Effects of Ignition Timing, Equivalence Ratio and Compression Ratio on RDH Engine Performance

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Procedure:

Our research examines the effects of spark timing, equivalence ratio and compression ratio on the performance of a single-cylinder research engine. The engine used was the spark-ignited, removable dome head (RDH) version of the Coordinated Fuels Research (CFR) engine. This engine is unique in several ways. First, the compression ratio is variable. For this research the compression ratio was varied from 5:1 to 9:1. Second, the engine was fitted with an in-cylinder pressure transducer. And optical shaft encoder in order to permit measurement of indicated performance parameters.

The engine was connected to a dynamometer that held the engine speed at 1800 RPM. The engine ran at wide-open throttle and fuel was metered by means of a single fuel injector in the intake manifold. We used 99-octane fuel, treated with an anti-knock additive. The high-octane fuel and additive were necessary due to the poor knock characteristics of this engine.

Various sensors were attached to the engine. Key sensors for acquiring data were: pressure transducer in the cylinder, airflow rate sensor, exhaust temperature sensor and crank angle encoder. In order to account for cycle-to-cycle variation, a pressure signal processing system (the Indiskop) averaged the data over 128 engine cycles. Figure 1 presents a line diagram of the experimental setup.

The effects of spark timing, equivalence ratio and compression ratio on performance were measured by varying one of these three conditions, while holding the other two constant.

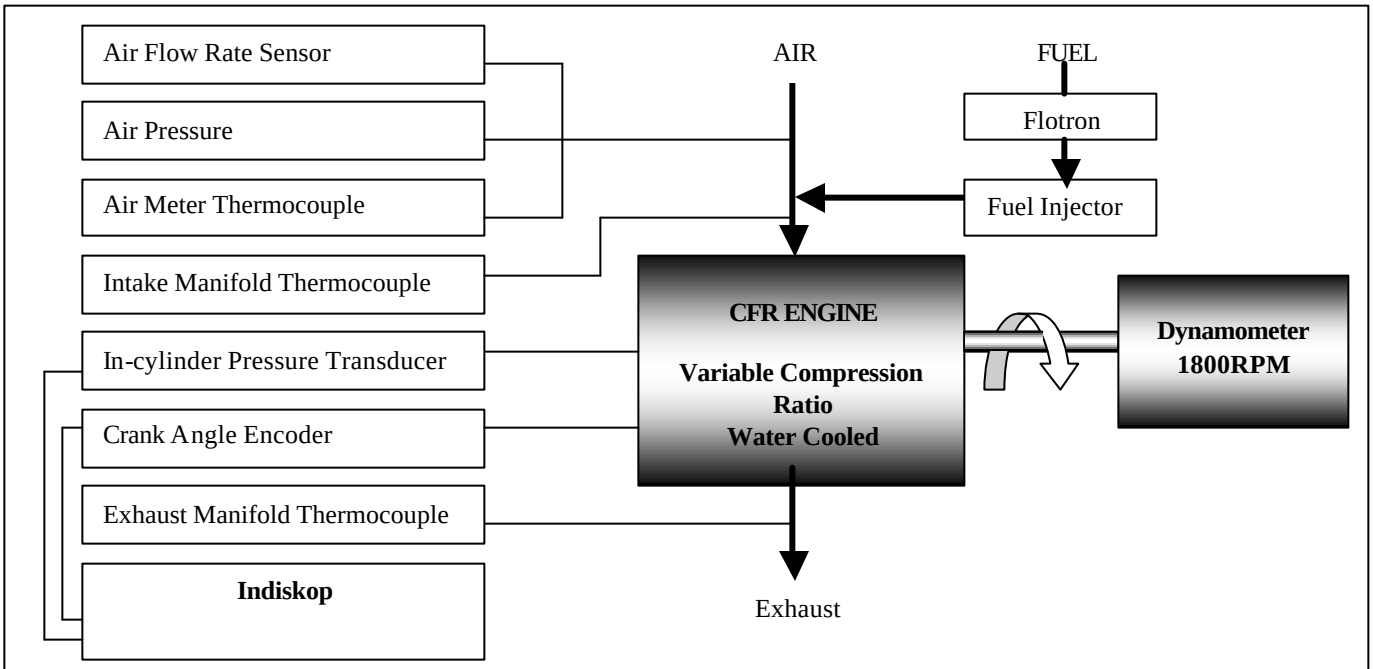

Figure 1: Line diagram of experimental setup

Results and Discussion:

For the first series of data the spark timing was varied from 16 to 34 degrees before top dead center (?BTDC). The compression ratio was held constant at 7:1 and the equivalence ratio fixed at one.

