

Lawrence Berkeley National Laboratory

Recent Work

Title

COAL SLURRY IGNITION IN A DIESEL ENGINE SHOLATOR

Permalink

<https://escholarship.org/uc/item/8472p2bj>

Author

Brehob, D.D.

Publication Date

1985-03-01



Lawrence Berkeley Laboratory

UNIVERSITY OF CALIFORNIA

RECEIVED

LAWRENCE
BERKELEY LABORATORY

APR 25 1985

LIBRARY AND
DOCUMENTS SECTION

APPLIED SCIENCE DIVISION

To be presented at the Central States Section and
Western States Section, the Combustion Institute,
1985 Spring Technical Meeting, San Antonio, TX,
April 22-23, 1985

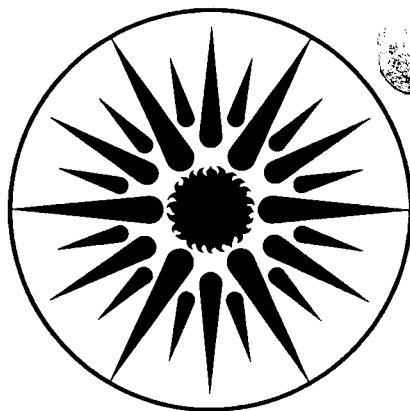
COAL SLURRY IGNITION IN A DIESEL
ENGINE SIMULATOR

D.D. Brehob, F. Robben, R.F. Sawyer and
P.M. Sherman

March 1985

TWO-WEEK LOAN COPY

This is a Library Circulating Copy
which may be borrowed for two weeks



APPLIED SCIENCE
DIVISION

LBL-19129
c.2

DISCLAIMER

This document was prepared as an account of work sponsored by the United States Government. While this document is believed to contain correct information, neither the United States Government nor any agency thereof, nor the Regents of the University of California, nor any of their employees, makes any warranty, express or implied, or assumes any legal responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by its trade name, trademark, manufacturer, or otherwise, does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof, or the Regents of the University of California. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof or the Regents of the University of California.

Central States and Western States Sections of the
Combustion Institute, 1985 Spring Meeting at
Southwest Research Institute, San Antonio, Texas.
April 22-23, 1985 Paper No. 4-1A.

COAL SLURRY IGNITION IN A
DIESEL ENGINE SIMULATOR *

D.D. Brehob, F. Robben, R.F. Sawyer,
and P.M. Sherman **

Lawrence Berkeley Laboratory and the Mechanical
Engineering Department at the University of
California, Berkeley

** On sabbatical leave from the University of
Michigan, Aeronautical Engineering Department

ABSTRACT

Compression ignition characteristics of three coal slurry fuels are compared to diesel reference fuel in a diesel engine simulator. The three slurries are 45 wt% coal (5 micrometers mean diameter) in water, diesel No. 2, and methanol carriers. Each experiment is a single, isolated combustion event. Heat transfer losses to the cold walls and mass losses past the square piston seals reduces the compressed gas temperature. Air preheating to 400 K for the diesel, to 450 K for the coal/methanol and coal diesel, and to 525 K for the coal/water fuels assures ignition. Activation temperatures (E/R) of 2,330 K, 2,270 K, 2,670 K, and 3,430 K for diesel fuel, coal/diesel, coal/methanol, and coal/water slurry, respectively, are found from ignition delay measurements.

* This work was supported by the Assistant Secretary for Energy Technology, Heat Engines Section, U.S. Department of Energy under Contract No. DE-AC03-76SF00098.

INTRODUCTION

Rudolph Diesel's 1892 patent covered the potentiality of using solid fuels in a compression ignition engine (reported by Soehngen (1976)). Severe contamination of the engine by coal particles in just seven minutes caused Diesel to discontinue testing with solid fuels. However, his one time co-worker, Rudolf Pawlikowski continued developing and promoting the concept. Largely because of Pawlikowski's efforts, extensive research and development (primarily supported privately) were carried out in Germany from 1930 to 1944. Soehngen (1976) discusses the German coal fired diesel efforts in detail. The Germans sought an economical fuel to achieve independence from foreign oil supplies.

Because of similar motives, the U.S. government has funded research on coal fired diesel engines. Robben (1983) reports that diesels used in transportation consume approximately 7.5% of the nation's oil supply. One third of the fuel is burned in low to medium-speed engines. The length of time available for combustion is longer in the lower speed engines making them prime targets for coal utilization. A large petroleum fuel savings would be realized if distillate burning engines could be converted to burning coal.

The German coal fired engines used a pulverized coal greater than 50 micrometers mean diameter. However, because the U.S. infrastructure is built around the use of liquid fuels, slurry fuels would greatly facilitate the transportation and storage over the coal powder. Also, slurry fuels are much safer than the explosive coal dust. Pulverized coal has been slurried with diesel fuel, alcohols, water, and blends of these carriers with coal mass loadings from 10-60%. The diesel fuel and alcohol slurries aid in the ignition and help sustain the coal combustion. However, these carriers increase the cost of the slurry. Thus, there may be insufficient economic incentive to overcome the injection, combustion, wear and lubrication, fuel storage, and fuel stability problems associated with introduction of coal particles. Coal/water slurry is the least expensive alternative; however, it presents the greatest combustion difficulties.

Results from a number of coal slurry fired diesel engine programs have been published since WW II. All of the researchers reported problems in at least one of the following areas (summarized by Caton and

Rosegay(1983)): fuel injection pump seizure or excessive wear, injector plugging, contamination of engine oil leading to high wear, incomplete burnout of the coal, or high gaseous and particulate emissions. Ceramic components are being incorporated in production engines which may reduce or eliminate some of the problems mentioned above. For example, an engine with ceramic combustion chamber surfaces and minimum cooling would increase coal conversion and aid in the ignition of the slurry. Ceramic surfaces may be less vulnerable to coal particle abrasion. These advances may warrant a reevaluation of coal fuels in diesel engines.

The goal of this project is to determine the conditions (temperature and pressure) for ignition of three slurries in comparison with diesel fuel. Ignition delay based on pressure rise is measured. In addition, high speed movies of the combustion event for the different fuels are examined to observe the location of ignition and the combustion duration.

EXPERIMENTAL FUELS AND FACILITIES

Fuels

The three slurries used in this study were provided by Bartlesville Energy Technology Center (now the National Institute for Petroleum and Energy Research). The coal loading is 45 wt% for the water, methanol, and diesel No. 2 slurries. The slurries contain less than 5 wt% additives: dispersants, viscosity improvers, lubricants, and stabilizing agents. The mean coal particle diameter is 5 micrometers with the maximum particle size less than 40 micrometers. The coal contains less than 0.7 wt% ash after cleaning by the Otisca process (hot acid wash).

Injection

The injection system on the compression expansion apparatus uses a conventional Lucas CAV 7mm bore jerk pump. A modified Stanadyne pencil injector is used. The standard nozzle tips are removed and replaced by single hole orifices made from hardenable steel. The machined orifices are interchangeable allowing orifice diameters to be varied from 0.254-0.508 mm. The fuel injector is situated in the engine's cylinder wall. The fuel is injected into a clearance height of 6.4 mm at top dead center (TDC). To reduce the amount of fuel spray penetration onto piston top and cylinder head surfaces, the spray cone angle is minimized. The length to diameter ratio of the nozzle tips are made as

large as possible with the available machining equipment: greater than 1.7 for all the orifices. A summary of the injection parameters used is contained in Table 1a.

