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COMPRESSION IGNITION OF COAL SLURRY FUELS

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ABSTRACT

About 40% of the oil consumed in the United States must be imported. Replacement of petroleum by domestically available energy sources such as coal is desirable. Slow- and medium-speed compression ignition engines have the potential for conversion to coal fueling. Previous engine studies on coal slurries have investigated wear, thermal efficiency, and injection performance without evaluating the ignition characteristics. The ignition delay times and conditions for ignition of 45 mass % coal in methanol, diesel No. 2, and water are compared to diesel No. 2 and methanol in the present study.

The slurries are evaluated using a 900 rpm, direct injection, square piston engine simulator operating for one combustion cycle per experiment. Both 16:1 and 22:1 compression ratios are used with 2 atm abs. inlet air at temperatures from ambient to 250°C. The square geometry accommodates windows on two opposite walls of the combustion chamber for complete optical access.

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All of the test fuels except coal/water slurry ignited at the operating conditions attainable in the engine simulator. The temperature at time of injection required to obtain ignition is approximately 680 K for diesel No. 2 and coal/diesel slurry, 725 K for coal/methanol slurry, and 825 K for neat methanol. Activation temperatures, T_a, in the Arrhenius-type expression are: 5559 K for diesel No. 2, 7685 K for methanol, 3541 K for coal/diesel, and 5330 K for coal/methanol by the pressure delay method and 4357 K for diesel No. 2, 3926 K for coal/diesel, and 5510 K for coal/methanol by the luminosity delay method. The results establish the compression ignitability of the coal/methanol and coal/diesel slurry fuels at conditions appropriate to medium-speed diesel engines.

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INTRODUCTION

Most of the work on coal fired diesel engines before 1945 was conducted by German industry using coal dust; see the review by Soehngen[1]. However since that time, most coal fueled diesel research and development has used coal slurries: various percentages of coal in oil, water, methanol, ethanol, or mixtures of the liquid carriers. The existing fuel storage and transportation infrastructure for diesel engines is designed for liquids. Thus, conversion to slurry usage would present fewer difficulties compared to pulverized coal. Coal slurries compared to dust give advantages in the areas of fuel handling, safety, fuel injection control and reliability with slight cost disadvantages and an ignitability disadvantage for coal/water slurry only. Coal slurried with water, methanol, and diesel No. 2 are the test fuels for this study.

A comprehensive summary of published work on diesel engine combustion of coal slurries is contained in the thesis by Brehob[2] which extends an earlier review by Caton and Rosegay.[3] The engine tests described provide valuable information concerning engine modifications to enhance slurry combustion and to improve engine and injection system performance with slurries. The constant volume bomb studies of Siebers and Dyer[4] and the computer model studies of Bell and Caton[5] provide more fundamental information on coal slurry combustion under diesel engine conditions. Detailed information about the ignition characteristics of coal slurries under compression ignition conditions is not available currently. The objectives of this study are: 1) igniton delay characteristics of the three coal slurries tested and 2) to establish engine conditions required for slurry combustion.

EXPERIMENT AND DATA ANALYSIS

Fuels

The coal slurries used in this work were prepared by the National Institute for Petroleum and Energy Research. They are 45 mass % subbituminous, pulverized coal (Pittsburgh seam HVA) suspended in water, diesel No. 2, and methanol. The coal in the liquid carrier is ground by a ball mill to 5 micrometers mean diameter with the largest coal particles at 40 micrometers. The elemental composition of the coal is: 79.7% C, 5.9% H, 12.9% O and N, 0.6% S, and the remainder is ash. The coal contains 40.7% volatiles, 55.9% fixed carbon, 2.7% moisture, and 0.7% ash. The low ash level is achieved by washing in a hot acid, specifically, using the Otisca process. In addition to the coal and liquid carrier, the fuels contain additives: surfactants, dispersants, viscosity improvers, and lubricants at 5 mass % maximum.

Square Piston Engine Simulator

A single cylinder, CLR (Coordinated Lubrication Research) engine, commonly used for evaluation of engine lubricants is used as the base for the square piston engine simulator,

Fig. 1. The CLR's cylinder head is removed and replaced with a square "cylinder" and piston. The square and CLR pistons are attached by an adjustable connecting rod which provides compression ratios from 4.75:1 to 24:1. Compression ratios of 16:1 and 22:1 are used in the present study. Lateral or thrust forces are absorbed by the CLR piston acting as a crosshead. A square geometry provides flat surfaces for mounting windows for optical access.

The square piston engine simulator is a 4-stroke, direct injection engine with a swept volume of 700 cm^3 and scavenged by the CLR inlet and exhaust poppet valves installed in the square head. The square piston design is similar to the piston described by Namazian[6], et al. Each of the three grooves contain four piston ring sections which overlap in the corners. Copper leaf springs placed behind the piston rings at the back of the grooves force the rings against the cylinder walls.

A slow or medium-speed compression ignition engine is preferred over high-speed engines for burning coal fuels because of the longer residence time for more complete coal burnout. The vast majority of these engines are supercharged or turbocharged which raises the inlet air pressure and temperature. Thus, the square piston engine facility is equipped with a 12 kW air heater which is supplied with compressed air. A pressure of 2 atm abs at the heater is maintained for all the experiments. Air temperatures of ambient to 375°C can be delivered to the inlet of the engine.

The engine block can be preheated up to 150°C; but, the majority of the experiments are conducted with the engine block at ambient temperature.

A Stanadyne pencil type injector with a single orifice of 0.33 mm is found to provide a suitable compromise between good atomization and minimal plugging with coal slurries. The high pressures required to open the fuel injector valve and to maintain a high velocity spray during the injection interval is provided by a 6 mm jerk pump. The pump is actuated by a cambox driven off the engine crankshaft sprocket at 900 rpm. It is necessary to avoid injection during the scavenging stroke before combustion as this is a form of pilot injection which is an ignition aid. Therefore, solenoids on each end of the rack pull it open and closed for one stroke only.