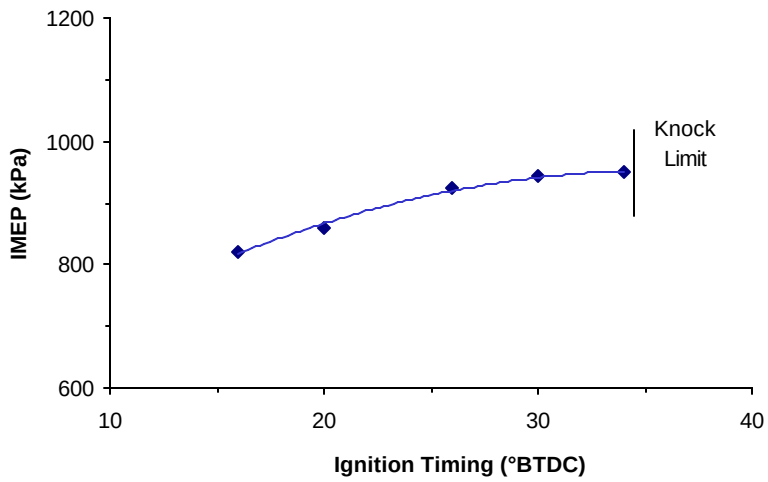

Figure 2: The relationship between IMEP and Ignition Timing. Wide open-throttle; Compression ratio of 7:1; Equivalence ratio of one.

The results show that IMEP tends to increase with spark advance between 15 and 35?BTDC. It is expected that IMEP should increase with spark advance to a point, and then drop off. Best performance will be achieved when the greatest portion of the combustion takes place near top dead center. If the spark is not advanced enough, the piston will already be moving down when much of the combustion takes place. In this case we loose the ability to expand this portion of the gas through the full range, decreasing performance. If the spark is too advanced, too much of the gas will burn while the piston is still rising. The work that must be done to compress this gas will decrease the net work produced. These competing effects cause there to be a maximum in the IMEP as a function of spark advance.

In our experiment we were unable to observe this maximum. Our test engine has very poor knock characteristics—primarily as a result of poor combustion chamber design. As the gas burns, pressure increases in the cylinder. The rising piston increases the cylinder pressure as well. These two events happen simultaneously at far advanced spark timing, combining to cause very large increases in pressure that contributes to knock. Consequently, as the spark advance was increased, knock was detected on the indiskop. Excessive knock was reached before the maximum could be determined. Figure 2 does, however, appear to be approaching a maximum.

The second-order polynomial fitted to the data has a maximum IMEP at an ignition timing of 35.9 ?BTDC. Minimum advance for best torque (MBT) is defined as the smallest advance that achieves 99% of the maximum power. Using the equation of the trend line, we obtained a value of 30.6 ?BTDC for MBT under these conditions.

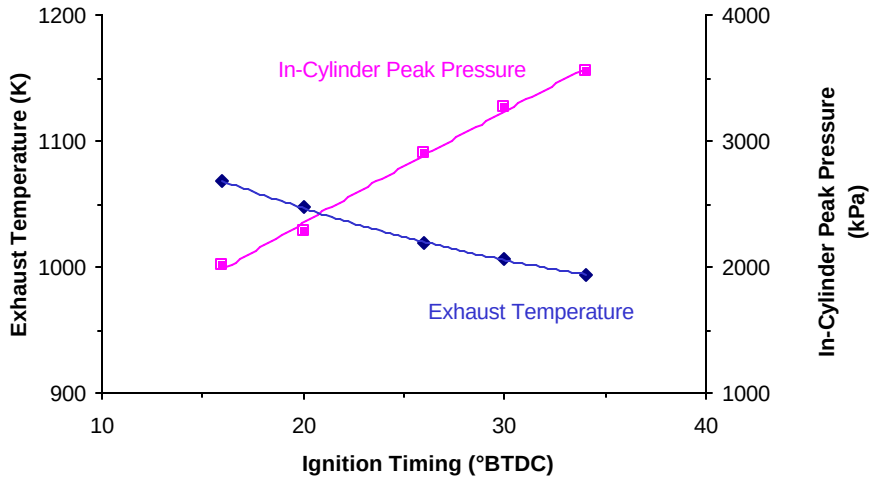

Figure 3: The relationship between Exhaust Temperature and In-cylinder Peak Pressure versus Ignition Timing. Wide open-throttle; Compression ratio of 7:1; Equivalence ratio of one

As is evident in Figure 3, the peak pressure increases with increasing spark advance. Maximum pressure would be reached if all of the gas were burned by the time the piston reached TDC. With less advanced spark timing the gas does not burn completely until the piston is on it's way down on the expansion stroke. As spark advance is increased we come closer to the maximum.