The injection equipment is evaluated outside the square piston engine in a pressurized vessel. A schematic of the bomb and associated equipment is contained in Figure 1. Taylor (1969) illustrates "the great importance of the density ratio [air density in bomb divided by fuel density] on the characteristics of the spray" by presenting NACA photographs of injection at a range of density ratios. A pressure of 15 atm at ambient temperature in the bomb closely approximates the density of the compressed air near top dead center (TDC) in the engine at the conditions described later.

The bomb experiments are single injections to emulate the engine conditions. A single injection is obtained by opening and closing the rack approximately half a revolution before and after the injection stroke, respectively. Timing of the solenoids is accomplished by the injection control system. An inductive pickup on the cambox pulley pulses once per revolution. A single inductive pickup signal is conditioned and sent to the injection control box by the operator enabling the circuit (labelled "signal conditioning" in Figure 1).

The injection line contains a tee approximately 10 cm upstream of the injector. The coal slurry fuels are introduced at this point thus saving the fuel pump from corrosion or abrasion. Normally the injection lines contain only diesel fuel. Slurries are loaded between tests by releasing tension on the injector valve. A syringe is loaded with the slurry and the high pressure valve at the tee is opened. The force applied to the syringe by hand is sufficient to purge the line and injector of the diesel fuel. Enough slurry is contained in the line and injector to operate for at least ten injections before contamination by diesel fuel.

The sprays from the injector are evaluated by analyzing the injection line pressure history and shadow photographs of the spray. The photographic setup is shown in Figure 2. The operator selected pulse from the inductive pickup as described above triggers the injection control system and the oscilloscope. In addition, it sends a pulse to an adjustable time delay which in turn pulses the power supply of the light source. A complete record of the spray can be obtained by photographing successive

Table 1a.

INJECTION PARAMETERS

Injection pump: 7 mm plunger CAV Lucas, FA0B060

Injector: Stanadyne pencil injector

Orifice: Standard nozzle tip removed, replaced
by interchangeable single hole orifices of
0.254 - 0.508 mm diameter.

Length/diameter: >1.7 for all nozzle tips.

Needle opening pressure: 100 - 200 atm.

Table 1b.

ENGINE CONDITIONS

16:1 Compression ratio

900 rpm engine and injection pump speed

Pressurized inlet air: 2 atm abs

Preheated inlet air: up to 525 K

Block temperatures: Ambient - 125 C

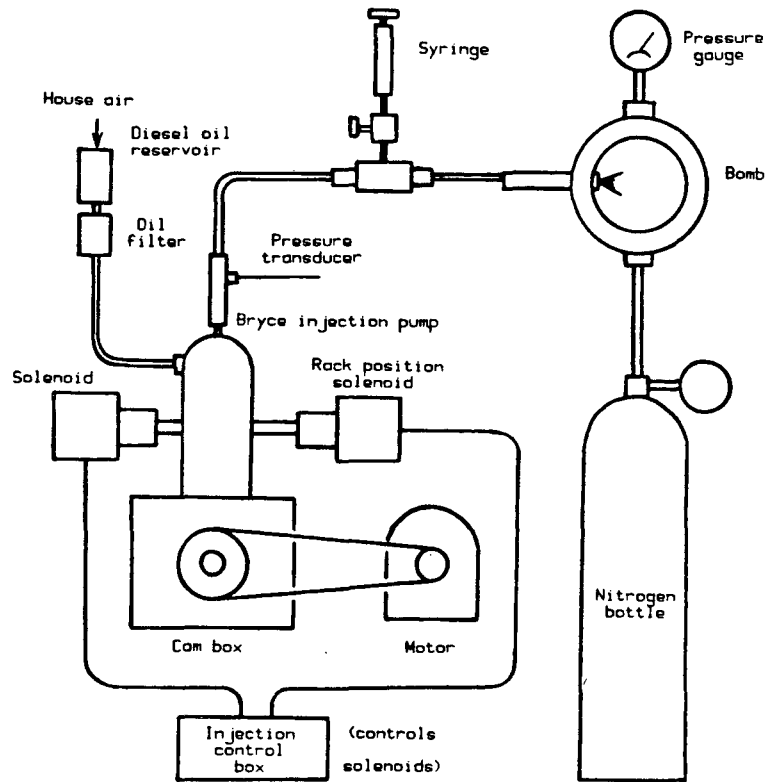


Figure 1. Schematic of Injection Test Facility.

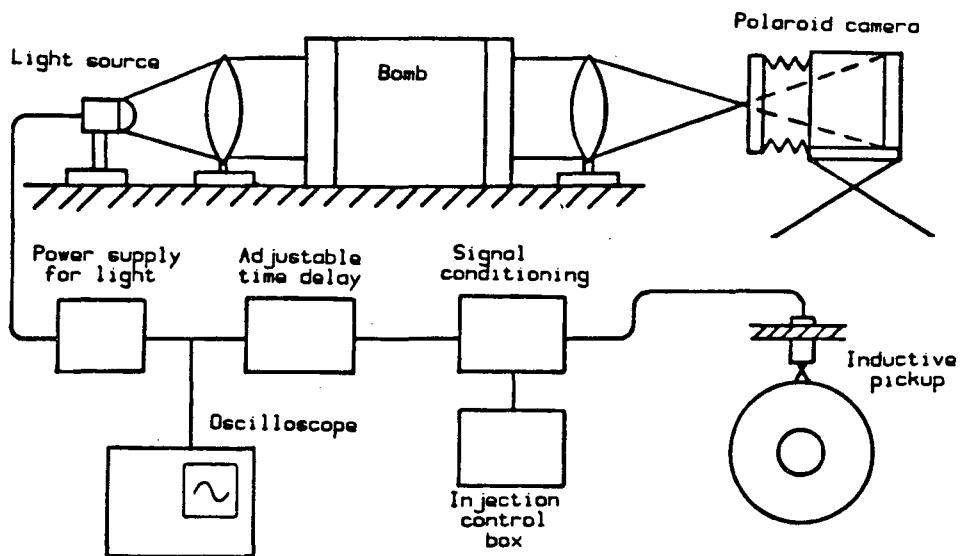


Figure 2. Photographic Setup for Injection Test Facility.

injections at different delay times. The light from the point light source is collimated, passed through the test section where it is scattered by particles, and imaged onto Polaroid film. The light from the flash persists for slightly more than one microsecond.

Engine

A square piston, compression-expansion apparatus was developed for both compression ignition and spark ignition simulation, Figure 3. The 89 mm square engine block is situated above a conventional Coordinated Lubrication Research (CLR) single cylinder engine minus the cylinder head. An adjustable connecting rod links the CLR and square pistons. The CLR piston acts as a crosshead piston absorbing the lateral thrust forces. By changing the length of the connecting rod, compression ratios from 5:1 to 24:1 can be selected. A compression ratio of 16:1 is used in this work. Two of the four "cylinder" walls are stainless steel and two are quartz windows. This feature provides complete optical access to the combustion event and allows techniques such as schlieren, direct, and shadow photography to be used.