Each experiment consists of approximately ten revolutions with data collected during two revolutions (one firing event). Cylinder pressure, injection line pressure, needle lift, and combustion luminosity are measured at each CAD (crank angle degree) by a LSI-11 microcomputer. Combustion luminosity is detected by a photodiode at one of the quartz windows. The axis of the photodiode is perpendicular to the axis of the injector to provide an unobstructed view of the fuel cloud because luminous combustion occurs along the injector axis. Cylinder line and injection line pressures are measured with piezoelectric transducers. Injector needle valve position is sensed by a proximity detector.

Ignition Delay Analysis

A suitable fuel for diesel engines is one with a short autoignition delay under compression ignition conditions. Ignition delay is the time between fuel delivery into the combustion chamber and the start of combustion. Starkman[7] postulates that the delay period is comprised of overlapping physical and chemical periods: vaporization and mixing for the physical delay and initiation reactions building a radical pool for the chemical delay. During the delay interval, the fuel becomes "prepared" for combustion, i.e., vaporizes and comes in contact with sufficient air to be flammable, as discussed by Lyn[8]. At the onset of ignition the premixed fuel burns rapidly. If the ignition delay is too long, a large portion of the fuel is prepared for combustion and the pressure rise associated with the premixed combustion is excessive resulting in uncontrolled combustion and diesel knock.

Due to the higher viscosity of coal slurries, the droplet size distribution issuing from the fuel injector nozzle is larger than for the liquid component alone. Nelson and coworkers[9] verify that coal/water slurry droplet sizes are larger than diesel fuel as measured by a Malvern laser diffraction analyzer at atmospheric pressure. Their results are replotted in Fig. 2. The droplet distributions are vastly different: while 90% of the mass of diesel droplets are contained in droplets less than 100 micrometers, less than 20 mass % of the coal/water slurry spray is composed of drops less than 100 micrometers. Nelson and coworkers also report on high resolution shadowgraphs of diesel and coal/diesel fuels. The maximum droplets observed in the developed region of the spray are approximately 650 and 250 micrometers for coal/oil slurry and neat diesel fuel, respectively. Larger drop size distributions prolong the physical delay period. Therefore, the slurry fuels may exhibit longer ignition delays than the corresponding neat fuels without coal due to longer physical delays.

In addition to droplet diameter, the ignition delay interval depends on many variables: air temperature, air pressure, turbulence levels, fuel type and fuel injection parameters such as spray formation, jet breakup, and impingement on combustion chamber surfaces, to name a few. For a given fuel and injection system, Lyn and Valdmanis[10] report that temperature is the dominant factor with pressure also playing a role. Henein and Bolt[11] summarize the functional relationships historically used to relate ignition delay to pressure and temperature. An Arrhenius type expression is the most widely used:

$$ID = \frac{A}{P^n} \exp(E_a/RT)$$

where ID is the ignition delay in msec, A the pre-exponential factor in msec-atm^n, P the pressure in atm, E_a the activation energy in J/kg, R the universal gas constant in J/kgmole-K, and T the temperature in K. The term activation energy strictly corresponds to an elementary reaction. For this system, the Arrhenius type expression is used to describe an extremely complex set of reactions. An "activation temperature", $T_a = E_a/R$, is used to emphasize the empirical nature of the activation energy.

The value of, n, is not believed to be a fuel dependent parameter. There is, however, considerable disagreement concerning the value of n as demonstrated in Table 1. Henein and Bolt[11] report that n increases with increasing turbulence levels, i.e., engine rpm. For this work at 900 rpm with relatively low turbulence levels, n of 1.5 is used:

$$ID = \frac{A}{P^{1.5}}\exp(T_a/T). \tag{1}$$

To determine A and T_a, four quantities are measured: start of injection and start of combustion to define delay interval, air temperature and air pressure. The pressure and temperature at the time of injection are used as suggested by Tsao and coworkers[17].

The CAD of needle lift and the pressure at injection are measured directly during each experiment. The temperature history of a 0.0375 mm diameter bead thermocouple placed into the center of the combustion chamber during a noncombusting engine cycle is corrected to estimate the motoring cycle gas temperature. Noncombusting cycle temperatures are measured for the range of conditions used in the square piston engine simulator. The temperature at the time of injection is found by interpolating between the two motoring cycle temperature histories with the most similar inlet conditions.

The start of combustion is measured in two ways for the purposes of this work: 1) pressure delay (the point in the combusting cycle where the slope of the pressure-time curve exceeds that of a corresponding noncombusting case) and 2) illumination delay (significant light emission is detected by a photodetector).

RESULTS AND DISCUSSION

Conditions for Ignition of Fuels

The longest ignition delays measured in the square piston engine simulator at 900 rpm are less than 10 msec (54 CAD). For injection at about 20^{o}BTDC (before top dead center), ignition occurs as late as 35^{o}ATDC (after top dead center). After this CAD, the lower temperatures and pressures from expansion do not permit substantial chemical activity. Ignition at 35^{o}ATDC is not of practical significance; it does, however, provide information concerning the ignition characteristics of the fuels.

Neither pressure rise nor light emission is observed when injecting coal/water slurry into the maximum temperature and pressure conditions attainable in the square piston engine simulator in the current configuration: i.e., up to 22:1 CR (compression ratio), 150^{o}C block temperature, and 250^{o}C inlet temperature at IVC (intake valve closing). Thus, coal/water slurry does not compression ignite when injected into air at up to 1150 K and 52 atm in the nearly 10 msec available.

Henein and Elias[18] discuss using inlet air preheat to extend the cetane scale. Following the approach of Henein and Elias, to evaluate diesel No. 2, methanol, coal/oil, and coal/methanol slurries, the inlet air temperature is increased at constant injection timing until ignition is obtained. The measured minimum temperatures required for combustion at 16:1 CR (approximately 30 atm pressure at the injection timing of 20^{o}BTDC) are:

Fuel	Temperature at IVC [K]	Temperature at injection [K]
Diesel No. 2	320	675
Coal/diesel	325	685
Methanol	450	825
Coal/methanol	360	725

The temperatures above are approximate due to the errors in measurement and the unrepeatable nature of ignition near the limiting temperature.

