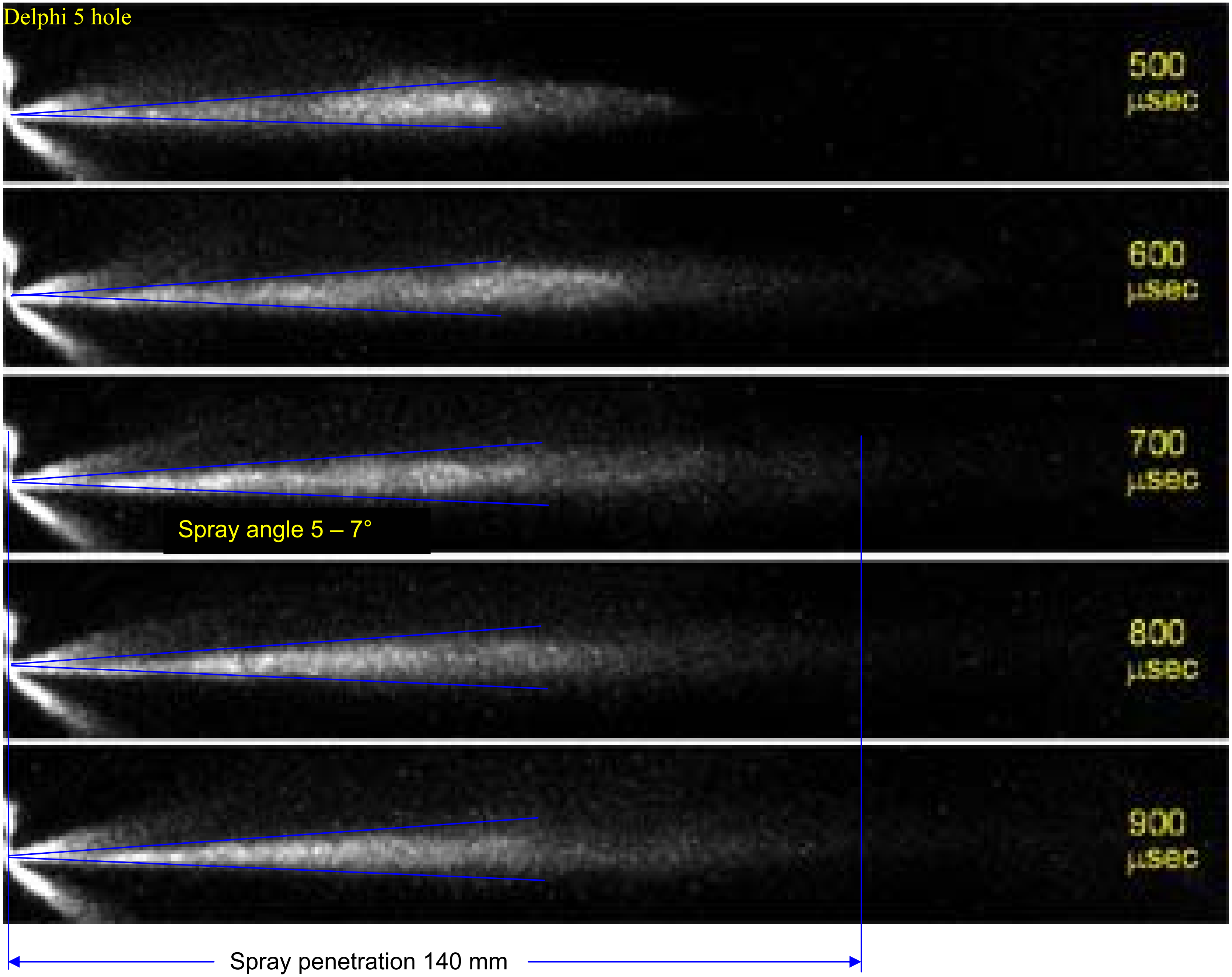
APPENDIX 6

“Combustion Test Chamber Data and Conclusions”

Combustion Test Chamber Data and Conclusions

Specifications for the prototype fuel injection system were outlined in Task 3. These included being able to inject 1mm^3 to 20mm^3, rail pressures of 400 to 1300 bar, and 50 – 250mm of spray penetration. In addition, the single cylinder tests were to provide data on spray angle and direction so that appropriate nozzles could be designed for optimizing the fuel propagation into the Cooper GMV clearance volume.

The Delphi system has met the specifications outlined. In addition, spay angles of each plume are 5 to 7 degrees. Spray penetration and angles are described in the picture on the following page. These images were taken in the CTC with the Mie Scattering technique using a constant light source.


Delphi 5 hole
500 µsec
600 µsec
700 µsec
800 µsec
900 µsec
Spray angle 5 – 7°
Spray penetration 140 mm

APPENDIX 7

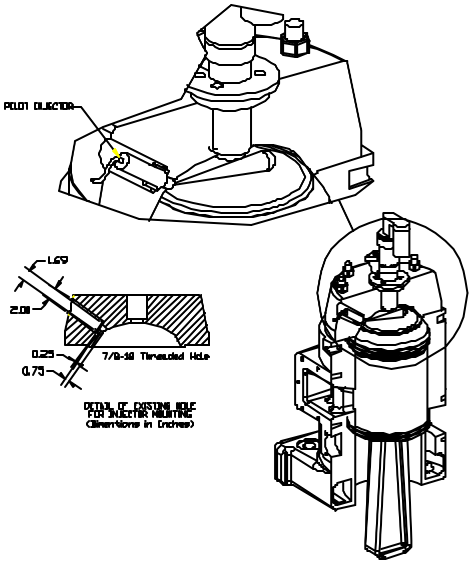
“4-Cylinder Prototype Specifications”

Engines and Energy Conversion Laboratory

Department of Mechanical Engineering
Fort Collins, Colorado 80523-1374
(970) 490-1418
FAX: (970) 493-6403
http://www.engr.colostate.edu/eecl

Micropilot Ignition Project
4 cylinder test specification sheet

The Micropilot Ignition Project being conducted at Colorado State University will assess the benefits of retrofitting diesel pilot ignition systems on large-bore, lean burn, natural gas engines. A three-year program has been developed to integrate the necessary research and development efforts. Under Phase I of the program, the single cylinder injection system will be converted to a four-cylinder, on-engine system.



The EECL will construct a 4-cylinder system (see attached schematic) that meets the following speculations:

Pump Motor – The current motor is 1.5 hp. A 2-3 hp motor will be required for the on-engine test due to the larger demand placed on the pump.

High Pressure Fuel Lines – Fuel lines capable of delivering fuel at 20 kpsi are required to distribute fuel the four cylinders.

Coolant Pump – A pump capable of circulating coolant through the cooling system will be required.

Coolant Tank and Lines – Due to the continuous operation of the engine, a cooling system may be required for proper operation of the injector. The system will require lines capable of withstanding the temperatures found around the engine.

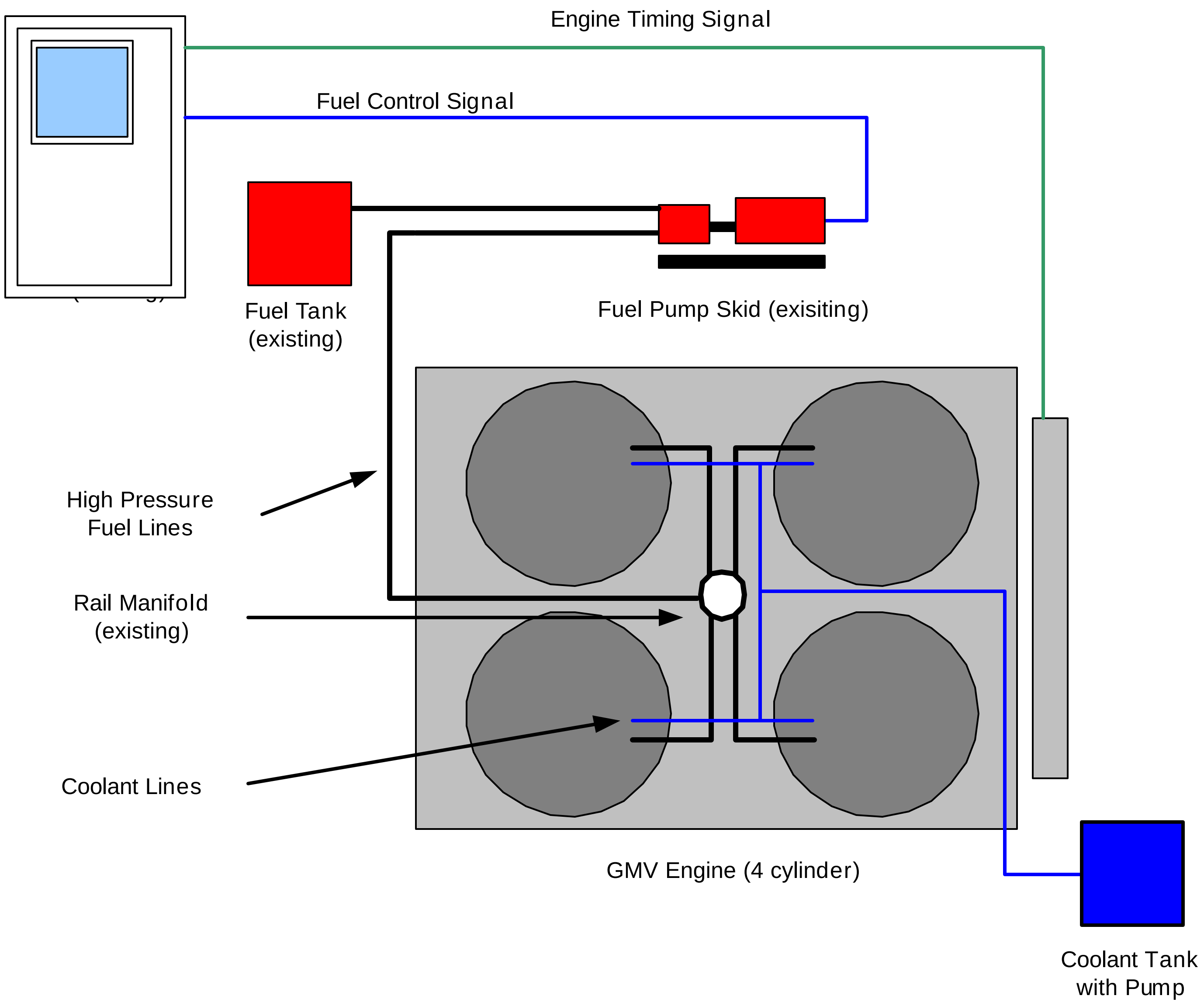
APPENDIX 8

“4-Cylinder Prototype Design Documentation”

Specification Sheet for DOE/CSU Micropilot Project

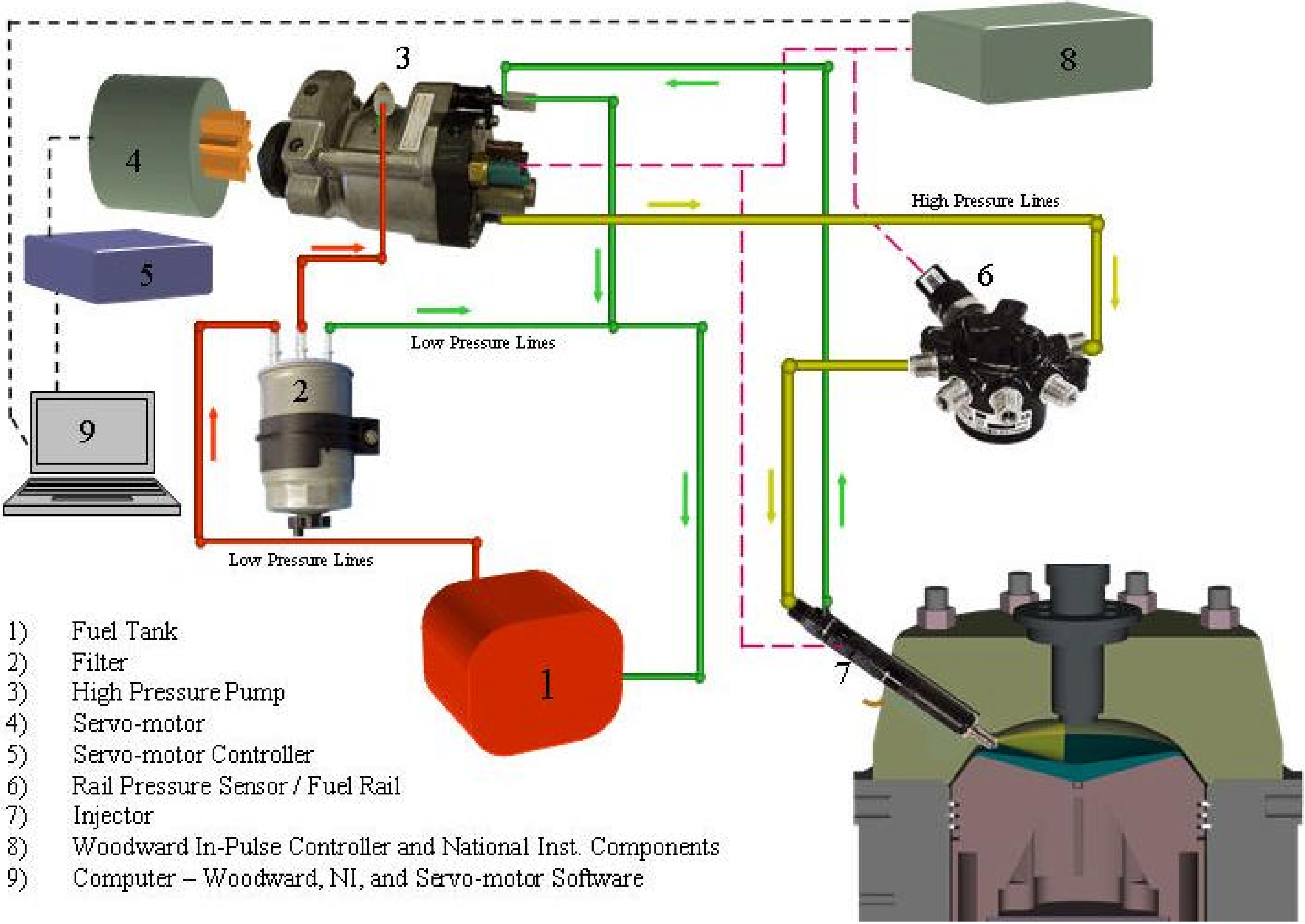
Revised April 7, 2002

4 cylinder, on-engine test



Additional Items
High Pressure Fuel Lines
Larger Pump Motor
Coolant Pump
Coolant Tank and Lines

Fuel System for Common Rail fuel-injection system



1) Fuel Tank
2) Filter
3) High Pressure Pump
4) Servo-motor
5) Servo-motor Controller
6) Rail Pressure Sensor / Fuel Rail
7) Injector
8) Woodward In-Pulse Controller and National Inst. Components
9) Computer – Woodward, NI, and Servo-motor Software

QTY	SUFFIX	NAME	QTY	SUFFIX	NAME
			2	D06	12:1 COMPRESSION PLATE
			2	D05	11:1 COMPRESSION PLATE
			2	D04	MODIFIED FUEL INJECTOR
			1	D03	INSERT INSTALLATION TOOL
			2	D02	FUEL INJECTOR INSERT
			2	D01	MODIFIED CYLINDER HEAD
			2	A03	MODIFIED PISTON ASSEMBLY
			2	A02	INJECTOR INSERT ASSEMBLY
			2	A01	MASTER ASSEMBLY

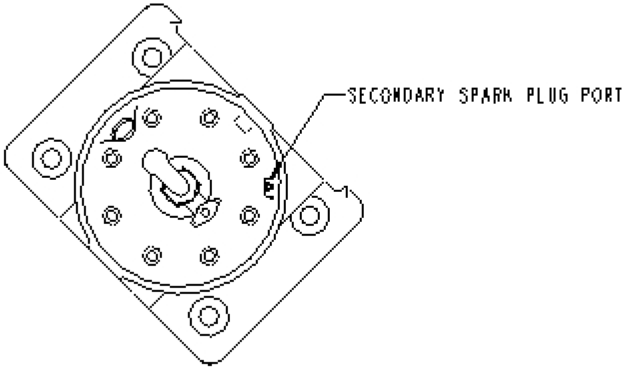
TITLE:

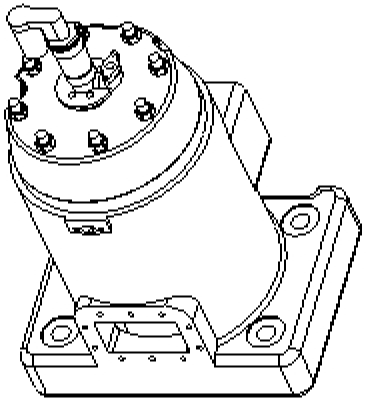
MASTER ASSEMBLY LIST
GMV MICROPILOT IGNITION PROJECT

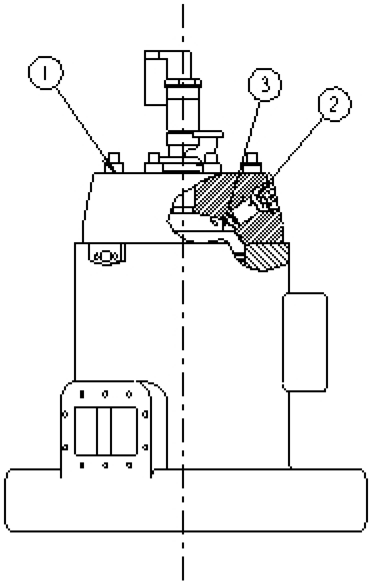
SCALE:	DWG #:		LAST REVISED:
NONE	EM2-LST		- - -
DESIGN:	DETAIL:	SHEET:	DRAWN:
WHULL	WHULL	1 OF 1	05/22/02

CSU ENGINES & ENERGY CONVERSION LAB

1	1	EM2-D01	MODIFIED CYLINDER HEAD
2	2	EM2-A02	INJECTOR INSERT ASSEMBLY
3	3	EM2-A03	MODIFIED PISTON ASSEMBLY







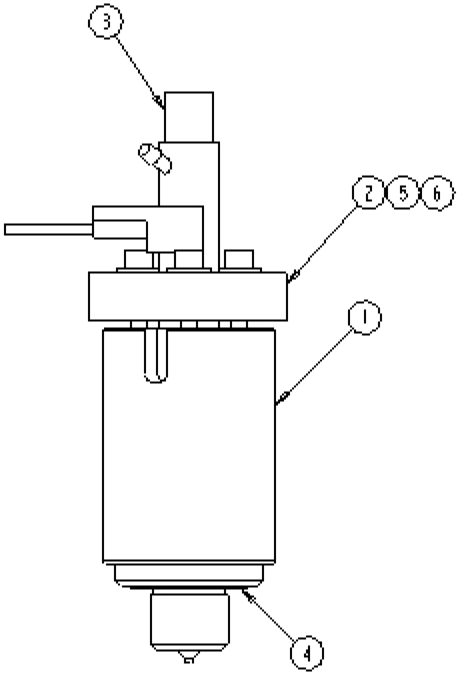
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

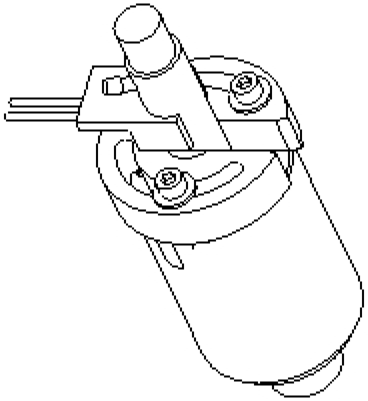
TITLE:
MASTER ASSEMBLY
GMV MICROPILOT IGNITION PROJECT

SCALE: 1:10	DWG #: EM2-A01		LAST REVISED: ---
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 05/23/02

CSU ENGINES & ENERGY CONVERSION LAB

1	1	EM2-D02	FUEL INJECTOR INSERT
2	1	EM3-D03	FUEL INJECTOR RETAINER
3	1	PURCHASE	AC DELCO 93082808 INJECTOR
4	1	PURCHASE	VITON -123 ORING
5	3	PURCHASE	SHCS #10-24 X 1-1/4 LG
6	3	PURCHASE	FLAT WASHER SAE #10





INJECTOR MUST BE ORIENTED SO THAT PILOT HOLE FACES TOP OF CYLINDER HEAD

TORQUE SPECIFICATIONS:
FUEL INJECTOR RETAINER SCREWS: 30-35 IN-LBS

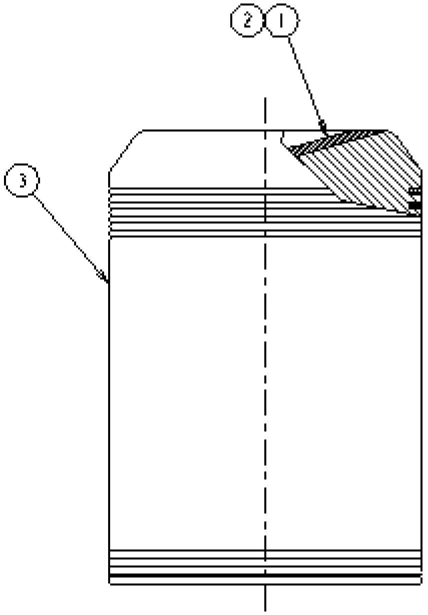
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ± .01
--ALL ANGLES ± 1°

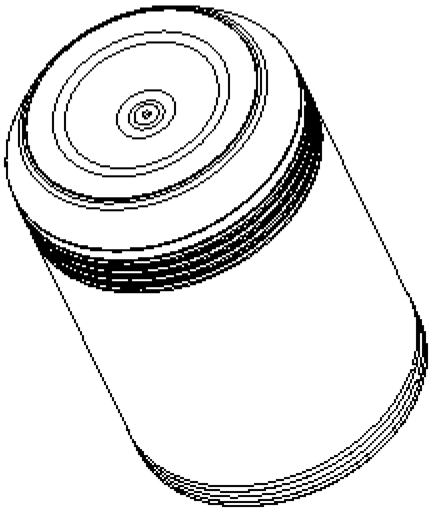
TITLE:
INJECTOR INSERT ASSEMBLY
GMV MICROPILOT IGNITION PROJECT

SCALE: 1:1	DWG #: EM2-A02		LAST REVISED: 06/03/02
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 05/21/02

CSU ENGINES & ENERGY CONVERSION LAB

1	1	EM2-D06	12:1 COMPRESSION PLATE
2	1	PURCHASE	FHCS 5/8-11 X 1-1/2 LG
3	1	STOCK	GMV PISTON & RINGS





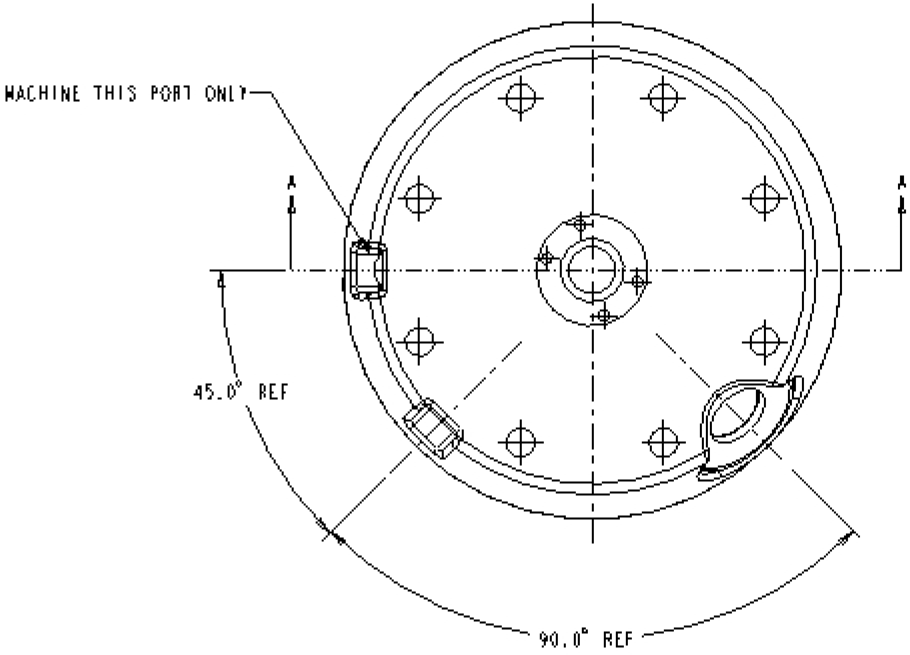
ITEM 1 IS INTERCHANGEABLE WITH EM2-D05
(11:1 COMPRESSION PLATE)

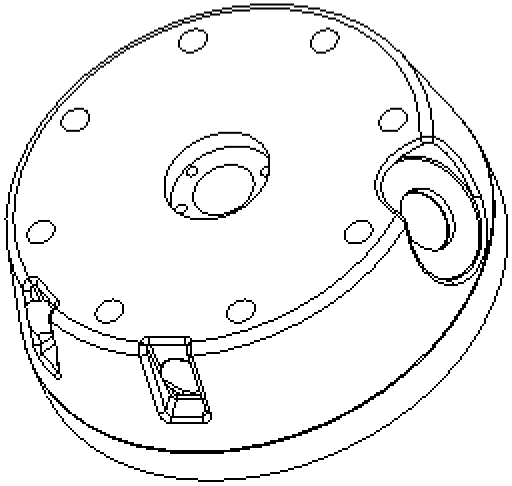
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

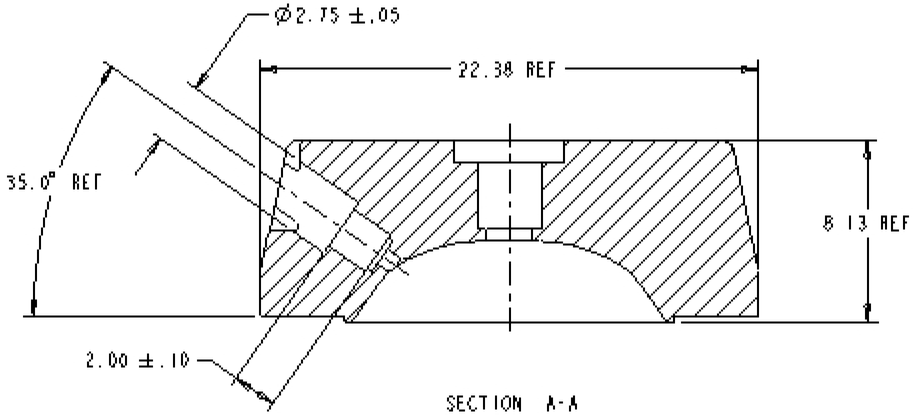
TITLE:
MODIFIED PISTON ASSEMBLY
GMV MICROPILOT IGNITION PROJECT

SCALE:	DWG #:		LAST REVISED:
1:4	EM2-A03		---
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 05/22/02

CSU ENGINES & ENERGY CONVERSION LAB







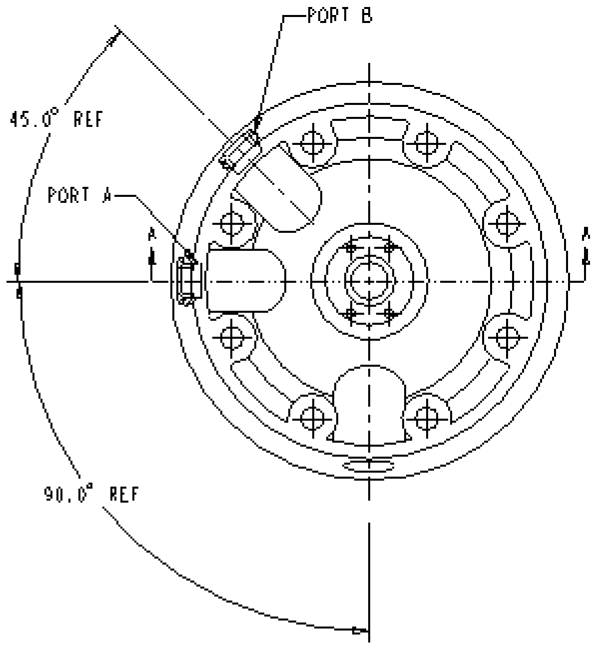
SECTION A-A

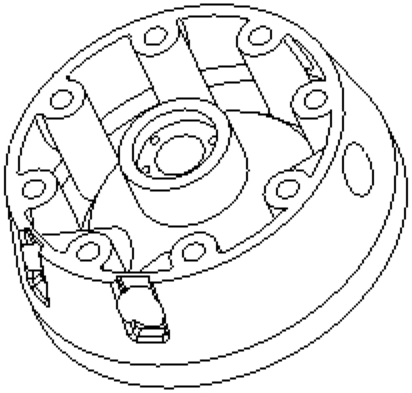
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

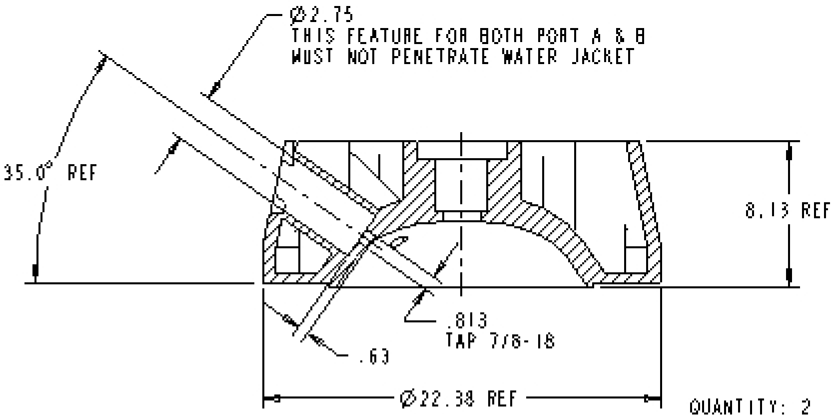
TITLE:
CYLINDER HEAD MODIFICATION
GMV MICROPILOT IGNITION PROJECT

SCALE: 1:4	DWG #: EN2-D01		LAST REVISED: ---
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 05/21/02