Experiments on the engine simulator are conducted as follows: a) the synchronous electric motor and flywheel are run at 1800 rpm; b) the motor is shut down and the flywheel slowly loses speed; c) at the predetermined flywheel speed the clutch is engaged and the engine simulator attains the desired speed (900 rpm for this experiment) within one revolution; d) after ten revolutions, one which is a firing event, the clutch is disengaged and the brake is applied. Engine rotation is measured by an optical shaft encoder. The encoder gives two signals: a pulse every one-half crank angle degree (CAD) of rotation and a single index pulse per revolution at TDC.

A summary of the engine conditions for these tests is contained in Table 1b. The inlet air for the engine can be pressurized and preheated to simulate supercharged operation by means of an air compressor and a resistance heater. Losses due to leakage and heat transfer to cylinder surfaces at ambient temperatures are offset by preheating the inlet air to temperatures as high as 525 K. Inlet pressures of 3 atm absolute are possible. However, 2 atm absolute is used throughout the work presented here. The cylinder surfaces and piston top can be preheated up to 400 K by opening both intake and exhaust valves and passing heated air through the engine.

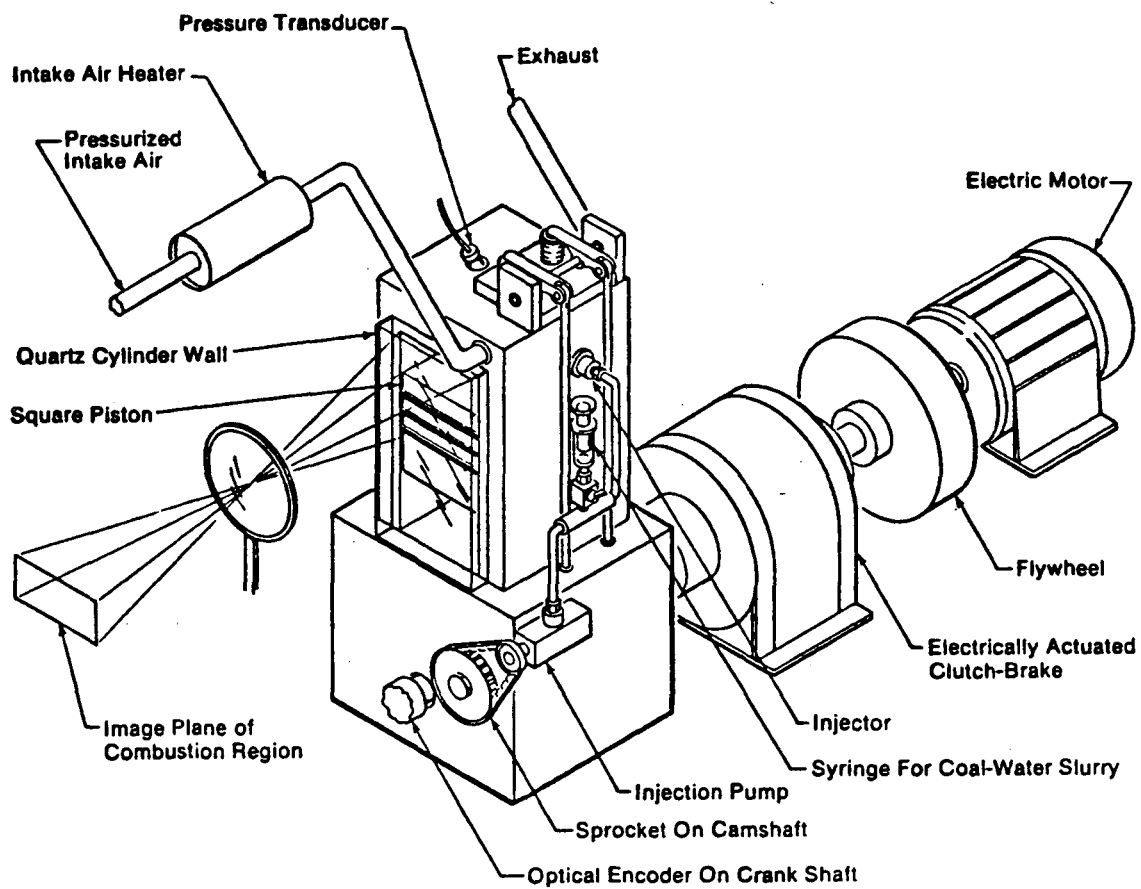


Figure 3. Square Piston, Diesel Engine Simulator.

A cambox operating at 900 rpm actuates the injection pump. Because of the single injection nature of the experiments, the cambox speed need not be one-half crankshaft speed as normally required on four-stroke diesels. The cambox is gear driven off the camshaft at twice camshaft speed. Injection timing is selected by rotating the camshaft and cambox gears relative to each other. The injection control box opens and closes the rack as described above. The slurry fuel is loaded by syringe, also described above.

Data Acquisition and Control

Two time bases are important to controlling the experiment, Figure 4a. A Universal Programmable Timer, based on real time, sequences the clutch brake, high speed camera, and charge amplifier reset. Another timer based on the TDC and one-half CAD pulses from the shaft encoder sequences other events based on engine CAD: analog to digital (A/D) conversions at one-half CAD intervals, injection control system timing, and spark light source timing.

The data acquisition system is shown schematically in Figure 4b. A LSI-11 microcomputer interfaces with the inductive pickup on the flywheel, a cylinder pressure transducer charge amplifier, and the optical shaft encoder. From these signals the following information is computed: flywheel rpm during coast down, pressure at one-half CAD intervals, engine rpm for the firing revolution.

MODELLING

The square piston geometry suffers from significant leakage past the ring pack. Computer modelling of the compression and expansion was undertaken to estimate the magnitude of the processes governing the observed pressure in the cylinder and to estimate the temperature as a function of CAD. Estimates of air temperature are important because it is the dominant variable controlling the ignition delay interval.

Engine geometry prescribes the volume above the piston as a function of CAD. From known inlet pressure and temperature, the program computes successive values assuming adiabatic compression. An empirical fit for the ratio of specific heats for air is used. Temperature and pressure are adjusted at each calculation step to correct for leakage and heat transfer. Heat transfer is estimated by using Woschni's (1967) correlation. The mass leak rate is

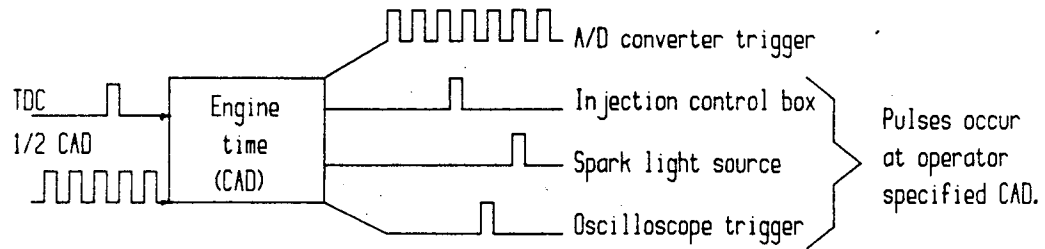
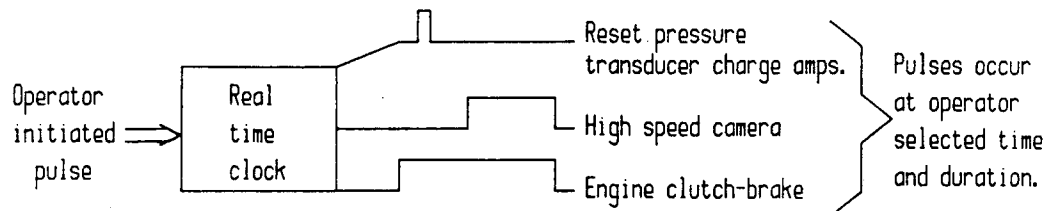


Figure 4a. Engine Control Electronics.

