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A NOVEL DIESEL-FUELED ENGINE FOR MICROCLIMATE COOLING FOR THE INDIVIDUAL SOLDIER

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Military mission performance in a chemically contaminated environment necessitates the wearing of chemically impermeable protective garments. Soldiers encapsulated in these protective garments in a hot environment will succumb to heat stress. Consequently a microclimate cooling system is being developed. In the present design, a vapor compression cycle chills water which is then circulated over the body to absorb heat via a cooling garment. One of the major components of the vapor compression cycle system is the prime mover, i.e. the engine. For battlefield use, the engine shall utilize battlefield fuel, JP-8. Commercially available engines of the required size, approximately 0.5 hp at 4000 rpm, such as model aircraft engines and small outdoor power equipment (string trimmers) engines do not start and run on JP-8. This effort established the technical feasibility of starting and operating a small, lightweight engine on diesel fuel and JP-8. A commercially available 1.09 cubic inch displacement was used; however, the stock head was replaced with a custom designed head employing a fuel injector and pre-chamber. The engine started by hand cranking from ambient temperature (70 °F) and ran without aids. 0.42 brake hp was developed at 3700 rpm. Brake specific fuel consumption was 0.76 lb/bhp-hr.

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PREFACE

The work described in this report was initiated in response to a solicitation from the U.S. Army Natick Research, Development and Engineering Center for an engine to power a microclimate cooling system for the individual soldier.

The project was a feasibility demonstration carried out between July 1991 and January 1992. It included the selection, conversion, and testing of a miniature engine operating on diesel-type fuels.

The Project Officer for U.S. Army Natick was Mr. Scott Bennett of the Special Projects Section.

A NOVEL DIESEL-FUELED ENGINE FOR MICROCLIMATE COOLING FOR THE INDIVIDUAL SOLDIER

INTRODUCTION AND SUMMARY OF RESULTS

1.1 BACKGROUND

There are combat situations that require soldiers to perform their mission while operating in elevated ambient temperature conditions and wearing a chemical-protective garment. To eliminate heat stress under these conditions, microclimate cooling equipment is essential.

The cooling specifications for the microclimate cooling system for the individual soldier are that it produce 300 watts of cooling at a supply temperature that is 40°F below ambient in an ambient temperature of 100°F. Since the system must be carried on the soldier's back under combat conditions, it must be lightweight, reliable, durable, quiet, and easy to start. The target weight, including fuel and/or batteries for a six-hour mission, is 10 pounds. For logistical reasons, the fuel must meet the military single-fuel specification, which is essentially diesel fuel (JP-8).

The microclimate cooling system for an individual soldier has three essential elements:

1. A cooling vest worn by the soldier.

2. A cooling system that provides a chilled stream of water or air to the cooling vest worn by the soldier.

3. An engine to drive the cooling system and any ancillary equipment such as fans and pumps.

This report is concerned with establishing the feasibility of using a small, high-speed two-stroke cycle diesel as the engine for the microclimate cooling system.

For this application, the engine must produce only 1/2 hp at 4000 rpm. As a result, the engine is much smaller than any commercially available high-speed two-stroke cycle diesel engine. The commercially available "diesel engines" sold to model enthusiasts are compression-ignition engines operating on glow plug fuel which has been doped with ether. The small size of the engine, approximately one-cubic-inch-displacement, and the requirement that it burn diesel fuel make it particularly difficult to design a combustion system and a fuel delivery system that will produce reliable, efficient, and clean combustion for many hundreds of hours with little or no maintenance.

This report describes a novel concept for converting a small, high-speed two-stroke cycle gasoline engine to diesel fuel operation. The key to the concept is in the innovative combustion system and fuel delivery system. The work described is aimed at demonstrating the feasibility of the concept by conducting laboratory tests on an experimental engine specifically designed for this purpose.

1.2 TECHNICAL OBJECTIVES

The objective of the work described in this report is to establish the technical feasibility of starting and operating a small, lightweight high-speed engine on diesel fuel, with the potential to deliver 0.5 brake horsepower at an output shaft speed of 4000 rpm and at a weight of less than 2 pounds.

The specific technical objectives are as follows:

- To determine if combustion can be initiated and sustained in an engine of one-cubic-inch-displacement at a speed of approximately 4,000 rpm.

- To determine if a commercially available miniature aircraft gasoline engine can serve as a test bed for the development of such an engine, and in modified form, as the ultimate power plant.

- To arrive at some preliminary evaluation of design parameters, such as compression ratio and combustion chamber geometry.

- To arrive at some preliminary evaluation of starting techniques and the problem of lubrication.

1.3 SUMMARY OF RESULTS

The following was achieved in the six-month program described in this report.

- A commercially available 1.09-cubic-inch-displacement miniature engine was identified for use in development of the concept.

- The engine has proved rugged enough to be used for the development of the combustion chamber and injection system.

- The modified engine operates as a purely compression-ignition engine giving consistent combustion on pump-grade diesel fuel.

- The modified engine, using pump diesel fuel, will start by hand cranking from ambient temperature (70°F) without aids of any type. A minimum compression pressure of approximately 530 psi is required for cold starting.

- The modified engine's best performance was a brake specific fuel consumption of 0.76 lb/bhp-hr at 3700 rpm and 0.42 brake horsepower.

2. TECHNICAL APPROACH

2.1 ENGINE SELECTION

The first requirement was to select an appropriate engine to serve as a test bed for the program. The required power level (0.5 bhp) and speed (4,000 rpm) indicated a displacement of about 1 cubic inch if the engine was a two-stroke. This means the choice is confined to miniature engines used primarily by modelers.

Since miniature model engines are available in either two-stroke or four-stroke configurations, this becomes an option in selecting a basic platform to work from. We believe the choice of the two-stroke cycle was appropriate for the following reasons:

- Higher compression temperatures due to inherent trapping of more hot exhaust, which gives better combustion.
- Better power density for meeting the weight goal.
- No fuel economy penalty with timed fuel injection.
- More flexible cylinder head for accommodating a pre-chamber.
- Fewer parts, more likely to meet durability goal.
- Easier to lubricate, since the air charge enters through the crankcase and can carry and disperse a small amount of lubricant. Typical 4-stroke miniature engines are lubricated by fuel mixture which leaks past the piston rings.

A search of miniature engine test reports and conversations with modelers were conducted to find an appropriate engine. The primary criteria were:

- Robust design.
- Flat top piston to minimize problems in designing a new cylinder head.
- Separate head and jacket so that the existing jacket and liner could be used with the new cylinder head.

The engine selected is an O.S. model 108F-SR-BX1 with the following characteristics:

Bore	- 1.142 in.
Stroke	- 1.063 in.
Displacement	- 1.09 $in.^3$
Weight	- 26.5 oz (less muffler)
Scavenging	- Reverse loop (Schnurrle) with crankcase compression. Crankcase ported through rotary valve in hollow crankshaft.
Cooling	- Air-cooled
Lubrication	- Castor oil in fuel
Construction Details	
Piston	- Aluminum, flat top with one piston ring and full floating steel wrist pin.
Liner	- Removable, steel.
Connecting Rod	- Single piece aluminum with pressed-in bronze wrist pin and crank pin bearings. Wrist Pin Diameter: 0.287 in. Crank Pin Diameter: 0.316 in.
Crankshaft	- Two rolling element main bearings with over-hung crankpin.
Cylinder Head	- Cast aluminum, eight head bolts.

The wrist pin and crank pin bearings are small for operation as a diesel engine. If one were to increase their size to bring them up to full size diesel engine proportions, the wrist pin and crankpin for this engine would be approximately 0.35 in. and 0.65 in. in diameter, respectively.

Despite the small bearings, it was felt that the engine was strong enough to serve as a developmental test bed for demonstrating the objectives of the program. There was still the option of substituting a needle bearing for the bronze crankpin bearing, if necessary.

2.2 COMBUSTION SYSTEM DESIGN

2.2.1 Introduction

Compression temperature, engine speed, engine size, and fuel atomization all interact to determine ignition delay and combustion duration in a reciprocating diesel engine. In order to achieve the required temperature and fuel/air mixing in a very small high-speed engine, we have used a thermally insulated pre-chamber to heat a portion of the charge to the required level. Fuel is introduced into the pre-chamber by a method described later in this chapter.

At the beginning of compression, the pre-chamber contains mainly residual gas from the previous cycle. As compression proceeds and fresh air is pushed into the pre-chamber, heat is transferred to the incoming charge from the throat and chamber walls of the pre-chamber, raising the air's temperature to higher levels than the adiabatic compression temperature. Further compression results in the charge in the pre-chamber being compressed from ever higher temperature levels, resulting in compression temperatures much higher than would be achievable in an open chamber engine of the same compression ratio.

The question of the optimum compression ratio for an engine of the size under discussion here would have to be determined experimentally and would probably involve a compromise between starting requirements and steady-state running requirements. That there is a real optimum, as opposed to an asymptotic optimum, is fairly certain. In an engine of this size, the surface-area-to-volume ratio effects will predominate at relatively low compression ratios as compared to automotive- or industrial-size engines. In addition, all of the standard clearances, piston to cylinder head, volume above the top ring, etc., become proportionately larger as engine size decreases, thus limiting the percentage of combustion chamber volume that can be placed in the pre-chamber.

The percentage of combustion chamber volume in the pre-chamber and above the piston would also have to be optimized experimentally once a detailed design study of a specific engine configuration has shown the practical limits. Neither issue, optimum compression ratio or relative volume between pre-chamber and main chamber, is resolvable within the scope of the program described here, which is basically a feasibility demonstration.

Another significant advantage of the pre-chamber approach is the mixing afforded by the air motion, which reduces the demands on the fuel injection system. This is amply demonstrated in larger engines where injection pressures in pre-chamber engines typically are in the 3,000 psi range even at speeds up to 5,000 rpm, while injection pressures in open chamber engines run between 15,000 and 20,000 psi at speeds of only 2,000 rpm.

Introducing the fuel into the pre-chamber is advantageous in that some of the fuel can get on the walls and still be burned due to the high wall temperature and high relative air velocity over the wall.

Virtually all of the successful high-speed diesel engines currently in production use a divided combustion chamber. Our approach is the same, except the problems peculiar to the relatively small size must be considered in scaling the system down. Primary among these is insulating the pre-chamber well enough to maintain wall temperatures (and compression temperatures) that are high enough for short ignition delays.

2.2.2 Analytical Results

A simple computer program was written to calculate pressure and temperature in the air charge as a function of crank angle during the compression stroke. This was done in order to help in deciding on the compression ratio, the division of volume between the main chamber and the pre-chamber, and the size of the nozzle connecting the main chamber and pre-chamber. The model includes heat transfer effects between the engine cylinder wall and pre-chamber wall, assuming a constant wall temperature. The model also includes a calculation of the velocity of air through the nozzle entering the pre-chamber as a function of crank angle.

Some sample results are shown in Figures 1 through 4, all for 3600 rpm since that was originally the speed at which we planned to do all of the testing. Figure 1 shows the temperature in the pre-chamber and main chamber as a function of crank angle; cylinder pressure is also shown. The compression does not begin until 100° before top dead center when the piston ring closes the exhaust port. The temperature of the residual gas in the pre-chamber is assumed to be 1,000°F. Varying this value has little effect on the temperature time history in the pre-chamber. During the early part of the stroke, before the exhaust port closes, no air is pumped into the pre-chamber, and the residual gas is heated by the pre-chamber wall, which is assumed to be 1350°F. Once fresh charge begins to enter the pre-chamber, the mass average temperature drops until the compression temperature in the main chamber has risen significantly.

Figure 2 shows how the fresh air and residual gas fractions vary in the pre-chamber for the case where the clearance volume is equally divided between the pre-chamber and the main chamber. Figure 3 shows the actual mass distribution between the pre-chamber and the main chamber.

Figure 4 shows the velocity of the gas entering the pre-chamber as a function of crank angle for various diameter throats between the pre-chamber and the main chamber. Here the tradeoff is between atomization of the fuel in the pre-chamber due to high air velocities and pumping losses due to those velocities.