CSU ENGINES & ENERGY CONVERSION LAB







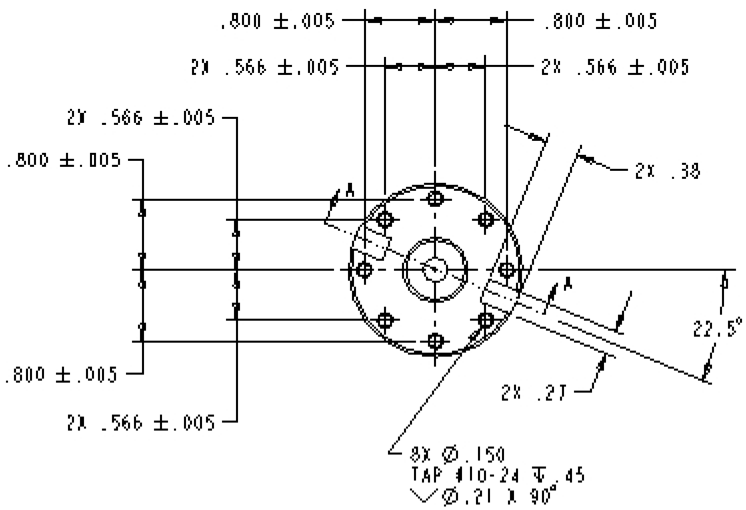
SECTION A-A

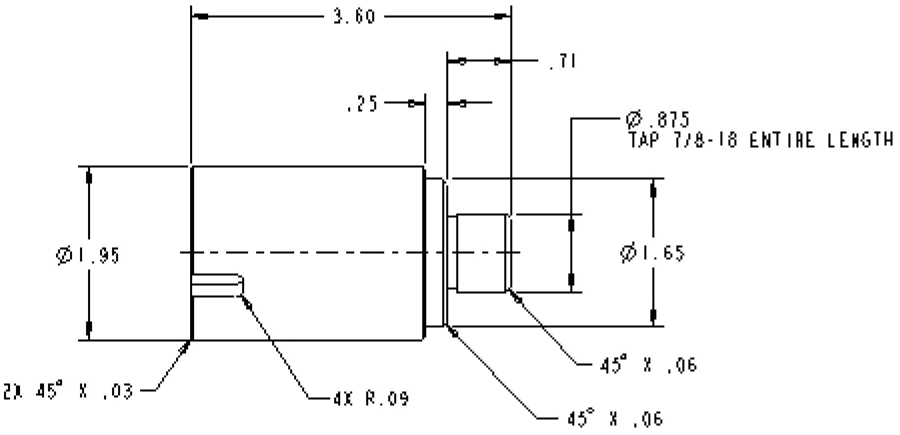
QUANTITY: 2

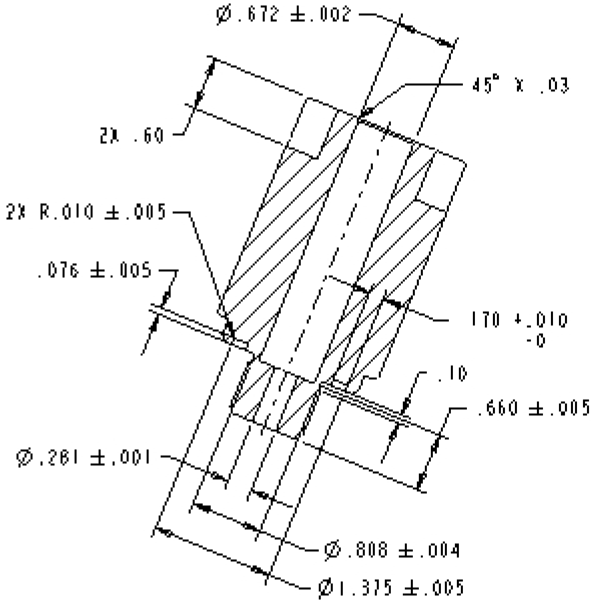
MODIFY PER DRAWING SPECIFICATIONS

UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

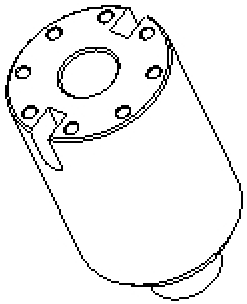
TITLE: CYLINDER HEAD MODIFICATION GMV MICROPILOT IGNITION PROJECT			
SCALE: 1:5	DWG #: EW2-D01_B		LAST REVISED: ---
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 12/11/02
CSU ENGINES & ENERGY CONVERSION LAB			







SECTION A-A



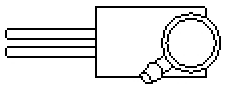
QUANTITY: 2

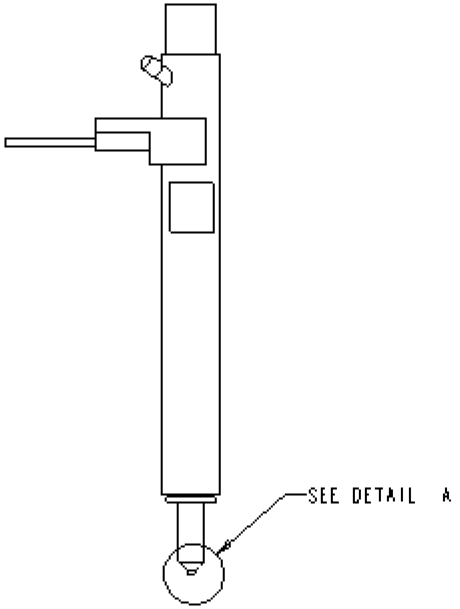
MATERIAL: AISI 1018 OR SIMILAR LOW CARBON STEEL

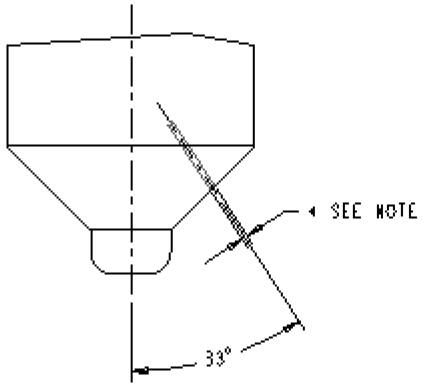
BREAK ALL SHARP EDGES

UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

TITLE: FUEL INJECTOR INSERT GMV MICROPILOT IGNITION PROJECT			
SCALE: 1:1	DWG #: EN2-D02		LAST REVISED: 07/11/02
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 04/26/02
CSU ENGINES & ENERGY CONVERSION LAB			







DETAIL A
SCALE 10.000

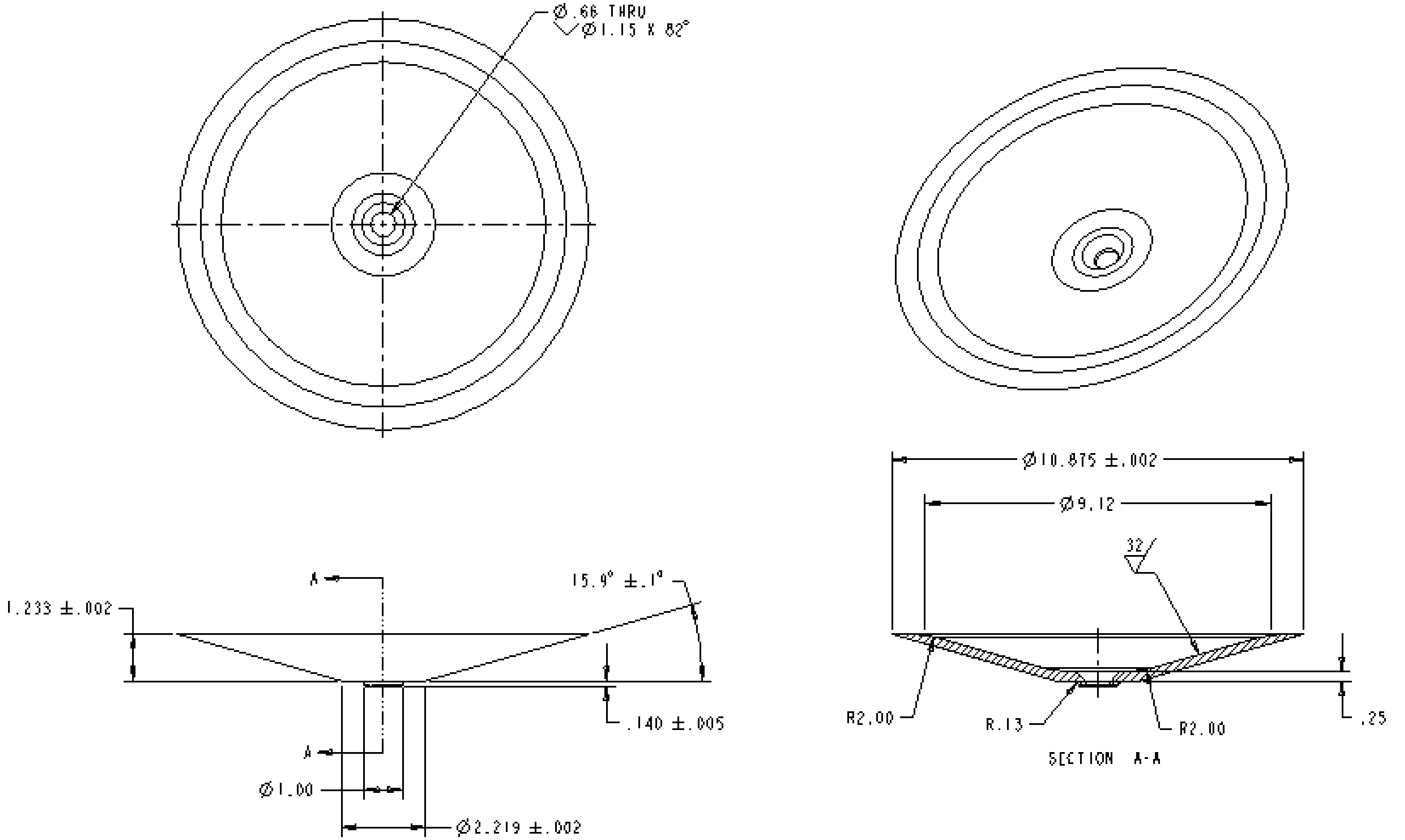
GEOMETRY IS APPROXIMATE

CONNECTOR ORIENTATION AS SHOWN

4 Ø.14 mm, Ø.165 mm, Ø.20 mm WILL BE REQUIRED
LOCATE AS NECESSARY WHILE MAINTAINING GIVEN
ANGULAR DIMENSION

UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

TITLE: INJECTOR HOLE REQUIREMENTS GMV MICROPILOT IGNITION PROJECT			
SCALE: 1:1	DWG #: EN2-D04		LAST REVISED: 05/21/02
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 03/07/02
CSU ENGINES & ENERGY CONVERSION LAB			



QUANTITY: 1

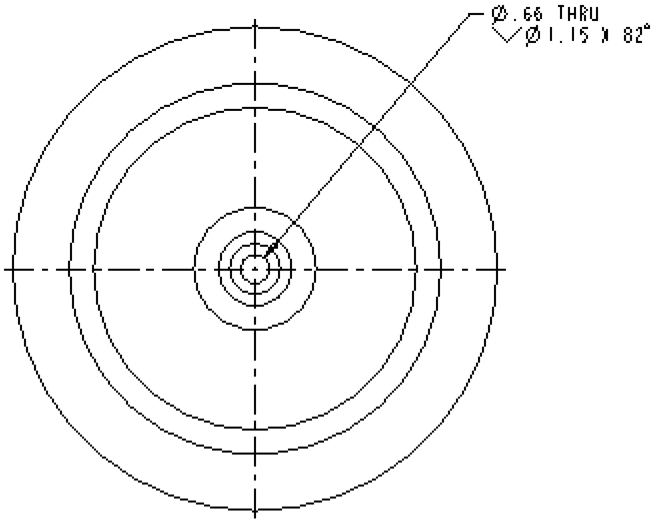
MATERIAL: AISI 1018 OR SIMILAR LOW CARBON STEEL

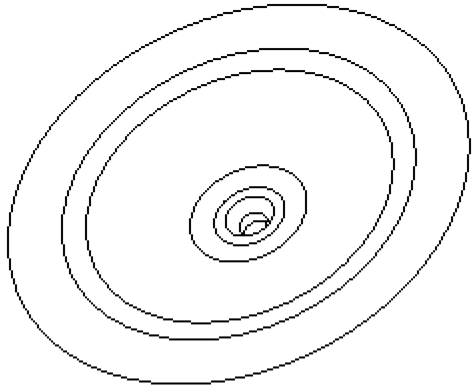
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

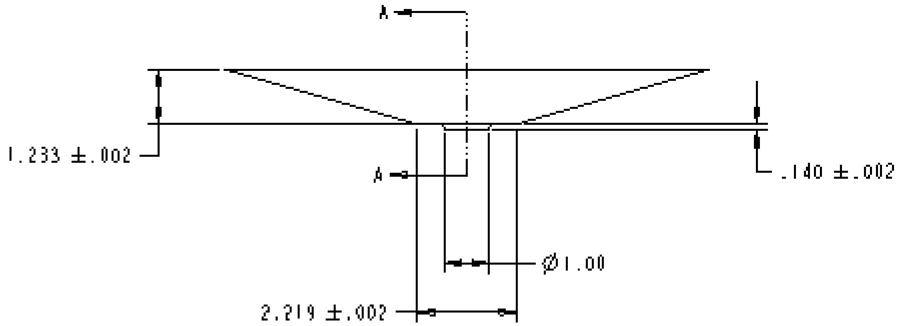
TITLE:
11:1 COMPRESSION PLATE
GMV MICROPILOT IGNITION PROJECT

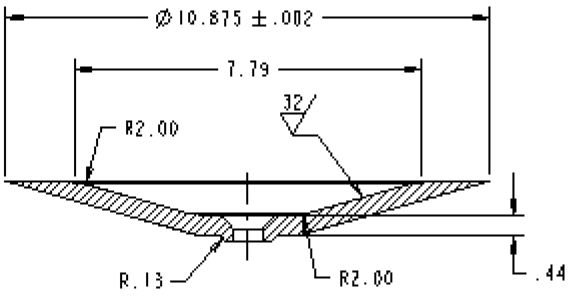
SCALE: 1:2	DWG #: EN2-D05		LAST REVISED: ---
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 05/22/02

CSU ENGINES & ENERGY CONVERSION LAB









SECTION A-A

QUANTITY: 1

MATERIAL: AISI 1018 OR SIMILAR LOW CARBON STEEL

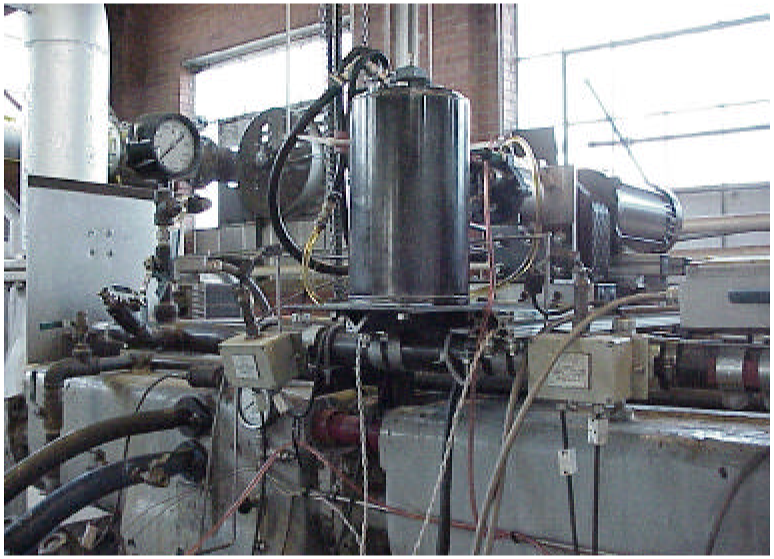
UNLESS OTHERWISE SPECIFIED:
--ALL DIMENSIONS ARE IN INCHES
--SCALE APPLIES TO FULL SIZE PLOTS ONLY
--ALL INTERNAL THREADS CLASS 2A
--ALL EXTERNAL THREADS CLASS 2B
--ALL DIMENSIONS ±.01
--ALL ANGLES ±1°

TITLE: 12:1 COMPRESSION PLATE GMV MICROPILOT IGNITION PROJECT			
SCALE: 1:2	DWG #: EN2-D06		LAST REVISED: ---
DESIGN: WHULL	DETAIL: WHULL	SHEET: 1 OF 1	DRAWN: 05/22/02
CSU ENGINES & ENERGY CONVERSION LAB			

APPENDIX 9

“4-Cylinder Prototype Installation Photographs”

4-CYLINDER PROTOTYPE INSTALLATION PHOTOGRAPHS



Pilot fuel pump and storage mounted on GMV test engine

Micropilot control system for engine operation

Closeup of Delphi pilot fuel injector installed in head

APPENDIX 10

“4-Cylinder Prototype Data and Conclusions”

4-Cylinder Prototype Data and Conclusions

Micro Pilot Ignition on 4 Cylinders of GMV
Colorado State University
12.19.2002

This testing was not an optimization of micro-pilot ignition, but a proof of technology study. However, the four-cylinder prototype data was encouraging for the micro-pilot ignition technology with respect to spark ignition. The benefits could likely be improved but the initial results showed:

- Brake specific fuel consumption of Natural Gas was improved from standard spark ignition across the map, 1% at full load and 5% at 70% load.
- 0% misfires for all points on Pilot injection. Fuel savings were most likely due to this percent misfire improvement.
- THC (Total Hydrocarbon) emissions were improved significantly at light load, 38% at 70% load.
- VOC (Volatile Organic Compounds) emissions were improved above 80% load.
- Coefficient of Variance for the IMEP (Indicated Mean Effective Pressure) was significantly less at lower loads, 76% less at 70%.