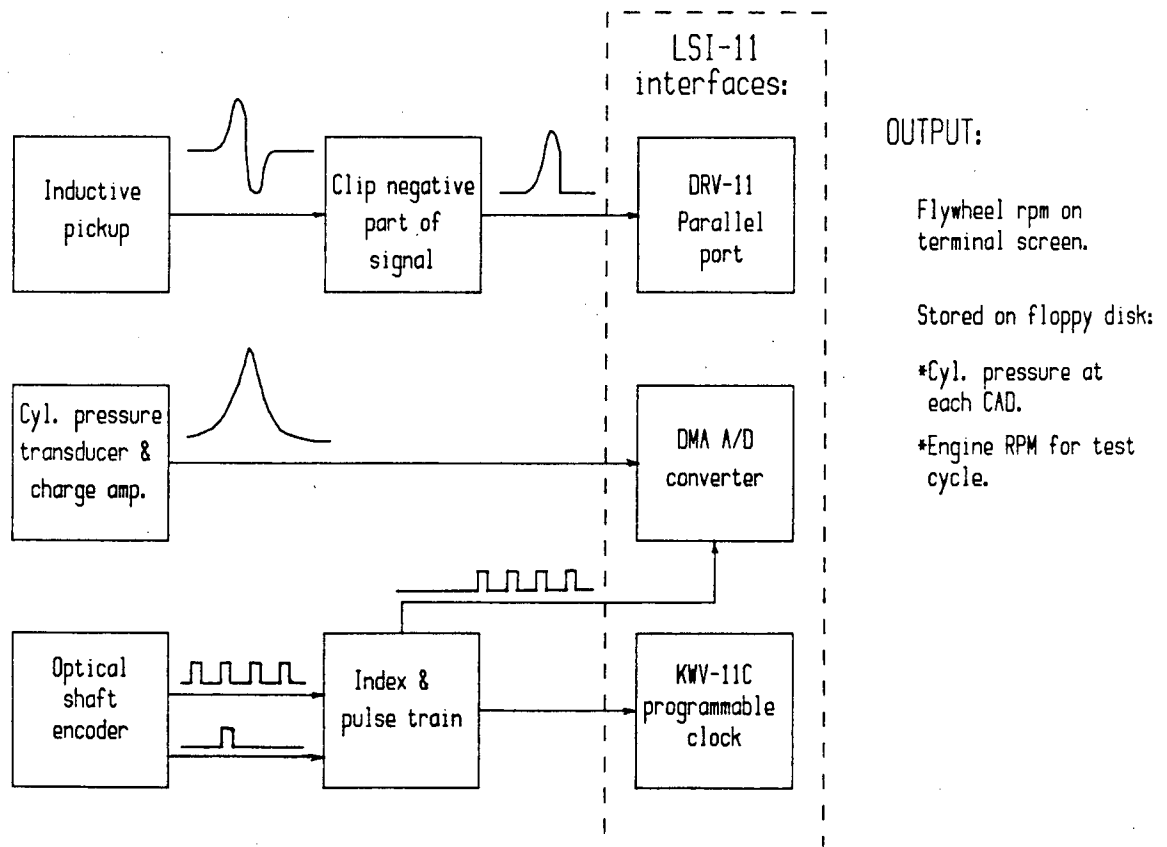


Figure 4b. LSI-11 Microcomputer for Data Acquisition.

assumed to be sonic, i.e., proportional to cylinder pressure. Static leak measurements at two to six atm, well below compression pressures, shows a leak rate of 0.5 standard liters/atm-sec. The leakage coefficient is adjusted so that the peak pressure calculated by the model matches the peak experimental pressure.

A comparison between the model and experiment for a motored case with ambient temperatures and 2 atm absolute inlet pressure is shown in Figure 5a. Also shown is the pressure for an adiabatic case with no leakage. The leakage past the ring pack accounts for the major proportion of the pressure losses from the adiabatic values. The air temperature in the cylinder and the mass of air trapped above the piston as a function of CAD are plotted in Figure 5b. At the end of the compression and expansion strokes, approximately 40% of the air leaks out of the combustion chamber. The temperature falls to 245 K at the end of the expansion stroke. Exhaust gas temperature measurements, at different operating conditions, also indicate temperatures significantly below ambient.

RESULTS AND DISCUSSION

Injection Tests

Orifices of 0.254 to 0.508 mm were tested. As expected, better atomization is obtained with the smaller orifice diameters. However with the coal slurries, plugging is encountered with the 0.254 mm nozzle. A 0.330 mm nozzle gives good atomization with infrequent occurrence of plugging. The spring tension on the injector valve governs the nozzle opening pressure. Opening pressures of 100 - 200 atm were tested. The atomization improves as the pressure is increased. Attempts at higher opening pressures are unsuccessful because the nozzle tip deforms under the higher stresses. Thus, a nozzle opening pressure of 200 atm is used. The best atomization is found at moderate rack settings. Hiroyasu and Kadota (1976) report that the Sauter mean diameter increases as the amount of fuel injected is increased. They attribute this to more particles agglomerating as the amount of fuel injected is increased.

Engine Tests

Compression ignition of all three slurry fuels and the reference fuel is possible only when inlet air preheat is used to overcome heat transfer and mass losses. Figure 6 shows the estimated temperature at the time of injection for the three inlet air

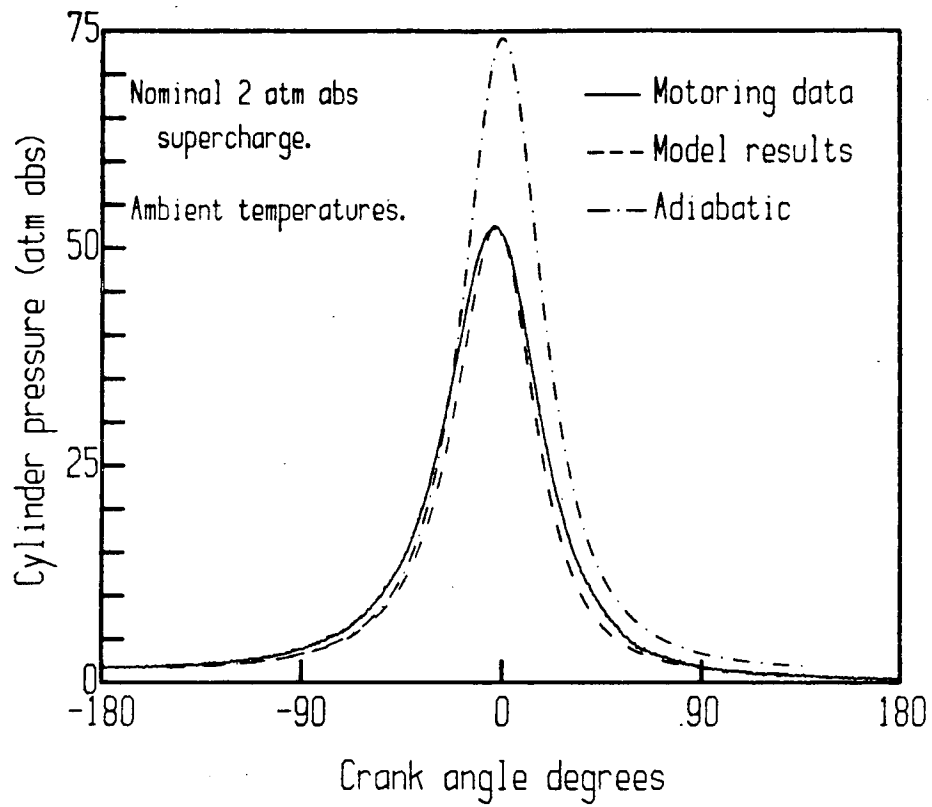


Figure 5a. Motoring Cycle Pressure--Comparison Between Model and Experimental Results.