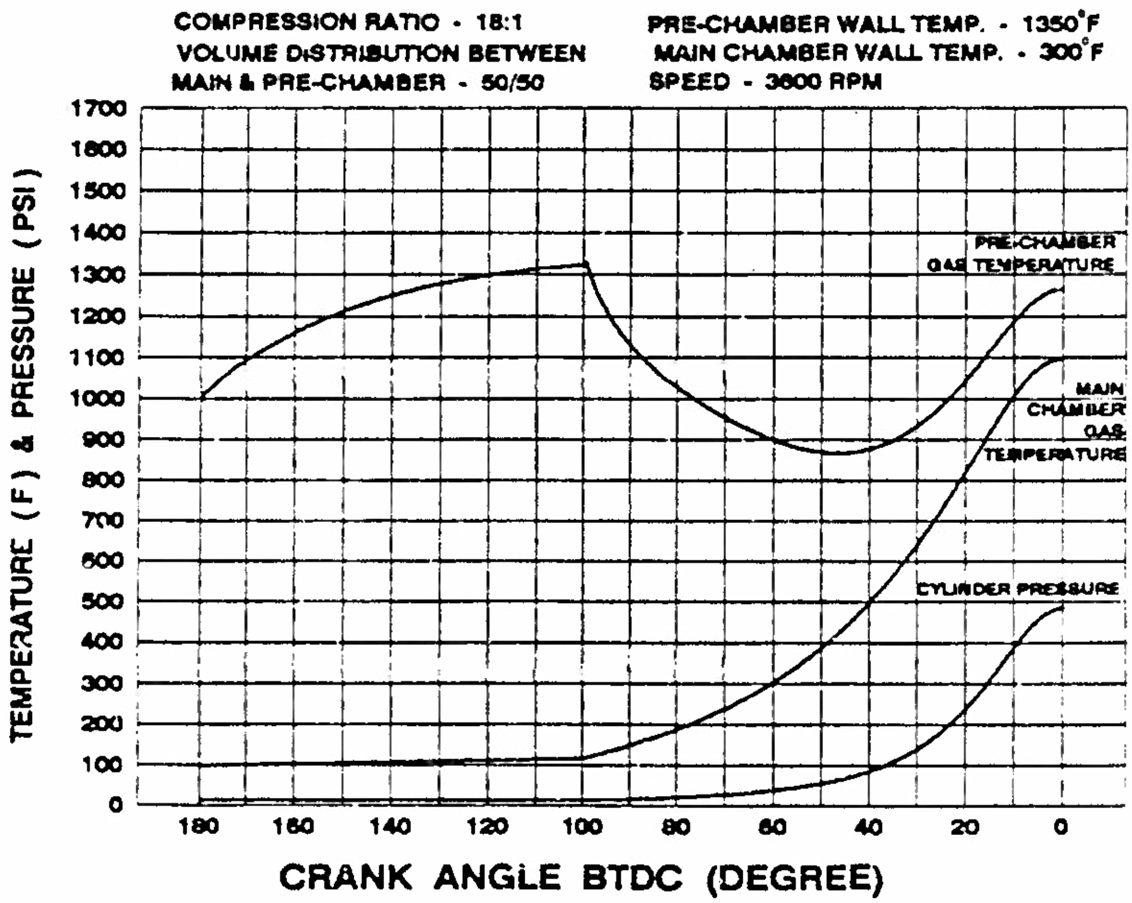


Figure 1. Chamber Temperature and Pressure vs. Crank Angle

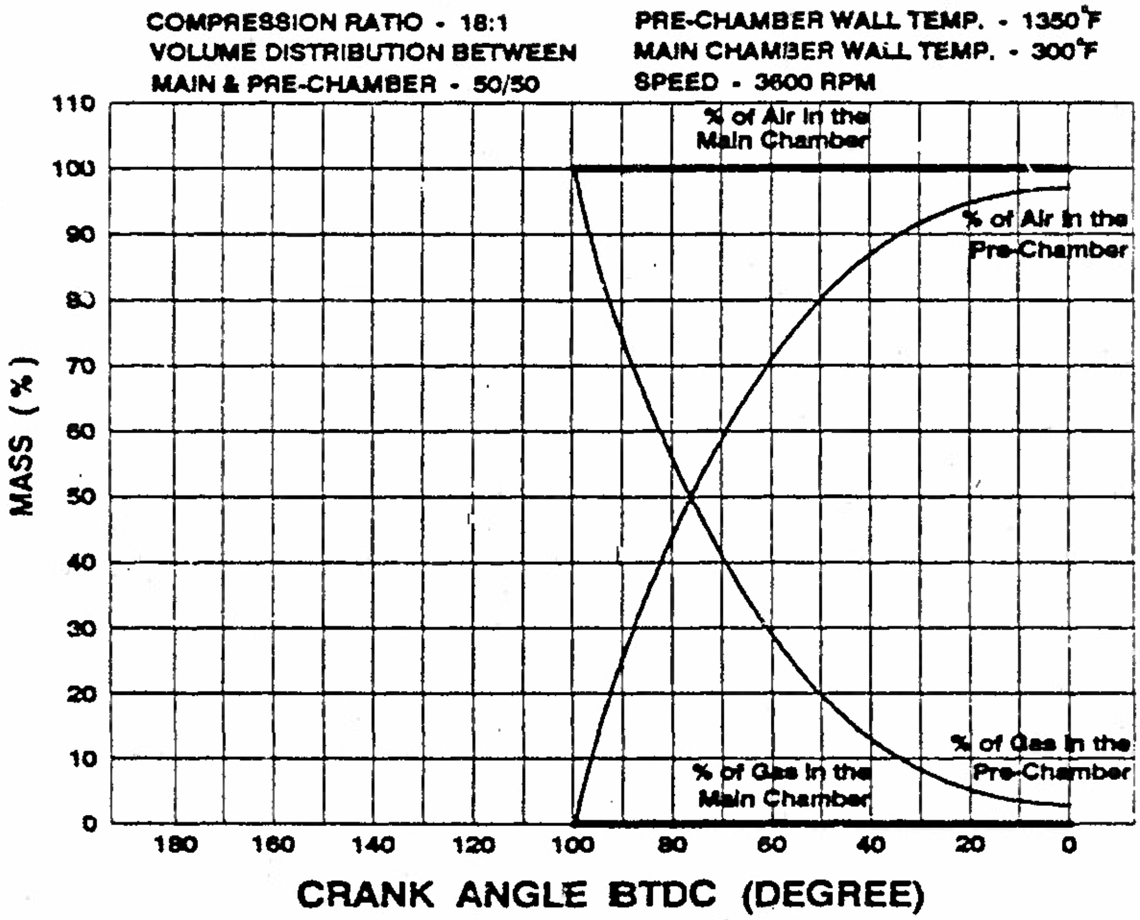


Figure 2. Percent Mass Distribution vs. Crank Angle

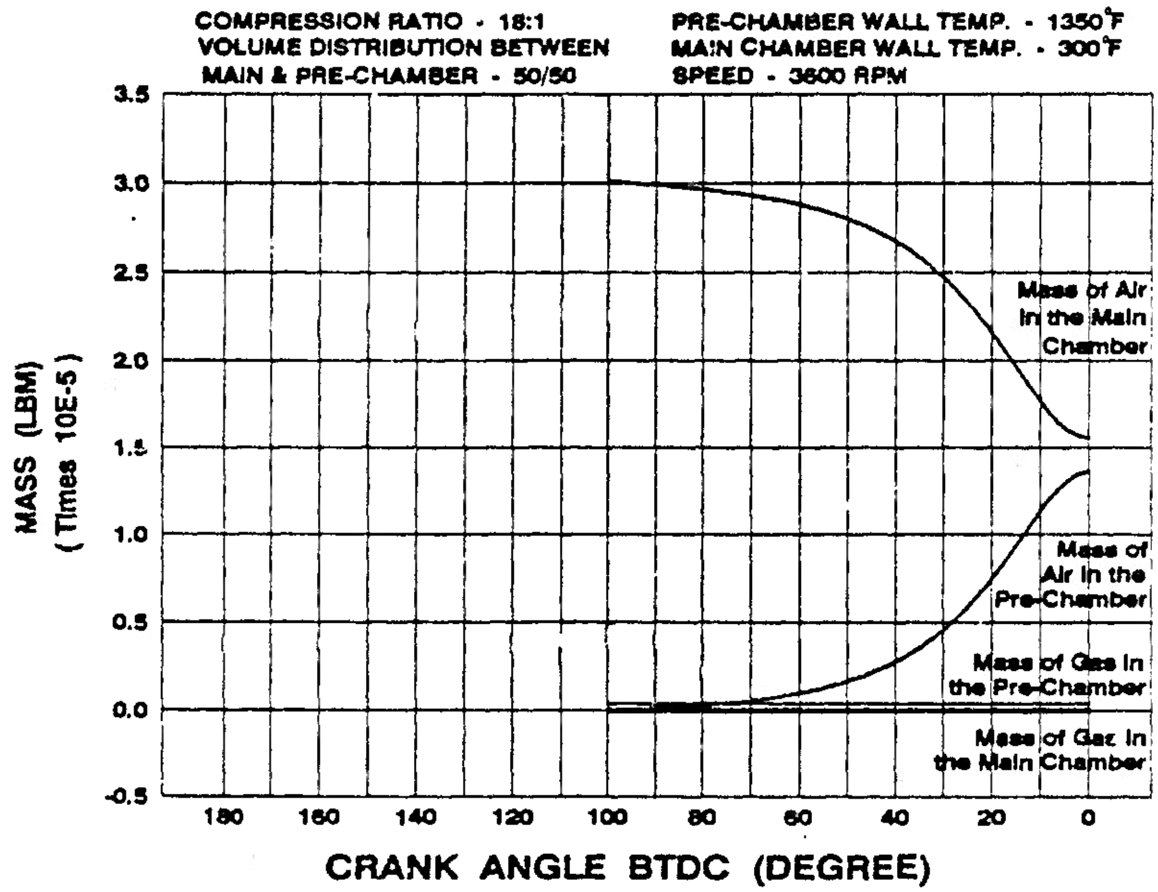


Figure 3. Mass Distribution vs. Crank Angle

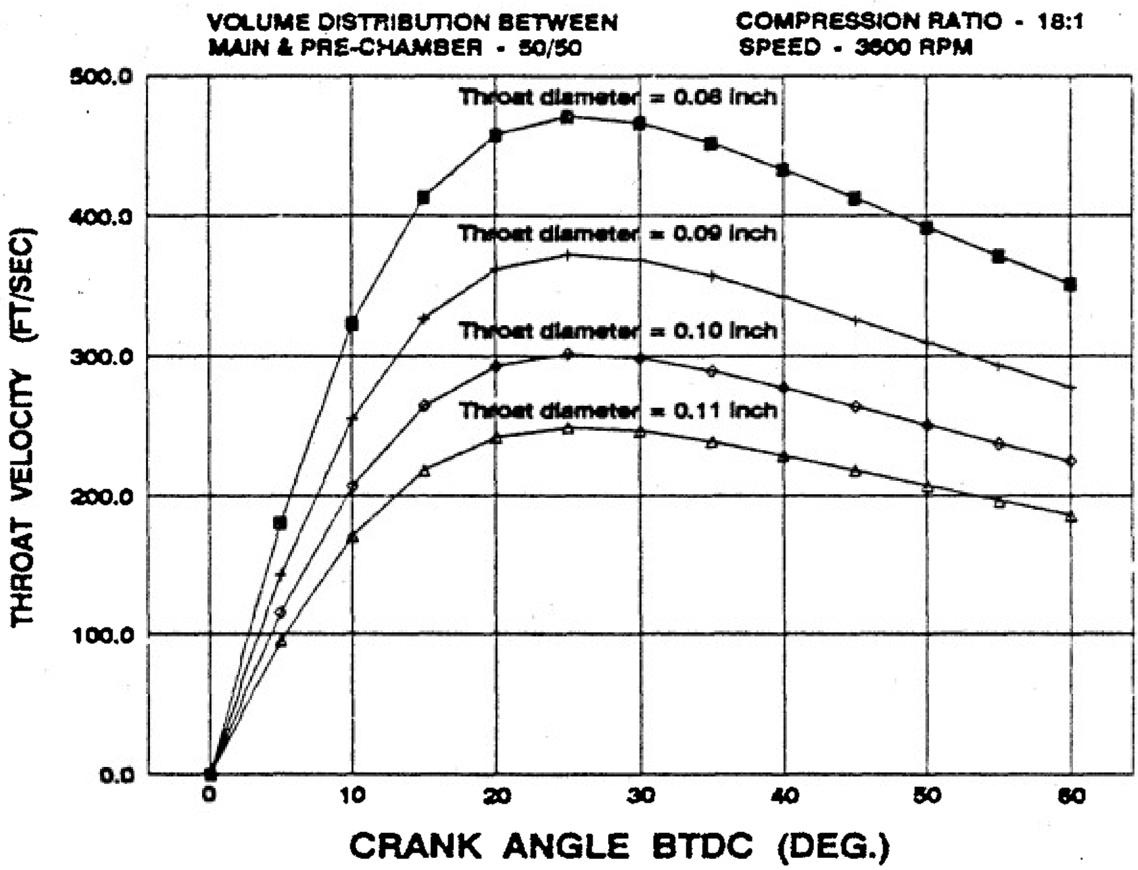


Figure 4. Throat Velocity vs. Crank Angle

Figures 1 through 4 are all for the selected compression ratio of 18 and a 50-percent distribution of the clearance volume. Varying these parameters in the computer program showed, among other things, that the temperature-time history in the pre-chamber is not a strong function of compression ratio once the engine is running, hence the decision to use a relatively low compression ratio for this type of combustion system.

The split in clearance volume between the main chamber and the pre-chamber did not have a large influence on the temperature-time history in the pre-chamber. Varying the temperature of the pre-chamber wall from 1050°F to 1650°F increased the temperature at the end of compression of the gas in the pre-chamber from 1220°F to 1320°F, but had a very significant effect on the temperature-time history.

The final design parameters chosen for the proof-of-concept experiment are as follows:

Compression Ratio (geometric) - 18
Relative Distribution of Clearance Volume - 50/50
Size of Pre-Chamber - 0.375 in. diameter
Size of Throat - 0.11 in. diameter
Piston to Head Clearance - 0.030 in.

2.2.3 Cylinder Head Design

Figure 5 shows the experimental cylinder head designed to the above specifications (roughly five times size). The scope of the program dictated the use of the existing liner and piston. The liner projects above the piston at top dead center (as Figure 5 shows) and this fact, together with the existing head bolt pattern, constrains the pre-chamber to be inside the diameter of the engine bore. This, in turn, means that the throat from the pre-chamber enters the main chamber near the center of the bore. Hot, partially burned gas exiting the pre-chamber will have a difficult time finding all of the air in the main chamber. Some consideration was given to using multiple throats between the main chamber and the pre-chamber to attempt better mixing, but this would have interfered with the swirl pattern and fuel-air mixing in the pre-chamber during compression.

A new cylinder head was machined from aluminum using a specially made carbon tool to electrically discharge machine (EDM) the cooling fin pattern. A 0.010-in. air gap was created between the cylinder head and the pre-chamber to reduce heat transfer and maintain a high pre-chamber temperature. The pre-chamber is pressed into the cylinder head from the combustion chamber side as shown in Figure 5.

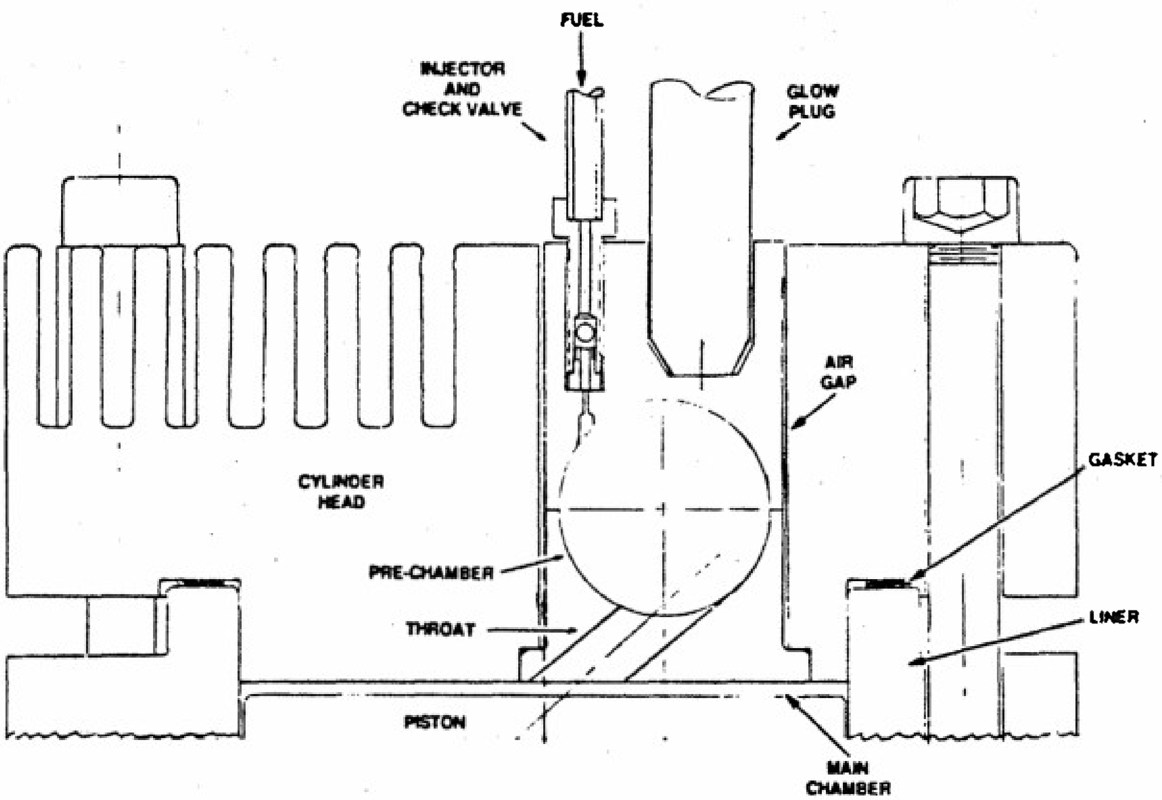


Figure . Experimental Cylinder Head

The pre-chamber insert assembly is made from 304 stainless steel and is beam welded together at the major diameter of the pre-chamber. Sample pieces wrre made and sectioned to insure that the weld penetration was total. As shown in Figure 5, a well was placed in the pre-chamber assembly so that a commercially available automotive glow plug could be used to heat the pre-chamber. The glow plug is supported by a simple bridge between two of the cylinder head studs and was preloaded with a spring to maintain physical contact with the pre-chamber, while not exerting excessive force on the assembly.

The injector is also shown in Figure 5 and consists of a simple ball check valve and narrow (0.012 in. diameter) opening into the pre-chamber. The ball is 0.031 in. in diameter and is made of tungsten carbide. The upper part of the injector is made from a standard No. 2-56 Allen head screw. The lower piece on which the ball rests in the open position is made from 304 stainless steel.

Fuel enters the injector from a constant pressure source (0 to 100 psi) and opens the ball valve whenever the gas pressure falls below the fuel pressure. Timing of injection, therefore, is completely uncontrolled. The simplicity of this device, compared to using a timed injector which would have to be designed, built, and tested from scratch, was believed justified for establishing proof-of-concept. If consistent ignition could not be obtained with the injector of Figure 5, we felt that it probably could not be obtained with a more sophisticated device.

The overall compression ratio can be controlled by varying the thickness of the annular cylinder head gasket and/or machining the annular recess in the cylinder head (see Figure 5). The gaskets were made of soft copper.

2.2.4 Starting

The approach taken in starting the engine is to use a 3600-rpm induction motor to crank the engine. The pre-chamber is heated by the work of compression and if desired, by the glow plug shown in Figure 5. This allows the engine to be started easily without using ether, so that the steady-state running characteristics and starting can be examined independently.

It is also possible to hand crank the engine to investigate various techniques for aided and unaided starting. No internal starting devices were included in the experimental engine beyond the glow plug shown in Figure 5.

2.3 LUBRICATION AND COOLING

The O.S. engine, like most miniature engines, is normally lubricated by castor oil mixed with the fuel, which typically contains castor oil (20 percent), methanol (75 percent), and nitromethane (5 percent). The fuel is aspirated through a carburetor where the air/fuel ratio is controlled by a needle valve. The mixture enters the crankcase where the heavier castor oil with its high boiling point is deposited, while much of the remaining fuel is evaporated. The oil thus provides an oil film that lubricates the moving parts while the excess oil is thrown off as droplets that find their way through the engine.

In these feasibility tests, we determined the approximate lube oil requirement by measuring the total fuel flow with the engine operating in its normal mode. From this we calculated the required castor oil flow and calibrated a drip tube to provide that flow rate to the crankcase in the normal manner using the needle valve on the carburetor to regulate the flow rate. This proved to work well and allowed us to operate without a lubrication problem during the feasibility tests.

Cooling is by forced air. A commercial electric fan is used to direct air at the cylinder head.

2.4 TEST FACILITY

2.4.1 Overall Arrangement

An overall view of the test facility is shown in Figure 6. The engine is connected to a 3/4-hp electric induction motor, which in turn is connected to an electric dynamometer. To the right of the dynamometer is its load control and power and speed read-out devices. The dynamometer is on loan from the U.S. Army Natick RD&E Center and is a Magtrol Model HD-700-6.

Figure 7 shows a close-up of the engine and flywheel assembly and the coupling between the flywheel and the induction motor. The flywheel inertia and coupling stiffness were selected to keep the torsional natural frequency of the engine vibrating against the load well below the minimum operating speed. The flywheel inertia is 0.0674 lb-in.-sec^2 and the coupling stiffness is 974 in.-lb/rad. The natural frequency is therefore somewhat below 1200 cycles per minute and the forcing frequency is once per revolution.