Four Delphi injectors (P/N: R01601Z) were installed in the 2-stoke Cooper GMV engine. The engine was also fitted with dual plug heads and compression raising piston pancakes. One plug hole was used for the injector adaptor and the second for a spark plug. The compression raising pancakes were attached to the top surface of each piston raising the compression ratio to approx 12:1.

The engine was started on normally timed spark ignition with the injectors retarded to 22 deg ATDC to eliminate any ignition assistance from the micro-pilot. A standard inlet air condition of 7.5 inch HG was used throughout testing. After the engine achieved running temperature (approximately 145 deg F water temperature) the micro-pilot injection was advanced to 9 BTDC. The spark timing was slowly retarded and the cylinder pressures were monitored to verify ignition from the pilot. When pilot ignition was confirmed, the spark timing was retarded to 15 ATDC. Two load maps were performed using loads of 70, 80, 90, and 100%, one with each ignition system. This created 8 test points, 4 for each map. During the spark map the micro-pilot injection was set to 3 micro liters at 22 deg ATDC.

The micro-pilot injectors had to be fired during the spark ignition to maintain a fuel flow. Fuel flow thru the injector is its major cooling mechanism. The significance of the 22 deg ATDC pilot injection during the spark map is mute. Combustion is still occurring at this time so the diesel will burn, contributing less than ¼% to the overall energy at this quantity (3 uL) and none to the ignition. The significance of retarded spark during the pilot ignition map is even less.

Using an FTIR emissions analyzer along with measurements of cylinder pressures and fuel and air flows, a data set was gathered for all 8 test points. The entire data set is shown in the two worksheets that follow. Two graphs show the greatest improvements of the micro-pilot injection. These were: fuel economy, NOx, THC, COV of IMEP, and VOC emissions as noted above. All the other points of comparison were very similar.

The pilot injection duration for each cylinder was individually tailored to maintain similar peak pressures in all cylinders. It is not surprising that different injection duration were used for each injector, each injector has a certain amount of variability with respect to others in the quantity it will inject for a given duration. Even with different injection durations it is possible that the injectors were all delivering very similar amounts of fuel for each combustion event. The quantity of pilot fuel injected for each combustion event that is shown in the following worksheets is interpolated from past measurements and averaged for all four cylinders.

When calculating the percent energy provided by the pilot injection, properties of the diesel, current natural gas supply, and brake horsepower where considered. These calculations are shown on a following worksheet. Because this testing was used only as a proof of technology, an optimization of pilot quantity was not attempted. A nominal value near 8 uL was used for the pilot at all points except at 70% load. At this light load, quantity had to be increased to maintain consistent combustion with the lean conditions. In the following worksheet, it is shown that:

- 100% load, 8.36 uL, 0.568% pilot energy
- 90% load, 8.36 uL, 0.611% pilot energy
- 80% load, 8.66 uL, 0.679% pilot energy
- 70% load, 12.76 uL, 1.078% pilot energy

At 8.36 uL per combustion event and 300 RPM, about 0.95 gal/day will be consumed per cylinder.

Calculation of MP percent energy per cylinder per cycle

GMV Testing 12.19.2002

Test Point: MP4_1, 100% Load

Inputs:

$BHp := 441hp$

$rev := 1 \quad \mu L := 1 \cdot 10^{-6} L$

$BSFC := 8070 \frac{BTU}{hp \cdot hr}$

$V_{mp} := 8.36 \mu L$

$RPM := 300 \frac{rev}{min}$

Energy of Natural Gas per cylinder per revolution:

$$NG_{energy} := BSFC \cdot BHp \cdot \frac{1}{RPM} \cdot \frac{1}{4}$$

$$NG_{energy} = 5.215 \times 10^{4} \frac{J}{rev}$$

Engine efficiency:

$$NG_{power} := BSFC \cdot BHp$$

$$NG_{power} = 1398.69 \, hp$$

$$\eta := \frac{BHp}{NG_{power}} \qquad \eta = 0.315$$

Energy of mp per cylinder per revolution:

$$\rho_{diesel} := .82 \frac{kg}{L} \qquad LHV_{diesel} := 43.2 \cdot 10^{6} \frac{J}{kg}$$

$$m_{mp} := \rho_{diesel} \cdot V_{mp} \qquad m_{mp} = 6.855 \, mg$$

$$mp_{energy} := m_{mp} \cdot LHV_{diesel}$$

$$mp_{energy} = 296.145 \, J$$

Percent energy from mp per cylinder per cycle:

$$\% := \frac{mp_{energy}}{NG_{energy}} \cdot 100$$

$$\% = 0.568$$

Test Point: MP4_2, 90% Load

Inputs:

$BHp := 398hp$ $\qquad$ $rev := 1 \quad \mu L := 1\cdot 10^{-6} L$

$BSFC := 8304 \frac{BTU}{hp\cdot hr}$ $\qquad$ $V_{mp} := 8.36\mu L$ $\qquad$ $RPM := 300 \frac{rev}{min}$

Energy of Natural Gas per cylinder per revolution:

$$NG_{energy} := BSFC\cdot BHp\cdot \frac{1}{RPM}\cdot \frac{1}{4}$$

$$NG_{energy} = 4.843 \times 10^{4} \frac{J}{rev}$$

Engine efficiency:

$$NG_{power} := BSFC\cdot BHp$$

$$NG_{power} = 1298.91\ hp$$

$$\eta := \frac{BHp}{NG_{power}} \qquad \eta = 0.306$$

Energy of mp per cylinder per revolution:

$$\rho_{diesel} := .82 \frac{kg}{L} \qquad LHV_{diesel} := 43.2\cdot 10^{6} \frac{J}{kg}$$

$$m_{mp} := \rho_{diesel}\cdot V_{mp} \qquad m_{mp} = 6.855\ mg$$

$$mp_{energy} := m_{mp}\cdot LHV_{diesel}$$

$$mp_{energy} = 296.145\ J$$

Percent energy from mp per cylinder per cycle:

$$\% := \frac{mp_{energy}}{NG_{energy}}\cdot 100$$

$$\% = 0.611$$

Test Point: MP4_3, 80% Load

Inputs:

$\mathrm{BHp} := 357\mathrm{hp}$ $\qquad$ $\mathrm{rev} := 1 \quad \mu\mathrm{L} := 1\cdot 10^{-6}\mathrm{L}$

$\mathrm{BSFC} := 8636\,\frac{\mathrm{BTU}}{\mathrm{hp}\cdot\mathrm{hr}}$ $\qquad$ $V_{mp} := 8.66\mu\mathrm{L}$ $\qquad$ $\mathrm{RPM} := 300\,\frac{\mathrm{rev}}{\mathrm{min}}$

Energy of Natural Gas per cylinder per revolution:

$$NG_{energy} := \mathrm{BSFC}\cdot\mathrm{BHp}\cdot\frac{1}{\mathrm{RPM}}\cdot\frac{1}{4}$$

$$NG_{energy} = 4.518\times 10^{4}\,\frac{\mathrm{J}}{\mathrm{rev}}$$

Engine efficiency:

$$NG_{power} := \mathrm{BSFC}\cdot\mathrm{BHp}$$

$$NG_{power} = 1211.69\,\mathrm{hp}$$

$$\eta := \frac{\mathrm{BHp}}{NG_{power}} \qquad \eta = 0.295$$

Energy of mp per cylinder per revolution:

$$\rho_{diesel} := .82\,\frac{\mathrm{kg}}{\mathrm{L}} \qquad LHV_{diesel} := 43.2\cdot 10^{6}\,\frac{\mathrm{J}}{\mathrm{kg}}$$

$$m_{mp} := \rho_{diesel}\cdot V_{mp} \qquad m_{mp} = 7.101\,\mathrm{mg}$$

$$mp_{energy} := m_{mp}\cdot LHV_{diesel}$$

$$mp_{energy} = 306.772\,\mathrm{J}$$

Percent energy from mp per cylinder per cycle:

$$\% := \frac{mp_{energy}}{NG_{energy}}\cdot 100$$

$$\% = 0.679$$

Test Point: MP4_4, 70% Load

Inputs:

$BHp := 315hp$

$rev := 1 \quad \mu L := 1 \cdot 10^{-6} L$

$BSFC := 9084 \frac{BTU}{hp \cdot hr}$ $V_{mp} := 12.76 \mu L$ $RPM := 300 \frac{rev}{min}$

Energy of Natural Gas per cylinder per revolution:

$$NG_{energy} := BSFC \cdot BHp \cdot \frac{1}{RPM} \cdot \frac{1}{4}$$

$$NG_{energy} = 4.193 \times 10^{4} \frac{J}{rev}$$

Engine efficiency:

$$NG_{power} := BSFC \cdot BHp$$

$$NG_{power} = 1124.6\, hp$$

$$\eta := \frac{BHp}{NG_{power}} \qquad \eta = 0.28$$

Energy of mp per cylinder per revolution:

$\rho_{diesel} := .82 \frac{kg}{L}$ $LHV_{diesel} := 43.2 \cdot 10^{6} \frac{J}{kg}$

$m_{mp} := \rho_{diesel} \cdot V_{mp}$ $m_{mp} = 10.463\, mg$

$$mp_{energy} := m_{mp} \cdot LHV_{diesel}$$

$$mp_{energy} = 452.01\, J$$

Percent energy from mp per cylinder per cycle:

$$\% := \frac{mp_{energy}}{NG_{energy}} \cdot 100$$

$$\% = 1.078$$

Colorado State University
12/19/2002
Spark 4 cylinders
Load Map

GMV-4TF				
ENGINE OPERATING PARAMETERS	SP4_1	SP4_2	SP4_3	SP4_4
Percent Load	100.00	90.00	80.00	70.00
Dynamometer Torque [ft-lbs]	7722	6946	6261	5532
Brake Horse Power [BHp]	441	397	357	316
BSFC [BTU/BHp-hr]	8147	8322	8706	9488
Engine Speed [RPM]	300	300	300	300
Timing of Pilot [degree ATDC]	22	22	22	22
Timing of Spark [degre BTDC]	10.10	10.10	10.10	10.10
Average LPP [degree]	17	18	17	14
A/F Stoic Total	16.0	16.0	16.0	16.0
A/F Stoich Combustibles	17.2	17.2	17.2	17.2
A/F Total (Urban & Sharp)	40.0	43.5	46.2	48.3
A/F Comb. (Urban & Sharp)	43.4	47.1	50.0	52.3
A/F (Wet) Carbon Balance	41.3	NaN	47.7	49.4
Pressures				
Air Manifold [in Hg]	7.52	7.51	7.53	7.54
Ambient Pressure [psia]	12.05	12.05	12.05	12.05
Exhaust Manifold [in Hg]	5.01	5.01	4.99	5.00
Fuel Manifold [psig]	23.90	21.80	20.79	20.03
Average Cylinder PP [psig]	528.7	460.7	406.5	386.0
Average Cylinder IMEP [psig]	84.5	77.5	71.3	64.5
Temperatures (deg F)				
Air Manifold [F]	110.1	110.5	109.8	110.1
Fuel Manifold [F]	121.4	121.3	121.3	121.2
Average Cylinder Exhaust [F]	719.2	699.4	683.9	667.4
Exhaust Stack [F]	594	578	566	544
Jacket Water Inlet [F]	141	143	142	143
Jacket Water Outlet [F]	151	151	150	151
Lube Oil Inlet [F]	142	143	144	143
Lube Oil Outlet [F]	155	154	154	153
Fuel Flow Measurements				
Pilot Quantity [uL/combustion event]	3	3	3	3
Static Fuel [psig]	43.6	44.0	44.2	44.4
Fuel Differential [in H20]	66.2	55.6	49.5	45.9
Orifice Temperature [F]	81.3	79.9	80.0	79.9
Fuel Flow [SCFH]	3927.6	3615.0	3400.6	3279.2
Higher Heating Value-Dry [Btu]	1014.7	1014.7	1014.7	1014.7
Lower Heating Value-Dry [Btu]	914.3	914.3	914.3	914.3
Fuel Tube I.D. [in]	3.0680	3.0680	3.0680	3.0680
Fuel Orifice O.D. [in]	0.5950	0.5950	0.5950	0.5950
Annubar Flow Rates				
Inlet Air Flow [SCFH]	1289.43	1296.51	1305.58	1303.70
Exhaust Flow [SCFH]	0.0	0	0	0
Ambient Conditions				
Air Manifold Relative Humidity [%]	3.56	3.41	3.40	3.33
Dry Bulb Temperature [F]	31.49	32.26	32.75	33.75
Relative Humidity [%]	19.17	17.83	17.21	16.52
Absolute Humidity [lb/lb]	0.0018	0.0018	0.0017	0.0017
Absolute Humidity [gr/lb]	0.88	0.86	0.83	0.82

GMV-4TF				
COMBUSTION ANALYSIS	SP4_1	SP4_2	SP4_3	SP4_4
Ignition Type	**MS**	**MS**	**MS**	**MS**
AVG./STD. Peak Pressure (psia)				
Cylinder 1	524.55 / 59.90	442.43 / 62.80	394.65 / 51.51	363.64 / 37.49
Cylinder 2	506.30 / 39.54	449.28 / 42.55	405.69 / 47.34	390.84 / 53.59
Cylinder 3	568.02 / 49.47	497.26 / 50.99	419.69 / 53.85	401.11 / 54.20
Cylinder 4	515.74 / 45.92	453.65 / 46.46	406.10 / 49.41	388.32 / 52.17
Engine Average	#DIV/0!	#DIV/0!	#DIV/0!	#DIV/0!

AVG./STD. Location Peak Pressure (Deg.ATDC)	AVG.	STD.	AVG.	STD.	AVG.	STD.	AVG.	STD.
Cylinder 1	17.75	3.34	16.78	7.23	12.42	9.76	4.82	9.16
Cylinder 2	17.73	2.13	19.56	2.50	20.72	3.63	20.52	4.98
Cylinder 3	16.00	2.14	17.26	2.71	16.35	6.39	14.16	8.13
Cylinder 4	17.73	2.12	18.67	3.05	19.16	4.65	18.05	6.35
Engine Average	17.30	2.43	18.07	3.87	17.16	6.11	14.39	7.16
AVG./STD. Cylinder IMEP								
Cylinder 1	85.69 / 2.17		77.25 / 7.15		70.11 / 10.30		50.04 / 26.57	
Cylinder 2	82.86 / 1.54		77.14 / 1.70		72.59 / 2.27		70.78 / 3.57	
Cylinder 3	87.89 / 1.24		80.75 / 1.57		73.16 / 4.24		69.81 / 8.59	
Cylinder 4	81.49 / 1.51		74.86 / 1.83		69.49 / 4.10		67.19 / 4.59	
Engine Average	#DIV/0!		#DIV/0!		#DIV/0!		#DIV/0!	
COV. Cylinder IMEP								
Cylinder 1	2.5		9.3		14.7		53.1	
Cylinder 2	1.9		2.2		3.1		5.1	
Cylinder 3	1.4		1.9		5.8		12.3	
Cylinder 4	1.9		2.5		5.9		6.8	
Engine Average	1.92		3.96		7.38		19.32	
AVG./STD. Burn Duration 0-10% (Degrees)								
Cylinder 1	15.18 / 2.84		17.61 / 4.40		19.04 / 6.26		16.41 / 11.31	
Cylinder 2	15.22 / 1.81		16.87 / 2.16		18.30 / 2.68		18.94 / 3.08	
Cylinder 3	12.80 / 1.75		14.69 / 2.25		17.43 / 3.75		17.92 / 4.75	
Cylinder 4	14.67 / 1.88		15.91 / 2.36		17.34 / 2.97		17.93 / 3.60	
Engine Average	#DIV/0!		#DIV/0!		#DIV/0!		#DIV/0!	
AVG./STD. Burn Duration 10-90% (Degrees)								
	SP4_1		SP4_2		SP4_3		SP4_4	
	AVG.	STD.	AVG.	STD.	AVG.	STD.	AVG.	STD.
Cylinder 1	14.80	5.80	18.88	8.08	21.81	10.69	18.85	19.35
Cylinder 2	14.15	4.87	16.79	3.98	19.22	4.25	20.35	5.33
Cylinder 3	11.70	4.17	14.58	4.36	19.53	7.24	20.58	8.14
Cylinder 4	13.76	4.55	16.23	4.20	18.84	5.36	20.00	5.8
Engine Average	13.60	4.85	16.62	5.16	19.85	6.89	19.95	9.66