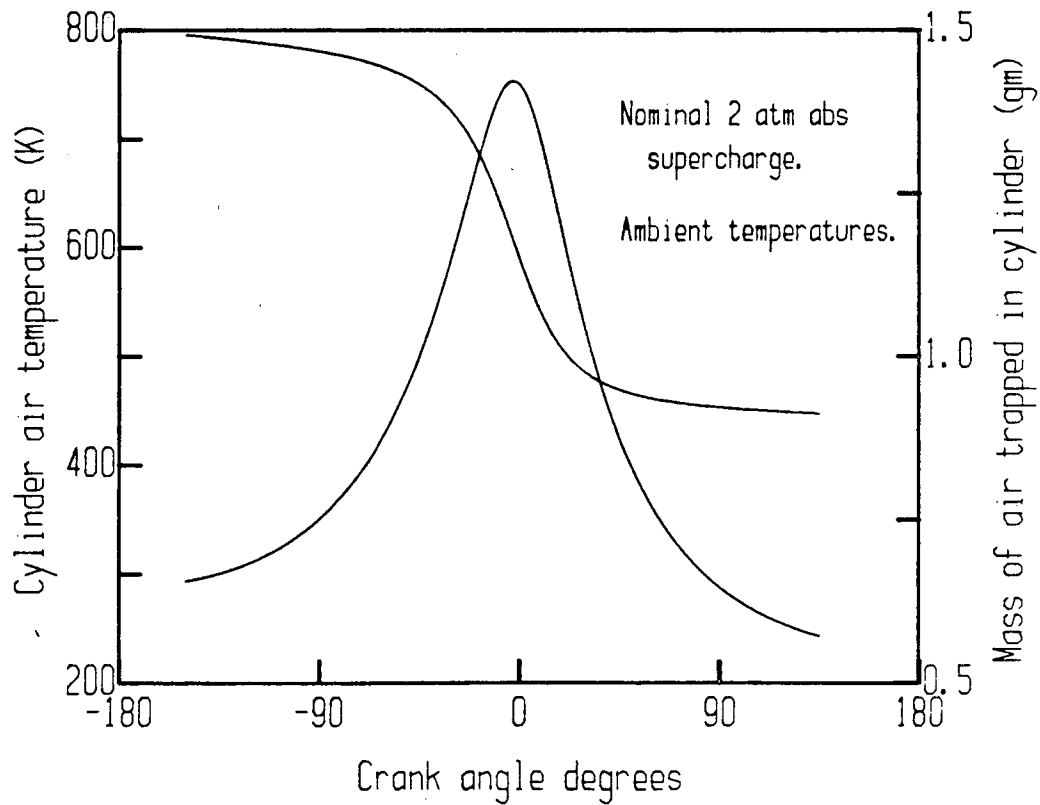


Figure 5b. Model Predictions of Motoring Cycle Temperature and Mass of Air in Combustion Chamber.

temperatures used: 400 K for diesel fuel, 450 K for coal/methanol and coal/diesel, and 525 K for coal/water slurry. Ignition of the diesel fuel requires inlet air temperatures of 400 K or above. Further heating to 525 K is required for coal/water slurry ignition. These results help substantiate results from computer simulations of coal combustion in a medium-speed CI engine by Caton (1984). Ignition of the coal particles is not obtained at the 16:1 compression ratio unless an ignition aid is used. Caton simulates pilot injection by introducing diesel fuel into the cylinder early in the compression stroke. With pilot injection quantities above 8% of the total fuel energy, combustion of the coal is initiated and sustained. The key consequence of the pilot injection is a higher air temperature at the time of coal injection.

Ignition delay for the four fuels is calculated from cylinder pressure and injection line pressure records. The definition used here is the time between start of injection and when cylinder pressure of the combusting cycle exceeds cylinder pressure for a motoring cycle. The combusting cylinder pressure drops below the motoring pressure as a result of the latent enthalpy of vaporization of the diesel fuel, methanol, or water. However, the motoring pressure is surpassed by the combusting pressure after sufficient energy release from the combustion. Sample cylinder pressure traces and the injection pressure trace in Figure 7 illustrate how ignition delay is calculated. The start of injection is considered to be the time when the line pressure exceeds the nozzle opening pressure. Introduction of the first droplets into the combustion chamber probably occurs after the line pressure exceeds the opening pressure. This introduces systematic error which will be discussed later.

An empirical Arrhenius expression correlates ignition delay time and experimental parameters:

$$t_i = C \exp (T_a / T),$$

where t_i is the ignition delay, C is the pre-exponential factor, and T_a is the activation temperature. The constant, C , contains the pressure dependency of ignition delay.

Figure 8 contains a summary of the data on all four test fuels. The temperature is that estimated by the computer model at the time of injection (Figure 6). A least squares fit of the data gives a slope (activation temperature) of 3430 ± 610 K, 2670 ± 1150 K, 2270 ± 810 , and 2330 ± 110 K for coal/water, coal/methanol,

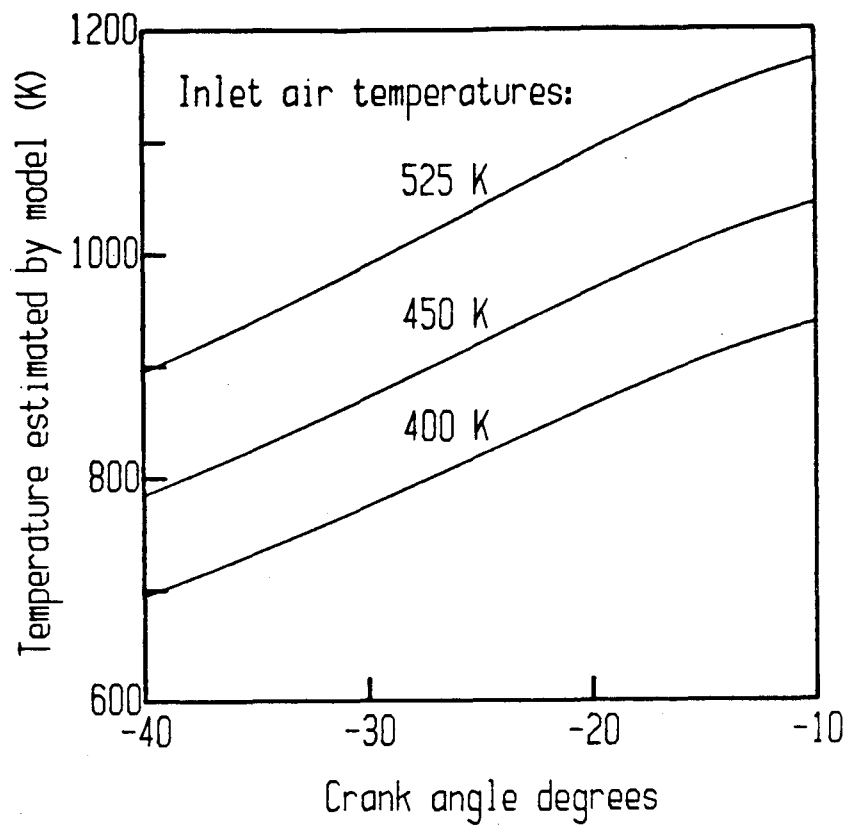


Figure 6. Model Predictions of Motoring Cycle Temperatures for Varied Inlet Air Temperature.