The engine is mounted rigidly to an aluminum stand which is in turn bolted rigidly to an aluminum base plate to which the motor and dynamometer are bolted. All of this rigid mounting makes for a lot of noise transmission and amplification

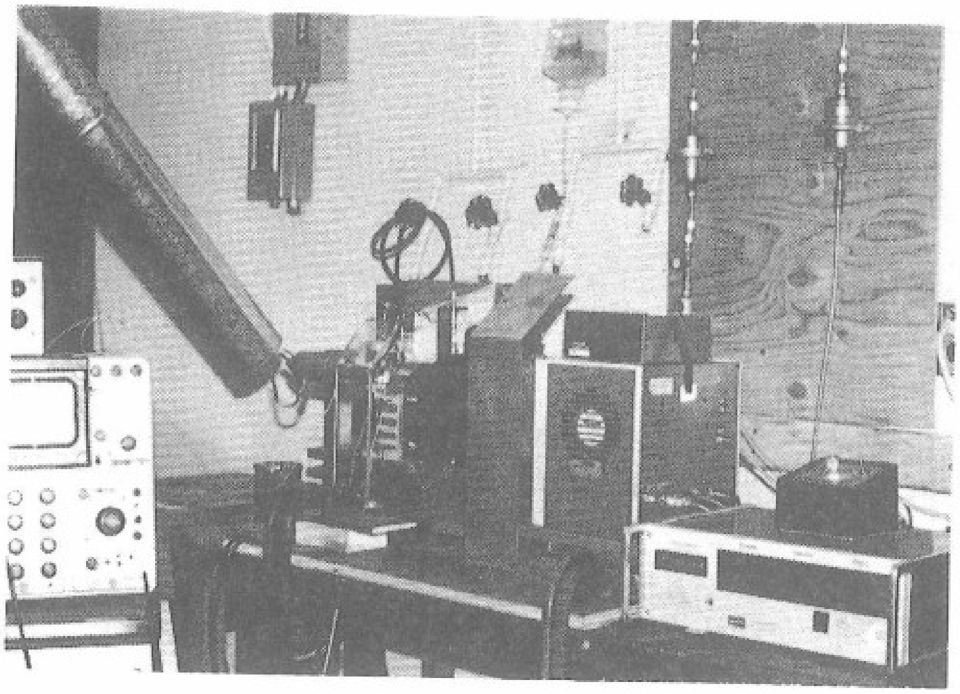


Figure 6. Engine Test Facility

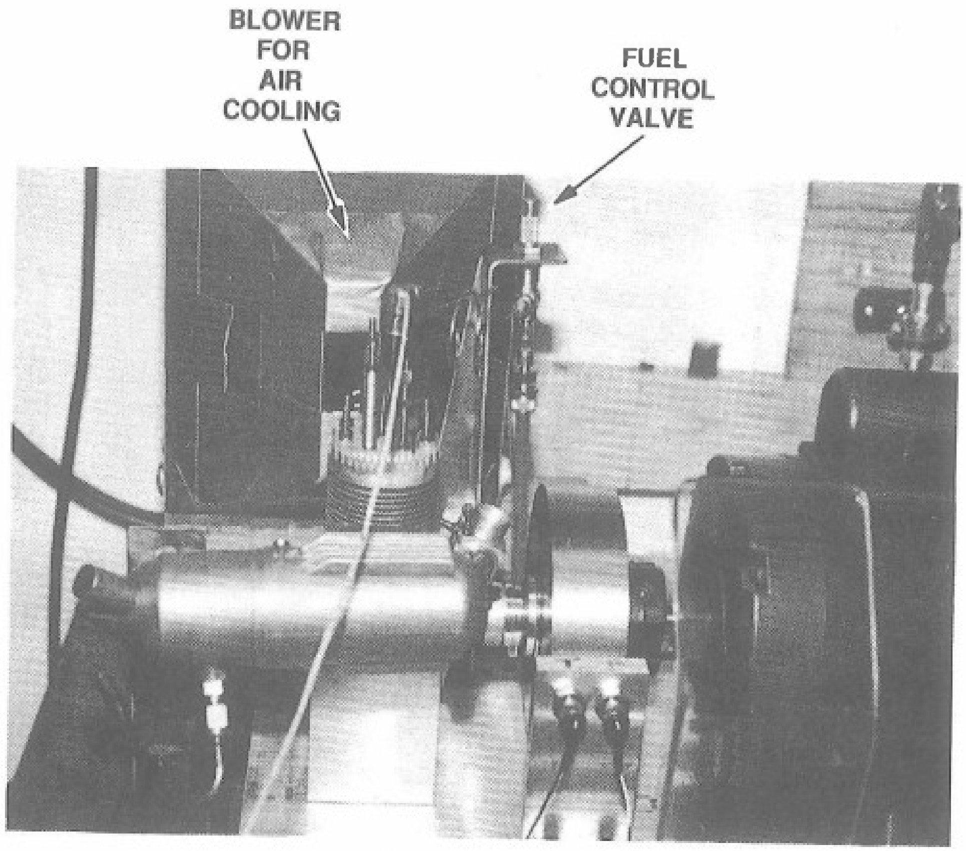


Figure 7. Test Engine with Flywheel and Coupling

but was necessary in this test setup to maintain alignment. The entire engine-motor-dynamometer assembly, as shown in Figure 6, was isolated from the bench by stock rubber vibration isolators. Figure 7 also shows magnetic pickups which sense timing marks placed in the flywheel for the oscilloscope display of the cylinder pressure trace.

2.4.2 Fuel System

The fuel system is shown in Figure 8. The fuel reservoir is a standard 300-cc sample cylinder that is manually filled with fuel through the cap shown adjacent to the pressure gauge. The reservoir is then pressurized by shop air that is introduced through a regulator and filter. The fuel pressure could thereby be varied from zero to 100 psi. Fuel flows from the reservoir through a filter, a shutoff valve, and a sight needle valve before passing to the fuel control valve (Figure 7) and injector.

The sight needle valve, shown in detail in Figure 8, is used to measure the fuel flow. It consists of a sight glass which traps an air bubble in the fuel line. By counting the drops of fuel passing through the air bubble in a unit of time, we could calculate the flow to the engine. The sight needle valve was calibrated by removing the cylinder head and allowing the fuel injected to run into a graduated cylinder. The fuel pressure was maintained at the same level as during an actual run (typically 50 psig).

The scale on the reservoir sight glass could also be used for fuel consumption measurements when run duration was extended.

Two fuels were tested during the program: a pump grade fuel and JP-8 supplied by the U.S. Army Natick RD&E Center. The specific gravity of the fuels tested is as follows:

Pump "Red Diesel" No. 2 – 0.86
JP-8 – 0.78

A military standard diesel fuel (DF-2) was supplied by the U.S. Army Natick RD&E Center, but was not tested; its specific gravity is 0.86.

2.4.3 Lubrication System

As already mentioned, lubrication of the engine was by aspiration of castor oil through the needle valve in the engine's carburetor. The castor oil was fed to the engine by gravity from the plastic reservoir shown in Figure 8. Flow rate was determined by counting drops through a sight glass as just described for the fuel

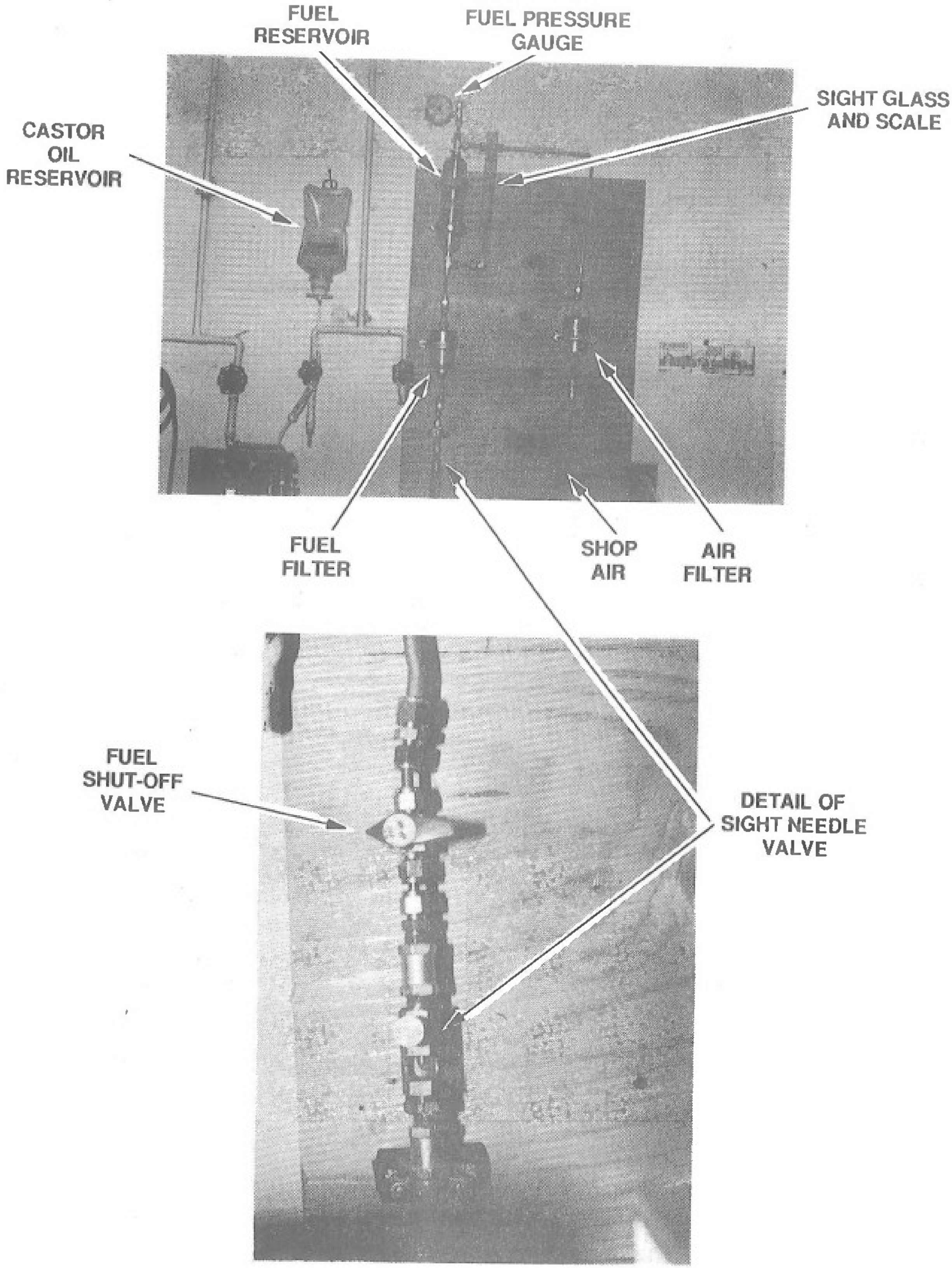


Figure 8. Fuel System

flow. As already discussed, we determined the lube oil flow by running the engine as a conventional glow plug engine and calculating the castor oil portion of the total fuel flow rate. The castor oil flow rate was also ratioed down with rpm since the tests were performed at 13,000 rpm (using a calibrated propeller). This flow rate proved to be a problem in that castor oil would enter the pre-chamber and clog the injector passage and ball. The castor oil flow rate was then reduced. The rate used for all the tests reported here was approximately 0.022 lb/hr. The fuel flow rate in the tests reported here varies from 0.22 to 0.47 lb/hr.

The specific gravity of the castor oil used in these tests is 0.96.

2.4.4 Instrumentation

Engine torque and speed were measured with the Magtrol dynamometer, which was provided with a calibrating torque arm to which known weights were added. It should be noted here that the electric motor between the engine and dynamometer, although switched off, still has an undetermined amount of friction and windage losses. This means that all of the horsepower and specific fuel consumption readings are somewhat low and high, respectively.

Fuel flow measurement is discussed above in the description of the fuel system.

The engine is equipped with a model 609 Kistler pressure transducer which was mounted in the cylinder head with the sensing element flush with the main chamber (not shown in Figure 5 for reasons of clarity).

Thermocouples are mounted on the cylinder barrel and cylinder head. A single thermocouple is mounted in a small diameter hole parallel to the glow plug well in the pre-chamber assembly. It is at the same depth as the glow plug. This thermocouple is the most important in that it is used to define the pre-chamber temperature at which ignition is effected. The external thermocouples are located on the cylinder head at the base of a fin at the outside edge and at the base of the top, middle, and bottom fins of the cylinder barrel. All were on the side of the engine facing the blower (see Figure 7).

The glow plug used was a GM automotive-type AC12G. It was powered by a variable-voltage power supply and had a maximum power consumption of about 80 watts when in contact with the pre-chamber. Bench tests were performed with a simulated pre-chamber to determine how long the glow plug could be left on at 12 volts before it would fail. This turned out to be over 5 minutes. Since the pre-chamber needs to be heated to only 320°F to effect a start and the glow plug could do this in about half a minute, glow plug life is not a problem even at maximum voltage.

3. TEST RESULTS

3.1 ENGINE PERFORMANCE

The engine configuration of Figure 5 was operated at three compression ratios and with two fuels. Initial runs were intended to debug the engine and test facility. Few difficulties were encountered. The two most important were:

- The quantity of castor oil fed to the crankcase was initially too high and tended to clog the injector and interfere with fuel atomization as discussed in Section 2.4.3. The quantity was reduced, and this problem disappeared.

- When the glow plug was used to start the engine, there was a tendency for the 0.012-inch-diameter injector passage to become plugged with carbon. A very thin cylindrical annulus was machined around the injector which thermally isolated it from the remainder of the pre-chamber in the radial direction. This modification was made with a specially constructed EDM tool. No further carboning of the passage, due to glow plug heat, was encountered.

The engine was started by cranking with the electric motor at 3600 rpm. Cranking for about one minute at that speed would get the pre-chamber temperature to about 320°F at which time the engine would begin to fire and become self-sustaining. Starting could be accelerated by supplementing the pre-chamber heat flux due to compression with heat from the glow plug. Because the glow plug offered only a minimal advantage in starting time, it was seldom used to effect a start.

During cranking, the engine would produce quantities of white smoke almost as soon as the fuel was switched on, but no significant rise in combustion pressure (or pressure spikes on the expansion stroke) was observed until the pre-chamber temperature reached 320°F.

Figure 9 shows a typical pressure-time trace taken under firing conditions. The following summarizes the combustion characteristics of the engine.

- Combustion is very rapid, with peak pressure occurring near or even before top dead center.

- Combustion is retarded as the fuel flow rate is increased.

SPEED 3980 RPM
LOAD 18.0 PSI, BMEP
COMPRESSION RATIO 17 TO 1

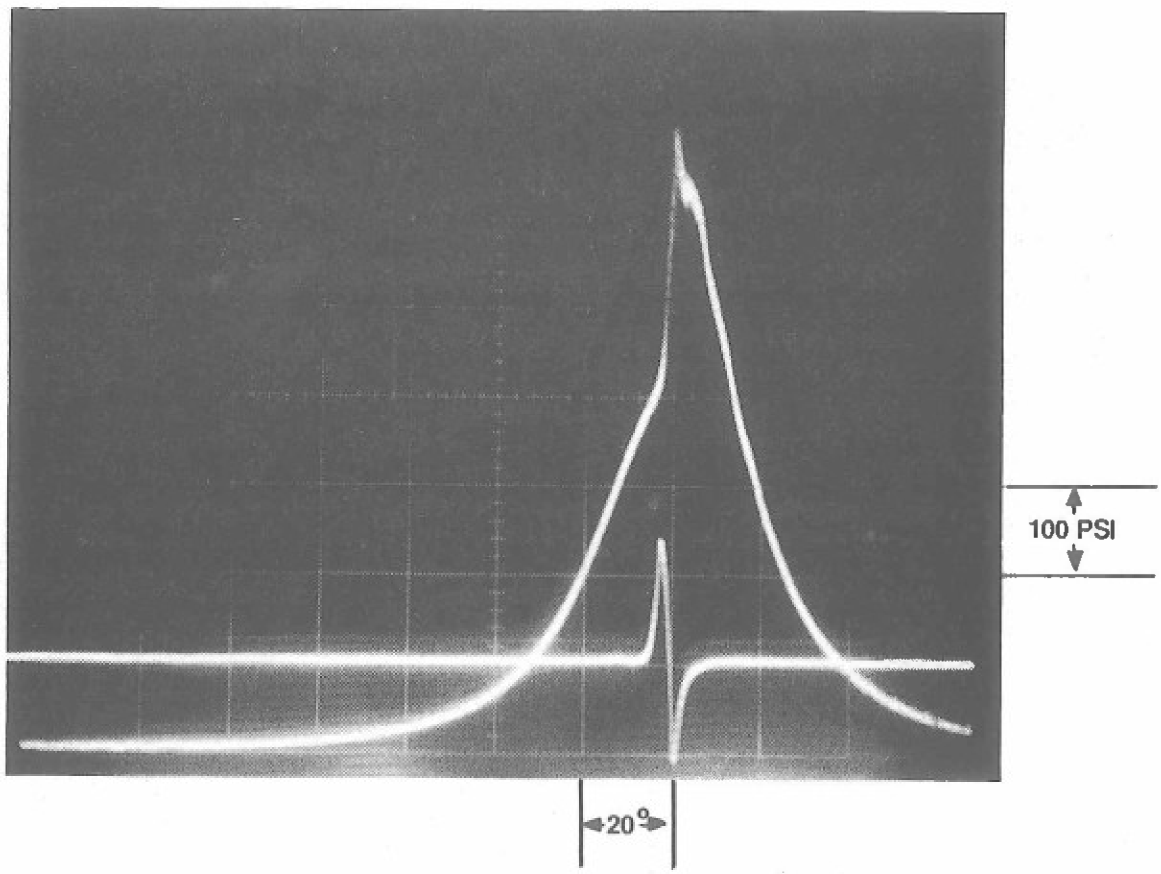


Figure 9. Cylinder Pressure vs. Time

- After the initial rapid burning near top dead center, combustion appears to be dominated by the ability of unburned and partially burned fuel to find air in the main chamber.