The temperatures for coal/diesel ignition are nearly the same as those for diesel No. 2. Coal/methanol and methanol fuels both require higher temperatures than diesel fuel for ignition as expected due to the low cetane number of methanol. However, coal/methanol ignites at temperatures well below those required for neat methanol.

The autoignition of coal/methanol slurry at lower temperatures than neat methanol is either due to the coal or the additive package in the slurry. As mentioned above, coal addition to the liquid carrier increases the viscosity dramatically causing the injected fuel droplets to be larger than with neat liquid fuel as shown in Fig. 2. Thus, any physical effect of the coal on the autoignition is most probably detrimental.

To investigate any chemical effects that the presence of coal might have on ignition, the volatile fraction must be characterized. Thurgood and Smoot[19] summarize data from 24 experiments in which the gaseous coal pyrolysis products are measured. Twenty of the reported experiments are conducted at rapid heating rates, i.e., greater than 10^4 K/sec. The products of devolatilization depend on coal type, heating rate, final temperature, and other factors. But, average values of the components are: 3% CO_2, 15% CO, 53% H_2, 14% CH_4, 10% C_2H_6, and 5% of other C_2 molecules. The final temperatures of the coal in these experiments is 1000 K or higher. The fuel at the periphery of the spray cone injected into the square piston engine simulator experiences very high heating rates: on the order of 10^5 K/sec. However, when the coal/methanol is injected into 725 K air, the final fuel droplet temperature is below the maximum 800 K of the gas. This is significantly below the 1000 K reported by Thurgood and Smoot. Kimber and Gray[20] report that at 1050 K final temperature and a reaction time of 70 msec, less than 25% of the coal's volatile matter is removed. Therefore, it is unlikely that enough combustible gas is devolatilized during the approximate 5 msec before ignition. The more plausible explanation for the autoignition characteristics of coal/methanol is that the additive package contains chemicals which ignite more readily than methanol.

Ignition Delay

The end of the ignition delay is measured in two ways: emission of light from combustion and pressure rise compared to the motoring trace. The amount of light emitted from methanol combustion is much less than that emitted from the other fuels. To obtain adequate sensitivity from the photodiode, the gain is raised by a factor of five for methanol tests. At this high gain with engine block heating, the photodetector senses the infrared radiation from warm combustion chamber surfaces giving erroneous ignition readings. Thus, only the pressure rise determination at start of combustion is used for methanol.

The diesel No. 2, methanol, coal/oil slurry, and coal/methanol slurry pressure delay results are shown in Fig. 3. Luminosity delay measurements are shown in Fig. 4 for the above fuels except for methanol. The following differences are significant at the 90% confidence level:

1.) coal/diesel has a lower activation temperature than diesel No. 2, coal/methanol, and methanol fuels by the pressure delay measurement.

2.) coal/diesel slurry has a lower activation temperature than coal/methanol when considering the luminosity delay.

No differences between the luminosity delay and pressure delay results are found within the measurement precision of these tests.

In both the luminosity and pressure delay analyses the activation temperature for the coal slurry is lower than the neat fuel alone. The ignition delay interval is comprised of overlapping physical and chemical periods. El Wakil, et al.[21] present calculated physical delay times for a 20 micrometers decane droplet. They also report that physical delay times are proportional to the droplet diameter to the 1.75 power. From their findings, Fig. 5 is constructed. From Fig. 2, 10% of the mass of diesel fuel droplets are contained in droplets less than 20 micrometers in diameter. In contrast, 10% of the mass of coal/oil droplets are found in 100 micrometers drop size or less. The physical delay for 20 micrometers droplets is almost negligible throughout the temperature range. However, at 100 micrometers, the physical delay interval is a sizeable portion of the total delay period. Temperature plays a lesser role in determining the physical delay period than droplet diameter. Therefore, the lower T_a of coal/oil and coal/methanol slurry compared to the neat fuels may be due to a relatively long physical delay interval which is less sensitive to temperature than the chemical delay.

Comparison to Reported Results

Siebers and Dyer[4] report ignition delay results from coal/water slurry combustion in a constant volume bomb using vitiated air. They define three delays: luminosity delay, pressure recovery delay, and pressure deficit delay (the point of lowest pressure after injection before pressure rises from combustion). Their results and a least squares fit of their data are shown in Fig. 6 along with dotted lines demarcating the operational limits of the square piston engine simulator. The maximum temperature attainable in the engine simulator is approximately 1100 K. The maximum delay time possible in the engine simulator is 10 msec because the temperatures become too low, due to expansion, for significant reaction to occur. The delay time most relevant to the square piston engine simulator, is the pressure recovery delay. As seen in Fig. 6, the pressure recovery delay curve does not pass through the shaded region, i.e., the conditions achievable in the engine simulator. Siebers and Dyer's results are consistent with the findings of the present work that coal/water slurry ignition does not occur in the engine simulator with the current configuration.

The activation temperature for diesel No. 2 from this work is compared with other reported results:

Researcher	T_a (Pres)	T_a (Lum)
Present	5560 K	4360 K
Wolfer[12]	4650 K	
Kuniyoshi, *et al.*[14]	4170 K	5130 K

The agreement among these three experiments is within the measurement accuracy.

CONCLUSIONS

1) Coal/methanol slurry ingites at significantly lower inlet temperatures than neat methanol: 360 K versus 450 K.

2) Coal/water slurry does not compression ignite at the conditions attainable in the square piston engine simulator.

3) Methanol compression ignition combustion emits low levels of light making accurate luminosity delay measurements difficult.

4) The activation temperatures for the test fuels are:

Fuel	T_a (Lum)	T_a (Pres)
Diesel No. 2	4360 K	5560 K
Coal/diesel	3925 K	3540 K
Methanol	7685 K	
Coal/methanol	5510 K	5330 K

The activation temperature for coal/diesel is lower than diesel and coal/methanol based on the pressure delay at the 90% confidence level.