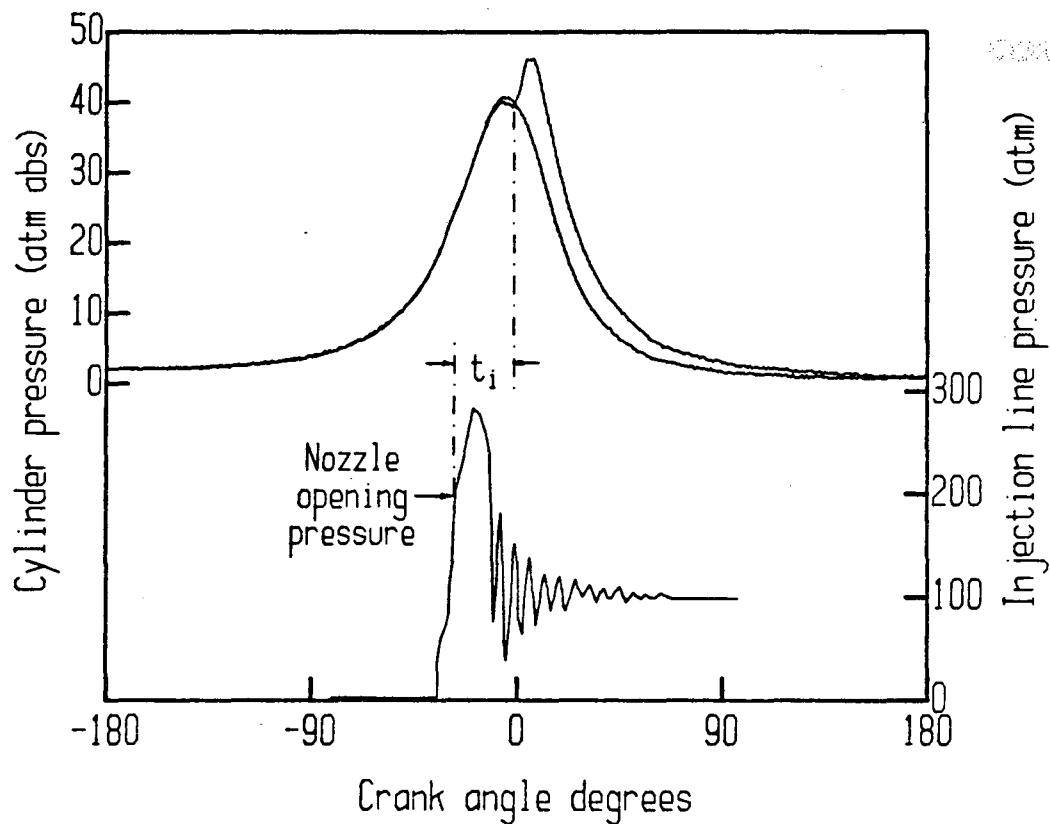
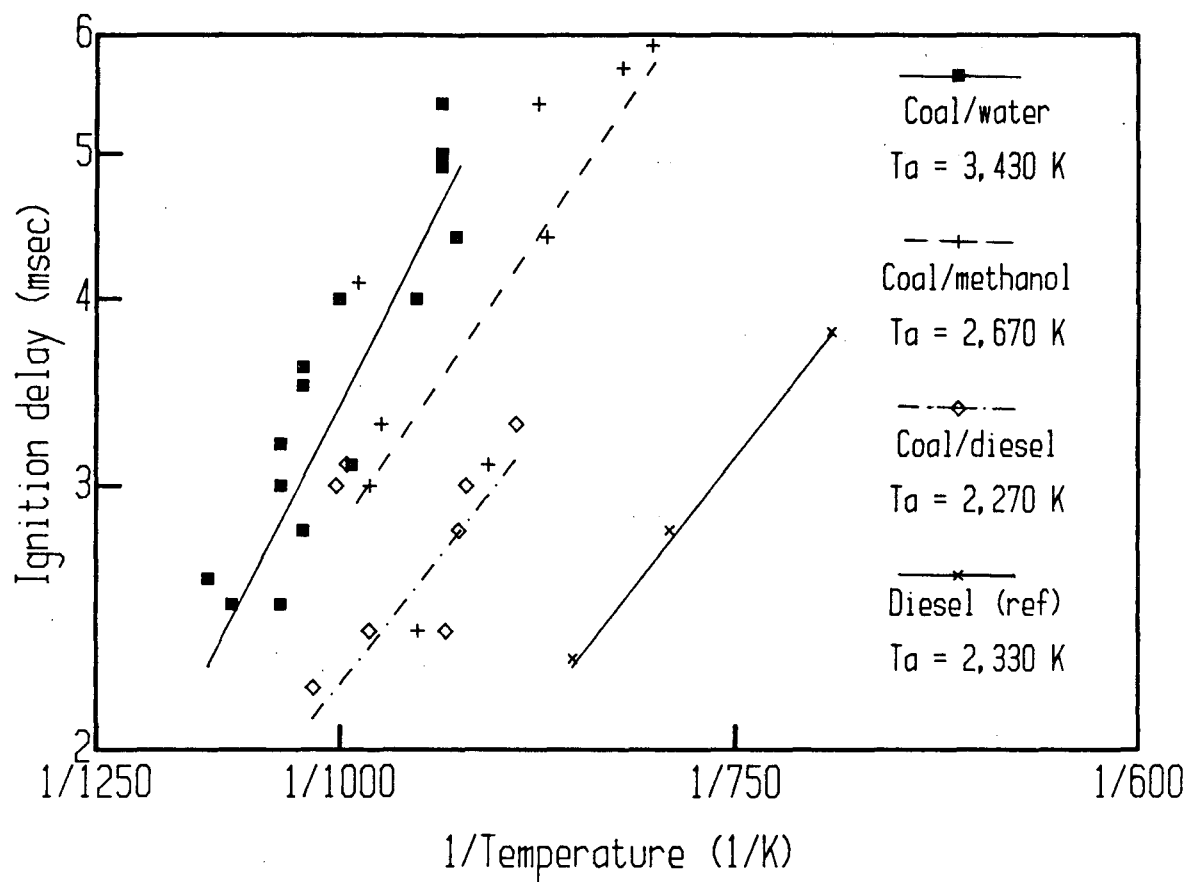


Figure 7. Definition for Ignition Delay.



coal/diesel, and diesel fuel, respectively, at the 90% confidence interval. (The two data points for coal/diesel slurry which give an ignition delay of approximately 3 msec for 1000 K temperature are not included in the least squares analysis.) The 90% confidence band on the least squares fit to the data is shown in Figure 9.