Figure 3 also shows that the exhaust temperature decreases with increasing spark timing between 15 and 35?BTDC. This is interesting when related to the IMEP over this range. The IMEP represents the work done on the piston. The exhaust gas temperature represents the enthalpy of the exhaust gas—since for ideal gases the enthalpy is a function of temperature only. The energy released by the combustion of the fuel must go into expansion work or the enthalpy of the exhaust gas. Since the IMEP (work) increases over the range studied, the temperature of the exhaust gas (enthalpy) must decrease if energy is to be conserved.

For the second series of data the air-fuel ratio was varied from approximately 17:1 to 7:1. This corresponds to varying the equivalence ratio from .9 to 2. This was accomplished by changing the fuel rate at constant airflow. The compression ratio was held constant at 7:1 and spark advance was held at 25 ?BTDC.

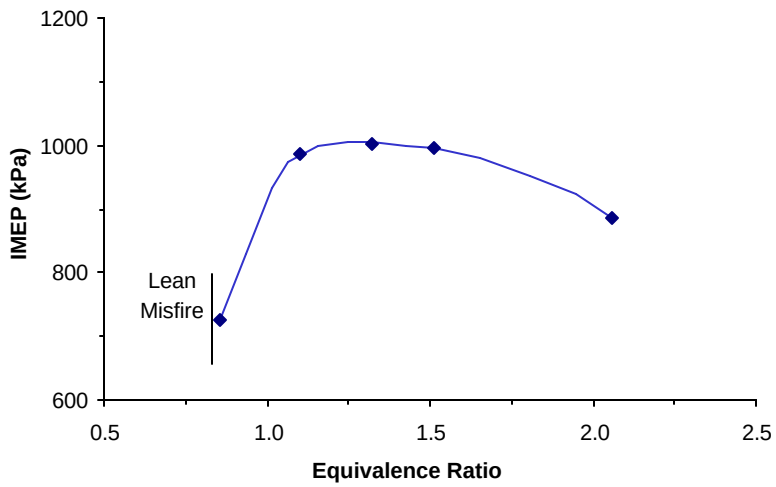

Figure 4: The relationship between IMEP and Equivalence ratio. Wide open-throttle; Compression ratio of 7:1; Spark timing of 25 ?BTDC.

The maximum IMEP for the CFR engine appears to occur close to an equivalence ratio of 1.25. Maximum performance is expected to be on the rich side of stoichiometric. At very low equivalence ratios the combustion process is limited by the available fuel. At equivalence ratios that are too high the combustion process is limited by the availability of oxygen. At intermediate equivalence ratios the combustion process is limited, to a degree, by the ease of forming free radicals. Running slightly rich means that there is more fuel available to accomplish this. These factors combine to cause the observed maximum in the IMEP as a function of equivalence ratio. For the CFR this maximum occurs at a higher equivalence ratio than for most spark ignited engines (equivalence ratio of 1.1). This may have to do with the poor turbulence and mixing characteristics of the combustion chamber as well as mixture preparation and charge cooling.