It is clear that the combustion characteristics are dominated by the uncontrolled nature of the fuel injection system. All of the fuel is presumably present in the pre-chamber before the compression stroke begins. The tendency for combustion to advance and retard with fuel quantity is due to the cooling effect of the fuel on the overall charge. The more fuel, the longer the ignition delay and the later peak pressure occurs with respect to top dead center. The output of the engine was typically fuel-limited at constant speed. There was always some fuel quantity beyond which the engine would cease to fire. Figure 10 is a pressure time trace taken fairly near this limiting fuel quantity. Note that combustion is later than that shown in Figure 9, but peak pressure is occurring at less than 10 degrees after top dead center, which means it is still somewhat advanced for optimum efficiency.

Figure 11 shows the brake specific fuel consumption vs. brake mean-effective pressure for the test engine for the range of speeds and three compression ratios tested in this program. The brake mean-effective pressure can be converted to horsepower by the following equation:

$$\text{bhp} = \text{bmep} \times \text{rpm} \times \text{displacement}/396{,}000$$

Thus, 45 psi brake mean-effective pressure corresponds to 0.5 brake horsepower at 4,000 rpm for a 1.09 in.^3 two-stroke engine.

The trend to lower brake specific fuel consumption with higher brake mean-effective pressure is due to two factors – the tendency for combustion to retard slightly to give peak firing pressures closer to the optimum crank angle, as discussed above, and the higher mechanical efficiencies as the load is increased at constant speeds. It should be noted that the horsepower (brake mean-effective pressure) and brake specific fuel consumption numbers are low and high, respectively, due to the unaccounted-for windage and friction losses in the induction motor between the engine and dynamometer.

Figure 11 shows two data points taken with the compression ratio set at 22 to 1. The engine was erratic in behavior at this condition and would not develop much more than the 21 psi brake mean-effective pressure shown. The reason for this is not understood. It should be noted that the distribution in volume between the pre-chamber and the main chamber is 60/40 at 22 to 1 vs. 50/50 at 18 to 1. It seems unlikely, however, that the combustion characteristics should be that strongly influenced by that rather marginal change.

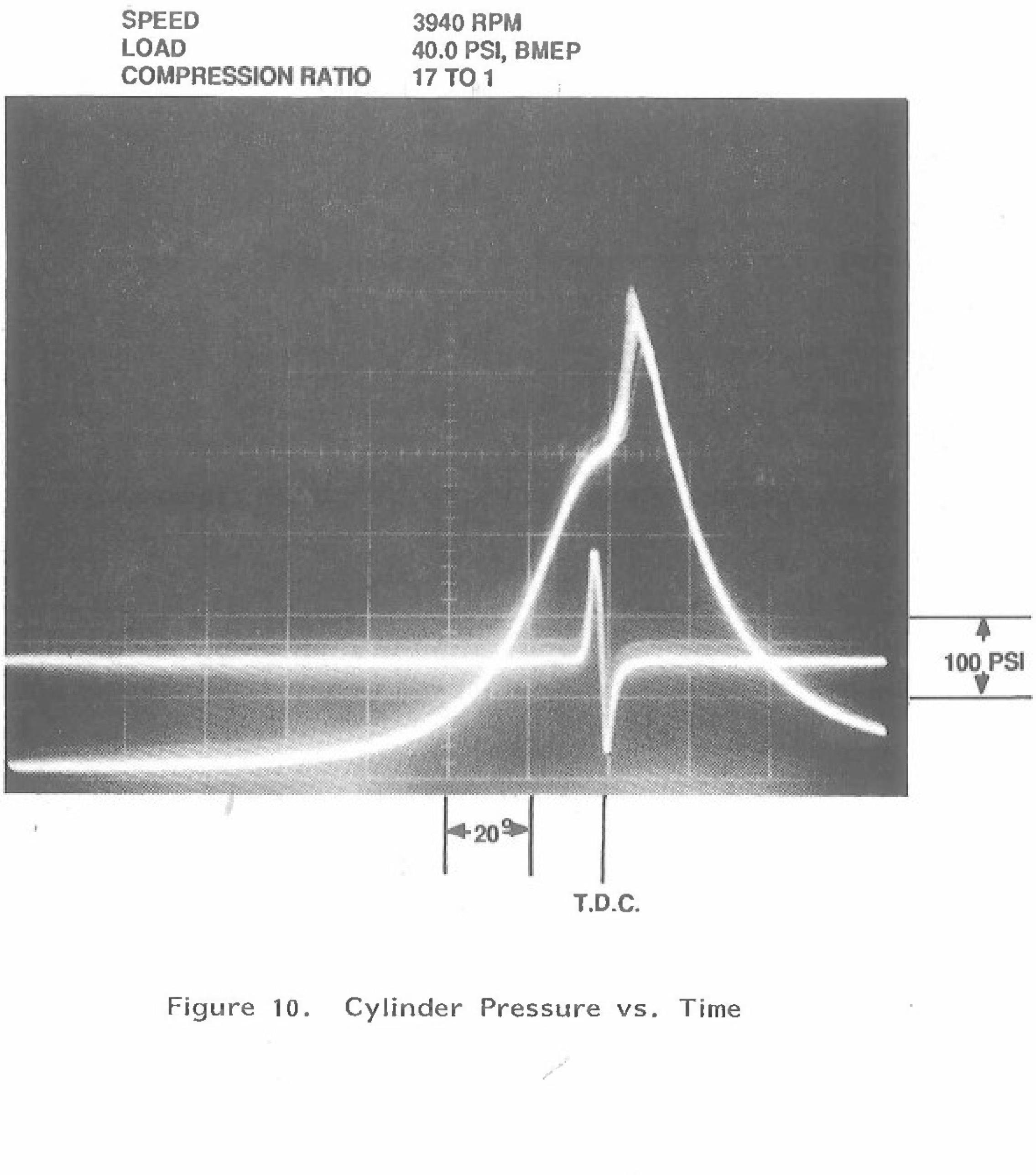


Figure 10. Cylinder Pressure vs. Time

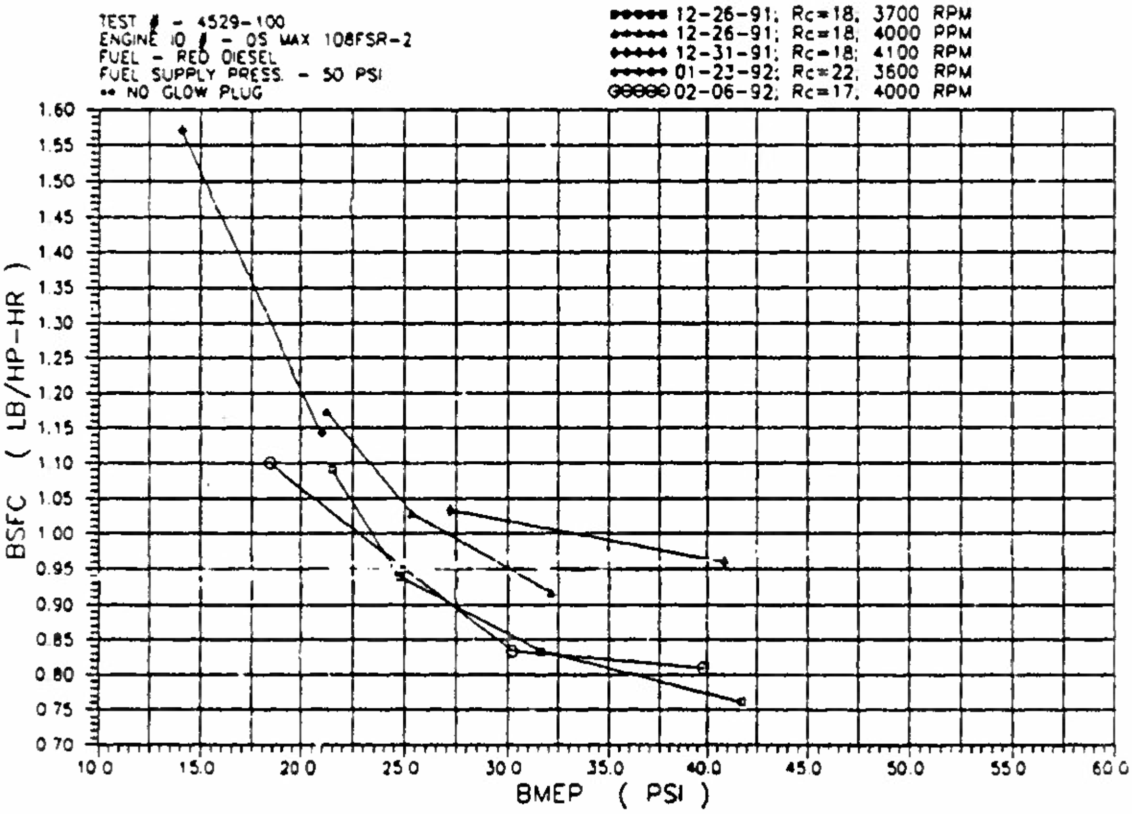


Figure 11. Brake Specific Fuel Consumption vs. Brake Mean-Effective Pressure

Figure 12 shows the peak firing pressure vs. brake mean-effective pressure for the same runs as Figure 11. The general trend to lower firing pressures with brake mean-effective pressure is due to the tendency for combustion timing to vary with load as already discussed.

Figure 13 shows the pre-chamber temperature (as measured by a thermocouple buried in the pre-chamber housing about 0.050 in. from the inner surface of the pre-chamber) for the runs of Figure 11. Again, the tendency for the pre-chamber temperature to drop with increasing load is probably due to retarding combustion timing as fuel flow is increased.

Figures 11 through 13 show that higher engine speed causes the pre-chamber to run hotter and the ignition to occur earlier, which results in higher firing pressure and higher brake specific fuel consumption.

None of the data shown in Figures 11 through 13 were taken with the glow plug in place. Once the engine was running, switching the glow plug on and off had no observable effect on combustion. Apparently, its contribution to the total heat flux in and out of the pre-chamber assembly was not significant. Presumably, the glow plug could only help overcome the tendency to stop firing at higher fuel flows since otherwise it would aid in advancing an already too-advanced combustion system.

Figure 14 shows the temperature of the engine at various locations at the base of the cooling fins (as well as the pre-chamber) as a function of brake mean-effective pressure for one of the conditions of Figure 11. Note that component temperatures follow the general shape of the pre-chamber temperature, but lag it somewhat. This again is due to a combination of load change and combustion timing.

Figure 15 shows how combustion timing, as measured by the crank angle at which peak cylinder pressure occurs, varies with speed. The higher brake specific fuel consumption of the 4100-rpm runs is most likely due to the relatively advanced crank angle at which peak pressure occurs.

Figures 16 and 17 show the calculated equivalence ratios in the pre-chamber and the main chamber at top dead center for the runs of Figure 11. The equivalence ratio is calculated by using the actual fuel flow, and the air flow is calculated by using the program described in Section 2.2.2 since the actual scavenging ratio and efficiency are not known for this engine as a function of speed. These curves indicate that the mixture becomes fuel-rich in the pre-chamber at very light loads, while the overall ratio was less than one for all the runs. These curves probably explain the rather flat combustion characteristic after the initial rapid pressure rise as fuel exits the prechamber and finds air in the main chamber.