AVG./STD. Burn Duration CA@50 (deg ATDC)				
Cylinder 1	24.38 / 4.09	30.48 / 6.63	35.44 / 9.77	32.83 / 18.92
Cylinder 2	25.51 / 2.60	29.12 / 3.10	31.59 / 3.92	32.61 / 4.71
Cylinder 3	20.86 / 2.77	26.16 / 3.41	34.42 / 5.90	33.97 / 7.59
Cylinder 4	22.14 / 2.81	27.43 / 3.52	30.64 / 4.49	32.14 / 5.46
Engine Average	#DIV/0!	#DIV/0!	#DIV/0!	#DIV/0!
Percent Misfires				
Cylinder 1	0.0	0.5	0.9	15.9
Cylinder 2	0.0	0.0	0.0	0.0
Cylinder 3	0.0	0.0	0.1	0.8
Cylinder 4	0.0	0.0	0.2	0.1
Engine Average Percent	0.00	0.13	0.30	4.20
Cylinder Exhaust Temperatures (Degrees oF)				
Cylinder 1	671.3	649.4	637.3	582.7
Cylinder 2	791.4	768.7	753.1	744.0
Cylinder 3	664.6	647.8	629.7	631.4
Cylinder 4	749.5	732.0	715.4	711.9
Engine Average	719.21	699.46	683.88	667.50

note: Data taken over 1000 engine cycles

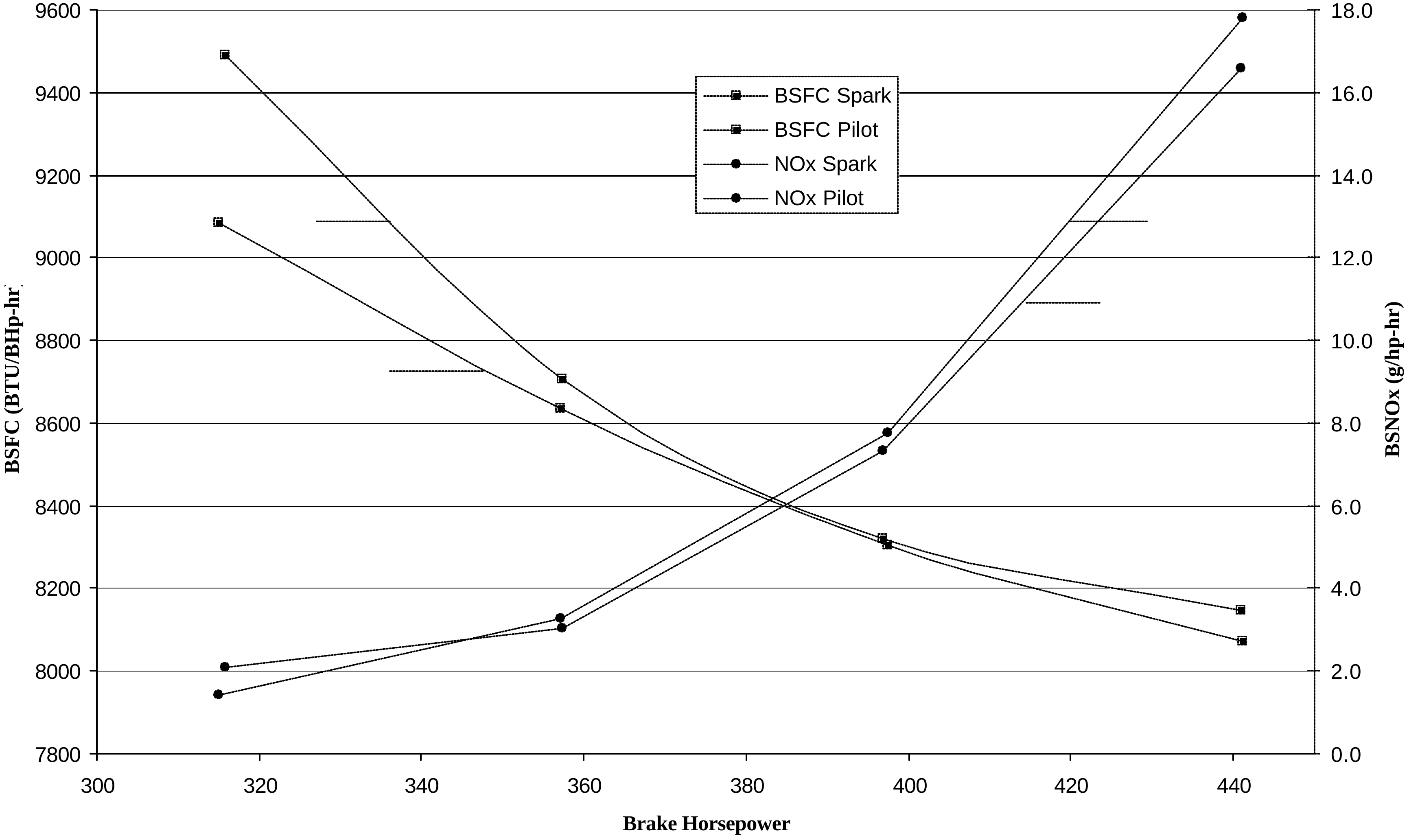
GMV-4TF				
MEASURED EMISSIONS	SP4_1	SP4_2	SP4_3	SP4_4
Ignition Type	**MS**	**MS**	**MS**	**MS**
Emissions Measured (Dry)				
NOx (ppm)	1403.16	555.41	206.57	124.48
CO (ppm)	87.78	111.44	137.54	163.00
THC (ppm)	945.96	1043.80	1258.71	2177.78
O2 (%)	13.00	13.64	14.10	14.52
CO2 (%)	4.21	3.83	3.57	3.31

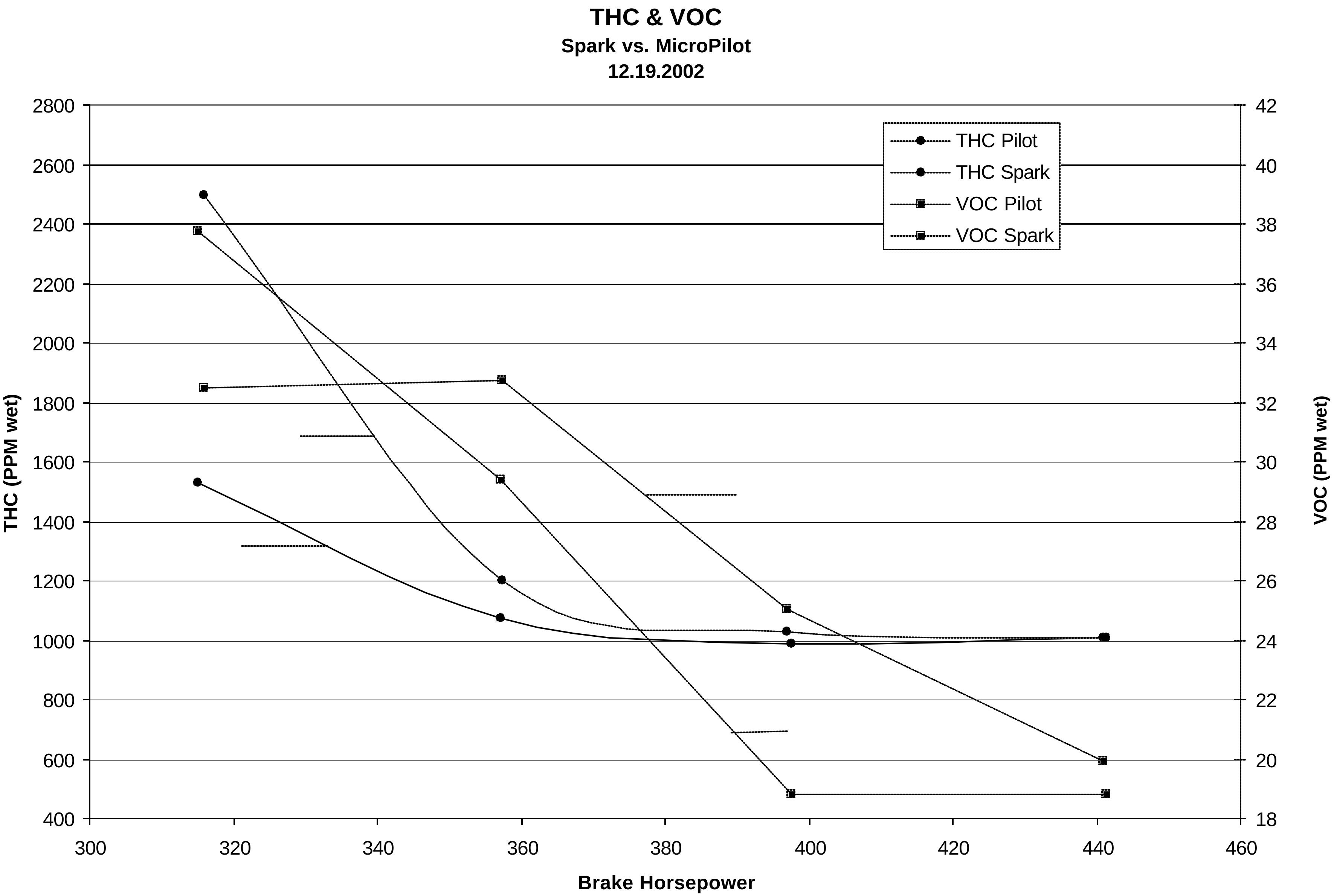
Calculated Carbon Balance Emissions				
NOX (g/hp-hr)	16.57	7.31	3.02	2.08
THC (g/hp-hr)	4.0	5.0	6.7	13.2
CO (g/hp-hr)	0.63	0.89	1.23	1.67
CH2O (g/hp-hr)	0.16	0.21	0.27	0.32
NOx Corr. to 15% O2 (ppm)	1048.72	451.07	178.42	114.84
FTIR Water (ppm, wet)	90136.8	84608.7	80113.3	76237.6
Air Flow (lbs/hr)	7402	7412	7403	7443
Trapped A/F	20.2	NaN	23.4	24.2
A/F Wet Carbon Balance	41.3	NaN	47.7	49.4
FTIR Measured Emissions (PPM, Wet)				
Carbon Monoxide low	80.73	105.16	128.27	150.05
(+-)Carbon Monoxide low	2.75	3.36	4.13	4.43
Carbon monoxide high	53.68	81.62	105.62	128.50
(+-)Carbon monoxide high	70.21	70.80	65.50	66.31
Carbon dioxide	40190.03	37758.36	35713.47	33778.99
(+-)Carbon dioxide	983.89	748.93	720.13	711.72
Nitric oxide	1349.63	512.74	173.30	127.98
(+-)Nitric oxide	35.60	27.62	15.63	12.93
Nitrogren dioxide	84.94	49.54	46.31	43.27
(+-)Nitrogren dioxide	6.38	5.13	4.81	4.80
Nitrous oxide	0.08	-0.02	-0.09	-0.08
(+-)Nitrous oxide	0.56	0.55	0.58	0.59
Methane	955.04	966.78	1100.34	2345.28
(+-)Methane	31.86	29.86	38.24	107.79
Ethyne	0.06	0.07	0.14	0.21
(+-)Ethyne	0.93	0.88	0.80	0.79
Ethene	6.41	7.79	10.25	11.36
(+-)Ethene	0.25	0.25	0.25	0.25
Ethane	17.74	24.24	38.12	77.20
(+-)Ethane	3.59	3.47	4.18	10.21
Propene	1.72	2.00	2.17	2.04
(+-)Propene	1.83	1.73	1.64	1.59
Formaldehyde	18.97	22.12	25.55	26.85
(+-)Formaldehyde	0.20	0.21	0.23	0.39
Water	90136.77	84608.69	80113.34	76237.60
(+-)Water	2744.07	2540.80	2383.07	2360.77
Propane	2.90	3.82	4.89	3.91
(+-)Propane	3.59	3.46	4.16	10.19
Ammonia	0.00	0.00	0.00	0.00
(+-)Ammonia	0.04	0.04	0.04	0.04
Acrolein	-0.13	0.04	0.36	0.39
(+-)Acrolein	0.52	0.49	0.56	1.15
Acetaldehyde	1.09	1.26	1.27	0.11
(+-)Acetaldehyde	0.51	0.52	0.59	0.97
Isobutylene	-0.20	-0.13	-0.13	-0.14
(+-)Isobutylene	0.67	0.63	0.60	0.58
1-3 Butadiene	-0.26	-0.30	-0.42	-0.37
(+-)1-3 Butadiene	1.03	0.97	0.92	0.89
SF6	-0.01	-0.01	-0.01	-0.01
(+-)SF6	0.01	0.01	0.01	0.01
Methanol	-0.83	-0.71	-0.46	-0.15
(+-)Methanol	1.24	1.23	1.21	1.20
NOx	1434.27	562.29	218.28	169.59
(+-)NOx	41.99	32.76	20.38	17.61
Total Hydrocarbons	1006.93	1027.59	1201.80	2494.46
(+-)Total Hydrocarbons	51.71	48.63	60.26	154.86
Non Methane Hydrocarbons	52.40	68.67	102.32	172.36
(+-)Non Methane Hydrocarbons	19.83	19.01	22.06	48.28
VOC	19.94	25.05	32.72	32.49
(+-)VOC	13.25	12.67	14.42	30.18