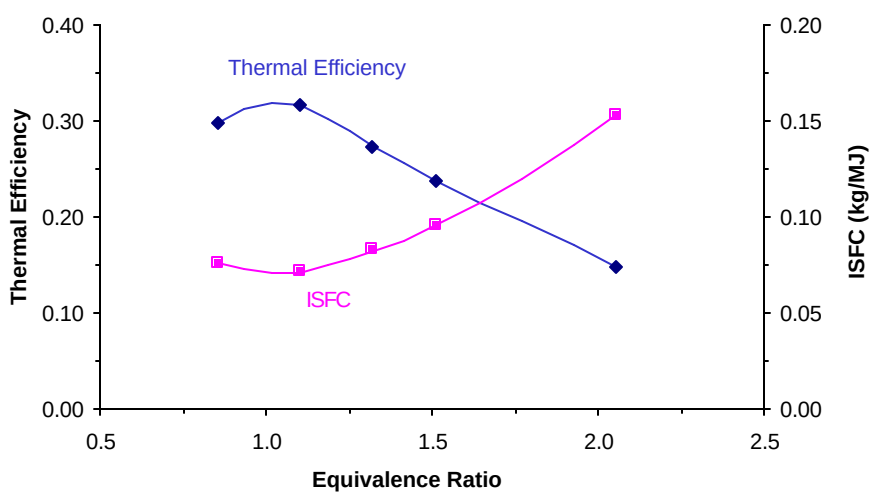

Figure 5: The relationship between Thermal Efficiency and ISFC versus Equivalence ratio. Wide open-throttle; Compression ratio of 7:1; Spark timing of 25 degrees BTDC.

Figure 5 shows that thermal efficiency (? $_{th}$) tends to decrease and indicated specific fuel consumption (ISFC) tends to increase with equivalence ratio over the range studied. Thermal efficiency is work-out divided by energy-in. In this case the energy-in is the product of the mass of fuel and the lower heating value. As the mixture is made richer a greater amount of the fuel does not burn. The energy contained in the bonds of this fuel is counted in the energy-in term but does not contribute to the work-out since it does not burn –thus the thermal efficiency decreases. At equivalence ratios that are too lean, the power output drops significantly causing the thermal efficiency to decrease. These two factors cause there to be a maximum value of thermal efficiency at an intermediate equivalence ratio. This maximum is achieved slightly rich of stoichiometric.

ISFC is the mass flow rate of fuel divided by the power output. Very lean mixtures have a low mass flow rate of fuel but also have low power output. Very rich mixtures have a greater mass flow rate of fuel but the additional fuel does not contribute to power output. These two factors cause there to be a minimum in ISFC at an intermediate equivalence ratio. The minimum value of ISFC appears to occur at an equivalence ratio slightly greater than one.

The fact that the critical points occur slightly on the rich is most likely a result of chemical kinetics. We speculate that this equivalence ratio corresponds roughly to the point at which the greatest percentage of the fuel taken into the cylinder is burned. There is a competing factor that would tend to contribute to greater thermal efficiency at leaner conditions. It can be seen in examining the effects of equivalence ratio on the specific heat ratio $(k=C_p/C_v)$. At richer mixtures k tends to be smaller due to the percentage of large fuel molecules in the mixture. If the expansion process is modeled as isentropic, the following equations hold: $T_2/T_1=(V_1/V_2)^{(k-1)}$ and $W=mC_v(T_1-T_2)$. From these relationships we see that increasing k (lower equivalence ratio) causes T_2 to decrease and work to increase, thus affecting the thermal efficiency. While this has an affect on thermal efficiency, the fact that thermal efficiency peaks rich of stoichiometric implies that it is not the dominant factor.

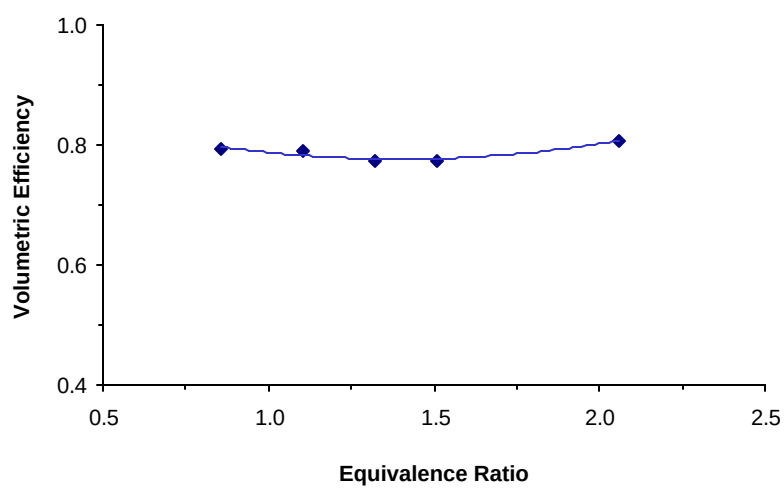

Figure 6: The relationship between Volumetric Efficiency and Equivalence ratio. Wide open-throttle; Compression ratio of 7:1; Spark timing of 25 degrees BTDC.