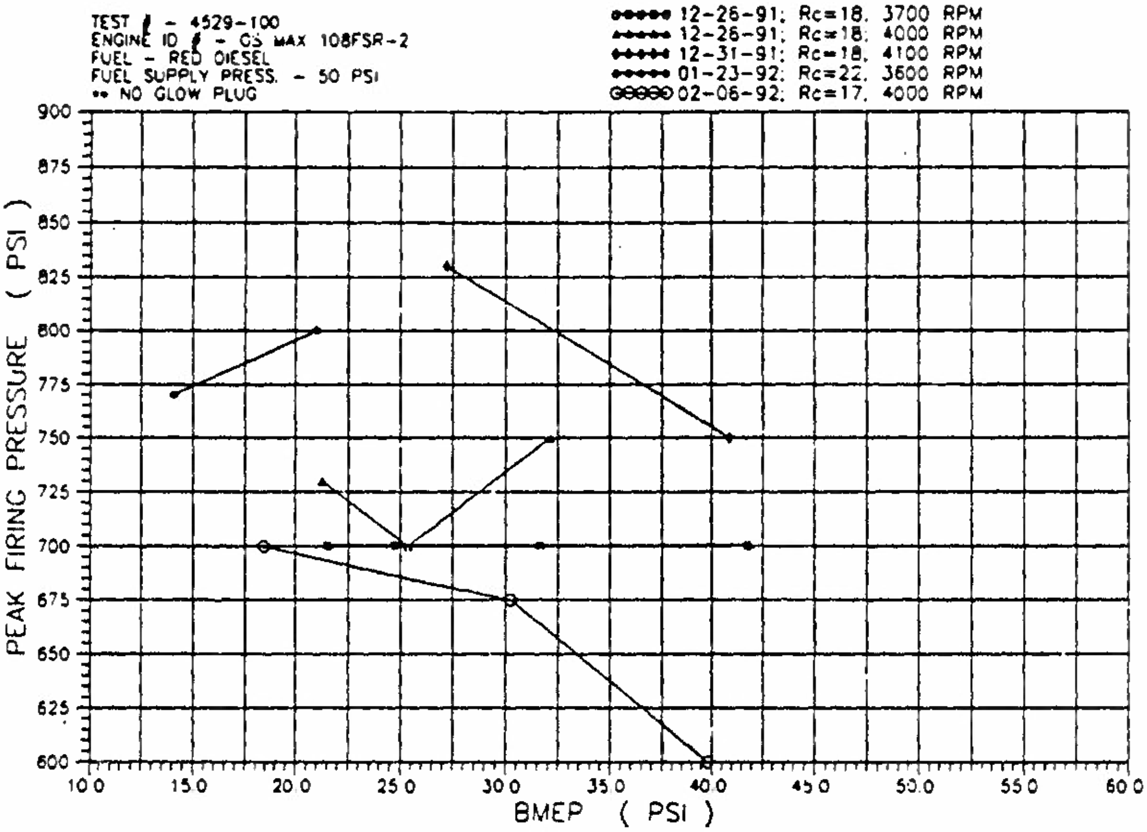


Figure 12. Peak Firing Pressure vs. Brake Mean-Effective Pressure

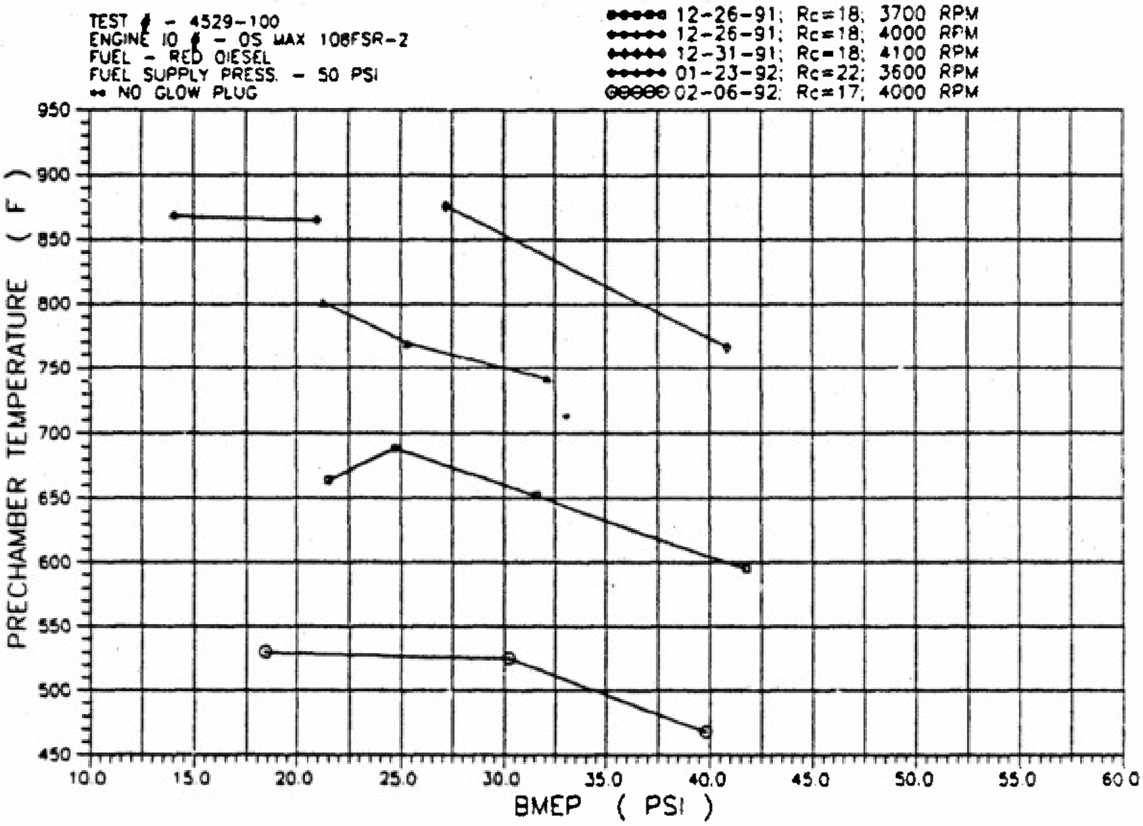


Figure 13. Pre-Chamber Temperature vs. Brake Mean-Effective Pressure

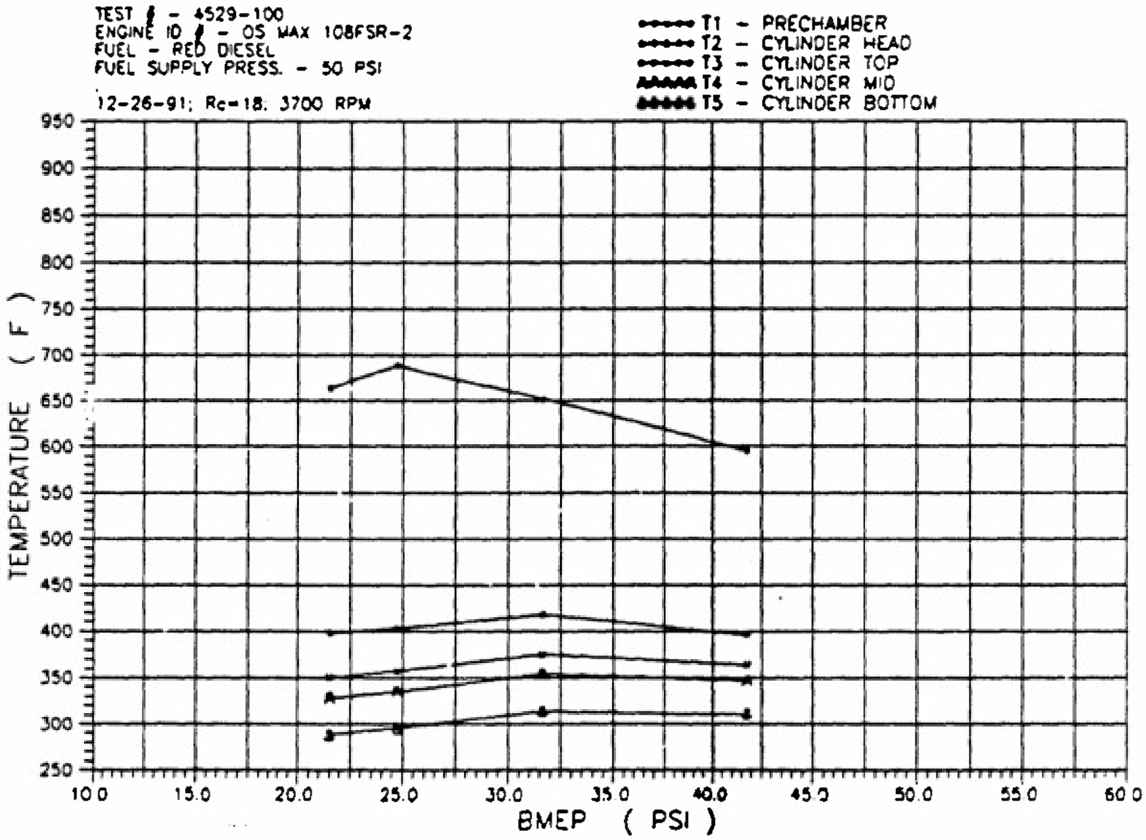


Figure 14. Engine Temperature vs. Power

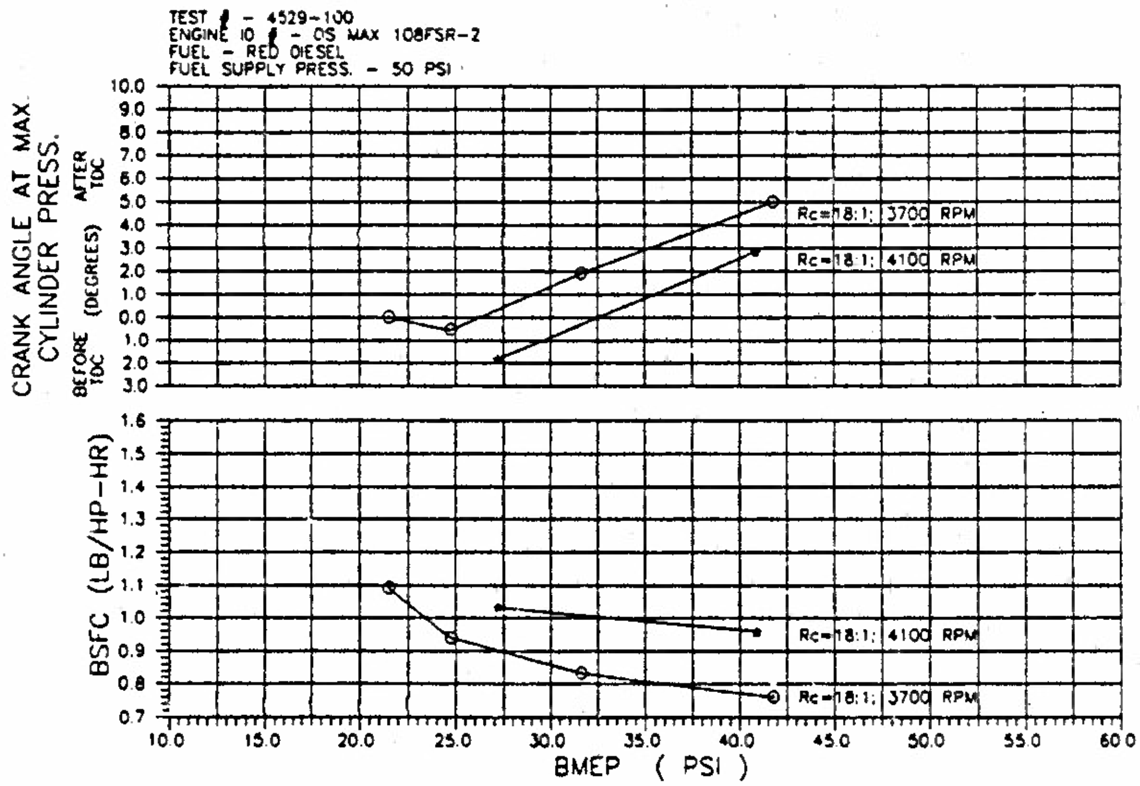


Figure 15. Brake Specific Fuel Consumption and Combustion Timing vs. Brake Mean-Effective Pressure

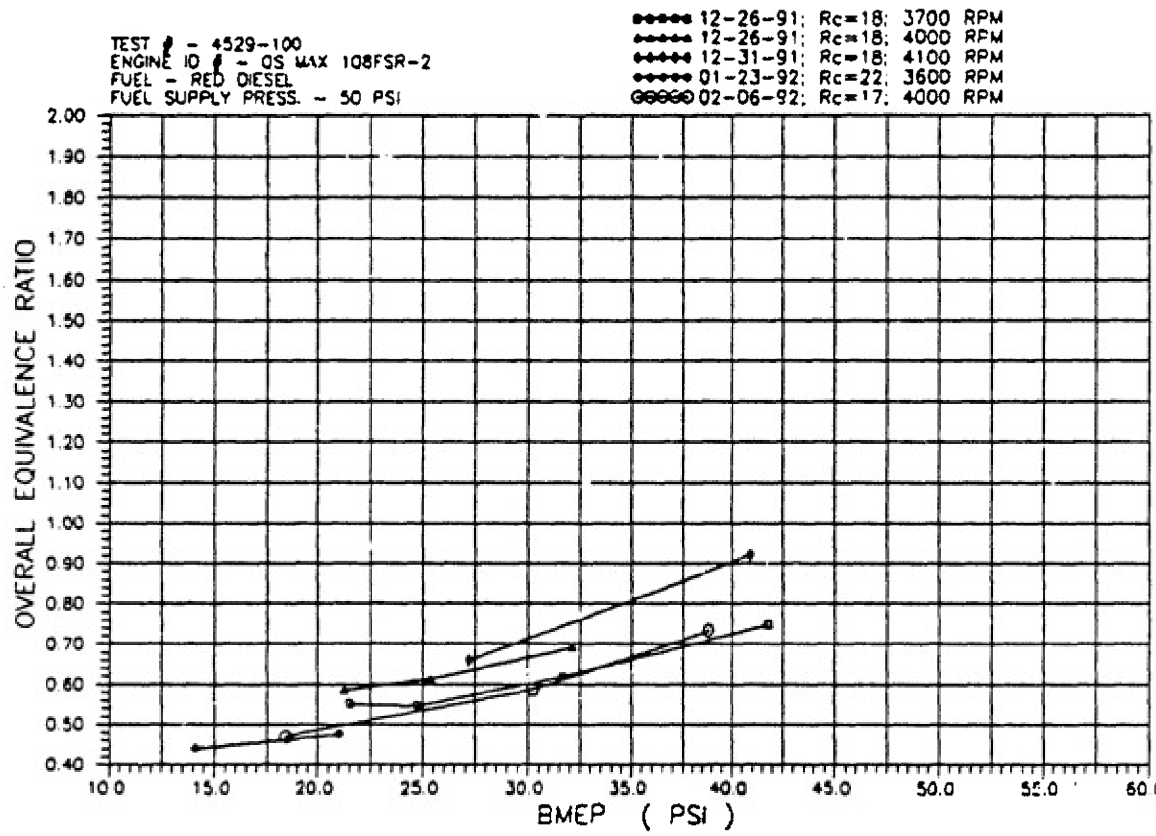


Figure 16. Calculated Overall Equivalence Ratio vs. Brake Mean-Effective Pressure

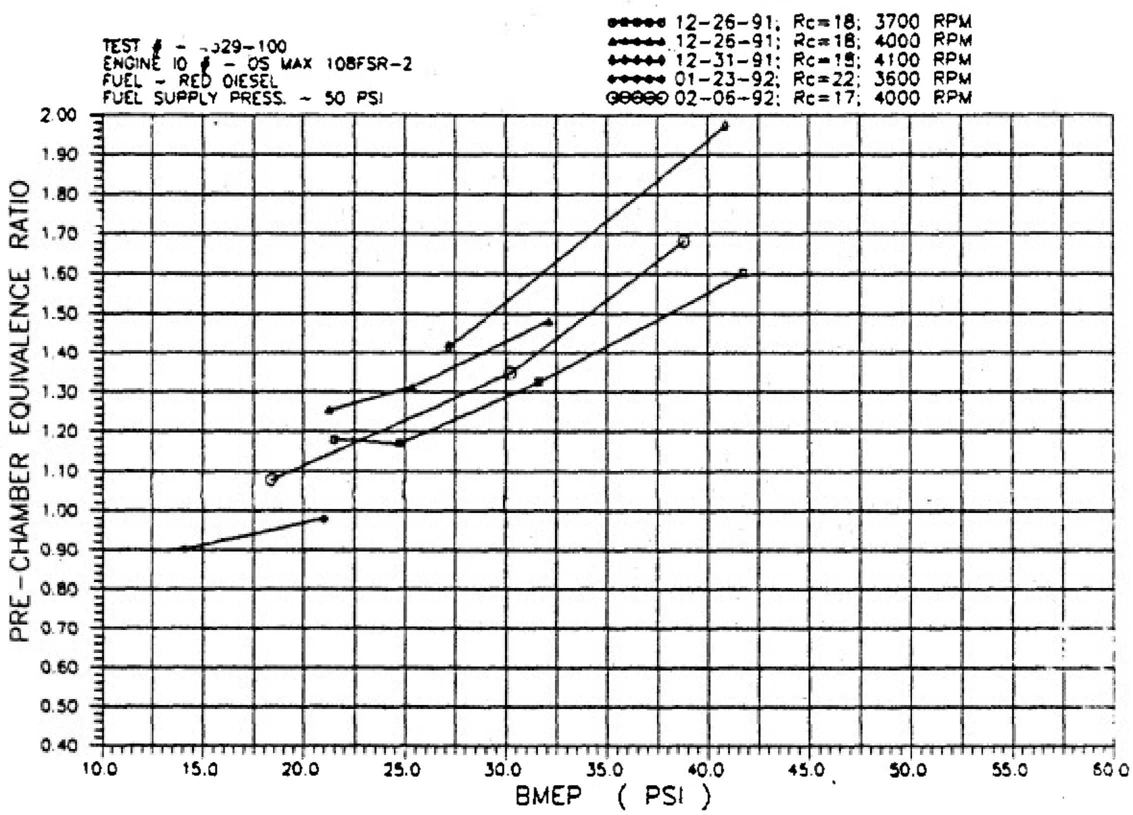


Figure 17. Calculated Pre-Chamber Equivalence Ratio vs. Brake Mean-Effective Pressure

Figures 18 through 20, which compare the pump diesel fuel (red diesel) and JP-8, show similar results. The engine began to fire at about the same pre-chamber temperature (320°F) with JP-8 as with the pump diesel when cranked at 3600 rpm. As Figure 18 indicates, the JP-8 seemed to give more advanced combustion at 28 psi brake mean-effective pressure, and this may explain the higher brake specific fuel consumption. Otherwise the two fuels seemed to perform equally well.

Throughout the testing, the exhaust was observed to be very similar to conventional diesel engines. The white smoke when the engine was cold had the same acrid odor. Warmed up and under load, the exhaust was a very light grey at light loads and became darker as load was increased. The engine never produced any black smoke. Under load, the exhaust had the same characteristic aldehyde odor as larger engines.

The exhaust produced quantities of castor oil from the lube system which was black with carbon. There was no evidence of any significant quantities of liquid diesel fuel in the exhaust.

3.2 STARTING

The way in which the engine was started when measuring performance with the dynamometer has been described; essentially the engine was motored at 3600 rpm until the pre-chamber was hot enough to effect ignition. This section of the report describes our activities in manually starting the engine.

A recoil-type hand-cranking device was obtained from the U.S. Army Natick R,D&E Center and modified to fit the shaft at the free end of the motor where the dynamometer is normally connected (see Figure 6). The cranking mechanism was attached to the free end of the starting motor in place of the dynamometer, as shown in Figure 21. Cranking tests were performed with the compression ratio set at its initial value of 18 to 1. With the engine clean (all deposits removed and the prechamber cleaned in an ultrasonic device), the compression pressure was 410 psi. The engine would not start without the glow plug and the prechamber had to be heated to 450°F before the engine would fire. After allowing the engine to idle for a few minutes, it was shut down. It was then repeatedly restarted without the glow plug as it cooled off, and the engine was restarted manually down to ambient temperature.

After these initial runs, the engine would start consistently from cold with a few pulls and without the glow plug. The compression pressure had increased to 530 psi, presumably from the effect of deposits. The predicted compression pressure at 18 to 1 is 480 psi (see Figure 1).