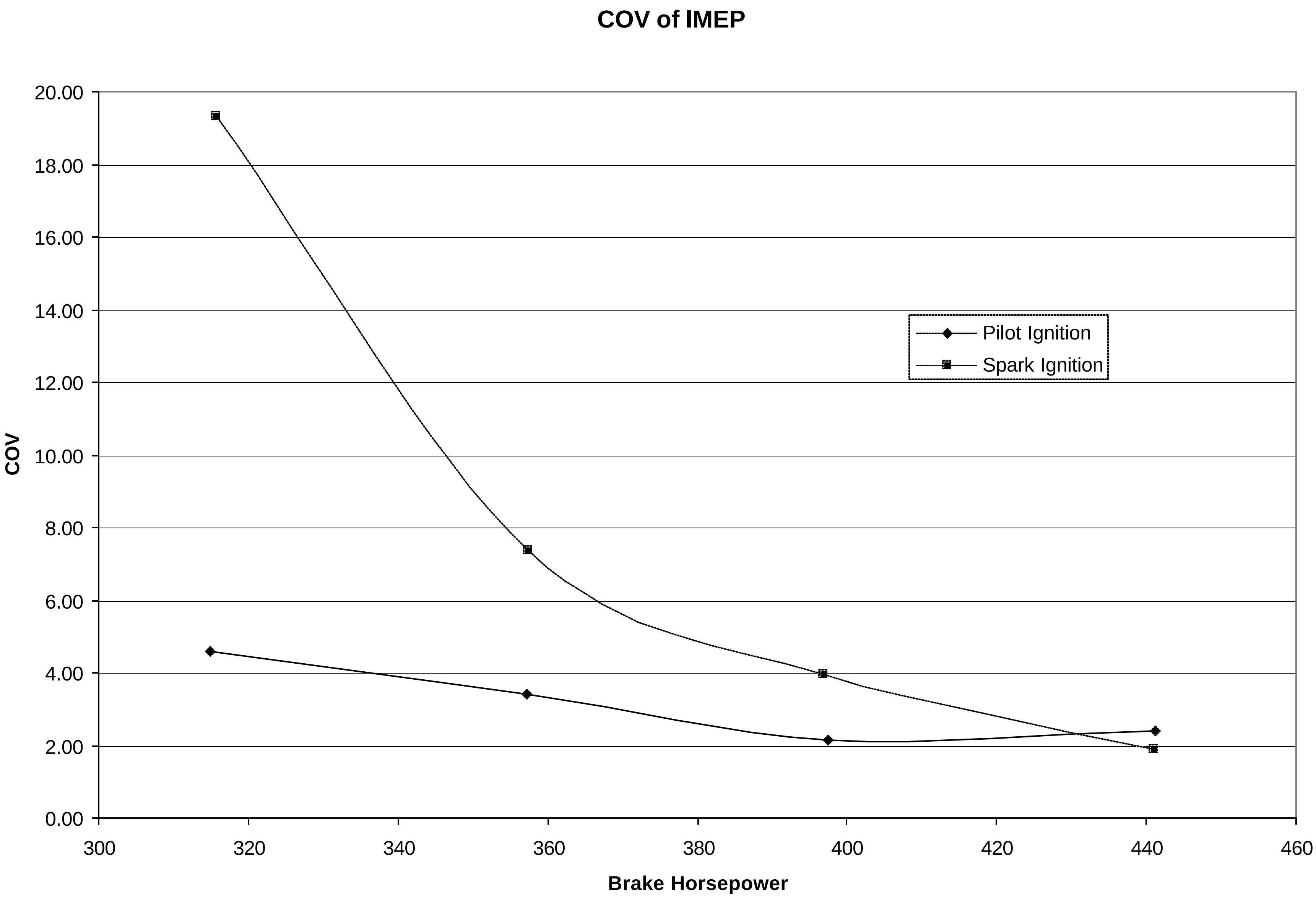
BSFC & BSNOx

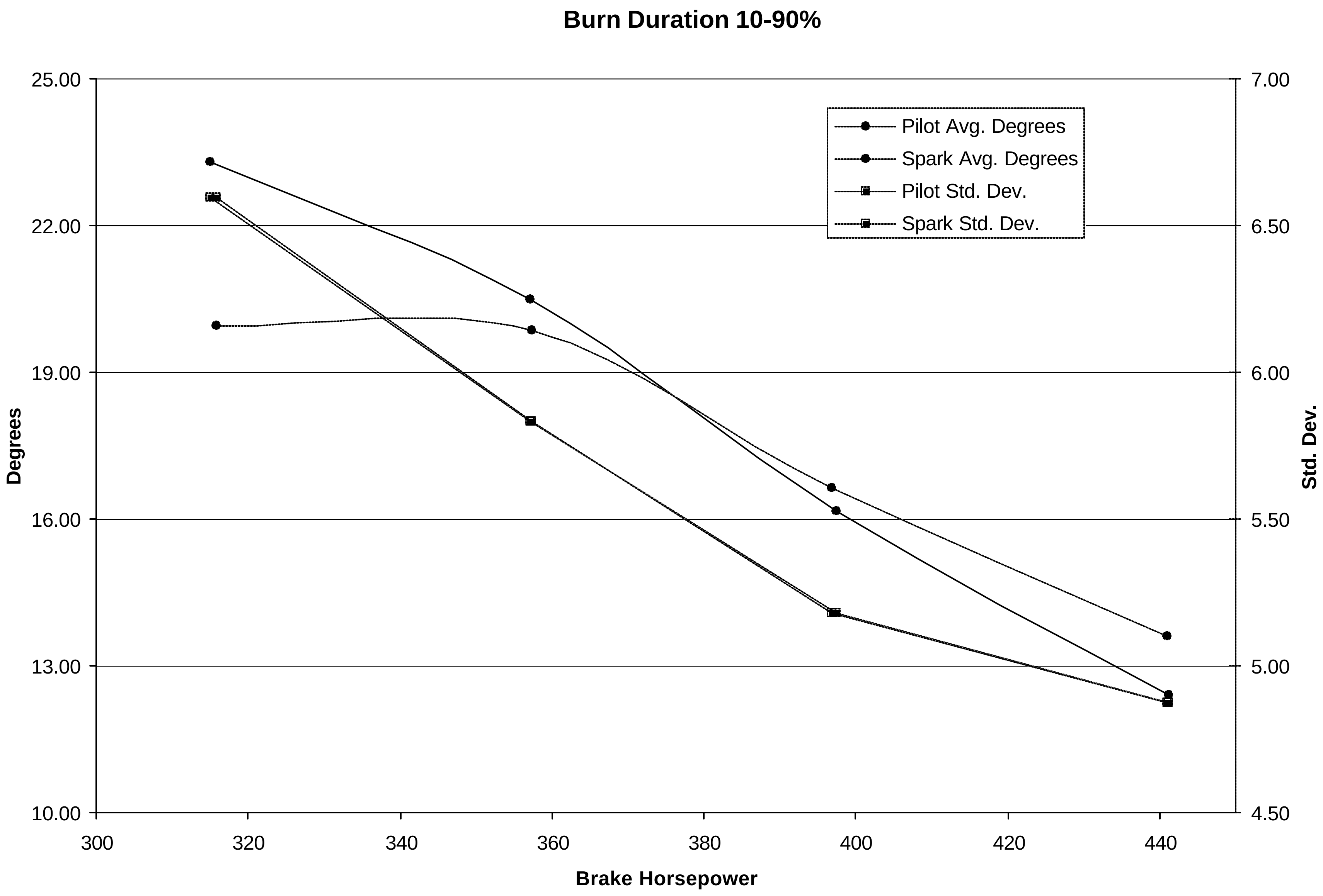
Spark vs. MicroPilot

12.19.2002




THC & VOC
Spark vs. MicroPilot
12.19.2002
THC (PPM wet)
VOC (PPM wet)
Brake Horsepower
2800
2600
2400
2200
2000
1800
1600
1400
1200
1000
800
600
400
42
40
38
36
34
32
30
28
26
24
22
20
18
300
320
340
360
380
400
420
440
460
THC Pilot
THC Spark
VOC Pilot
VOC Spark


COV of IMEP
COV
Brake Horsepower
0.00
2.00
4.00
6.00
8.00
10.00
12.00
14.00
16.00
18.00
20.00
300
320
340
360
380
400
420
440
460
Pilot Ignition
Spark Ignition


Burn Duration 10-90%
Pilot Avg. Degrees
Spark Avg. Degrees
Pilot Std. Dev.
Spark Std. Dev.
Degrees
25.00
22.00
19.00
16.00
13.00
10.00
Std. Dev.
7.00
6.50
6.00
5.50
5.00
4.50
300
320
340
360
380
400
420
440
Brake Horsepower

Colorado State University
12/19/2002
MicroPilot 4 cylinders
Load Map

GMV-4TF				
ENGINE OPERATING PARAMETERS	MP4_1	MP4_2	MP4_3	MP4_4
Percent Load	100.00	90.00	80.00	70.00
Dynamometer Torque [ft-lbs]	7726	6958	6254	5517
Brake Horse Power [BHp]	441	398	357	315
BSFC [BTU/BHp-hr]	8070	8304	8636	9084
Engine Speed [RPM]	300	300	300	300
Timing of Pilot [degree BTDC]	8.88	8.88	9.13	9.13
Timing of Spark [degre ATDC]	15.00	15.00	15.00	15.00
Average LPP [degree]	17	18	19	15
A/F Stoic Total	16.0	16.0	16.0	16.0
A/F Stoich Combustibles	17.2	17.2	17.2	17.2
A/F Total (Urban & Sharp)	39.8	43.2	46.5	50.1
A/F Comb. (Urban & Sharp)	43.1	46.8	50.3	54.3
A/F (Wet) Carbon Balance	40.9	44.6	47.9	51.6
Pressures				
Air Manifold [in Hg]	7.52	7.51	7.51	7.53
Ambient Pressure [psia]	12.05	12.05	12.05	12.05
Exhaust Manifold [in Hg]	4.99	4.98	4.98	5.03
Fuel Manifold [psig]	24.60	22.23	20.75	19.21
Average Cylinder PP [psig]	551.7	462.2	397.4	364.5
Average Cylinder IMEP [psig]	84.8	77.7	71.4	64.4
Temperatures (deg F)				
Air Manifold [F]	109.5	110.4	109.1	108.7
Fuel Manifold [F]	120.3	120.4	120.6	120.8
Average Cylinder Exhaust [F]	717.7	697.8	683.6	655.7
Exhaust Stack [F]	592.1	576.6	562.4	540.7
Jacket Water Inlet [F]	140.4	140.9	141.3	142.4
Jacket Water Outlet [F]	150.1	149.6	149.6	149.8
Lube Oil Inlet [F]	142.0	140.8	143.8	142.6
Lube Oil Outlet [F]	155.0	152.7	153.5	152.5
Fuel Flow Measurements				
Pilot Quantity [uL/combustion event]	8.36	8.36	8.66	12.76
Static Fuel [psig]	43.7	44.0	44.3	44.5
Fuel Differential [in H20]	65.2	55.6	48.3	41.8
Orifice Temperature [F]	79.6	80.2	80.6	79.9
Fuel Flow [SCFH]	3894.4	3610.2	3369.9	3132.0
Higher Heating Value-Dry [Btu]	1014.7	1014.7	1014.7	1014.7
Lower Heating Value-Dry [Btu]	914.3	914.3	914.3	914.3
Fuel Tube I.D. [in]	3.0680	3.0680	3.0680	3.0680
Fuel Orifice O.D. [in]	0.5950	0.5950	0.5950	0.5950
Annubar Flow Rates				
Inlet Air Flow [SCFH]	1286.04	1293.58	1297.98	1312.94
Exhaust Flow [SCFH]	4.5	0	0	4.456231
Ambient Conditions				
Air Manifold Relative Humidity [%]	4.10	3.58	3.58	3.21
Dry Bulb Temperature [F]	34.29	36.09	35.02	34.49
Relative Humidity [%]	17.29	15.60	16.49	15.29
Absolute Humidity [lb/lb]	0.0020	0.0018	0.0018	0.0016
Absolute Humidity [gr/lb]	1.00	0.89	0.86	0.76

GMV-4TF				
COMBUSTION ANALYSIS	**MP4_1**	**MP4_2**	**MP4_3**	**MP4_4**
Ignition Type	**MS**	**MS**	**MS**	**MS**
AVG./STD. Peak Pressure (psia)				
Cylinder 1	566.45 / 30.00	478.47 / 30.34	399.08 / 28.12	362.60 / 16.27
Cylinder 2	575.75 / 36.27	444.26 / 36.78	418.79 / 41.37	352.27 / 36.52
Cylinder 3	525.31 / 32.35	464.10 / 32.14	394.29 / 29.27	359.14 / 19.91
Cylinder 4	539.44 / 89.46	461.85 / 27.83	377.53 / 28.06	384.15 / 26.36
Engine Average	#DIV/0!	#DIV/0!	#DIV/0!	#DIV/0!

AVG./STD. Location Peak Pressure (Deg.ATDC)				
Cylinder 1	16.50 / 1.28	17.89 / 1.63	16.86 / 4.12	10.30 / 5.48
Cylinder 2	16.78 / 1.35	19.19 / 2.70	20.21 / 3.50	20.97 / 5.34
Cylinder 3	17.61 / 1.58	18.30 / 1.61	17.99 / 4.23	12.06 / 7.14
Cylinder 4	15.82 / 2.74	17.30 / 1.42	19.12 / 2.63	17.83 / 2.32
Engine Average	#DIV/0!	#DIV/0!	#DIV/0!	#DIV/0!
AVG./STD. Cylinder IMEP				
Cylinder 1	88.61 / 1.51	79.87 / 1.19	71.49 / 2.18	61.11 / 4.66
Cylinder 2	86.18 / 1.78	76.87 / 2.92	75.05 / 4.22	67.18 / 2.70
Cylinder 3	87.94 / 1.62	80.57 / 1.26	72.78 / 1.71	64.82 / 2.62
Cylinder 4	76.49 / 3.00	73.64 / 1.33	66.31 / 1.72	64.52 / 1.76
Engine Average	#DIV/0!	#DIV/0!	#DIV/0!	#DIV/0!
COV. Cylinder IMEP				
Cylinder 1	1.7	1.5	3.1	7.6
Cylinder 2	2.1	3.8	5.6	4.0
Cylinder 3	1.9	1.6	2.4	4.0
Cylinder 4	3.9	1.8	2.6	2.7
Engine Average	2.39	2.16	3.40	4.60
AVG./STD. Burn Duration 0-10% (Degrees)				
Cylinder 1	12.71 / 0.55	14.52 / 0.78	17.48 / 1.26	18.39 / 1.59
Cylinder 2	12.02 / 0.75	16.22 / 2.12	17.32 / 2.72	19.71 / 2.70
Cylinder 3	14.83 / 0.84	15.95 / 1.02	18.36 / 1.30	19.45 / 1.76
Cylinder 4	12.30 / 1.19	13.81 / 0.77	17.29 / 1.27	15.40 / 1.02
Engine Average	#DIV/0!	#DIV/0!	#DIV/0!	#DIV/0!

AVG./STD. Burn Duration 10-90% (Degrees)								
	AVG.	**STD.**	**AVG.**	**STD.**	**AVG.**	**STD.**	**AVG.**	**STD.**
Cylinder 1	12.31	4.06	15.97	6.28	21.45	7.63	25.88	8.55
Cylinder 2	10.69	3.70	16.67	4.30	18.18	4.45	22.73	5.17
Cylinder 3	14.30	4.62	17.00	5.40	21.75	5.80	25.48	6.75
Cylinder 4	12.25	7.12	15.02	4.73	20.50	5.45	19.12	5.91
Engine Average	12.39	4.88	16.17	5.18	20.47	5.83	23.30	6.60

AVG./STD. Burn Duration CA@50 (deg ATDC)				
Cylinder 1	22.04 / 1.48	29.47 / 2.12	39.11 / 3.37	45.03 / 5.11
Cylinder 2	18.36 / 1.41	29.81 / 3.08	29.94 / 3.90	35.91 / 4.10
Cylinder 3	23.89 / 1.91	30.61 / 2.35	38.29 / 3.13	45.25 / 4.29
Cylinder 4	22.52 / 2.68	30.29 / 1.97	35.01 / 2.74	34.27 / 2.66
Engine Average	#DIV/0!	#DIV/0!	#DIV/0!	#DIV/0!
Percent Misfires				
Cylinder 1	0.0	0.0	0.0	0.0
Cylinder 2	0.0	0.0	0.0	0.0
Cylinder 3	0.0	0.0	0.0	0.0
Cylinder 4	0.0	0.0	0.0	0.0
Engine Average Percent	0.00	0.00	0.00	0.00
Cylinder Exhaust Temperatures (Degrees oF)				
Cylinder 1	674.9	641.5	622.0	581.1
Cylinder 2	795.4	773.0	769.3	734.2
Cylinder 3	684.2	660.5	643.0	623.2
Cylinder 4	716.2	716.1	700.1	684.8
Engine Average	717.64	697.77	683.61	655.81

note: Data taken over 1000 engine cycles

GMV-4TF				
MEASURED EMISSIONS	**MP4_1**	**MP4_2**	**MP4_3**	**MP4_4**
Ignition Type	**MS**	**MS**	**MS**	**MS**
Emissions Measured (Dry)				
NOx (ppm)	1535.83	594.92	223.14	84.78
CO (ppm)	85.34	102.80	134.58	209.70
THC (ppm)	948.32	934.99	1019.05	1518.64
O2 (%)	12.99	13.61	14.11	14.66
CO2 (%)	4.25	3.88	3.57	3.24