The correlation between volumetric efficiency (? $_V$) and equivalence ratio is shown to be very weak. There is a weak trend that the volumetric efficiency tends to

increase slightly as you move to values of the equivalence ratio that are either very rich or very lean. At very lean conditions there is very little fuel in the mixture. This means that the cylinder takes in a greater amount of air for a given amount of mixture. At very high equivalence ratios the effects of charge cooling may be significant. This decrease in temperature increases the density of the mixture, thus increasing volumetric efficiency. These trends are not strong and more tests would be warranted as confirmation.

For the third series of data the compression ratio (CR) was varied from 5:1 to 9:1. The equivalence ratio was held close to unity and spark advance was held at 20?BTDC.

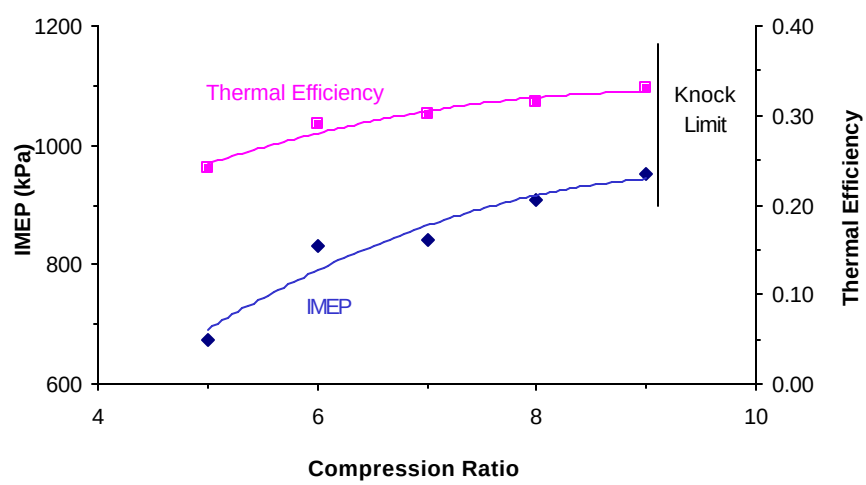

Figure 7: The relationship between IMEP and Thermal Efficiency versus Compression ratio. Wide open-throttle; Spark timing of 25 degrees BTDC; Equivalence ratio of one

The variation of IMEP with compression ratio can be seen in the figure 7 above. Greater IMEP was achieved at higher compression ratios. As we increase the compression ratio more negative compression work must be done, but the IMEP still tends to increase. The increase in work-out at higher compression ratios overrides the additional required compression work. This is a result of the properties of the fluid and is best seen in examining thermal efficiency and the ideal Otto cycle.

For the ideal Otto cycle the thermal efficiency is a function of compression ratio ($?_{th}=1-1/(r^k-1)$). As the compression ratio is increased the thermal efficiency also increases as can be seen in figure 7. Thermal efficiency is work out divided by energy-in. The energy-in (the product of mass of fuel and lower heating value) was held constant, and thus increasing thermal efficiency means the net work out must have increased—as is seen by an increasing IMEP.

The IMEP trend line seems to be approaching a maximum value. We did not run the engine up to a compression ratio high enough to reach this maximum since the knock characteristics at a compression ratio of 9 were already considerable. It is expected that IMEP would tend to decrease after reaching a maximum due to increasing heat losses through the cylinder walls. As the surface area-to-volume ratio increases, greater amounts of heat are conducted out of the cylinder. The loss of this thermal energy decreases the amount of work that can be extracted from the system.

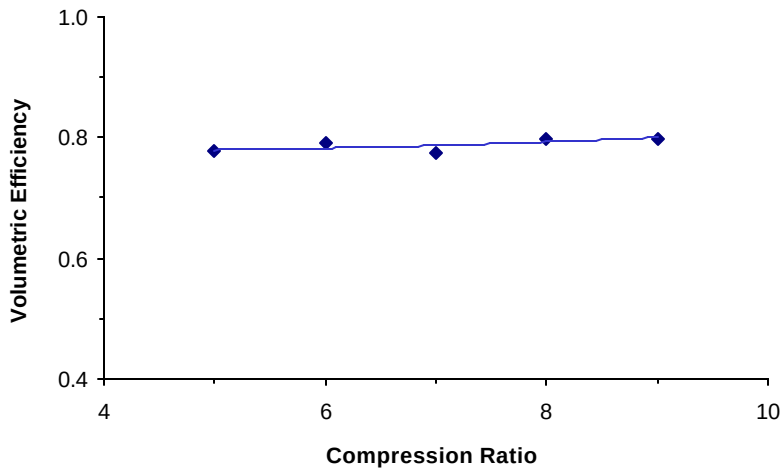

Figure 8: The relationship between Volumetric Efficiency and Equivalence ratio. Wide open-throttle; Compression ratio of 7:1; Spark timing of 25 degrees BTDC.