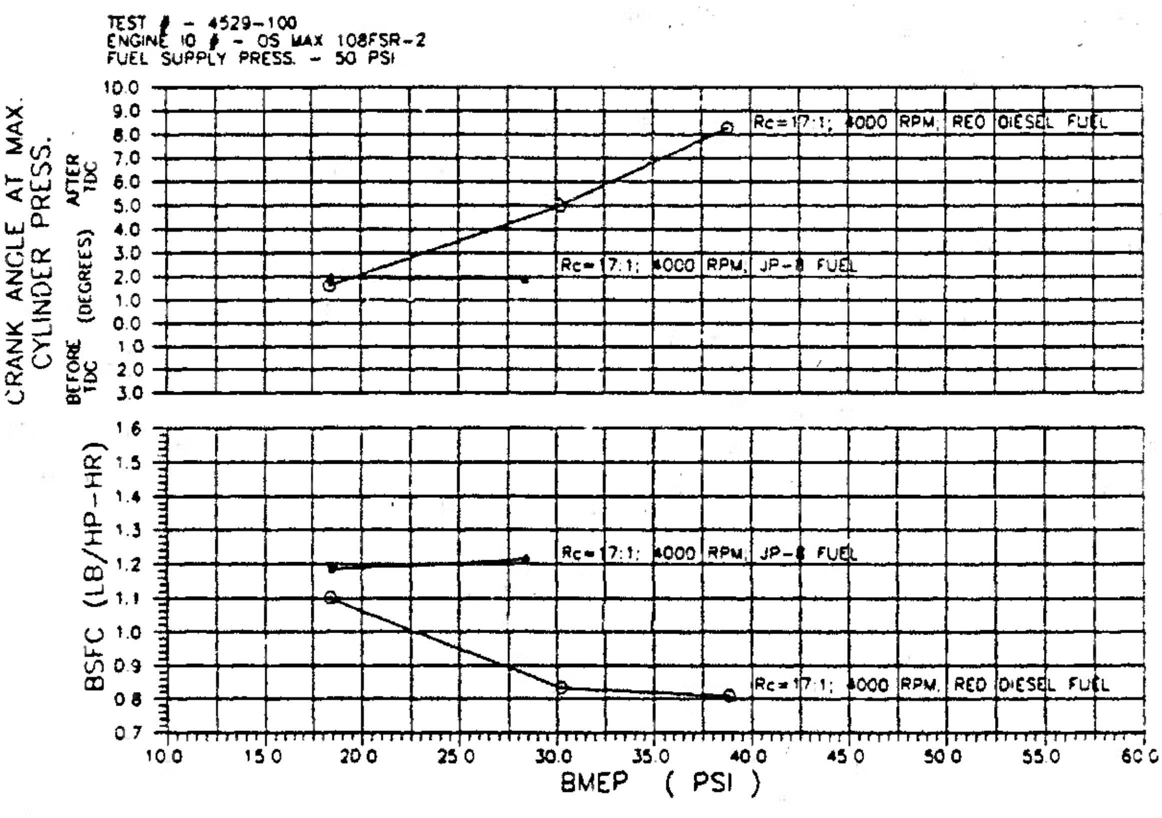


Figure 18. Brake Specific Fuel Consumption and Combustion Timing vs. Brake Mean-Effective Pressure (Comparison of Different Fuels)

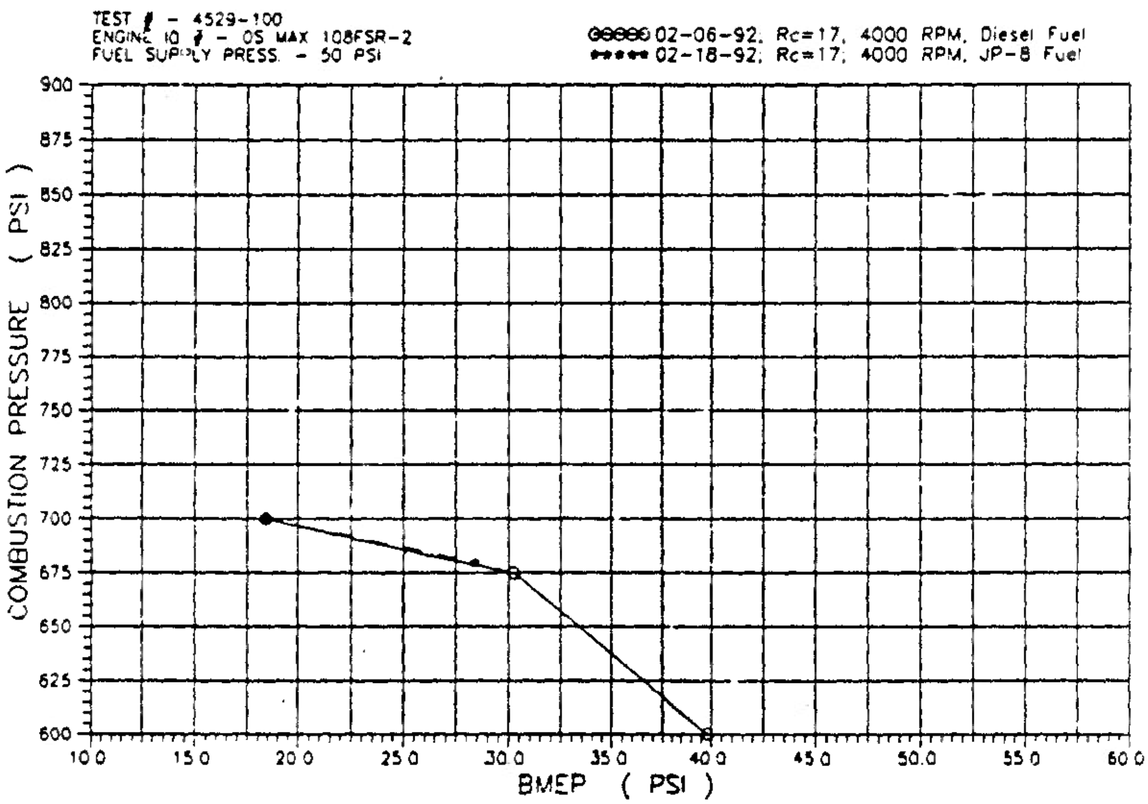


Figure 19. Combustion Pressure vs. Brake Mean-Effective Pressure (Comparison of Different Fuels)

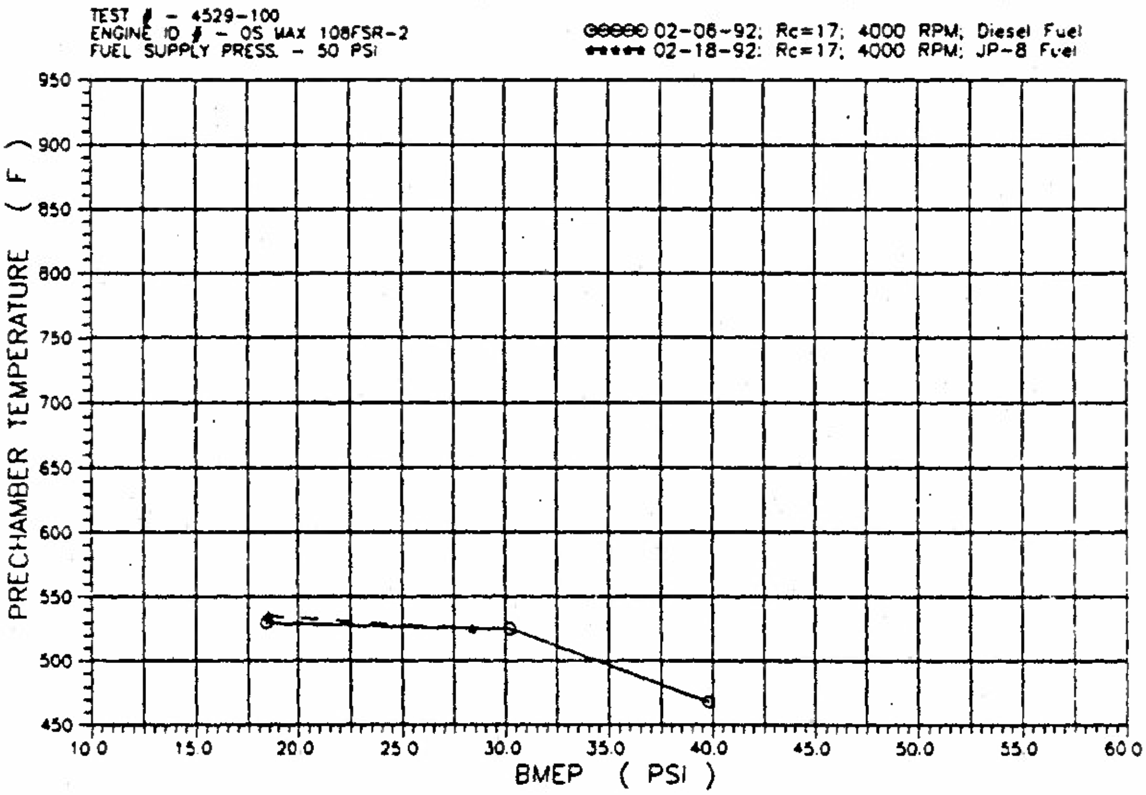


Figure 20. Pre-Chamber Temperature vs. Brake Mean-Effective Pressure (Comparison of Different Fuels)

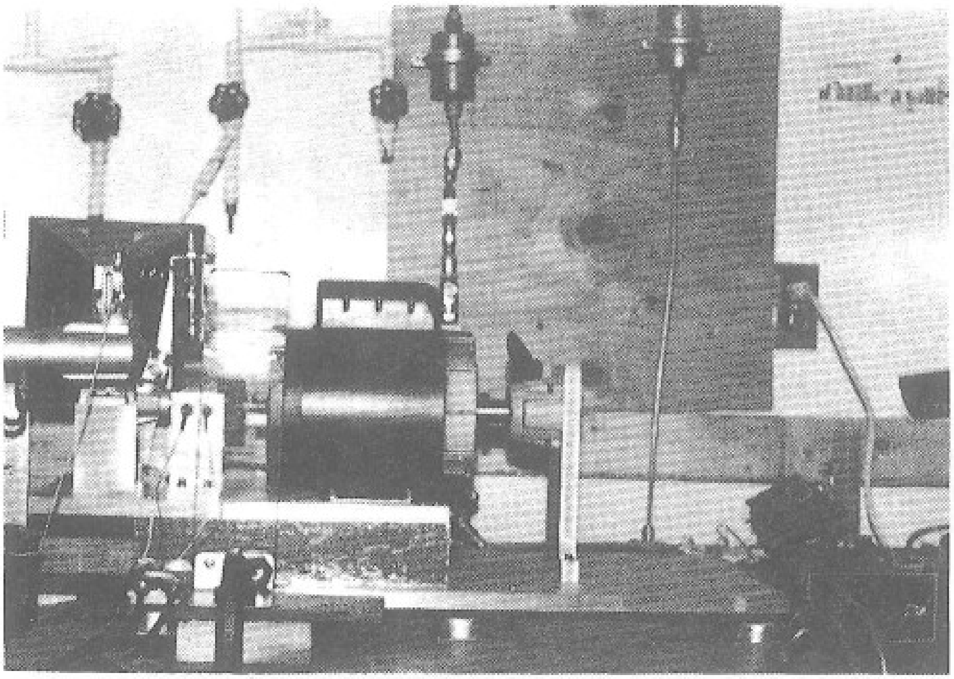


Figure 21. Test Facility Showing Hand-Cranking Device

When started by hand, the engine idles at 1200 ± rpm but is extremely sensitive to fuel flow. Any increase or decrease in fuel flow will cause the engine to stall. Combustion is obviously very poor as evidenced by excessive grey-black smoke. If the speed is increased to 3600 rpm with the motor, the engine will run as before but the engine cannot boot-strap itself to that speed. This behavior is due to poor atomization and could presumably be overcome by reducing the throat size between the main chamber and the pre-chamber to increase air velocity. This would aggravate the tendency we already have observed for the engine to advance itself at higher speeds.

When idling at 3600 rpm, the engine speed can be decreased to about 1800 rpm and it will reaccelerate. Below 1800 rpm, the engine simply drops to ± 1200 rpm and will not reaccelerate. It is clear that this phenomenon is related to the simple untimed fuel injection system, which will be one of the main developmental items in the next step of the program.

The compression ratio was raised to 22 to 1 to determine its effect on starting and performance. This compression ratio was selected because the analysis program indicated it would give the 530-psi compression pressure that had already resulted in consistently unaided hand starting. Unfortunately, the performance runs were made prior to the starting tests. While the compression pressure was about 530 psi with the engine clean, it had dropped considerably by the end of the two runs shown in Figure 11. When the dynamometer was removed and the hand-crank device was installed, the compression pressure had dropped to around 400 psi and the engine would not start. The engine was torn down for examination and found to have a scuffed piston with the piston ring stuck in its groove.

Some idling tests had been carried out at 22 to 1 and indicated the same basic behavior as at 18; namely, the tendency of the engine to sit at one speed (1200 rpm) independent of fuel flow and to refuse to accelerate. It appeared not to stall as readily when the fuel flow was increased beyond that required to idle.

3.3 ENGINE DURABILITY

The O.S. engine proved to be more than adequate as a test bed for the feasibility tests that were the objective of this program. Approximately 12 hours of firing operation were obtained on two engine assemblies.

The first engine failure occurred at slightly more than four hours of running time. The mode of failure was crank pin bearing seizure. Since the electric motor was energized at the time, it continued to turn the engine after the bearing seized, and this did considerable damage to the connecting rod. Some damage was also done to the keyway in the engine crankshaft. The damaged rod is shown with the engine in Figure 22.

A second failure occurred in the second engine after 3 hours and 50 minutes. This was the scuffed piston and liner referred to in the previous section where a drop in compression pressure was observed. The engine had been disassembled and examined when the compression ratio was increased to 22, and the piston and liner were in good condition at that point. We know, therefore, that the scuffing occurred during the one hour of tests at 22 to 1. Figures 12 and 13 indicate that firing pressure and pre-chamber temperature were relatively high during these runs as compared to all but the relatively short run time at 4100 rpm and 18 to 1 compression ratio.

Figure 23 shows the scuffed piston. Both thrust (top photo) and anti-thrust sides were scuffed and the piston ring was jammed in its groove. Apparently the piston overheated and began to seize in the bore. There was castor oil in the crankcase when the engine was disassembled. It may be that cylinder head temperatures in the 450°F range are excessive for this engine. When work is continued, some experiments with an instrumented engine operating as a conventional glow plug engine could be performed to determine what cylinder head temperatures are typical for this engine and how much they can be increased before the piston will scuff.

Another interesting phenomenon was observed with the piston. What appeared to be a crack in the skirt in the first piston turned out to be a corroded area in the region bordering the unloaded and loaded areas of the skirt. This "crack" appeared when carbon was cleaned from the area. This is shown in Figure 24 at 10x magnification. Apparently, the products of combustion that collected in the unloaded area of the skirt corroded the piston over a period of days. A similar attack was seen in the unloaded areas of the wrist pin bores.

The only component of the engine known to be inadequate at this point is the crankpin bearing. For further testing, a titanium connecting rod with a rolling element bearing could be utilized (these are commercially available). The alternative of increasing the bearing size would require a crankcase modification.

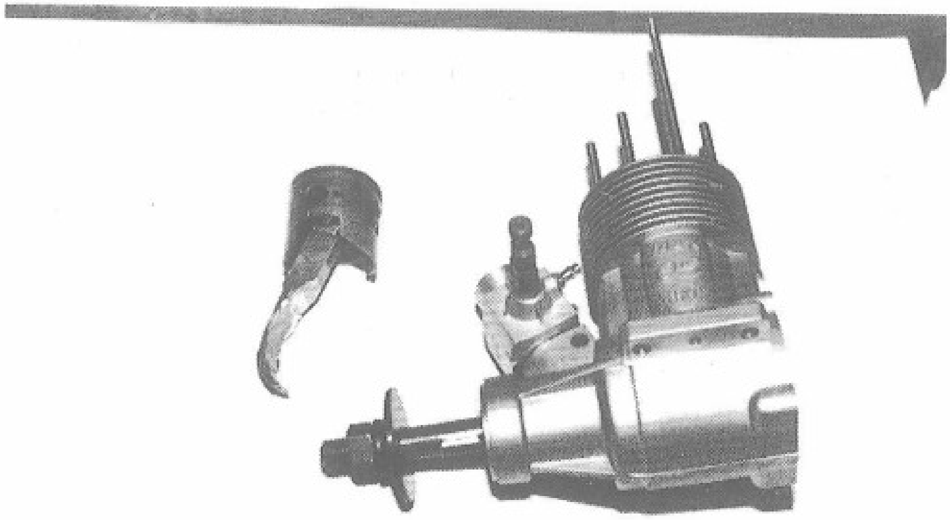
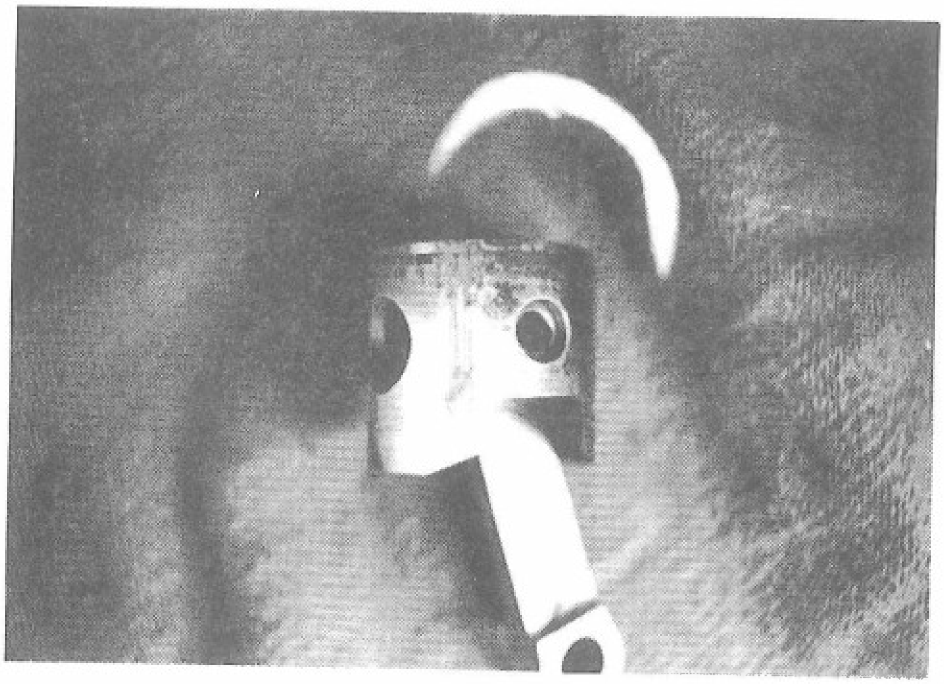
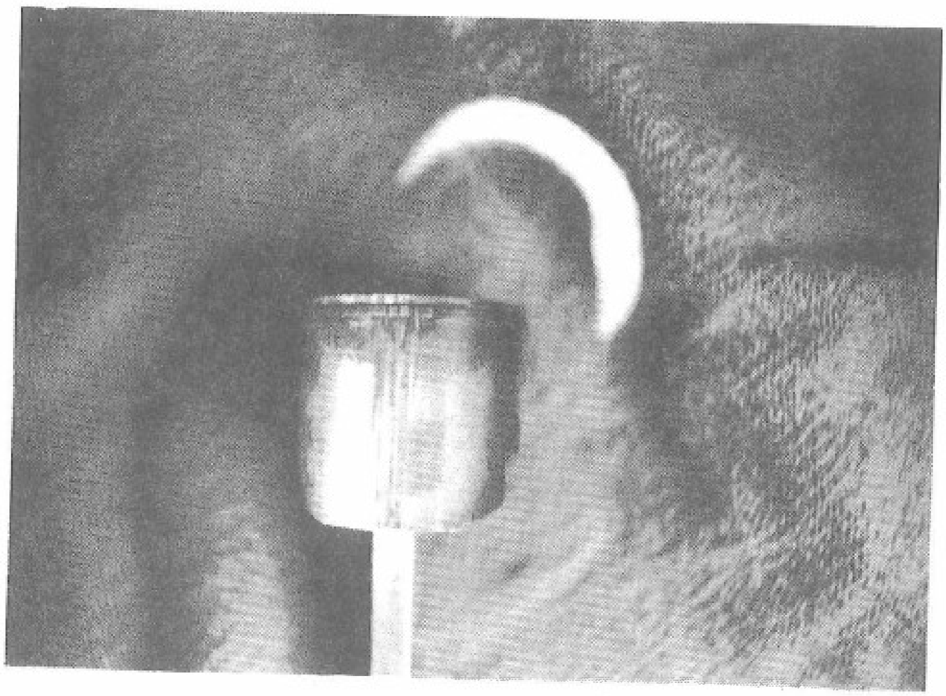