Calculated Carbon Balance Emissions				
NOX (g/hp-hr)	17.79	7.75	3.27	1.42
THC (g/hp-hr)	4.0	4.4	5.4	9.2
CO (g/hp-hr)	0.60	0.82	1.20	2.13
CH2O (g/hp-hr)	0.15	0.18	0.25	0.35
NOx Corr. to 15% O2 (ppm)	1146.15	481.43	194.08	80.23
FTIR Water (ppm, wet)	91628.3	85486.4	79861.3	74686.8
Air Flow (lbs/hr)	7276	7340	7378	7389
Trapped A/F	20.2	22.0	23.6	25.4
A/F Wet Carbon Balance	40.9	44.6	47.9	51.6
FTIR Measured Emissions (PPM, Wet)				
Carbon Monoxide low	77.21	94.49	124.51	192.57
(+-)Carbon Monoxide low	2.59	3.13	3.67	5.44
Carbon monoxide high	61.83	77.18	104.62	167.23
(+-)Carbon monoxide high	73.72	68.52	69.95	70.10
Carbon dioxide	40521.40	37773.17	35675.19	33281.84
(+-)Carbon dioxide	1029.15	756.88	821.74	696.36
Nitric oxide	1480.78	549.21	195.61	53.83
(+-)Nitric oxide	38.15	27.98	16.64	8.72
Nitrogren dioxide	87.21	48.00	42.33	43.33
(+-)Nitrogren dioxide	6.50	4.95	4.72	4.59
Nitrous oxide	0.08	-0.01	-0.07	-0.08
(+-)Nitrous oxide	0.61	0.56	0.58	0.58
Methane	950.62	931.55	983.10	1406.30
(+-)Methane	30.76	29.27	31.59	49.97
Ethyne	0.41	0.33	0.26	0.24
(+-)Ethyne	0.92	0.88	0.83	0.78
Ethene	6.06	6.30	9.60	12.72
(+-)Ethene	0.25	0.24	0.25	0.26
Ethane	19.92	19.44	32.34	48.19
(+-)Ethane	3.26	3.54	3.47	5.13
Propene	1.64	1.66	1.94	2.06
(+-)Propene	1.87	1.73	1.64	1.55
Formaldehyde	17.87	19.78	24.27	29.57
(+-)Formaldehyde	0.20	0.20	0.22	0.28
Water	91628.26	85486.39	79861.34	74686.78
(+-)Water	2811.96	2536.18	2445.82	2232.06
Propane	2.35	2.28	3.98	5.09
(+-)Propane	3.26	3.53	3.47	5.12
Ammonia	0.08	0.03	0.01	0.00
(+-)Ammonia	0.04	0.04	0.04	0.04
Acrolein	-0.51	-0.31	-0.06	0.27
(+-)Acrolein	0.50	0.50	0.52	0.65
Acetaldehyde	1.04	1.10	1.43	1.08
(+-)Acetaldehyde	0.50	0.51	0.55	0.69
Isobutylene	-0.19	-0.15	-0.23	-0.16
(+-)Isobutylene	0.68	0.63	0.60	0.57
1-3 Butadiene	-0.30	-0.36	-0.35	-0.50
(+-)1-3 Butadiene	1.05	0.97	0.92	0.87
SF6	-0.01	-0.01	-0.01	-0.01
(+-)SF6	0.01	0.01	0.01	0.01
Methanol	-1.08	-0.76	-0.57	-0.07
(+-)Methanol	1.28	1.23	1.21	1.23
NOx	1565.72	596.18	237.41	97.46
(+-)NOx	44.66	32.88	21.29	13.30
Total Hydrocarbons	1006.38	986.11	1071.77	1528.73
(+-)Total Hydrocarbons	49.18	48.75	50.80	76.43
Non Methane Hydrocarbons	55.35	54.44	88.58	125.70
(+-)Non Methane Hydrocarbons	18.34	19.40	19.04	26.45
VOC	18.83	18.82	29.38	37.73
(+-)VOC	12.33	12.90	12.64	17.04

Colorado State University
12/19/2002
Spark 4 cylinders
Load Map

GMV-4TF				
ENGINE OPERATING PARAMETERS	SP4_1	SP4_2	SP4_3	SP4_4
Percent Load	100.00	90.00	80.00	70.00
Dynamometer Torque [ft-lbs]	7722	6946	6261	5532
Brake Horse Power [BHp]	441	397	357	316
BSFC [BTU/BHp-hr]	8147	8322	8706	9488
Engine Speed [RPM]	300	300	300	300
Timing of Pilot [degree ATDC]	22	22	22	22
Timing of Spark [degre BTDC]	10.10	10.10	10.10	10.10
Average LPP [degree]	17	18	17	14
A/F Stoic Total	16.0	16.0	16.0	16.0
A/F Stoich Combustibles	17.2	17.2	17.2	17.2
A/F Total (Urban & Sharp)	40.0	43.5	46.2	48.3
A/F Comb. (Urban & Sharp)	43.4	47.1	50.0	52.3
A/F (Wet) Carbon Balance	41.3	NaN	47.7	49.4
Pressures				
Air Manifold [in Hg]	7.52	7.51	7.53	7.54
Ambient Pressure [psia]	12.05	12.05	12.05	12.05
Exhaust Manifold [in Hg]	5.01	5.01	4.99	5.00
Fuel Manifold [psig]	23.90	21.80	20.79	20.03
Average Cylinder PP [psig]	528.7	460.7	406.5	386.0
Average Cylinder IMEP [psig]	84.5	77.5	71.3	64.5
Temperatures (deg F)				
Air Manifold [F]	110.1	110.5	109.8	110.1
Fuel Manifold [F]	121.4	121.3	121.3	121.2
Average Cylinder Exhaust [F]	719.2	699.4	683.9	667.4
Exhaust Stack [F]	594	578	566	544
Jacket Water Inlet [F]	141	143	142	143
Jacket Water Outlet [F]	151	151	150	151
Lube Oil Inlet [F]	142	143	144	143
Lube Oil Outlet [F]	155	154	154	153
Fuel Flow Measurements				
Pilot Quantity [uL/combustion event]	3	3	3	3
Static Fuel [psig]	43.6	44.0	44.2	44.4
Fuel Differential [in H20]	66.2	55.6	49.5	45.9
Orifice Temperature [F]	81.3	79.9	80.0	79.9
Fuel Flow [SCFH]	3927.6	3615.0	3400.6	3279.2
Higher Heating Value-Dry [Btu]	1014.7	1014.7	1014.7	1014.7
Lower Heating Value-Dry [Btu]	914.3	914.3	914.3	914.3
Fuel Tube I.D. [in]	3.0680	3.0680	3.0680	3.0680
Fuel Orifice O.D. [in]	0.5950	0.5950	0.5950	0.5950
Annubar Flow Rates				
Inlet Air Flow [SCFH]	1289.43	1296.51	1305.58	1303.70
Exhaust Flow [SCFH]	0.0	0	0	0
Ambient Conditions				
Air Manifold Relative Humidity [%]	3.56	3.41	3.40	3.33
Dry Bulb Temperature [F]	31.49	32.26	32.75	33.75
Relative Humidity [%]	19.17	17.83	17.21	16.52
Absolute Humidity [lb/lb]	0.0018	0.0018	0.0017	0.0017
Absolute Humidity [gr/lb]	0.88	0.86	0.83	0.82

GMV-4TF				
COMBUSTION ANALYSIS	SP4_1	SP4_2	SP4_3	SP4_4
Ignition Type	**MS**	**MS**	**MS**	**MS**
AVG./STD. Peak Pressure (psia)				
Cylinder 1	524.55 / 59.90	442.43 / 62.80	394.65 / 51.51	363.64 / 37.49
Cylinder 2	506.30 / 39.54	449.28 / 42.55	405.69 / 47.34	390.84 / 53.59
Cylinder 3	568.02 / 49.47	497.26 / 50.99	419.69 / 53.85	401.11 / 54.20
Cylinder 4	515.74 / 45.92	453.65 / 46.46	406.10 / 49.41	388.32 / 52.17
Engine Average	#DIV/0!	#DIV/0!	#DIV/0!	#DIV/0!

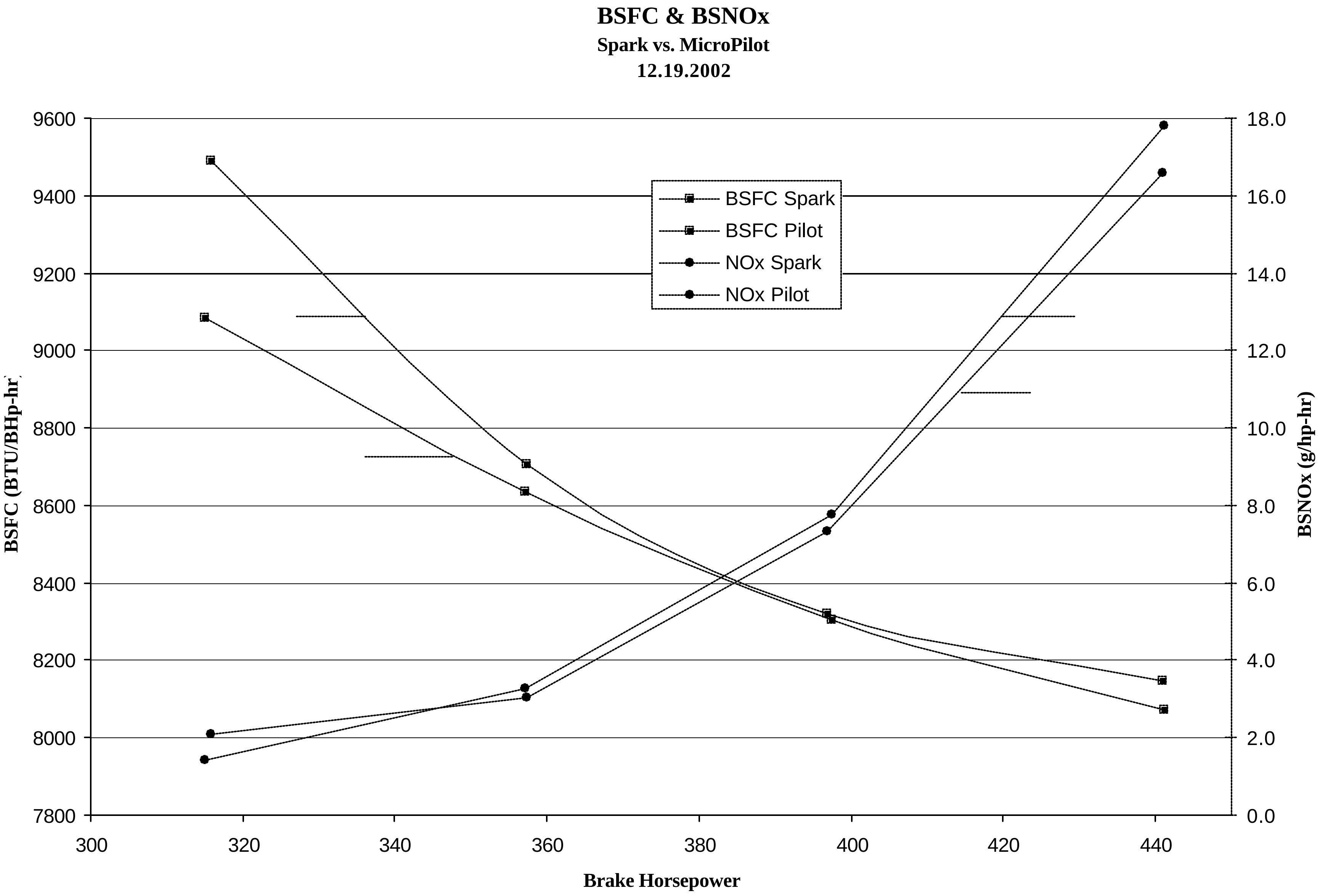
AVG./STD. Location Peak Pressure (Deg.ATDC)	AVG.	STD.	AVG.	STD.	AVG.	STD.	AVG.	STD.
Cylinder 1	17.75	3.34	16.78	7.23	12.42	9.76	4.82	9.16
Cylinder 2	17.73	2.13	19.56	2.50	20.72	3.63	20.52	4.98
Cylinder 3	16.00	2.14	17.26	2.71	16.35	6.39	14.16	8.13
Cylinder 4	17.73	2.12	18.67	3.05	19.16	4.65	18.05	6.35
Engine Average	17.30	2.43	18.07	3.87	17.16	6.11	14.39	7.16
AVG./STD. Cylinder IMEP								
Cylinder 1	85.69 / 2.17		77.25 / 7.15		70.11 / 10.30		50.04 / 26.57	
Cylinder 2	82.86 / 1.54		77.14 / 1.70		72.59 / 2.27		70.78 / 3.57	
Cylinder 3	87.89 / 1.24		80.75 / 1.57		73.16 / 4.24		69.81 / 8.59	
Cylinder 4	81.49 / 1.51		74.86 / 1.83		69.49 / 4.10		67.19 / 4.59	
Engine Average	#DIV/0!		#DIV/0!		#DIV/0!		#DIV/0!	
COV. Cylinder IMEP								
Cylinder 1	2.5		9.3		14.7		53.1	
Cylinder 2	1.9		2.2		3.1		5.1	
Cylinder 3	1.4		1.9		5.8		12.3	
Cylinder 4	1.9		2.5		5.9		6.8	
Engine Average	1.92		3.96		7.38		19.32	
AVG./STD. Burn Duration 0-10% (Degrees)								
Cylinder 1	15.18 / 2.84		17.61 / 4.40		19.04 / 6.26		16.41 / 11.31	
Cylinder 2	15.22 / 1.81		16.87 / 2.16		18.30 / 2.68		18.94 / 3.08	
Cylinder 3	12.80 / 1.75		14.69 / 2.25		17.43 / 3.75		17.92 / 4.75	
Cylinder 4	14.67 / 1.88		15.91 / 2.36		17.34 / 2.97		17.93 / 3.60	
Engine Average	#DIV/0!		#DIV/0!		#DIV/0!		#DIV/0!	
AVG./STD. Burn Duration 10-90% (Degrees)								
	SP4_1		SP4_2		SP4_3		SP4_4	
	AVG.	STD.	AVG.	STD.	AVG.	STD.	AVG.	STD.
Cylinder 1	14.80	5.80	18.88	8.08	21.81	10.69	18.85	19.35
Cylinder 2	14.15	4.87	16.79	3.98	19.22	4.25	20.35	5.33
Cylinder 3	11.70	4.17	14.58	4.36	19.53	7.24	20.58	8.14
Cylinder 4	13.76	4.55	16.23	4.20	18.84	5.36	20.00	5.8
Engine Average	13.60	4.85	16.62	5.16	19.85	6.89	19.95	9.66

AVG./STD. Burn Duration CA@50 (deg ATDC)				
Cylinder 1	24.38 / 4.09	30.48 / 6.63	35.44 / 9.77	32.83 / 18.92
Cylinder 2	25.51 / 2.60	29.12 / 3.10	31.59 / 3.92	32.61 / 4.71
Cylinder 3	20.86 / 2.77	26.16 / 3.41	34.42 / 5.90	33.97 / 7.59
Cylinder 4	22.14 / 2.81	27.43 / 3.52	30.64 / 4.49	32.14 / 5.46
Engine Average	#DIV/0!	#DIV/0!	#DIV/0!	#DIV/0!
Percent Misfires				
Cylinder 1	0.0	0.5	0.9	15.9
Cylinder 2	0.0	0.0	0.0	0.0
Cylinder 3	0.0	0.0	0.1	0.8
Cylinder 4	0.0	0.0	0.2	0.1
Engine Average Percent	0.00	0.13	0.30	4.20
Cylinder Exhaust Temperatures (Degrees oF)				
Cylinder 1	671.3	649.4	637.3	582.7
Cylinder 2	791.4	768.7	753.1	744.0
Cylinder 3	664.6	647.8	629.7	631.4
Cylinder 4	749.5	732.0	715.4	711.9
Engine Average	719.21	699.46	683.88	667.50

note: Data taken over 1000 engine cycles

GMV-4TF				
MEASURED EMISSIONS	SP4_1	SP4_2	SP4_3	SP4_4
Ignition Type	**MS**	**MS**	**MS**	**MS**
Emissions Measured (Dry)				
NOx (ppm)	1403.16	555.41	206.57	124.48
CO (ppm)	87.78	111.44	137.54	163.00
THC (ppm)	945.96	1043.80	1258.71	2177.78
O2 (%)	13.00	13.64	14.10	14.52
CO2 (%)	4.21	3.83	3.57	3.31