The effect of compression ratio on volumetric efficiency is essentially negligible, as can be seen in figure 8. Volumetric efficiency is the mass ratio of the air inducted into the cylinder to the theoretical amount of ambient air that would fill the displacement volume. Changing the compression ratio does not change the displacement volume, thus an appreciable change in volumetric efficiency would not be expected. The amount of residual gas in the cylinder, may affect the volumetric efficiency somewhat. The RDH is designed such that at lower compression ratios the amount of residual exhaust gas in the cylinder should be greater. This hot residual would increase the temperature of the fresh mixture inducted into the cylinder, and decrease its density. This implies that volumetric efficiencies should be lower at lower compression ratios. This trend is slightly evident in figure 8, although the magnitude of this effect on volumetric efficiency is quite small.

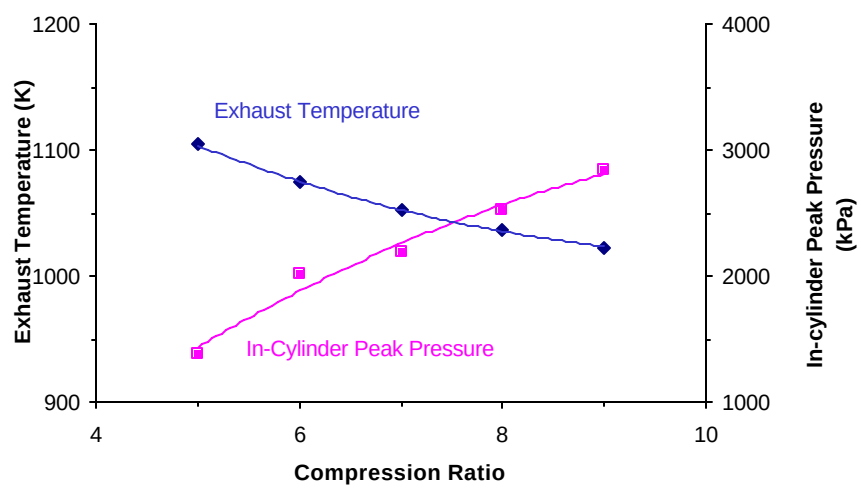

Figure 9: The relationship between Exhaust Temperature and In-cylinder Peak Pressure versus Compression ratio. Wide open-throttle; Compression ratio of 7:1; Spark timing of 25 degrees BTDC.

The effects of compression ratio on exhaust temperature and in-cylinder peak pressure are much more significant. The amount of energy released by the fuel is constant and can contribute to work-out or to increasing the enthalpy of the exhaust gas. IMEP (work) increases with increasing compression ratio (figure 7). If more energy is being converted to work, the enthalpy of the exhaust gas–and thus its temperature—must decreases if energy is to be conserved.

It is interesting to note that the maximum in-cylinder peak pressure occurs at a higher compression ratio than does the IMEP. The surface-to-volume ratio is greater at higher compression ratios. The energy released is more readily dissipated through heat loss and thus we speculate that the peak pressure is not sustained as long, resulting in a lower IMEP.

The most important trends garnered from the above research are that better performance is generally achieved at MBT, at slightly rich air fuel ratios and at high compression ratios. To illustrate the synthesis of these trends we ran the engine with a spark advance of 25?BTDC, at an equivalence ratio of 1.3 and a compression ratio of 9. The IMEP achieved was 10.58 bar—about .5 bar above the highest IMEP attained during our previous experimentation. Under these conditions, however, the knock characteristics of the engine were considerable and it seemed unwise to attempt to modify any of the parameters further. While optimal IMEP is what we desire for the greatest work output, the knock limit provides a practical constraint. The poor knock characteristics of the RDH engine emphasize the importance of combustion chamber design for reaching optimal performance. A further study of the effects of knock would be a good compliment to this research.