Figure 22. Failed Connecting Rod



Thrust Side



Anti-Thrust Side

Figure 23. Scuffed Piston Assembly

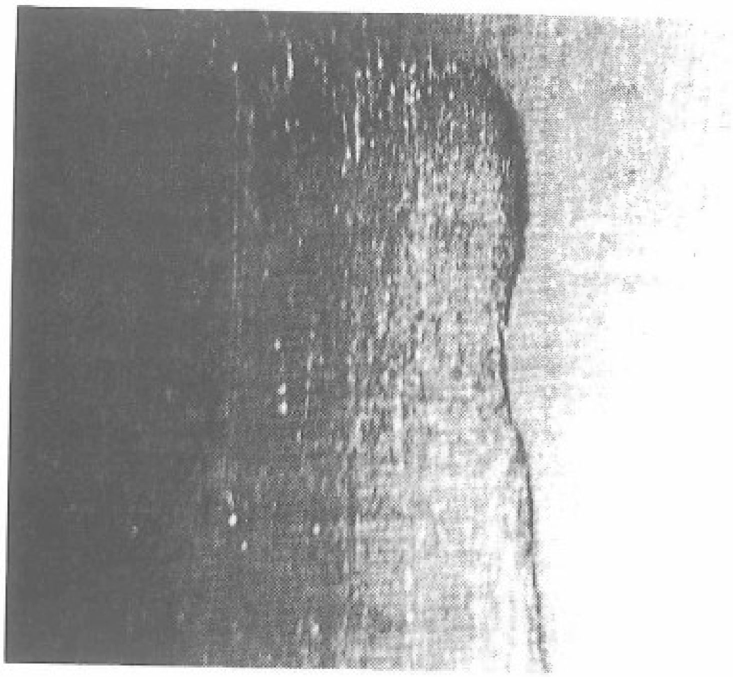


Figure 24. Corrosion Damage on Piston

3.4 ENGINE WEIGHT AND NOISE

The stock O.S. engine used in these tests weighs 26.5 ounces without the muffler. The engine with modified cylinder head weighs 29.3 ounces. The muffler used in these tests weighs 8.6 ounces, giving a total weight less flywheel and hand-cranking mechanism of 37.8 ounces, or almost 6 ounces heavier than the stated goal of two pounds. The hand-cranking system used for these tests would add about 8 ounces to the weight of the engine. The flywheel used in these tests was not representative of what might be required in an actual system. That flywheel's design characteristics would be determined by the compressor's inertia, the method of coupling, etc.

The stock muffler does not provide adequate sound attenuation at the speeds of these tests. When operated as a conventional glow plug engine at circa 13,000 rpm, the engine seems more quiet despite the much higher power output (~3.5 bhp). At this point, it is difficult to say whether a muffler designed for the 4,000 rpm condition would be heavier or lighter than the one used here.

What is important here is not only engine weight but also engine efficiency since that influences the weight of fuel the soldier must carry for a specific mission. The specific fuel consumption reported here (0.75 lb/bhp-hr) is somewhat better than the value (1.0 lb/bhp-hr) used in arriving at the two-pound engine weight goal. A savings of 0.1 lb of fuel per hour could allow an extra 9.6 ounces of engine weight for a 6-hour mission.

Other ways in which the engine weight might be reduced are as follows:

- Use a smaller displacement engine at higher speed. If this involves a speed change to the compressor then the weight of that system must be added in. The encouraging results regarding combustion indicate higher speeds are practical, particularly with a timed injection system.

- Substitute titanium for aluminum while using thinner sections. This is not likely to save more than a few ounces.

- Eliminate the throttle valve in the carburetor since it is not necessary (probably about 1 ounce).

- A design study would be required, but liquid cooling might reduce the overall weight provided the coolant and heat exchange area did not exceed the savings in eliminating the fins.

4. CONCLUSIONS AND RECOMMENDATIONS

4.1 SUMMARY

Sections 1.2 and 1.3 provide a concise statement of the technical objectives and results of this feasibility program. The program has shown that an insulated pre-chamber type of reciprocating engine operating as a purely compression-ignition engine will start by hand from ambient conditions without aids and run efficiently at the required speed and load with good overall efficiency, despite a rather crude fuel injection system.

The program has also shown that a commercially available miniature engine is suitable as a development tool for combustion system optimization and starting and lubrication studies. The engine used here was only about 20 percent heavier than the stated goal of two pounds.

The success with this concept to date would appear to justify further development. The results presented here indicate the need for a timed fuel injection system to achieve optimum combustion timing and the flexibility to provide characteristics needed for starting and coming up to full load. Once a timed injection system has been developed, then the entire combustion system can be optimized.

With an optimized combustion system in hand, the issues of lubrication, cooling, noise, weight, and engine durability can be addressed. As the development program progresses, the overall engine characteristics will evolve. This requires a "prototype" engine design to ensure that overall goals are kept on track.

The specific technical recommendations are outlined in the following sections.

4.2 COMBUSTION SYSTEM REFINEMENT

4.2.1 Injection System Refinement

The present injection system consists of a simple ball check valve and 0.012-in. orifice located adjacent to the pre-chamber as described in Section 2.2.3. Fuel pressure is increased until the engine begins to fire, and the power level is fine-tuned with a needle valve a short distance upstream of the check valve in the fuel line. This system results in uncontrolled injection timing and poor fuel atomization, as discussed in Section 3. Its use here was to prove that compression ignition of diesel fuel can be established with an insulated pre-chamber for a small high-speed engine.

A preliminary design for a timed injection system is shown in Figure 25. In this design, the injector plunger is actuated by the piston at 15 to 20 degrees before top dead center where the piston velocity at 4,000 rpm is only a few feet per second. The plunger would follow the piston motion through the action of a spring. On the compression stroke, the piston would contact the plunger and move its ports past the cut-off ports to the fuel supply system maintained at relatively low pressure. The trapped fuel would then be pumped through the delivery check valve located at the plunger pump outlet to the orifice at the pre-chamber. Removing the check valve to the pump side should be beneficial in reducing the tendency of the check valve to carbon-up due to heat transfer from the hot pre-chamber.

As the plunger retracts, the delivery valve would close under spring and differential pressure forces, and a partial vacuum would be produced above the plunger, which would cause fuel from the supply port to flow in through the clearances. Some fuel-flow control could probably be obtained by varying the fuel supply pressure, although precise fuel control over a range of flows is not particularly important in this application.

The timed system of Figure 25 is obviously much more complex than the proof-of-concept system used in the work reported here. Some fine-tuning of the timed system in the experimental stage would obviously be required. Variables such as plunger diameter, stroke, port timing, spring force, etc., would have to be explored in a motored engine.

At the stage of having a motored, timed injection engine, it would also be desirable to look at the atomization of the fuel in conjunction with the air motion in the pre-chamber. Variables such as the size and location of the injection nozzle should be explored as part of the combustion system optimization. Since piston speeds are not high in this engine, there are obvious tradeoffs between swirl velocity, heat loss, and atomization of the fuel in the pre-chamber.

Other timed injection systems would also be investigated. The design complexity, component size, and overall cost of the system are the important factors to be considered.

4.2.2 <u>Combustion Chamber Geometry Refinement</u>

The chamber geometry tested here and shown in Figure 5 was arrived at using a simple analytical tool described in Section 2.2.2. The only parameter varied in the course of the work described was compression ratio, and that over a fairly narrow range. Future development should include the evaluation of various changes in the combustion chamber geometry including:

- Compression ratio
- Pre-chamber/main chamber volume distribution
- Throat size between main chamber and pre-chamber

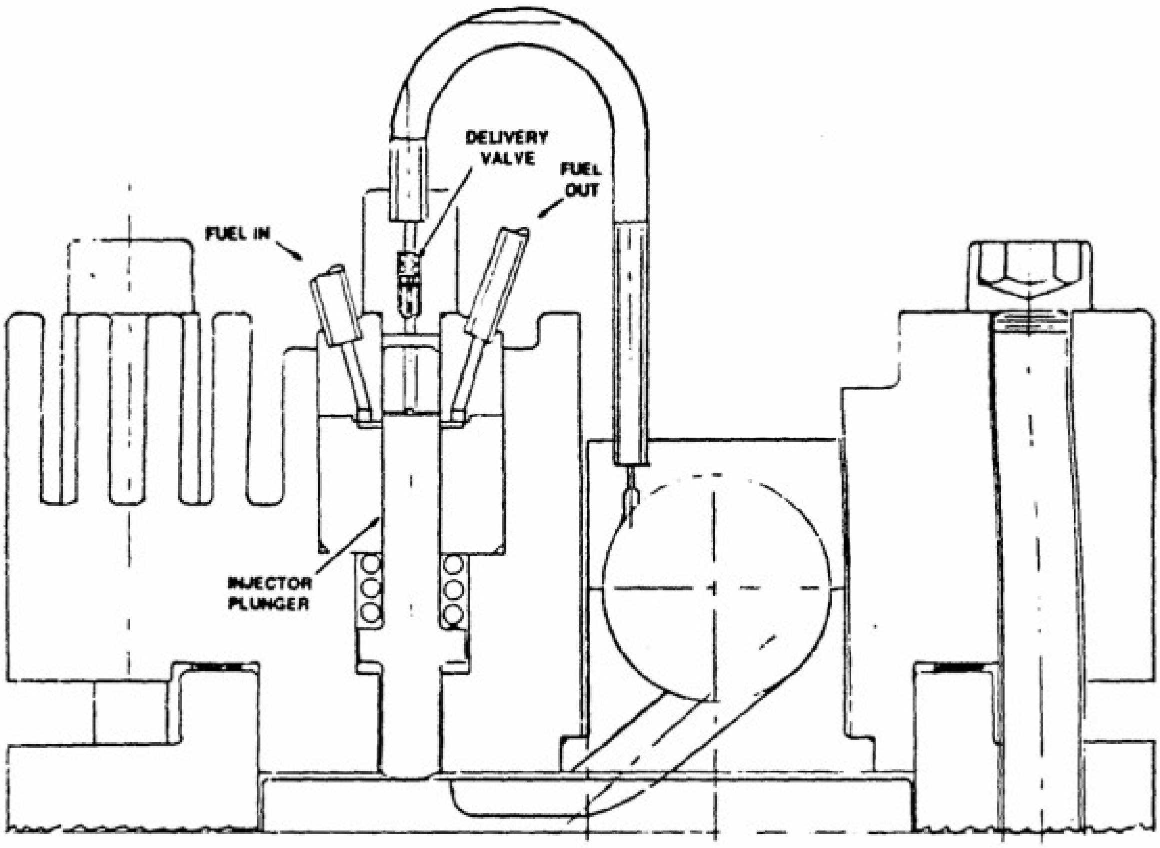


Figure 25. Experimental Cylinder Head, Timed Injection

This document reports research undertaken at the US Army Natick Research, Development and Engineering Center and has been assigned No. NATICK/TR-92/036 in the series of reports approved for publication.

There should be some attempt made to achieve better mixing in the main chamber in any future work. Figure 25 shows a new piston design with tighter piston-to-head clearance than that of Figure 5, and a bowl at the exit of the pre-chamber.

Testing to date indicates that some improvement in insulating the pre-chamber from the remainder of the cylinder head is desirable. At the present time, the pre-chamber is insulated by an air gap, as shown in Figure 5, with a solid conduction path through the flange which holds it in place in the cylinder head. Various techniques for reducing heat transfer from the pre-chamber to the head should be investigated. These could include ceramic coatings, intermediate low-conductivity inserts, etc.

4.3 STARTING

The demonstrated ability to hand start the cold engine without aids indicates that this should remain the primary approach to starting. There are probably some tradeoffs to be made between starting ability, full-load running, and durability of the engine, especially with regard to compression ratio. Most diesel engines have a higher compression ratio than is optimum for efficiency and durability in order to achieve good starting characteristics. This engine is not likely to be different in that regard.

One other phenomenon observed in this program, which is worth following up, is the tendency for a dirty engine (i.e., one which has been run) to hand start easier than a clean engine. Compression pressure does not seem to be a major factor here, but deposits could reduce heat loss and result in higher compression temperatures. This indicates that some type of thermal coating on the interior of the pre-chamber wall could be beneficial.

4.4 LUBRICATION

The engine is currently lubricated by allowing a small amount of castor oil (its normal lubricant) to enter the crankcase through the carburetor needle valve. Castor oil has a relatively high oxidation temperature as compared to petroleum base oils or diesel fuel. The preferred approach to lubrication in this application would be to use diesel fuel (or JP-8), which would eliminate the logistical problem of supplying another fluid in the field.

Tests to date show that with the crude injection system currently being used, some unburned diesel fuel finds its way to the crankcase. This raises the possibility of not having to introduce the fuel separately to the crankcase.

If the diesel fuel approach does not work and supplying a more suitable lubricant is not logistically feasible, then a separately lubricated sump would have to be designed. This would presumably reduce the consumption of a second fluid, but would not eliminate its presence. Virtually any engine in use by the military requires the presence of at least two fluids (fuel and lubricant), and in many cases, three (coolant).

4.5 PROTOTYPE ENGINE DESIGN

The purpose of this effort would be to keep a running record of the ultimate engine configuration as the experimental program progresses. The present status of this effort is still relatively primitive, since none of the major issues have been completely resolved. The present prototype design is basically a stock O.S. engine with the cylinder head shown in Figure 5, less the glow plug.