5) The pressure and luminosity delay intervals are not different within the precision of the measurements in the square piston engine simulator.

ACKNOWLEGEMENTS

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Table 1. Summary of reported values for the pressure exponent, n, in the Arrhenius type relation.

Reference	System	n
Wolfer[12]	Heated constant volume bomb	1.19
Mullins[13]	Flow reactor	1.0
Henein and Bolt[11]	Indirect injection engine	1.5-1.8
Kuniyoshi, *et al.*[14]	Heated constant volume bomb	0-1
Spadaccini and TeVelde[15]	Flow reactor	2.0
Parker, *et al.*[16]	Heated constant volume bomb	0

FIGURE CAPTIONS

Figure 1. Square piston engine simulator.

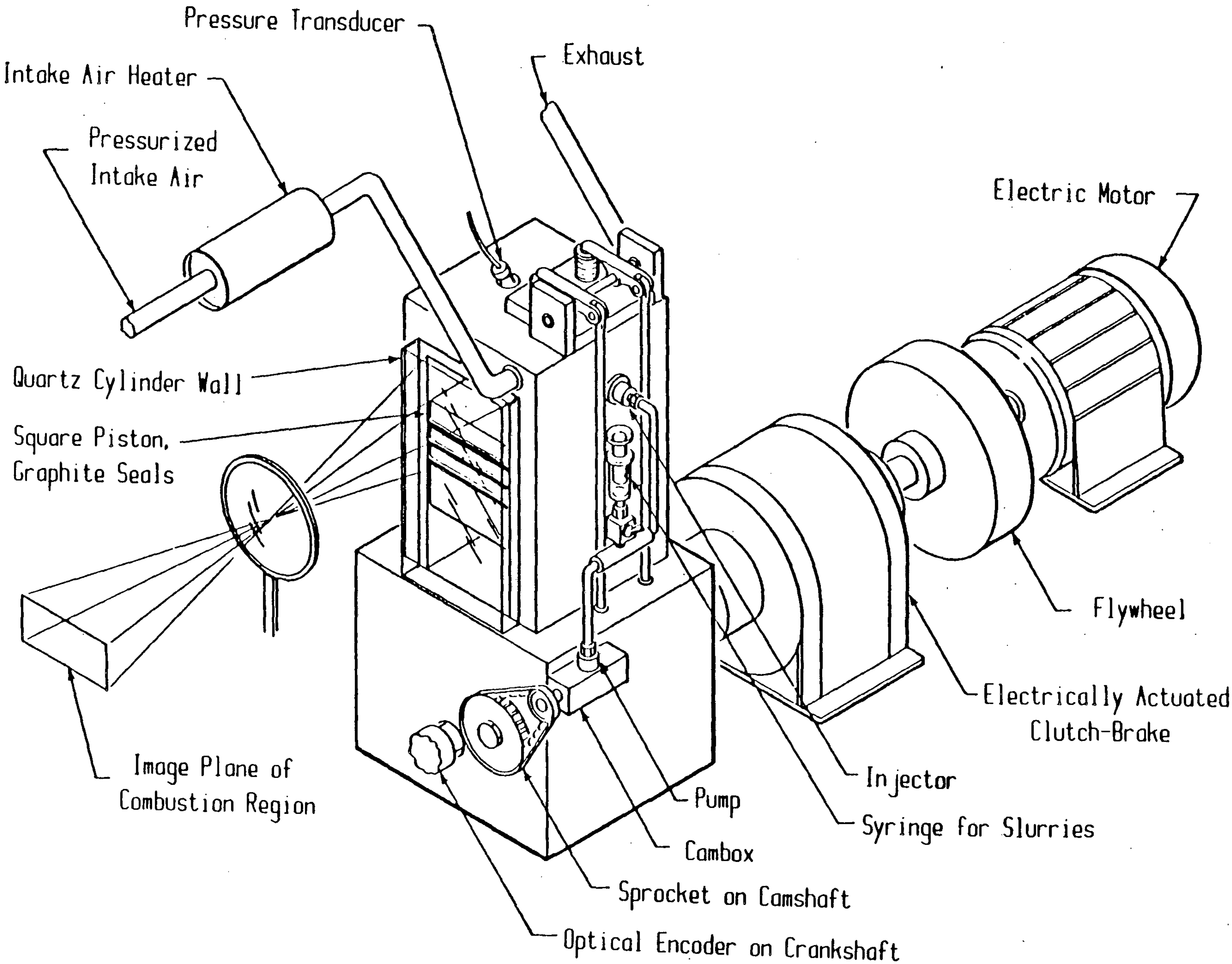
Figure 2. Fuel droplet size distributions for diesel No. 2 and coal/water slurry, from Nelson, et al.[9].

Figure 3. Temperature dependence of pressure defined ignition delay with pressure dependence to the 1.5 power for diesel No. 2, coal/diesel slurry, methanol, and coal/methanol slurry fuels in the square piston engine simulator.

Figure 4. Temperature dependence of luminosity defined delay period with pressure dependence to the 1.5 power for diesel No. 2, coal/diesel slurry, and coal/methanol slurry in the square piston engine simulator.

Figure 5. Physical delay time for decane droplets at varying droplet temperatures with drop diameter as a parameter. (Data taken from Fig. 7 of paper by El wakil, et al.[21]).