As mentioned above, the actual start of injection may lag the delivery line pressure rise. In this case the actual ignition delay is shorter. A systematic error in this direction would reduce the pre-exponential factor greatly and increase the activation temperature slightly.

The results on diesel fuel from Figure 8 are replotted in Figure 10 along with estimated delays computed from correlations given by Henein and Bolt (1969) and Tsao, et al. (1962). The correlation given by Tsao, et al. is an empirical formula relating ignition delay to air temperature, air pressure, and engine rpm. Henein and Bolt report an apparent activation energy (reformulated here as an activation temperature in deg K) and a pre-exponential factor for diesel fuel:

$$t_i = 0.146[\text{msec}] \exp (1467[\text{K}] / T);$$

Results from this equation are vastly different from either Tsao's correlation or the present work. However, the above equation applies to cylinder pressures of 48 atm. In a previous work, Henein and Bolt (1967) report that ignition delay is inversely proportional to cylinder pressure to the 1.46 power at 600 rpm and 1.77 power at 1000 rpm. To correct the correlation for the approximate average pressure of 25 atm used here, a factor of $(48 / 25)$ raised to the 1.7 power was multiplied to the right hand side of the equation. With the correction, the two correlations are in close agreement. The results of the present work lie within 10% of either correlation in the range of temperatures tested. However, the present work indicates a steeper slope (higher activation temperature) than either correlation.

Siebers and Dyer (1985) report ignition delay measurements as a function of temperature for coal/water slurry. (The fuel is identical to the one used in this study.) Three different definitions of ignition delay are plotted in their Figure 9: luminosity delay, pressure deficit delay (time from start of injection to the minimum pressure), and pressure recovery delay (time from start of injection

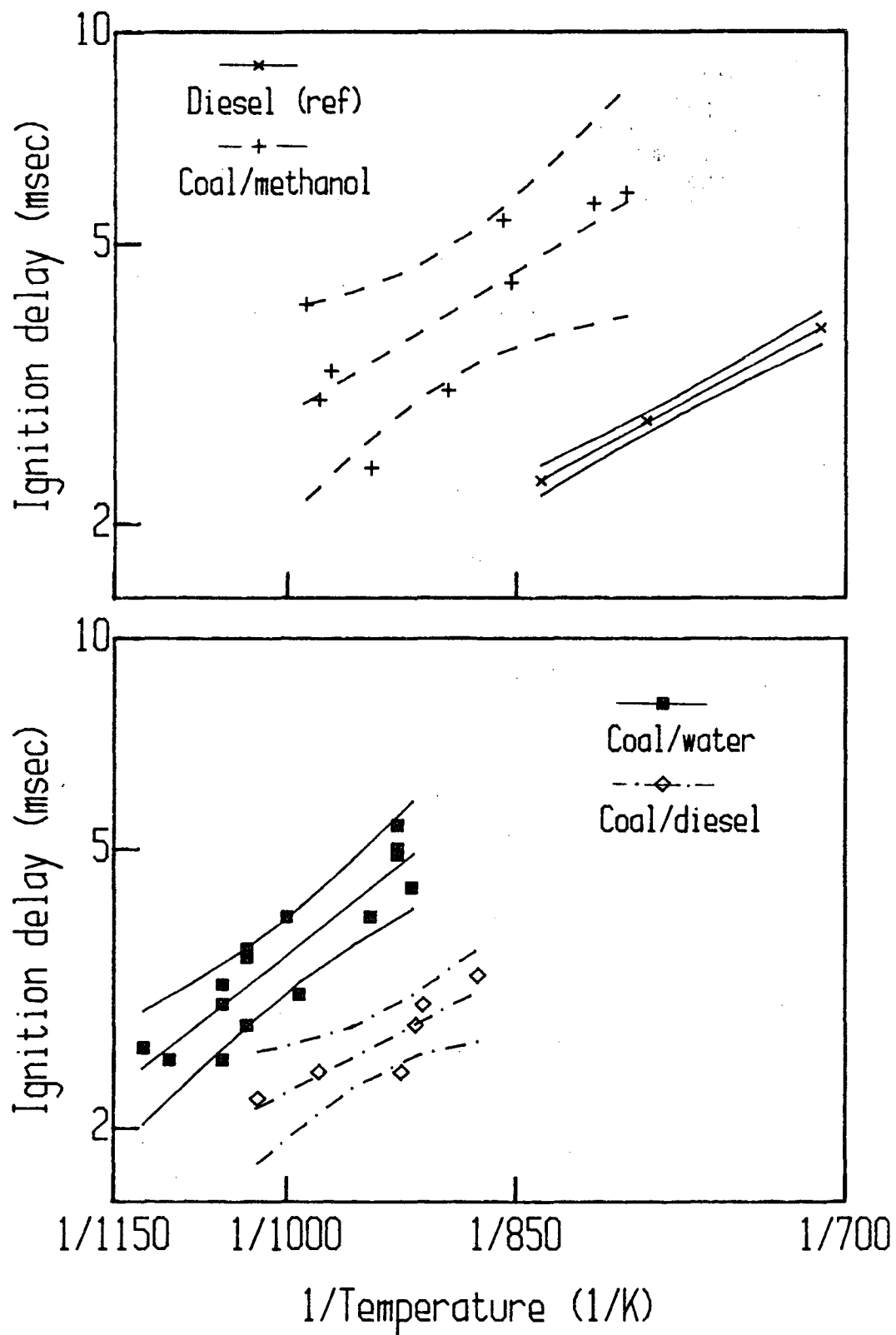


Figure 9. Arrhenius Plots of Ignition Delay for Four Fuels in the Engine Simulator with 90% Confidence Level Bands.