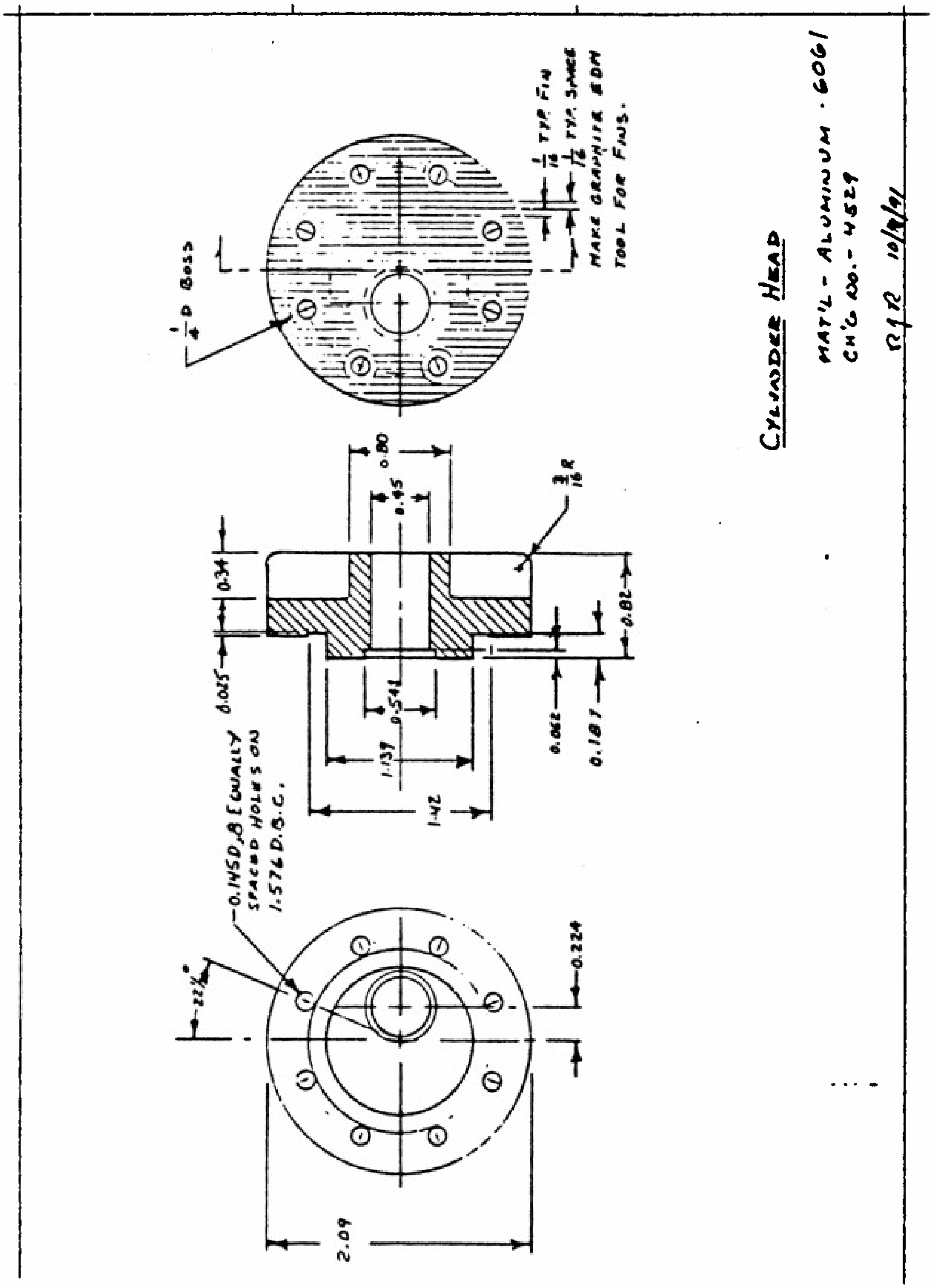
As the issues begin to be resolved (starting, lubrication, cylinder head geometry, fuel supply, etc.), the engine system configuration must be made compatible with its application or mission. The prototype engine design effort would begin to look at how the engine would be integrated with the microclimate cooling system. Some aspects of the system – such as starting and cooling of the engine – cannot be solved without considering the overall cooling system. For example, rejecting the engine heat through the condenser of the cooling system might be more effective than having a separate engine cooling system. Similarly, starting the engine requires some consideration of what happens to the compressor while the engine is being cranked.

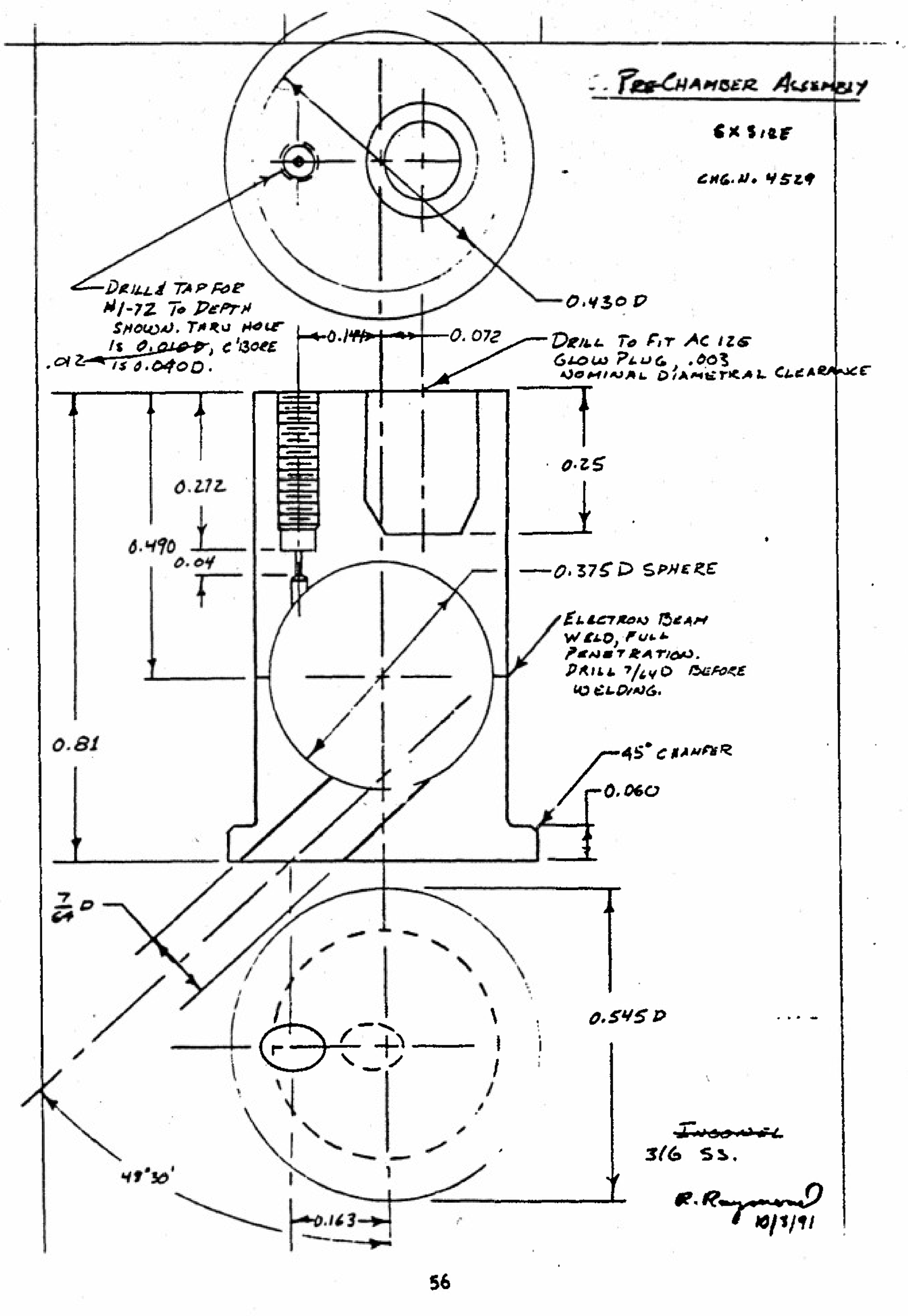
We feel that this feasibility demonstration program can evolve from the base of information presented here to provide a simple, cost-effective engine for the microclimate cooling system.

APPENDIX

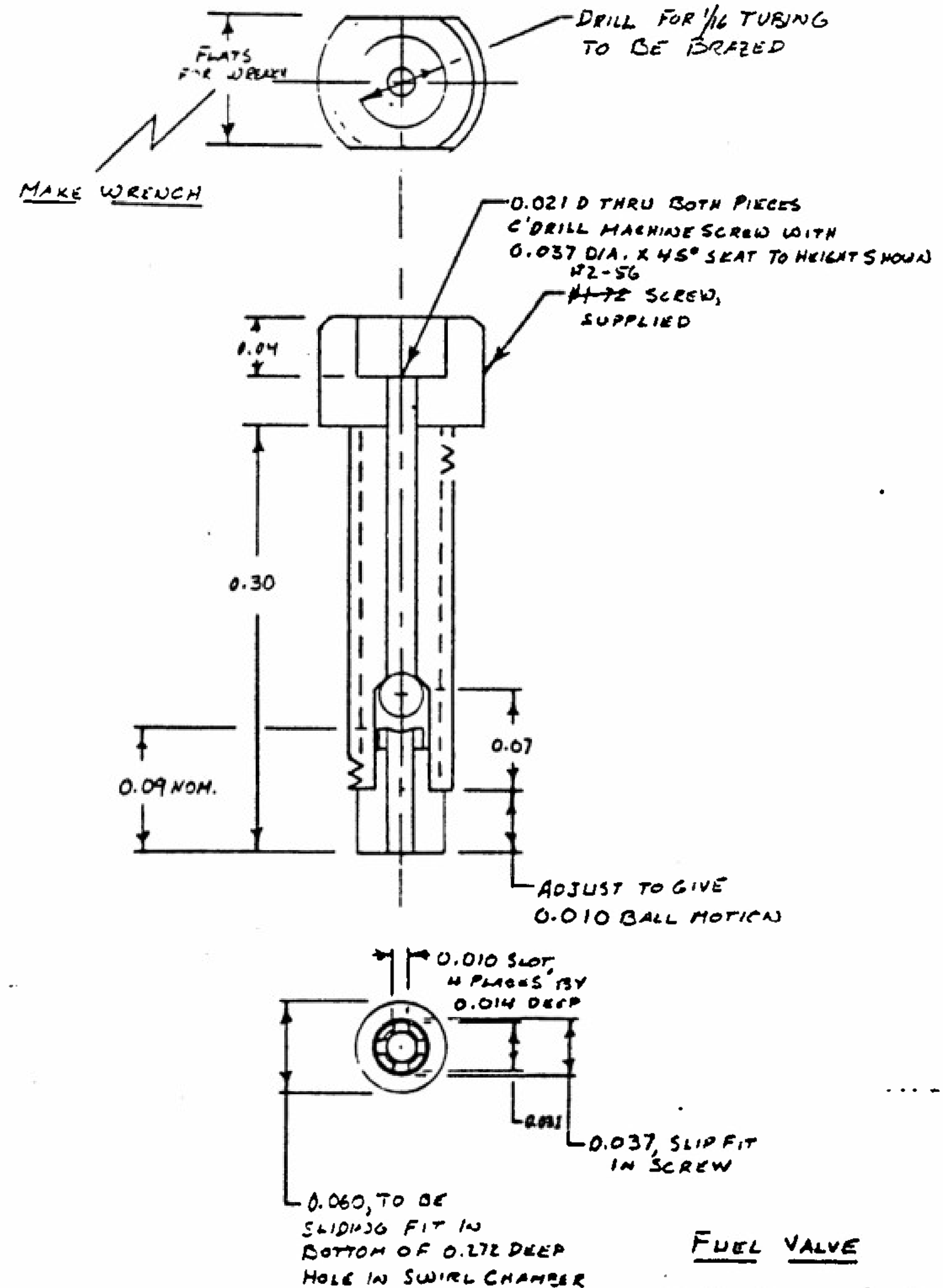
DRAWINGS

1. Cylinder Head
2. Pre-Chamber
3. Fuel Valve
4. Ball Check Design
5. Ball Check Design (Continued)


CYLINDER HEAD
MAT'L - ALUMINUM · 6061
CH'G NO. - 4529
RJR 10/8/91
1/16 TYP. FIN
1/16 TYP. SPACE
MAKE GRAPHITE EDM TOOL FOR FINS.
1/4 D BOSS
0.80
0.45
3/16 R
0.34
0.82
0.025
0.541
1.139
1.42
0.062
0.187
0.145 D, 8 EQUALLY SPACED HOLES ON 1.576 D.B.C.
22½°
0.224
2.09



. PRE-CHAMBER ASSEMBLY
6X SIZE
CHG. N. 4529
DRILL & TAP FOR
#1-72 TO DEPTH
SHOWN. THRU HOLE
IS ~~0.010D~~, C'BORE
.012
IS 0.040D.
0.430 D
0.144
0.072
DRILL TO FIT AC 126
GLOW PLUG, .003
NOMINAL DIAMETRAL CLEARANCE
0.25
0.272
0.490
0.04
0.375 D SPHERE
ELECTRON BEAM
WELD, FULL
PENETRATION.
DRILL 7/64 D BEFORE
WELDING.
0.81
45° CHAMFER
0.060
7/64 D
0.545 D
~~INCONEL~~
316 SS.
48°30'
0.163
R. Raymond
10/3/91



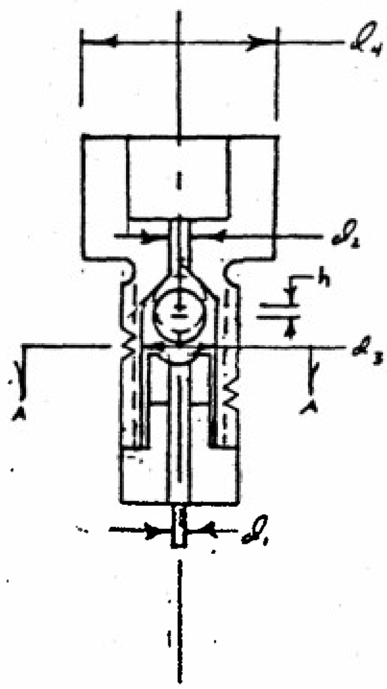
FUEL VALVE

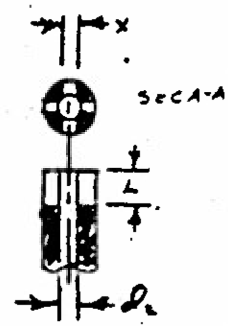
MAT'L - SCREW - SUPPLIED
BALL STOP - 304 SS

O J R 10/9/01

BALL CHECK DESIGN

- TO FIT INSIDE STANDARD UNF SCREWS (HEX HEAD)
- AREA OF d_2 TO BE 3X AREA OF NOZZLE (d_1)
- AREA OF d_3 - BALL SIZE TO BE 3X AREA OF NOZZLE





$d_2 = d_1 \sqrt{3}$

$d_3^2 = 3\, d_1^2 + d_{ball}^2$

$4 \times L = \frac{\pi\, d_2^2}{4}$

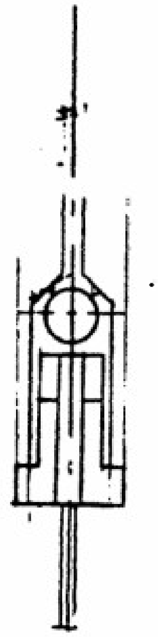
$d_{ball} = 0.031$

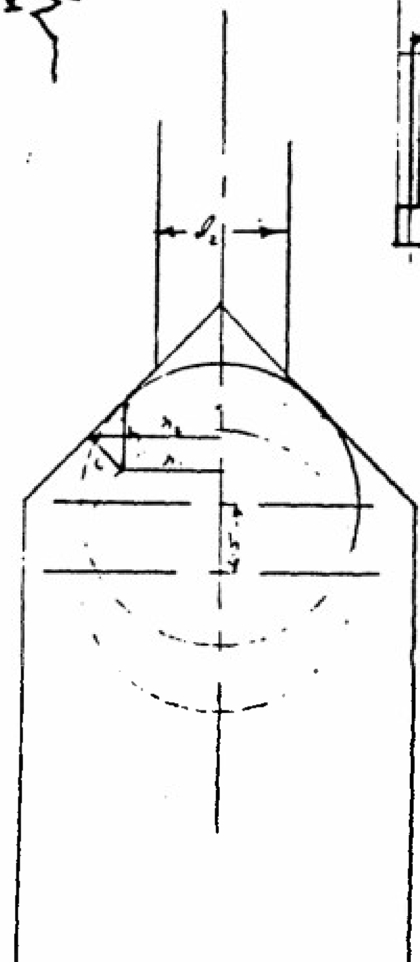
$x = 0.010$, $L = 0.009$ > $d_2 = 0.021$

SCREW SIZE	MAJOR DIA	MINOR DIA	d_1	d_2	d_3	d_4	h (BALL TRAVEL)	
0-80	.060	.045	.008	.014	.034	.096	.003	
"	"	"	.012	.021	.037	.096	.007	
1-72	.073	.056				.118		

$$A_t = 0.000154\ (0.014\ d_2)$$
$$= 0.000346\ (0.021\ d_2)$$

h	A_f
0.002	0.000102
0.004	0.000213
0.006	0.00033
0.008	0.00046
0.003	0.000156
0.007	0.000395





$$A_f = \frac{\pi\ d_2^2}{4}$$

$$c = h \cos 45°$$

$$r_1 = r_b \sin 45°$$

$$r_2 = r_1 + c \sin 45°$$

$$A_f = \frac{1}{2}\ \pi\ c(2)\ [r_1 + r_2]$$

$$A_f = \pi\ h \cos 45°\ [r_b \sin 45° + r_1 + c \sin 45°]$$

$$A_f = \pi\ h \cos 45°\ [2\ r_b \sin 45° + h \sin 45 \cos 45°]$$

$$A_f = \pi\ h \cos 45°\ [d_b \sin 45° + h \sin 45 \cos 45°]$$

$$A_f = 2.22\ h\ [0.022 + h.5]$$

1 (2)