Figure 6. Ignition delay results on coal/water slurry adapted (least squares analysis applied) from Siebers and Dyers[4] with possible operating conditions in square piston engine simulator superimposed.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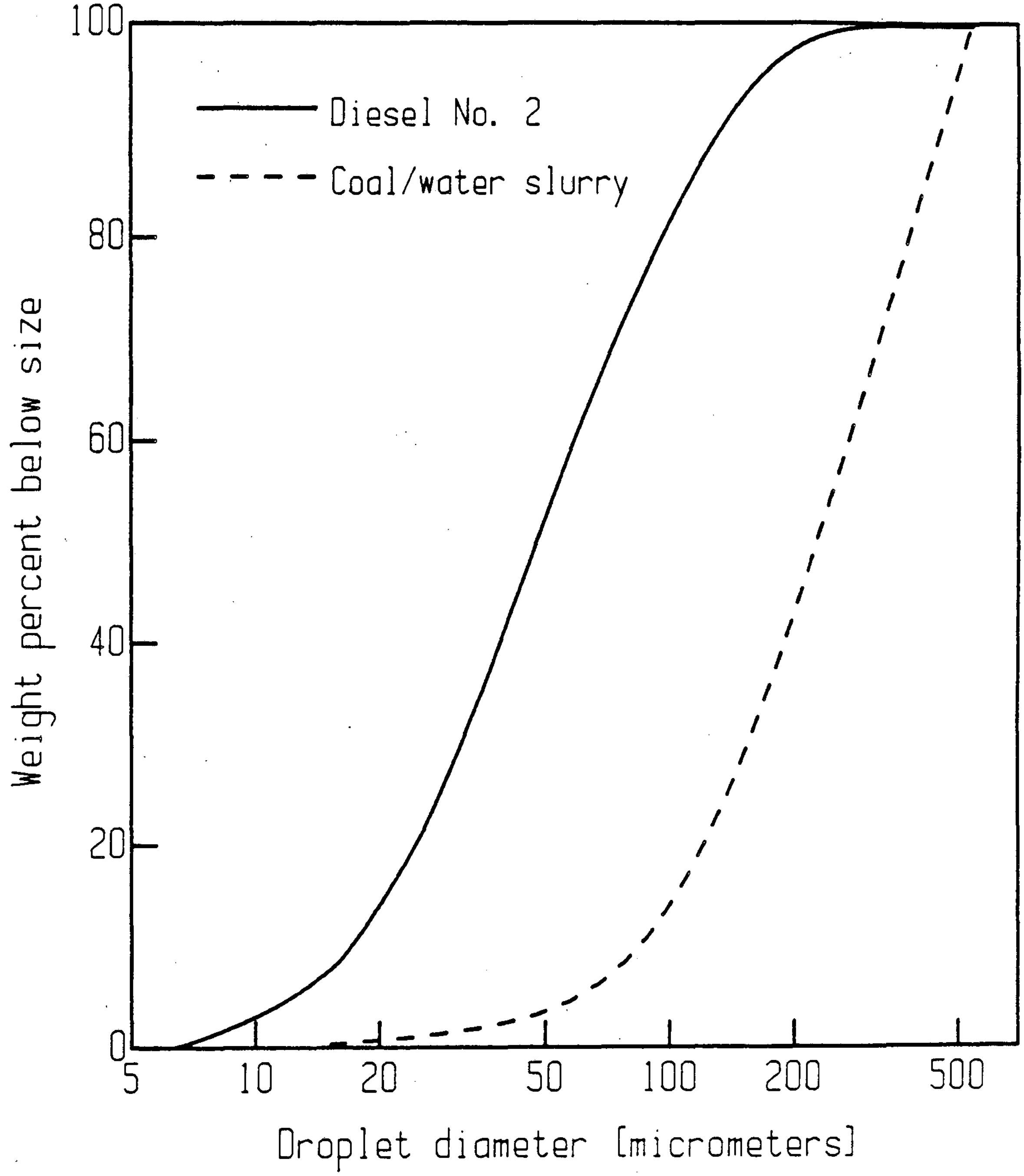
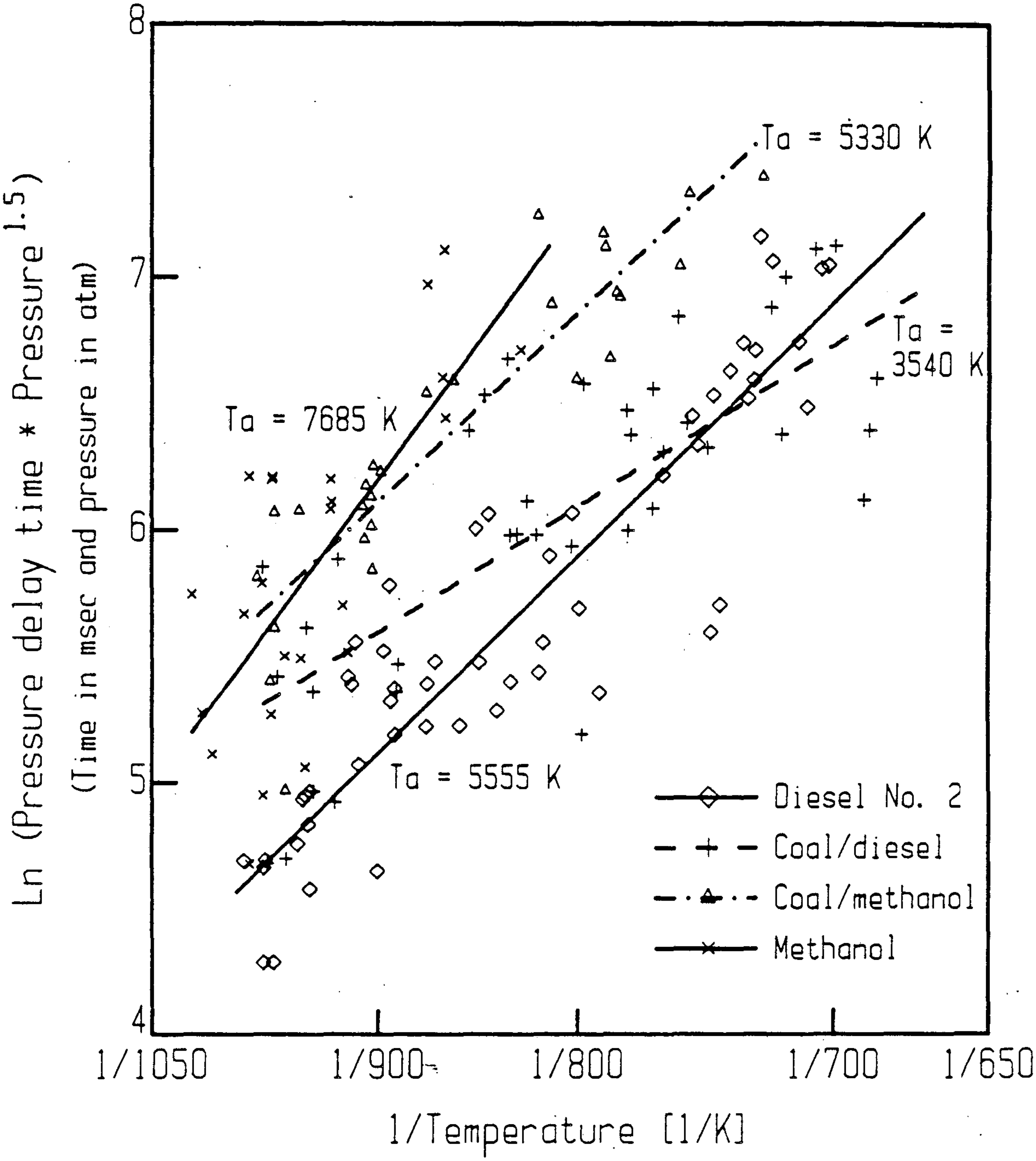