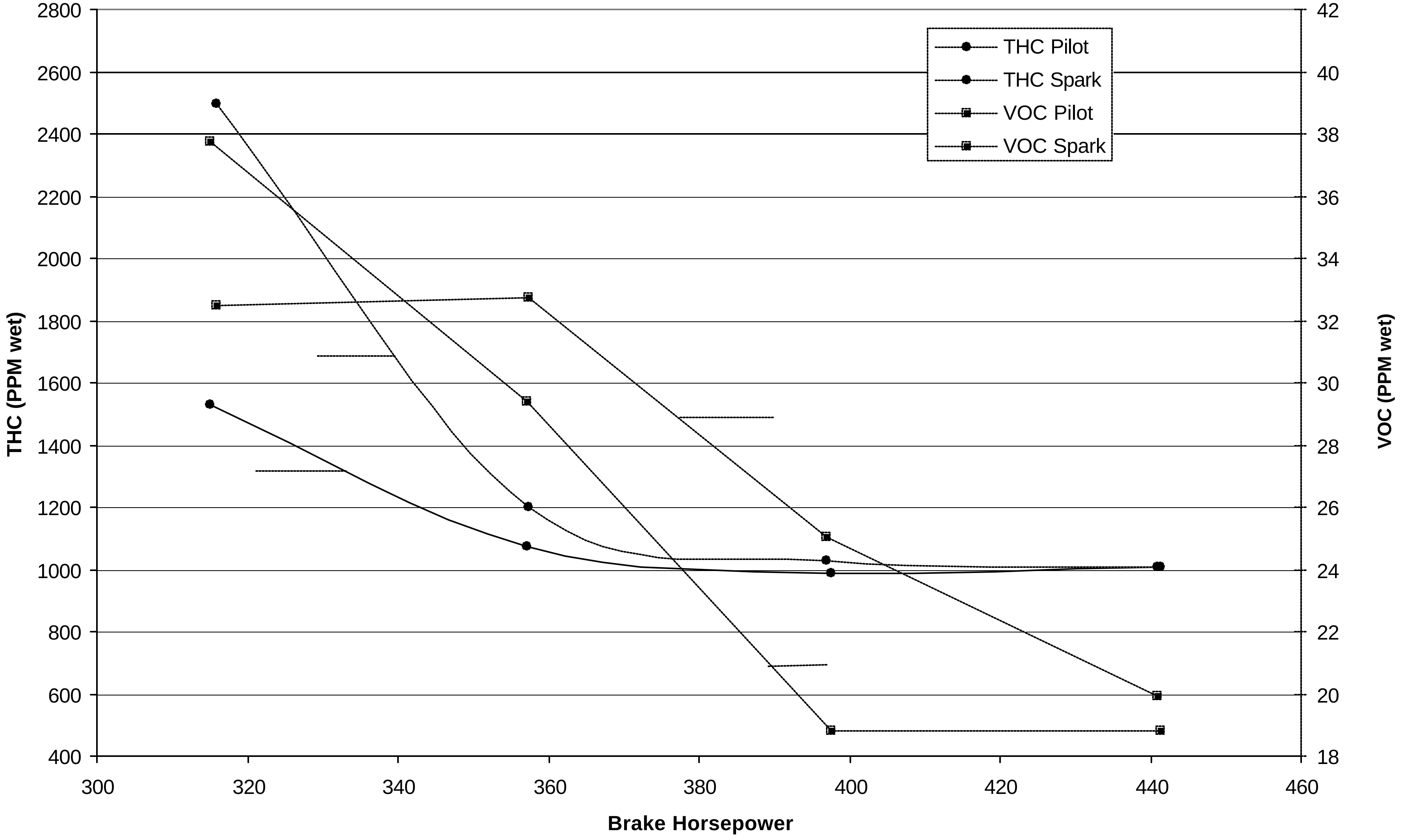
Calculated Carbon Balance Emissions				
NOX (g/hp-hr)	16.57	7.31	3.02	2.08
THC (g/hp-hr)	4.0	5.0	6.7	13.2
CO (g/hp-hr)	0.63	0.89	1.23	1.67
CH2O (g/hp-hr)	0.16	0.21	0.27	0.32
NOx Corr. to 15% O2 (ppm)	1048.72	451.07	178.42	114.84
FTIR Water (ppm, wet)	90136.8	84608.7	80113.3	76237.6
Air Flow (lbs/hr)	7402	7412	7403	7443
Trapped A/F	20.2	NaN	23.4	24.2
A/F Wet Carbon Balance	41.3	NaN	47.7	49.4
FTIR Measured Emissions (PPM, Wet)				
Carbon Monoxide low	80.73	105.16	128.27	150.05
(+-)Carbon Monoxide low	2.75	3.36	4.13	4.43
Carbon monoxide high	53.68	81.62	105.62	128.50
(+-)Carbon monoxide high	70.21	70.80	65.50	66.31
Carbon dioxide	40190.03	37758.36	35713.47	33778.99
(+-)Carbon dioxide	983.89	748.93	720.13	711.72
Nitric oxide	1349.63	512.74	173.30	127.98
(+-)Nitric oxide	35.60	27.62	15.63	12.93
Nitrogren dioxide	84.94	49.54	46.31	43.27
(+-)Nitrogren dioxide	6.38	5.13	4.81	4.80
Nitrous oxide	0.08	-0.02	-0.09	-0.08
(+-)Nitrous oxide	0.56	0.55	0.58	0.59
Methane	955.04	966.78	1100.34	2345.28
(+-)Methane	31.86	29.86	38.24	107.79
Ethyne	0.06	0.07	0.14	0.21
(+-)Ethyne	0.93	0.88	0.80	0.79
Ethene	6.41	7.79	10.25	11.36
(+-)Ethene	0.25	0.25	0.25	0.25
Ethane	17.74	24.24	38.12	77.20
(+-)Ethane	3.59	3.47	4.18	10.21
Propene	1.72	2.00	2.17	2.04
(+-)Propene	1.83	1.73	1.64	1.59
Formaldehyde	18.97	22.12	25.55	26.85
(+-)Formaldehyde	0.20	0.21	0.23	0.39
Water	90136.77	84608.69	80113.34	76237.60
(+-)Water	2744.07	2540.80	2383.07	2360.77
Propane	2.90	3.82	4.89	3.91
(+-)Propane	3.59	3.46	4.16	10.19
Ammonia	0.00	0.00	0.00	0.00
(+-)Ammonia	0.04	0.04	0.04	0.04
Acrolein	-0.13	0.04	0.36	0.39
(+-)Acrolein	0.52	0.49	0.56	1.15
Acetaldehyde	1.09	1.26	1.27	0.11
(+-)Acetaldehyde	0.51	0.52	0.59	0.97
Isobutylene	-0.20	-0.13	-0.13	-0.14
(+-)Isobutylene	0.67	0.63	0.60	0.58
1-3 Butadiene	-0.26	-0.30	-0.42	-0.37
(+-)1-3 Butadiene	1.03	0.97	0.92	0.89
SF6	-0.01	-0.01	-0.01	-0.01
(+-)SF6	0.01	0.01	0.01	0.01
Methanol	-0.83	-0.71	-0.46	-0.15
(+-)Methanol	1.24	1.23	1.21	1.20
NOx	1434.27	562.29	218.28	169.59
(+-)NOx	41.99	32.76	20.38	17.61
Total Hydrocarbons	1006.93	1027.59	1201.80	2494.46
(+-)Total Hydrocarbons	51.71	48.63	60.26	154.86
Non Methane Hydrocarbons	52.40	68.67	102.32	172.36
(+-)Non Methane Hydrocarbons	19.83	19.01	22.06	48.28
VOC	19.94	25.05	32.72	32.49
(+-)VOC	13.25	12.67	14.42	30.18

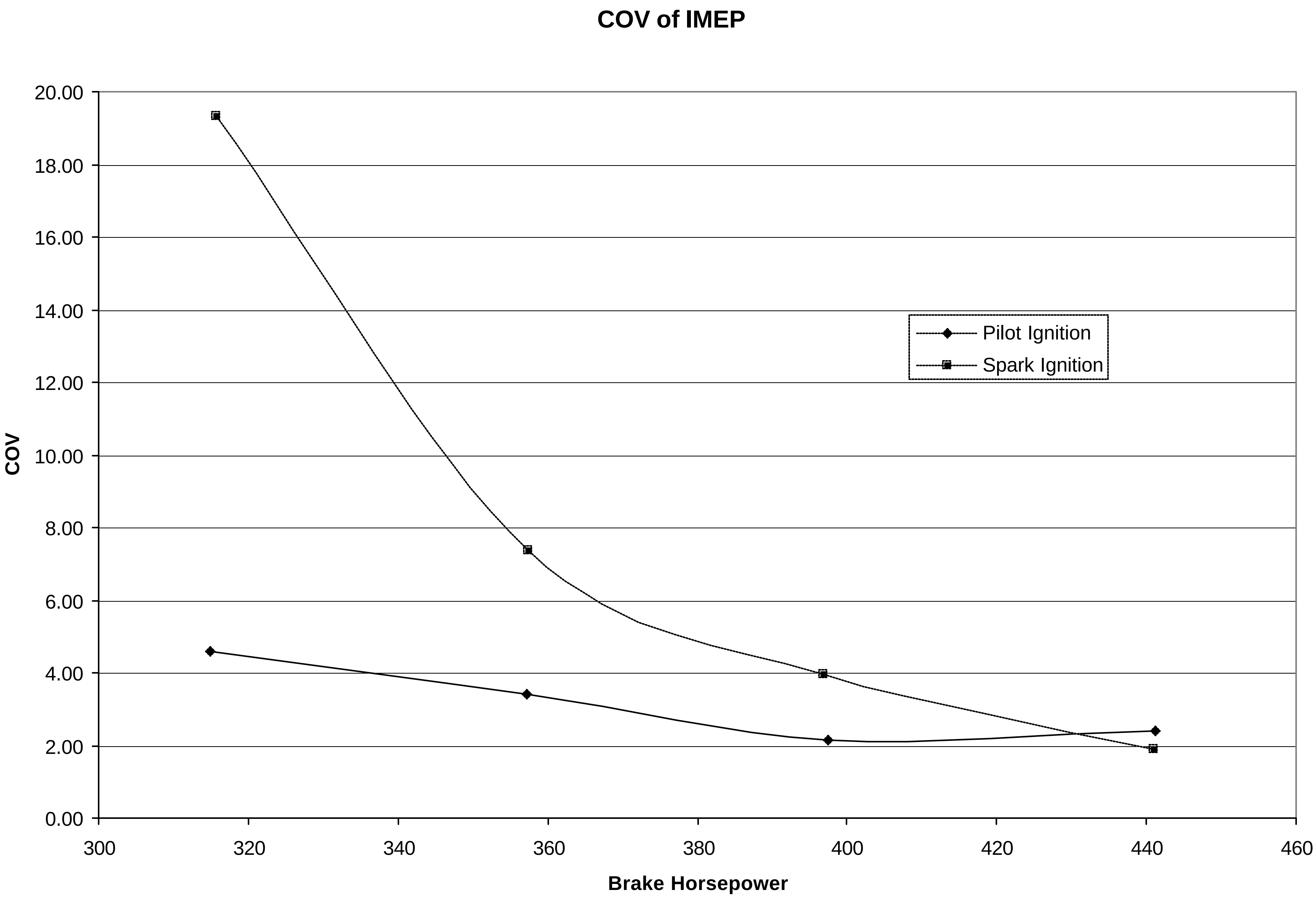

BSFC & BSNOx
Spark vs. MicroPilot
12.19.2002
BSFC (BTU/BHp-hr)
9600
9400
9200
9000
8800
8600
8400
8200
8000
7800
BSNOx (g/hp-hr)
18.0
16.0
14.0
12.0
10.0
8.0
6.0
4.0
2.0
0.0
300
320
340
360
380
400
420
440
Brake Horsepower
BSFC Spark
BSFC Pilot
NOx Spark
NOx Pilot

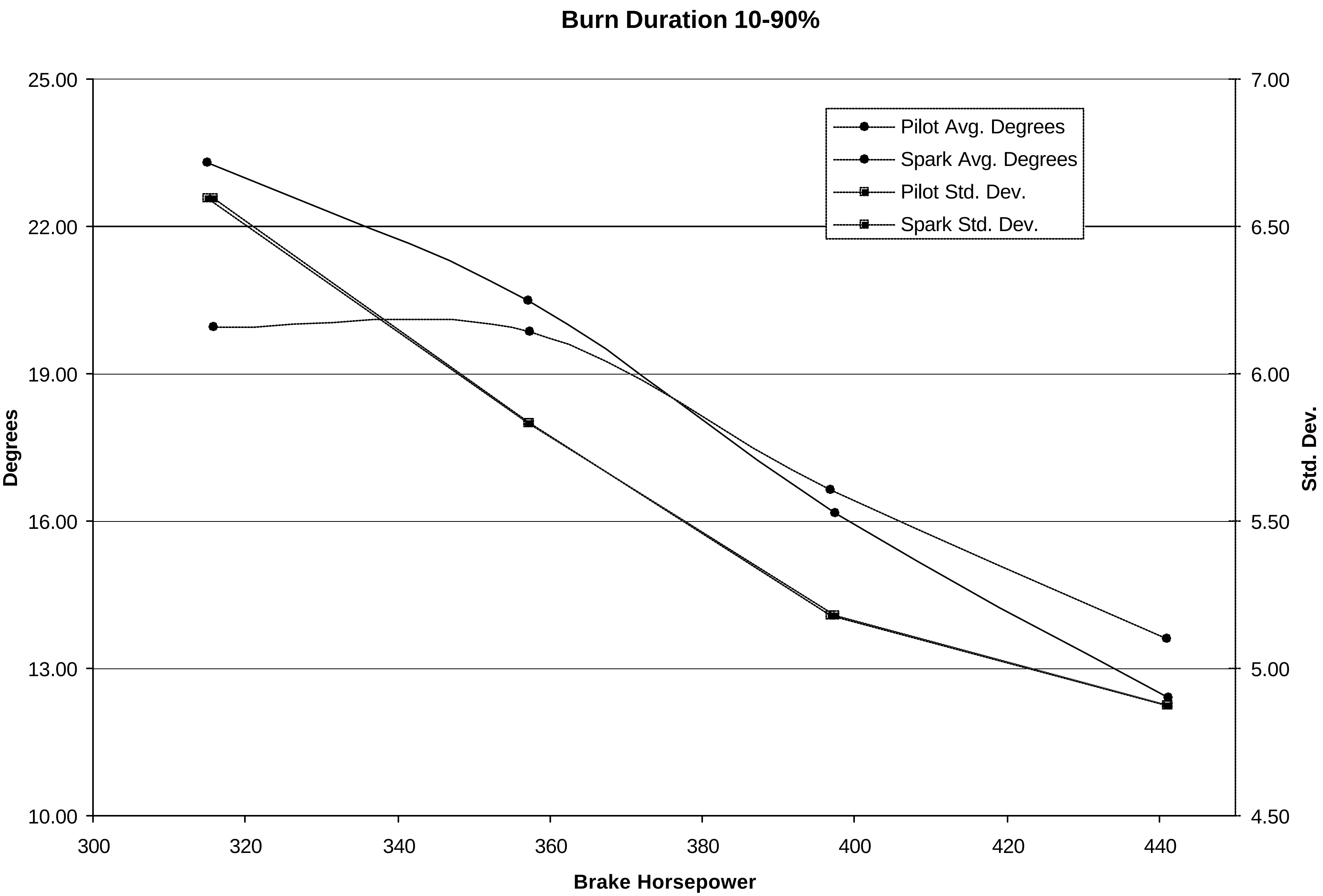
THC & VOC

Spark vs. MicroPilot

12.19.2002




COV of IMEP
COV
20.00
18.00
16.00
14.00
12.00
10.00
8.00
6.00
4.00
2.00
0.00
300
320
340
360
380
400
420
440
460
Brake Horsepower
Pilot Ignition
Spark Ignition


Burn Duration 10-90%
Pilot Avg. Degrees
Spark Avg. Degrees
Pilot Std. Dev.
Spark Std. Dev.
Degrees
25.00
22.00
19.00
16.00
13.00
10.00
Std. Dev.
7.00
6.50
6.00
5.50
5.00
4.50
300
320
340
360
380
400
420
440
Brake Horsepower