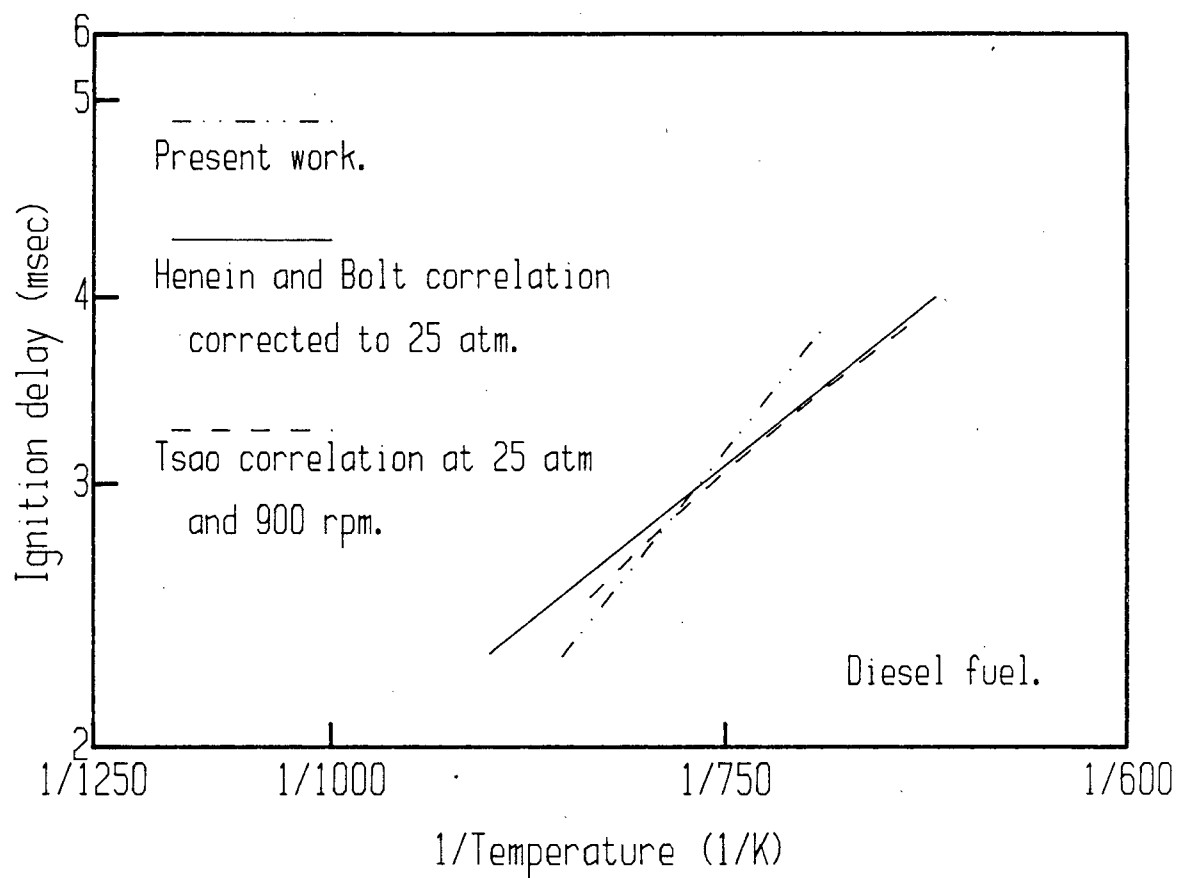


Figure 10. Ignition Delay Measurements on Diesel Fuel Compared to Results Computed from Reported Correlations.

to when pressure of combusting case exceeds case with no fuel injection). Replotting the data in the customary Arrhenius plot, log of ignition delay against the reciprocal of temperature, three activation temperatures are evident:

Definition	Approximate T_a
Luminosity	2460 K
Pressure Deficit	3520 K
Pressure Recovery	4540 K

Siebers and Dyer's measurements on diesel fuel indicate ignition delays much shorter with diesel fuel. However, the activation temperatures are similar:

Definition	Approximate T_a
Pressure deficit	2730 K
Pressure recovery	4360 K

(The degree of uncertainty in evaluating activation temperatures from Siebers and Dyer's Figure 9 is quite large for the luminosity delay with coal/water slurry and for the two diesel fuel delays. Figure 9 is not an Arrhenius plot. Thus, the quantities with short ignition delays are difficult to evaluate accurately from the graph.) The ignition delay definition most consistent with the definition used in the compression-expansion apparatus is the pressure recovery delay. The compression expansion apparatus gives T_a of 3430 K as compared to the 4340 K in the constant^a volume bomb for the coal/water slurry. The pressure recovery ignition delay for diesel fuel is 4360 K compared to 4540 K for the coal/water slurry. The compression expansion apparatus gives a wider separation: 2330 K for diesel fuel and 3430 K for coal/water slurry.

Direct movies of the luminous combustion were taken for all of the fuels at framing rates of 2500 and 5000 frames/sec. In nearly all of the movies, ignition occurs near the injector. Combustion spreads to the remainder of the combustion chamber in several CAD. The smaller particles do not penetrate across the chamber and remain close to the injector. The smaller droplets ignite first because they are prepared for combustion before the larger droplets.

The duration of the combustion can be measured from the luminous period observed in the film:

Fuel	Approx. burn duration	
Diesel No. 2	75 CAD	11 msec
Coal/oil	60 CAD	9 msec
Coal/methanol	45 CAD	7 msec
Coal/water	35 CAD	5 msec

Extinguishment of the coal/water and coal/methanol slurries before burn out is verified by viewing the compression stroke which follows the combustion event. As the temperature increases on the compression, the unburned fuel particles glow. This is not seen with diesel fuel. Whether coal particles remain from the previous stroke because of quenching in the bulk gas or in the crevice volume can not be determined. These single shot experiments are not suitable to the determination of fuel burnout characteristics.

CONCLUSIONS

The ignition results are summarized below:

a) Compression ignition of diesel fuel is possible when the inlet air temperature is raised to at least 400 K at the 16:1 compression ratio in the square piston engine simulator.

b) Additional air preheating is required for ignition of coal/water slurry: 525 K as opposed to 400 K for the reference fuel.

c) The activation temperatures for the four fuels are:

Diesel No. 2	2330 K
Coal/diesel	2270 K
Coal/methanol	2670 K
Coal/water	3430 K

The high speed movies reveal the following information:

a) Ignition is observed near the injector in nearly all of the films.

b) The combustion duration for the fuels is in this order:

diesel > coal/oil > coal/methanol > coal/water

ACKNOWLEDGEMENTS

The authors wish to acknowledge the invaluable advice and support received from the late Prof. Arthur Saarsten. Ned Dickson is appreciated for his role in designing, fabricating, and testing the injection equipment. Also, Mark Fan and Hitoshi Takagi are acknowledged for their assistance in conducting experiments.

REFERENCES

- Caton, J.A., "Cycle Simulation of a Coal Particle Fueled Reciprocating, Locomotive Engine," ASME Paper 84-DGP-19 (1984).
- Caton, J.A., Rosegay, K.H., "A Review and Comparison of Reciprocating Engine Operation Using Solid Fuels," SAE Paper 831362, (1983).
- Henein, N.A., Bolt, J.A., "Correlation of Air Charge Temperature and Ignition Delay for Several Fuels in a Diesel Engine," SAE Paper 690252 (1969).
- Henein, N.A., Bolt, J.A., "Ignition Delay in Diesel Engines," SAE Paper 670007 (1967).
- Hiroyasu, H., Kadota, T., "Fuel Droplet Size Distribution in Diesel Combustion Chamber," Bulletin of the JSME, 19, 1064-72 (1976).
- Robben, F., "Coal-Fueled Diesel Engines," SAE Paper 831747 (1983).
- Siebers, D.L., Dyer, T.M., "The Autoignition and Combustion of Coal-Water Slurry Under Simulated Diesel Engine Conditions," ASME Paper 85-DGP-15 (1985).
- Soehngen, E.E. "Development of Coal-Burning Diesel Engines in Germany," Energy Research and Development Administration Report FE/WAPO/3387-1 (1976).
- Taylor, C.F., The Internal-Combustion Engine in Theory and Practice, The M.I.T. Press (1969).
- Tsao, K.C., Myers, P.S., Uyehara, O.A., "Gas Temperatures During Compression in Motored and Fired Diesel Engines," SAE Transactions, 70, 136-45 (1962).
- Woschni, G., "A Universally Applicable Equation for the Instantaneous Heat Transfer Coefficient in the Internal Combustion Engine," SAE Paper 670931 (1967).

This report was done with support from the Department of Energy. Any conclusions or opinions expressed in this report represent solely those of the author(s) and not necessarily those of The Regents of the University of California, the Lawrence Berkeley Laboratory or the Department of Energy.

Reference to a company or product name does not imply approval or recommendation of the product by the University of California or the U.S. Department of Energy to the exclusion of others that may be suitable.

TECHNICAL INFORMATION DEPARTMENT
LAWRENCE BERKELEY LABORATORY
UNIVERSITY OF CALIFORNIA
BERKELEY, CALIFORNIA 94720