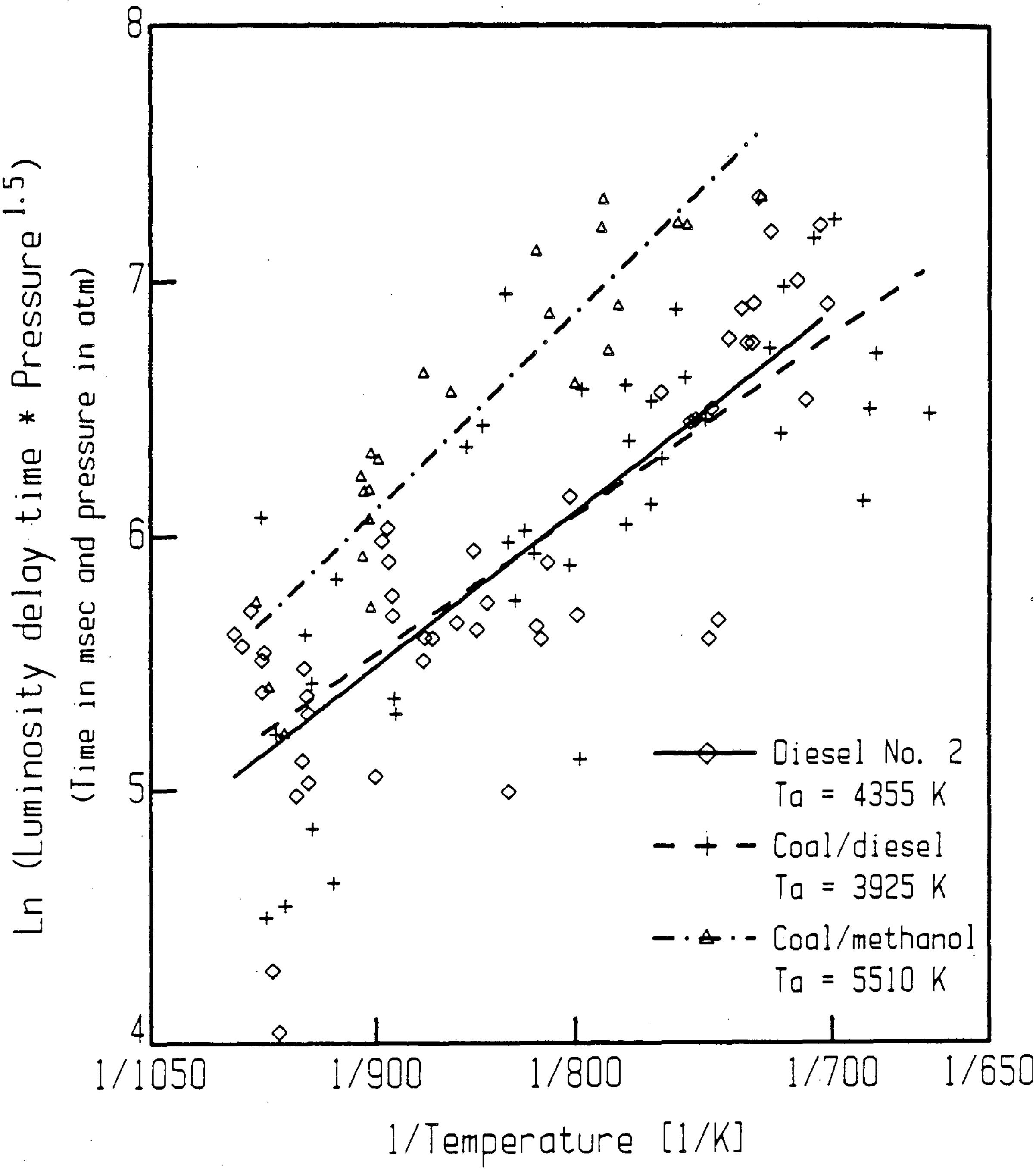
Exhaust
Intake Air Heater
Pressurized
Intake Air
Electric Motor
Quartz Cylinder Wall
Square Piston,
Graphite Seals
Flywheel
Electrically Actuated
Clutch-Brake
Image Plane of
Combustion Region
Injector
Pump
Syringe for Slurries
Cambox
Sprocket on Camshaft
Optical Encoder on Crankshaft



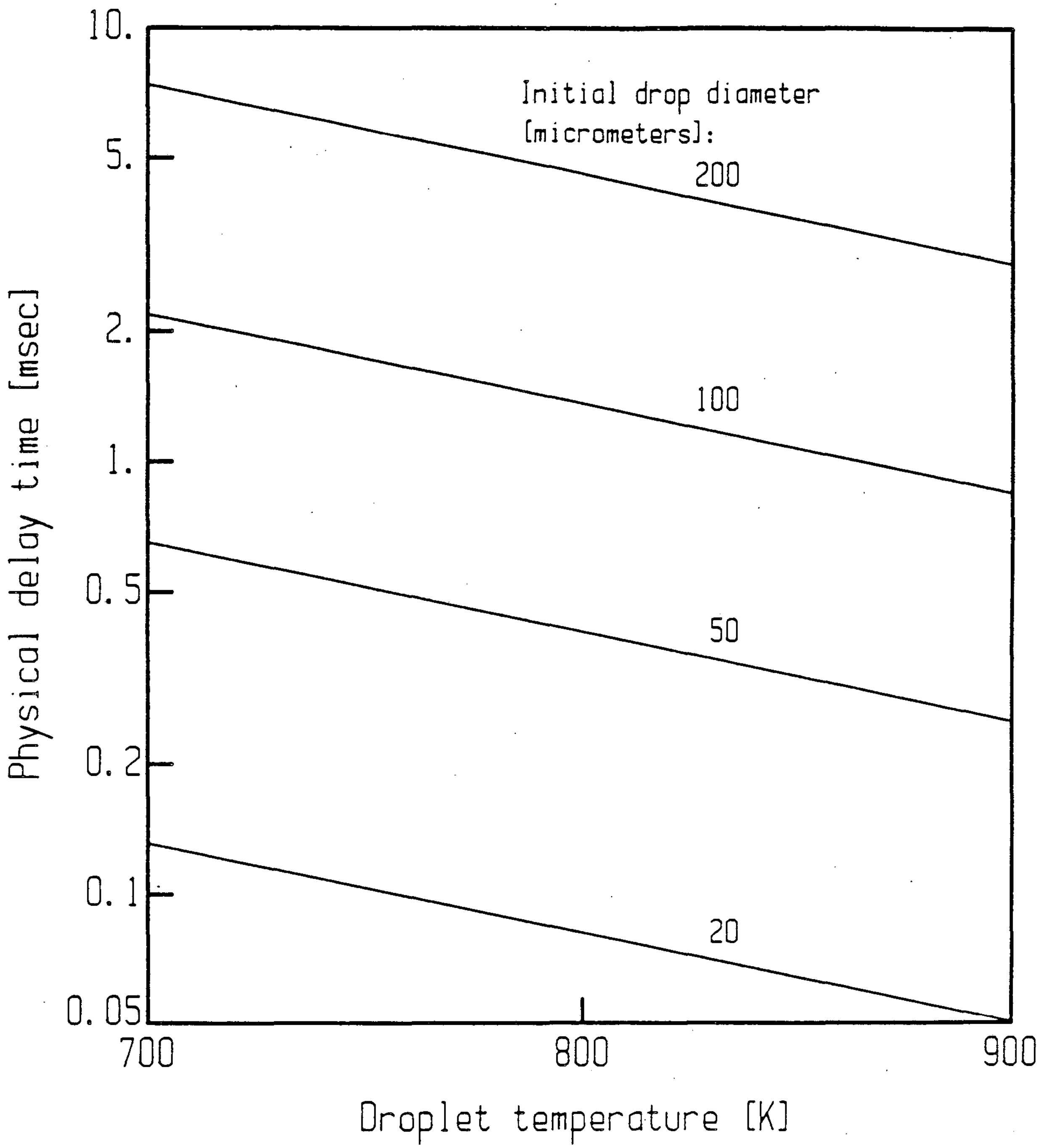
Brehob and Sawyer, Figure2



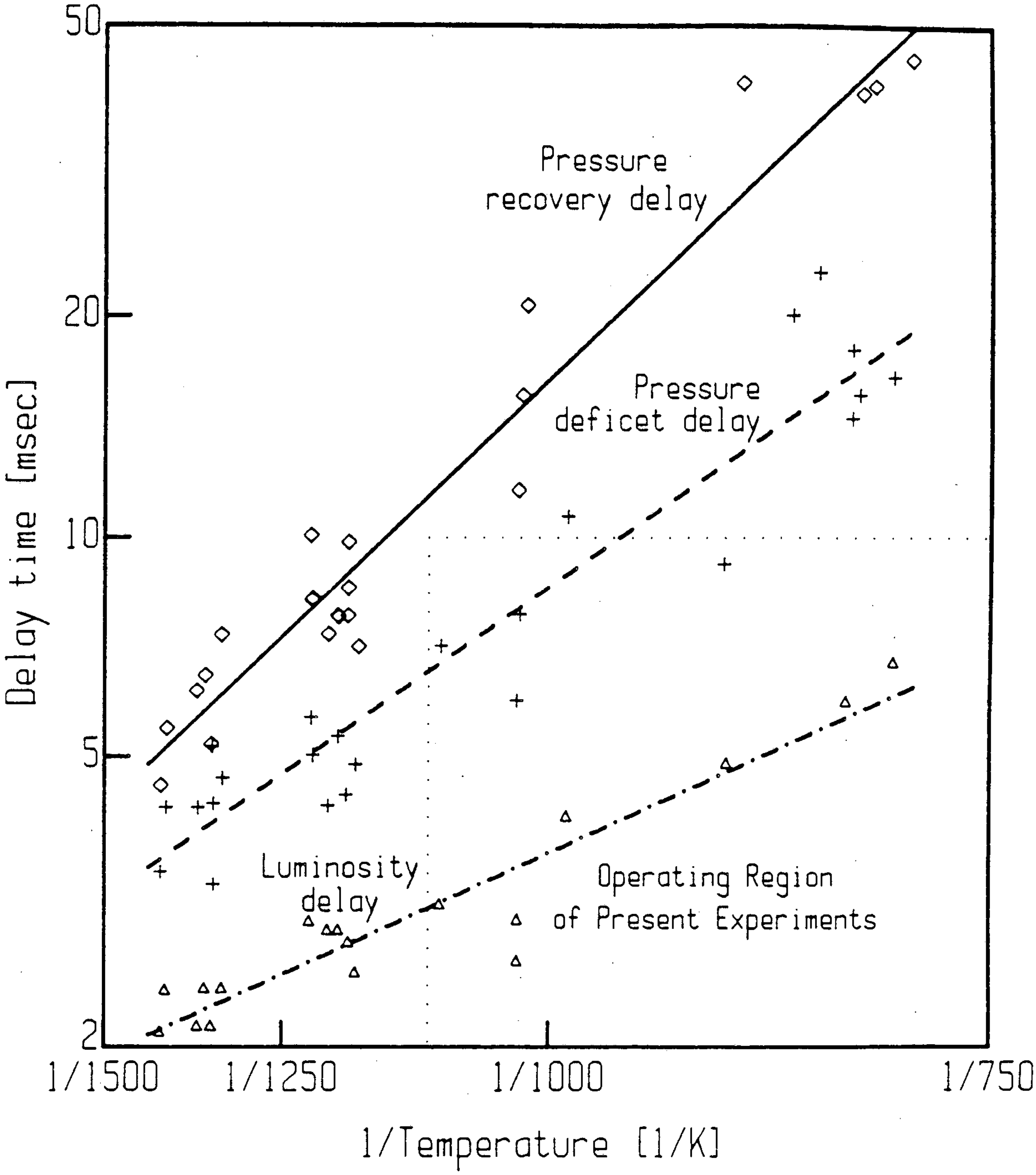
Brehob and Sawyer, Figure 3



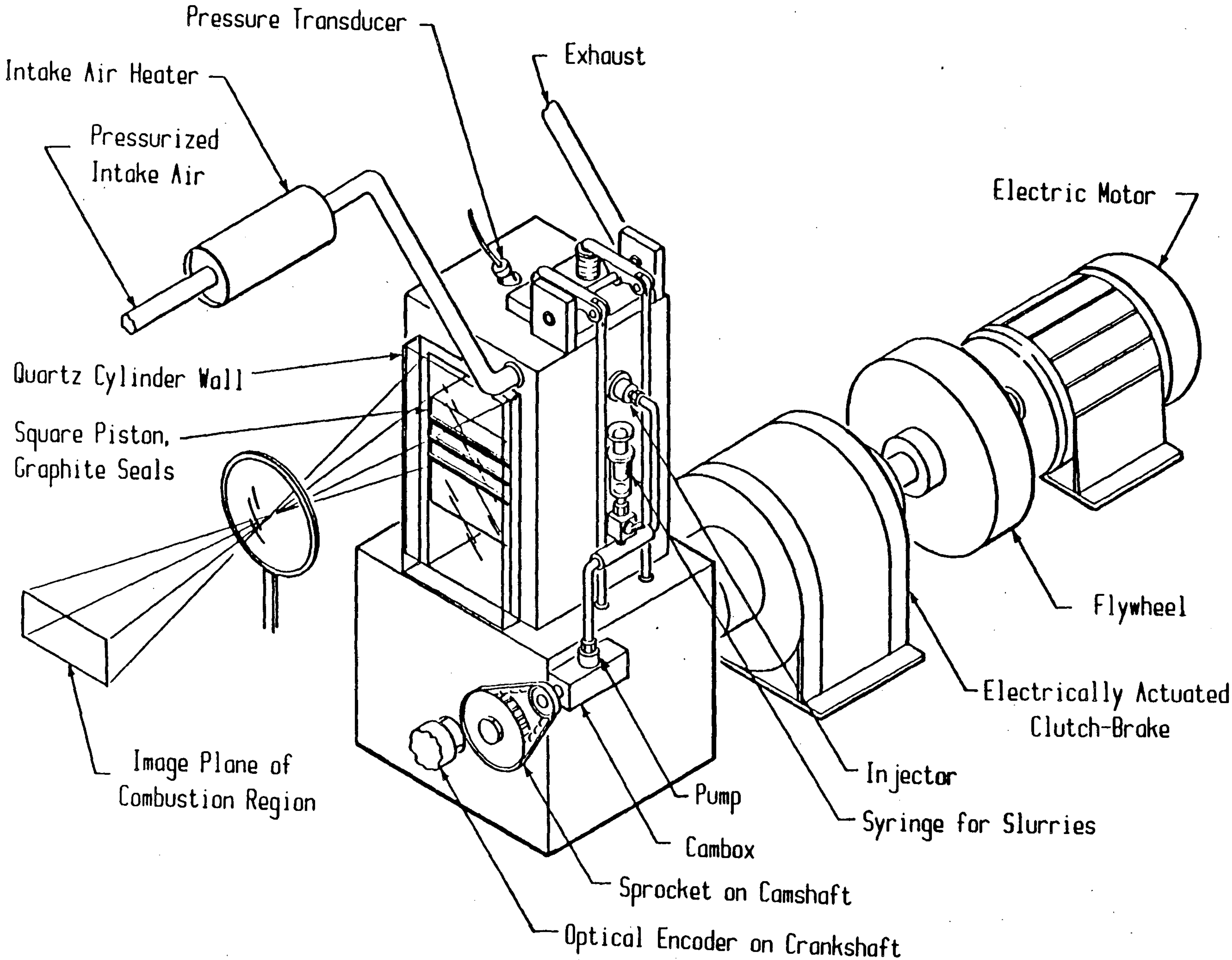
Brehob and Sawyer, Figure 4



Brehob and Sawyer, Figure 5



Brehob and Sawyer, Figure 6


Pressure Transducer
Exhaust
Intake Air Heater
Pressurized Intake Air
Electric Motor
Quartz Cylinder Wall
Square Piston, Graphite Seals
Flywheel
Electrically Actuated Clutch-Brake
Image Plane of Combustion Region
Injector
Pump
Syringe for Slurries
Cambox
Sprocket on Camshaft
Optical Encoder on Crankshaft